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Techno-Economic Analysis of Integrated Heat Pump with Solar Collectors and Energy Wells

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Master of Energy and Environmental Engineering
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MASTER THESIS

for

Student Monica Tjørhom

Spring 2014

Techno-economic analysis of integrated heat pump with solar collectors and energy well

Tekno-økonomisk analyse av varmepumpe integrert med solfangere og energibrønn

Background and objective

Liquid to liquid heat pump system combined with solar collectors and energy wells for seasonal storage of excess heat is a very efficient technology. Internationally, there has been several research reports related to heat pumps supplemented by solar collectors and integrated with energy wells. In Norway however, it is more common to design liquid to liquid heat pump systems with solar collectors *without* utilization of seasonal storage.

Passive house standard requires high investment costs for the building construction. Heat pump supplemented by solar collectors and boreholes as energy supply to a building requires high investment costs, but allows for low operational expenses. The student will analyze the heat pump system with solar collectors and energy wells. If possible the student might also analyze other heat pump systems connected with the solar collectors. The master thesis will perform simulation of a building model that is energy supplied with the system mentioned above. The simulation might be performed in IDA-ICE.

The project assignment was done in collaboration with the consultant company Rambøll in Trondheim. During the project assignment, the student started to analyze a highly efficient kindergarten that will be built in Trondheim. The student will continue to work on the same project.

The purpose of this study is to analyze and simulate the building energy supply system equipped with heat pump that is supported by solar collectors for Norwegian conditions. Different heat pump technologies might be considered. Challenges, conditions, and benefits of such technologies will be identified. Economic analysis should be performed.

The following tasks are to be considered:

1. Literature study of liquid to liquid heat pump system integrated with solar collectors and energy wells. Literature study should include an overview of technical as well as practical and financial conditions.
2. Implement the collected data about the kindergarten in the simulation model. If necessary develop and improve the developed model in IDA-ICE.

3. Analyze the developed model. Include the collected data into the analysis. Some starting points for the analysis might be: occupant behavior, system design, and system operation.
4. Present and discuss the results. Specifically present the results of the economic analysis. Suggest ideas for the future work.

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

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

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

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

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

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

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

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

Department of Energy and Process Engineering, 16. January 2014

Olav Bolland

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PREFACE

This master thesis is written in the spring of 2014 as a final assessment in the master program Energy and Environmental Engineering at Norwegian University of Science and Technology (NTNU). The thesis is a continuation of my project thesis with the same title, written autumn 2013.

The subject of the thesis is made up through collaboration between Rambøll Norge AS, NTNU and me. The purpose of the thesis is to gain knowledge about and analyse an energy supply system consisting of a heat pump combined with solar collector and energy wells for seasonal storage of thermal energy. The analysis is done by simulating a case building in Trondheim, using the simulation program IDA ICE.

I chose this research topic because I wanted to learn more about renewable heat supply systems, especially the concept of storing heat seasonally. Heat pumps have caught my interest ever since my parents installed a ground source heat pump as the first to do so in our municipality in year 2002.

I would like to thank my supervisor Natasa Nord at NTNU for good advises and guidance. My co-supervisor Line Solberg Ohnstad at Rambøll in Trondheim also deserves great thanks for helping me retrieving all information about the case building. All co-workers at Rambøll Trondheim have been very forthcoming, and special thanks go out to Tor Lystad for being so helpful regarding heat pump analysis. I would also like to thank the caretaker at Haukåsen kindergarten, Stian Sandnes, for showing me around the kindergarten.

Last I would like to thank my family, especially my sister Camilla for proofreading and Vidar for good personal support all the way.

Trondheim, June 17th 2014

Monica Tjørhom

Monica Tjørhom

ABSTRACT

The European Union appointed in 2007 an objective to reduce the energy consumption with 20 % and increase the utilization of renewable energy resources with 20 % within year 2020. This master thesis analysed a passive house kindergarten in Trondheim built in 2013 with a heat supply system based on renewable heat sources, solar collectors combined with a ground source heat pump. The possibility of storing solar heat seasonally in energy wells was also taken into consideration.

The kindergarten was modelled in the simulation program IDA ICE, in order to perform simulations and analyse the building's heat supply system. The model was initiated through the author's preliminary project thesis. As the aim was to make the model as realistic as possible, all documentation used as basis for the input data was received from Rambøll, who did the projecting of the heat supply system. Issues particularly of interest were the resulting net annual energy need of the building, heating loads and the performance of the heat plant. Indoor environment in the occupied zones and ventilation control strategies were also examined. Parameters regarding system design and operation were changed in order to study how this affected the results. At last, an economic evaluation of the heat supply system was carried out too see whether the heat supply system was economically preferable.

The net annual energy need according to IDA ICE was 57,4 kWh/m^2, in which the heating need was 33,1 kWh/m^2. Judging by IDA ICE results, the installed radiators at Haukåsen kindergarten have oversized capacity, while the heating coil and domestic hot water load was dimensioned with too low heating load. Out of the net annual heating need, the results showed that the heat pump covered 81,5 %, the boiler 12,5 % and the solar collector 6 %. As the heat pump coverage was found to be only 24 % of the heating load, the high coverage of the net annual heating need indicated an oversized capacity in the heat pump. This may cause earlier wear out of the compressor due to part-load operation most of the time.

The results related to analysis of the solar collector showed that the solar collector contribution was 1 608 kWh/year, but the theoretical efficiency implied that a contribution of 4 241 kWh/year could be expected. Either doubling of the collector area or optimization of the tilting angle gave noticeably higher contribution. Neither did changing of the shape factor for the hot water tank. Thus the default control strategy of the solar collector circuit in IDA ICE was questioned, and ought to be further studied.

As the zones in the kindergarten have demand controlled ventilation based on temperature, presence and CO_2 concentration, different ventilation control macros were developed and implemented in the IDA ICE model. This resulted in more energy efficient ventilation and 1 388 kWh was saved each year due to decreased energy need for fan operation. Realization of night set-back contributed to decrease the net annual heating

need with 5 969 kWh/year. Nevertheless, the low annual energy need was at the sacrifice of the indoor environment. The night set-back implied too low zone air temperatures during wintertime, while the occupancy controlled ventilation led to excess temperature during summertime.

Simulation of underground thermal energy storage was carried out by changing the IDA ICE plant macro. A ground heat exchanger ensured transfer of heat from the solar collector circuit to the brine return pipeline. The result showed a 78 % increase in annual solar heat contribution and 0,08 °C increase in ground temperature over a year. This indicated that the heat pump COP would remain high for a longer time period than in the model without the ground heat exchanger. To confirm this, further studies on the subject should involve simulations over longer time periods.

The economic analysis showed that the existing heat plant in the kindergarten has a global cost of 452 892 NOK and a pay-off period of 25 years. If the solar collector had not been installed, 25 667 NOK could have been saved in global cost and 1,3 years in pay-off period. Nevertheless, installation of solar collectors was a deciding factor when the building received the label *Very good* according to the building classification system BREEAM. On this basis the solar collector was considered a valuable investment. Sensitivity analysis showed that an increase in real interest rate gave lower global costs and a higher pay-off period.

SAMMENDRAG

EU satte i 2007 et mål om å redusere energiforbruket med 20 % og øke bruken av fornybare energiressurser med 20 % innen 2020. Denne masteroppgaven analyserer en passivhus-barnehage med oppvarmingsanlegg basert på fornybare energiressurser, solfanger kombinert med en bergvarmepumpe. Muligheten for å sesonglagre solvarme i energibrønner ble også tatt i betraktning.

Barnehagen ble modellert i simuleringsprogrammet IDA ICE for å utføre simuleringer og analysere bygningens oppvarmingsanlegg. Denne modellen ble påbegynt under forfatterens innledende fordypningsprosjekt. Ettersom målet var å bygge en mest mulig realistisk modell, ble alt underlag for inndata mottatt fra Rambøll, som utførte prosjekteringen av oppvarmingsanlegget. Temaer av spesiell interesse var netto årlig energibehov, effektbelastninger og ytelsen til oppvarmingsanlegget. Innemiljøet i bygningen og strategier for ventilasjonsstyring ble også undersøkt. Parametre i anleggets systemdesign og –drift ble endret for å studere hvordan det påvirket ytelsen. Til slutt ble det utført en økonomisk evaluering av oppvarmingsanlegget for å se om systemet var å foretrekke ut fra et økonomisk perspektiv.

Netto årlig energibehov var ifølge IDA ICE på 57,4 kWh/m^2 per år, hvorav oppvarmingsbehovet var på 33,1 kWh/m^2. Ut ifra resultatene i IDA ICE har de installerte radiatorene i barnehagen overdimensjonert kapasitet, mens varmebatteriet og tappevannslasten ble underdimensjonert med tanke på varmeeffektuttak. Prosentvis dekte varmepumpa 81,5 %, den elektriske kjelen 12,5 % og solfangeren 6 % av netto årlig energibehov. Ettersom resultatet viste at varmepumpa kun dekte 24 % av effektbelastningen, indikerte den høye dekningen av energibehovet at varmepumpa er overdimensjonert. Det kan føre til tidlig slitasje på grunn av uønsket dellastdrift mesteparten av tiden.

Resultatene relatert til analysen av solfangeren viste et bidrag på 1 608 kWh/år, mens den teoretiske virkningsgraden tilsa at et bidrag på 4 241 kWh/år kunne forventes. Verken dobling av solfangerarealet eller optimalisering av installert helningsvinkel gav merkbart høyere bidrag. Det gjorde heller ikke endring av formfaktoren til varmtvannstanken. Av den grunn bør den innebygde reguleringsstrategien til solfangeren i IDA ICE undersøkes nærmere.

Ettersom barnehagen har behovsstyrt ventilasjon basert på tilstedeværelse, temperatur og CO_2-konsentrasjon, ble ulike makroer utviklet og implementert i IDA ICE-modellen. Dette resulterte i mer energieffektiv ventilasjon og 1 388 kWh ble spart hvert år grunnet lavere energibehov til vifter. Innføring av nattsenking i modellen bidro til å redusere det netto årlige oppvarmingsbehovet med 5 969 kWh. Disse innsparingene gikk dog på bekostning av innemiljøet. Nattsenkingen medførte for lave lufttemperaturer i

oppholdssonene om vinteren, mens den behovsstyrte ventilasjonen førte til overtemperaturer om sommeren.

Simulering av sesonglagring ble utført ved å endre systemskissen til oppvarmingsanlegget direkte i IDA ICE. En varmeveksler sikret overføring av varme fra solfangerkretsen til returføringen til energibrønnene. Resultatene viste at solfangerbidraget økte med 78 % og temperaturen i berggrunnen økte med 0,08 °C over ett år. Dette medførte at varmepumpas virkningsgrad vil holde et høyt nivå over lengre tid enn i modellen uten sesonglagring. For å bekrefte dette, bør videre studier involvere simuleringer over lengre tidsperioder.

Den økonomiske analysen viste at eksisterende oppvarmingsanlegg har en nåverdikostnad på 452 892 NOK og en tilbakebetalingstid på 25 år. Dersom solfangeren ikke hadde blitt installert, ville 25 667 NOK vært spart i nåverdikostnad og 1,3 år i tilbakebetalingstid. Likevel var installasjonen av solfangere av avgjørende betydning for at bygningen fikk betegnelsen *Very good* i henhold til klassifiseringssystemet BREEAM. Av den grunn ble investeringen i solfangerne betraktet som verdifull. En sensitivitetsanalyse viste at økt realrente gav lavere nåverdikostnader og høyere tilbakebetalingstid.

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NOMENCLATURE

Base load: The (more or less) constant share of a buildings total effect load during a year
Boiler: A device that burns gas, oil, coal or is heated by electricity to provide hot water
Brine: Heat transfer medium between ground and heat pump
Carnot process: Theoretical, ideal thermodynamic cycle without losses
Compressor: Component in the heat pump, used for compression of the refrigerant
Condenser: Heat exchanger transferring heat from the refrigerant in the heat pump to the hot water distribution system
Dry-bulb temperature: What is spoken of as air temperature
Evaporator: Heat exchanger transferring heat from the brine to the refrigerant in the heat
Expansion valve: Component in the heat pump, used for expansion of the refrigerant
Heat exchanger: A devise for transferring heat from one medium to another
Outdoors kindergarten: A kindergarten based on outdoor activities most of the day
Present value factor: Annual costs and incomes are multiplied with this factor in order to be referred to the starting year
Refrigerant: Cooling medium in the heat pump
Thermal bridge: Junctions in building construction where insulation is not continuous and causes heat loss
Thermal mass: Describes a material's capacity to absorb, store and release heat
U-pipe: Piping in the ground where the brine circulates.
U-value: Coefficient of thermal transmittance

ABBREVIATIONS

ACH: *Air Changes per Hour*
AHU: *Air Handling Unit*
BIM: *Building Information Model*
BREEAM: *Building Research Establishment Environmental Assessment Method*
CAV: *Constant Air Volume,* cf. constant ventilation air flows
COP: *Coefficient of Performance*
DCV: *Demand Controlled Ventilation*
GSHP: *Ground Source Heat Pump*
HVAC: *Heating, Ventilation and Air-Conditioning*
IFC: *Industry Foundation Classes*, (IFC-model implies a digital data file containing several subjects related to building construction, such as architectural model, structural model, technical systems etc.)
LMTD: *Logarithmic mean temperature difference*

PMV: *Predicted Mean Vote*
PPD: *Predicted Percentage Dissatisfied*
UTES: *Underground Thermal Energy Storage*
VAV: *Variable Air Volume*, cf. variable ventilation air flows

SYMBOLS

α = The absorber's degree of absorption
Δ = Delta, which stands for an interval (for example ΔT means a temperature interval)
η = Efficiency
τ = Degree of transmission of the glass
A = Area (m^2)
C = Cost (NOK)
c_p = Specific heat capacity (kJ/kgK])
E = Annual energy consumption (kWh/year)
h = Depth (m)
I = Irradiance (W/m^2)
k = Heat loss coefficient (W/m^2K)
$\dot{m}$ = Mass flow rate (kg/s)
M_p = Preventive maintenance cost (NOK/year)
n = Calculation period in economic analysis (years)
n_0 = Pay-off period (years)
Pr = Energy price (NOK/kWh)
PV = Present value factor (1/year)
Q = Air flow (m^3/h) or heat effect (W)
$\dot{Q}$ = Thermal power (W/m^2)
r = Real interest rate (-)
S = Savings in operational cost (NOK/year)
T = Temperature (K)
U = U-value (W/ m^2K)
W = Compressor work (W)

SUBSCRIPTS

c = Condensation
e = Evaporation
eff = Effective
f = Fixed (as in fixed costs)
I = Investment
loss = Losses (for example: $\dot{Q}_{loss}$ means thermal losses)
m = Maintenance
min = Minimum
max = Maximum
O = Operating (as in operating costs)
rv = Residual value
tot = Total
w = Water

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1. INTRODUCTION

1.1 BACKGROUND

In 2007 the European Union decided that within year 2020, the goal is to have 20 % reduction of energy consumption, and 20 % increase in utilizing renewable energy resources [Europaportalen, 2013].

Solar energy can be utilized for heating purposes by installing solar collectors, but in Norway the access to solar energy is limited in the winter months. In the summer months on the other hand, the irradiance is so high that the incident of excess heat occurs. One way of utilizing this excess heat is to store the heat seasonally in thermal energy wells. In that way the heat can be used for heating purposes during the winter, as the energy wells function as heat source for a ground source heat pump (GSHP).

This heat supply system will contribute to decrease the specific electricity need, which is an advantage in Norway where the electricity need keeps increasing. If a heat pump installation needs a greener profile, or one wishes to replace i.e. a wood-fired heating plant, this system is applicable.

1.2 OBJECTIVE

The purpose of this study is to gain knowledge about and analyse a heat supply system consisting of a heat pump combined with solar collector and energy wells. A reference model of a case building in Norway is to be established, in order to perform simulations of the building with the chosen heat supply system.

Haukåsen kindergarten located in Trondheim is the chosen case building of this project, and the simulation tool being used is IDA ICE. Issues particularly of interest are the resulting net annual energy need of the building, heating loads and the performance of the heat plant. Indoor environment in the occupied zones and ventilation control strategies are also to be examined. The kindergarten has ventilation control based on motion sensors in most of the zones, which makes it interesting to challenge the simulation tool and see whether or not the ventilation concept works in practice. Parameters regarding system design and operation are to be changed, in order to study how this affects the results. At last, an economic evaluation of the heat supply system is to be done, too see whether the system is economically preferable.

1.3 CONTINUATION FROM THE PROJECT THESIS

An IDA ICE model of the case building was established during the project thesis work, but further development of this model is highly necessary. One main task is to improve the ventilation control macros, by implementing minimum mean air flow rates during operating hours. Since the project work was carried out, a newer version of IDA ICE has been launched. This brings along a larger register of climate files, making it possible to achieve more accurate results. Also, implementation of night set-back and more exact occupancy presence ought to be examined, making the model more similar to the actual case building. Last, implementing a macro that contains the possibility of underground thermal energy storage (UTES) ought to be performed.

1.4 OUTLINE

First presented is a literature study where all the main components of the heat plant are described, along with research on existing passive houses. The aim with this chapter is to establish a basic understanding for how the heat supply system functions. Secondly, a method chapter called Simulations is given, describing the aim with the case building simulation and main approach on how to build up the model in the IDA ICE simulation tool. Then a chapter with a review of the case building is given, followed by all input data for external verification. Thereafter the results of the simulations are presented, and the significance of the results is evaluated in the Discussion chapter. Finally the Conclusion summarizes the impact of the results and further work on the subject is proposed.

In the report it is assumed that the reader has basic knowledge regarding building engineering, heat and mass transfer and thermodynamics. Although some terminology is provided, the reader should know technical terminology regarding heat pumps, air handling units, ventilation principles and thermal comfort index values like predicted mean vote (PMV) and predicted percentage dissatisfied (PPD).

1.5 LIMITATIONS

The literature study is limited to describing only technologies and components that are relevant for the heat supply system in this project. The focus lays on the main components of the system, i.e. the solar collector, accumulator tank, heat pump, energy wells, and the concept of thermal seasonal energy storage.

Collecting measuring data on the kindergarten was initially planned, to compare the data with the results from the simulations. As it turns out, the measuring instruments are not yet installed in the kindergarten, so logging of the components is not possible to carry out at this point. This implies that measuring data is given less space in the report, and the focus is directed more to IDA ICE simulations and the economic analysis.

2. LITERATURE REVIEW

This chapter presents a literature study of the heat supply system; solar collector combined with a ground source heat pump (GSHP). The aim is to give a description of all main components of the system and establish a basic understanding for how the heat supply technology works. Some of the content in this chapter was written in the project thesis, and is reproduced in this thesis.

Outline

First an introduction to passive houses and attainment criteria is given along with a description of the building classification system BREEAM-NOR. This theory is included due to the case building being a passive house and classified as *Very Good* according to BREEAM-NOR standard. Second the conversion of solar radiance to heat via solar collectors is explained, and different types of solar collectors are discussed. Third the GSHP with its main components is presented, and the principle of thermal energy storage is introduced. After this, a summary of the combined system will be given. At last a brief literature study of different simulation tools is presented.

2.1 PASSIVE HOUSES

A passive house consists of a well-insulated building body with minimum air leakages and thermal bridges and has a high-efficient heat recovery (80-90 %). This leads to a reduced need for space heating and in many cases airborne space heating via ventilation air is sufficient to heat the building [Dokka et al., 2009].

Paramount requirements for passive house standard are in preparation and are expected to take effect from year 2015. Temporarily requirements are recommendations stated in a project report published by *SINTEF* [Dokka et al., 2009]. Some of the values in this report are based on the Norwegian building code *NS3031* [Standard Norge, 2011]. Relevant requirements for this master thesis are stated in chapter 5 (about input data for the reference model).

Figure 1 shows how the specific energy need has changed in office buildings built before 1987 and till today's passive house standard.

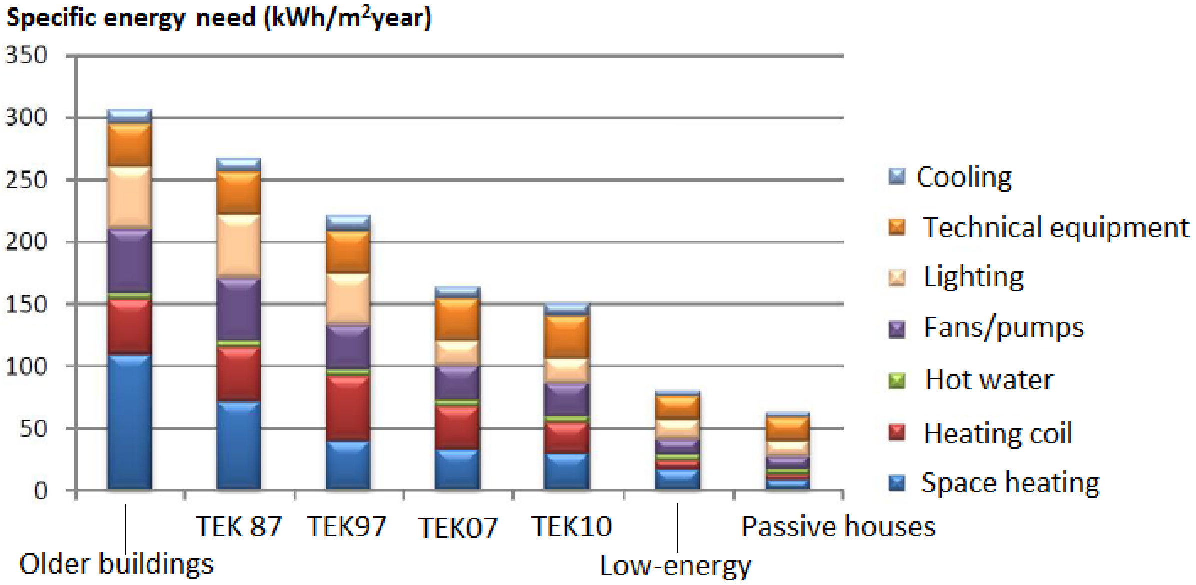

Figure 1 – Energy need development for office buildings [Stene, 2013]

The trend shown in figure 1 can be assumed to be similar for all non-residential buildings, including kindergartens. The need for domestic hot water heating remains the same for all the building types in figure 1, but the space heating need decreases as the technical regulations demands thicker insulation of the building body. Having a low-temperature space heating distribution system will also contribute to this decrease. More energy efficient lighting and technical equipment implies lower energy need on this field in newer buildings. Heat recovery of the extracted air leads to a much lower energy need for the heating coil in the air handling unit (AHU). Minimizing the air flow rate outside the operating hours and installing presence-, temperature- and CO_2 sensors, contributes to more energy efficient ventilation.

2.1.1 RESEARCH ON EXISTING PASSIVE HOUSES

When the results of the simulations are being evaluated in chapter 7, the question of *calculated* energy consumption versus *actual* energy consumption is brought up. Therefore this section presents research that has been performed on existing passive houses. The following studies deals with residential passive houses, as there is not much information to retrieve about existing kindergartens built to passive house standard,.

Measurements performed on residential passive houses in Denmark

Tine Steen Larsen et al. [2012] analyse eight residential passive houses in Denmark in detail from *year* 2008-2011. The results relevant for this master thesis are given in table 1. Only the best and the worst case out of the eight houses are cited, in other words the passive houses where the measured values differ respectively least and most from the calculated values.

Table 1 – Measurements on existing passive houses in Denmark

Category	Best case		Worst case	
	Calculated value	Measured value	Calculated value	Measured value
Thermal indoor climate				
Percentage of operating time when air tempe-rature exceeds 25 °C	0 %	4 %	1 %	32 %
Recommended air flow rates in occupied periods	Minimum 0,35 l/sm^2	0,24 l/sm^2	Minimum 0,35 l/sm^2	0,13 l/sm^2
Specific energy need				
Space heating	32 kWh/m^2year	17 kWh/m^2year	26 kWh/m^2year	55 kWh/m^2year
Primary energy need	113 kWh/m^2year	86,4 kWh/m^2year	105 kWh/m^2year	262 kWh/m^2year

The high deviation regarding indoor air temperature does not necessarily count for all the rooms in the house, since the value is an average value. Thus the thermal indoor climate varies in the different rooms in the house, which is why indoor environment control in the critical zones is recommended. The low air flow rates are explained with insufficient natural ventilation during wintertime. The high ventilation and space heating need in the worst case building is a result of frost damages to the water-based pre-heating surface, which made it necessary to give the electrical heating element first priority. This also affected the primary energy need, since the need for electrical energy was much higher in this house [Larsen et al., 2012].

Nevertheless the passive houses exceeded the calculated energy need with an average of 34 %. The results show that the users of the passive house affect the energy consumption considerably, T. Steen Larsen [2012, p. 67] states that "users behavior can mean a factor of 3-4 variation in the energy consumption".

Measurements performed on residential passive houses in Sweden

Another research on passive houses is performed by Patrik Rohdin et al. [2013] in Linköping, Sweden. Nine recently built passive houses are compared to thirty recent conventional houses built to local building code. The focus is on indoor environment and energy consumption.

The study indicates that the thermal comfort in the passive houses is within the limits defined in the local building code, but the occupant's perception is somewhat different. High indoor temperatures during summertime and cold floors were reported in general by the residents in the passive houses one year after the buildings were completed. Varying indoor temperatures were observed in a higher degree in the passive houses than in the conventional houses, which is assumed to be normal due to a more airtight building body. The high temperatures during summertime can also be explained by lack of external shading installed.

Whether or not the buildings fulfil the passive house criteria regarding energy consumption depends on the heating set points determined by the occupants. When 20 °C was used as set point, the annual energy use for heating was about 21 kWh/m^2. When increasing the set point to 24 °C, the annual energy use increased to about 35 kWh/m^2. The trend among the residents was a heating set point of 22-23 °C, which implied that the energy use exceeded the Swedish passive house design value of 21 kWh/m^2 per year.

Economic analysis of low-energy houses and passive houses in Belgium

Conventional houses, low-energy houses and passive houses are being compared in an analysis by A. Audenaert et al. [2007], in order to determine the economic viability of the three building types. The geometry of the houses is kept identical, but the building material differs from one house to another. A. Audenart et al. describes a low-energy house as "a building built according to special design criteria aimed at minimizing the buildings operating energy", while a passive house is "a type of low-energy building; design is oriented to make minimum exploitation of passive technologies". Figure 2 shows a graphical illustration of additional costs for low-energy houses and passive houses.

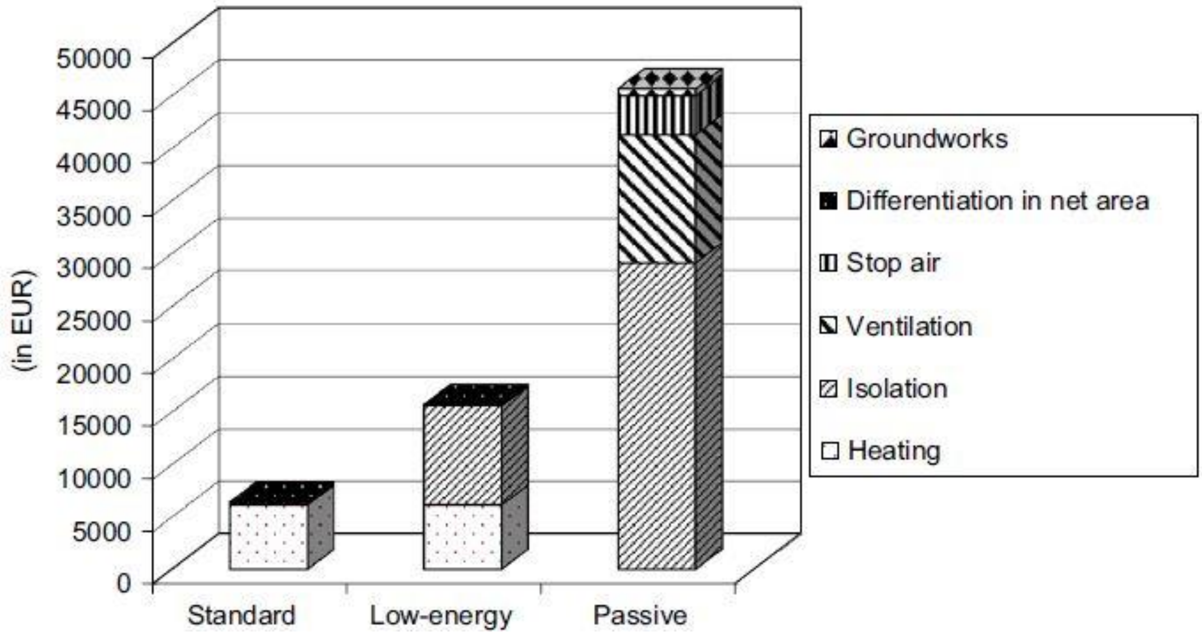

Figure 2 – Additional costs for low-energy houses and passive houses [Audenaert et al., 2007]

The costs are divided into six categories; groundworks, differentiation in net area, stop air, ventilation, insulation and heating. The largest surplus for the passive house is the costs for insulation and ventilation, at respectively 64 % and 27 % [Audenaert et al., 2007].

The report analyses different scenarios based on different growth rates from 0-25 %. Gas is the primary energy source for heating in Belgium, so a mean gas price of 0,04 €/kWh is used for calculating energy costs. Subsidies from the government are also taken into account; 721 € for the standard house and up to 2600 € for the low-energy and passive house. The economical calculation period is set to 20 years. With constant energy costs, the break-even time for the low-energy house is 12,3 years and 29,9 years for the passive house. With an annual energy price increase of 10 %, the break-even time is 9,5 years for the low-energy house and 24 years for the passive house. If the growth rate is set to 25 %, the passive house starts paying off after 12 years. Seeing that the energy price is difficult to predict, the report states that safest choice would be a low-energy house.

2.1.2 BREEAM-NOR

The case building for this project, which will be described in chapter 4, is BREEAM-NOR certified. A briefing about this system is therefore presented.

According to Norwegian Green Building Council [2012, p. 8-13], BREEAM is the world's leading building classification system regarding the environment. The purpose of the system is to promote sustainability and environmentally friendly buildings. BREEAM-NOR is a Norwegian adaption of this system, where the Norwegian building codes are implemented. The system consists of 10 categories with different criteria in each category, and the building receives points according to how many criteria it fulfills. The total score settles the building's label; Pass, Good, Very Good, Excellent and Outstanding. The most relevant categories for the case building in this project thesis are *Health and indoor environment*, *Energy utilization* and *Materials*.

2.2 SOLAR ENERGY

In Norway the energy from the sun is 1500 times larger than what is utilized [Andresen, 2008]. Solar energy can be used for electricity production through photovoltaic cells, or to heat buildings via solar collectors.

2.2.1 SOLAR IRRADIANCE

Figure 3 shows the annual irradiation on the earth surface.

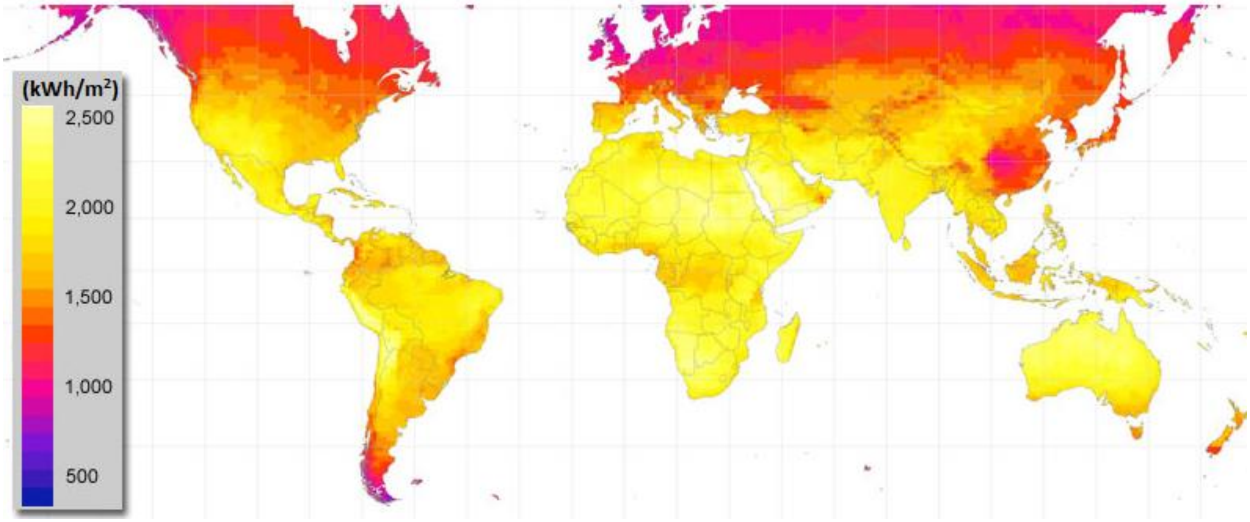

Figure 3 – Annual global irradiance [Kjellsson, 2009]

Notice that the radiation decreases with the distance from the equator. The amount of radiation reaching areas on the earth varies with the latitude, since the angle of incidence leads to a lower irradiance at high latitudes, see illustration in Figure 4.

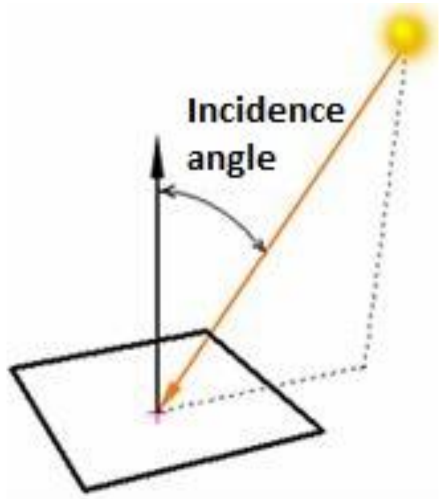

Figure 4 – Angle of incidence solar radiation [Natural Frequency, 2013]

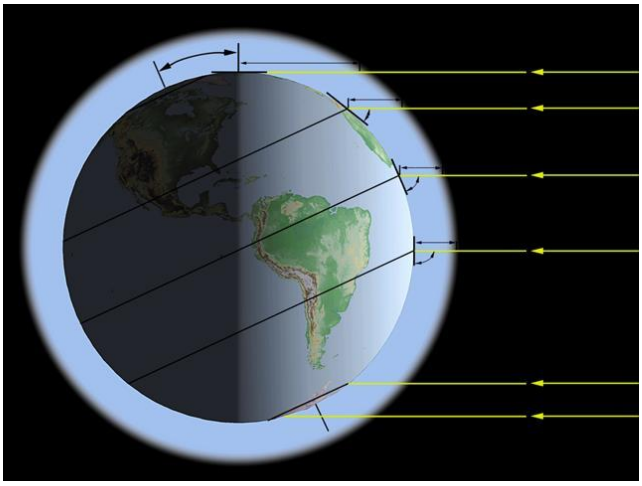

Figure 5 – Angle of incidence solar radiation [Natural Frequency, 2013]

The solar radiation increases as the angle of incidence moves closer to 0°. With 0°, which is the maximum solar radiation towards earth, the irradiance on horizontal ground is highest possible.

Another argument for decreased annual radiation in north is that the solar radiation has to travel a longer distance through the atmosphere before it reaches areas at high latitudes, as illustrated in figure 5. Longer travel time in the atmosphere leads to increased absorption and reflection before the radiation reaches the earth [Kjellsson, 2009]. A country like Norway that ranges from a latitude from 57° to 71° will have lower yearly irradiance than countries close to the equator line [Thorsnæs, 2013]. Trondheim's latitude is 63,28 °north [Allmetsat, 2013].

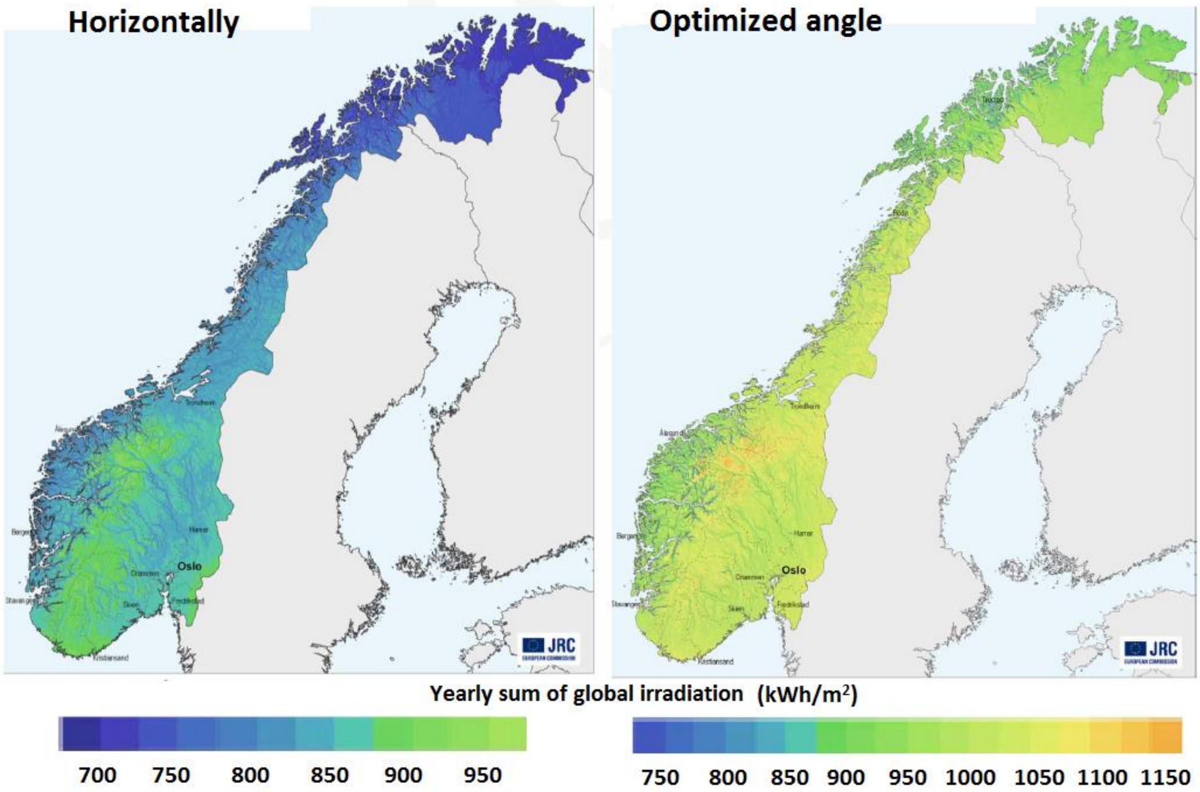

Figure 6 – Yearly irradiation in Norway - horizontally vs. optimized angle of solar collector [Zijdemans, 2012]

The way to compensate for high incident angle in Norway is to tilt the solar collectors till the incidence angle equals zero. As figure 6 shows, tilting the solar collectors to an optimal angle will affect the irradiation noticeably. According to Zijdemans, D. [2012, p. 110], the yearly irradiance in Trondheim will increase with 24 % when the solar collectors have an optimal tilting angle at 44°, compared to when it is installed horizontally.

2.2.2 SOLAR COLLECTORS

The solar collector is the element that converts solar irradiance to thermal energy. A heat transfer medium carries the heat in pipes to transfer it to an accumulator tank. Figure 7 shows an example of a heating system with solar collectors in which the solar energy is used to heat tap water.

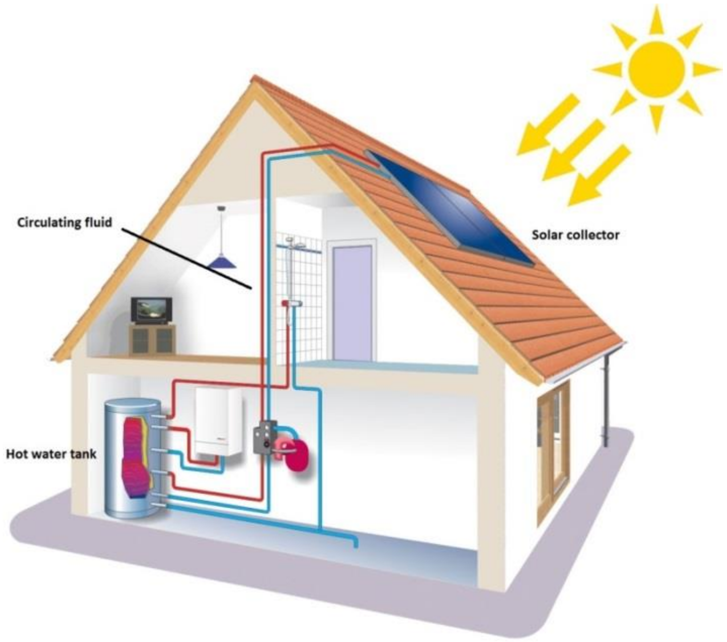

Figure 7 – Solar water heating system [Blue Sky Energy, 2013, edited by the author]

The solar collectors with circulating heat carrier fluid are placed on the roof to utilize as much as possible of the solar irradiance. The heated fluid is then pumped through a heat exchanger to emit heat to a hot water tank, which passes on hot tap water through the building's distribution system. Now the solar fluid has been cooled down, so the circle fulfilled when the fluid is once again heated up by circulating through the solar collectors.

Direct and indirect circulation system

Figure 8 and 9 shows respectively a direct and indirect solar heating system.

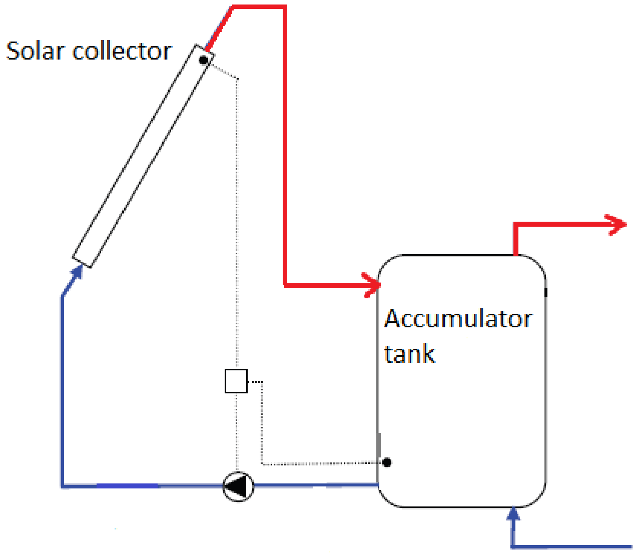

Figure 8 – Direct solar heating system [Andresen, 2008, p. 13, edited by the author]

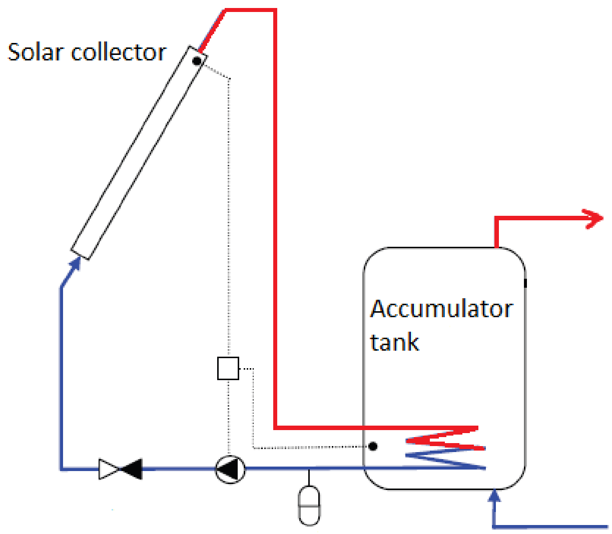

Figure 9 – Indirect solar heating system [Andresen, 2008, p. 12, edited by the author]

The solar heating system shown in figure 9 is an indirect system, which is most common. In this system the fluid in the solar circuit is physically separated from the water in the hot water tank via a heat exchanger as shown in figure 9. The heat exchanger can either be placed inside the hot water tank or separately on the outside. An outer covering on the hot water tank may also be used for transferring heat. With an indirect system, one can choose a frostproof fluid, i.e. water-glycol mixture. That will be beneficial during cold winters in Norway [Andresen, 2008, p. 11].

In a direct system (figure 8) the hot water from the solar collectors is led directly into an accumulator tank. This implies that the fluid circulating through the solar collectors has to be water, because it is in direct connection with the water that is being used as tap water in the building. Usually there is an extra hot water tank though, to avoid this direct mixing of water. To avoid that the water freezes during winter time, the pump that ensures circulation in the solar circuit is stopped. Then the water will drain off, and the hot water tank needs heat transfer from another heat source, for example by an electrical heating element.

Components of a solar collector
A solar collector consists mainly of three parts; an absorber, glazing and insulation, see figure 10.

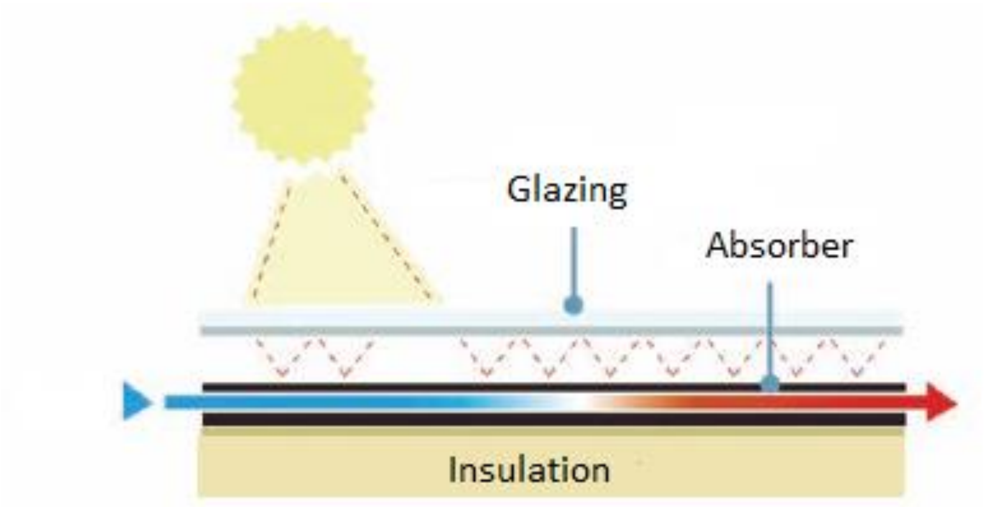

Figure 10 – Components of a solar collector [Andresen, 2008, p. 14, edited by the author]

The most central part is the absorber, as this is where the converting from solar radiation to heat occurs. The absorber is a thin metal plate with a special surface that absorbs around 90-95 % of the visual light from the sun [The German Solar Society, 2005, p. 20]. This spectral-selective surface coating emits less infrared radiation than other surfaces, which leads to lower heat loss from the solar collector.

In order to avoid low efficiency in cold climates it is important to insulate the solar collectors to avoid heat loss. Using highly transparent glazing of glass or plastic will also contribute to a higher efficiency. The mode of operation is to let the short-wave radiation inside and at the same time blocking the long-wave heat radiation from getting out [Andresen, 2008, p. 13]. Other benefits with the glazing are the resistance against wind, protection from moisture and physical protection against for example broken branches [The German Solar Society, 2005, p.22].

There are three main types of solar collectors; flat-plate collector, evacuated tube collector and low-temperature collectors.

Flat plate solar collector

This type of solar collector is installed on the case building, Haukåsen kindergarten. Due to the cold climate in northern countries, it is necessary that the fluid circulating through the solar collectors has a low freezing point. The flat plate solar collector allows such fluids. The operating range is from below -20 °C ambient air to around 80 °C hot water temperature. Unfortunately this solar collector has a higher heat loss than other collectors, which causes a lower efficiency [Auråen, 2013]. A cross-section of a typical flat plate solar collector can be seen in figure 11.

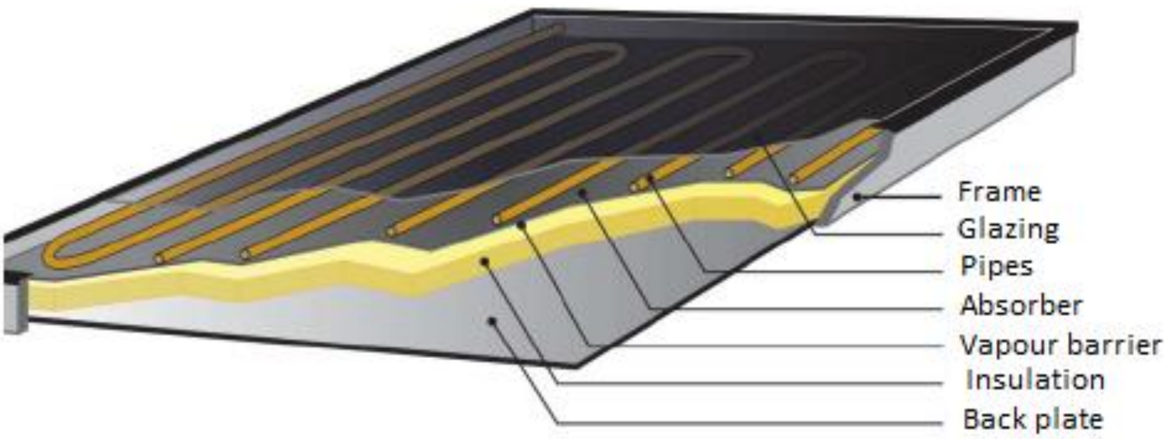

Figure 11 – Cross-section of flat plate solar collector [Zijdemans, 2012, edited by the author]

The glazing is attached to the front, with the absorber and piping right beneath. The heat carrying fluid flows through the pipes to collect heat from the absorber. The frame and back plate is robust and well-insulated to avoid heat loss. Other flat-plate solar collectors on the market are air collectors and vacuum collectors. Advantages and disadvantages of flat plate solar collectors are described in table 2.

