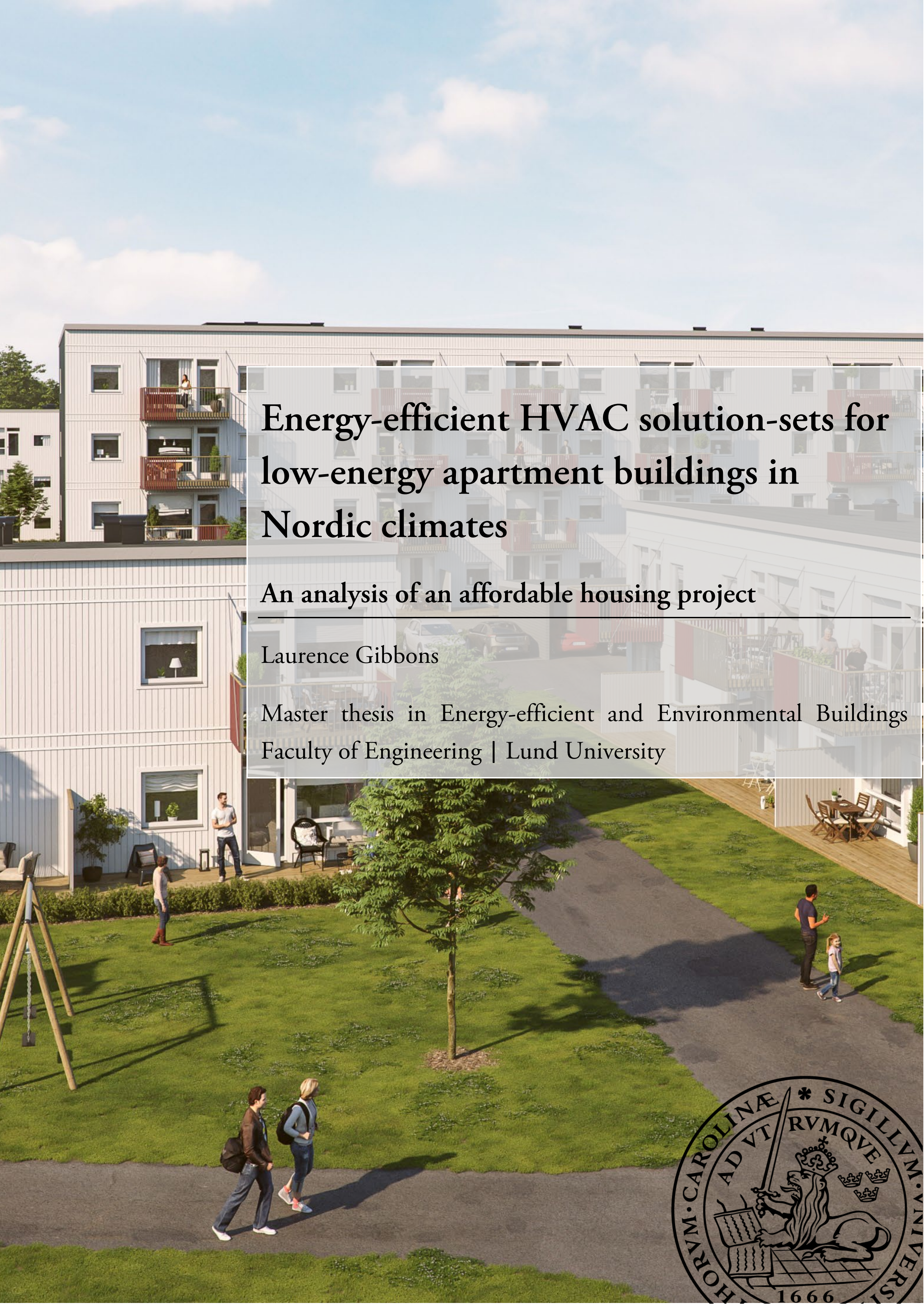

Energy-efficient HVAC solution-sets for low-energy apartment buildings in Nordic climates

An analysis of an affordable housing project

Laurence Gibbons

Master thesis in Energy-efficient and Environmental Buildings
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Lund University

Lund University, with eight faculties and a number of research centres and specialized institutes, is the largest establishment for research and higher education in Scandinavia. The main part of the University is situated in the small city of Lund which has about 112 000 inhabitants. A number of departments for research and education are, however, located in Malmö. Lund University was founded in 1666 and has today a total staff of 6 000 employees and 47 000 students attending 280 degree programmes and 2 300 subject courses offered by 63 departments.