Table 2 – Advantages and disadvantages with flat plate solar collectors [The German Solar Society, 2005, p. 25]

Advantages	**Disadvantages**
➢ Cheaper than an evacuated tube collector ➢ Multiple mounting options; on-roof, into the roof, facade mounting and free installation ➢ Good performance compared to price ➢ Possible for private persons to install by themselves	➢ Lower efficiency than evacuated tube collectors ➢ A supporting system is necessary for flat roof mounting (anchoring or counterweights) ➢ Not suitable for generating higher temperatures (steam generation, heat supply to absorption-type refrigerating machines) ➢ Requires more roof space than evacuated tube collectors

Evacuated tube solar collector

An evacuated tube collector consists of several glass tubes connected together, where the absorber is placed in vacuum inside the glass cylinders. The vacuum gives better thermal insulation, which leads to a lower heat loss from the solar collector and therefore a better efficiency [The German Society, 2005, p. 26]. There are mainly two types of evacuated tube collectors; heatpipes and u-pipes, see sketches in figures 12 and 13.

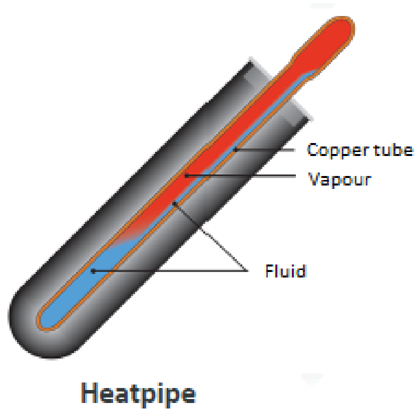

Figure 12 – Heatpipe evacuated tube collector [Zijdemans, 2012]

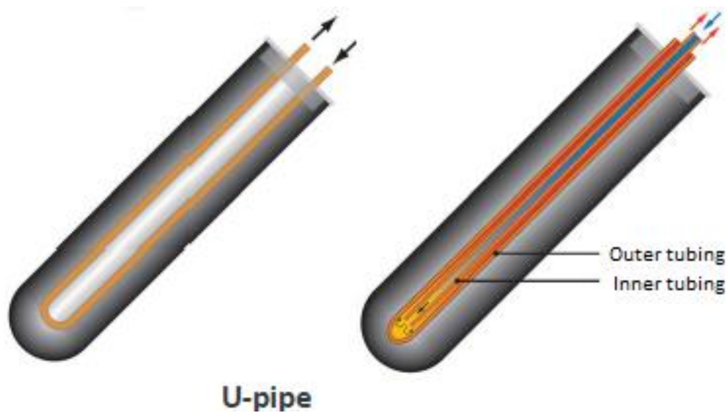

Figure 13 –U-pipe evacuated tube collector [Zijdemans, 2012]

Heatpipe evacuated tube collectors contain copper tubes with vacuum and a small quantity of fluid. The fluid evaporates at low temperatures (around 30 °C) because of the vacuum, and the vapour is led upwards in the tube to a heat exchanger on top of the solar collector. The vapour transfers heat via the heat exchanger by condensing to liquid phase. Then the fluid drains down to the bottom of the tube to start evaporating again. For making heat-pipes collectors function appropriate, they must be installed with a slope of minimum 20°. U-pipe collectors on the other hand, can be mounted horizontally on a flat roof. The heat transfer medium flows through metal tubing inside the larger vacuum cylinder, and heat is transferred in the same way as with flat plate solar collectors. Table 3 lists up advantages and disadvantages with an evacuated tube collector.

Table 3 – Advantages and disadvantages with evacuated tube solar collectors [Auråen, 2013, p. 15]

Advantages	Disadvantages
➢ High efficiency, even with high temperature difference between the absorber and surroundings ➢ Can achieve high efficiency even at low irradiations ➢ Supports space heating elements more effectively than the flat plate collectors ➢ Low weight gives it high mobility and makes it easy to transport ➢ U-pipes can be mounted horizontally, which implies lower wind load and lower installation costs	➢ More expensive than flat plate solar collector ➢ Cannot be used for in-roof installation ➢ Heat-pipes must be installed with minimum 20° slope, not aesthetic ideal for flat roof

There are also low-temperature solar collectors on the market utilized for example to heat swimming pools. Seeing that the subject of this master thesis is high-temperature solar circuits, the low-temperature collectors will not be further studied.

Efficiency of solar collectors

The methodology for calculating the efficiency of the solar collector is presented in this section. With stationary conditions, the effective thermal power, $\dot{Q}_{eff}$, from a solar collector is described by the following formula:

$$\dot{Q}_{eff} = I_{eff} - \dot{Q}_{loss} \left(\frac{W}{m^2}\right) \tag{1}$$

Where I_{eff} is the available solar irradiance (W/m^2), which can be calculated with formula (2). $\dot{Q}_{loss}$ represents the thermal losses (W/m^2), see formula (3).

$$I_{eff} = I * \tau * \alpha \left(\frac{W}{m^2}\right) \tag{2}$$

Here, I means irradiance directly on the glass pane (W/m^2), τ equals the degree of transmission of the glass and α represents the degree of absorption of the absorber.

$$\dot{Q}_{loss} = k * \Delta T \left(\frac{W}{m^2}\right) \tag{3}$$

The symbol k is the heat loss coefficient (W/m^{2}K), and ΔT is the temperature difference (K) between the absorber and the ambient air.

Formula (2) and (3) implemented in formula (1) gives:

$$\eta = \frac{\dot{Q}_{eff}}{I} = \frac{I*\tau*\alpha - k*\Delta T}{\mathrm{I}} = \tau * \alpha - \frac{k*\Delta T}{I} \tag{4}$$

With high absorber temperatures, the thermal losses will no longer increase linearly with the temperature difference between the absorber and ambient air. Therefore the constant k can be written as:

$$k = k_1 + k_2 \tag{5}$$

Where k_1 is the linear heat loss coefficient (originally k) and k_2 represents the quadratic heat loss coefficient. Inserting formula (5) in (4) gives the efficiency of a solar collector [The German Solar Society, 2005, p. 24]:

$$\eta = \tau * \alpha - \frac{k_1 * \Delta T}{I} - \frac{k_2 * \Delta T}{I} \tag{6}$$

Figure 14 shows typical solar collector efficiencies for different areas of application; heating of swimming pool, heating of buildings and process heating. The efficiency depends on the temperature difference between the absorber and the ambient air, corresponding to ΔT in formula (4) and (6).

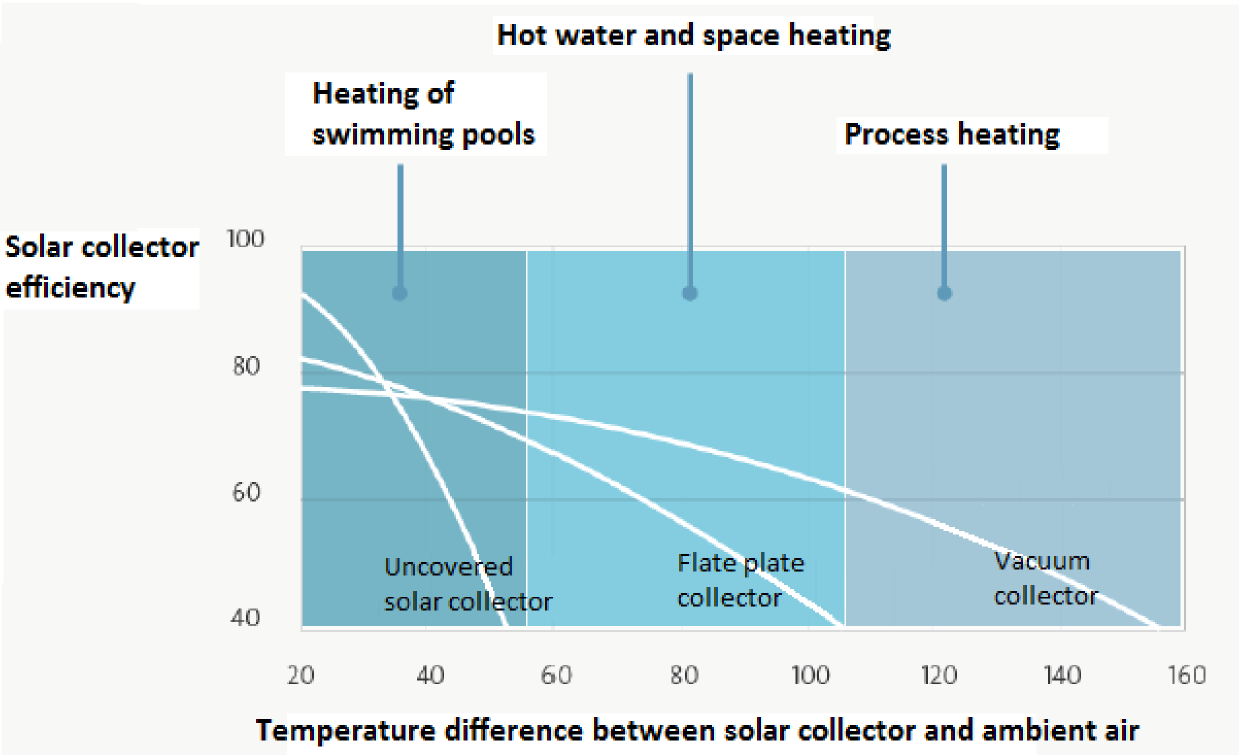

Figure 14 – Efficiency of different solar collectors depending on temperature difference between collector and ambient air [Andresen, 2008, p. 15, edited by the author]

2.2.3 SUMMARY SOLAR ENERGY

What can be pointed out from this literature review regarding solar energy for heating of buildings is that an indirect solar circulation system is most suited for Norwegian conditions. This is to avoid freezing of the heat transfer medium. A flat plate solar collector is cheaper than an evacuated tube collector, but has a lower efficiency. The efficiency of solar collectors varies with collector type, area of application and temperature difference between solar collector and ambient air.

2.3 GROUND SOURCE HEAT PUMP

A GSHP utilizes the ground as heat source. There are three types of heat sources within the term *ground heat source [Lystad, 2000]*:

- Top layer of soil in the ground, with horizontal piping and an indirect, closed-loop circuit
- Ground water, vertical piping and direct circuit
- Ground/rock, vertical piping and indirect circuit

This master thesis looks at energy wells as heat source, which means that the heat source is ground/rock, referred to as *ground source heat* or *geothermal heat*. The focus in this section will therefore be only on the ground/rock type of GSHP. The heat extraction from the ground is done by cooling the bore hole and the surrounding mountain formation [Stene, 2001, p. 4-19].

All types of heat pumps can in fact be combined with solar collectors, as long as the solar collectors are used for heating of domestic hot water. The only system that can be used for interseasonal storage of heat though, is the GSHP with vertical pipes. Recharging is possible with horizontal piping as well, but the recharging will mainly be natural, as the pipes lay closer to the surface with a higher temperature during summertime [Kjellsson, 2009].

2.3.1 MAIN PRINCIPLE

Figure 15 shows a section of a GSHP.

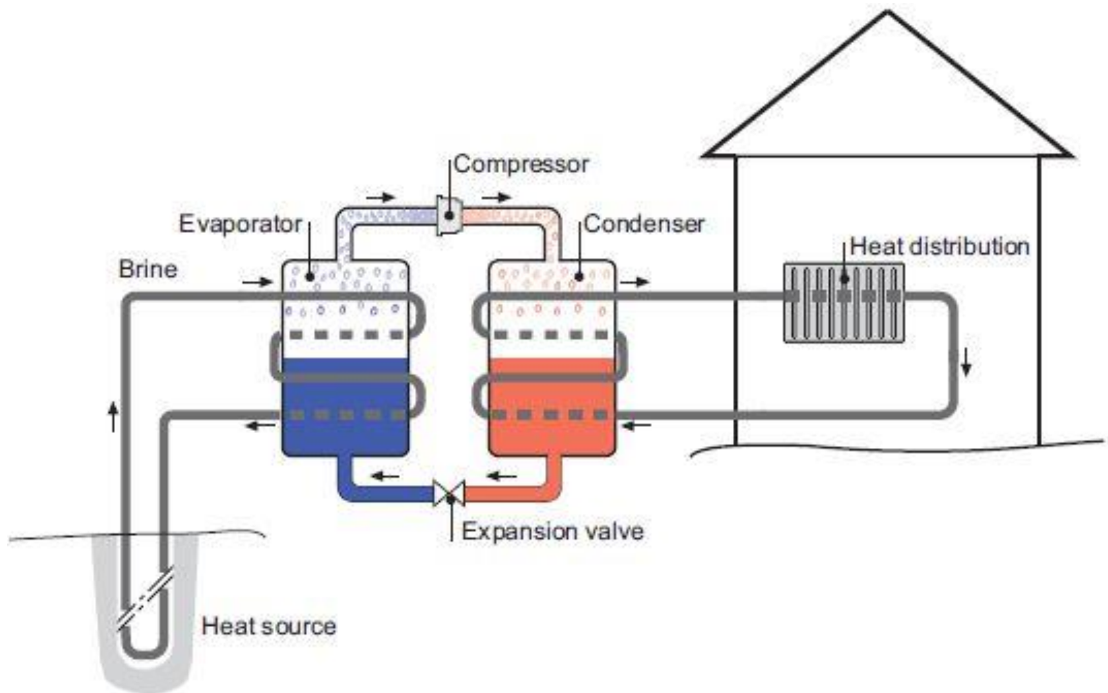

Figure 15 – Section of a ground source heat pump [Kjellsson, 2009]

Heat is brought from the ground to the evaporator in the heat pump with a brine being pumped in a close-looped circuit called u-pipes. The heat transfer medium in the circuit is an anti-freeze solution to prevent freezing of the medium during wintertime. When heat is transferred from the brine to the heat pump refrigerant, the brine is cooled down to below ground temperature and then led back to collect heat from the surrounding ground again. Advantages with this indirect circuit are that it prevents precipitation and corrosion. In addition the pump work is moderate [Lystad, 2000].

The heat pump refrigerant evaporates to a low-temperature vapour, and is then led in to the compressor. Here, an electrical powered motor provides compression of the refrigerant, which leads to increased pressure and temperature of the vapour. The refrigerant is then brought through the condenser, where the hot vapour is condensed by transferring heat to the buildings heat distribution system. The phase of the refrigerant is now liquid, but the pressure is still high. An expansion valve ensures pressure drop, which also leads to lower temperature of the refrigerant. The circuit is now fulfilled. The refrigerant is back to low-pressure and low-temperature liquid phase when again entering the evaporator.

If the heat pump is designed to deliver both heating and cooling, the ground is used as heat source during wintertime and heat sink at summertime. If this combined system is chosen, the depth of the bore holes can be reduced due to lower net heat extraction. The possible heat extraction from the ground depends on the grounds thermal conductivity and specific heat capacity, which again depends on the rock type and contents of quartz. Other factors that influences the heat extraction is the bedrock fracturing, the size of the drainage basin, terrain slope and distance between the bore holes [Stene, 2001, p. 4-20].

2.3.2 SIZING OF THE HEAT PUMP

A heat pump is normally sized to cover 50-70 % of the maximum heating load, which implies that 85-95 % of the total yearly heating need is covered by the heat pump [Stene, 2001]. Depending on the building and insulation, auxiliary heat is normally needed when the outside temperature is less than about 0 °C to -5 °C. The recent trend in residential buildings is however to install slightly larger heat pumps, to cover a higher percentage of the heating effect and thereby some additional minus degrees can be covered [Kjellsson, p. 49].

A disadvantage when installing these slightly larger heat pumps is that the heat pump has to be driven at part load most of the time. According to Jørn Stene part load implies decreased performance of the compressor [2013]. Figure 16 shows how the compressor performance varies with the compressor shaft power.

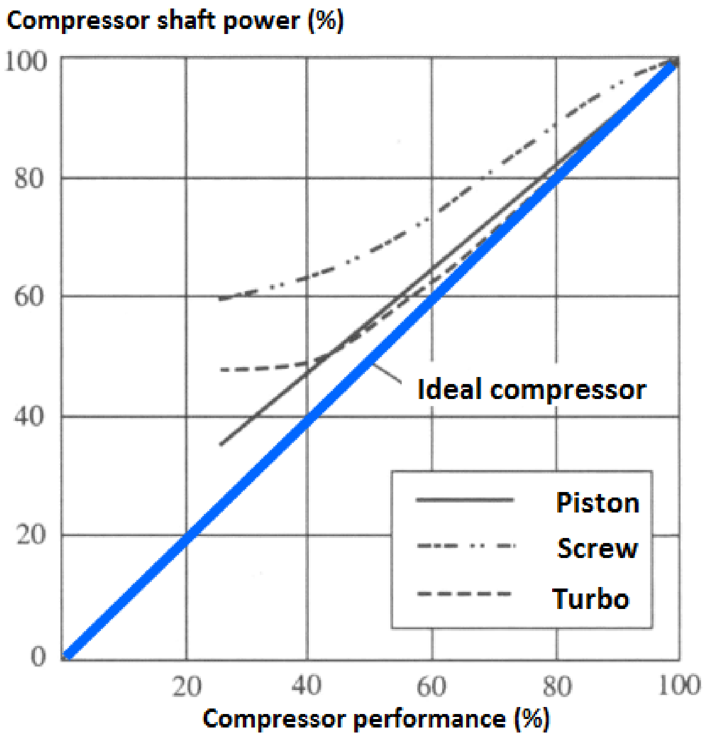

Figure 16 – Part load performance of compressors [Stene, 2013, p. 10]

With an ideal compressor, the performance would be 50 % at 50 % part load (see figure 16). Unfortunately this is only possible for the ideal Carnot process. In a real heat pump process, there will always be volumetric losses and friction losses. Therefore the compressor works with minimum losses when it operates at full heating power capacity.

The optimal sizing of the heat pump is determined by specific investment cost, operating costs (depending on energy consumption and energy prizes), maintenance costs and the building's heating load. The issue of the optimal sizing will be further discussed in chapter 7.

2.3.3 TEMPERATURE LEVEL AND IMPACT ON PERFORMANCE

A pressure-enthalpy diagram of the refrigerant R134a can be seen in figure 17.

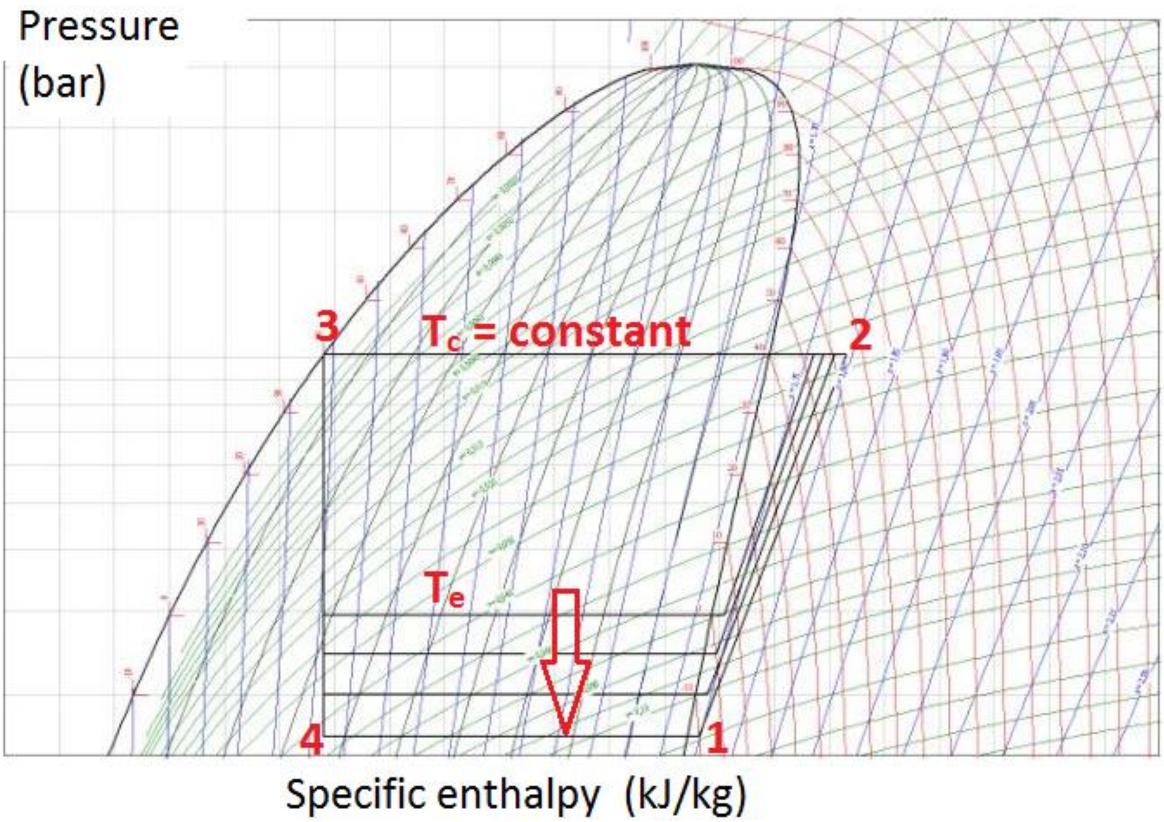

Figure 17 – Pressure - enthalpy diagram of R134a [Stene, 2013, p. 18]

The temperature difference between the evaporation temperature T_e (point 4 to 1 on figure 17) and condensation temperature T_c (point 2 to 3 on figure 17) is denoted the *temperature lift*. This temperature lift affects the heat pump coefficient of performance (COP). If T_e drops 1 °C, the COP decreases with about 2-3 %, same as for 1 °C increased T_c [Stene, 2013, p. 18]. A higher temperature lift leads to a higher compression work. With a higher compression work, the COP decreases:

$$COP = \frac{Q_c}{W} \tag{7}$$

Thus, low-temperature heat distribution systems are preferable, to obtain the lowest possible temperature lift in the heat pump. Floor heating and low-temperature radiators are examples of such distribution systems.

The possible temperature lift of the water in the distribution system depends on the return temperature of the water. Figure 18 illustrates the temperature levels of the working fluid and the distribution fluid when heat exchanging in the condenser.

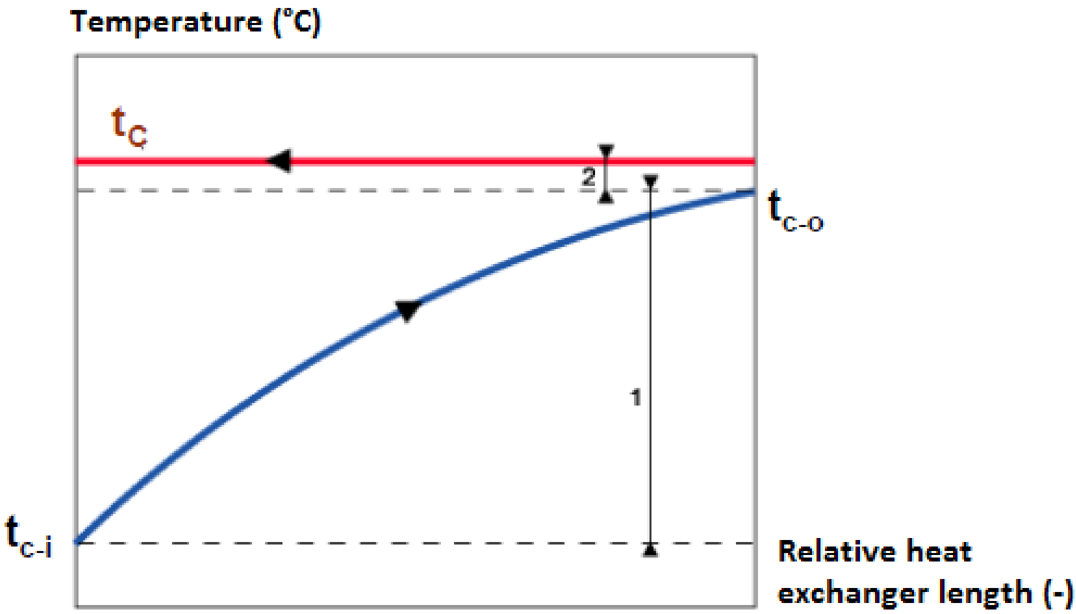

Figure 18 – Working fluid temperature (red line) and distribution circuit temperature (blue line)

The refrigerant temperature is illustrated by the red line. Seeing as the fluid is condensing, this red line indicates a constant temperature. The blue line represents the temperature of the fluid in the distribution circuit, which is being heated by heat exchanging with the working fluid.

The supply temperature, $t_{c\text{-}o}$, to the distribution circuit depends on the return temperature, $t_{c\text{-}I}$, as a result of limited maximal possible temperature lift indicated by *1* on figure 17. Other factors influencing the possible supply temperature to the distribution system are the mass flow rate and the size (surface area) of the condenser.

Condenser heating effect

Formula 7 gives the condensation heat effect, Q_c (kW):

$$Q_c = \dot{m} * c_p * \Delta T = U * A * LMTD \qquad (8)$$

Here $\dot{m}$ is the mass flow rate (kg/s), c_p is the specific heat capacity (kJ/kg K) and ΔT is the temperature difference (K) between inlet and outlet pipeline to the condenser. The condensation heat effect can also be expressed as the overall thermal transmittance of the condenser, U (W/m^{2}K) times the area, A (m^2), of the condenser times the logarithmic mean temperature difference (LMTD). This LMTD value is expressed with the following formula:

$$LMTD = \frac{\Delta T_{in} - \Delta T_{out}}{\ln\left(\frac{\Delta T_{in}}{\Delta T_{out}}\right)} = \frac{(T_c - T_{w,in}) - (T_c - T_{w,out})}{\ln\left(\frac{T_c - T_{w,in}}{T_c - T_{w,out}}\right)} \qquad (9)$$

Where $T_{w,in}$ and $T_{w,out}$ equals respectively the inlet and outlet water temperature (K) to the condenser.

Ground temperature level

The main advantage with using the ground as heat source is that it holds a stable temperature throughout the year, except in the upper level right beneath the surface. This gives stable working conditions for the heat pump. According to T. Lystad [2000], the ground temperature can be described by the following formula:

$$T_{ground} = T_0 + 1 + 0{,}02 * h \tag{10}$$

Where T_0 equals the mean annual outdoor temperature in degrees Celcius and h the depth below the surface in metres. Have in mind that the bedrock temperature differs with geographical location, depending on the latitude.

2.3.4 ENERGY WELLS

Figure 19 shows a sketch of a well.

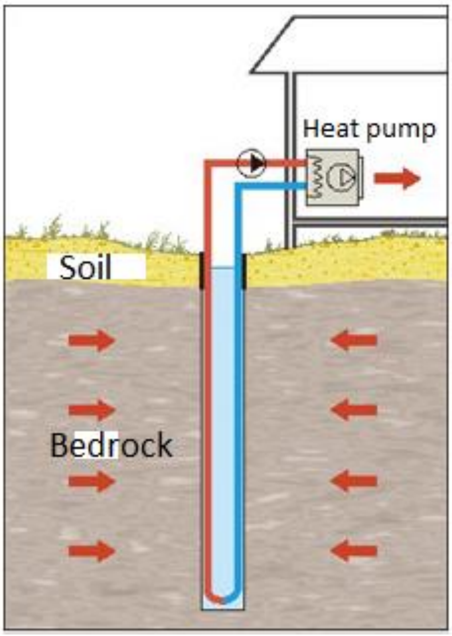

Figure 19 – Sketch of energy well [Norges Geologiske Undersøkelse, 2008, edited by the author]

Drilling bore holes for energy wells requires sufficient building plot area and suitable soil conditions. The wells are usually 80-300 metres deep, and typical dimensions of the bore holes are 115-165 mm [Stene, 2013]. The possible heat contribution from the upper layer above ground water level is minimal. In cases where the ground water level lies very deep, heat-conducting backfilling consisting of bentonite and quartz sand can be necessary [Lystad, 2000].

As mentioned in section 2.3.1, the heat extraction from the bedrock depends on among others the thermal conductivity of the ground, which varies with different rock and mineral compositions. According to *NGU* [Norges Geologiske Undersøkelse, 2008, p. 33], the thermal conductivity of the bedrock in Norway varies between 2-4 W/mK, with main area between 2,5-3,5 W/mK. The higher thermal conductivity, the less bore hole depth is needed, which implies savings in drilling costs. Still, NGU recommends drilling a few extra metres, to ensure sufficient heat extraction on the coldest days of the year.

The natural temperature of the bedrock will also affect the possible heat extraction. Normally, the temperature of the ground is 1-2 Celsius degrees higher than the mean annual air temperature. In places where the ground temperature is high, drilling costs are lower than in areas with lower ground temperature. If the bore holes are to be used as heat sink as well as heat source, the significance of the natural ground temperature is lower [NGU, 2008].

Depth

The required depth of the bore holes depends on [Kjellsson, 2009]:

- the building's heat load
- heat pump thermal effect output
- thermal conductivity and temperature in the ground
- distance to other GSHP systems
- depth of covering soil layer
- ground water level
- geological conditions and drilling costs

The exact bore hole depth needed can be calculated if the yearly specific effect transmission (W/m) is known, along with the evaporator capacity (W).

Costs

The costs of drilling bore holes depend on:

- depth of the soil above the bedrock that will be lined with steel pipes
- depth of the well
- type of bedrock and soil
- whether casing pipes are necessary, seen in figure 20

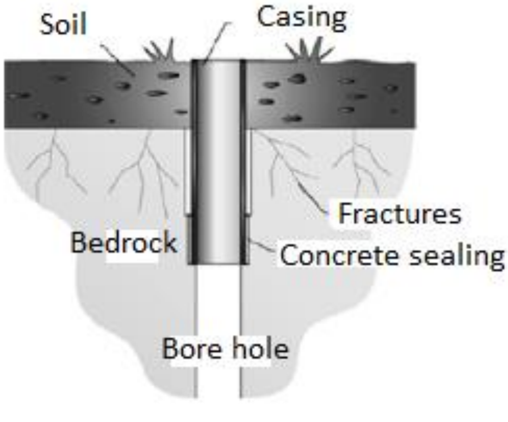

Figure 20 – Bore hole with steel casing [Stene, 2013, p. 76, edited by the author]

In Norway, the distance from the surface to the bedrock is in general quite low, 65 % of the built-up area have less than 30 metres depth to bedrock. The exceptions are valleys and parts of eastern Norway along with the areas *Trøndelag, Jæren, Østlandet og Finnmark*, where the thickness of the soil can be a considerable challenge *[NGU, 2008, p. 10]*. When drilling through soil, stainless steel casing is necessary, as shown in figure 20.

Distance between bore holes

Figure 21 shows how the brine temperature varies with the formation of the bore holes.

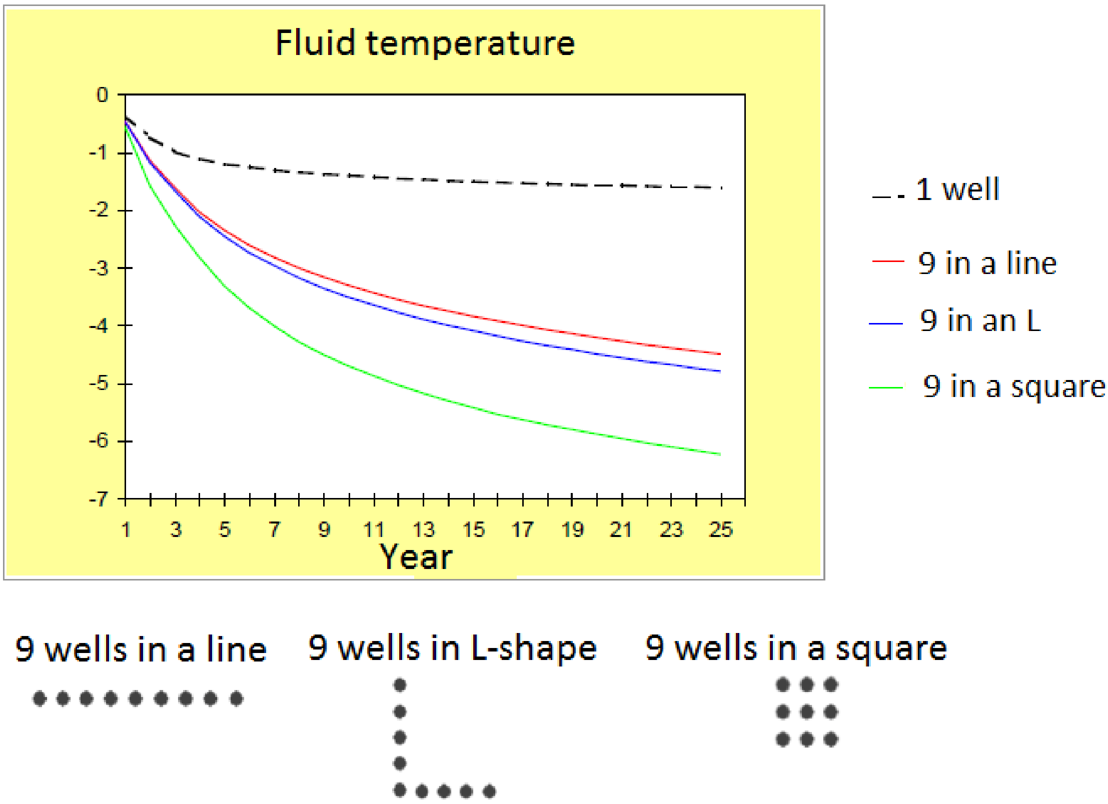

Figure 21 – The fluid temperature as a function of number of bore holes and shapes of bore hole collection [Stene, 2013, p. 95, edited by the author]

As figure 21 shows, the formation of the bore hole collection play an important role for the temperature of the brine. Over a long time period, the brine temperature drops more rapidly when bore holes are placed in a square than on a line. A lower brine temperature implies a lower evaporation temperature, which again leads to a lower efficiency of the heat pump [Stene, 2013]. Have in mind that the ground is not a perpetual heat source, so the ground temperature will indeed drop a few degrees over a long time period (25 years), no matter which bore hole formation is chosen. The only compensation is to inject heat back to the ground during the summer, see section 2.4.

Advantages and disadvantages

A disadvantage with energy wells as heat source is that the investment cost is very high. *Norges Geologiske Undersøkelse* [2008] estimates a cost of 150 000 – 250 000 NOK for a single family house, complete with energy wells and heat pump. Advantages are long lifetime (40 years), reliable heat source (in terms of stable temperature) and that the wells are suitable for plant dimensions from 6 kW and up to many megawatts.

2.3.5 GROUND HEAT EXCHANGER AND BRINE

Figure 22 shows a section of respectively single u-pipe to the left and double u-pipes to the right.

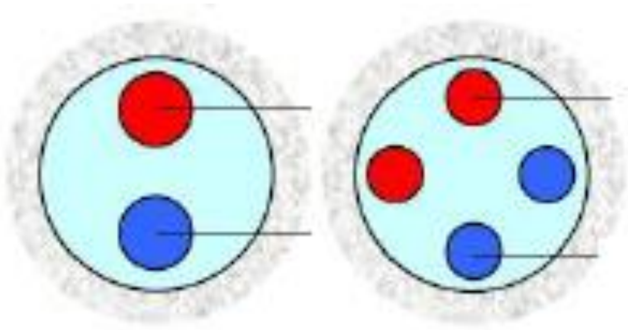

Figure 22 – Section of single and double u-pipes [Stene, 2013]

Double u-pipes are used when the building have a high effect need for cooling by dumping heat to the ground. The u-pipe inside the well acts as a heat exchanger to the wall of the bore hole. To achieve the best heat transfer, the material should not be too thick, and have a high thermal conductivity. The material is usually polyethylene (PEM). The thickness of the u-pipes depends on the pressure class, typically from 2 - 3,7 mm [Kjellsson, 2009].

Coaxial pipes are also on the market, but these are more expensive than the u-pipes. In Norway, only u-pipes are used. They are delivered with a bottom weight of 10-20 kg, to prevent buoyancy in case of icing on the pipes. The bottom weight also leads to simplified installation of the u-pipe [Stene, 2013].

The brine is the anti-freeze fluid circulating between the ground and the evaporator in the heat pump. To transfer the most heat, the solution should have a high specific heat capacity, high thermal conductivity and density, and a low viscosity. The brine should not be toxic, with regard to leakage.

2.3.5 SUMMARY GROUND SOURCE HEAT PUMP

Heat is brought from the ground to the heat pump with a brine being pumped in a close-looped circuit. The heat transfer medium is an anti-freeze solution usable for Norwegian bedrock. Possible heat extraction from the bedrock depends on the thermal conductivity of the ground, which varies with different rock and mineral compositions. The necessary depth of the energy wells depends on among others the buildings heat load, ground water level, depth of covering soil layer and distance to other ground source heat pump systems.

2.4 UNDERGROUND THERMAL ENERGY STORAGE

The term *underground thermal energy storage* (UTES) implies storing of excess heat from solar collectors during the summer, and utilizing the same heat during wintertime. The solar collectors are then connected to a shunt valve that leads the heat carrying fluid either to the hot water tank or to the ground. A system sketch with further explanation is presented in section 2.5. The building can also be cooled by injecting the excess heat to the ground during summertime, in which the ground functions as a heat sink. Figure 23 shows an illustration of heating and cooling of the building by respectively extracting and injecting heat to the bore holes.

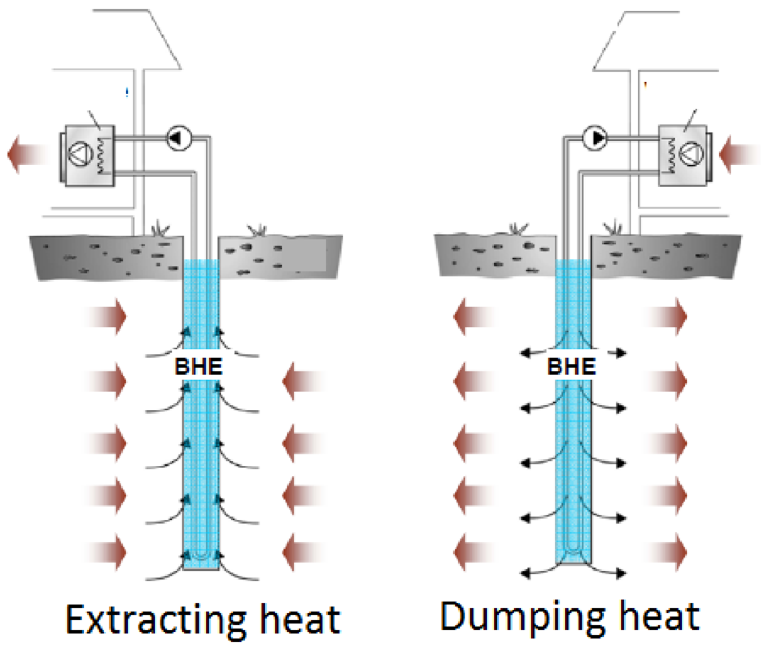

Figure 23 – Heating and cooling of a building with energy wells [Stene, 2013, p. 75, edited by the author]

By heating the ground like shown in figure 23, one can limit unwanted cooling of the ground over a long time period. Even though the ground will be heated naturally during the summer, this amount of heat is not enough to cover up for the extracted heat during the winter. The brine temperature to the evaporator can be held more stable when the ground temperature is held stable, unlike the example in figure 21. Without reclamation of heat to the ground during summertime, one can expect an effect output of 20-50 W per metre bore hole, and an energy output of 100-200 kWh per metre bore hole [Lystad, 2000].

Figure 24 gives an overview of the seasonal thermal energy storage concept, in chronological states from A-D.

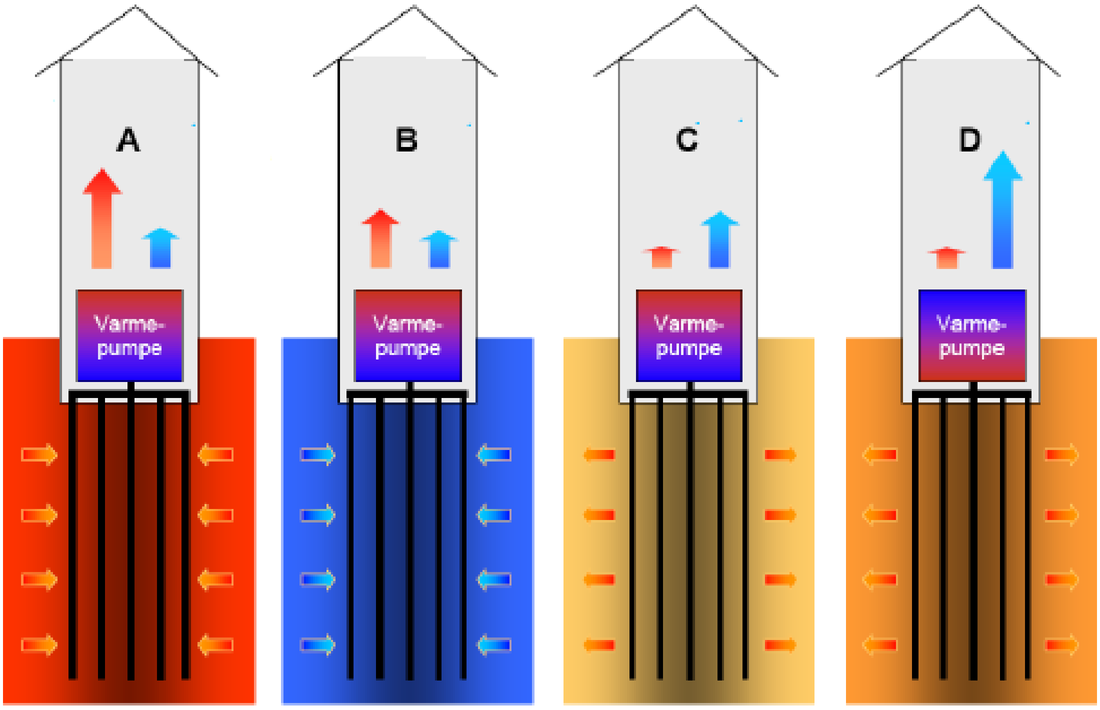

Figure 24 – Seasonal thermal energy storage [Stene, 2013, p. 99, edited by the author]

In state A the ground is at its warmest (in the autumn), and the heating load of the building at its largest. The ground is in this case used as heat source. In state B, the heating load of the building is still larger than the cooling load, so the ground works as heat source. As heat already have been extracted for some time by now (during state A), the blue colour of the ground symbolizes a colder ground than in state A. In state C, the cooling load is larger than the heating load, so heat is dumped back to the ground. This continues in state D, where the cooling load is at its largest, during summertime and early autumn.

2.5 COMBINED SYSTEM

This section describes the combined system of GSHP and solar collectors.

V. Trillat-Berdal [2006] performed an experimental analysis of a GSHP combined with solar collectors used in a 180 m^2 single family house from year 2004-2005. Storing of excess heat was performed. When the solar collectors had warmed up the hot water till the setpoint temperature, the excess solar heat was injected into the ground. The results showed that 34 % of the extracted heat from the ground came from solar heat that was injected into the ground.

Figure 25 shows a system sketch of the combined system; heat pump with solar collector and seasonal thermal energy storage in bore holes.

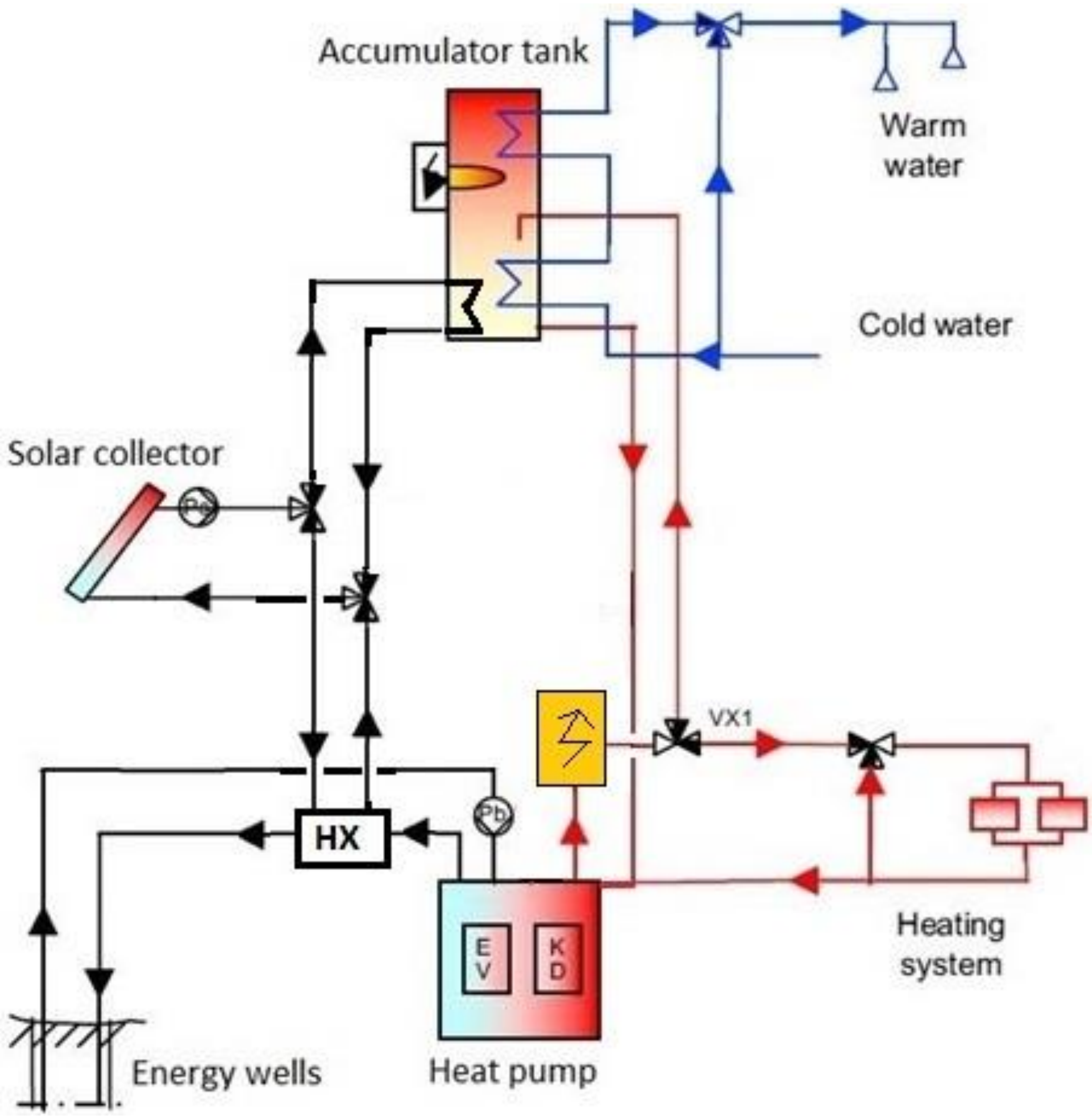

Figure 25 – Heat pump combined with solar collector for seasonal thermal energy storage [Kjellsson et al., 2009, edited by the author]

A shunt valve directs the heat from the solar collector either to the ground via a ground heat exchanger or to the hot water tank. In this system the solar collector do not contribute directly to space heating and the heating coil, this base load is covered by the heat pump alone. Stored heat from the solar collector in the ground will though indirectly contribute to space heating as the ground is the heat pump's heat source. The peak load effect is covered by an auxiliary heater (often being an electrical heater or boiler), placed directly after the heat pump in the supply direction of the distribution circuit (yellow box on figure 25). The hot water distribution circuit delivers hot water to the accumulator tank for hot water tapping, and also low-temperature water to the heating coil and space heating components of the building, i.e. radiators or floor heating.

A control system is needed to direct where the heat from the solar collector is being transferred. As with V. Trillat-Berdals system, a setpoint temperature of the hot water can be a reference point for where the solar heat used. Then, all the excess heat can be dumped to the bore holes.

2.5.1 ACCUMULATOR TANK WITH BUILT-IN HEAT EXCHANGERS

All solar water heating systems need a hot water accumulator tank, as the need for hot water does not necessarily happen simultaneously as the sun is up. Several different storage tanks exist, but the most common type for solar water heating systems is an accumulator tank with built-in heat exchangers [Andresen, 2008]. Figure 26 shows a cross-section of such an accumulator tank.

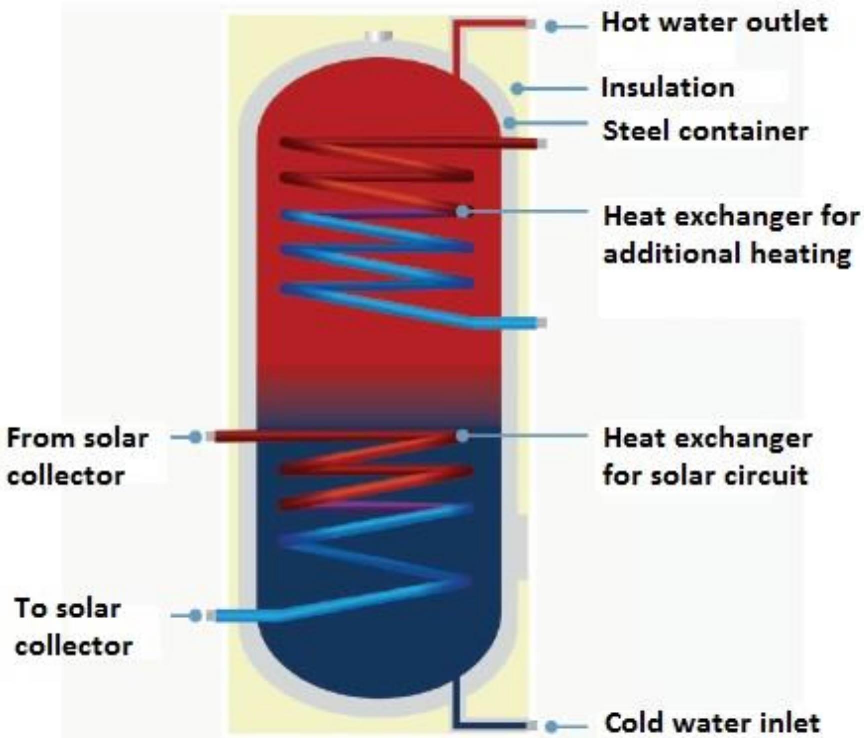

Figure 26 – Cross-section of an accumulator tank with built-in heat exchangers [Andresen, 2008, translated by the author]

Cold water is brought in to the bottom of the tank and is preheated by the solar collector circuit. The heated water flows to the upper part of the tank and is further heated by an additional heat source if necessary. The hot water being distributed is tapped from the top of the tank, and holds a minimum temperature of 60 °C to avoid legionary bacteria.

The shape and size of the tank depends on the heating need, solar collector area, system design and available space. In some cases the preheating and additional heating stage must be divided into two tanks, i.e. in engine rooms with low ceiling height [Andresen, 2008].

Temperature stratification

Two factors are important to consider when choosing a hot water storage tank; the first is temperature stratification. In a stratified tank, the cold water stays in the lower part and the hot water in the upper part, with minimum stirring in between (cf. figure 26). This is important in a system containing solar collectors, as the collector efficiency will be higher with a low inlet temperature. An accumulator tank with large height is preferable to maintain the stratification.

Thermal insulation

A second factor important to remember when choosing a tank is the thermal insulation. A well-insulated tank contributes to a lower heat loss. I. Andresen [2008] recommends minimum 5-20 cm of mineral wool or equivalent as insulation for the accumulator tank.

2.5.2 SOLAR CIRCUIT COMPONENTS AND CONTROL

The piping should have good thermal insulation in order to reduce heat losses in the system. I. Andresen [2008] recommends minimum 20-30 mm insulation depending on the pipe diameter. Other components in the solar circuit are circulation pump, expansion tank, safety valve, filters, temperature and pressure gauges and valves for filling and tapping of heat transfer medium.

A solar heating system is controlled by switching on and off the circulation pump dependent on the temperature difference between the hot water tank and the solar collector. When the temperature in the solar collector exceeds the temperature in the lower part of the tank (blue-coloured section on figure 26), the circulation pump is on. A more detailed description of the control strategy with examples from Haukåsen kindergarten is given in section 4.3.2.

2.5.3 HEAT PUMP CONTROL

Jørn Stene [2001, p. 5-2] describes three main rules of heat pump control:

1. Always utilize the heat pump at maximum capacity before any peak load unit is enabled
2. The supply temperature for heat distribution should be kept as low as possible while still maintaining an acceptable thermal comfort in the building
3. The heat pump shall have the lowest possible electricity consumption

Space heating

When a heat pump is used for space heating, the control is based on an outdoor compensation curve. Thus, the supply temperature to the distribution system depends on the outdoor air temperature. When the ambient air temperature is low, the need for space heating is high, which implies that a higher supply temperature is needed in the distribution system. When the ambient air temperature increases, the need for space heating decreases and the heat pump is set to deliver a lower temperature. This ensures an energy efficient operation of the heat pump [Stene, 2001].

As the outdoor compensation curve is based on the distribution temperature in the building and the local climate, each heat plant has its own curve. A more detailed description of the curve and typical values are given in section 4.3.1, where the control of the heat pump at Haukåsen kindergarten is described.

Tap water heating
Heat pumps can be used also for combined space heating and tap water heating. The most common system is having a heat pump preheat the tap water and installing an additional heat source for further heating, such as an electrical heating element. With this system, the heat pumps percentage coverage of the tap water heating depends on the outdoor compensation curve. If the heat pump ought to cover a higher percentage of tap water heating, a lower temperature limit must be defined in the outdoor compensation curve. This concept is further explained in section 4.3.1.

Night set-back
One should be careful when combining a heat pump with night set-back control. As explained in section 2.3.2, the heat pump is sized to cover 50-70 % of the maximum heating load. When night set-back is implemented, a high heating load is needed in the morning in order to compensate for the lower nighttime heating setpoint. Due to the heat pump being sized for 50-70 % effect coverage, it will have difficulties meeting the high morning heating load. This implies a higher share of peak load effect, which further implies higher operational costs [Stene, 2001].

2.6 SIMULATION TOOLS

The simulations in this master thesis are carried out in the program *IDA ICE*. This section presents a brief literature study of the chosen simulation program and other available simulation tools.

2.6.1 IDA ICE

IDA *Indoor Climate and Energy* (IDA ICE) is a simulation tool developed by EQUA Simulations AB. The program is used to study a buildings energy consumption and thermal indoor climate of individual zones within the building, through a detailed, dynamic multi-zone simulation over a whole year [EQUA Simulation AB, 2013].

An advantage with IDA ICE is that the program is compatible with BIM-files. All common 2D and 3D CAD files can be imported, and also complete IFC-models. Take into account that if an IFC-model is implemented, the file should be stripped for all other subjects except for the architectural model. This is due to that IDA ICE can only read the architectural model, and all other models implemented within the file will slow down the simulations considerably.

According to EQUA Simulation AB [2013, p. 9-11], the user interface is divided into three different levels, wizard level, standard level and expert level, with different support and scopes. In this master thesis, the model is defined at standard level. That implies that default AHU and primary plant system are automatically inserted when a new model is initiated. With ESBO (as in the case for this project), more complex plants can be built. Figure 27 shows the user interface of IDA ICE 4.591, the version used in this project.

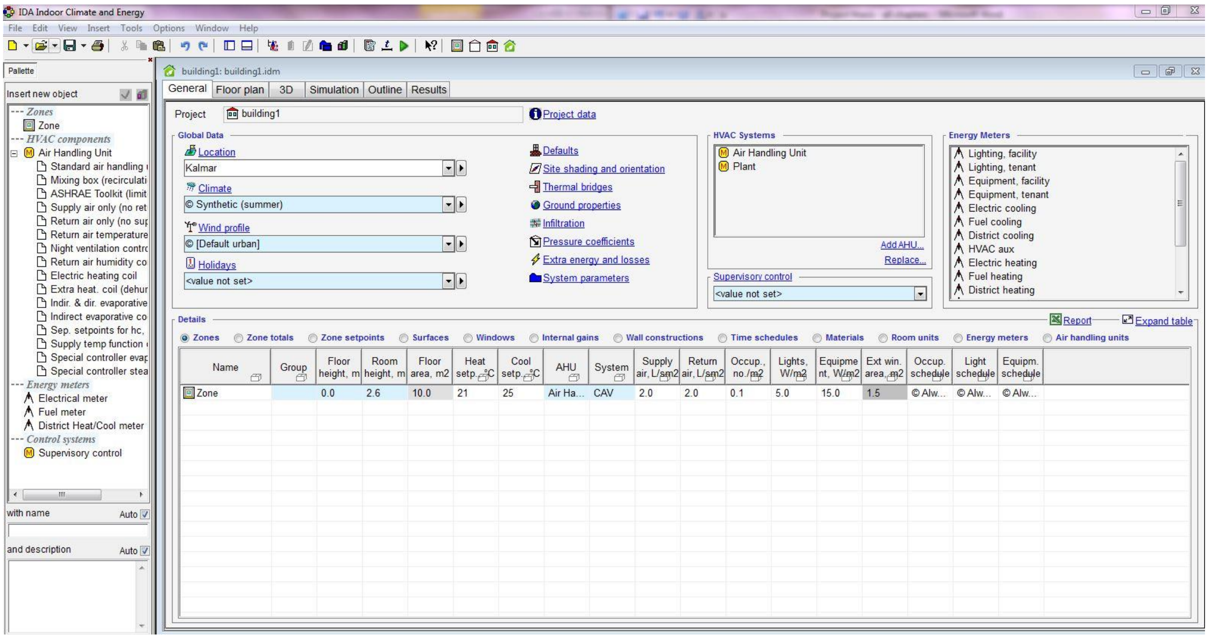

Figure 27 – IDA ICE user interface

The interface consists of a palette on the left side, an upper menu line on the top, and the main window in the front. The main window has different tabs for maneuvering between the general window, floor plan, 3D view, simulation inputs, outline and the results window. The 3D-view makes it easier to verify the building body and zone inputs, and also to study animations, such as shading animation, over a 24-hour period.

2.6.2 SIMIEN

SIMIEN is a Norwegian energy simulation program developed and sold by Programbyggerne ANS. The program calculates energy consumption and power requirement, as well as analyzing the indoor environment of a building with multiple thermal zones. Other areas of application are setting an energy label of the building, and dimensioning the heating and ventilation system [Programbyggerne, 2013]. When typing input values of the building, the program constantly evaluates all input according to the Norwegian building code. According to Nord et al.[2010, p. 9], this is what makes SIMIEN the most commonly used energy simulation program in Norway.

Nord et al. [2010, p. 9] further states that "SIMIEN has a very user friendly interface, which makes it quite quick and easy to use". SIMIEN does however have some disadvantages. The energy supply system and ventilation system cannot be modeled in detail. The only input values are yearly system efficiencies and COP values. Specialized heating system solutions and advanced VAV-control cannot be designed in the program. Therefore SIMIEN may not be the best suited program for simulations in this project, as the system solution of the case building is quite advanced, both regarding energy supply, heating and ventilation.

2.6.3 ENERGY PLUS

EnergyPlus is used for modeling a buildings heating and cooling need, lighting, ventilation and other energy flows. The program is a stand-alone simulation program that reads inputs and outputs as text files. Some of the simulation possibilities are for example time steps of less than an hour, modular systems and plant integrated with heat balance based zone simulation, multizone air flows, thermal comfort and photovoltaic systems [Nord et al., 2010].

The program contains three basic components; a heat and mass balance module, and a building systems simulation module, and a simulation manager that controls the entire simulation process. According to Nord et al. [2010, p. 12], the simulation results of EnergyPlus agreed to within 1 % of the analytical results except for the mean zone humidity ratio of 3 %.

2.6.4 TRNSYS

The simulation program *TRNSYS* was developed at the University of Wisconsin, and has been available for more than 35 years. The program can be used to design and simulate buildings with multiple thermal zones and different internal heat gains. One can analyze alternative energy supply systems, different ventilation control strategies, multizone airflows, solar design and thermal performance of the building [Klein et al., 2010].

A benefit with TRNSYS is that the source code is available to the user. In that way the users can add customized components to the model by using all common programming languages (like C, C++, PASCAL, FORTRAN etc.). A disadvantage on the other hand, is that even though some default information is provided by TRNSYS, the user still needs to have very detailed information about the building and system and know where to implement this information [Nord et al., 2010, p. 15].

3. SIMULATIONS

In order to analyse the energy need, heat plant design and operation of a building, a case model is established. Simulations are carried out in the simulation program *IDA ICE 4.6.* Chapter 4 contains information and descriptions about the case building, Haukåsen kindergarten. Measuring values and operational experiences are also included in this chapter, to obtain a better understanding of how the heat plant and ventilation system works out for the occupants. A detailed review of the input data used for simulations is given in chapter 5, making it possible to verify the simulation results.

A reference model is first developed, where the aim is to make the model as equal as possible to the real case building, to achieve realistic results. The detail engineering report by Rambøll [Wormdal and Smits, 2013] is mainly used as basis for the input data, as well as the Norwegian building code [NS3031, 2011] and Norwegian passive house criteria [Dokka et al., 2009]. All building sketches, data sheets and other documentation about Haukåsen kindergarten are received from Rambøll.

After the reference model of the building is developed, different parametres are changed to test how the system is affected. The results of this analysis are given in chapter 7.

4. CASE BUILDING: HAUKÅSEN KINDERGARTEN

The case building used for simulations is a 1 000 m^2 kindergarten located at Haukåsen, Trondheim, completed in august 2013. The heated area of the building is 813 m^2, with a base area of 330 m^2. A picture of the building is shown in figure 28, displaying the north side closest to the photographer.

Figure 28 – Haukåsen kindergarten (photographed by author 2013-22-09)

Haukåsen kindergarten has two floors, an attic where the heat plant and AHU are placed, an outdoor workshop room connected to the south facade (see figure 28) and toolshed/trolley parking in the north-west building (the grey building element to the right in figure 28). The kindergarten houses 15 employees and 56 children of age 0-6 years old, but as the concept of the kindergarten is *outdoors kindergarten,* it implies that the building is left almost deserted most of the day.

The author chose this building as case building because its heating concept corresponds to the subject of the master thesis; solar collector combined with a heat pump that utilizes energy wells as heat source. This chapter gives a description of the building, while more detailed input data used for simulations follows in chapter 5.

4.1 BREEAM-NOR CLASSIFICATION

The municipality of Trondheim wanted an energy efficient and environmentally friendly building, with BREEAM classification "Very good". With the chosen solutions, Haukåsen kindergarten is the first in Norway of its kind to be BREEAM-NOR certified. During the projecting phase, the earnings of the solar collector was questioned due to the high investment costs. But in order to achieve the label "very good", a solar collector needed to be installed, as this gives the building a greener profile. Being BREEAM certified also implies that the building's construction fulfil Norwegian passive house criteria [Dokka et al., 2009]. The buildings energy label is a green A.

4.2 CONSTRUCTION

Haukåsen kindergarten has woodwork as main material both outdoors and indoors. The inner supporting walls, flooring and roof constructions are all made of massive wood. More details of the construction follow in section 5.3.1. Pictures taken inside the kindergarten can be seen in figures 29 and 30.

Figure 29 – Meeting room

Figure 30 – Workshop room

4.3 HEAT PLANT

The heat plant at Haukåsen kindergarten differs somewhat from the system explained in section 2.5, in that there is no coupling from the solar collector to the energy wells. This clears the possibility of storing solar energy seasonally. A simplified sketch of the heat plant can be seen in figure 31, the original plant sketch can be found in appendix A.1-A.2.

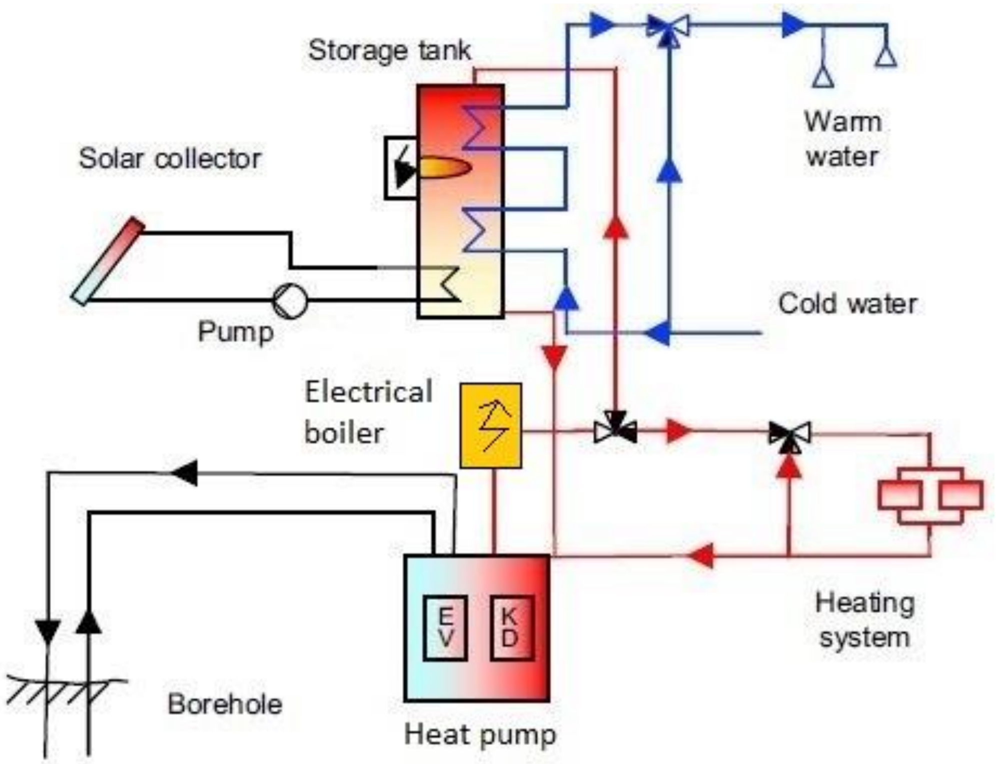

Figure 31 - Sketch of heat supply system Haukåsen kindergarten [Kjellsson et al., 2009, edited by the author]

The base load of the kindergarten's yearly heating need is covered by a GSHP, while the peak load is covered by an electrical boiler. The solar collector contributes only to preheating of tap water and not to space heating and the heating coil circuit. The actual plant differs somewhat from figure 31 as it has one accumulator tank after the heat pump, and two hot water tanks; one for water preheating by the heat pump and solar collector and one for additional heating carried out by an electrical heating element. The tap water temperature is 60 °C from the top of the hot water tank. More details about the tap water system follow in section 4.3.2.

Figure 32 shows a picture taken in the technical room, displaying the heat pump to the right, electrical boiler to the left and solar collector circuit with water heater tanks in the back. Detailed descriptions of the components and the control strategy are given in the following sections.

Figure 32 – Heat plant

4.3.1 HEAT PUMP AND ELECTRICAL BOILER

The GSHP with its connection to the bore holes can be seen in figures 33 and 34.

Figure 33 – Heat pump of brand Airwell

Figure 34 – Connection to bore holes

The green component in figure 34 is the pump ensuring circulation of the heat transfer medium between the u-pipes in the ground and the heat pump. This heat transfer medium is a mixture of water and ethylene glycol (20 %) that flows through an indirect circuit in the black-coloured tubing. The inlet and outlet pipes to the heat pump evaporator can be seen in figure 33. All the components of the heat pump, i.e. compressor, condenser, evaporator and expansion valve, come as a built-in box (white box on figure 33) from the supplier. The heat pump contributes to heating of tap water, as well as transferring heat for space heating purposes and the heating coil in the AHU. R410A is the refrigerant in the heat pump.

The electrical boiler (figure 35) is placed directly after the heat pump on the supply side (cf. figure 31), to ensure sufficient heat supply when peak loads occur. Thus this component also contributes to tap water heating, space heating and the heating coil.

Figure 35 – Electrical boiler

Both the heat pump and the electrical boiler operate according to the same outdoor compensation curve, given in figure 36.

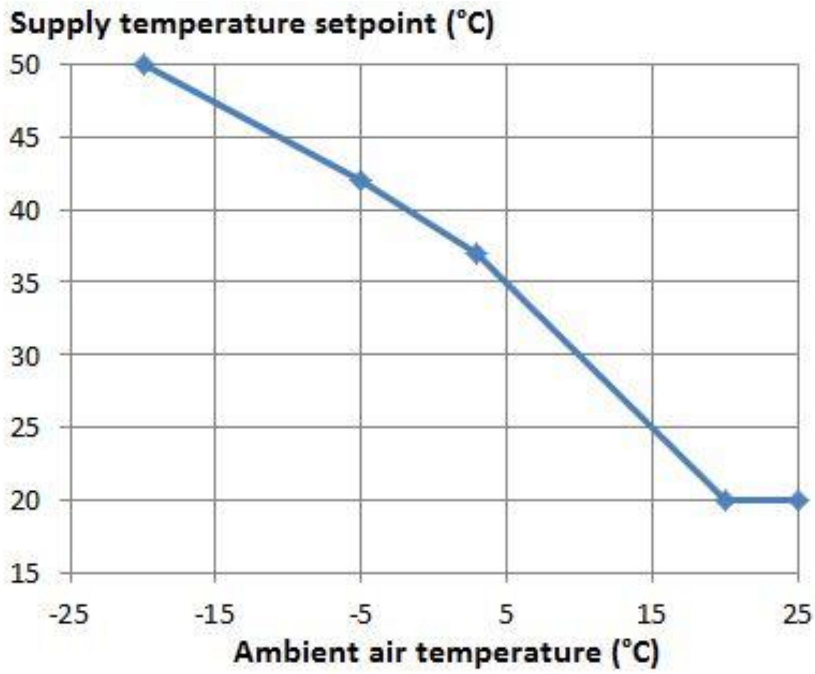

Figure 36 – Outdoor compensation curve [Schneider Electric, 2013]

The curve illustrates the setpoint supply temperature for space heating dependent on the ambient air temperature. When the ambient air temperature reaches the dimensioning outdoor temperature of -20 °C, the supply temperature to the distribution system shall be at its maximum level of 50 °C. As the ambient air temperature increases, the necessary supply distribution temperature will decrease due to higher heat gains in the building. The supply temperature still has a lower block limit of 20 °C, due to the system being a combined system. By this means both heating of ventilation air, space heating and preheating of tap water. In order to maintain the preheating of tap water at a certain minimum temperature level, this lower block limit is defined.

As mentioned earlier, both the heat pump and the electrical boiler operate according to this curve. Due to limited achievable temperature lift in the heat pump condenser (see section 2.3.3), the electrical boiler contributes when needed. The temperature sensor, from which the boiler receives its signal, is placed directly after the boiler on the supply side.

4.3.2 SOLAR COLLECTOR CIRCUIT

Figure 37 shows the circulation pump and measuring instrument of the solar collector circuit, which is an indirect circuit (see section 2.2.2). The circulating medium transfers heat from the solar collector to a preheating storage tank. Figure 38 shows this preheating tank, along with the heating tank from which the domestic hot water is being tapped. A larger tank would be more preferable for a correct temperature stratification (see section 2.5.1), but the technical room at Haukåsen kindergarten has two tanks due to low ceiling height.

Figure 37 – Solar collector pump and measuring instrument

Figure 38 – Tap water heaters: preheating and additional heating

The control strategy of the tap water heating is explained by showing an excerpt from the sanitary sketch (see appendix A.1), see figure 39.

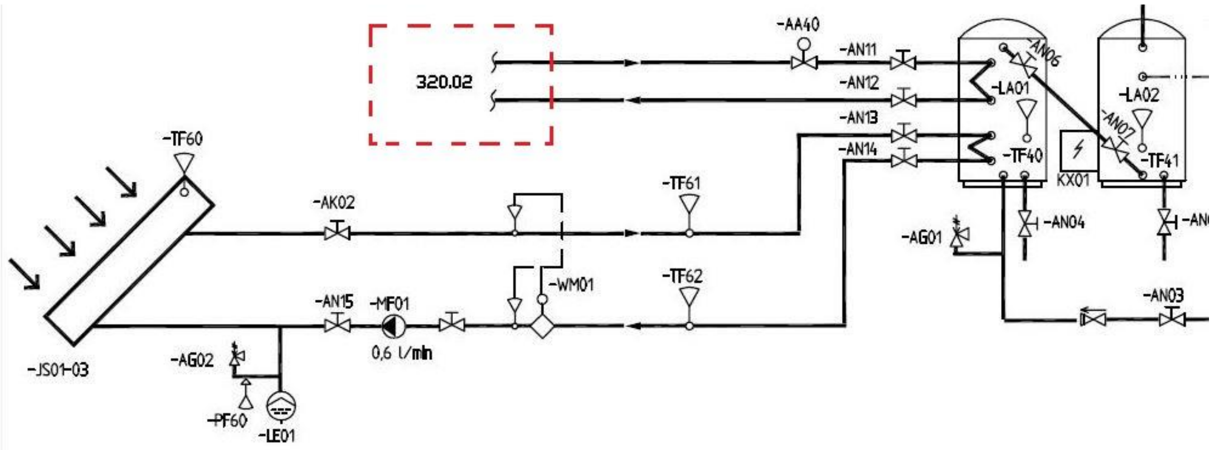

Figure 39 – Excerpt from the sanitary sketch at Haukåsen kindergarten

The circuit named *320.02* (red circle in figure 39) is the supply and return water connected to the heat pump and electrical boiler. This circuit and the solar collector circuit are both connected to the preheating storage tank, seen to the left of the two tanks in figure 38. The water in circuit *320.02* and the solar fluid are not circulating at the same time, due to the planned control principle of the heat plant.

The control strategy is to let the solar collector contribute as much as possible to heating of tap water. This is carried out by having circulation of the solar fluid as long as the temperature in the solar collector (sensor *TF60*) is higher than in the preheating tank (sensor *TF40*). In that case, the valve *AA40* closes and the pump *MF01* is set into operation. Otherwise, the pump *MF01* is stopped, and valve *AA40* is opened to let the water in circuit *320.02* preheat the tap water. In other words, there is a sequential control between the solar circuit and heat pump circuit, preferably making the solar collector contribute as much as possible.

Placing of solar collector

A matter worth further reflection is the placing of solar collector. Figure 40 shows the south-west facade, where the solar collector are mounted vertically (tilted 90° from horizontal level) on the outside of the workshop room. Theoretically a more ideal placing would be on the roof, as explained in section 2.2.1.

Figure 40 – Solar collector (red circle) placed on south-west facade

4.3.3 ENERGY SUPPLY DISTRIBUTION

According to Rambølls detail engineering report regarding energy supply [2013, p. 12] dated 2013-05-16, the percentage coverage of each energy supply alternative was planned to be as in table 4.

Table 4 - Planned percentage distribution of energy in Haukåsen kindergarten

	Electricity	**Heat pump**	**Solar collector**
Space heating (%)	2	98	0
Heating of tap water (%)	27	43	30
Heating coil (%)	2	98	0
Total heating need (%)	10	80	10
Energy for electric purposes (%)	100	-	-

This distribution meets the requirement for being qualified as a passive house according to *Prosjektrapport 42 [Dokka et al., 2009]*, since minimum 40 % of the buildings total heat need is supplied with a sustainable energy source.

4.3.4 HEAT DISTRIBUTION SYSTEM

An excerpt from the heat supply system sketch (appendix A.2) can be seen in figure 41, displaying the heat distribution circuits.

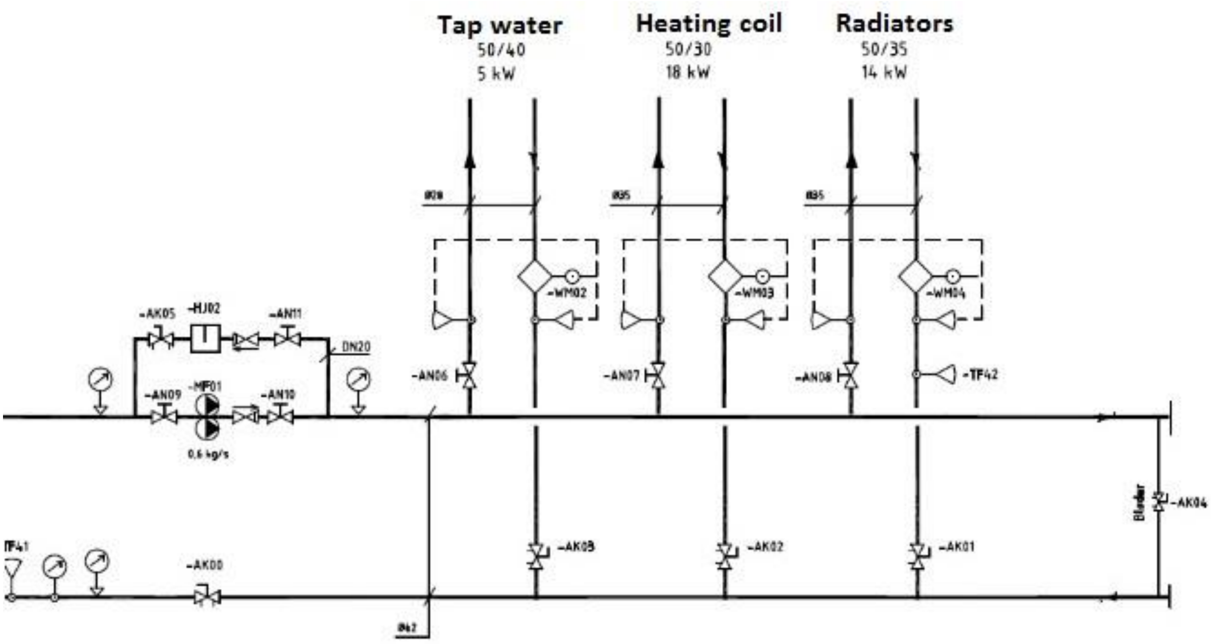

Figure 41 – Excerpt from the heat supply system sketch displaying the heat distribution

Heat is being distributed to the tap water circuit (tap water heater), space heating circuit and heating coil circuit in parallel, in order to supply all three circuits with the same water temperature. The twin pump *MF01* ensures sufficient pressure for supplying all three circuits, and the remaining water runs through the bleeder valve (to the right on the figure). The piping for the three circuits can be seen in figure 42. The six pipes are respectively supply and return pipes to radiators, supply and return to the heating coil and supply and return to the tap water heater.

Figure 42 – Pipelines for heat distribution

Figure 43 – Placing of radiators

The radiators are low-temperature radiators, which mean that they have a large surface area in order to emit enough heat to the surrounding space. Figure 43 shows a picture taken in a meeting room at first floor, showing the size and placing of a radiator. The placing underneath a window is to counteract cold air flow from the window.

4.4 AIR HANDLING UNIT

Figure 44 displays the AHU system sketch [Fläktwoods, 2012] and figure 45 shows a picture of the installed AHU in the technical room of the kindergarten.

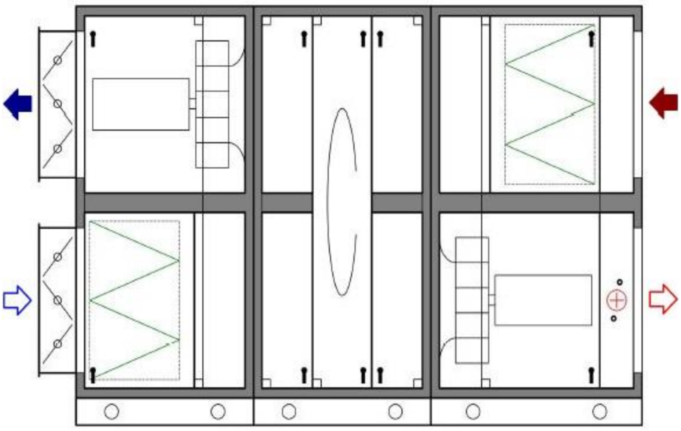

Figure 44 - AHU system sketch [Fläktwoods, 2012]

Figure 45 – Air handling unit

The AHU consists of inlet and outlet grids, dampers, filters, fans, a heat recovery unit and a heating coil. Maximum air flow capacity is of 8400 m^3/h. The heat recovery unit is a rotating unit with an temperature efficiency of 83,2 %. The control strategy for heating of supply air is to let the recovery unit run at full rotational speed before the heating coil is enabled for supplementary heating. This will lead to a lower heat need from the heat plant, which further gives a more energy efficient system.

4.5 CENTRAL CONTROL SYSTEM

The central control system can be seen in figure 46.

Figure 46 – Control system

The box seen in figure 46 contains all the controllers sending out signals to maintain the wanted operation of the system. The controllers receive their input signal from measuring instruments, being for example outdoor temperature, temperature in zone 7 or temperature after the heat recovery unit.

4.6 MEASURING VALUES

The author went in February 2014 to the kindergarten along with the caretaker Stian Sandnes [2014] for an inspection of the technical room and to hear about the operation experience since the acquisition in August 2013. The objective was also to receive access to measuring data in order to compare the data with the simulation results.

As a result of the HVAC contractor being declared bankrupt, the installation of energy metres was unfortunately not completed. After 9 months in operation the metres are still not functioning (till today's date). This implies that logging of the plant components is not possible to perform, as no flow meters are connected (cf. formula 7, section 2.3.3). The only measuring data that exists are the total supplied electricity and the input power to the heat pump compressor, AHU and the water heater. The input power values (measuring date: 2014-02-24) are given in table 5 [Sandnes, 2014].

Table 5 – Energy supplied to heat pump compressor, water heater and AHU

Heat pump compressor (kWh)	Water heater (kWh)	AHU (kWh)
13 578	9 041	18 564

As the kindergarten's first six months in operation consists of testing and balancing the heating and ventilation system, the values from table 5 cannot be used directly for comparison with IDA ICE results. Taking the AHU as an example, it runs at full capacity for 24 hours during these six months, instead of having demand controlled ventilation.

The kindergarten's total supply of electricity is more suitable for comparison with IDA ICE simulations though. The electricity supply is logged weekly by the energy consultancy company *Entro* (the author received access to the measuring values from the caretaker at the kindergarten *[Sandnes, 2014]*). As the building was set into operation in august 2013, the available measuring data is from 2013-08-26 until today's date. With a mean weekly energy use of 813 kWh until 2013-05-19, the total annual energy consumption is estimated to be 42 276 kWh.

Operational experiences

When it comes to the operational perception so far, both the occupants and the caretaker were satisfied. They experienced good thermal comfort and indoor air quality. According to S. Sandnes, the ventilation system was not correctly balanced from the start, which led to underpressure in some zones. After adjusting the supply- and return air pressure, the pressure level was satisfactory.

5. INPUT DATA

This chapter gives a review of the input data implemented in the IDA ICE model. The purpose is to give a thoroughly and detailed description, so that verification of the model can be performed. This chapter is therefore of most interest to readers wanting to verify the model. Other readers might have more interest in jumping directly to the next chapter. The structure of the chapter is given in an attempt to make external verification as easily as possible to carry out.

The entire input data report printed out directly by IDA ICE is given in appendix B.1. As this report does not make a complete basis for understanding the input data, the author chose to give an entire review in this chapter. Note that the first step when setting up the model is to start a new building with a single zone. In that way, a lot of default information is defined in IDA ICE. Only the input data that have been changed from the programs initial set-up is presented.

Modifications from the model used in the project thesis

A reference model was first established during the project thesis work, but some changes are made to improve the model, in order to resemble the actual kindergarten to a greater extent. The most important changes are the following:

- **Correct climate file**
  Östersund was used as climate file in the old model, as *Trondheim* was not available due to an older version of IDA ICE.
- **Night set-back of heating setpoint temperature**
  22 °C heating setpoint during operating hours and 19 °C otherwise.
- **Correct supply temperature to heating coil and radiators**
  Previously this was automatically set to 60 °C by IDA ICE. This is now corrected to 50 °C.
- **More exact occupancy hours**
  The old model contained occupants present during the entire operating hours from 07:00-16:30, while the new model is based on the actual occupancy (see section 5.6.4)
- **Ventilation macros ensuring minimum air flow rates during operating hours**
- **Reduced lightning power input**

Basis for the input data

All documentation is received from Rambøll. Unless otherwise stated, all input data is based on the detail engineering report by Rambøll [Wormdal and Smits, 2013], as mentioned in chapter 3. This detail engineering report corresponds with the Norwegian building code and Norwegian passive house requirements [Dokka et al., 2009]. The following table summarizes the requirements and recommendations given by T.H.

Dokka. Only the demands that are relevant for simulation results of the kindergarten is listed.

Table 6 – Requirements and recommendations for kindergartens built to passive house standard

Element	Requirement	Recommendation
Specific heating need	Maximum *25 kWh/m²year*	
Specific cooling need	Maximum *0 kWh/m²year*	
Air flow rates		Minimum mean during operating hours: *6 m³/hm²* Minimum mean outside operating hours: *1 m³/hm²*
CO_2 level	Maximum *1 000 ppm (vol)*	
Thermal comfort		Maximum number of hours with air temperature over 26 °C: *50 hours*
Expected internal heat gains		Lighting: 6 W/m² Equipment: 2 W/m² Occupants: 6 W/m²

Another requirement is that at least 40 % of the total heating need has to be covered by renewable energy resources [Dokka et al., 2009].

5.1 FLOOR PLAN

The building body was drawn automatically by the program when importing an IFC-file made by the architect (file received from Rambøll). The internal walls were on the other hand not defined by the IFC-model, so all zones are drawn manually. The toolshed/trolley parking in the north-west building (see figure 28) is excluded from the building body, as it is not defined as available area of the building. Available area of all zones is noted in table 7. Figure 47 shows the division of zones on each floor plan.

Table 7 – Zone areas

Zones	Available area (m²)
1	173,6
2	35,8
3	174,0
4	182,2
5	69,8
6	60,6
7	58,6
8	120,1
TOTAL	**874,7**

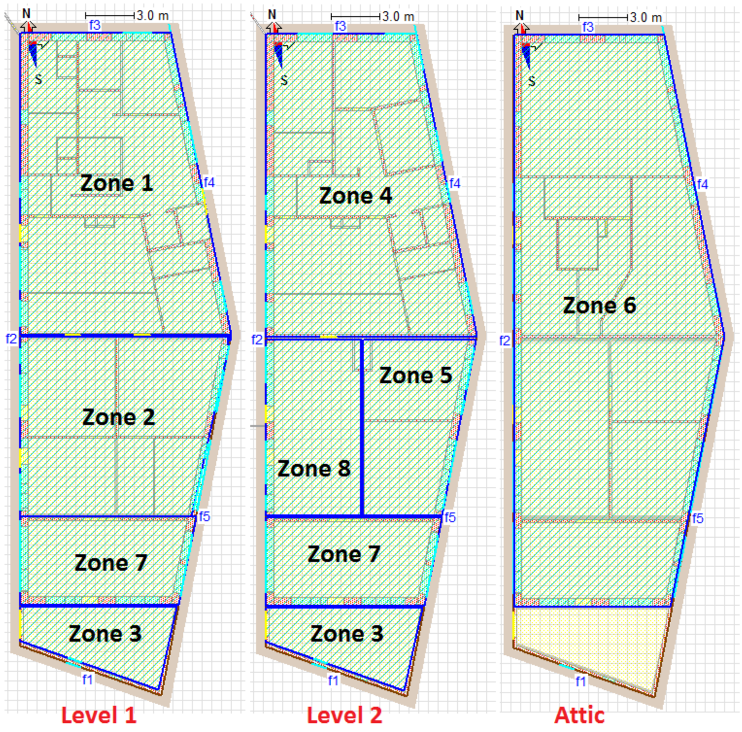

Figure 47 – Floor plans with zone division in IDA ICE

The zone division was made on the basis of the planned zone control projected by Rambøll. The division also fulfills the demand from NS3031 about separating the zones based on the exposure for direct sunlight [Standard Norge, 2011]. Complete sketches can be found in appendix B.2, showing type and placement of sensors in the rooms. A further description of this control will follow in section 5.6 about zone inputs. The division of zones is as follows:

Floor 1

- Zone 1: Reception, locutory, wardrobes and bathrooms/WC
- Zone 2: Workshop room, group rooms and common room
- Zone 3: Outdoor workshop room (over two floors)
- Zone 7: Common room (over two floors)

Floor 2

- Zone 4: Workrooms/meeting rooms, wardrobes and bathrooms/WC
- Zone 5: Group rooms
- Zone 8: Kitchen/common room

Attic

- Zone 6: Engine room

3D view

A 3D view of the building body from IDA ICE is shown in figure 48, displaying the south-west and west facade, and the toolshed/trolley parking to the left in the figure.

Figure 48 – 3D view of Haukåsen kindergarten in IDA ICE

The toolshed is only visible due to the IFC-file, it does not form a part of the IDA ICE building body. The solar collector can be seen to the right, mounted on the south-west facade.

5.2 GLOBAL DATA AND OPERATING TIME

The location and climate file is set to Trondheim, while the alternative named *suburban* defines the wind conditions around the kindergarten.

The Norwegian building code NS3031 states that the operating time for a kindergarten is 10 hours, 5 days a week, 52 weeks a year [2011, p. 40]. But since the aim with the simulations is to create a realistic model, the actual holidays of the kindergarten were implemented. The operating time of the kindergarten is from 07:00 to 16:30 5 days a week [Trondheim Trondheim Kommune]. Complete list over all the holidays can be found in appendix B.3.

5.3 DEFAULTS

This section contains default values implemented in the general tab in IDA ICE. The detail engineering report is used as basis for the input data. According to Jörgen Ericsson [2014], the zone model fidelity has to be set to *Energy*, since the *Climate model* only works with rectangular zones, which is not the case in this kindergarten.

5.3.1 ELEMENTS OF CONSTRUCTION

Table 8 indicates the U-values of the construction.

Table 8 – U-values external construction

Element	External walls	External roof	External floor	Windows and doors
U-value (W/m^2K)	0,09	0,09	0,07	0,7

Defining default walls of the building is done by selecting materials from the IDA ICE database that lies closest to the materials in the detailed sectional drawings. These drawings can be seen in appendix B.4. Then, the thickness of each material is adjusted so that the total U-value is as close as possible to the respective U-value from table 8. One should also have in mind that the total thickness of the construction element cannot differ too much from the reality, as this will affect the buildings total weight and thermal mass. Table 9 displays the chosen default construction. The materials are listed in a sequence from floor top/wall inside to floor bottom/wall outside.

Table 9 – Default construction IDA ICE

Element	**Material and thickness**		**Total U-value (W/m²K)**
External walls	Wood Light insulation Frames cc600 insul Wood	100 mm 300 mm 50 mm 100 mm	0,09
External walls toolshed	Extruded polystyrene	50 mm	0,64
Internal walls	Gypsum Wood Gypsum	20 mm 100 mm 20 mm	0,94
External floor	Floor coating Light insulation Concrete	50 mm 500 mm 250 mm	0,07
Internal floors	Floor coating Chip board Light insulation Wood	2 mm 22 mm 50 mm 160 mm	0,35
Roof	Wood Light insulation Gypsum	160 mm 350 mm 5 mm	0,09

The windows are made *detailed default*, to have more design options, such as draw control and shading. Name of chosen window type is *Triple low-e (argon) – deflected (WIN7)*. This triple glazed window was chosen because its U-value of 0, 698 W/m²K corresponds to table 8.

5.3.2 THERMAL BRIDGES, INFILTRATION AND PRESSURE COEFFICIENTS

Normalized thermal bridge value for the entire building envelope is set to 0,03 W/m²K. The pressure test showed that air changes per hour (ACH) in Haukåsen kindergarten is 0,14 [Wormdal and Smits, 2013], which meets the requirement from NS3031 of 0,2 ACH. Regarding the pressure coefficients, all facades and roof are autofilled with the option *sheltered*. The reason is that the forest makes the kindergarten more sheltered from the wind [Eriksson, 2014].

5.3.3 SITE SHADING AND ORIENTATION

To achieve the most realistic shading for this building, trees ought to be implemented in south-west, west, and north-west direction. Jörgen Ericsson warned on the other hand that IDA ICE will then calculate the shadow from each and every leaf, which will lead to considerable longer simulation time. Some objects of the type *shading building* has therefore been implemented, with space in between to achieve the same effect as streak of light from the forest. Figure 49 shows a freeze-frame from a shadow animation carried out in the 3D view of IDA ICE, which illustrates how the building is shaded by the objects. The solar collector can be seen on the south-west facade to the right in the figure. Point in time is 16:41, 2013-03-22.

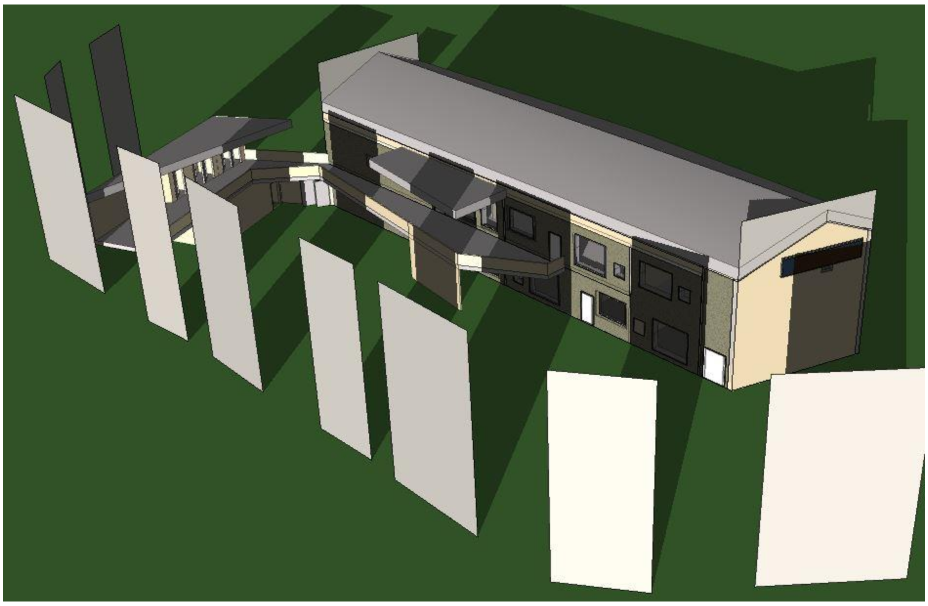

Figure 49 – Shadow animation in IDA ICE

5.3.4 GROUND PROPERTIES

Under the basement slab, the default *Ground layers* (0,7 metres soil and 10 metres stone) is chosen from the IDA ICE database. Outside basement slab, *Ground outside walls* (default soil, 1 metre) is chosen. According to J. Eriksson, only 0,5 metres of the deepest layer will be taken into account when the simulation runs. Therefore there is no consequence of choosing a default ground layer, as long as the order of the materials and the depth of the top layers are correct.

5.3.5 GENERATOR EFFICIENCIES

As for generator efficiencies, one can choose between electric, fuel or district heating. As the heat pump has electrical driven motor and in addition the top heating is an electrical boiler, electrical heating is set with a COP of 2,11 [Wormdal and Smits, 2013].

5.3.6 DOMESTIC HOT WATER USE AND DISTRIBUTION LOSSES

All distribution system losses are set to *Good*. As this is a newly built kindergarten, one would expect low losses. Based on NS3031, the average hot water use per year is set to 10 kWh/m^2. When it comes to the hot water schedule, J. Eriksson recommended avoiding a flat curve, as this will give an unrealistic picture in a system with solar collector. The leader at Haukåsen kindergarten, Pål Gerhard Rystad [2014], was contacted to define the peak loads of the hot water use correctly. Figure 50 shows a screenshot of the schedule from IDA ICE.

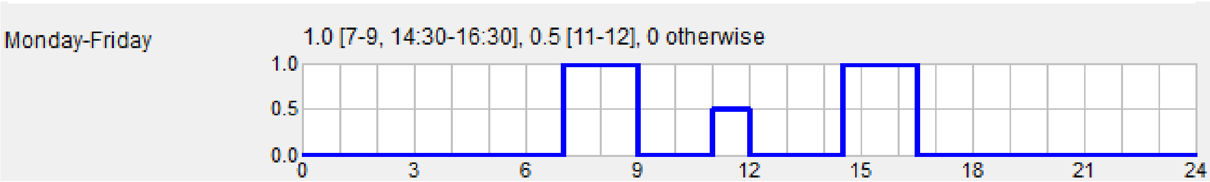

Figure 50 – Domestic hot water schedule Haukåsen kindergarten

5.4 PLANT

The plant is equal to the heat supply system sketch in figure 31. As explained in section 4.3, a heat pump stands for the entire base heating and the top heating is covered by an electrical boiler. The solar collector is used for supplementary heating of tap water. This differs somewhat from the real plant that has *sequential control* between the heat pump and the solar collector (c.f. section 4.3.2). Unfortunately this was too complex to carry out in IDA ICE. The advantage is on the other hand that utilization of solar energy is possible to obtain all year around, making the most of this free energy source. The disadvantage is that the aim with the model was to make it as realistic as possible to the real building.

Inputs regarding the heat plant are made by replacing original plant with *ESBO Plant* and selecting *flat plate solar collector*, *brine to water heat pump* as base heating, *generic electric heater* as top heating and *ground source bore hole loop* as ground heat exchange. The generic hot water tank volume is 0,4 m^3, with a shape factor of 2,33 (height divided by diameter) and an insulation U-value of 0,3 W/m^{2}K. Complete plant sketch screenshot from IDA ICE can be found in appendix B.5. As mentioned in the introduction to chapter 5, one modification from the old model is reduction of the supply temperature to the radiators and heating coil. This is made by typing in 50 °C manually on the setpoint out from the hot water tank.

5.4.1 TOPUP HEATING

The electrical boiler has 45 kW capacity and an efficiency of 0,98.