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This international programme provides knowledge, skills and competencies within the area of energy-efficient and environmental building design in cold climates. The goal is to train highly skilled professionals, who will significantly contribute to and influence the design, building or renovation of energy-efficient buildings, taking into consideration the architecture and environment, the inhabitants' behaviour and needs, their health and comfort as well as the overall economy.

The degree project is the final part of the master programme leading to a Master of Science (120 credits) in Energy-efficient and Environmental Buildings.

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Cover image of computer render of BoKlok project provided by BoKlok Norway.

Abstract

To meet the aims of the Paris climate agreement, it is necessary to reduce energy use in buildings. Efficient HVAC systems are a critical part in achieving a low delivered energy use. However, there is a lack of comparative information of the best HVAC systems for apartment buildings, which represent an increasing share of newly built floor area in the Nordic countries. Elements of HVAC systems were found through a statistical review of national EPC databases and a literature review. Resulting solution-sets were analysed through simulation of an affordable housing project in Sørum, Norway, built using a modular construction. The solution-sets were compared for energy use and energy cost. The potential for energy flexibility using thermal energy storage was also examined.

The solution-sets were comprised of five heat emitter options, three ventilation options, six schedules and seven energy supply systems. Thermal energy storage was also identified as an important element in the sizing of the energy supply system.

Underfloor heating and fan coils were the most efficient heat emitters. Fan coils were a more practical solution for the case study. Solutions using balanced ventilation system were more efficient than those with exhaust ventilation, although the difference could be offset by using an exhaust air heat pump. Maintaining a constant setpoint or allowing it to setback 2 °C during the night affected both the energy demand and peak demand. The constant setpoint required more energy but had a lower peak demand, allowing for a smaller heating system. For systems with a fixed size or low system cost, a variable setpoint was better.

All the energy supply systems could have lower energy costs than the standard system used in each apartment, a 100 L electric immersion water tank. For exhaust air heat pump systems, this was only possible when combined with district heating. Ground source heat pump systems had the lowest delivered energy and energy costs. Using solar thermal collectors to provide DHW and recharge the boreholes reduced the total borehole depth while increasing the temperature of the brine. The improved heat pump performance resulted in a further reduction of the delivered energy and energy cost. Compact systems offered a low energy cost and the best thermal comfort with the possibility of heating and cooling all year round. However, its use was hindered by a high price and practical issues.

The potential of consumer driven energy flexibility was shown to be limited due to the small cost savings possible and the large tank sizes required. An optimisation of the demand profile to use the lowest hourly cost of electricity did produce savings but these were outweighed by the higher charges for the high peak demand. When applied to the base case with a minimum tank temperature of 55 °C, a 5 % energy cost saving was possible.

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Abbreviations

ACH	Air Changes per Hour
AHU	Air Handling Unit
ASHP	Air-Source Heat Pump
A_{temp}	Heated floor area
COP	Coefficient of Performance
DH	District Heating
DHW	Domestic Hot Water
EAHP	Exhaust Air Heat Pump
EL	Direct Electric heating
EPBD	Energy Performance of Buildings Directive
EPC	Energy Performance Certificate
FP	Flat Plate solar thermal collector
GSHP	Ground Source Heat Pump
HP	Heat Pump
HR	Heat Recovery
HVAC	Heating, Ventilation and Air Conditioning
LTDH	Low Temperature District Heating
nZEB	nearly Zero-Energy Building
NZEB	Net Zero-Energy Building
PEF	Primary Energy Factor
PV	Photovoltaic solar panel
PVT	Photovoltaic Thermal collector
SAGSHP	Solar Assisted Ground Source Heat Pump
SC	Solar thermal Collector
SCOP	Seasonal Coefficient of Performance
TES	Thermal Energy Storage
WSHP	Water-Source Heat Pump
VT	Vacuum tube solar thermal collector

1 Introduction

Globally the energy used by buildings is responsible for 28 % of energy related CO_2 emissions with absolute emissions still increasing (IEA, 2020). As built floor area is increasing at 2.5 % annually, the energy efficiency of buildings needs to be improved by a greater percentage in order to reduce emissions and meet the goals of the Paris Agreement (UNFCCC, 2015). The European Union and Norway have legislated for energy-efficient buildings since 2002 through the Energy Performance of Buildings Directive (EPBD) (EU, 2002). Although not a member of the EU, Norway has implemented much of the directive in national legislation and is a member of the Concerted Action EPBD (Brekke et al., 2016). According to the 2010 recast, all new buildings in the EU must be nearly Zero-Energy Buildings (nZEB) from the start of 2021 (EU, 2010). Similar nZEB legislation was expected in Norway in 2020 but has been delayed (Lotherington, 2020).