5.4.2 BASE HEATING

All the input values are based on the heat pump datasheet, see appendix B.6. The heat pump utilizes 3 bore holes with depth 80 metres as heat source. Total heating capacity is 26,4 kW, with a COP of 4,06 with optimal working conditions. The compressor type is in reality a scroll compressor, but the correct type could not be found in the IDA ICE database. Therefore screw compressor is chosen. When it comes to the evaporation temperature and condensation temperature, the following interpretation of IDA ICE terms (in italics) is done:

- *Brine (cold) unit temperature difference*:
  Temperature difference between brine inlet (3 °C) and brine outlet (0 °C), which equals 3 °C.
- *Water (hot) unit temperature difference:*
  Temperature difference between hot water supply (43 °C) and return water (35 °C), which equals 8 °C.

5.4.3 SOLAR COLLECTOR

A new resource is made, named *Solar Collector SOLC 220* after the actual one mounted on Haukåsen kindergarten. Figure 51 shows solar collector input data, based on the data sheet in appendix B.6. Three solar collector units of total area 6,5 m² are implemented with 90° tilt, and directed 28° to the west (where south is 0°).

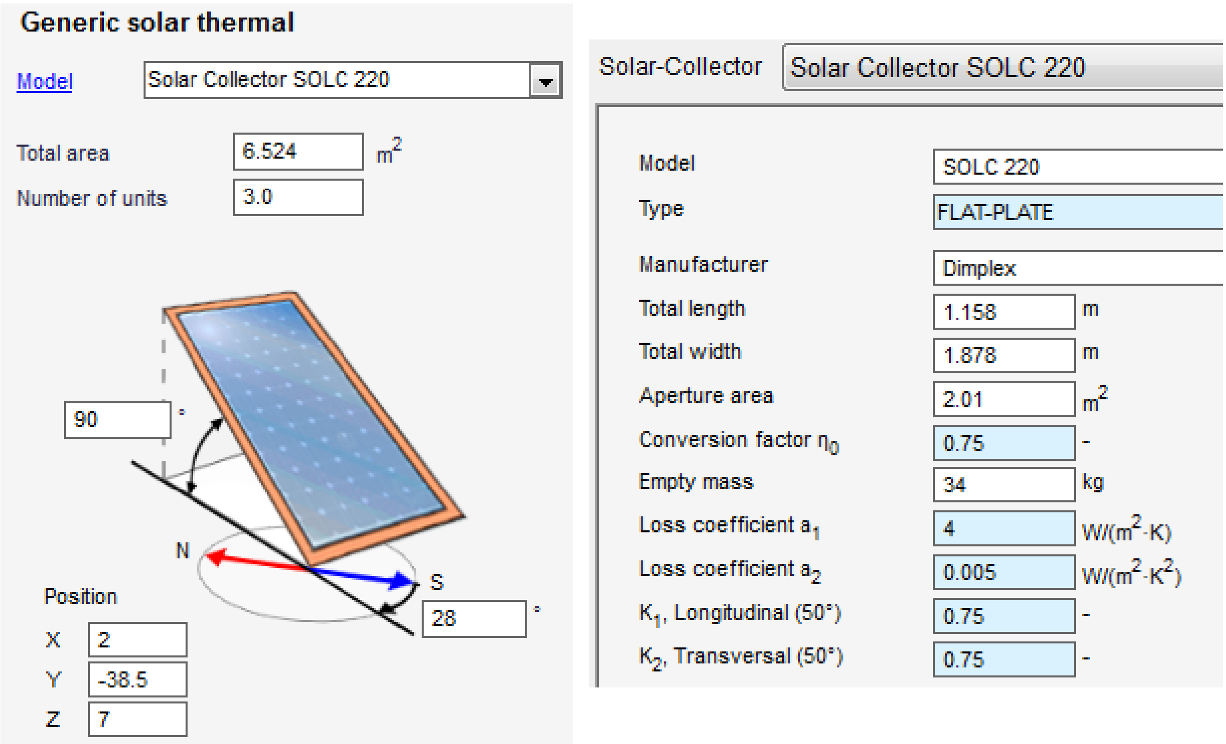

Figure 51 – Input data solar collector

5.4.4 HEAT DISTRIBUTION SYSTEM

The supply temperature for heat distribution to the heating coil and domestic hot water is maximum 50 °C. Figure 52 shows an IDA ICE screenshot of the heat distribution temperature level for space heating, according to the outdoor compensation curve (figure 36) [Schneider Electric, 2013]. The pump pressure head is set to 60 000 Pa [Ohnstad, 2013].

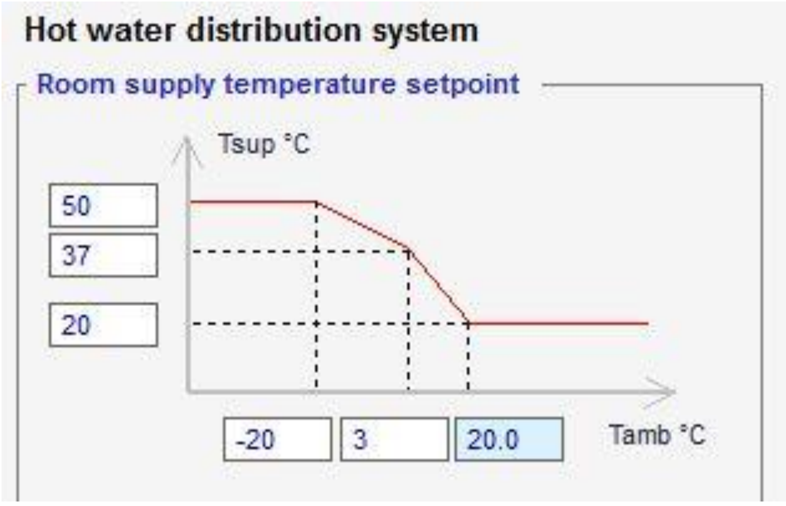

Figure 52 – Temperature level distribution of space heating

5.5 AIR HANDLING UNIT

To achieve the correct air flow rate based on the given zone control (see appendix B.2), it is necessary to start by defining the schedule precedence regarding ventilation. Figure 53 illustrates this sequence, which is discovered by logging the output signals of the components in multiple simulations.

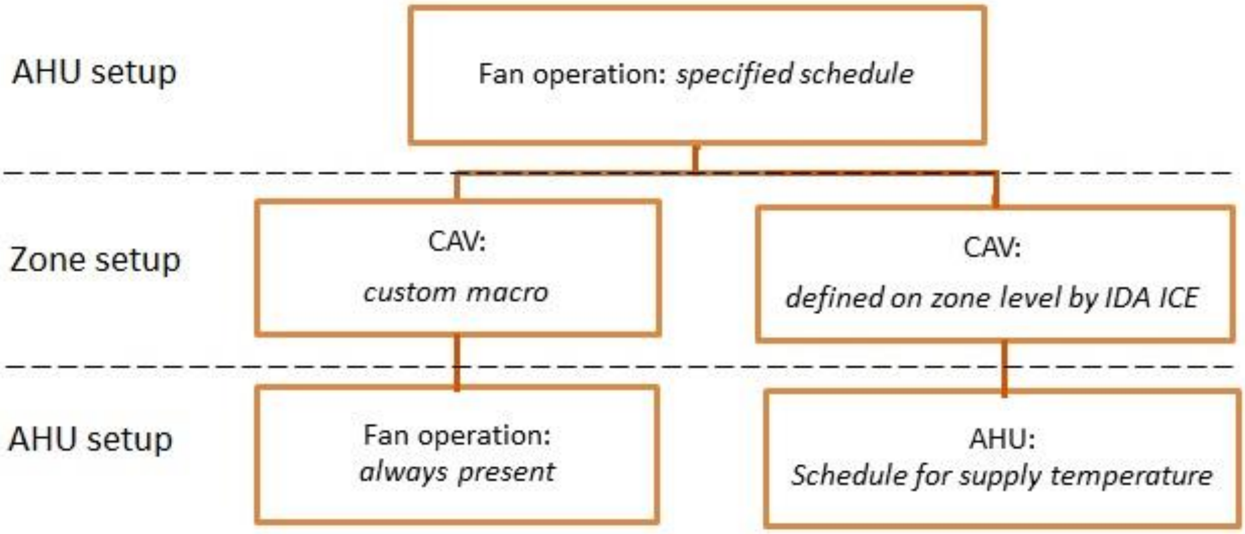

Figure 53 – Schedules hierarchy regarding ventilation in IDA ICE

In other words, defining a fan operation schedule in the AHU setup will oversteer all attempts on zone control implemented on zone level. In order to attain the desired zone control, the following input is implemented in the AHU setup:

- Fan input:
    - Operation is set to *always present*
    - Rated flow (available VAV, see appendix B.7): around 1250 l/s
- Heat recovery unit input (see datasheet in appendix B.6):
    - Effectiveness: 0,832
    - Capacity: 2,33 m^3/s

5.6 ZONES

All zone inputs can be found in the complete input data report in appendix B.1. This section presents assumptions, calculations and control strategies that form the basis of the input.

5.6.1 CONTROL STRATEGIES FOR VENTILATION

An overview of the planned zone ventilation control can be seen in table 10.

Table 10 – Summary of ventilation macros implemented in zones

CAV with variable minimum air flow rates	VAV with control based on occupancy and temperature level	DCV with control based on occupancy, temperature level and CO_2 concentration
Zone 1, 4 and 6	Zone 2 and 5	Zone 7 and 8

As explained in section 5.5, a precedence of schedules that affects the zone air flow rate is defined. The outcome on zone level can be summarized with the following points:

- In zones where constant air volume (CAV) is used, a custom ventilation macro has to be developed to ensure minimum air flow rates outside normal operating time. The built-in CAV function of IDA ICE does not provide reduction outside operating hours.
- The built-in VAV function does not take occupancy into account, therefore custom VAV and DCV macros must also be developed.

Three different ventilation control strategies are designed and implemented in the model based on the actual control strategies used in the building.

1) CAV custom macro

Figure 54 shows a screenshot of the custom CAV macro.

Figure 54 – CAV custom macro

The operating hours (07:00-16:30) are defined in the schedule block to the left in figure 54. The output signal from this block is 1 when the point in time is within operating hours and 0 otherwise. This implies that the AHU supplies the zone of interest with maximum air flow rate within operating time and minimum otherwise. The upper and lower air flow rate limits are defined specifically for each zone, in the zones *controller setpoints* (see section 5.6.3). This macro is implemented in zone 1, 4 and 6.

2) VAV custom macro with control based on occupancy and temperature level

The macro implemented in zone 2 and 5 can be seen in figure 55.

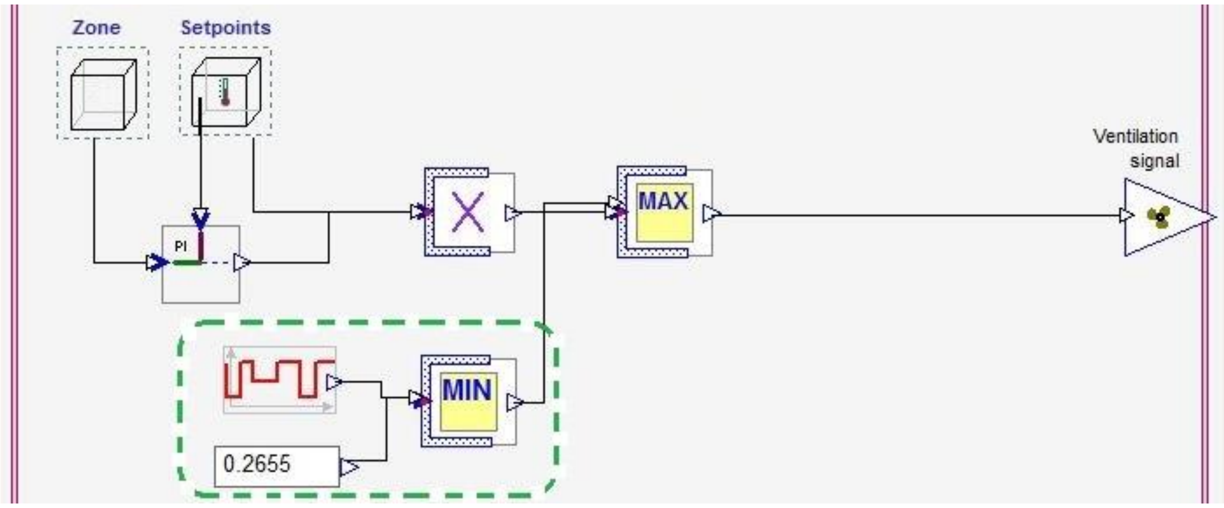

Figure 55 – VAV control based on occupancy and temperature level

The input signal to the PI-controller is the air temperature of the zone. This signal is then compared to the reference signal, which is defined by maximum allowed temperature in the zone controller setpoints. The output signal from the PI-controller is then a value from 0-1, which represents a percentage value between minimum and maximum air flow rate needed to achieve desirable temperature level. Mode of operation of the PI controller is set to *cooling mode*.

Multiplying the signal from the PI-controller with the *IN USE* signal from the zone ensures that occupancy has a higher precedence than the air temperature. When no people are in the zone, the output signal from the multiplier is 0, regardless of how high the zone air temperature is. With occupants present, the registered air temperature will determine the necessary air flow rate from the AHU. This *IN USE* signal from the zone is based on *occupant* room units implemented within the zone. Each occupant group has a defined schedule (given in section 5.6.4), where the given value has to be either 1 for presence or 0 for no presence.

As the required minimum air flow rate is lower outside operating hours than within (cf table 5); the minimum air flow rate limit defined in the controller setpoints needs to be set to the lowest of these two requirements to obtain the most energy efficient control strategy. This implies that when the final ventilation signal (the block to the right in figure 55) is 0, the AHU supplies minimum air flow to the zone.

To ensure sufficient minimum supply of air flow also during operating hours, the three blocks in the green dotted line circle is implemented. The schedule block defines the operating hours, its mode of operation is explained under figure 54. The constant block right beneath the schedule defines the necessary minimum air flow rate specifically for each zone during operating hours, given in percentage between minimum and maximum air flow rate (see table 12). These two signals are then compared, to detect the minimum value. There are two possible outcomes; either 0 when outside operating hours or the constant (in this case 0,2655) within operating hours. At last, the signal from the green dotted line circle is compared to the signal from the multiplier. The *maximum value* block detects the higher of the two input signals, to ensure sufficient ventilation at all times.

To sum up the mode of operation, the precedence and four different scenarios are given.

Precedence

1) Operating hours
2) Occupancy
3) Zone air temperature

Case 1: The timing is outside operating hours and no occupants are present:
The *IN_USE* signal from the zone equals 0 when no occupants are present. The schedule in the green dotted line circle will also have 0 as output signal, because the timing is outside operating hours. This implies that both the multipliers outcome and the *minimum value* blocks outcome is 0. The final ventilation signal will be 0, which corresponds with the minimum air flow rate defined in the zone controller setpoints.

Case 2: The timing is outside operating hours, but occupants are present:
The schedule will oversteer the fact that there are occupants present, and the final ventilation signal equals 0, which is the minimum air flow rate according to table 6.

Case 3: The timing is within operating hours, but there are no occupants present:
The *IN_USE* signal from the zone equals 0 when no occupants are present, which implies 0 as outcome from the multiplier. Output signal from the green dotted line circle will be 0,2655, which will oversteer the multiplier output signal when the two signals are compared in the *maximum value* block. The final ventilation signal will in this case be 0,2655, which means that the AHU will supply the zone with an air flow that is 26,55 % between minimum and maximum.

Case 4: The timing is within operating hours with occupants present:
Now the zone air temperature will be the deciding factor when it is multiplied with the *IN_USE* signal (which equals 1). The schedule blocks outcome will equal 1, but is oversteered by the *minimum value* block. The final ventilation signal will be defined by the PI controller. In any case the *maximum value* block ensures minimum required air flow rate within operating hours.

3) DCV custom macro with control based on occupancy, temperature level and CO_2 concentration
The only difference from macro number 2 is that in addition to temperature control, this macro (figure 56) is also controlled by CO_2 concentration in the zone.

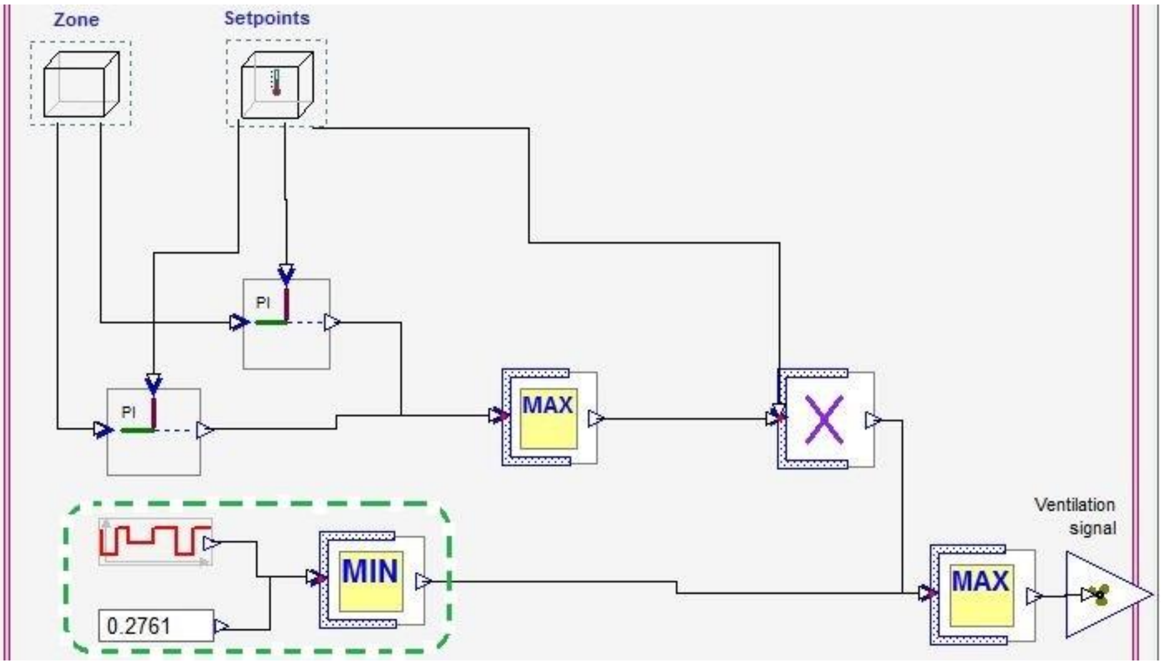

Figure 56 – VAV control based on occupancy, temperature level and CO_2 level

Operating hours still has the highest precedence, then occupancy next, while the CO_2 concentration and temperature has equal precedence. This is carried out by adding another PI-controller, which receives input signal of the CO_2 concentration in the zone and a reference signal containing maximum allowed CO_2 concentration from the controller setpoints. The output signals from the two PI- controllers are sent to a *maximum value* control block to detect the maximum value of these to input signals. Then sufficient air flow rate is provided independent of which control criteria that has the highest output signal from the PI- controller. A multiplier is brought in to ensure that the occupancy is taken into account (cf. CAV custom macro). Then the green dotted line circle is once again brought in to ensure minimum required air flow rate during operating hours. This macro is implemented in zone 7 and 8.

5.6.2 HEATING UNITS

The kindergarten has low-temperature water based radiators. First, *ideal heaters* were added to all zones that are to be heated. Then a winter simulation (from 1st of November to 31st of march) was carried out, to find the necessary heat capacity that should be implemented in the radiators in each zone. Table 11 shows the necessary installed heat capacities based on this winter simulation. Zone 3 and 6 are not to be heated [Ohnstad, 2013].

Table 11 – Necessary radiator heat capacity

Zone	**Heat capacity (W)**
1	2 200
2	1 150
4	2 250
5	700
7	1 650
8	500
TOTAL	**8 450**

After the winter simulation, the ideal heaters are replaced with water based radiators with the respective heating capacity as in table 11. Water supply temperature to the radiators at maximum power is 50 °C, and return temperature at maximum power is 35 °C. In the final model, there are no ideal coolers or heaters in any of the zones.

5.6.3 ZONE CONTROLLER SETPOINTS

The following parametres are kept unchanged because they correspond well with the kindergarten:

- Pressure difference envelope: -20 to -10 Pa
- Daylight at workplace: 100 to 10 000 Lux
- CO_2 concentration (lower limit means when the air flow rate is starting to increase to maximum value): 700 – 1 000 ppm (vol)
- Relative humidity: 20 – 80 %

Variable temperature setpoints

The maximum temperature limit defined in all zone controller setpoints is 25 °C. In order to achieve the same night set-back as in the detail engineering report, variable minimum temperature setpoints have been implemented in all zones except zone 3 and

6. Zone 3 is not to be heated, and zone 6 has a constant minimum temperature setpoint of 19 °C. The minimum temperature limit for the remaining zones is 22 °C from 06:00-16:30 and 19 °C otherwise. Have in mind that the operating hours are from 07:00-16:30. In order to reach the desired minimum air temperature of 22 °C in time for the occupants arriving at 07:00, the night set-back schedule stops at 06:00.

Air flow rates

The maximum air flow rates in table 12 are based on the air flow rate table given in appendix B.7.

Table 12 – Air flow rates in zones

Zone	OUTSIDE OPERATING HOURS		WITHIN OPERATING HOURS			
	Minimum air flow rate		**Minimum air flow rate**		**Maximum air flow rate**	
	(m^3/hm^2)	**(m^3/h)**	**(m^3/hm^2)**	**(m^3/h)**	**(m^3/hm^2)**	**(m^3/h)**
1	2	347,2	6	1 041,6	8,93	1 550
2	2	240,2	6	720,6	16,92	2 030
4	2	348	6	1044	7,02	1 220
5	2	117,2	6	351,6	17,06	1 000
6	2	364,4	6	1 093,2	2,48	450
7	2	139,6	6	418,8	16,49	1 150
8	2	121,2	6	363,6	16,49	1 000

The sum of all the maximum air flow rates equals the maximum capacity of the AHU; 8400 m^3/h. The minimum air flow rates outside working hours are based on NS3031, while the minimum air flow rates within operating hours are based on values presented in table 5. Zone 3 (the outdoor workshop) has no ventilation.

Zones with VAV and DCV macros contain a constant that defines the percentage value of necessary air flow (see description of macro number 2). These constant are defined individually for each zone according to the following formula:

$$Minimum\ air\ flow\ rate\ percentage\ = \frac{Q_{min,w} - Q_{min,o}}{Q_{max,w} - Q_{min,o}} * 100\ \% \qquad (11)$$

Where $Q_{min,w}$ represents the minimum required air flow within operating hours, $Q_{min,o}$ is the minimum required air flow outside operating hours and $Q_{max,w}$ equals the maximum air flow capacity of the zone. All the air flows are given in table 12. The following table shows the resulting air flow rate constant for each zone having VAV ventilation macro.

Table 13 – Minimum air flow rate percentage constant used in VAV macros

Zone	Minimum air flow rate percentage (%)
2	26,81
5	26,55
7	27,61
8	27,61

5.6.4 INTERNAL HEAT GAINS

The internal heat gains are implemented on zone level on the basis of values given in table 5. The exception is internal gains from occupants, which have been defined as realistic as possible. Seeing that zone 3 is only an outdoor workshop room with no heating and cooling, no internal gains are added to this zone. Same applies for zone 6, which is the technical room of the building that houses the AHU and plant.

Occupants

Table 14 summarizes all input regarding occupants.

Table 14 – Occupants input overview

Zone	Adults			Children		
	Number	Schedule	Activity level	Number	Schedule	Activity level
1	4	Young children	1,8	14	Young children	1
2	4	Young children	1,8	14	Young children	1
3	0	0	0	0	0	0
4	4	Older children (3 adults) Leader Office (1 adult)	1,8	12	Older children	1,8
5	1	Older children	1,8	6	Older children	1,8
6	0	0	0	0	0	0
7	1	Older children	1,8	4	Older children	1,8
8	1	Older children	1,8	6	Older children	1,8

Number of adults and children, along with schedules are information that is retrieved from the leader at the kindergarten [Rystad, 2014]. Activity levels are estimated on the basis of values given by Novakovic et al. [2007, p. 107]. The values for clothing proposed by IDA ICE are kept as they are; 0,85 ± 0,25 clo for each occupant. These values correspond with what is suggested by Novakovic et al. and covers both winter clothing and summer clothing.

The schedules are as follows:

- *Young children*: present from 07:00-09:00, 11:00-12:00 and 14:30-16:30
- *Older children*: present from 07:00-09:00 and 14:30-16:30
- *Leader office*: present from 11:00-13:00

Lighting and equipment

IDA ICE defines the number of lighting and equipment units automatically when these are added to a zone. The rated power input is defined based on the expected internal heat gains from table 5 along with the number of units defined by the program. The whole input data report can be found in appendix B.1.

5.7 MODEL WITH UNDERGROUND THERMAL ENERGY STORAGE

In addition to the reference model, a model with the possibility of storing excess heat from the solar collector is made. All input data are the same as in the reference model, only the plant sketch is changed. Figure 57 on the next page shows how the plant sketch looks in the UTES model.

All components that are added can be found in the palette in IDA ICE. A heat exchanger is placed between the brine circuit and solar fluid circuit. The brine in the return pipeline is led via the heat exchanger, heated by the solar fluid and then this heat is transferred to the ground. As for the solar circuit, a pipe split on the supply pipeline directs the solar fluid either to the heat exchanger in the hot water tank or to the ground heat exchanger. The circulation pump placed on the return pipeline to the solar collector (placed right above the ground heat exchanger on figure 57) is controlled by the PI-controller to the left of the pump. Other than that, the control strategy for directing solar heat to the hot water tank is exactly equal to the reference model.

The PI-controller receives the outgoing temperature from the solar collector as its input signal, and compares it to the constant, here defined 62. This means that when the outgoing temperature from the solar collector exceeds 62 °C, the outsignal from the PI-controller equals 1. Consequently the circulation pump is set into operation, and heat is transferred from the solar fluid to the brine via the ground heat exchanger. The reason that the limit is set to 62 °C, is that the temperature of the tap water is 60 °C. Seeing as it it only the *excess heat* that is being stored in the ground, the limit has to be higher than the tap water temperature.

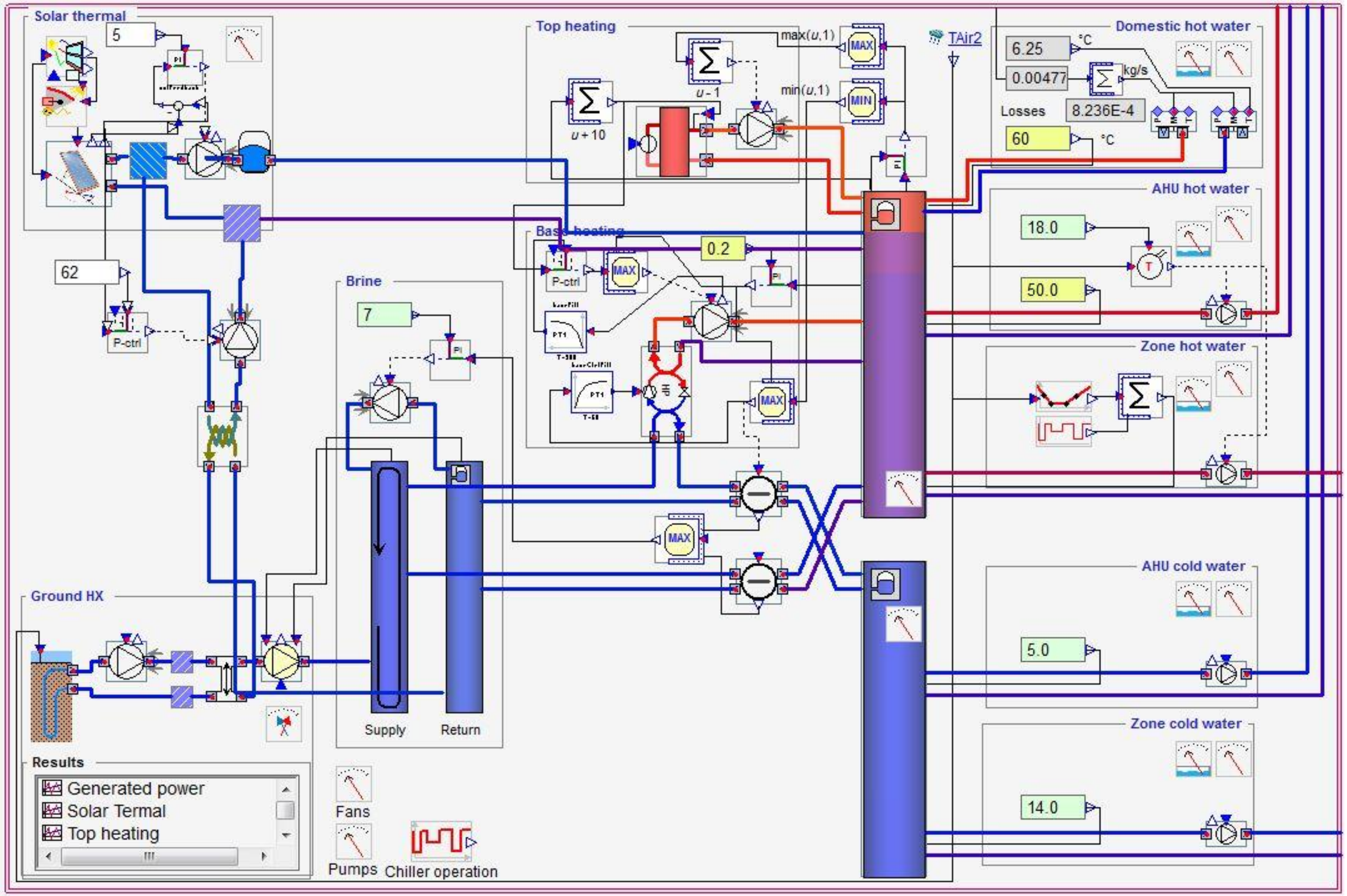

Figure 57 – Plant sketch - model with UTES (made by the author with help from J. Eriksson [May 2014])

6. METHODOLOGY FOR ECONOMIC ANALYSIS

Section 6.1 gives an explanation of all parametres involved and the relevant formulas needed for the economic analysis. All necessary collected data are given in section 6.2.

6.1 METHODOLOGY

The economic methodology is summed up in a step by step presentation in figure 58

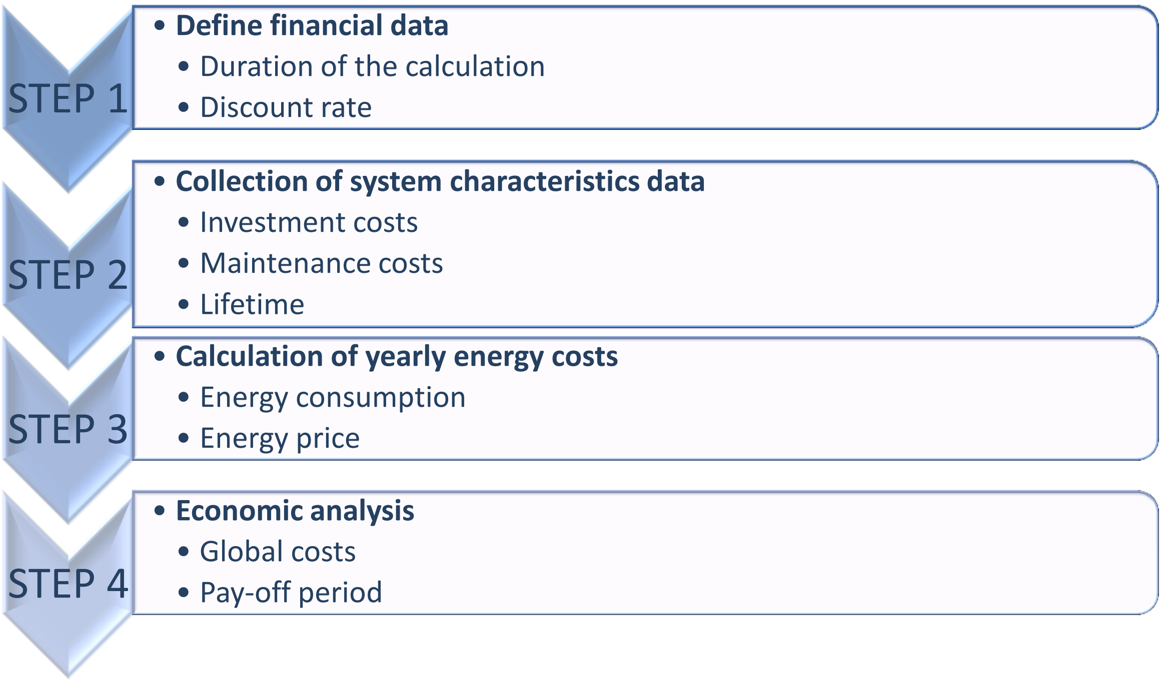

Figure 58 – Economic methodology step by step

The first step is to determine financial data, such as the calculation period and the discount rate. The discount rate represents the nominal interest rate corrected for inflation, relative changes in energy price and taxes. For projects initiated by a municipality like Trondheim, the discount rate is defined by the Norwegian Finance Ministry [Novakovic et al., 2007]. The discount rate, r, and the calculation period, n, affect the present value factor.

Present value factor

$$PV = \frac{1-(1+r)^{-n}}{r}\ (year^{-1}) \tag{12}$$

The next step is to collect investment costs, maintenance costs and expected lifetime of the components in the heat plant. In this case, the system to be considered consists of solar collector, heat pump, energy wells and an electrical boiler. Investments costs should include design, purchase of components, connection to suppliers and installation. Maintenance costs are for restoring the desired quality of the installation and include inspection, cleaning, repair and preventive maintenance. The lifetime is defined as the expected lifespan of the component specified in years [Standard Norge, 2007].

In the third step the annual energy costs are calculated, based on yearly energy consumption and energy prices. More information about energy prices will follow in section 6.2. As the kindergarten has not been in operation for a whole year yet (set into operation august 2013), the analysis is based on *expected* annual energy consumption according to simulations in IDA ICE.

At last, the heat supply system is evaluated economically by calculating the global costs and pay-off period.

6.1.1 GLOBAL COST

The global cost, C_G, equals the sum of the present value of all costs including the investment cost:

$$C_G = \sum_{i=1}^{c} (C_{I,i} + C_{O,i} + C_{m,i} + C_{rv,i}) + C_f \ (NOK) \tag{13}$$

Where: c = component, the costs for all components are summarized
C_I = investment cost (NOK)
C_O = operating cost (NOK)
C_m = maintenance cost (NOK)
C_{rv} = residual value at the end of calculation period (NOK)
C_f = fixed cost, for example fixed electricity costs (NOK)

All expenses have a negative sign and incomes have a positive sign. When comparing different heat supply systems, the one with the lowest global cost is economically preferable.

Operating cost

$$C_{O,c} = E * Pr * PV \ (NOK) \tag{14}$$

Where E (kWh/year) is the annual energy consumption, *Pr* (NOK/kWh) is the energy price and PV is the present value factor (formula 12).

Maintenance cost

$$C_{m,c} = M_p * PV \ (NOK) \tag{15}$$

As the maintenance cost is given as an annual cost to prevent breakdown, M_p (see section 6.2), it is multiplied with the present value factor to discount it to global cost.

Residual value

$$C_{rv,c} = \left(-C_{I,c} * \left(1 - \frac{n}{l}\right)\right) * (1 + r)^{-n}\ (NOK) \qquad (16)$$

Information about residual values of the components was difficult to obtain, therefore it is assumed to be the percentage investment value of remaining lifetime years. The last term in the formula is the marginal rate of return. The negative sign in front of the investment cost is due to the investment being a *cost*, while now the object is to find the residual value, hence an *income*.

6.1.2 PAY-OFF PERIOD

The pay-off period represents the time it takes before the present value of all future savings is equal to the investment cost of the measures [Novakovic et al., 2007, p. 86]. The new measure must be compared to another, initial project, in order to calculate the savings. In this case the optional measure used for calculations is to use electricity for heating of the building.

To calculate the pay-off period, n_0, the sum of the present values of all savings is set equal to the total investment cost:

$$S_{tot} * \frac{1-(1+r)^{-n_0}}{r} = C_{I,tot} \qquad (17)$$

$$\Rightarrow n_0 = -\frac{\ln\left(1 - \frac{C_{I,tot} * r}{S_{tot}}\right)}{\ln(1+r)} \qquad (18)$$

Where S_{tot} (NOK/year) is the total annual savings in operational cost compared to the electricity heat supply system, $C_{I,tot}$ is the total investment cost of the measure and r is the discount rate. For this calculation, both incomes and outcomes are marked with a positive sign, as the formula takes this into consideration.

6.2 COLLECTION OF DATA

This section presents all collected data that form the basis of the calculations.

6.2.1 FINANCIAL DATA

The discount rate is defined as 4 % [Finansdepartementet, 2012], while the calculation period is to be chosen according to what time perspective found interesting to look at. To begin with, the calculation period is set to 20 years, equal to the lowest lifetime of all the components in the plant (see section 6.2.2).

6.2.2 HEAT PLANT DATA

Table 15 summarizes the heat plant data. The investment costs and lifetime of the components are retrieved from the supplier; the exact sources of information can be seen after the table. The maintenance costs are found in table A.1 in NS-EN 15459 [Standard Norge, 2007], given as *annual preventive maintenance cost* in percentage of initial investment. Hence any repair cost is considered included in these annual preventive maintenance costs.

Table 15 – Plant component costs

Component	**Investment cost (NOK)**	**Lifetime (years)**	**Preventive maintenance cost[3] (NOK/year)**
EXISTING HEAT SUPPLY			
Solar collector	20 200[1]	30[1]	0,5 % of $C_{I,c}$ = 101
Accumulator tank (connected to solar circuit)	19 100[1]	30[1]	1 % of $C_{I,c}$ = 191
Heat pump	50 800[2]	20[3]	2 % of $C_{I,c}$ = 1016
Energy wells (drilling job)	192 000[4]	30[5]	1 % of $C_{I,c}$ = 1920
Electrical boiler	39 760[6]	20[3]	1 % of $C_{I,c}$ = 397,6
ALTERNATIVE HEAT SUPPLY: DISTRICT HEATING			
Substation	40 000[7]	30[7]	1 % of $C_{I,c}$ = 400

Sources of information:

1) [Petersen, 2014]
2) [Olsen, 2014]
3) [Standard Norge, 2007]
4) [Grandetrø, 2014]
5) [Norges Geologiske Undersøkelse, 2008]
6) [Nyhus, 2014]
7) [Dønnestad, 2014]

The installation investment cost was not possible to obtain, due to that the HVAC contractor has been declared bankrupt. With this in mind, the installation costs for the different heat supply alternatives will have to be assumed equal in order to compare the different plants. The components that are common for all heat supply systems, such as radiators, AHU, pipes etc. are not included in table 15, due to the fact that these are common for all heat supply systems.

District heating is chosen as alternative heat supply, in order to compare the chosen heat plant with other alternatives. Have in mind that the location of the kindergarten might not be suited for district heating, but the analysis is still included as it might be valuable for other projects in the municipality of Trondheim.

To verify that the information retrieved from the suppliers is reliable, the investment costs are compared to the prices given in the Norwegian Price Book *[Norconsult Informasjonssystemer AS, 2010]*. These prices are given in table 16.

Table 16 – Component prices according to Norwegian Price Book

Component	**Price (NOK)**
Solar collector (vacuum tube collector)	16 225
Solar heat tank 500 litres	7 048
• Heat pump with 30 kW capacity at 50 °C • 2 bore holes • Pipelines • Accumulator tank	269 461
Electrical boiler	N/A
Heat exchanger (in substation)	40 009

The price for electrical boiler could not be found. When comparing the prices in table 16 with the costs retrieved from the suppliers, the substation price and heat pump price is mostly in accordance. However, the price for solar collector with accumulator tank from table 16 is 41 % less than what the supplier indicated. The reason may be that the cost given by the supplier includes more than what is indicated in the Norwegian Price Book, i.e. the solar fluid, measuring instruments, circulation pump etc

6.2.3 ENERGY PRICES

The electricity prices are based on the mean electricity price for the last ten years in Trondheim [Nord Pool Spot, 2013]. The net price depends on the geographical location, and is given as a fixed price per year plus a price dependent on the electricity consumption [TrønderEnergi Nett AS, 2014]. The district heating price is based on the mean price from year 2006 – 2012 calculated by Statistics Norway, received from supervisor N. Nord [2014]. Table 17 gives an overview of the prices.

Table 17 – Energy prices

Type	**Price**
Variable electricity price	0,3218 NOK/kWh
Fixed net price (electricity)	2140 NOK/year
Variable net price (electricity)	0,21 NOK/kWh
District heating price	0,5831 NOK/kWh

This gives a total variable electricity price of 0,53 NOK/kWh. The spreadsheet used in the economic analysis can be found in appendix C.2, and the results of the economic analysis are given in section 7.5.

7. RESULTS AND ANALYSIS

This chapter presents the results of the IDA ICE simulations and the economic evaluation.

First, the reference model is analysed, looking at the net annual energy need of the building, heating loads and the performance of the heat plant. The indoor environment is also looked into, and a thorough review of the ventilation control strategy is carried out.

Secondly, different parametres are changed in the system design and operation, to see the impact on net annual energy need, heat plant performance and utilization of free energy. Another purpose with the changes is also to test IDA ICE's capability as a simulation tool. The parametres looked into in this thesis are area and angle of the solar collector, the shape factor of the hot water tank, ventilation control, night set-back and impact of internal heat gains.

Third, the model with UTES is analysed. Utilization of solar heat, impact on delivered energy and ground temperature are among the subjects looked into in this section. Last, an economic evaluation and sensitivity analysis is carried out, based on the expected net annual energy use retrieved from the reference model.

7.1 REFERENCE MODEL

One of the main purposes with the master thesis was to establish a reference model of Haukåsen kindergarten. This section presents the IDA ICE results on this reference model.

7.1.1 NET ENERGY NEED

A one-year simulation (year 2013) has been performed to look at the expected net energy need of the kindergarten. The result is then compared to SIMIEN simulations, passive house criteria and measuring data. The impact of changes being done in IDA ICE is discussed and the utilized free energy is presented.

In IDA ICE the heating need can be found in the tab *Systems energy*, while the specific electricity needs can be found in D*elivered energy*. The resulting net energy need can be seen in table 18, along with the net energy need retrieved from SIMIEN simulations [Wormdal and Smits, 2013].

Table 18 – Calculated energy need at Haukåsen kindergarten year 2013

	SIMIEN simulations [Wormdal and Smits, 2013]		Result from IDA ICE simulation	
	Net energy need (kWh)	**Net specific energy need (kWh/m²)**	**Net energy need (kWh)**	**Net specific energy need (kWh/m²)**
Space heating	10 963	13,5	10 223	12,6
Heating coil	6 541	8	5 733	7,1
Domestic hot water	8 144	10	10 873	13,4
Fans	10 083	12,4	5 392	6,6
Pumps	412	0,5	65	0,1
Lighting	12 727	15,7	10 956	13,5
Technical equipment	4 243	5,2	3 460	4,3
TOTAL	**53 113**	**65,3**	**46 702**	**57,4**

SIMIEN versus IDA ICE

Table 18 shows quite similar energy budgets in SIMIEN and IDA ICE. The largest difference is the electricity need for pumps and fans. One explanation for this is that IDA ICE is a more advanced simulation program containing more input data than SIMIEN, hence more accurate results can be expected. This appears especially for the fan and pump operation. SIMIEN does not perform dimensioning of the components, only simple calculations of energy needs based on the input data from the Norwegian building code.

Aside from that, the low energy need for fans in the IDA ICE model can be explained by the implemented ventilation macros based on occupancy. Due to low occupancy in the building because of the kindergarten being an outdoors kinder-garten, the air flow rate is kept at a minimum when the building is unoccupied. This gives an energy efficient ventilation system.

The fact that calculations in SIMIEN are based on the Norwegian building code is revealed also in the energy need for heating of domestic hot water. The specific energy need is 10 kWh/m^2 according to SIMIEN, directly as given in NS3031 [Standard Norge, 2011]. IDA ICE on the other hand, takes the hot water schedule (see section 5.3.6) into consideration. This leads to a more accurate calculation, as long as the implemented schedule is close to the reality.

Comparison to passive house criteria

The values in table 18 correspond well with the passive house energy need indicated in figure 1 (section 2.1), which has a total specific energy need of 65 kWh/m^2 upper limit. The net annual energy need from table 16 is also presented as a bar chart (see figure 59), equal diagram as in figure 1. With this, comparison between the two figures is easier to perform.

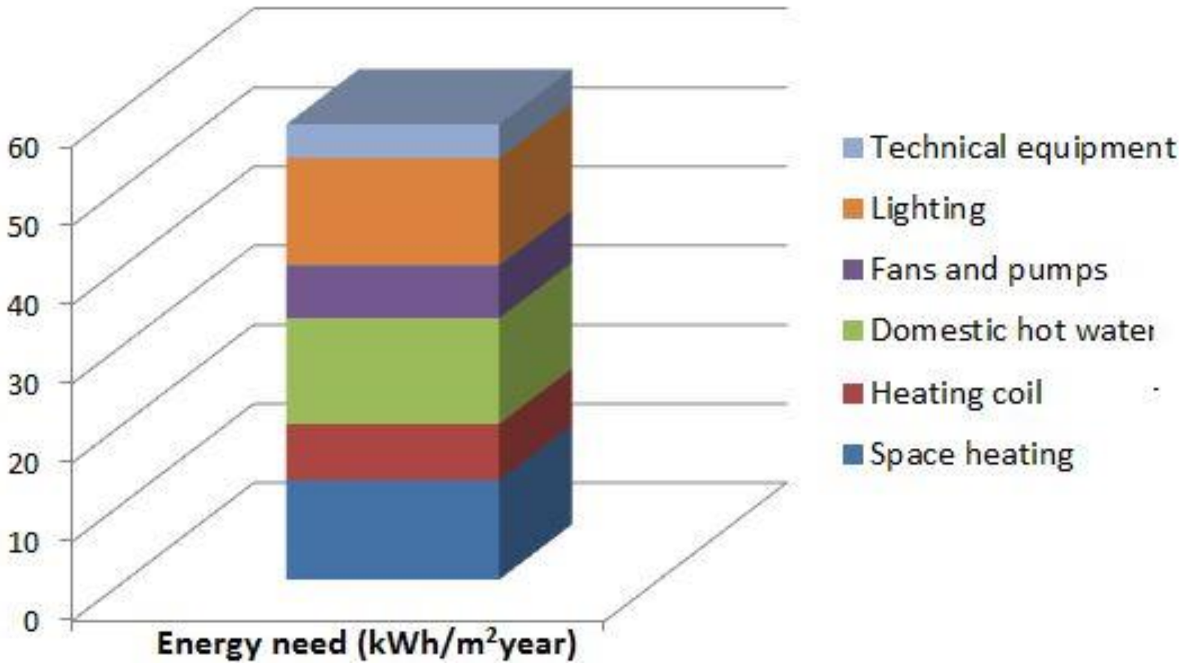

Figure 59 – Net annual energy need according to IDA ICE model

Even though figure 1 is based on office buildings, the operational hours are roughly the same as in a kindergarten. The office building has a larger energy need for technical equipment, while the kindergarten has a higher need for heating of domestic hot water. The result does however not fulfil the passive house demand of maximum 25 kWh/m^2 annual specific heating need, as the heating need in the IDA ICE model is estimated to 33,1 kWh/m^2 according to table 18.

When referring to the passive house research in Denmark (section 2.1.1), 12,6 kWh/m^{2}year specific energy need for space heating is lower than their best case passive house. The total specific energy need in Haukåsen kindergarten is also lower than the measured value of the best passive house. What remains to see is what the *actual* energy use is, seeing that most of the passive houses in Denmark exceeded their estimated energy need. The actual energy need will be discussed later on in this section.

Impact of improved IDA ICE model

The most important changes implemented in the new IDA ICE model compared to the model in the project thesis are the following (cited from chapter 5):

- **Correct climate file**
  Östersund was used as climate file in the old model, as *Trondheim* was not available due to an older version of IDA ICE.
- **Night set-back of heating setpoint temperature**
  22 °C heating setpoint during operating hours and 19 °C otherwise.
- **Correct supply temperature to heating coil and radiators**
  Previously this was automatically set to 60 °C by IDA ICE. This is now corrected to 50 °C.

- **More exact occupancy hours**
  The old model contained occupants present during the entire operating hours from 7:00-16:30, while the new model is based on the actual occupancy (see section 5.6.4)
- **Ventilation macros ensuring minimum air flow rates during operating hours**
- **Reduced lightning power input**

The net specific energy need from the old model (used in the project thesis) and new model is given in table 19.

Table 19 – Comparison of net specific energy need in old and new IDA ICE model

	Net specific energy need (kWh/m²)	
	Old model	**New model**
Space heating	13,4	12,6
Heating coil	30,6	7,1
Domestic hot water	9,0	13,4
Fans	4,3	6,6
Pumps	0,1	0,1
Lighting	17,3	13,5
Technical equipment	5,0	4,3
TOTAL	**79,7**	**57,4**

Looking at table 19, the net specific energy need is reduced by 28 % in the new model. The largest difference is the energy need for the heating coil. The first thing to remember is that *Östersund* was used as climate file in the old model, and Trondheim's mean annual temperature is 1,5 °C higher than Östersund's [Forbrukerrådet, 2013]. This gives a higher inlet temperature to the AHU and consequently a lower energy need for the heating coil.

Even though the ventilation macros lead to a slightly higher energy need for fan operation in the new model, it is insignificant compared to the large energy need reduction for the heating coil. Implementation of correct lightning power input made the estimated energy need for lightning drop with 22 %. The night set-back involves a lower need for space heating in the new model.

Delivered energy and comparison to measuring data

The energy delivered for heating purposes, that being electricity to the heat pump compressor and electrical boiler, is found in respectively the *Base heating* tab and the T*op heating* tab. The result from the one-year simulation is given in table 20.

Table 20 – Delivered electricity for heating purposes

	Delivered energy (kWh)
Heat pump compressor	8 749
Electrical boiler	3 408
TOTAL	**12 157**

According to table 20, the electricity supplied to the compressor and electrical boiler covers 45 % of the annual net heating need, which means that the rest will have to be covered by solar heat and ground heat. When the annual energy delivered for fan and pump operation, lightning and equipment are added (see table 18), the total delivered energy is estimated to be 32 030 kWh per year.

Based on the measuring from 2013-08-26 until 2014-05-19, the kindergartens annual delivered energy is expected to be 42 292 kWh (c.f. section 4.6). If this is the case, the actual delivered energy will exceed IDA ICE's estimate with 32 %. One should however have in mind that the summer months from mid-May to mid-August is not included in the measuring data, as the kindergarten was set into operation last August.

Thus the energy use in the months June till August is subtracted from the IDA ICE results, and the mean energy use for the rest of the year is used as basis (calculations can be found in appendix C.1) to have equal standard of comparison. This gives an expected annual energy use of 34 649 kWh, proving that the annual expected energy use based on measuring data is still 22 % higher than the estimate from IDA ICE under the same circumstances. This result corresponds with the research in Denmark (section 2.1.1), showing that most of the passive houses exceeded their estimated energy use.

Utilized free energy

Table 21 presents the utilized free energy, hence the heat recovered in the AHU and the collected heat from the solar collector and ground, found in the tab Systems energy in IDA ICE.

Table 21 – Utilized free energy reference model year 2013

	Utilized free energy (kWh)
AHU heat recovery	91 898
Solar heat	1 608
Ground heat	13 024
TOTAL	**106 530**

The utilized free energy from solar heat and ground heat covers 55 % of the annual net heating need, which corresponds with the 45 % coverage by the heat pump compressor and electrical boiler (c.f. table 20). To verify the numbers in table 21, the tabs *Solar termal* and *Ground heat exchange* are also checked. The data in these tabs are in indeed in accordance with the results given in table 21.

7.1.2 HEATING LOADS

Figure 60 shows the heating effect duration curve at the kindergarten, made on the basis of values from the *Total heating and cooling* tab in IDA ICE.

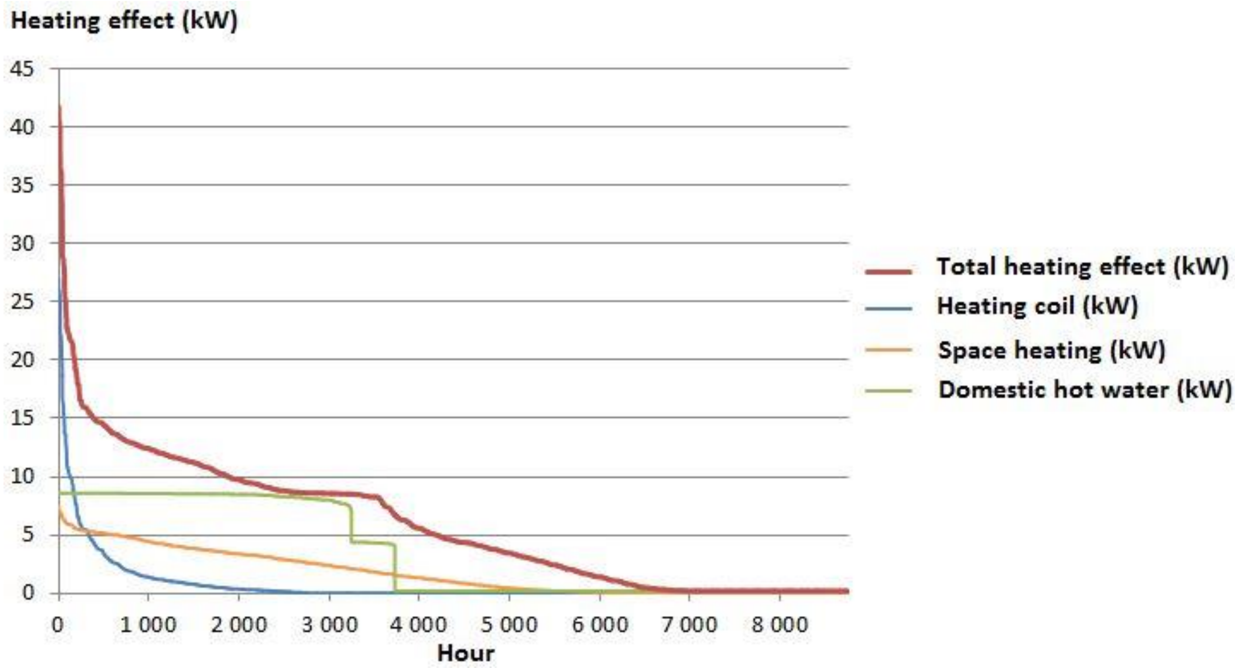

Figure 60 – Heating effect duration curve (year 2013)

The total heating effect peak load is 41,7 kW, while the peak loads for the other three curves can be seen in table 22. The distribution of heating effect coverage between the heat pump and electrical boiler is given in section 7.1.3. As the peak loads for the heating coil, radiators and hot water does not necessarily happen simultaneously, the sum of the three will not form the total heating effect curve. The block shape of the hot water curve is a result of the hot water schedule defined in IDA ICE (c.f. section 5.3.6).

Estimated heating loads versus actual installed capacity

Table 22 shows the actual installed capacity at Haukåsen kindergarten (see appendix A.2) versus the resulting maximum effect load estimated by IDA ICE.

Table 22 – Installed heating capacity versus needed capacity

	Installed capacity (kW)	Maximum heating load according to IDA ICE model (kW)
Radiators	14	7,5
Heating coil	18	27,0
Domestic hot water	5	8,6

The total installed heating capacity at Haukåsen kindergarten is 37 kW, while the simulations in IDA ICE (figure 60) shows a maximum heating effect of nearly 42 kW.

Judging by IDA ICE results, the installed radiators at Haukåsen are oversized. Remember that the necessary installed capacity were found by first implementing ideal heaters (cf. section 5.6.2), which resulted in totally 8,45 kW installed zone heating effect. Table 22 shows that with the water radiators, the maximum heating effect were only 54 % of the actual installed capacity.

Table 22 also indicates large differences in the heating coil capacity, as the result from IDA ICE indicates 50 % higher maximum heating effect than installed capacity. In order to examine this difference more closely, the heating coil effect load is plotted against the ambient air temperature, see figure 61.

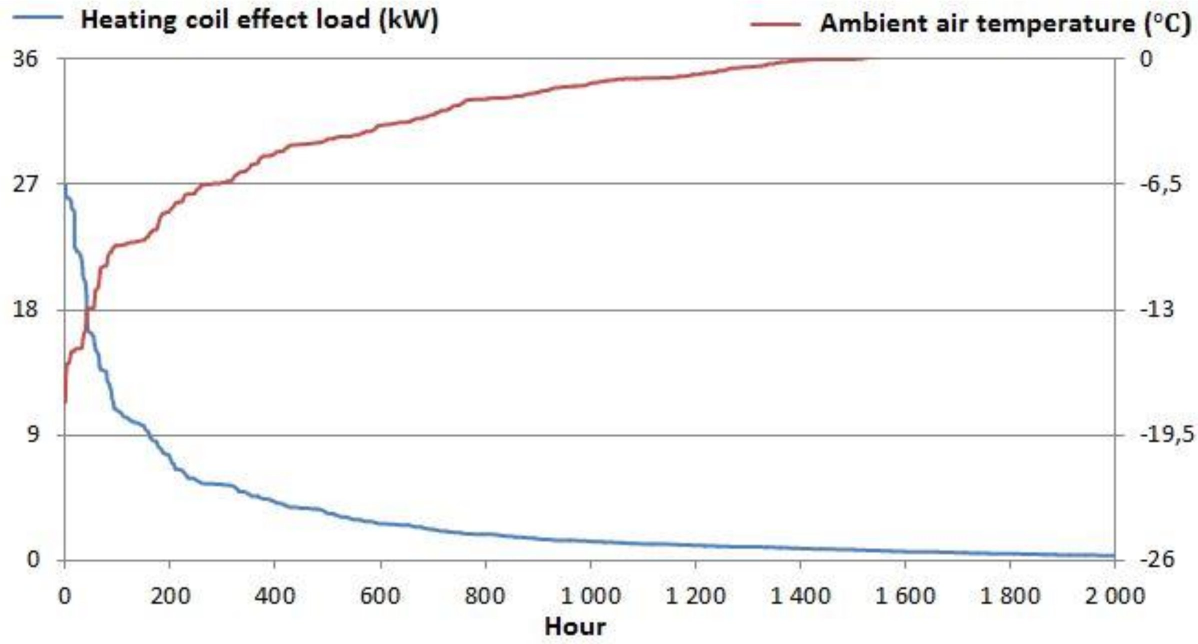

Figure 61 – Heating coil effect load versus ambient air temperature

The y-axes in figure 61 have been adjusted to make the crossing point between the two plots easier to detect. When the outdoor air temperature is -13 °C or lower, the heating coil effect load is higher than the installed capacity of 18 kW. This is the case for about 40 hours a year during January and February. As Trondheim has a dimensioning outdoor temperature of -20 °C, one would expect a lower crossing point.

7.1.3 HEAT PLANT ANALYSIS

A one-year simulation of year 2013 in IDA ICE is used as a basis for analysing the coverage of the heating need, and the performance of the heat pump, solar collector and electrical boiler.

Coverage of heating need

Figure 62 presents the distribution between the heat pump and electrical boiler for coverage of the heating effect during one year.

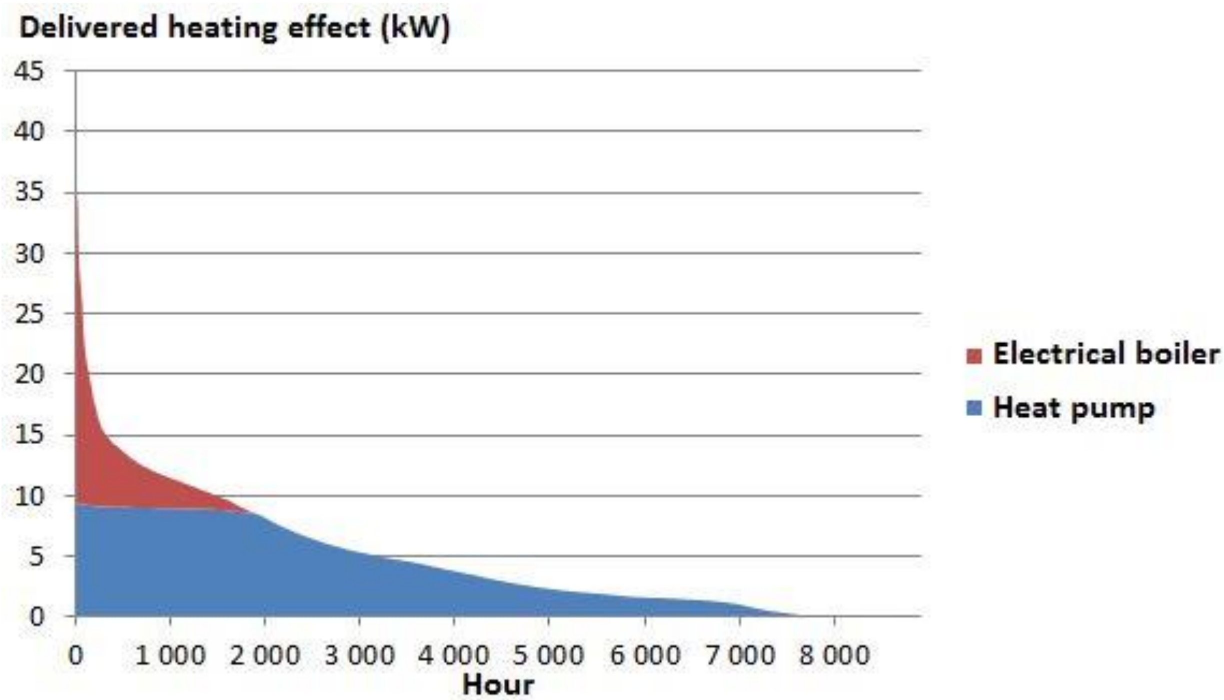

Figure 62 – Delivered heating effect during year 2013

As figure 62 shows, the heat pump clearly covers the base load, while the peak load is covered by the boiler. The solar collector contributes to the cover the base load to some extent. More exact values regarding the net annual heating need can be seen in table 23, including the solar collector contribution. The values are found in *Top heating, Base heating – Condenser power* and *Solar Termal* in the results tab in IDA ICE.

Table 23 – Expected coverage of heating need at Haukåsen kindergarten

	Electrical boiler	**Heat pump**	**Solar collector**
Energy output	3 408 kWh	22 163 kWh	1 608 kWh
Coverage of heating need	12,5 %	81,5 %	6,0 %
Maximum effect output	33,1 kW	9,9 kW	3,1 kW

Table 23 indicates that the total delivered energy for heating purposes is 27 129 kWh per year, while table 18 indicates that the net annual heating need is 26 829 kWh. The larger amount of delivered energy is due to losses in the system, such as heat losses through pipes in the hot water circuit.

The planned percentage distribution according to Rambølls detail engineering report (section 4.3.3, table 4) was to let the boiler cover 10 %, the heat pump 80 % and the solar collector 10 % of the net annual heating need. Comparing this to table 23, it is mostly in accordance, but the difference is largest for the solar collector.

One explanation for the low solar collector contribution can be that the collector is not mounted at an optimum angle. Another reason is that the efficiency of the collector depends on the temperature difference between the collector and the outdoor air temperature, which implies that the efficiency drops rapidly when the air is very cold (c.f. figure 14 section 2.2.2). If the collector was mounted at an optimum angle, one would expect a heating capacity of around 1 000 kWh/m^2 according to figure 6. The area of the collector is 6.524 m^2. With a mean annual outdoor temperature of 4,1 °C, the average efficiency is expected to be around 65 %, which implies a yearly expected contribution of 4 241 kWh. The impact of the solar collector angle and area are further analysed in section 7.2.

In the old model from the project thesis, the percentage coverage of net annual heating need was 28 % for the electrical boiler, 70 % for the heat pump and 2,4 % for the solar collector. It may seem as though the solar collector has a large increase judging by the results from the new model (table 23), but the *delivered energy* differs with only 6 %. The reason for the high coverage in the new model is that the total annual heating need has decreased with 57 % (c.f. *impact of improved IDA ICE model* in section 7.1.1).

Heat pump

As explained in section 2.3.2 regarding sizing of the heat pump, it is usually dimensioned to cover 50-70 % of the maximum heating load. This will often imply 85-95 % coverage of the net yearly heating need. The maximum installed capacity of the heat pump is 26 kW, so at first sight the installed capacity seems reasonable, seeing as the peak load is 42 kW.

Despite its maximum capacity of 26 kW, the maximum effect output from the heat pump is only 9,9 kW (c.f. table 23). The first rule of heat pump control (c.f. section 2.5.3) is to always utilize the heat pump at maximum capacity before any peak load unit is enabled. Seeing as the heat pump manages to cover 81,5 % of the annual heating need with only 9,9 kW as maximum effect output, this is an indication that the installed heat pump has oversized capacity. To analyse this further, the condenser effect output during one year is given in figure 63.

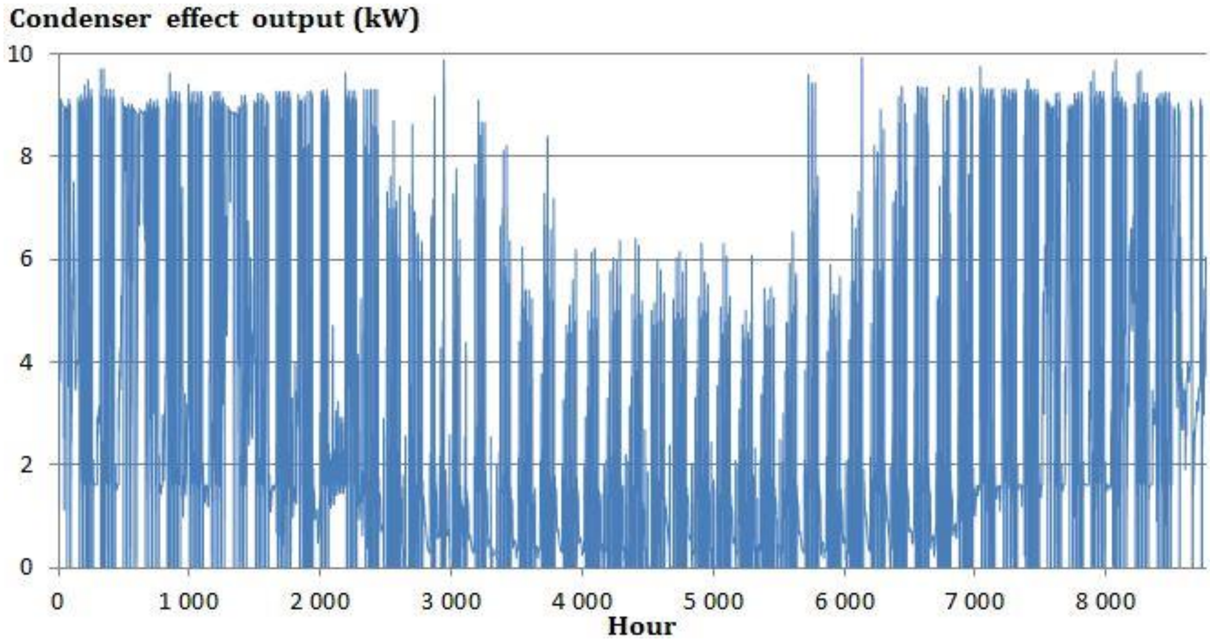

Figure 63 – Condenser effect output during year 2013

Figure 63 shows that the heat pump works around 9,9 kW most of the year, with stable working conditions. If the heat pump had covered a higher share of the effect load, it would have given more unstable working conditions and higher variations in the temperature lift (c.f. section 2.3.3). Stable working conditions and low temperature lift are important to maintain a high heat pump COP. Figure 64 shows how the COP varies with the temperature lift. Values are retrieved from the tab *Base heating.*

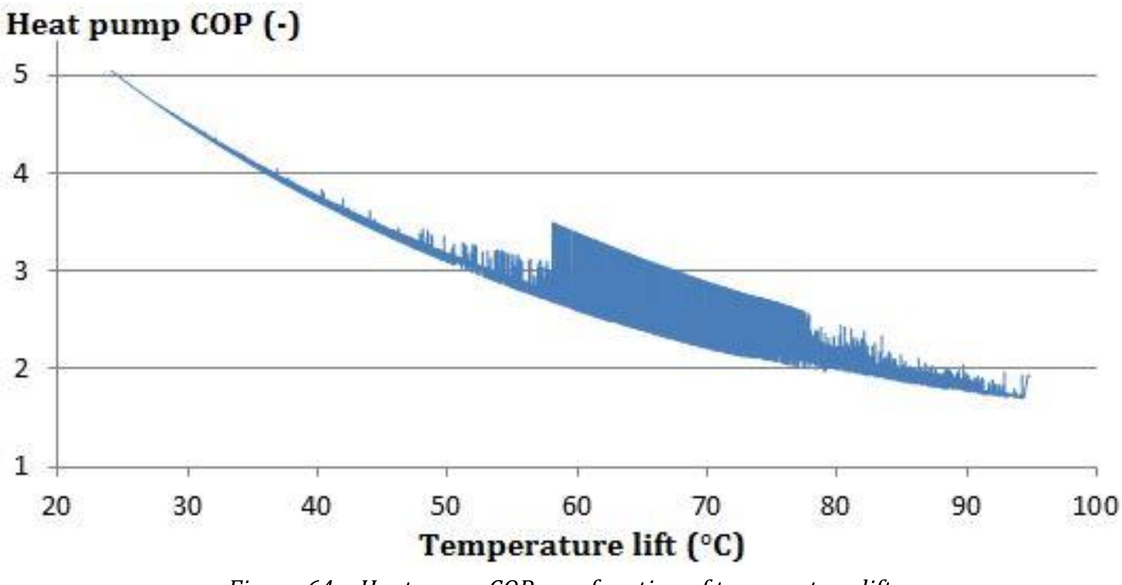

Figure 64 – Heat pump COP as a function of temperature lift

According to IDA ICE, the mean COP over the year is 2,7. Figure 64 shows that the COP drops below 2,7 when the temperature lift is higher than 55 °C. To have a closer look at the temperature lift, the evaporation and condensation temperature is plotted against the ambient air temperature in figure 65.

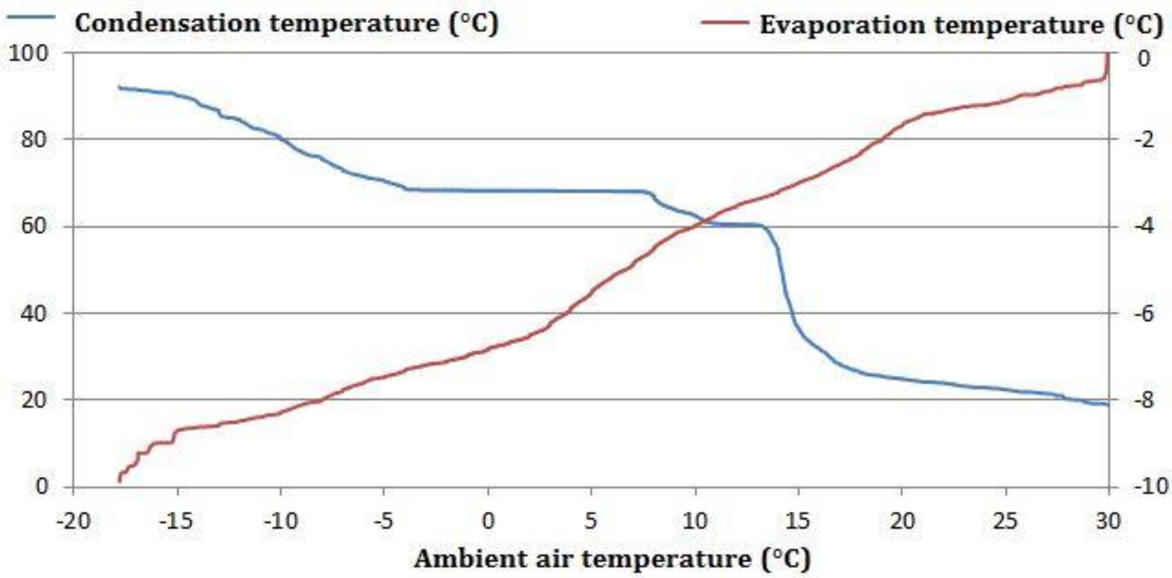

Figure 65 – Condensation and evaporation temperature in the heat pump

As figure 65 shows, the condensation temperature decreases with increasing ambient air temperature. This implies a lower temperature lift and thereby a higher COP of the heat pump.

On the other side, having the heat pump work at part load most of the time like it does in this case will decrease the performance of the compressor, c.f. section 2.3.2. To sum up, the heat pump appears to have oversized capacity in this case. By decreasing the capacity, part load operation can be avoided to a higher degree, while the heat pump still covers the same share of the annual heating need.

Solar collector

Judging by the default heat plant sketch in IDA ICE (see appendix B.5), it looks like the solar collector circuit contributes as a peak load, seeing as placing of the heat exchanger is at the top of the hot water tank. The concept in the kindergarten is to utilize solar collector for sequential *preheating* of hot water (c.f. section 5.4). This made it necessary to examine the default control strategy implemented in the solar collector circuit more closely. An excerpt from the plant in IDA ICE is shown in figure 66.

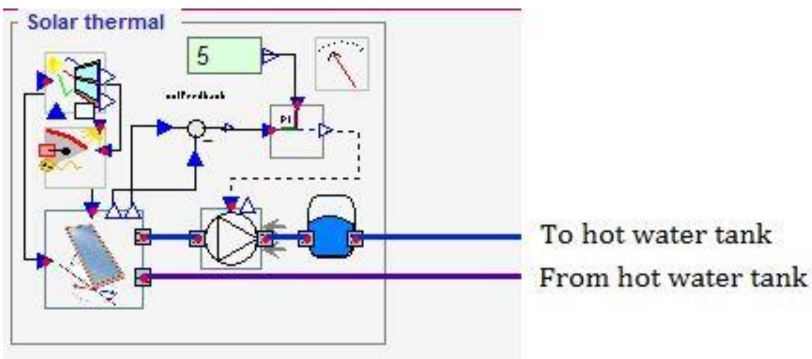

Figure 66 – Solar collector circuit in IDA ICE plant

The PI-controller's input signal is the temperature difference between inlet and outlet temperature to the collector. The reference signal is a constant numbered *5*, which implies that when the temperature difference between the inlet and outlet flow to the collector is higher than 5 °C, the circulation pump is turned on. Otherwise, there is no flow in the circuit. This verifies that the solar collector is indeed used for preheating the tap water and not for peak loads, as the solar fluid circulates independent of the temperature in the upper layer of the tank. Figure 67 confirms this theory, where the mass flow rate of the solar fluid is plotted against the supply and return temperatures to the hot water tank.

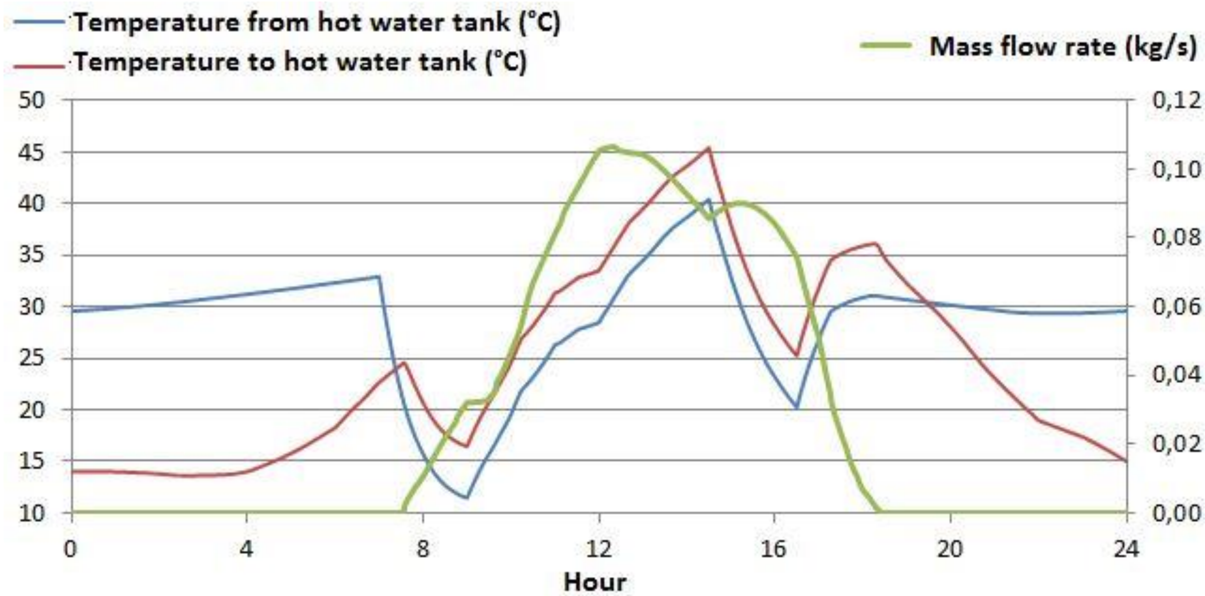

Figure 67 – Temperatures and mass flow rate in solar circuit, 2013-07-09

As can be seen in figure 67, the mass flow rate starts increasing when the temperature difference exceeds 5°C. The decrease of mass flow rate around 12:00 is due to time delay in the controller, and is caused by the small stagnation of temperature difference. As shown in the figure, there is circulation in the solar collector circuit even when the supply temperature to the tank is under 50 °C. Seeing as the tapping temperature from the upper layer of the tank is set to 60 °C, one can say for sure that the solar collector contribute to preheat the water. This concept differs somewhat from the strategy at the kindergarten, where the circulation pump runs as long as the temperature in the collector is higher than in the hot water tank (c.f. section 4.3.2). Trying to make the IDA ICE plant more similar to the kindergarten was too complex to carry out. On the other hand, the advantage with this IDA ICE default plant is that it utilizes as much as possible of the solar energy.

Figure 68 shows the annual effect output from the solar collector.

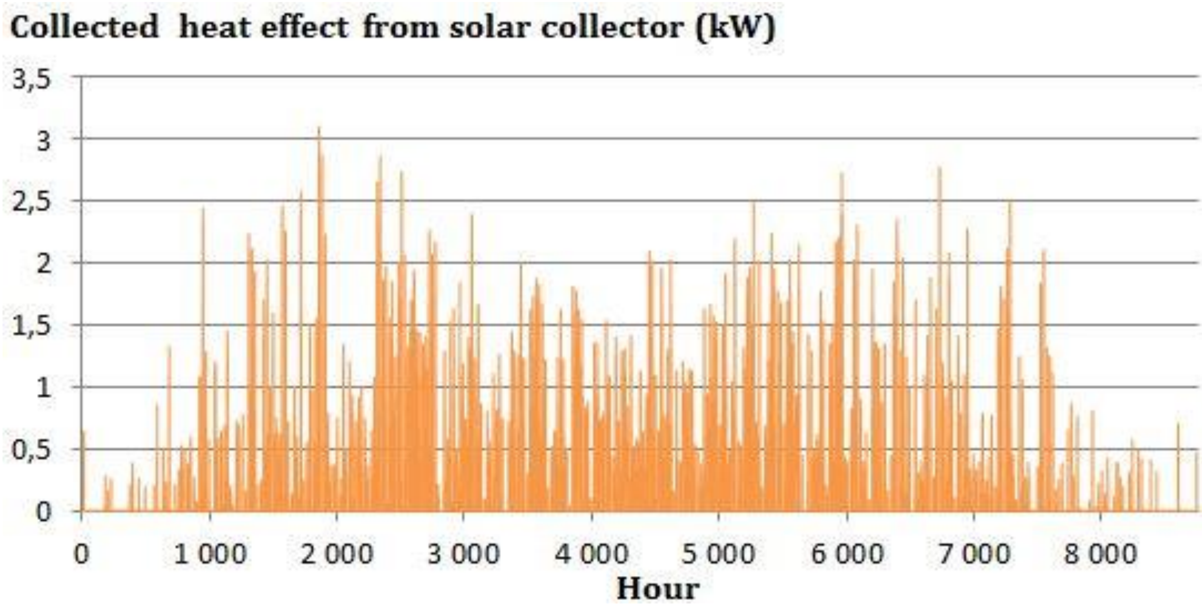

Figure 68 – Collected heat from the solar collector during year 2013

One would expect a higher share of heat effect during the summertime from 4000-6000 hours, as this is when the solar radiation is at its peak. What must be remembered though, is that the need for heating is lowest during the summer. Nevertheless, as the solar collector contributes to preheating, it should deliver heat at maximum capacity all the time. The collected heat was calculated manually to confirm the results, by using the following formula:

$$Q = \dot{m} * c_p * \Delta T \tag{19}$$

The collected heat, Q, was calculated for each time step from the table in *Solar termal*, containing the mass flow rate and temperature levels. The specific heat capacity of the solar fluid is found in *Solar collector outline* in IDA ICE; 3 444 j/kgK. This gives a mean collected heat of 288 W, while the mean value calculated by IDA ICE is 294 W. This error of 2,1 % is considered acceptable, which means that the results in figure 68 are reliable according to the given control strategy.

Electrical boiler

The heating effect output from the electrical boiler is given in figure 69.

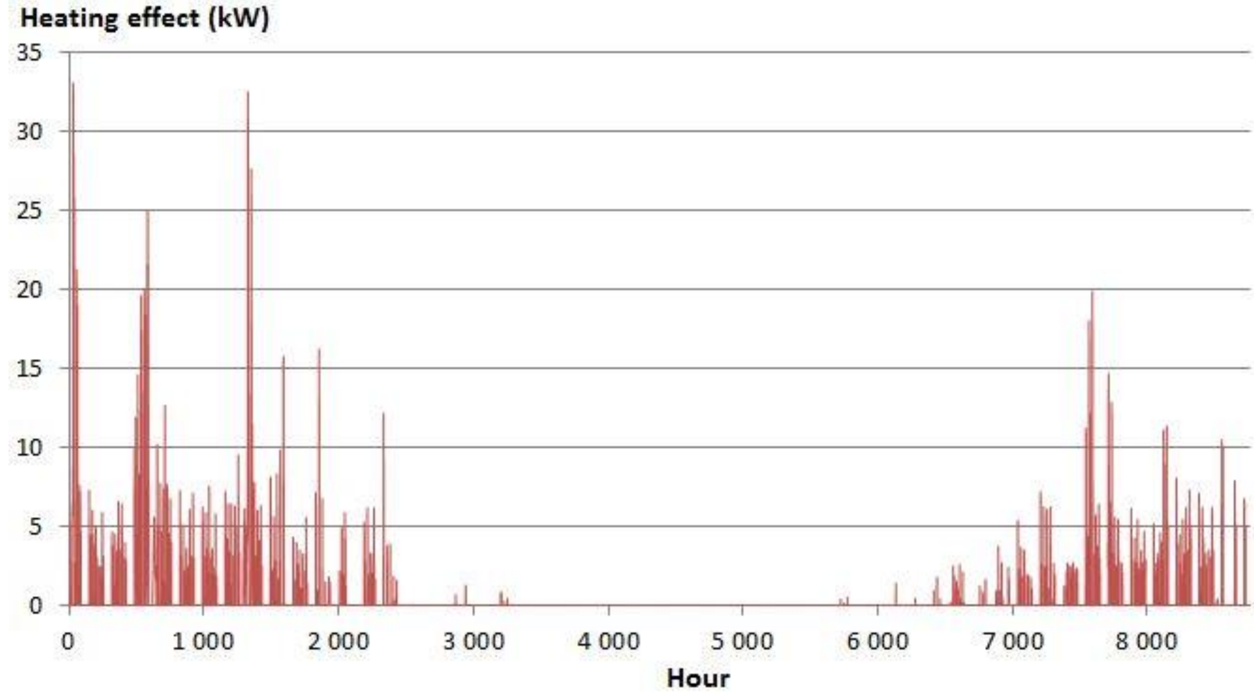

Figure 69 – Electrical boiler heating load (year 2013)

The shape of the heating effect output from the boiler reminds of a typical peak load curve, in that it contributes during the coldest periods of the year. Remember that the heat pump delivers around 10 kW most of the time during the winter and somewhat smaller effect output during the summer. The peak loads are covered by the boiler, to maintain stable working conditions for the heat pump.

7.1.4 AIR FLOW RATES

A 24 hour simulation was performed at the warmest day of the year (2013-07-09), to analyse the zone ventilation control. The warmest day was chosen as simulation day because this makes it possible to study the effect of VAV with temperature control.

CAV custom macro

The following figures present the results of the custom CAV macro defined in section 5.6.1.

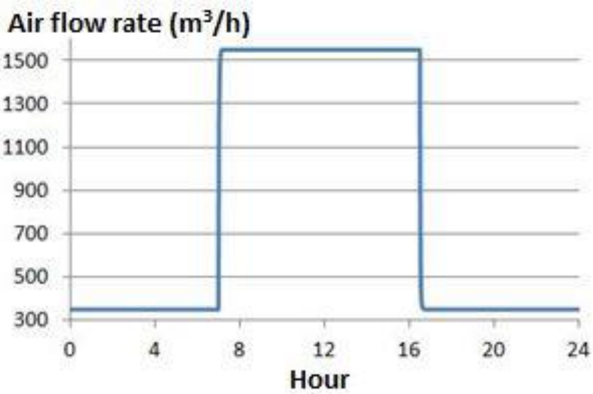

Figure 70 – Air flow rate in zone 1, 2013-07-09

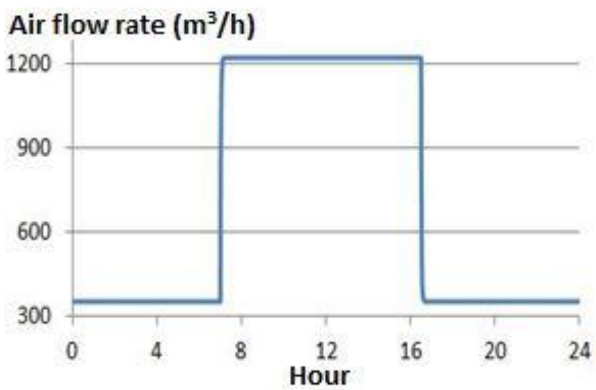

Figure 71 – Air flow rate 2013-07-09 in zone 4

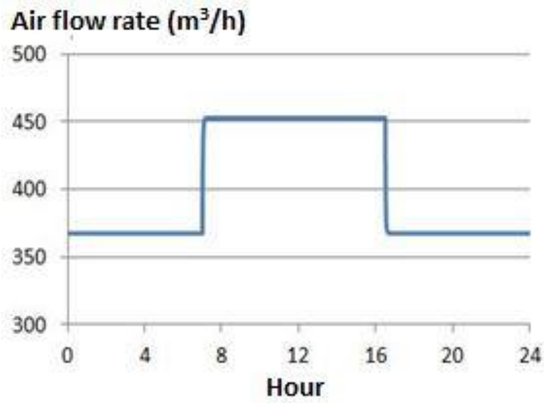

Figure 72 – Air flow rate 2013-07-09 in zone 6

Figure 70, 71 and 72 show the air flow rate in respectively zone 1, 4 and 6, the three zones with CAV. The aim with this macro was to have constant ventilation during the working hours, and a minimum air flow rate outside the occupancy period. As the figures shows, the correct principle with regard to working hours is achieved. The AHU supplies maximum ventilation between 07:00-16:30, and minimum in the remaining hours. The correct air flow rates of each zone are also obtained, cf. table 12 in section 5.6.3.

VAV custom macro with control based on occupancy and temperature level

Figure 73 illustrates the air flow rate in zone 2 in proportion to the indoor air temperature, to see whether the implemented VAV control behaves as wanted.

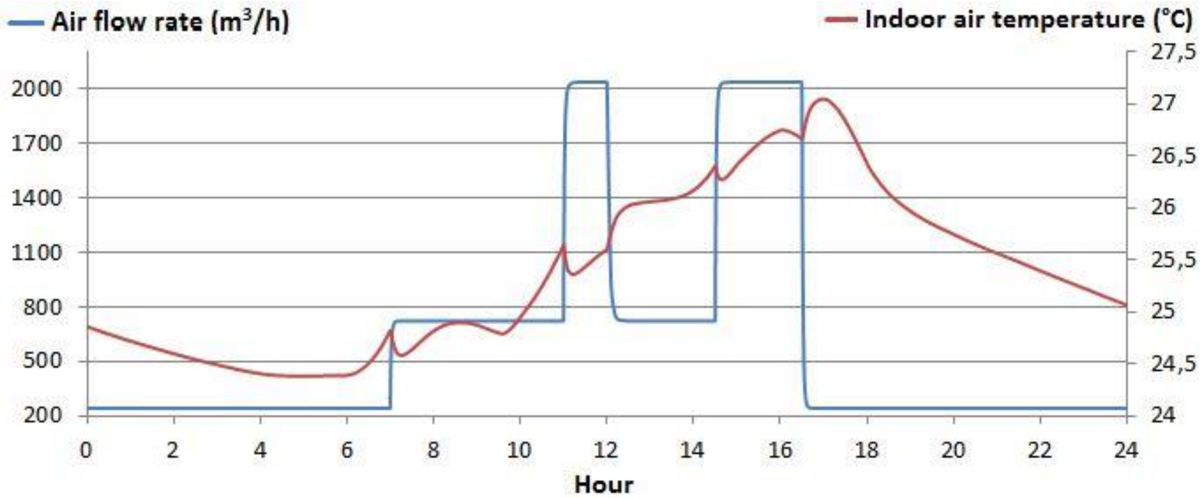

Figure 73 – Air flow rate in proportion to indoor air temperature, 2013-07-09 zone 2

The occupants in zone 2 are present according to the schedule called "Young children" (c.f. section 5.6.4), which implies that the zone is occupied from 07:00-09:00, 11:00-12:00 and 14:30-16:30. In reality the occupancy is based on signal from a movement sensor in the zone, but in IDA ICE a schedule has to be defined. Figure 73 indicates that the occupancy is the determining factor, as the air flow rate is always kept at a minimum when no people are present. Have in mind that the minimum air flow rate is 2 m^3/hm^2 outside the operating hours and 6 m^3/hm^2 within the operating hours. An overview of minimum and maximum air flow rates for each zone is given in table 12, section 5.6.3.

When zone 2 is occupied, the indoor air temperature will determine the air flow rate. The maximum temperature limit within operating hours is 25 °C, which is exceeded around 10 o'clock. The air temperature keeps increasing throughout the operating hours. Hence the air flow rate is kept at a maximum level in every occupied time period as long as the temperature level is too high.

The other zone controlled by temperature and occupancy is zone 5, shown in figure 74.

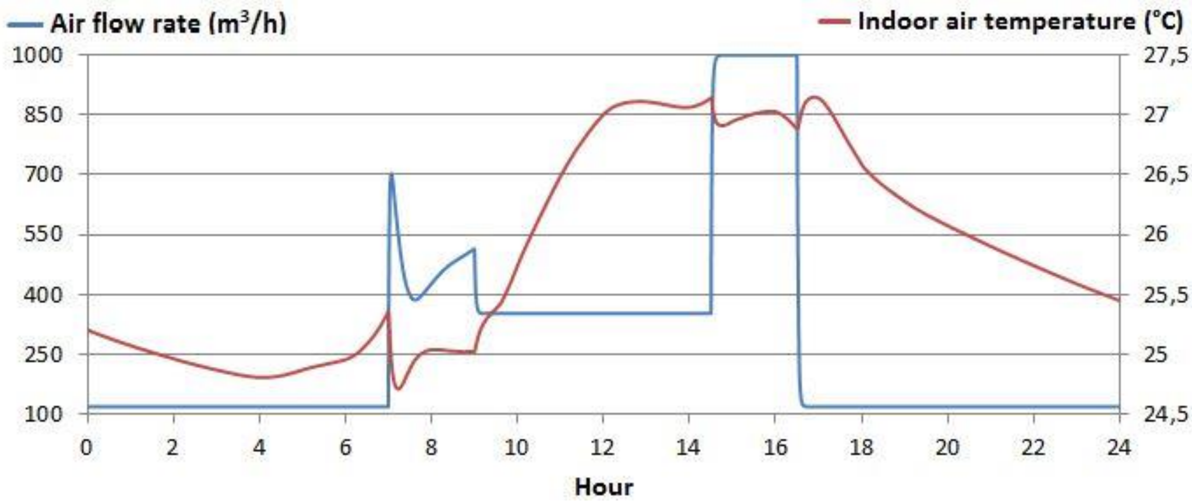

Figure 74 – Air flow rate in proportion to indoor air temperature, 2013-07-09 zone 5

The occupancy in zone 5 is defined by the schedule "Older children", which implies presence from 07:00-09:00 and 14:30-16:30. As with zone 2, the air flow rate is kept at a minimum level (defined in table 12) when there are no occupants present. When the people arrive at 07:00, the temperature level is higher than the maximum limit, which causes a rapid increase in air flow. Between 07:00 and 09:00 the temperature is hovering around the maximum limit, which results in a varying air flow rate. At the second period with occupancy, the air flow rate is kept at a maximum level all the time due to high temperature in the zone. Still, this is not enough to decrease the temperature to an acceptable level. One reason may be that the unoccupied period lasts too long, where the air flow is limited to a lower level independent of the air temperature.

An attempt on decreasing the temperature is carried out by letting the air temperature be the deciding factor during operating hours. The resulting air flow rate in proportion to the temperature is shown in figure 75.

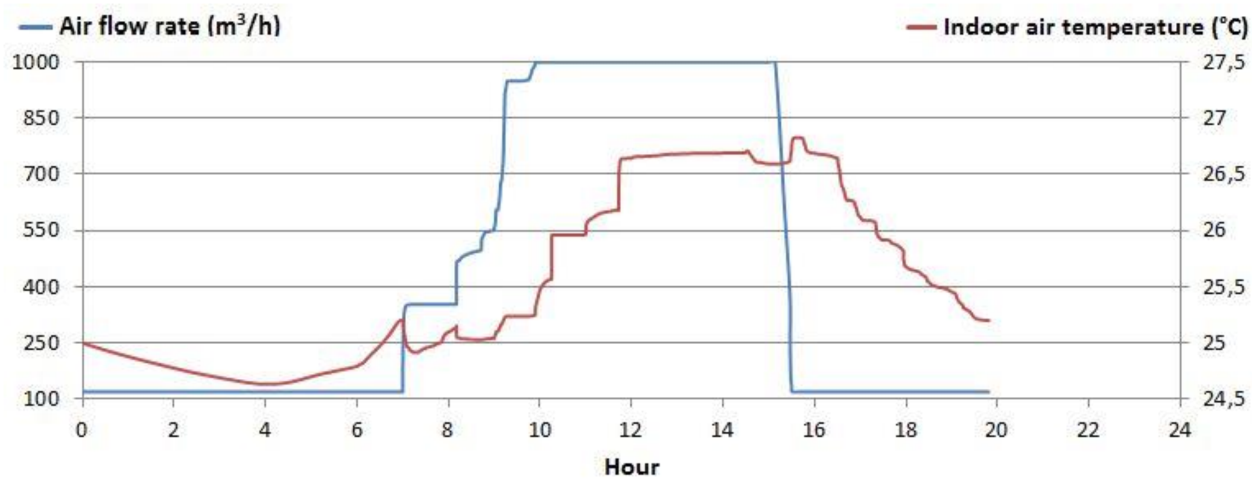

Figure 75 – Zone 5 - with temperature as the deciding factor during operating hours 2013-07-09

As figure 75 shows, the air temperature is too high, even when occupancy is not taken into account in the operating hours. The air flow rate increases gradually as the temperature increases, but the maximum air flow rate is not high enough to cool down the zone. This makes the original control (figure 74) preferable, as this leads to more energy efficient ventilation. Another argument is that replacement of polluted air will still take place, even though the zone temperature is too high.

VAV custom macro with control based on occupancy, temperature level and CO_2 level

The air flow rate in zone 7 is shown in proportion to respectively indoor air temperature in figure 76 and CO_2 concentration in figure 77.