The need for best practice solutions for achieving nZEBs has resulted in increased research, and development of exemplar projects. Energy demands have been dramatically decreased for heating, through improving the U-value and air tightness of the building thermal envelop; and for cooling, through improved specification of glazing and shading strategies (Economidou et al., 2011). However, further improvements with passive measures are subject to diminishing rates of return, meaning that both a cost and environmental optimum can be found (Ylmén et al., 2021). In turn, greater air tightness has increased the requirement for mechanical ventilation (Lechner, 2015). Therefore, the focus for additional energy savings has shifted to the optimisation of the mechanical systems and their energy supply. Any improvements in efficiency of these systems should not reduce thermal comfort, indoor air quality nor user satisfaction. To achieve the future target of Net Zero-energy Buildings (NZEB), the remaining demand must be supplied by on-site renewable energy generation. As the share of renewable energy, both on-site and imported, increases, the energy balance of the system also becomes an important factor in the HVAC design. Most importantly, for these new efficient solutions to be implemented in a large proportion of the building stock, they must be cost-effective.

This thesis focused on the optimisation of HVAC solutions for a multi-family apartment building for energy use and cost, in the challenging Nordic climate. Apartment buildings account for more than a third of residential floor area in Europe, which accounts for 75 % of total floor area (Economidou et al., 2011). The increased rate of urbanisation and densification in the Nordics, requires denser residential buildings such as apartment buildings (Smas et al., 2016). The cold climate of the Nordics requires particular attention on reducing heating demand and the effective supply of heating.

Optimizing system design in apartment buildings is particularly complex because systems can be designed as completely centralised through to completely decentralised (apartment-based) systems, resulting in a large number of possible solution-sets. The most cost-effective and energy-efficient solution depends on the characteristics of the building and is highly sensitive to technological development. The small roof area relative to floor area further increases the need for energy efficiency as the possibility to compensate with solar energy is limited (Erhorn and Erhorn-Kluttig, 2018).

Therefore, apartment buildings are an important typology to study. Despite this, only one comparable study has been undertaken, the EU Horizon 2020 project ConZEBS (Cost reduction of new Nearly Zero-Energy buildings), which presented cost-effective solution-sets for multi-family houses in Denmark, Germany, Italy and Slovenia (Gutierrez et al., 2019). Each country had its own unique set of solutions, demonstrating that there is not a silver bullet to achieve nZEBs. Therefore, there is a need to expand such research to define solutions for each country and climatic region.

1.1 Background

The HVAC system design of apartment buildings is subject to several competing factors, outlined in this section. Although these factors are general, they are explained from the perspective of the Norwegian market reflecting the location of the case study building.

1.1.1 Regulation push and Market pull

The need for developers to construct better performing buildings in the residential sector is due to a combination of regulation push and market pull. Building regulations have become increasingly strict. In Norway, the first specific energy requirements in the building regulations were introduced in 2007 (DiBK, 2007), although stipulations on U-values have been in place since 1997 (DiBK, 1997). These were updated in 2010 (DiBK, 2010) and maintained in the current regulations (DiBK, 2017). The development is shown in Table 1.1. It is possible to meet the energy requirements in two ways: either by achieving the minimum net energy requirement, U-values and airtightness (Method A); or achieving lower U-values and better airtightness with no net energy requirement (Method B). In the second method, there are also requirements for ventilation heat recovery, fan power and thermal bridges.

Apartment buildings are also required to have an energy performance certificate (EPC) for each apartment. EPC registration started in December 2009 and was mandatory for residential property sold or rented after July 2010 (Brekke et al., 2016). The rating is composed of a letter A through G, representing energy use, and five colours, which indicate the type of heating system used (Enova, 2009a). The boundaries for the letters and typical systems for the colours for apartment buildings are shown in Table 1.2 and Table 1.3. All letter and colour combinations are possible.