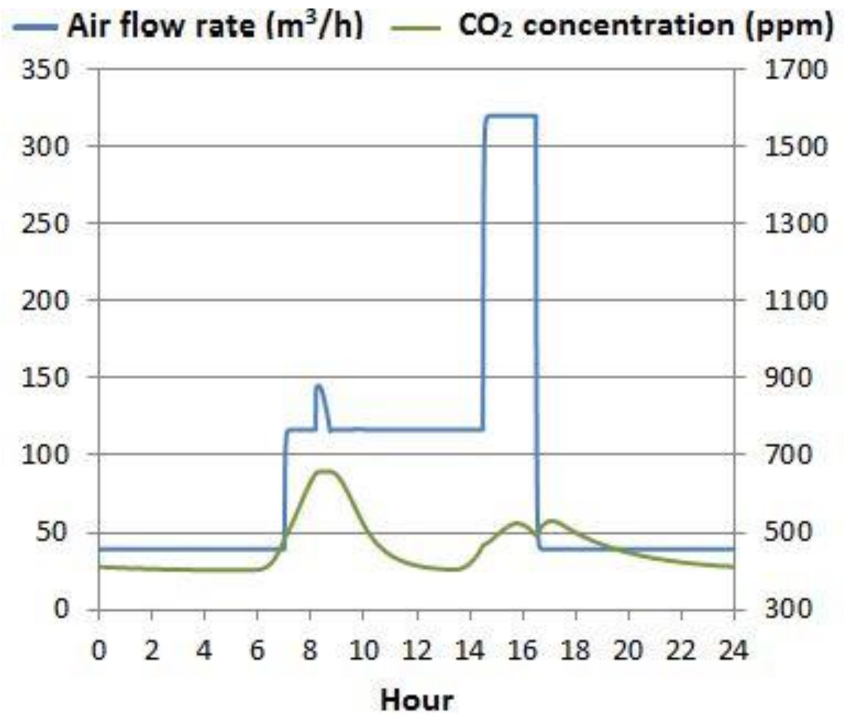

Figure 76 – Air flow rate zone 7 in proportion to CO_2 concentration, 2013-07-09

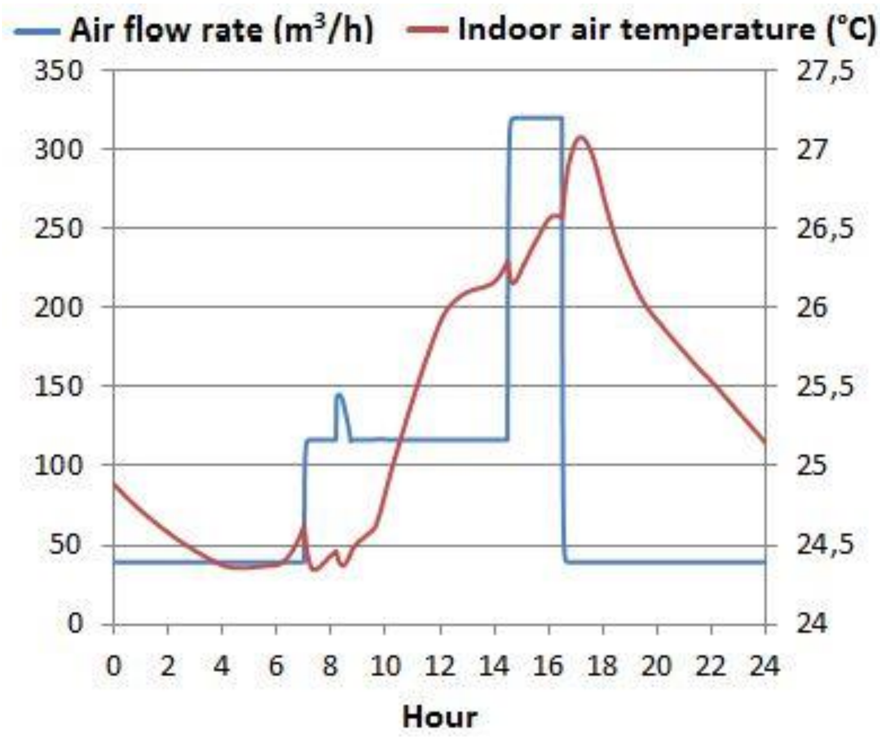

Figure 77 – Air flow rate zone 7 July in proportion to indoor temperature, 2013-07-09

The occupancy schedule for this zone defines that there are people in the zone present from 07:00-09:00 and 14:30-16:30. It becomes clear from figure 76 and 77 that occupancy is held as the highest precedence, as was the intention with this ventilation control. Figure 76 shows that the CO_2 concentration never touches its maximum limit, which makes the indoor temperature the determining factor over CO_2 concentration for the air flow rate in zone 7. The small peak in air flow between 8:00 and 9:00 is due to increased CO_2 concentration (c.f. figure 76) when the temperature was below the maximum limit, caused by a larger group of people entering the zone at 7:00. Seeing as the group leaves the zone at 9:00, there is not enough time for a larger increase of CO_2 concentration.

Another way of verifying the control principle is to plot the output signal from the temperature and CO_2 PI controllers against each other, as illustrated in figure 78. The blue line shows the final ventilation output signal, when minimum air flow rate demands are taken into account (c.f. table 12).

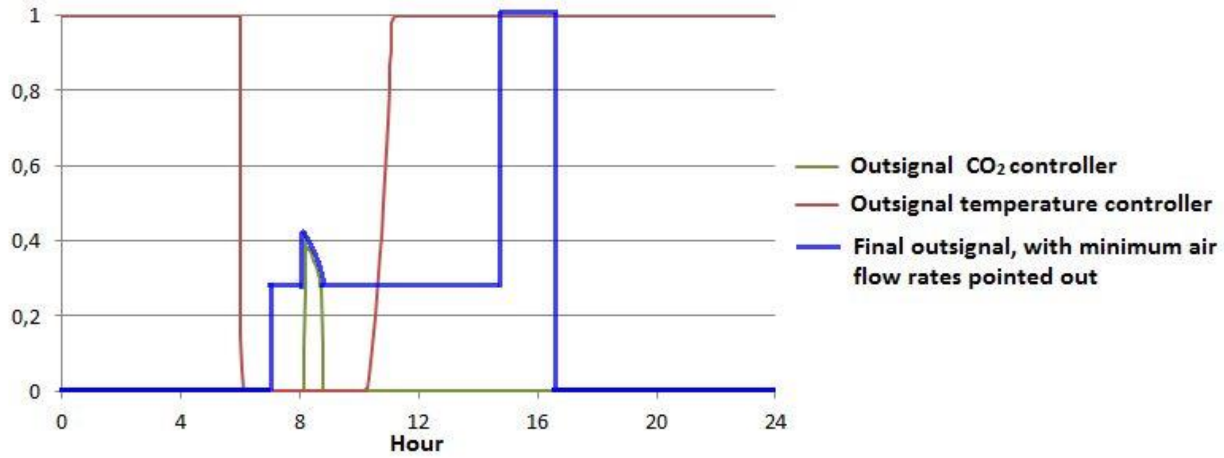

Figure 78 – Output signal from CO_2 and temperature controllers, with resulting final ventilation signal in zone 7, 2013-07-09

The blue line is drawn along the temperature and CO_2 curve, tangent to the highest value between the two signals at each time step while at the same time maintaining the minimum air flow rate demanded. The form of this curve becomes exactly equal to the air flow rate curve shown in figures 76 and 77. For this reason the behaviour of the macro has proven to be correct in zone 7, as the precedence of the temperature controller and CO_2 controller switches according to the one with the highest output signal value.

Zone 8 is the other zone controlled by temperature, CO_2 concentration and occupancy. The occupants are assumed present at the same time as in zone 7; from 07:00-09:00 and 14:30-16:30. Figure 79 shows the air flow rate demand in the zone, while the output signal from the controllers are given in figure 80.

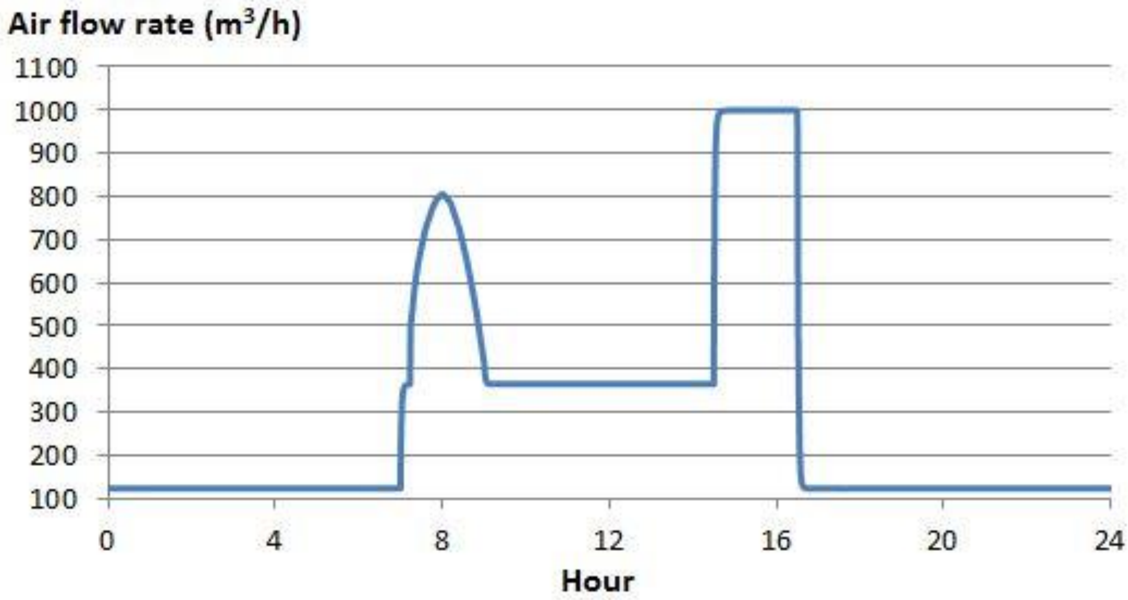

Figure 79 – Air flow rate zone 8, 2013-07-09

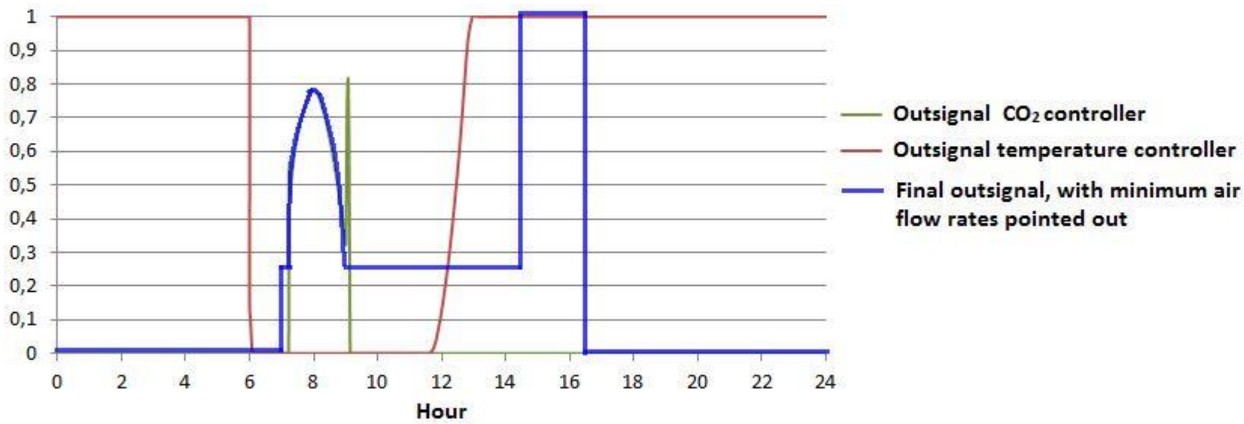

Figure 80 – Output signal from CO2 and temperature controllers, with resulting final ventilation signal in zone 8, 2013-07-09

As with zone 7, the final ventilation signal curve in zone 8 (figure 80) becomes exactly equal to the air flow rate curve (figure 79). Have in mind that the occupancy has the highest precedence, so outside the occupancy hours the air flow rate will be set to a minimum level regardless of the temperature and CO_2 concentration in the zone. Given these points, the implemented control strategy behaves as intended.

Altogether all three ventilation control strategies works as intended. The aim with the ventilation macros was to obtain energy efficient ventilation system while at the same time fulfilling the minimum requirement regarding air flow rates (c.f. table 6). Table 24 lists the mean annual air flow rates during operating hours for every zone, based on a whole year simulation (year 2013). Zone 3 is not included as this is a non-ventilated zone.

Table 24 – Mean annual air flow rates during operating hours

Zone	**Mean annual air flow rate during operating hours (m^3/hm^2)**
1	9,0
2	6,1
4	7,1
5	6,2
6	2,5
7	6,2
8	7,9

Hence all zones fulfil the minimum requirement of 6 m^3/hm^2 mean minimum air flow rate, except zone 6. As zone 6 is a technical room with minimum occupancy, this is considered satisfactory. The mean air flow supplied to the kindergarten over a year is 3841 m^3/h, which equals 46 % of the capacity of the AHU.

7.1.5 INDOOR ENVIRONMENT

A one-year simulation was performed to analyse the indoor environment of the kindergarten. Looking at indoor temperatures, CO_2 concentration, PMV and PPD values, the thermal comfort of the building can be pointed out.

Indoor temperatures

Table 25 shows the mean annual temperatures of the occupied zones.

Table 25 – Mean annual temperature in the occupied zones

Occupied zone	Mean annual temperature (°C)
1	21,0
2	21,5
4	21,3
5	21,8
7	21,1
8	21,5

Compared to the results from the project thesis, all zone temperatures have now decreased with an average of 1,6 °C. Implementing a night set-back of 19 °C (c.f. section 5.6.3) is the main reason for this reduction. Whether or not the heat plant is able to supply sufficient heat to achieve an acceptable temperature during the winter period is questionable. Hence the temperature throughout a whole year in the coldest zone is studied in figure 81 on next page. Have in mind that the minimum temperature setpoint during occupied hours is 22 °C.

Figure 81 shows that the temperature in zone 1 is below 22 °C constantly during seven months, from September (around 6 000 hours) to May (around 2 800 hours). What must be pointed out is that a heat pump works best under stable conditions, meaning that its weakness occurs with frequent changes in heating need. When the heating setpoint is raised 3 °C during the operating period of only 9,5 hours, it may not be enough time for the heat pump to meet the demand before the setpoint is lowered again.

The heat pump operation is further analysed in section 7.3.2 when the night set-back is removed. Meanwhile, a whole year simulation is performed with only 2 °C night set-back, meaning that the heating setpoint is set to 20 °C outside operating period. The resulting mean annual temperatures in occupied zones have all increased with 0,3 °C. Still, the temperature in the coldest zone never reaches 22 °C constantly during five months, from October to March.

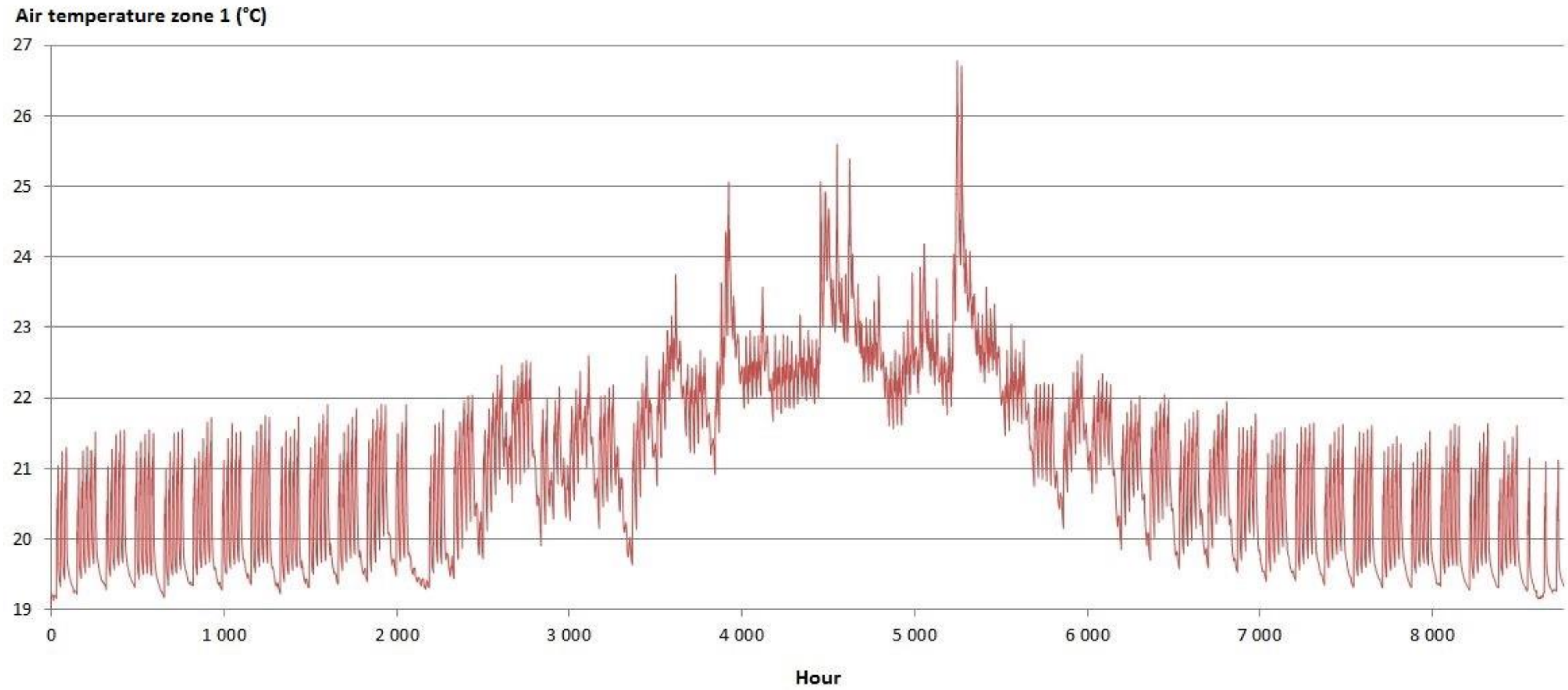

Figure 81 – Air temperature in zone 1, year 2013

An advantage with the night set-back is that the indoor temperature during summertime is lowered, compared to the results in the project thesis where night set-back was not implemented. A simulation from June throughout August shows that the mean temperature in the hottest zone during the summer months is 24,2 °C, with a maximum of 28,2 °C. In comparison, the project thesis showed over 29 °C for the hottest zone.

Nevertheless, this model exceeds the maximum temperature limit. High temperatures during summertime are expected in a passive house, as the building body is very airtight. Another key point is that there is no window opening control defined in the model. The integrated window shading in IDA ICE is controlled by the light, but whether or not this works optimally is uncertain.

CO_2 concentration

A simulation throughout year 2013 shows that all zones have a satisfactory CO_2 level, as none of them at any time exceed the maximum limit of 1 000 ppm (vol) CO_2. In view of the kindergarten being an outdoors kindergarten, the defined schedules for each zone (c.f. section 5.6.4) implies a low occupancy level. Whenever occupants enter a zone, they leave again before the CO_2 concentration reaches the maximum level. Figure 82 shows the CO_2 concentration through a 24-hour period for the zone with the highest mean CO_2 concentration; zone 5.

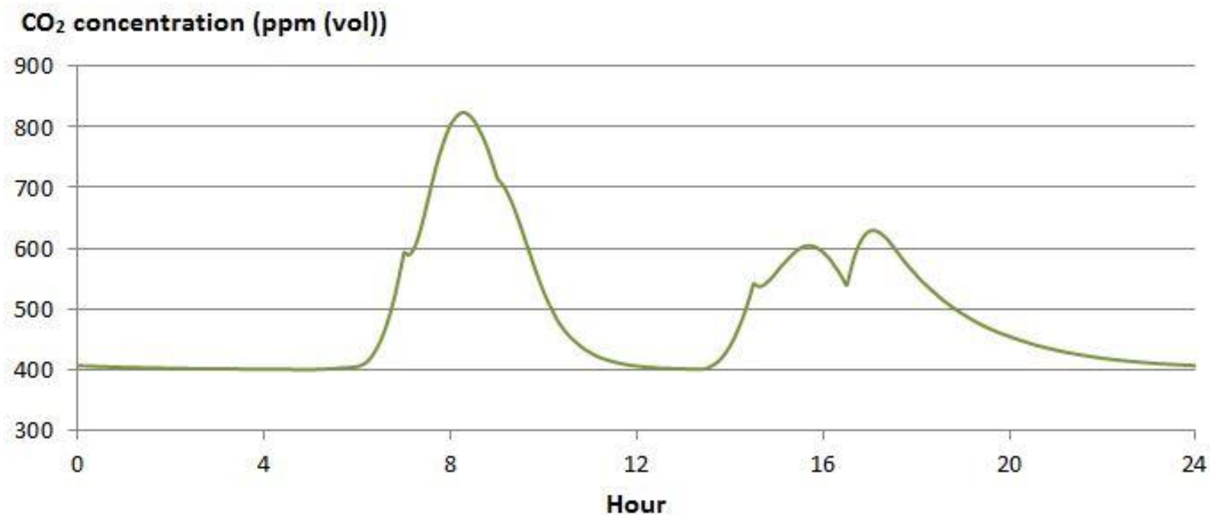

Figure 82 – CO_2 concentration zone 5, 2013-07-09

The curve corresponds with the occupancy schedule in the zone; the occupants are present from 07:00-09:00 and from 14:30 -16:30.

Thermal comfort

When looking at the thermal comfort, values from IDA ICE based on PPD and PMV (IDA ICE own source is the standard *NS-EN 15251:*2007) are taken into account. In table 26 the result from a one-year simulation is given.

Table 26 – Thermal comfort zone 5

Comfort category	No. of hours over a year
I - best	4 829
II – good	6 536
III – acceptable	8 684
IV - unacceptable	76

The values in table 26 are from zone 5, which has the highest amount of hours in the category "unacceptable" out of the occupied zones. In percentage, 76 hours a year account for only 0,9 % of the time with unacceptable PMV and PPD values. In comparison, the percentage was 2 % with the old model with no night set-back of heating setpoint. As much as 55 % of the time is characterized as the best thermal comfort. Even so, the air temperature in zone 5 exceeds 26 °C for 103 hours during a year, which is 53 hours above the maximum limit (c.f. table 6). Whether these hours are within the operating hours or not is difficult to say, as the result only gives the total number of hours during a year.

Summary indoor environment
The IDA ICE results regarding thermal comfort are very similar to the calculations that were performed on the passive houses in Denmark before the measuring (section 2.1.1). The calculated percentage of time when the indoor temperature was expected to exceed upper limit in Denmark was between 0-1 %, whereas IDA ICE shows 0,9 % for Haukåsen kindergarten. The *measured* values are however a lot higher, 32 % of the time the indoor temperature is unacceptable in the worst case of the passive houses, due to deviant user behaviour. In other words, the simulations on Haukåsen kindergarten cannot guarantee that 99 % of the operating time is going to be acceptable, it can differ a lot in the reality judging by the research in Denmark.

7.1.6 CLIMATE FILE ANALYSIS
In order to see how the climate affects the heating need, one-year simulations were performed with Oslo and Östersund climate. These two locations are chosen as they were also used in the project thesis work, making it possible to compare the result also with the old model. Table 27 summarizes the heating needs for the three locations.

Table 27 – Heating need for the kindergarten at different locations

	Heating need (kWh)		
	Trondheim	**Oslo**	**Östersund**
Space heating	10 223	11 002	14 579
Heating coil	5 733	7 048	15 943
Domestic hot water	10 873	10 801	11 381
TOTAL	**26 829**	**28 851**	**41 903**

It seems unreasonable that the domestic hot water need differs at the three locations. And the fact that a kindergarten in Oslo has a higher heating need than the same kindergarten placed in Trondheim does not seem reliable, as Trondheim has a lower mean annual temperature. According to *Forbrukerrådet* [2013], the mean annual temperature in Trondheim is 5,6 °C and in Oslo 6,2 °C. These values are based on 30 years of temperature reading. The temperature data for the default climate files in IDA ICE (Results → AHU temperatures → Outside air dry-bulb temperature) was examined more closely, which revealed design temperatures that deviate from what is expected. Table 28 shows the temperature data logged for each location.

Table 28 – Temperature data logged from IDA ICE for Trondheim, Oslo and Östersund

	Trondheim	**Oslo**	**Östersund**
Dry-bulb minimum temperature (°C)	-17,8	-17,0	-25,7
Dry-bulb maximum temperature (°C)	30,0	28,2	26,5
Mean annual temperature (°C)	6,6	7,1	3,8

The mean annual temperature in Trondheim is 1 °C higher than expected, but at the same time the mean annual temperature in Oslo is 0,9 °C higher. With this, the climate files cannot be said to be the cause of the strange result in table 27. With Östersund on the other hand, the connection between the climate files and the resulting heating need seems more reasonable. Why the heating need is higher in Oslo than in Trondheim and why the need for hot water differs between the locations, is a subject that ought to be further studied. At this point, the cause could not be detected.

7.2 SYSTEM DESIGN ANALYSIS

Different parametres in the system *design* of the model are changed separately to see the consequences. This section presents inquiry regarding solar collector area and tilting angle. How the shape of the hot water tank affects the result is also examined.

7.2.1 SOLAR COLLECTOR AREA

The total area of the solar collector was doubled to 13,05 m^2, by doubling the number of solar collector units when building the *ESBO Plant*.

A one-year simulation shows that the solar collector with doubled area collects only 11 kWh more than the original collector, even with exactly the same mass flow rate. A higher heat contribution was expected, seeing as it takes longer time for the solar fluid to flow through a collector of doubled area. Hence the fluid will be further heated, resulting in a higher outgoing temperature from the collector. With the same mass flow rate, a higher outgoing temperature gives a higher temperature difference over the collector, which results in a higher heat contribution (c.f. formula 18).

To examine this more closely, the supply temperature to the tank is compared with the reference model. The mean temperature over a year is only 0,07 °C higher than in the reference model, which substantiates the low amount of collected heat from the solar collector. Nevertheless, this subject ought to be further studied.

7.2.2 SOLAR COLLECTOR TILTING ANGLE

As explained in section 2.2.1, the tilting angle will influence the radiation per square meter to the solar sollector. According to D. Zijdemans [2012, p. 110], the optimal tilting angle in Trondheim is 44°. This angle was adjusted directly in the solar collector unit when building the *ESBO Plant* to see how it affects the annual collected heat from the solar collector.

The heat contribution from the solar collector in the reference model was 1 608 kWh, so a higher heat contribution is expected when the solar collector are mounted at an optimal angle. The result after a one-year simulation is however not as expected, as the heat contribution has now decreased to only 459 kWh. The incoming radiation is plotted in figure 83 to analyse this further.

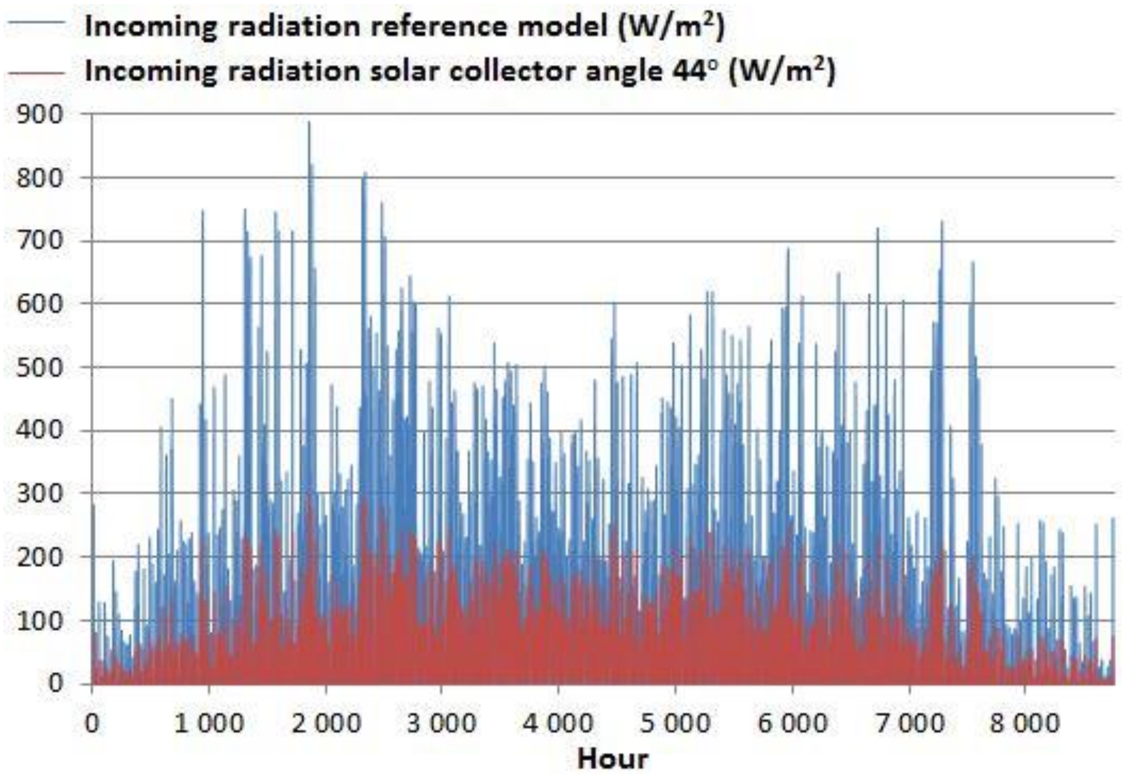

Figure 83 – Incoming annual radiation in reference model and model with optimal solar collector angle

Figure 83 shows an unexpected result, in that the incoming radiation is higher in the reference model all year around. This is inconsistent with the theory on the subject, and further investigation regarding the solar collector angle ought to be performed.

7.2.3 SHAPE FACTOR HOT WATER TANK

Why the shape factor of the hot water tank was examined further, can be explained by quoting section 2.5.1 regarding stratification: "In a stratified tank, the cold water stays in the lower part and the hot water in the upper part, with minimum stirring in between. This is important in a system containing solar collectors, as the collector efficiency will be higher with a low inlet temperature. An accumulator tank with large height is preferable to maintain the stratification".

In the reference model the hot water tank volume is 0,4 m^3, with a shape factor of 2,33 (height divided by diameter) and an insulation U-value of 0,3 W/m^2K. While the volume and insulation were kept constant, the shape factor was changed to 1,6 to see how this affects the temperature layers and stratification in the tank.

The results in this model are unexpected exactly the same as for the reference model. All eight temperature layers evolve alike for both models. The temperature levels to and from the solar collector are also identical independent of the shape of the hot water tank. This result is inconsistent with the stratification concept, as one would expect the shape of the hot water tank to influence the temperature layers to a higher degree.

7.3 SYSTEM OPERATION ANALYSIS

Changes in the *operation* of the system are carried out to analyse the impact of these changes. Removing occupancy controlled ventilation and night set-back of heating setpoint are among the operation strategies studied in this section. The impact of internal gains is also examined.

7.3.1 VENTILATION CONTROL

Motion sensors are removed from the zones, to see how this affects the energy need, especially for space heating, heating coil and fan operation. The impact on air flow rates is also examined.

This implies that the ventilation is controlled only by temperature and/or CO_2 concentration. In order to maintain the same ventilation strategies but without the motion sensors, the macros were changed as shown in figure 84.

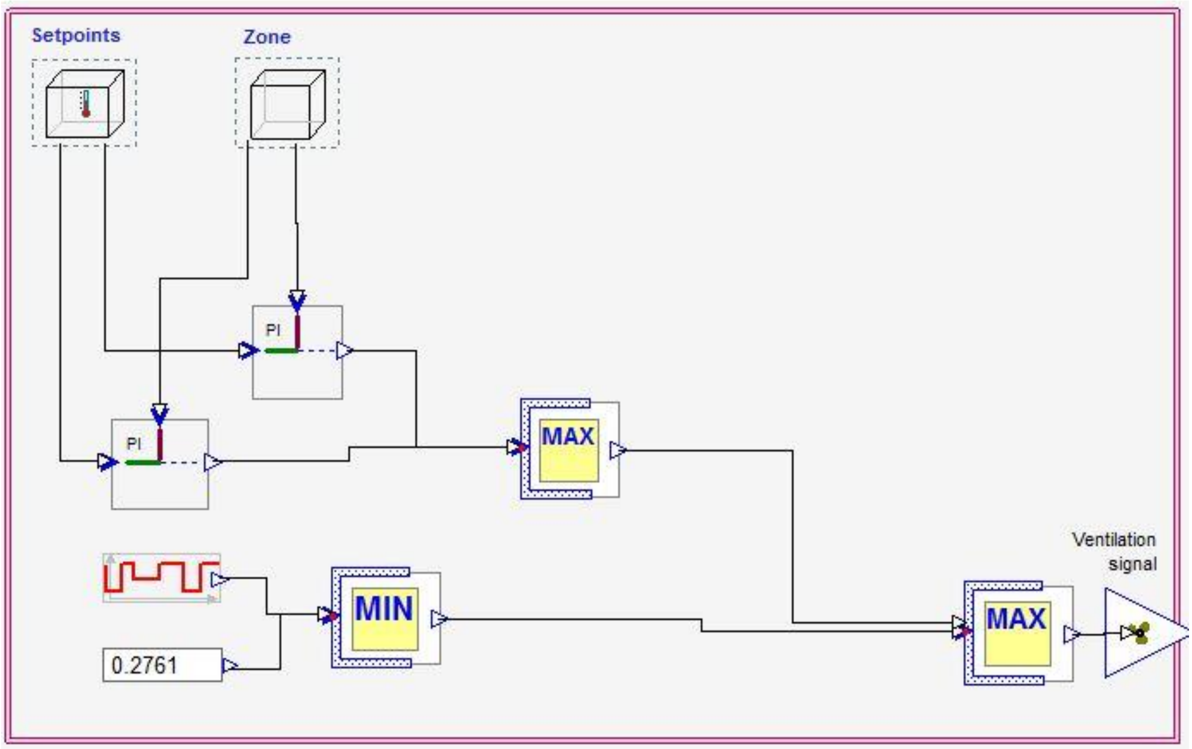

Figure 84 – Ventilation macro without motion sensors (zone 7 and 8)

Compared to the original macro (figure 56 in section 5.6.1), only the multiplier block with zone occupancy as input signal is removed. The same applies for the macro implemented in zone 2 and 5, where the control is based only on the air temperature in the zone.

The results after a one-year simulation shows that the mean air flow rate supplied to the kindergarten has increased from 3 841 m^3/h to 4741 m^3/h. How the changes affect the net annual energy need is given in table 29. Energy need for technical equipment, lightning and heating of domestic hot water are not included in the table, as these categories are independent of the changes and thereby remain the same.

Table 29 – Net annual energy need after removing motion sensors

	Net annual energy need (kWh)	
	Reference model	**Removal of motion sensors**
Space heating	10 223	8 628
Heating coil	5 733	5 300
Fans	5 392	7 914
Pumps	65	62

Table 29 shows an increase of 47 % in energy need for fan operation, which was expected due to higher mean air flow rate. The energy need for space heating and the heating coil has decreased. Lower energy need for pump operation is connected to the decreased energy need for the heating coil. To sum up, the total delivered energy is raised to 33 418 kWh/year, compared to 32 030 kWh/year which was the case in the reference model. Altogether the motion sensors contribute to a more energy efficient building.

7.3.2 NIGHT SET-BACK

The night set-back implemented in the reference model will presumably save energy for space heating. More difficult working conditions for the heat pump can on the other hand be expected, as it will have to adjust to more frequent temperature variations. The COP might also be decreased as a result of high starting current every morning. Given these points, the annual heating need, temperature lift and COP for the heat pump are factors being examined in this section.

The net annual heating need is given in table 30.

Table 30 – Net annual heating need in reference model compared to model without night set-back

	Net annual energy need (kWh)	
	Reference model	**Model with no night set-back**
Space heating	10 223	18 255
Heating coil	5 733	3 667
Domestic hot water	10 873	10 876
TOTAL	**26 829**	**32 798**

Table 30 shows that the space heating need in the model without night set-back has increased with 79 %. An increase was expected, due to higher heating setpoint outside operating hours. Energy need for the heating coil has on the other hand decreased. This is presumably caused by a higher mean air temperature. Hence the exhaust air holds a higher temperature, which is recovered in the heat recovery unit.

As for delivered energy for heating purposes, energy delivered to the heat pump compressor has increased with 35 %, while the energy to the electrical boiler has decreased with 30 % (values from *Base heating* and *Top heating* tab). The total amount of delivered energy has increased with 2014 kWh, an enhancement of 16,6 %. The advantage with no night set-back is however that the mean air temperatures in the zones have been raised to an acceptable level, as all mean air temperatures are above the minimum limit of 22 °C.

When looking at the mean annual heat pump COP, it has actually decreased in the model with no night set-back. In the reference model the mean annual COP was 2,74, while it is now down to 2,67. This was unexpected given that the compressor probably has a high starting current every morning in the reference model. The question is whether the decreased mean supply temperature to the hot water tank in the reference model make up for this starting current. To analyse this further, the heat pump temperature lift from the reference model is compared to the model with no night set-back, see figure 85.

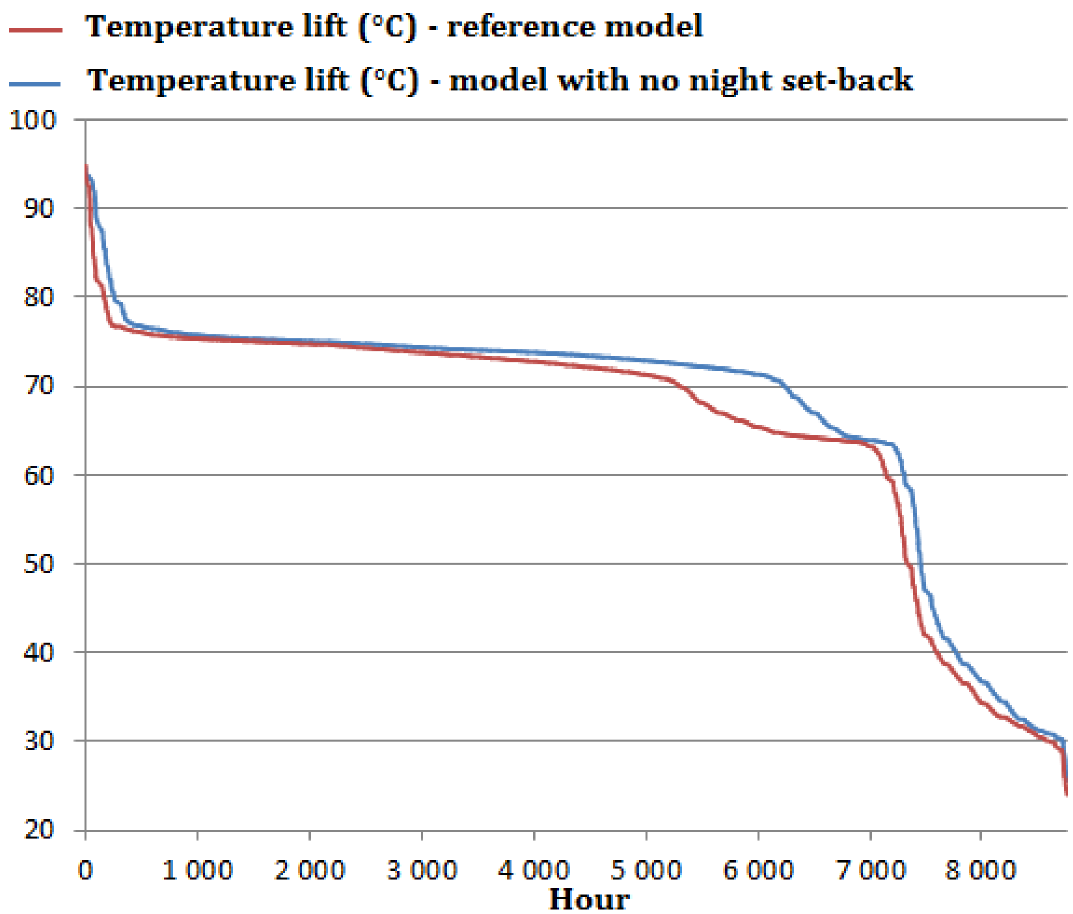

Figure 85 – Temperature lift in the heat pump for the reference model and the model with no night set-back

As can be seen in figure 85, the temperature lift throughout the year is indeed higher in the model with no night set-back, which explains the lower COP. Nevertheless, one should be careful using night set-back when a heat pump is involved, as it may wear down the compressor over time. Looking at the compressor work over a 24-hour period shows this principle, see figure 86.

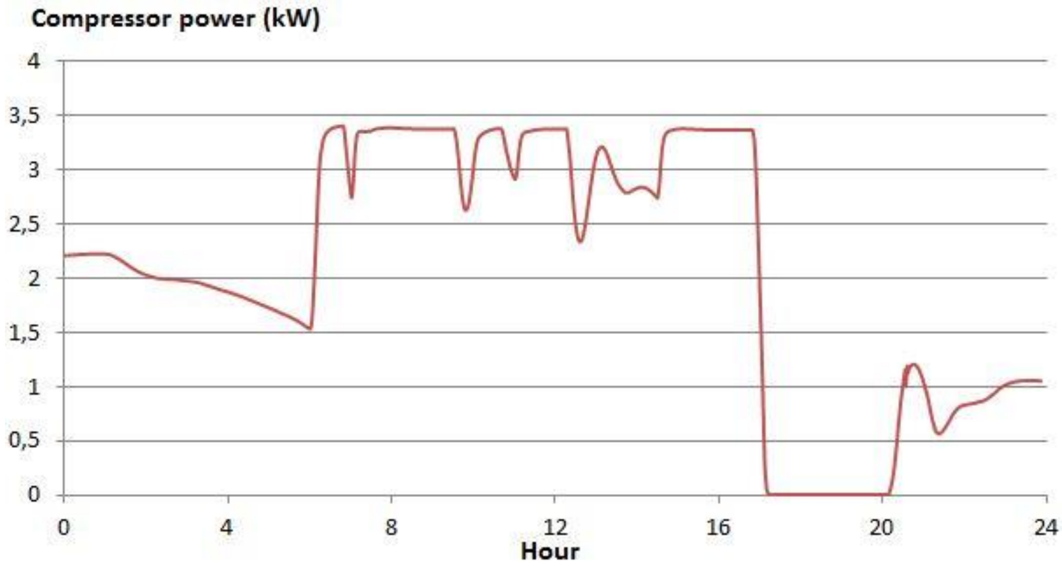

Figure 86 – Compressor power over a 24 hour period (2013-01-04)

Figure 86 shows the large variations in compressor power due to the night set-back. The curve follows the schedule implemented, as the compressor power increases at 6:00 and decreases at 16:30 (c.f. section 5.6.3).

7.3.3 INTERNAL HEAT GAINS

If the kindergarten was not an outdoors kindergarten, the occupants would be present to a higher extent. Or if the municipality of Trondheim determines to change the area of application for the building in the future, this will also lead to different internal heat gains, i.e. higher occupancy during operating hours. Hence the occupant schedules are all set to 7:00-16:30, to see how this affects the temperature and CO_2 concentration in the zone and thereby the zone heating and ventilation need. The resulting mean annual temperatures after a one-year simulation can be seen in table 31, and the net annual heating need in table 32.

Table 31 – Mean annual temperatures in reference model compared to model with higher internal gains

	Mean annual temperature (°C)	
Occupied zone	**Reference model**	**Model with higher internal gains**
1	21,0	21,3
2	21,5	22,0
4	21,3	21,7
5	21,8	22,2
7	21,1	21,3
8	21,5	21,7

Table 32 – Net annual heating need in reference model compared to model with higher internal gains

	Net annual heating need (kWh)	
	Reference model	**Model with higher internal gains**
Space heating	10 223	8 207
Heating coil	5 733	5 128
Domestic hot water	10 873	10 878
TOTAL	**26 829**	**24 213**

According to table 32, both the space heating need and the energy need for the heating coil have decreased. This result was expected, as higher internal gains contribute to a higher air temperature in the zone, as seen in table 31. Thus the exhaust air holds a higher temperature, in which more heat can be recovered by the heat recovery unit. As the heat recovery unit and the heating coil are controlled sequentially, this implies that the heat recovery unit always runs at full capacity before the heating coil starts to contribute. The reason for the reduced space heating need is the heat contribution from the occupants.

The higher internal heat gains could however bring about worse indoor environment if the temperatures exceed the maximum limit a large amount of the operating hours during the summer. Hence the thermal comfort in the worst zone, that being zone 5, is given in table 33.

Table 33 – Thermal comfort in the reference model compared to the model with higher internal gains

	No. of hours over a year	
Comfort category	**Reference model**	**Model with higher internal gains**
I - best	4 829	5 412
II – good	6 536	7 118
III – acceptable	8 684	8 637
IV - unacceptable	76	123

Even though a larger number of hours belong to the best category in the model with higher internal gains, the number of hours with unacceptable thermal climate still exceeds the reference model. This is due to high temperatures in the summer. The advantage with higher internal gains is on the other hand that the zone air temperatures during wintertime are now closer to the minimum limit. Have in mind that most of the zones were too cold during the winter (c.f. section 7.1.5).

In addition to lower net annual heating need, another advantage with higher occupancy in the building is a 6,2 % lower energy need for pump operation due to less circulation to the radiators and heating coil. Nevertheless, the energy need for pump operation was so small in the first place (c.f. table 18, section 7.1.1), that this benefit becomes insignificant in the big picture. Also, the solar collector still transfers the same amount of heat, but as the total heating need has decreased this solar energy contribution has increased to 6,6 % (previously 6,0 %). The electrical boiler's coverage has decreased from 12,5 % to 11,5 %, and the heat pump contribution is now 83,5 % compared to 81,5 % in the reference model.

7.4 UNDERGROUND THERMAL ENERGY STORAGE

This section presents the result after a one-year simulation of the model with UTES. The utilization of solar heat, delivered energy and impact on the ground temperature are among the subjects being analysed.

7.4.1 UTILIZATION OF SOLAR HEAT

Table 34 presents the utilized energy from the solar collector and the ground, and table 35 shows where the solar heat is directed, either to the ground or to the hot water tank. The values in table 34 are found directly in the tab *Systems energy*, while table 35 is made on the basis of the mean effect outputs in each of the heat exchangers over a year.

Table 34 – Annual utilized free energy in reference model compared to model with UTES

	Utilized free energy (kWh)	
	Reference model	**Model with UTES**
Solar heat	1 608	2 867
Ground heat	13 024	13 353

Table 35 – Where the solar heat is directed in the UTES model

Direction	**Heat collected from solar collector (kWh)**
Heat exchanger in hot water tank	1 733
Ground heat exchanger	1 365

As shown in table 34, the solar collector contributes to a higher degree when UTES is implemented than in the reference model. Table 35 shows that 44 % of this heat is directed to the ground heat exchanger. The solar heat stands for 10,2 % of the extracted heat from the ground. In comparison, the analysis performed by V. Trillat-Berdal (section 2.5) showed that 34 % of the extracted ground heat came from the solar collector.

When looking closely at the values in table 35, one can see that the total delivered solar heat differs from the utilized solar heat given in table 34. As the values in table 34 are found directly in the *Systems energy* tab, it is difficult to tell exactly how these values are calculated. The numbers in table 34 are on the other hand based on 172 000 time steps during a year, and can therefore be considered reliable.

The reason for the higher heat contribution from the solar collector in the UTES model is a higher temperature difference between the ingoing and outgoing temperatures to the collector. The mean mass flow rate is equal to the reference model. In order to examine more detailed how the solar heat is utilized, the mass flow rates through the tank heat exchanger and ground heat exchanger are plotted in figure 87.

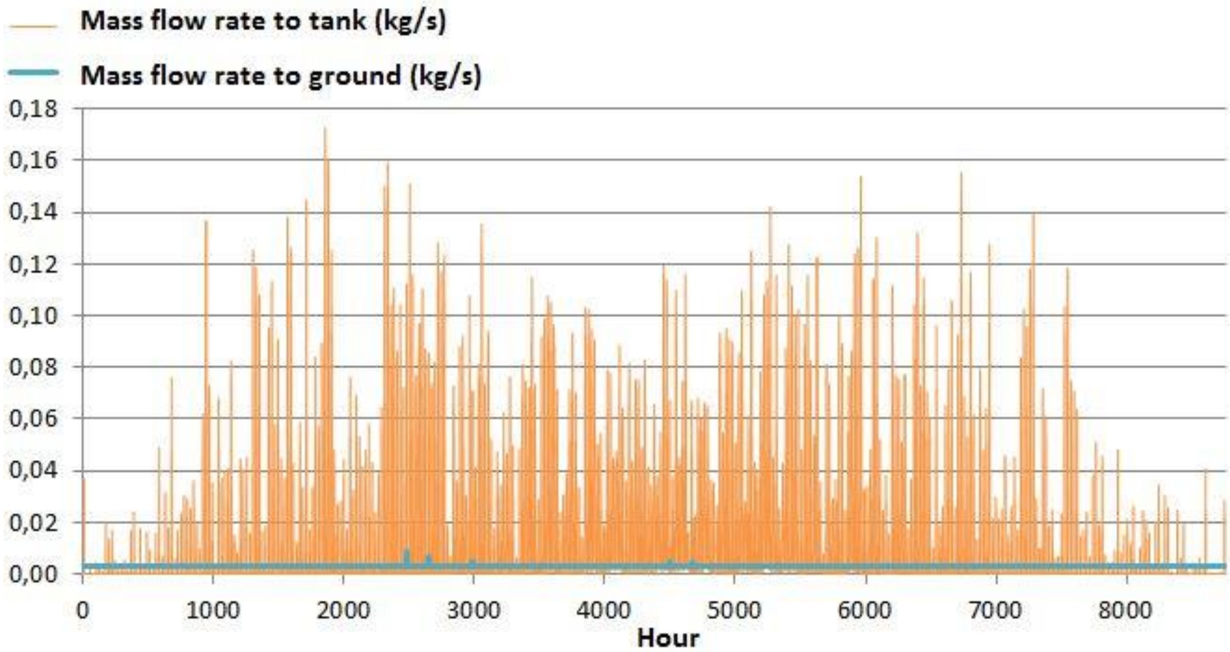

Figure 87 – Mass flow rate through hot water tank and ground heat exchanger – UTES model year 2013

The shape of the curve showing the mass flow rate to the hot water tank is identical to the one in the reference model (c.f. figure 68). The mass flow rate through the ground heat exchanger is on the other hand constant at 0,003 kg/s all year around. The fact that this mass flow rate is constant is incompatible with the intended control principle. The purpose is to direct the *excess* heat to the ground, more exactly when the outgoing temperature from the solar collector exceeds 62 °C.

The tab *Solar termal* shows that the maximum outgoing temperature is 61,1 °C, which indicates that the control principle does not work as intended. Judging by the result in figure 87 it seems like the circulation pump runs on a minimum level all year around, which ought to be further examined. Nevertheless, the heat directed from the solar collector to the ground heat exchanger will still contribute to heating of the ground to some extent. This will be verified in the following subsections.

7.4.2 IMPACT ON BRINE TEMPERATURE AND HEAT PUMP COP

To prove that heat is transferred from the solar fluid to the brine, the brine supply temperature to the evaporator is plotted throughout a year with and without UTES, see figure 88.