Table 1.1. Energy requirements of Norwegian building regulations since 1997.

	TEK 97	**TEK 07**		**TEK 10 / TEK 17**	
		Method A	Method B	Method A	Method B
Net Energy requirement / (kWh/(m²·year))	n/a	120	n/a	95	n/a
U-value / (W/(m²·K))					
- Wall	0.22	0.22	0.18	0.22	0.18
- Roof	0.15	0.18	0.13	0.18	0.13
- Ground	0.15	0.18	0.15	0.18	0.10
- Windows and doors (maximum window area allowed in relation to heated floor area)	1.60 (n/a)	1.60 (n/a)	1.20 (20%)	1.20 (n/a)	0.80 (25%)
Air tightness / ACH at 50 Pa	n/a	3.0	1.5	1.5	0.6
Heat recovery	n/a	n/a	70 %	n/a	80 %
Specific Fan Power / (kW/(m³/s))	n/a	n/a	2.5	n/a	1.5
Normalised thermal bridges / (kWh/m²)	n/a	n/a	n/a	n/a	0.07

As these have become understood by occupants, a value has been placed on buildings with good EPC grades. A-grade apartments in Norway are associated with 2.7 % higher rents than D-grade apartments and those with a green colour have 3.3 % higher rents than non-green apartments (Khazal and Sønstebø, 2020). Whether this premium is entirely due to the EPC is widely debated (Olaussen et al., 2019). Property buyers are likely driven by the associated lower energy costs and new construction than environmental ambitions. As the price premium is small, the cost of improved energy efficiency must also be small to be of interest to a developer. Similarly, there is little incentive for a developer to greatly exceed the requirement for an A, as this is not visible in the market.

Table 1.2. Letter grade for Norwegian energy performance certificates for apartments (Enova, 2009a).

Grade	**A**	**B**	**C**	**D**	**E**	**F**	**G**
Delivered energy / (kWh/m²)	85	95	110	135	160	200	>F
Addition depending on heated Floor Area (A)	600/A	1000/A	1500/A	2200/A	3000/A	4000/A	

Table 1.3. Typical systems for the Norwegian energy performance certificate colours (Enova, 2009b).

Dark Green	- Hydronic heating from biomass boiler. - District heating.
Light Green	- Hydronic heating from ground source heat pump with solar thermal collectors.
Yellow	- Hydronic heating from ground source heat pump. - Air to air heat pump, wood stove and direct electrical heating. - Hydronic heating form pellet oven. - Solar thermal collectors and air to water heat pump.
Orange	- Direct electrical heating and wood stove. - Solar thermal collectors and direct electrical heating.
Red	- Air to air heat pump and direct electrical heating. - Direct electrical heating. - Fossil fuel boiler.

1.1.2 Cost from different perspectives

Cost is an important factor and often a barrier for any new energy-efficient solution to be adopted (Intrachooto and Horayangkura, 2007). For a solution to be cost-effective it must cost the same or less than the typical solution. The cost-optimal solution is the one with the lowest cost. Energy-efficient solutions often require higher investment costs which are offset by lower ongoing energy costs, requiring a life cycle perspective. In apartment buildings, this is complicated as those investing in technologies and those benefiting from them are often not the same party. Developers often give weight to the capital investment cost as this involves less risk and a faster return on investment. However, this affects the energy and maintenance costs paid for by the building owner / occupants. What constitutes a cost-effective solution is therefore a matter of perspective, as shown in Figure 1.1. A focus on investment cost can result in a higher energy use than the life cycle cost optimum.

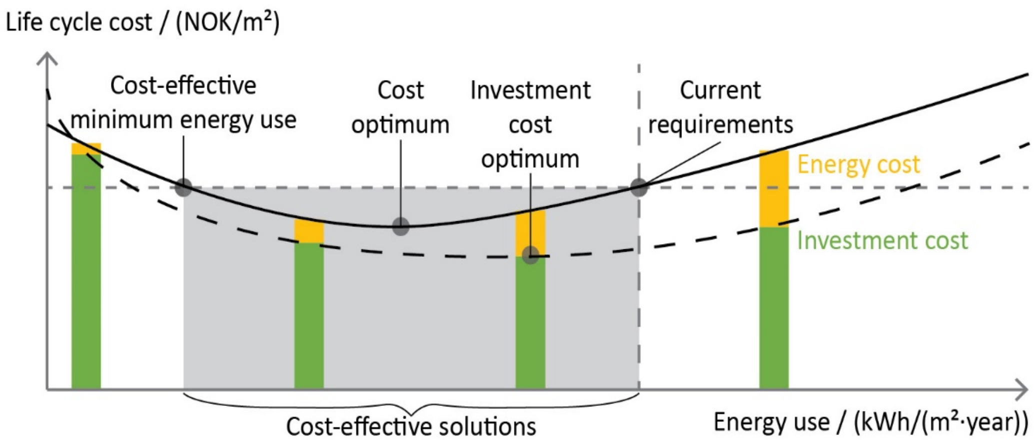

Figure 1.1. Energy vs cost curve showing cost-effective and cost optimum solutions.

The EPBD recast defines cost optimal as: "the energy performance level which leads to the lowest cost during the estimated economic lifecycle" (EU, 2010). A complete life cycle costing includes the investment costs, operating costs, maintenance costs and energy costs for a set lifespan, defined by each country (Wittchen and Engelund Thomsen, 2012). The directive suggests that minimum energy performance requirements should be set at this cost optimal point. Where this is not the case, countries must justify the difference or outline steps to change the regulations. As technological development and price changes shift this optimum point, regular assessment is required, leading to a reduction of the energy use requirement towards NZEB. The move to heating systems using renewable energy sources (e.g. heat pumps, solar technologies) is a key driver of this reduction (Simson et al., 2019).

1.1.3 Energy-efficient systems

The energy demand is the focus of current regulations. The energy requirements of both the Norwegian building regulations and EPC are based on the energy required for heating, domestic hot water (DHW), fans, pumps, lighting and equipment. The calculation is standardised to use an Oslo climate file, with fixed demands for DHW (29.8 kWh/(m²·year)), lighting (11.4 kWh/(m²·year)) and equipment (17.5 kWh/(m²·year)) (Standard Norge, 2014). However, the building regulation requirement is for **net energy demand** while the EPC requirement is for **delivered energy demand**. In between these two terms, is the **Gross energy demand.** The scope of these terms is illustrated in Figure 1.2.

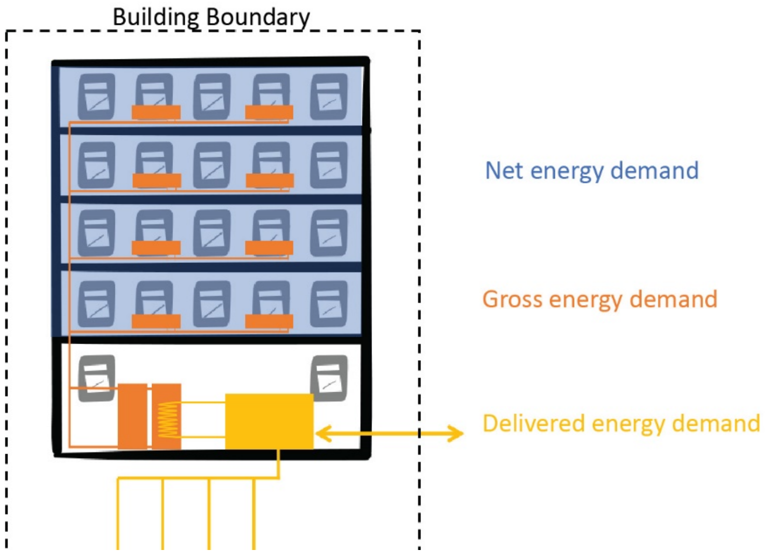

Figure 1.2. Illustration of the scope of each energy demand definition.

Net energy demand describes the energy required to supply the heated floor area or zone. As the loads are standardised, it is affected by the solar gain, ventilation rate, ventilation heat recovery and the U-values and heat capacity of the thermal envelope. **Gross energy demand** is the sum of net energy demand and the distribution losses for the pipes and tanks. **Delivered energy demand** is the net amount of imported energy delivered to the building. It can also be described as the product of the gross energy demand and the working efficiencies of the heating system. Where energy is exported from excess production of on-site energy generation, this is subtracted from the imported energy. These three demands can also be expressed in primary energy terms, by scaling the energy use by a primary energy factor according to the type of energy used. Primary energy factors are not applied in Norway.