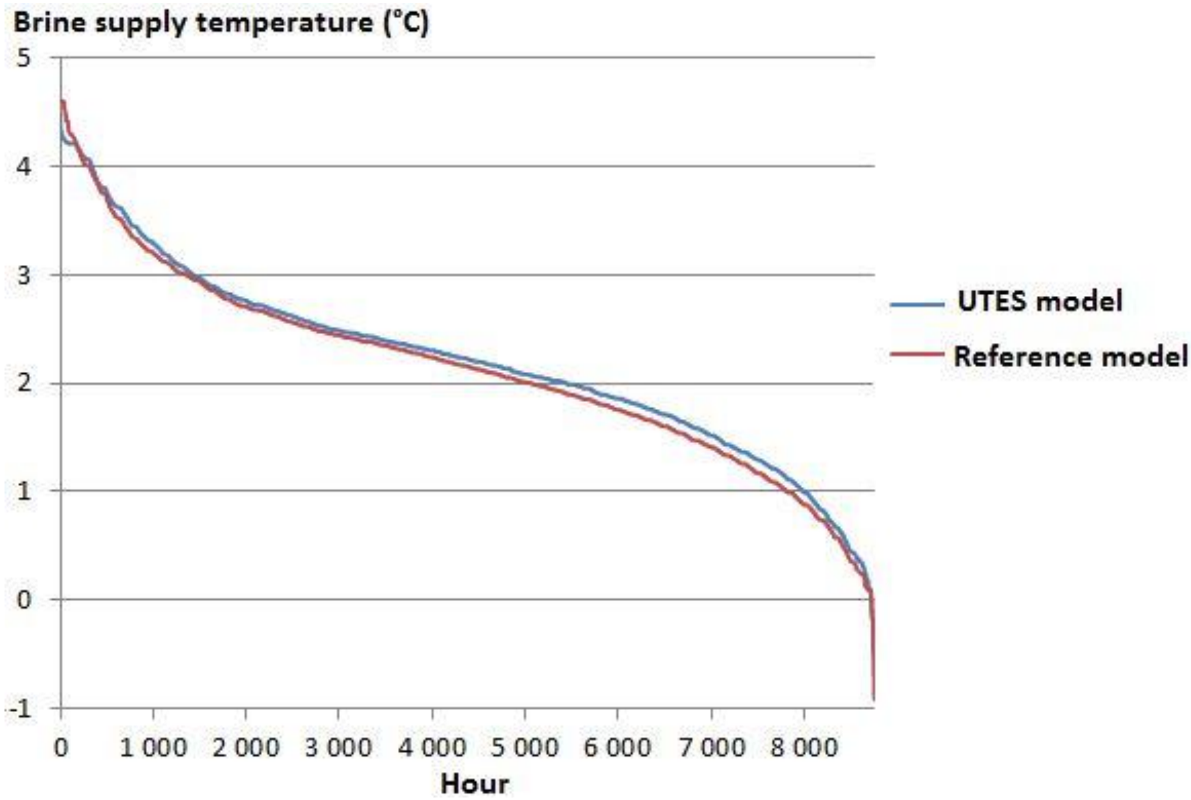

Figure 88 – Brine supply temperature throughout year 2013 with and without UTES

Figure 88 shows that the brine temperature in the UTES model is slightly higher than in the reference model. For the UTES model, the mean brine supply temperature to the heat pump evaporator is 2,21 °C and the mean return brine temperature is -0,25 °C. For the reference model, the same values are respectively 2,16 °C and -0,27 °C. The brine mass flow rate is exactly equal for the two models, which proves that the brine is indeed receiving heat from the solar fluid.

What can also be interpret from the brine temperatures is that more heat is being transferred to the working fluid in the heat pump, as the brine temperature difference over the heat pump evaporator is higher for the UTES model. This corresponds with table 34 showing a higher amount of ground heat utilized in the UTES model. Seeing as the net annual energy need is found to be exactly the same in the two models, the distribution of the heating need should also remain approximately the same. This subject needs to be further examined. Table 36 gives the distribution between the three energy sources in the UTES model and reference model after a one-year simulation.

Table 36 – Comparison of distribution of energy for heating purposes

	Delivered energy (kWh)	
	Reference model	**Model with UTES**
Electrical boiler	3 408	3 447
Heat pump	22 163	22 657
Solar collector	1 608	1 733
TOTAL	**27 179**	**27 837**

As table 36 shows, the delivered energy is actually higher for all three heat sources in the model with UTES. Even so, the total delivered heat is only 2,4 % higher in the UTES model. Hence one can say that the two models are quite similar. What remains to examine is whether the UTES model has a positive influence on the ground temperature.

7.4.3 IMPACT ON GROUND TEMPERATURE

As explained in section 2.3.4, the ground is not a perpetual heat source. The ground temperature will drop a few degrees over a long time period (25 years), and the only compensation is to transfer heat back to the ground. With this in mind, simulation over long time periods were attempted to carry out to see how the ground temperature develops. Unfortunately, simulations lasting longer than one year resulted in system breakdown. Nevertheless, the ground temperature during a one-year simulation is given in figure 89.

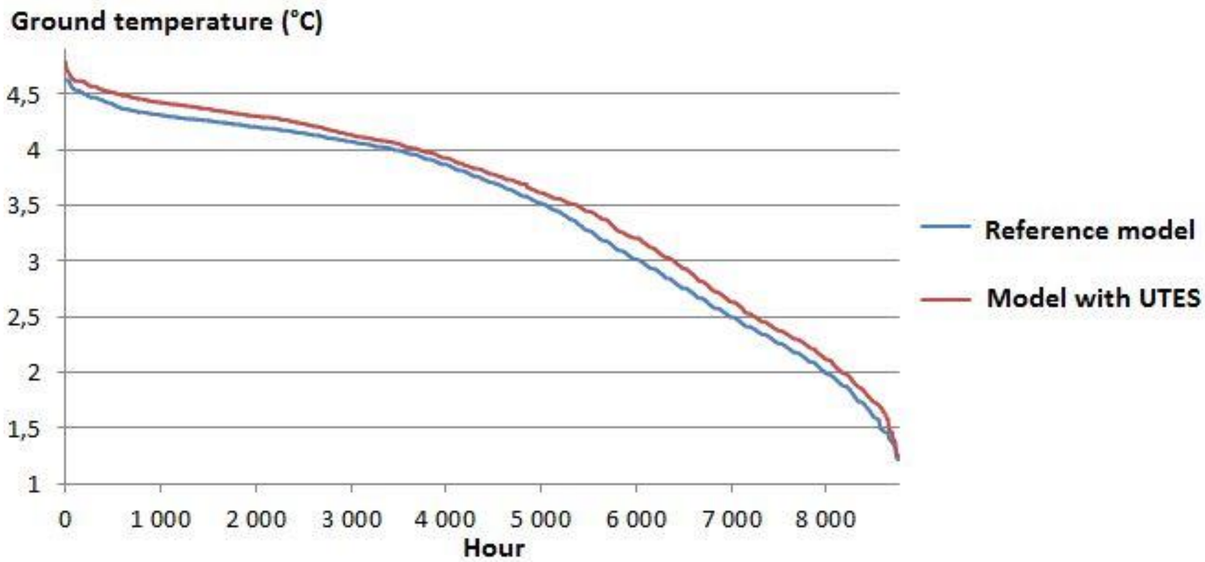

Figure 89 – Comparison of ground temperature with and without UTES

The mean ground temperature during a year in the reference model is 3,44 °C, while it is 3,52 °C in the model with UTES. Hence the plant with the UTES possibility has proven to be a better choice for preventing the ground temperature to drop over time. This implies that the heat pump COP will remain higher for a longer time period than in the model without UTES. To confirm this, further studies on the subject should involve simulations over longer time periods.

7.5 ECONOMIC ANALYSIS

Economy is one of the main deciding factors in a building project. Section 7.5.1 presents the economic aspect of the heat supply analysis, focusing mainly on global cost and pay-off period. A sensitivity analysis based on real interest rate and market prices are given in section 7.5.2.

As the kindergarten has not been in operation for a whole year yet (set into operation august 2013), the analysis is based on *expected* annual energy consumption according to simulations in IDA ICE. Especially table 18 and 20 that summarizes respectively the annual energy need and the delivered energy to the kindergarten can be used to determine whether or not the heat supply system is economically preferable. The energy consumption is quoted in table 37.

Table 37 – Energy consumption at Haukåsen kindergarten according to IDA ICE simulation year 2013

Type	**Energy consumption (kWh/year)**
Electrical boiler	3 408
Heat pump	8 749
Total heating need (from table 18)	26 829

Energy need for pump operation and controller equipment is neglected. The spreadsheet used for calculations can be found in appendix C.2. Methodology and collected data is reviewed in chapter 6.

7.5.1 GLOBAL COSTS AND PAY-OFF PERIOD

The resulting global cost and pay-off period is given in table 38. The reason that district heating is chosen as alternative heat supply is explained in section 6.2.2.

Table 38 – Global cost and pay-off period

Energy supply alternative	**Global cost (NOK)**	**Pay-off period (years)**
Solar collector, heat pump and electrical boiler	452 892	24,8
District heating	251 958	28,8

Table 38 shows that the existing heat supply alternative is the most favourable one when looking at the pay-off period, as it will be paid off within 25 years. When looking at the global cost however, the district heating alternative is cheaper, even with higher operational costs. The reason is the high investment cost in the existing heat plant, especially for the heat pump with energy wells. A discussion regarding the sizing of the heat pump was brought up in section 7.1.3, and this economic analysis confirms the importance of correct heat pump sizing. A further discussion about this subject will follow in chapter 8.

Remember that the alternatives are compared to heating with electricity, in order to calculate the savings and pay-off period (c.f. section 6.1). Both the fixed price and variable price related to electricity is taken into account. The maintenance costs are ignored, as they are assumed to be applicable for all alternatives. The calculations imply a real interest rate of 4 % and a calculation period of 20 years (c.f. section 6.2.1).

The passive house report performed in Belgium (c.f. section 2.1.1) resulted in a pay-off period of 29,9 years for the passive house and 12,3 years for the low-energy house. It is however difficult to compare the result with the kindergarten, as the focus in the report from Belgium is mainly on the differences between different *building types* and not heat supply systems.

Further reading contains different case studies regarding the heat supply alternatives, to see how changes in the plant affect the global cost and pay-off period.

Case 1: No solar collector
It is here assumed that the electrical boiler will cover the annual heat contribution from the solar collector, which means that an additional 1 608 kWh is added to the annual energy consumption for the boiler.

Resulting global cost is 427 224 NOK and a pay-off period of 22,8 years. In other words 25 668 NOK is saved in present value costs and 1,3 years in pay-off period. Seeing as the solar collector give the kindergarten a BREEAM certification labelled *Very good* (c.f. section 4.1) as the first building in Norway of its kind, investing in solar collector is still considered valuable for the municipality of Trondheim.

Case 2: Electrical boiler as back-up in district heating alternative
Even though district heating can supply heat at the highest temperature level needed (for tap water purposes), a back-up alternative is always wise to have if for example the substation should break down. Hence an electrical boiler is added to the global cost calculation for the district heating alternative. This gives a global cost of 326 205 NOK and a pay-off period of as much as 41,9 years. This alternative is in other words undesirable.

7.5.2 SENSITIVITY ANALYSIS

The global cost and pay-off period is very dependent on the real interest rate. Figure 90 shows how these factors vary with the real interest rate.

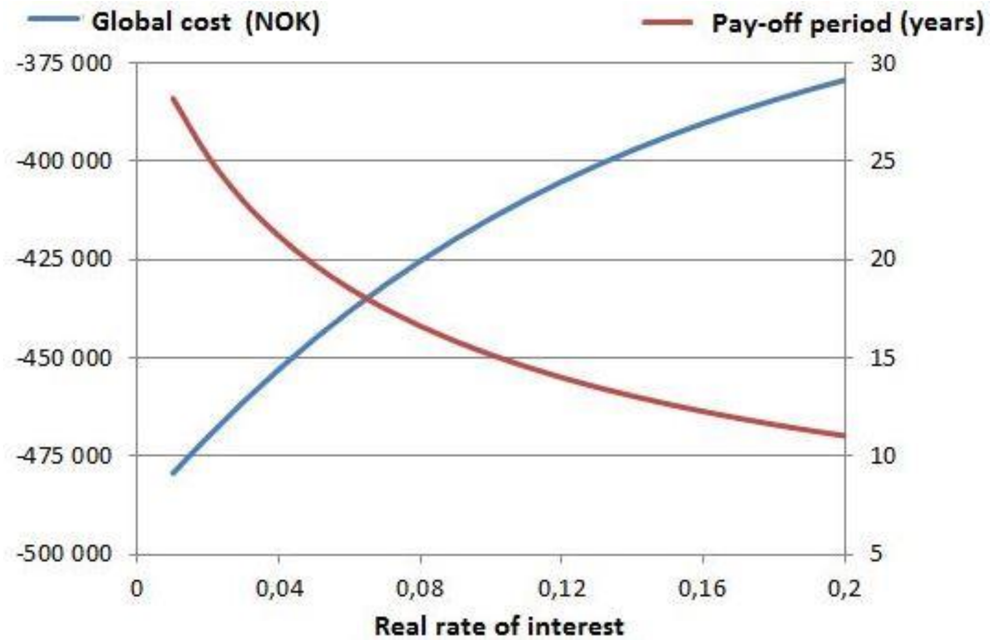

Figure 90 – Global cost and pay-off period according to real interest rate

The global cost on the left y-axis is shown with a negative sign to point out that it is *expenses*. Thus the global cost decreases with increasing real interest rate. The pay-off period will on the other hand decrease with higher real interest rate. In other words, changes in the real interest rate will not involve both the advantage of lower global cost and a lower pay-off period.

Another variable in the economic profibility is the electricity price and district heating price. In figure 91 the prices are plotted against the pay-off period to see how the two heat supply alternatives are affected.

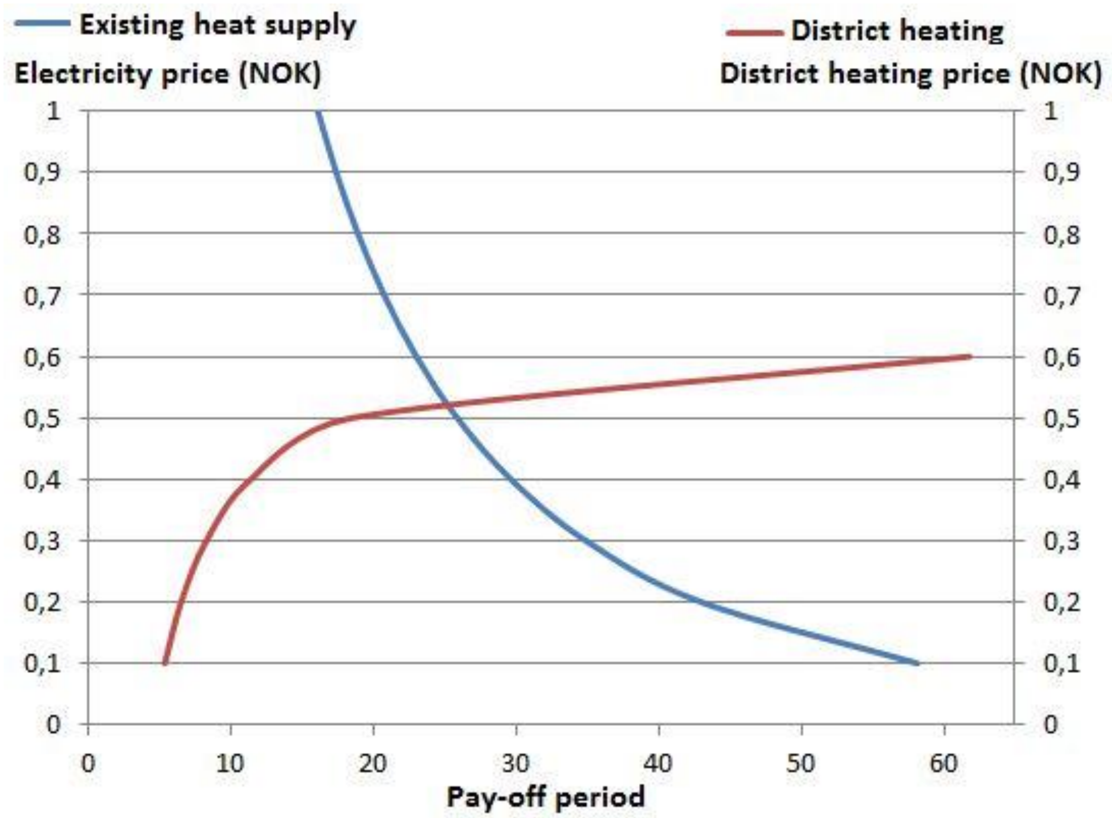

Figure 91 – Pay-off period for existing heat supply and district heating alternative based on respectively electricity price and district heating price

What is important to realize, is that the curve of the existing heat supply (blue curve) is only subject to changes in the *electricity price*. Likewise the district heating curve is only subject to changes in the *district heating price,* in order to compare the two alternatives.

A short pay-off period is favourable for both heat supply alternatives. There are four cases of interest from figure 91:

1) Electricity price and district heating price are both below 0,52 NOK
This makes the district heating alternative favourable.

2) Electricity price and district heating price are both above 0,52 NOK
In this case the existing heat supply is a better option.

3) Electricity price above 0,52 NOK and district heating price below 0,52 NOK
If the district heating price is below 0,5 NOK, this is the most favourable heat supply alternative, otherwise both alternatives are applicable.

4) Electricity price below 0,52 NOK and district heating price above 0,52 NOK

This is a somewhat unrealistic picture, as the district heating companies aim at a lower district heating price than the electricity price. Have in mind that the district heating price presented in section 6.2.3 are 0,05 NOK above the electricity price. However, when the fixed net electricity price is added, the district heating price remains the lowest. Nevertheless, case 4 is unwanted for both heat supply alternatives as they both have a high pay-off period.

7.5.3 SUMMARY ECONOMIC ANALYSIS

To summarize, the existing heat supply has a global cost of 452 892 NOK and will be paid off within 25 years with a real interest rate of 4 % and an electricity price of 0,53 NOK plus 2140 NOK in fixed costs. There are however uncertainties regarding the maintenance costs and residual value of the plant components, along with installation costs. If the solar collector had not been installed, 25 667 NOK would have been saved in global costs and 1,3 years in pay-off period. Nevertheless the solar collector gives the building the BREEAM label *Very good,* and is therefore considered a valuable investment.

The sensitivity analysis shows that both the global cost and the pay-off period are highly dependent on the real interest rate, but in opposite favour of each other. An increase in real interest rate gives lower global costs and a higher pay-off period, and vice versa. The prices of electricity and district heating will also influence the pay-off period. In order to justify the existing heat supply, the best case is that both prices remain above 0,52 NOK.

8. DISCUSSION

This chapter looks into the significance of the results and to what degree the results can be useful. IDA ICE is evaluated as a simulation tool, and the matter of heat pump sizing and ventilation strategies in passive houses are discussed.

8.1 IDA ICE'S ABILITY AS SIMULATION TOOL

IDA ICE is an advanced simulation tool with a wide range of application. Seeing as all the results are based on IDA ICE simulations, it is of interest to discuss the capability and reliability of IDA ICE as a simulation tool.

The subject of the programs reliability was questioned several times when the results was analysed. Looking at the reference model, most of the IDA ICE results were in accordance with existing theory or measuring data. This accounts for the net energy need and heating loads, which proved to agree with both the Norwegian passive house criteria and the existing theory regarding passive houses, presented in the literature review. Especially the coverage of heating loads between the electrical boiler and the heat pump was exactly as expected. The fact that the effect output from the heat pump was lowered to ensure more stable working conditions, while still covering 81,5 % of the annual heating need, is remarkable. This distribution of heating load is considered a smart decision and requires a very advanced simulation tool.

On the other hand, what differed from the expectations was the solar collector contribution, which was much less than anticipated according to the theoretical potential. In addition the default control strategy of the solar collector circuit did not work optimally and as intended, meaning that the circulation pump was switched on even when the temperature in the solar collector was lower than in the hot water tank. Surely, this may be the reason for the unexpected result. Nevertheless, the reliability of IDA ICE was questioned at this stage. An additional reason is the unexpected result when the area and tilting angle of the solar collector were changed, which gave incompatible results with the existing theory reviewed in chapter 2.

All things considered, IDA ICE has the capability of performing advanced simulations, but the reliability is dependent on the user's knowledge regarding the input data. The user must be certain about the input data and the range of use. A great advantage is on the other hand the possibility to define custom-made macros. This regards both the heat plant and ventilation signal as was performed in this project, but also for the AHU and to have more detailed zone control.

8.2 DIMENSIONING HEAT PUMP CAPACITY

The heat plant analysis showed that the heat pump covers only 24 % of the heating load, but this implies as much as 81,5 % of the net annual heating need. This indicates an oversized capacity in the heat pump, as 81,5 % coverage in heating need normally would imply around 50-70 % effect load coverage. Having the heat pump work at part load a large amount of the time will decrease the performance of the compressor, (c.f. section 2.3.2) and cause earlier wear out. Given that the investment cost of heat pump plus energy wells constitute 75 % of the total heat plant investment (c.f. section 6.2.2), it is important to size the heat pump's capacity correctly.

Perhaps one or two bore holes would be sufficient, and a smaller heat pump capacity would still be enough to cover the same share of the heating load. Thus the importance of correct sized heat pump influences both the heat pump's lifespan and exceedingly the heat plant investment cost.

8.3 VENTILATION STRATEGIES IN PASSIVE HOUSES

The indoor environment analysis showed that the occupied zones have too low air temperatures during wintertime and too high temperatures during summertime. This brings about a discussion regarding the ventilation strategies in passive houses.

Because of the passive house criteria of maximum 65 kWh/m^2 net annual energy need, it is necessary to have very energy efficient ventilation of the building. Ventilation control based on occupancy contributes to meet this demand. But as the building analysed in this project is an outdoors kindergarten, it is left deserted a large amount of the day. Consequently the air flow rate is kept at a minimum most of the day, when there are no occupants present. This makes it difficult to avoid high zone air temperatures during summertime. A measure to avoid high temperatures could be to increase the minimum air flow rate during summertime. The question is whether energy efficient ventilation is to be desired or higher thermal comfort.

8.4 MASTER THESIS SIGNIFICANCE

The master thesis shows that utilization of solar collectors combined with a GSHP is a possible contribution for increasing the utilization of renewable energy resources and reducing the specific electricity need in Norway. Thus this is a step in the right direction for reaching the EU's goals for year 2020.

Whether or not the solar collector was a valuable investment was questioned to begin with. The economic analysis showed that if the solar collector had not been installed, 25 667 NOK would have been saved in global costs and 1,3 years in pay-off period. Nevertheless, the solar collector gives the building the BREEAM label *Very good,* and is therefore considered a valuable investment for the municipality of Trondheim.

Also the matter of seasonal thermal energy storage was examined. Even though the building's net annual heating need remains the same, one can prevent decreasing of the heat pump COP over time. This is done by warming up the surrounding ground around the bore holes and thus a higher evaporation temperature is provided. Only a one-year simulation was possible to carry out in this project, but it still showed a 0,08 °C increase in ground temperature. Further studies ought to be carried out on this field, looking at longer time periods and taking the economic costs versus profitability into consideration.

9. CONCLUSION

A reference model of Haukåsen kindergarten has been simulated in IDA ICE. The construction is built to passive house standard and the heat supply system consists of solar collectors combined with a ground source heat pump. When modelling the building, the aim was to make a reference model as close as possible to the real kindergarten. This chapter summarizes the impact of the results and analysis.

The net annual energy need is calculated to be 46 702 kWh, whereas the heating need stands for 26 829 kWh per year. Comparing this to the measuring data, the actual energy use is 22 % higher than the estimate from IDA ICE. The passive house requirement is maximum 65 kWh/m^2 annual energy need, which means that the specific energy need of 57,4 kWh/m^2 is within this limit. The annual heating need of 33,1 kWh/m^2 does however not fulfil the passive house demand of maximum 25 kWh/m^2.

Judging by IDA ICE results, the installed radiators at Haukåsen kindergarten have oversized capacity, while the heating coil and domestic hot water load is dimensioned with too low heating load. Plotting the heating coil effect load against the ambient air temperature showed that the 18 kW installed capacity of the heating coil is exceeded when the outdoor temperature is -13 °C or lower. As Trondheim has a dimensioning outdoor temperature of -20 °C, one would expect a lower crossing point.

The coverage of the heating need showed that the heat pump covers the base load up to 9,9 kW and the electrical boiler covers the peak load up to 41,7 kW, as intended. In percentages, the heat pump covered 81,5 %, the boiler 12,5 % and the solar collector 6 % of the net annual heating need. Even though the heat pump covered only 24 % of the heating load, it still implied as much as 81,5 % of the net annual heating need. This indicates an oversized capacity in the heat pump, which may cause earlier wear out of the compressor due to unwanted part-load operation most of the time.

The solar collector contribution of 1 608 kWh/year was expected to be higher, as the theoretical efficiency would imply an expected contribution of 4 241 kWh/year. Either doubling of the collector area or optimization of the tilting angle gave noticeably higher contribution. Neither did changing of the shape factor for the hot water tank. The temperature levels to and from the solar collector were identical independent of the shape of the hot water tank. This result is inconsistent with the stratification concept, as one would expect the tank shape to influence the temperature layers to a higher degree. Thus the control strategy of the solar collector circuit is questioned, and ought to be further studied.

IDA ICE's capability as an advanced simulation tool satisfied the expectations when it comes to designing macros. Especially the ventilation control macros based on occupancy all worked as intended, which led to more energy efficient ventilation.

1 388 kWh delivered energy is saved each year by having occupancy controlled ventilation. The implementation of night set-back also contributes to decrease the net annual heating need, with a reduction of 5 969 kWh/year. Nevertheless, the low annual energy need is at the sacrifice of the indoor environment. The night set-back implied too low zone air temperatures during wintertime, while the occupancy controlled ventilation led to excess temperature during summertime. 76 hours a year was characterized with an unacceptable comfort category, based on PMV and PPD values.

Another macro being tested was change of the IDA ICE plant sketch in order to store heat seasonally in bore holes. The result showed 78 % increase in solar heat contribution and 0,08 °C increase in ground temperature over a year. This indicates that the heat pump COP will remain high for a longer time period than in the model without UTES. To confirm this, further studies on the subject should involve simulations over longer time periods.

The economic analysis resulted in that the existing heat supply has a global cost of 452 892 NOK and will be paid off within 25 years with a real interest rate of 4 %. . If the solar collector had not been installed, 25 667 NOK would have been saved in global costs and 1,3 years in pay-off period. Nevertheless the solar collector gives the building the BREEAM label *Very good,* and is therefore considered a valuable investment. The sensitivity analysis showed that an increase in real interest rate gives lower global costs and a higher pay-off period, and vice versa. In order to justify the existing heat supply, both the electricity price and the district heating price should remain above 0,52 NOK.

10. FURTHER WORK

Subjects that ought to be further studied are primarily matters in IDA ICE. First, the control strategy of the solar collector circuit should be further developed, in order to take the temperature in the hot water tank into consideration. This makes sequential control of the solar collector circuit possible to carry out. Furthermore this may give answers to why the solar collector area and tilting angle have so little impact on the collected heat.

Secondly, an optimal solar collector control will improve the UTES model, as presumably more heat can be stored in the ground. Another improvement regarding the UTES model is to develop a connection for dumping excess heat from the building back to the ground. Hence the building is cooled during summertime. Also, simulations over longer time periods ought to be carried out, to examine how the ground temperature is affected.

Third, matters regarding climate files should also be further studied, i.e. why the heating need for exactly the same building is higher in Oslo than in Trondheim, when Oslo has a higher mean annual air temperature. Why the need for hot water differs between the locations should also be looked into. Furthermore the subject of temperature stratification in the hot water tank ought to be examined, as the shape factor did not have an effect on the temperature layers at all.

Last, installation of measuring instruments in the kindergarten will give the opportunity to verify calculated energy need versus actual energy need. This also implies that more detailed plant component analysis can be performed.

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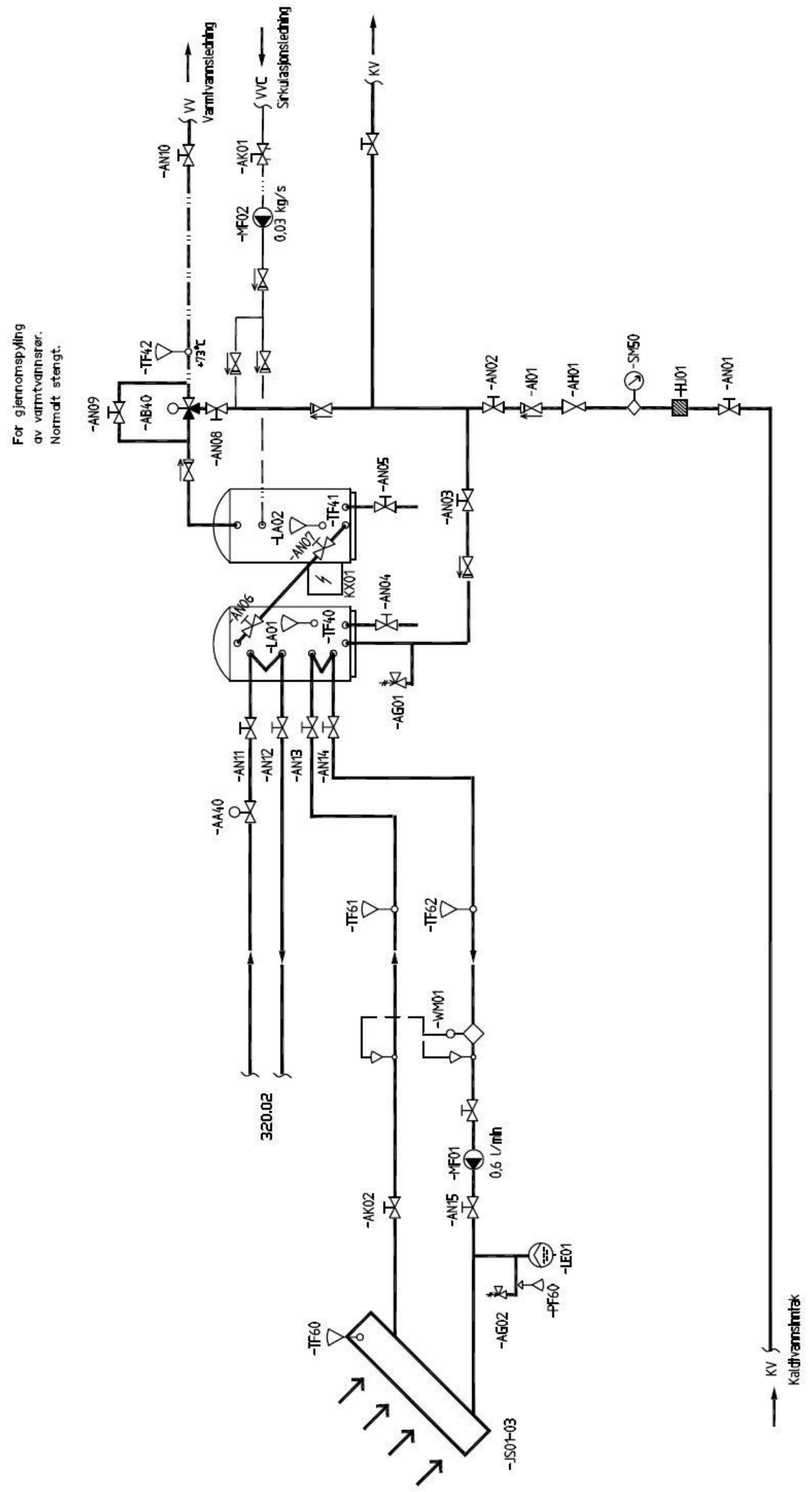
APPENDICES

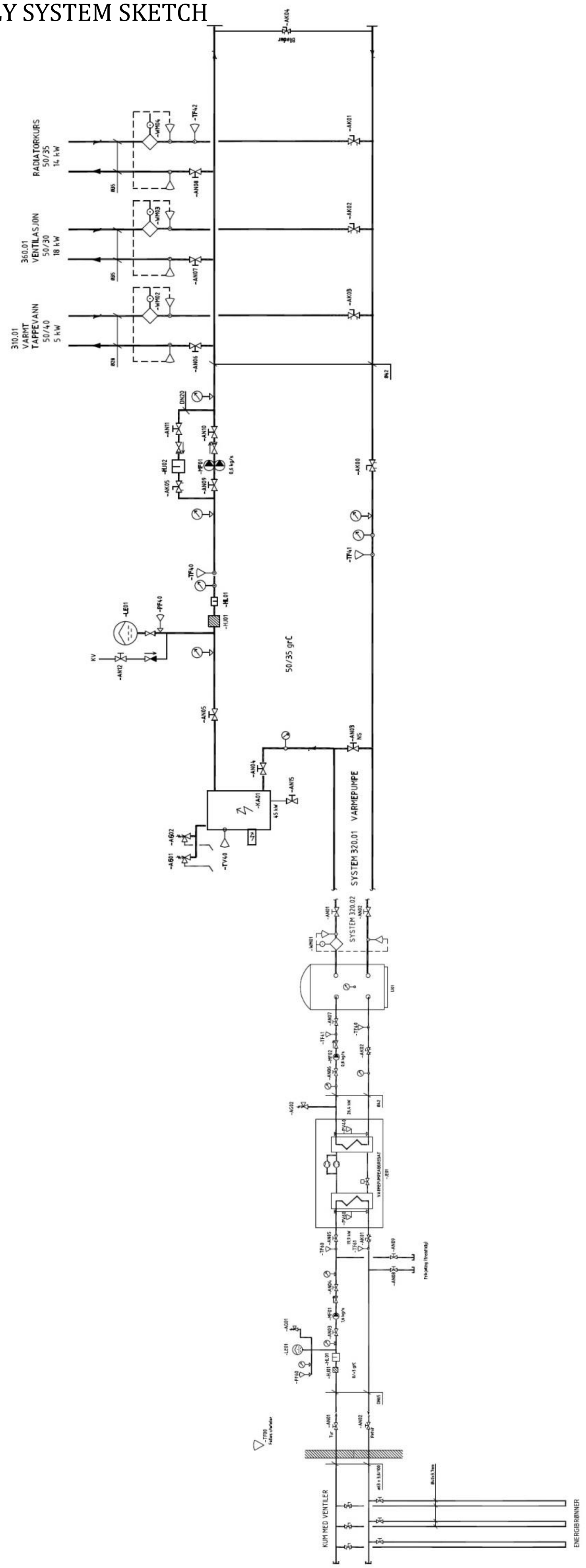
A. CASE BUILDING: HAUKÅSEN KINDERGARTEN

All sketches are received from Rambøll.

A.1 SANITARY SYSTEM SKETCH



A.2 HEAT SUPPLY SYSTEM SKETCH



B. INPUT DATA

B.1 IDA ICE INPUT DATA REPORT

Zones

Name	Group	Floor height, m	Room height, m	Floor area, m2	Heat setp., °C	Cool setp., °C	AHU	System	Supply air, L/sm2	Return air, L/sm2	Occup., no./m2	Lights, W/m2	Equipment, W/m2	Ext win. area, m2	Occup. schedule	Light schedule	Equipm. schedule
Zone 1		0	3.33	173.6	n.a.	n.a.	Air Handling	CAV custom	2.48	2.48	0.1037	6	0	20.65	Young childr	Schedule 7-16:30	
Zone 3		0	6.63	35.8	22	25	No central A	n.a.			0	0	0	0.9			
Zone 4		3.33	3.3	174	n.a.	n.a.	Air Handling	CAV custom	1.95	1.95	0.09195	5.999	0.1149	32.21	<mixed>	Schedule 7-1	© Always on
Zone 6		6.63	2.6	182.2	19	25	Air Handling	CAV custom	0.69	0.69	0	6	2.058	0		Schedule 7-8	© Always on
Zone 7		0	6.63	69.8	n.a.	n.a.	Air Handling	Zone 7 and 8	4.58	4.58	0.07163	6	0	22.29	Older childre	Schedule 7-16:30	
Zone 8		3.33	3.3	60.6	n.a.	n.a.	Air Handling	Zone 7 and 8	4.58	4.58	0.1155	6	0	10.05	Older childre	Schedule 7-16:30	
Zone 5		3.33	3.3	58.6	n.a.	n.a.	Air Handling	Zone 5 contr	4.74	4.74	0.1195	6	0	13.06	Older childre	Schedule 7-16:30	
Zone 2		0	3.33	120.1	n.a.	n.a.	Air Handling	Zone 2 contr	4.7	4.7	0.1499	6	0	21.21	Young childr	Schedule 7-16:30	

Zone totals

Name	Group	Floor height, m	Zone multiplier, M	M*Area, m2	M*Volume, m3	M*Walls above gr., m2	M*Walls below gr., m2	M*Ext win. area, m2	M*Ext door area, m2	M*Roof area, m2	M*Ground area, m2	M*floor to amb., m2	M*Tot env. area, m2	M*UAtot, W/K	M*UAwall above gr., W/K	M*UAwall below gr., W/K	M*UAext. window, W/K	M*UA door, W/K	M*UA roof, W/K
Zone 1		0	1	173.6	693.9	131.07	0	20.65	5.88	0	208.38	0	365.98	47.964	11.842	0	17.102	0.53126	0
Zone 3		0	1	35.8	294.5	132.4	0	0.9	3.57	36.54	33.588	0	207	21.678	11.963	0	0.74538	0.32255	3.3043
Zone 4		3.33	1	174	691.8	122.01	0	32.21	2.31	0.28739	0	0	156.82	40.442	11.024	0	26.676	0.20871	0.025989
Zone 6		6.63	1	182.2	732.9	66.931	0	0	0	432.22	0	0	499.15	53.143	6.0472	0	0	0	39.085
Zone 7		0	1	69.8	372.2	50.494	0	22.29	0	0	56.136	0	128.92	28.491	4.5621	0	18.461	0	0
Zone 8		3.33	1	60.6	210.2	23.491	0	10.05	2.31	0	0	0	35.851	11.232	2.1224	0	8.3234	0.20871	0
Zone 5		3.33	1	58.6	212.6	23.463	0	13.06	0	0.16813	0	0	36.691	13.541	2.1199	0	10.816	0	0.015204
Zone 2		0	1	120.1	435.1	49.644	0	21.21	2.31	0	130.65	0	203.82	33.439	4.4854	0	17.566	0.20871	0
TOTAL			8	874.7	3643.2	599.503	0	120.37	16.38	469.21552	428.754	0	1634.232	249.93	54.166	0	99.68978	1.47994	42.430493

Name	Group	M*UAfloor to ground, W/K	M*UAfloor to amb., W/K	M*Thermal bridges, W/K	M*Supply air, L/s	M*Return air, L/s	M*Occupants, items	M*Lights, W	M*Equipment, W
Zone 1		12.598	0	5.8901	430.5	430.5	18	1041.6	0
Zone 3		2.0305	0	3.3123			0	0	0
Zone 4		0	0	2.5075	339.3	339.3	16	1043.8	20
Zone 6		0	0	8.0101	125.7	125.7	0	1093.2	375
Zone 7		3.3937	0	2.0749	319.7	319.7	5	418.8	0
Zone 8		0	0	0.57699	277.5	277.5	7	363.59	0
Zone 5		0	0	0.5894	277.8	277.8	7	351.59	0
Zone 2		7.8985	0	3.2802	564.5	564.5	18	720.63	0
TOTAL		25.9207	0	26.24149	2335	2335	71	5033.21	395

Zone setpoints

Name	Group	Setpoints	Heat setp., °C	Cool setp., °C	Min VAV air return, L/(s.m2)	Max VAV air return, L/(s.m2)	Min VAV air supply, L/(s.m2)	Max VAV air supply, L/(s.m2)	Min humidity, %	Max humidity, %	Min CO2, ppm(vol)	Max CO2, ppm(vol)	Min light, lx	Max light, lx	Min pressure diff, Pa	Max pressure diff, Pa	Var. heat setpoint	Var. cool setpoint
Zone 1		[local for zc	n.a.	n.a.	0.56	2.48	0.56	2.48	20	80	700	1000	100	10000	-20	-10	Minimum t	Maximum temperature in zone
Zone 3		[local for zc	22	25	0.56	0.56	0.56	0.56	20	80	700	1000	100	10000	-20	-10	<value not	<value not set>
Zone 4		[local for zc	n.a.	n.a.	0.56	1.95	0.56	1.95	20	80	700	1000	100	10000	-20	-10	Minimum t	Maximum temperature in zone
Zone 6		[local for zc	19	25	0.56	0.69	0.56	0.69	20	80	700	1000	100	10000	-20	-10	<value not	<value not set>
Zone 7		[local for zc	n.a.	n.a.	0.56	4.58	0.56	4.58	20	80	700	1000	100	10000	-20	-10	Minimum t	Maximum temperature in zone
Zone 8		[local for zc	n.a.	n.a.	0.56	4.58	0.56	4.58	20	80	700	1000	100	10000	-20	-10	Minimum t	Maximum temperature in zone
Zone 5		[local for zc	n.a.	n.a.	0.56	4.74	0.56	4.74	20	80	700	1000	100	10000	-20	-10	Minimum t	Maximum temperature in zone
Zone 2		[local for zc	n.a.	n.a.	0.56	4.7	0.56	4.7	20	80	700	1000	100	10000	-20	-10	Minimum t	Maximum temperature in zone

Surfaces

Name	Group	Type	Wetted area, m2	Connected to	Azimuth, Deg	Slope, Deg	Construction	U, W/m2°K	Thickness, m	Layer material	Layer thickness, m	Layer material	Layer thickness, m	Layer material	Layer thickness, m	Layer material	Layer thickness, m
Zone 1.Floor		Ext. floor	208.4	Ground		0	[Default] ©	0.06904	0.8	© Floor coa	0.05	© Light insı	0.5	© Concrete	0.25		
Zone 1.Ceiling		Int ceiling	208.4	Zone 4		180	[Default] ©	0.347	0.234	© Wood	0.16	© Light insı	0.05	© Chip boa	0.022	© Floor coa	0.002
Zone 1.Wall 1		Ext. wall	27.75	Building bc	8.7	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light insı	0.3	© Frames c	0.05	© Wood	0.1
Zone 1.Wall 2		Ext. wall	52.34	Building bc	87.365227	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light insı	0.3	© Frames c	0.05	© Wood	0.1
Zone 1.Wall 3		Ext. wall	1.252	Building bc	109.74094	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light insı	0.3	© Frames c	0.05	© Wood	0.1
Zone 1.Wall 4		Int. wall	1.172	Zone 2	278.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 1.Wall 5		Int. wall	38.99	Zone 2	188.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 1.Wall 6		Ext. wall	49.73	Building bc	278.7	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light insı	0.3	© Frames c	0.05	© Wood	0.1
Zone 3.Floor		Ext. floor	33.59	Ground		0	[Default] ©	0.06904	0.8	© Floor coa	0.05	© Light insı	0.5	© Concrete	0.25		
Zone 3.Ceiling.r1@Buil		Roof	17.03	Building bc	98.345107	156.58199	[Default] ©	0.09043	0.515	© Gypsum	0.005	© Light insı	0.35	© Wood	0.16		
Zone 3.Ceiling.r2@Buil		Roof	19.51	Building bc	278.35241	156.99526	[Default] ©	0.09043	0.515	© Gypsum	0.005	© Light insı	0.35	© Wood	0.16		
Zone 3.Wall 1		Int. wall	83.41	Zone 7; Zon	8.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 3.Wall 2		99% Ext. wa	43.39	Building bc	111.46557	90	<mixed>	<mixed>	0.55 / 0.55								
Zone 3.Wall 3		99% Ext. wa	77.49	Building bc	208.36876	90	<mixed>	<mixed>	0.55 / 0.55								
Zone 3.Wall 4		Ext. wall	11.52	Building bc	278.7	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light insı	0.3	© Frames c	0.05	© Wood	0.1
Zone 4.Floor		Int. floor	209.6	Zone 1		0	[Default] ©	0.347	0.234	© Floor coa	0.002	© Chip boa	0.022	© Light insı	0.05	© Wood	0.16
Zone 4.Ceiling.r1@Buil		Roof	0.2874	Building bc	98.341765	156.55043	[Default] ©	0.09043	0.515	© Gypsum	0.005	© Light insı	0.35	© Wood	0.16		
Zone 4.Ceiling.ceiling		Int ceiling	209.4	Zone 6		180	[Default] ©	0.347	0.234	© Wood	0.16	© Light insı	0.05	© Chip boa	0.022	© Floor coa	0.002
Zone 4.Wall 1		Ext. wall	21.35	Building bc	8.7	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light insı	0.3	© Frames c	0.05	© Wood	0.1
Zone 4.Wall 2		Ext. wall	49.79	Building bc	87.365227	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light insı	0.3	© Frames c	0.05	© Wood	0.1
Zone 4.Wall 3		Ext. wall	0.3655	Building bc	109.71711	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light insı	0.3	© Frames c	0.05	© Wood	0.1
Zone 4.Wall 4		Int. wall	40.85	Zone 5; Zon	188.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 4.Wall 5		Ext. wall	49.49	Building bc	278.7	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light insı	0.3	© Frames c	0.05	© Wood	0.1
Zone 6.Floor		Int. floor	394.9	Zone 5; Zone 8; Zone 7;		0	[Default] ©	0.347	0.234	© Floor coa	0.002	© Chip boa	0.022	© Light insı	0.05	© Wood	0.16
Zone 6.Ceiling.r1@Buil		Roof	215.6	Building bc	98.340512	156.5863	[Default] ©	0.09043	0.515	© Gypsum	0.005	© Light insı	0.35	© Wood	0.16		
Zone 6.Ceiling.r2@Buil		Roof	216.7	Building bc	278.34037	156.99606	[Default] ©	0.09043	0.515	© Gypsum	0.005	© Light insı	0.35	© Wood	0.16		
Zone 6.Wall 1		Ext. wall	18.94	Building bc	8.6690025	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light insı	0.3	© Frames c	0.05	© Wood	0.1
Zone 6.Wall 2		Ext. wall	12.52	Building bc	87.364273	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light insı	0.3	© Frames c	0.05	© Wood	0.1
Zone 6.Wall 3		96% Ext. wa	10.73	Building bc	110.13263	90	<mixed>	<mixed>	0.55 / 0.55								
Zone 6.Wall 4		Int. wall	19.29	Zone 3	188.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 6.Wall 5		96% Ext. wa	24.74	Building bc	278.7	90	<mixed>	<mixed>	0.55 / 0.55								
Zone 7.Floor		Ext. floor	56.14	Ground		0	[Default] ©	0.06904	0.8	© Floor coa	0.05	© Light insı	0.5	© Concrete	0.25		
Zone 7.Ceiling		Int ceiling	56.14	Zone 6		180	[Default] ©	0.347	0.234	© Wood	0.16	© Light insı	0.05	© Chip boa	0.022	© Floor coa	0.002
Zone 7.Wall 1		Int. wall	71.95	Zone 2; Zon	8.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		

Zone 7.Wall 2	Ext. wall	26.99	Building bc	109.70974	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light ins	0.3	© Frames c	0.05	© Wood	0.1
Zone 7.Wall 3	Int. wall	64.93	Zone 3	188.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 7.Wall 4	Ext. wall	23.5	Building bc	278.7	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light ins	0.3	© Frames c	0.05	© Wood	0.1
Zone 8.Floor	Int. floor	63.71	Zone 2		0	[Default] ©	0.347	0.234	© Floor coa	0.002	© Chip boa	0.022	© Light ins	0.05	© Wood	0.16
Zone 8.Ceiling	Int ceiling	63.71	Zone 6		180	[Default] ©	0.347	0.234	© Wood	0.16	© Light ins	0.05	© Chip boa	0.022	© Floor coa	0.002
Zone 8.Wall 1	Int. wall	17.25	Zone 4	8.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 8.Wall 2	Int. wall	35.85	Zone 5	98.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 8.Wall 3	Int. wall	19.35	Zone 7	188.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 8.Wall 4	Ext. wall	23.49	Building bc	278.7	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light ins	0.3	© Frames c	0.05	© Wood	0.1
Zone 5.Floor	Int. floor	64.43	Zone 2		0	[Default] ©	0.347	0.234	© Floor coa	0.002	© Chip boa	0.022	© Light ins	0.05	© Wood	0.16
Zone 5.Ceiling.r1@Buil	Roof	0.1681	Building bc	98.330057	156.5515	[Default] ©	0.09043	0.515	© Gypsum	0.005	© Light ins	0.35	© Wood	0.16		
Zone 5.Ceiling.ceiling	Int ceiling	64.27	Zone 6		180	[Default] ©	0.347	0.234	© Wood	0.16	© Light ins	0.05	© Chip boa	0.022	© Floor coa	0.002
Zone 5.Wall 1	Int. wall	23.04	Zone 4	8.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 5.Wall 2	Ext. wall	23.39	Building bc	109.70636	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light ins	0.3	© Frames c	0.05	© Wood	0.1
Zone 5.Wall 3	Int. wall	16.08	Zone 7	188.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 5.Wall 4	Int. wall	35.85	Zone 8	278.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 2.Floor	Ext. floor	130.7	Ground		0	[Default] ©	0.06904	0.8	© Floor coa	0.05	© Light ins	0.5	© Concrete	0.25		
Zone 2.Ceiling	Int ceiling	130.7	Zone 5; Zone 8		180	[Default] ©	0.347	0.234	© Wood	0.16	© Light ins	0.05	© Chip boa	0.022	© Floor coa	0.002
Zone 2.Wall 1	Int. wall	38.5	Zone 1	8.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 2.Wall 2	Int. wall	1.568	Zone 1	98.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 2.Wall 3	Int. wall	0.3111	None	14.22754	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 2.Wall 4	Ext. wall	25.15	Building bc	111.30252	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light ins	0.3	© Frames c	0.05	© Wood	0.1
Zone 2.Wall 5	Int. wall	35.11	Zone 7	188.7	90	[Default] ©	0.938	0.14	© Gypsum	0.02	© Wood	0.1	© Gypsum	0.02		
Zone 2.Wall 6	Ext. wall	24.49	Building bc	278.7	90	[Default] ©	0.09035	0.55	© Wood	0.1	© Light ins	0.3	© Frames c	0.05	© Wood	0.1

Windows

Name	Group	Zone	Face	Sill height from ground, m	Sill height from floor, m	Azimuth, Deg	Slope, Deg	Area, m2	Glazing	g (SHGC)	T	Glazing U, W/m2°K	Frame fract., 0-1	Frame U, W/m2°K	Win total U, W/m2°K	Int. shading	Control	Schedule	Ext. shading	Opening control	Opening schedule
Building body.f1.DetWi		Zone 3	Building bc	4.28	4.28	199.29913	90	0.9	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 7	Building bc	0.31	0.31	270	90	6.16	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 7	Building bc	0.96	0.96	270	90	0.9	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 2	Building bc	1.07	1.07	270	90	3.52	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 2	Building bc	0.38	0.38	270	90	5.72	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 2	Building bc	0.62	0.62	270	90	0.9	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 1	Building bc	0.45	0.45	270	90	3.52	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 1	Building bc	0.42	0.42	270	90	5.72	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 1	Building bc	1.72	1.72	270	90	0.81	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 7	Building bc	4.54	4.54	270	90	0.81	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 8	Building bc	4.27	0.94	270	90	0.81	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 7	Building bc	4.24	4.24	270	90	4.68	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 8	Building bc	3.84	0.51	270	90	5.72	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 8	Building bc	4.35	1.02	270	90	3.52	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 4	Building bc	3.77	0.44	270	90	3.52	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 4	Building bc	3.78	0.45	270	90	5.72	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f2.DetWi		Zone 4	Building bc	4.99	1.66	270	90	0.81	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f3.DetWi		Zone 1	Building bc	0.94	0.94	0	90	3.24	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f3.DetWi		Zone 4	Building bc	3.38	0.05	0	90	6.16	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open
Building body.f3.DetWi		Zone 4	Building bc	4.24	0.91	0	90	3.2	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exteri	Schedule	© Never open

Building body.f4.DetWi	Zone 1	Building bc	0.9	0.9	78.665227	90	3.2	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f4.DetWi	Zone 1	Building bc	0.47	0.47	78.665227	90	4.16	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f4.DetWi	Zone 4	Building bc	4.24	0.91	78.665227	90	4.48	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f4.DetWi	Zone 4	Building bc	4.25	0.92	78.665227	90	3.2	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f4.DetWi	Zone 4	Building bc	4.26	0.93	78.665227	90	1.92	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f4.DetWi	Zone 4	Building bc	4.24	0.91	78.665227	90	3.2	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f5.DetWi	Zone 7	Building bc	0.81	0.81	101.00457	90	4	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f5.DetWi	Zone 7	Building bc	1.2	1.2	101.00457	90	0.9	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f5.DetWi	Zone 2	Building bc	0.71	0.71	101.00457	90	4.16	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f5.DetWi	Zone 2	Building bc	0.9	0.9	101.00457	90	0.81	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f5.DetWi	Zone 2	Building bc	0.44	0.44	101.00457	90	5.2	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f5.DetWi	Zone 2	Building bc	0.65	0.65	101.00457	90	0.9	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f5.DetWi	Zone 5	Building bc	3.63	0.3	101.00457	90	5.72	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f5.DetWi	Zone 5	Building bc	4.22	0.89	101.00457	90	0.81	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f5.DetWi	Zone 5	Building bc	4.64	1.31	101.00457	90	0.81	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f5.DetWi	Zone 5	Building bc	3.66	0.33	101.00457	90	5.72	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open
Building body.f5.DetWi	Zone 7	Building bc	3.61	3.61	101.00457	90	4.84	© Triple lo	0.235	0.111	0.698	0.1	2	0.8282		Light AND S	© Always d	© No exterr	Schedule	© Never open

Internal gains

Name	Type	Number of units	Power	Activity level	Control	Schedule	Energy meter	Description
Zone 1.Adu	Occupant	4		1.8		Young children schedu	Children an	
Zone 1.Ligh	Light	20.84	49.98		Schedule	Schedule 7-	[Default] Lighting, facili	
Zone 1.Chil	Occupant	14		1		Young children schedu	Occupant	
Zone 4.Adu	Occupant	3		1.8		Older children schedule		
Zone 4.Ligh	Light	20.96	49.8		Schedule	Schedule 7-	[Default] Lighting, facili	
Zone 4.Chil	Occupant	12		1.8		Older children schedul	Occupant	
Zone 4.Lea	Occupant	1		1.3		Leader office		Occupant
Zone 4.Equi	Equipment	1	20			© Always o	[Default] E	Equipment
Zone 6.Equi	Equipment	5	75			© Always o	[Default] Equipment, te	
Zone 6.Ligh	Light	20	54.66		Schedule	Schedule 7-	[Default] Lighting, facili	
Zone 7.Ligh	Light	5.614	74.6		Schedule	Schedule 7-	[Default] Lighting, facili	
Zone 7.Chil	Occupant	5		1.8		Older children schedul	Occupant	
Zone 8.Adu	Occupant	1		1.8		Older children schedule		
Zone 8.Ligh	Light	6.371	57.07		Schedule	Schedule 7-	[Default] Lighting, facili	
Zone 8.Chil	Occupant	6		1.8		Older children schedul	Occupant	
Zone 5.Adu	Occupant	1		1.8		Older children schedule		
Zone 5.Ligh	Light	6.443	54.57		Schedule	Schedule 7-	[Default] Lighting, facili	
Zone 5.Chil	Occupant	6		1.8		Older children schedul	Occupant	
Zone 2.Ligh	Light	18.84	38.25		Schedule	Schedule 7-	[Default] Lighting, facili	
Zone 2.Adu	Occupant	4		1.8		Young children schedu	Occupant	
Zone 2.Chil	Occupant	14		1		Young children schedu	Occupant	

Time schedules

Name	Workdays	Saturday	Sunday	Holidays	Vacation
/© Always	1	1	1	1	
/© Always	0	0	0	0	
Schedule 7-	1 [7-16:30],	0	0	0	
Schedule 7-	1.0 [9-15], 0	0	0	0	
Schedule h	1.0 [7-9, 14:	0	0	0	
Young chilc	1 [7-9, 11-12	0	0	0	
Older child	1 [7-9, 14:30	0	0	0	
Leader offi	1 [11-13], 0	1 [11-13], 0	1 [11-13], 0	1 [11-13], 0 otherwise	
Minimum t	22.0 [6-16:3	19	19	19	
Maximum t	25.0 [6-16:3	21	21	21	

Materials			
Name	Heat conduct., W/mK	Density, kg/m3	Specific heat, J/kgK
/© [Defaul	0.13	1000	1300
/© [Defaul	2	2000	1000
/© [Defaul	0.036	32	750
© Floor coa	0.18	1100	920
© L/W conc	0.15	500	1050
© Concrete	1.7	2300	880
© Render	0.8	1800	790
© Gypsum	0.22	970	1090
© Air in 30	0.17	1.2	1006
© Light ins	0.036	20	750
© Wood	0.14	500	2300
© Frames c	0.044	56	1720
Light insula	0.033	20	750
© Brick	0.58	1500	840
© Heavy in	0.052	92	982
© Extruded	0.036	20	1200
© Chip boa	0.13	1000	1300
© Sand	0.33	1500	800
© Soil	1	1500	800
© Stone	3	2700	880

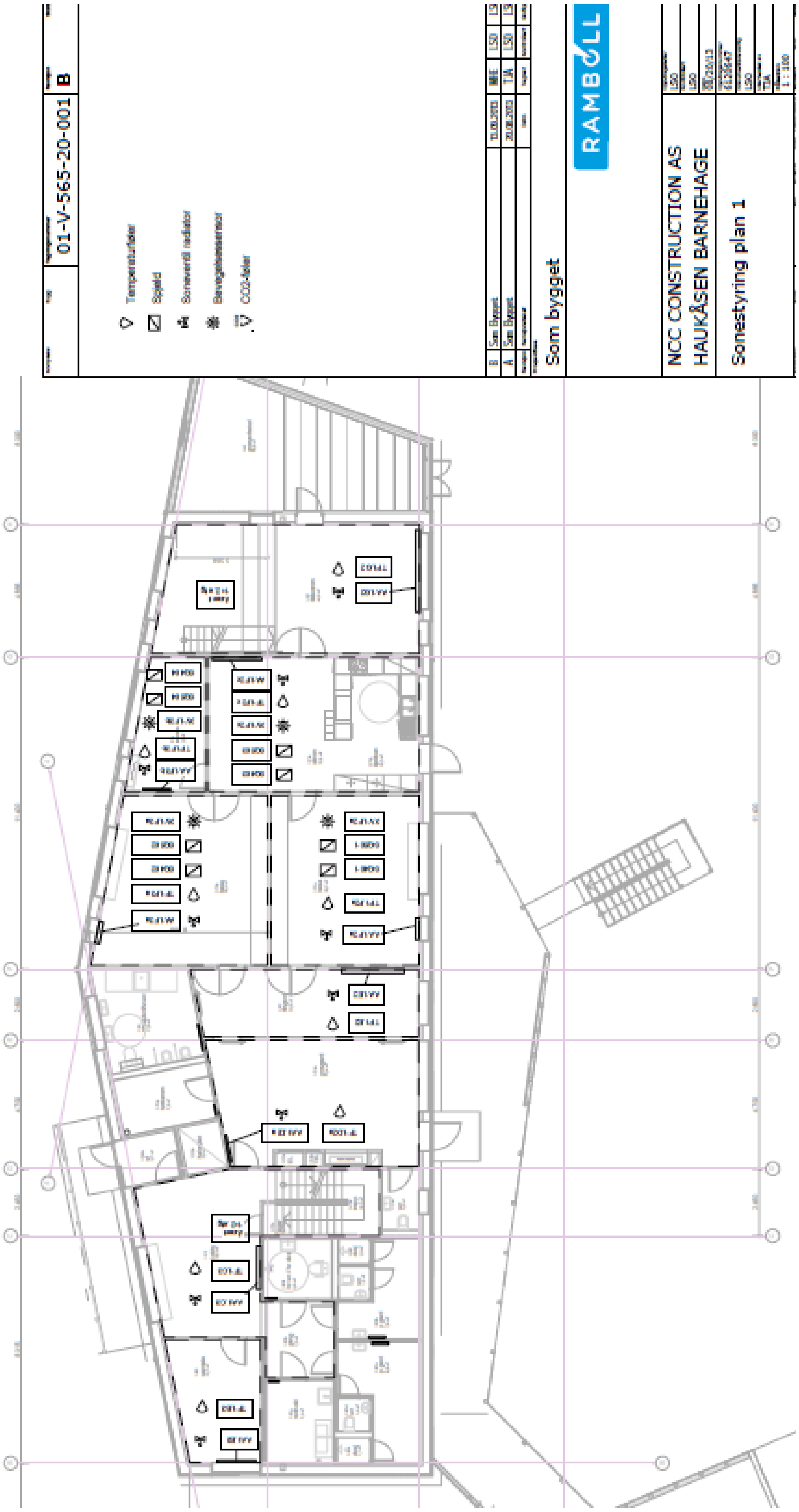
Room units						
Name	Type	Cooling power	Heating power	Controller	Energy meter	Description
Zone 1.Wat	Water Radiator			Proportional		Water Radiator
Zone 4.Wat	Water Radiator			Proportional		Water Radiator
Zone 7.Wat	Water Radiator			Proportional		Water Radiator
Zone 8.Wat	Water Radiator			Proportional		Water Radiator
Zone 5.Wat	Water Radiator			Proportional		Water Radiator
Zone 2.Wat	Water Radiator			Proportional		Water Radiator

Energy meters						
Name	Type	Rate plan	Primary energy factor	CO2 emission per kWh	Role	Color
Lighting, fa	Electrical m	<value not set>			Facility	
Equipment,	Electrical m	<value not set>			Tenant	
HVAC aux	Electrical m	<value not set>			Facility	
Electric hea	Electrical m	<value not set>			Facility	
Fuel heatir	Fuel meter	<value not set>			Facility	
Domestic h	Fuel meter	<value not set>			Facility	
CHP electri	Electrical m	<value not set>			PRODUCED	
Fans	Electrical m	<value not set>			Facility	
Electrical b	Electrical m	<value not set>			Facility	
Heat pump	Electrical m	<value not set>			Facility	

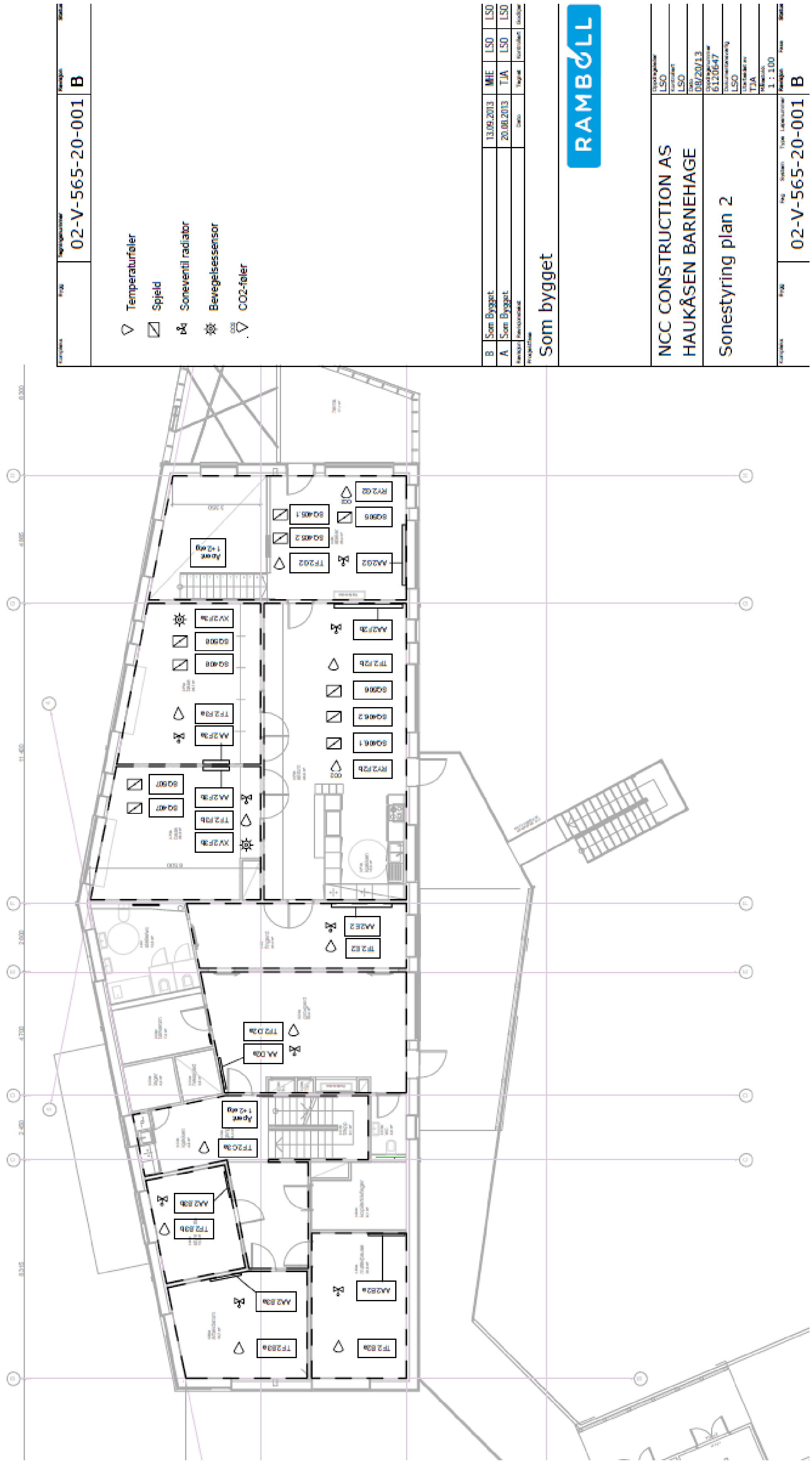
Air handling units								
Name	Served zones	Area, m2	AHU rated supply, L/s	Zone required min supply, L/s	Zone required max supply, L/s	AHU rated return, L/s	Zone required min return, L/s	Zone required max return, L/s
Air Handlin	7	838.9	1275	469.78	2335	1250	469.78	2335

B.2 ZONE CONTROL

Floor 1



Floor 2


02-V-565-20-001
B
Temperaturføler
Spjeld
Soneventil radiator
Bevegelsessensor
CO2-føler
Som Bygget
Som bygget
NCC CONSTRUCTION AS
HAUKÅSEN BARNEHAGE
Sonestyring plan 2
1 : 100
02-V-565-20-001
B

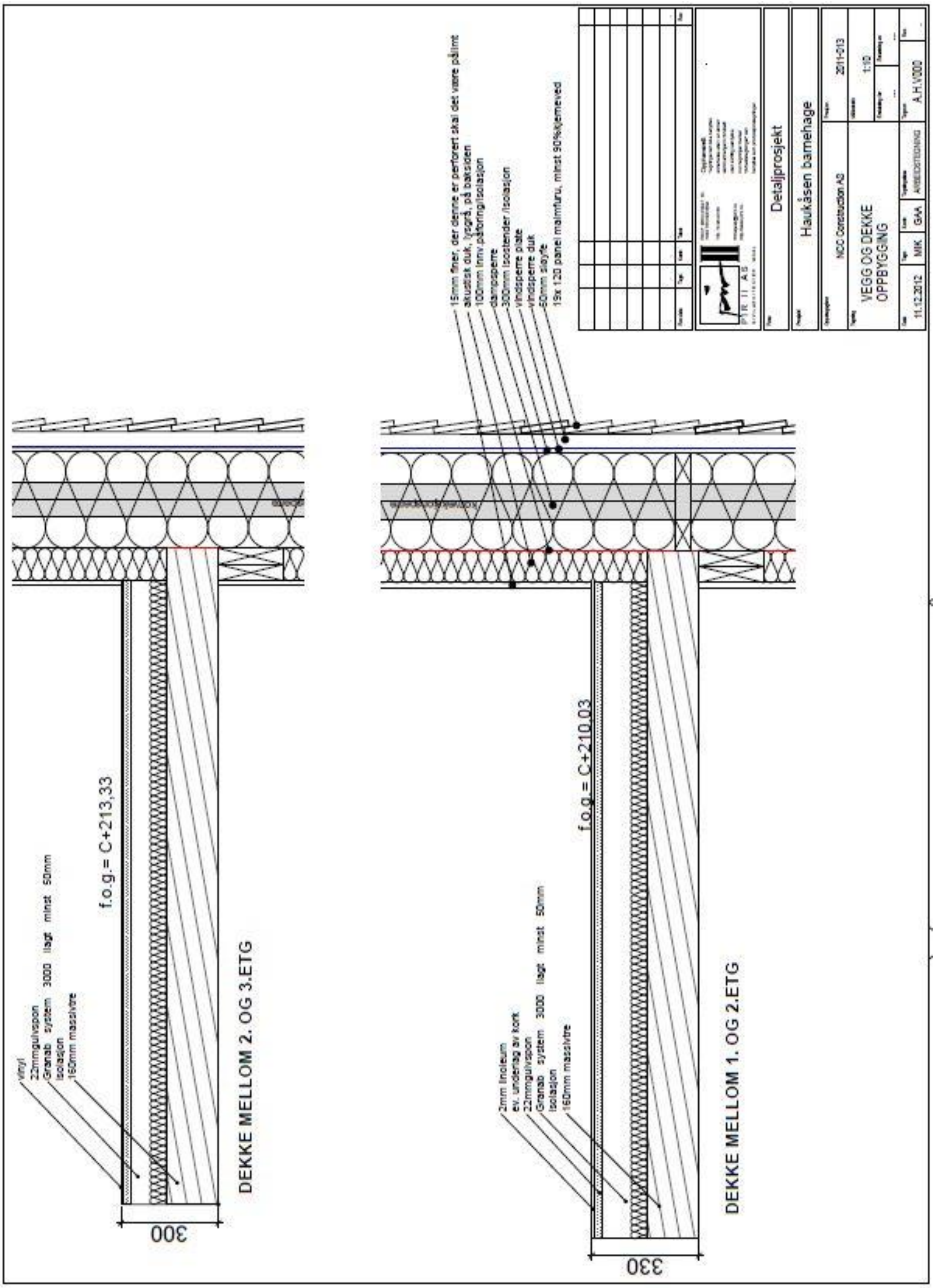
B.3 LIST OVER HOLIDAYS AT HAUKÅSEN KINDERGARTEN

List of holidays

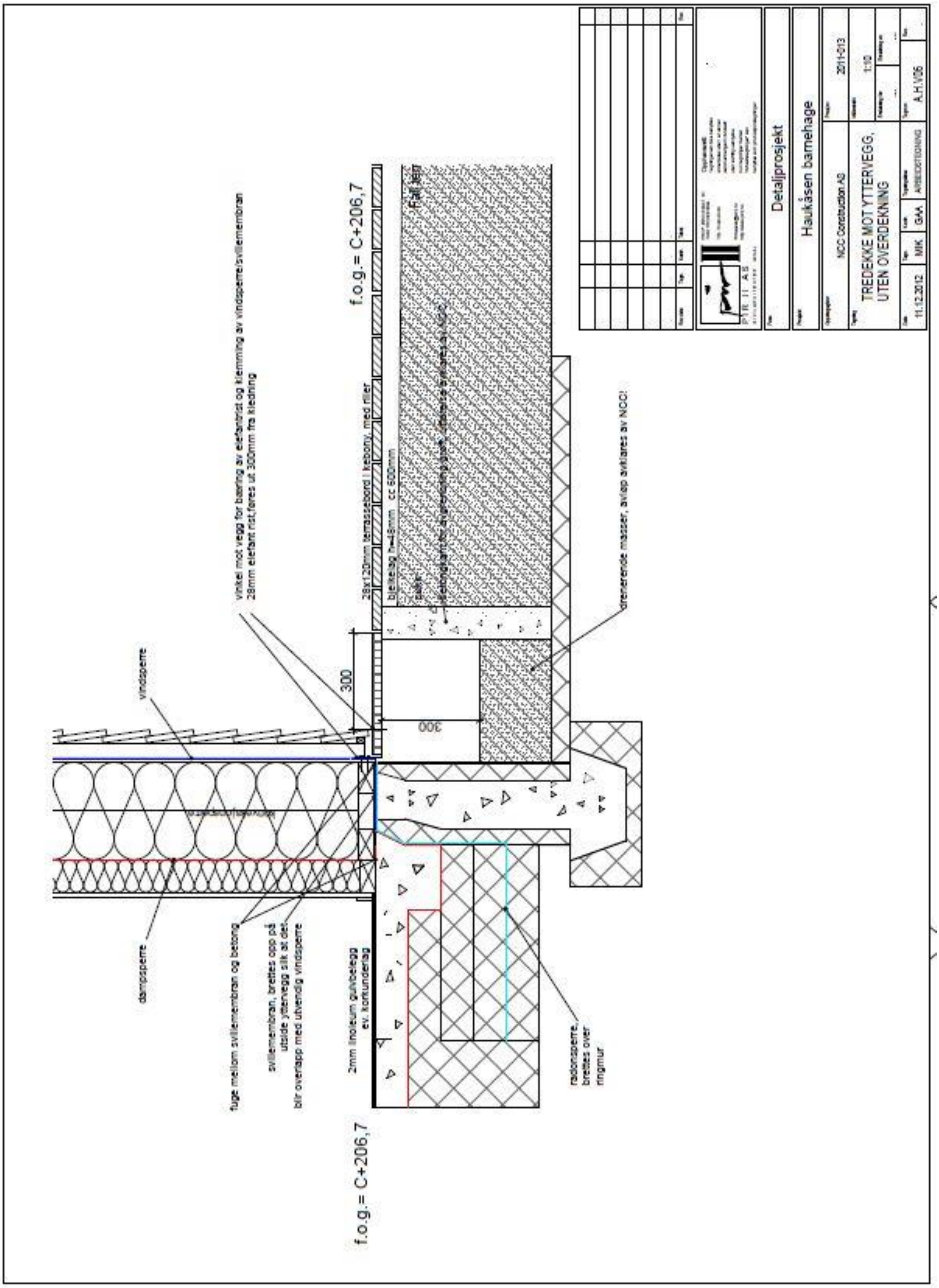
Name	Date	Rule
New Year	1 Jan	1 Jan
Maundy Thursday	28 Mar	Thu 28 Mar - 3 Apr
Good Friday	29 Mar	Easter (W) - 2
Easter Sunday	31 Mar	Easter (W)
Easter Monday	1 Apr	Easter (W) + 1
International Workers' Day	1 May	1 May
Ascension Day	9 May	Easter (W) + 39
Constitution day	17 May	17 May
Pentecost	19 May	Easter (W) + 49
Pentecost 2. day	20 May	Easter (W) + 50
Christmas Eve	24 Dec	24 Dec
Christmas Day	25 Dec	25 Dec
Boxing Day	26 Dec	26 Dec
New Year's Eve	31 Dec	31 Dec

B.4 DETAILED SECTIONAL DRAWINGS

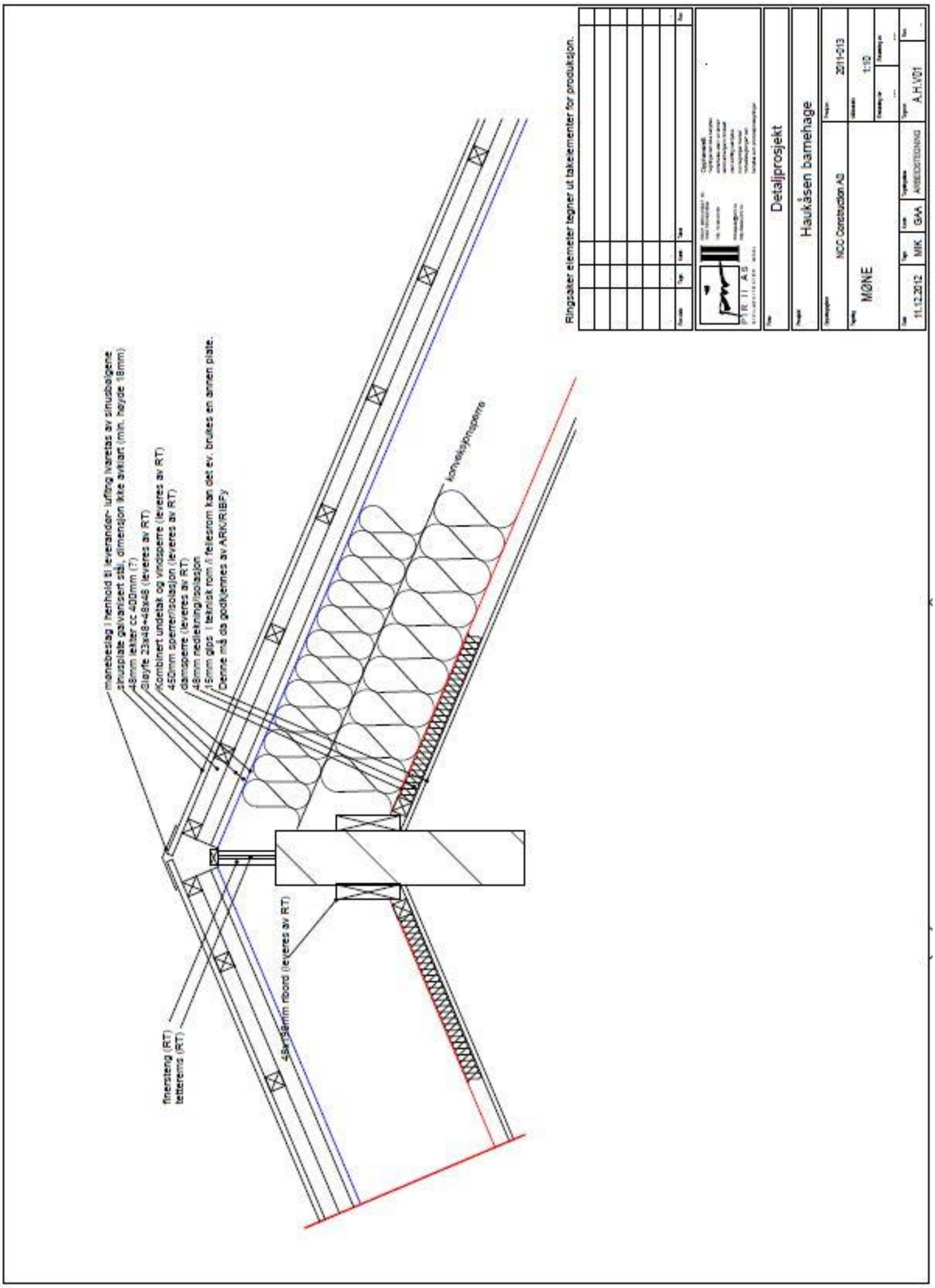
External wall and internal floor



External floor

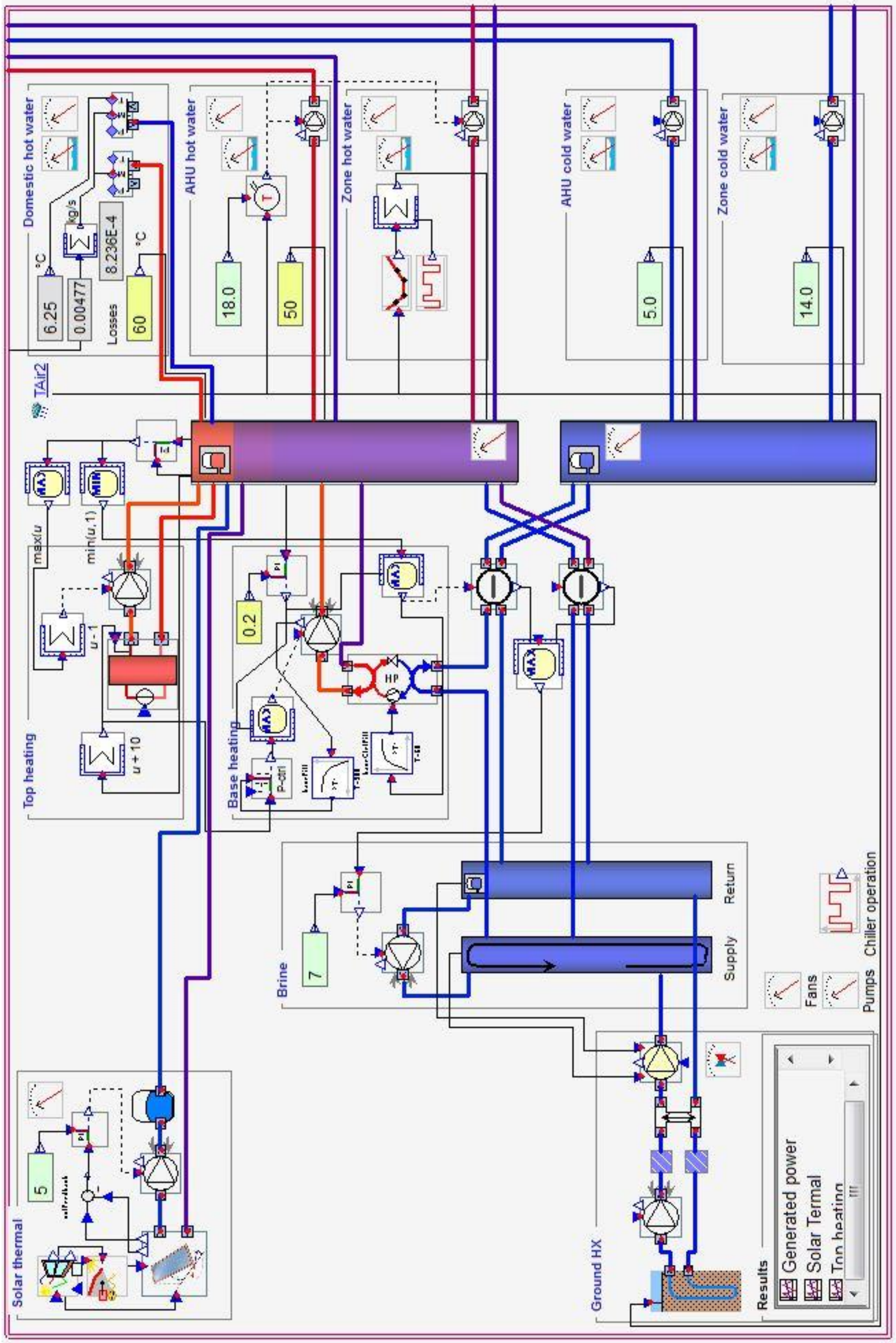


Roof



Ringsaker elementer tegner ut takelementer for produksjon.
mønebeslag i henhold til leverandør- lufting ivaretas av sinusbølgene
sinusplate galvanisert stål, dimensjon ikke avklart (min. høyde 18mm)
48mm lekter cc 400mm (?)
Sløyfe 23x48+48x48 (leveres av RT)
Kombinert undertak og vindsperre (leveres av RT)
450mm sperre/isolasjon (leveres av RT)
dampsperre (leveres av RT)
48mm nedlekning/isolasjon
15mm gips. I teknisk rom i fellesrom kan det ev. brukes en annen plate.
Denne må da godkjennes av ARK/RIBFy
konveksjonssperre
finersteng (RT)
tetteremz (RT)
48x198mm riboard (leveres av RT)
Detaljprosjekt
Haukåsen barnehage
NCC Construction AS
2011-013
1:10
MØNE
11.12.2012
MIK
GAA
ARBEIDSTEGNING
A.H.V01

B.5 IDA ICE PLANT SKETCH



B.6 DATA SHEETS RELEVANT FOR INPUT DATA IN IDA ICE

Solar collector

Flat plate collector (meander with collection manifold, 4 connections)

Order reference: SOLC 220

Article no.: 360520

High-performance flat plate collector for roof mounting, flat-roof or free-standing installation, highly selective, coated and laser-welded full-surface absorber. The universal-use collector is suitable for use with larger surfaces, as well as individual installation. The connections enable quick and hydraulically safe installation. The operation takes place with pre-mixed SOLHT 20 solar fluid and provides the necessary frost protection. The collector casing is made of anthracite-coloured, powder-coated aluminium. The structured solar safety glass provides reliable protection for the absorber.

Flat plate collector with meander pipe and two collection manifolds, max. 10 collectors can be hydraulically connected in series, 4 x 22 mm connections on the side (plug-in system with double O-ring seal), stagnation temperature 202 °C, nominal capacity 1.7 l

- Absorber design: Single meander with collection manifold
- Width: 1158 mm
- Height: 1878 mm
- Depth: 95 mm
- Aperture surface: 2,01 m²
- Gross surface: 2,18 m²
- Connections: 4 x 22 mm lateral (plug-in system. double O-ring)
- Nominal capacity: 1,7 l
- Weight: 34 kg
- Degree of efficiency: 78,1 %
- Nominal flow: 50–120 l/h
- Pressure drop: 28 Pa
- Stagnation temperature: 202 °C

SOLC 220

Reg.Nr.: 011-7S809 F

Heat pump

WQL R410A BLN 30

General technical data		
Cooling Capacity	kW	19,9
Input Power*	kW	6,5
ESEER		4,93
IPLV		4,98
Number of refrigerating circuits	n°	1
Part load steps	n°	2
Power supply	V/f/Hz	400V/3/50Hz
Nominal running current	A	22
Startup current	A	118
Refrigerant	Type	R410A

* Value referred to compressors only

Compressors		
Number	n°	2
Type		Scroll
Startup type		Direct

Evaporator		
Number	n°	1
Type		Plates
Chilled fluid		Water + Ethylene Glycol 20%
Chilled fluid flow rate	l/s	1,65
Pressure drop on the chilled fluid side	kPa	50,61
Chilled fluid inlet temperature	°C	3
Chilled fluid outlet temperature	°C	0
Fouling factor	m²°C/KW	0,044
Water connections Type		Victaulic
Inlet diameter		1"1/2
Outlet diameter		1"1/2

Condensator		
Number	n°	1
Type		Plates
Heated fluid		Water
Heated fluid flow rate	l/s	0,79
Pressure drop on the heated fluid side	kPa	11
Heated fluid inlet temperature	°C	35
Heated fluid outlet temperature	°C	43
Fouling factor	m²°C/KW	0,044
Water connections Type		Victaulic
Inlet diameter		1"1/2
Outlet diameter		1"1/2

Dimensions and Weights		
Length	mm	821
Width	mm	455
Height	mm	1350
Operating weight	kg	179

Sound data		
Sound power level	dB(A)	67
Distance from the sound source	m	1
Sound pressure level**	dB(A)	52

** Value calculated with a parallelepiped as a reference surface. Directionality factor Q=2

All specification and data are subject to change without notice.

Heat recovery unit

FläktWoods

EUROVENT CERTIFIED PERFORMANCE

LUFTBEHANDLINGSAGGREGAT eQ

Projekt	2218 (3240017) / HAUKÅSEN BARNEHAGE	*2.6.120906.1*
AOC	ACON-01137226	
Aggregat	4 (01B) / 120911 360.01 ENHETS 3-DELT U AUTO	2012-09-28
Storlek	041	Sida 6

Roterande värmeväxlare REGOTERM — **EQRB-041-1-3-1-1-0-1-1-1-1-1-1**

Aggregatstorlek: 041
Rotorutförande: ej hygroskopisk
Effektvariant: Effektvariant 5 (1,55)
Spänning/frekvens : 1 x 230 V, 50Hz
Drivtyp/Klassning, motor : Reglerbart varvtal
Funktionslängd: standard (enbart rotor)
Tilluftsplacering: undervåning
Leveransutförande: hellindad rotor, komplett levererad
Material: fz stålplåt/Alzn
Inspektionssida: höger
Versionsnummer: Med kabelkanal

	Vinter	
Temperaturverkningsgrad	83,2	%
Effekt reduktion	111,3	kW
Fuktverkningsgrad	48,6	%
Luftflödesöverföring	0,21	m^3/s
Tilluft	**Vinter**	
Tryckfall	108	Pa
Lufttemperatur	-19,0 / 15,1	°C
Relativ fuktighet	70,1 / 25,0	%
Frånluft	**Vinter**	
Tryckfall	114	Pa
Lufttemperatur	22,0 / -9,3	°C
Relativ fuktighet	30,0 / 99,9	%

Tomdel — **EQTC-041-030-0-0-0-0-0-0-1-1-1**

Aggregatstorlek: 041
Längd: 030
Inspektionsdörr: med (för undertryck)
Inspektionssida: höger

B.7 AIR FLOW RATE TABLE

RAMBØLL

Prosjekt nr:	6120647		Utført av:
Byggherre:	TRONDHEIM KOMMUNE	Dato: 24.06.2011	LSO
Prosjekt:	HAUKÅSEN BARNEHAGE	Rev.dato: 28.09.2012	LSO

Type bygg: Barnehage

Luftmengder:

Plan	[m³/h]	Aggregat	Tilluft [m³/h]	Avtrekk [m³/h]		Inkl. 25%	Areal	m3/hm2
U1		360.01				m3/h		
1	4330	360.01				m3/h		
2	3620	360.01				m3/h		
3	450	360.01				m3/h		
Sum	8400					**m3/h**		

Rom nr.	Romfunksjon:	Plan	Areal [m²]	Till.temp [°C]	Ant. pers.	Dimensjonering L/pers. [m³/h-p]	Dimensjonering L/areal i bruk [m³/h-m²]	Luftbehov i bruk Tot. [m³/h]	Tilluft i bruk Tot. [m³/h]	Avtrekk i bruk Tot. [m³/h]	Opptredende L/pers. [m³/h-p]	Opptredende L/areal i bruk [m³/h-m²]	Aggr. nr.
1.D3a	Heissjakt	1	3,8	22	0	26	2,5	10	0	0	0,0	0,0	1
1.D3b	VF	1	3,8	22	0	26	2,5	10	0	0	0,0	0,0	1
1.C3	Lobby	1	25,3	22	0	26	2,5	63	400	0	0,0	15,8	1
1.B3	Samtale	1	13,7	22	6	26	2,5	190	200	200	33,3	14,6	1
1.B2g	Renhold	1	7,4	22	0	26	2,5	19	0	150	0,0	0,0	1
1.B2a	Personalgarderobe	1	9,9	22	0	26	2,5	25	200	0	0,0	20,2	1
1.B2c	dusj	1	1,1	22	0	26	2,5	3	0	100	0,0	0,0	1
1.B2d	wc	1	1,4	22	0	26	2,5	4	0	100	0,0	0,0	1
1.B2b	Personalgarderobe	1	7,9	22	0	26	2,5	20	200	0	0,0	25,3	1
1.B2f	dusj	1	1,1	22	0	26	2,5	3	0	100	0,0	0,0	1
1.B2e	wc	1	1,3	22	0	26	2,5	3	0	100	0,0	0,0	1
1.B2i	HCWC / HC dusj	1	5,7	22	0	26	2,5	14	0	100	0,0	0,0	1
1.B2h	Gang	1	7,3	22	0	26	2,5	18	0	0	0,0	0,0	1
1.C2b	Trapp	1	6,6	22	0	26	2,5	17	0	0	0,0	0,0	1
1.C2c	Lager	1	3,6	22	0	26	2,5	9	0	0	0,0	0,0	1
1.C3c	wc	1	3,3	22	0	26	2,5	8	0	100	0,0	0,0	1
1.D2a	Grovgarderobe	1	32,5	22	0	26	2,5	81	0	150	0,0	0,0	1
1.E2	Fingarderobe	1	20	22	0	26	2,5	50	550	0	0,0	27,5	1
1.D3c	Tørkerom	1	5,6	22	0	26	2,5	14	0	100	0,0	0,0	1
1.E3	wc/stelle	1	13	22	0	26	2,5	33	0	200	0,0	0,0	1
1.F2a	Base	1	33,7	22	19	26	2,5	578	600	600	31,6	17,8	1
1.F3a	Base	1	36,4	22	19	26	2,5	585	600	600	31,6	16,5	1
1.F2b+1:F2c	Allrom/kjøkken	1	51,5	22	19	26	2,5	623	630	630	33,2	12,2	1
1.F3b	Verksted	1	12,3	22	6	26	2,5	191	200	200	32,5	16,3	1
1.G2	Fellesrom	1	44	22	22	26	2,5	682	750	0	34,1	17,0	1
1.H2	Grovverksted	1	35,3	22	0	26	2,5	88	0	0	0,0	0,0	1
2.C3b	Kjøkken	2	4,6	22	0	26	2,5	12	0	0	0,0	0,0	1
2.C3a	Gang	2	18,2	22	0	26	2,5	46	0	0	0,0	0,0	1
2.B3b	Styrer/møte	2	12,1	22	4	26	2,5	134	150	150	37,5	12,4	1
2.B3a	Arbeidsrom	2	19,6	22	6	26	2,5	205	200	200	33,3	10,2	1
2.B2a	Møte/pause	2	20,7	22	10	26	2,5	312	320	320	32,0	15,5	1
2.B2b	Arkiv/kopi/lager	2	7,8	22	0	26	2,5	20	0	150	0,0	0,0	1
2.C2b	Trapp	2	9,3	22	0	26	2,5	23	0	0	0,0	0,0	1
2.C2a	wc	2	3,3	22	0	26	2,5	8	0	100	0,0	0,0	1
2.D2a	Grovgarderobe	2	32,5	22	0	26	2,5	81	0	150	0,0	0,0	1
2.D3a	Heissjakt	2	3,8	22	0	26	2,5	10	0	0	0,0	0,0	1
2.D3b	Lager	2	2,9	22	0	26	2,5	7	0	0	0,0	0,0	1
2.E2	Fingarderobe	2	20	22	0	26	2,5	50	550	0	0,0	27,5	1
2.D3c	Tørkerom	2	5,6	22	0	26	2,5	14	0	100	0,0	0,0	1
2.E3	wc/stelle	2	13	22	0	26	2,5	33	0	200	0,0	0,0	1
2.F3a	Base	2	29,2	22	16	26	2,5	489	500	500	31,3	17,1	1
2.F3b	Base	2	29,7	22	16	26	2,5	490	500	500	31,3	16,8	1
2.F2a+ 2.F2b	Allrom/kjøkken	2	59,7	22	30	26	2,5	925	1000	1000	33,5	16,8	1
2.G2	Atelier	2	25,8	22	13	26	2,5	400	400	1150	31,0	15,5	1
00	0	0	0,0	0	0	26	2,5	0					
3.B2a + 3.B2b	Teknisk rom/disp	3	162,0	22	0	26	2,5	405	250	250	0,0	1,5	1
3.C2	Trapp	3	12,9	22	0	26	2,5	32	0	0	0,0	0,0	1
00	0	0	0,0	0	0	26	2,5	0					
00	0	0	0,0	0	0	26	2,5	0					
00	0	0	0,0	0	0	26	2,5	0					
3.D2a	Telerom	3	7,3	22	0	26	2,5	18	200	200	0,0	27,4	1
00	0	0	0,0	0	0	26	2,5	0					
	Totalt		**887,5**					**7052**	**8400**	**8400**			
	Inkl 25 % tillegg							**8815**	**10500**	**10500**			

4580	**3820**	**4313**	**total luftmengde CAV+VAV**	**271**
luftmengder tilgjengeleg for VAV	**Luftmengder CAV**	**5**	**luftmengde per areal CAV + VAV i enkelte områder**	

C. CALCULATIONS

C.1 ANNUAL ENERGY USE FROM IDA ICE WHEN SUBTRACTING SUMMER MONTHS

Table 39 shows the mean effect load per month retrieved from a one-year simulation in IDA ICE, and the energy use calculations are given beneath.

Table 39 - Mean effect load per month (from IDA ICE)

Month, n	**Mean effect load, P_{mean} (W)**
January	6 202
February	5 126
March	3 619
April	3 208
May	2 607
June	2 691
July	2 860
August	2 820
September	2 891
October	3 476
November	4 272
December	4 199

Mean effect load over the year (W):

$$P_{\text{mean,annual}} = \frac{\sum_{i=month}^{n} P_{mean,i}}{n} \tag{20}$$

When the summer months are not taken into account, $P_{mean,annual}$ equals 3955 W versus 3664 W when all months are considered.

Annual energy use (kWh):

$$W = P_{mean,annual}(W) * 8760 \left(\frac{h}{year}\right) * 10^{-3} \left(\frac{kW}{W}\right) \tag{21}$$

This gives an annual expected energy use of 34649 kWh when the summer months are not taken into account, versus 32097 kWh when all months are considered.

C.2 SPREADSHEET USED IN ECONOMIC ANALYSIS

Economic analysis

Haukåsen Kindergarten

2014-06-10

Real interest rate	0,04
Calculation period (years)	20
Present value factor	13,59

FIXED COSTS	
Energy cost (NOK/year)	-2140

INPUT DATA FOR EXISTING HEAT SUPPLY

Component	Investment cost (NOK)	Lifetime (years)	Energy consumption (kWh/year)	Energy price (NOK/kWh)	Operational cost (NOK/year)	Preventive maintenance cost (NOK/year)	Residual value (NOK)		Global costs
Solar collector	-20200	30	0	0,5318	0,00	-101	6733		-18499,6
Accumulator tank	-19100	30	0	0,5318	0,00	-191	6367		-18790,1
Heat pump	-50800	20	8749	0,5318	-4652,72	-1016	0		-127839,7
Energy wells	-192000	30	0	0,5318	0,00	-1920	64000		-188884,7
Electrical boiler	-39760	20	3408	0,5318	-1812,37	-397,6	0		-69794,3
								SUM:	**-452891,7**

INPUT DATA FOR ALTERNATIVE HEAT SUPPLY (NUMBER 2)

Component	Investment cost (NOK)	Lifetime (years)	Energy consumption (kWh/year)	Energy price (NOK/kWh)	Operational cost (NOK/year)	Preventive maintenance cost (NOK/year)	Residual value (NOK)	Global costs
Heat exchanger	-40000	30	26829	0,5831	-15643,99	-400	13333,3	-251957,8993
								-251957,90

Pay-off period

Real interest rate	0,04
Optional heating source:	Electricity
Heating need (kWh/year):	26829
Energy price (NOK/kWh):	0,5318
Operational cost (NOK/year	-14267,6622

Energy supply alternative	Investment cost (NOK)	Savings in operational costs (NOK/year)	Pay-off period (years)
1	-321860	7802,57	24,8
2	-79760	763,67	41,9