The term energy efficiency can be applied to any of these energy demands but it is possible to negate an energy-efficient net energy demand with poor distribution and energy system performance. An efficient delivered energy is only possible with an efficient gross and net energy demand. Delivered energy demand most accurately reflects the demand of a building on the communal electrical and thermal networks. However, this is complicated by the potential of using large on-site production, notably roof-top photovoltaic systems, to compensate for a large, imported energy. Here the delivered energy is low despite a large interaction with communal networks. This issue is amplified in Nordic climates, due to the combination of long cold winters with short days and a lack of sunlight. This places peak demand at a point where there is the smallest production from solar energy systems. Conversely in the summer with long daylight hours, there is a potential for large solar production when the demand is low. This large imbalance between when energy is required and when it is produced requires either long-term storage or large interaction with the grid. This interaction can be positive or negative depending on the overall demands on the grid.

Such situations can be identified by examining the energy peak load and the energy balance. Peak load and energy balance are becoming more relevant as society increasingly electrifies, putting added pressure on the electricity grid. A lower peak and good energy balance requires a smaller system to supply the energy and less interaction with the grid. Strategies to reduce peaks on a large scale can minimise the need for further energy infrastructure. In light of this, peak load has been proposed as a replacement for the current colour marks in the Norwegian energy certificate (Enova, 2019).

1.1.4 Energy flexibility

The current generation of electricity and thermal energy aims to match supply to demand. As the share of intermittent renewable generation, such as wind and solar, increases, this relationship needs to be reversed so that demand matches supply (Bleys et al., 2018). This is particularly acute for electricity grids due to the limited storage possibilities. District heating systems can use the thermal mass of the grid as a short-term storage (Balić et al., 2017).

The ability for buildings to respond to the available supply in energy grids is known as energy flexibility. Flexible buildings can shift their demand to periods of higher supply, putting reduced stress on the grid during high demand periods. Although direct demand management by power suppliers has long been practiced with large industry and office buildings, it is not practical for residential buildings with their many units and small demands (Lund et al., 2015). For consumer driven demand side control to be adopted, it has to be incentivised by cheaper energy costs, made possible by improvements in information and communications technology. Norway was the first country to implement a market-based power system in 1991 (Energifakta Norge, 2021). Since the start of 2019, all Norwegian properties now have smart meters which allow for hourly electricity pricing. The changing price can be used as a control signal for the energy system, serving as a good proxy for the level of supply in the grid.

The demand profile can be shaped to the available supply through shifting demands and energy storage. The potential for shifting demands in homes is limited to appliances which have operation times independent of the resident, such as dishwashers and washing machines. These loads can be managed by computer control but this may not suit all residents (Ahlbom, 2015). Also, the potential savings to the individual are small as the shifted energy use is relatively small. The economic feasibility of battery storage is highly dependent on energy price pattens and is lower than demand shifting due to high initial costs (O'Shaughnessy et al., 2018). The batteries in electric vehicles could be utilised instead, as they stand idle for long periods, so called vehicle-to-grid (Lund et al., 2015). However, the pricing model and technology are still in development. The extra wear placed on the battery through increased charging and discharging cycles, could mean that price responsive charging is preferred by most vehicle owners. This charges the vehicle to the required state of charge by a specified time by using an optimised charging profile to use the cheapest electricity.

As space heating and DHW represent the largest proportion of energy demand, using thermal energy storage offers considerable potential for energy flexibility in residential buildings. Thermal energy can be stored in the buildings thermal mass or water tanks (Romanchenko et al., 2018). This is the only flexibility option that closely interacts with the HVAC system and affects its design. Further background information about thermal energy storage in Nordic climates is found in Section 2.2.4.

1.2 Aim and Research Questions

This thesis was conducted as part of a research project aiming to identify cost-effective HVAC solutions for apartment buildings in Nordic climates. A comparison of a wide variety of complete solution-sets, which incorporates the current energy price structure in Norway, does not exist. In order to carry out a life-cycle cost analysis, it was first necessary to identify the solutions to be tested and calculate the effect of each parameter on the delivered energy demand and energy cost through simulation. Furthermore, the research on energy flexibility is limited with a lack of comparison of the effect on the cost of different systems (Kathirgamanathan et al., 2021). The potential for energy flexibility was assessed for the found solution-sets. These aims were investigated through the following research questions:

- What HVAC solutions are common in low-energy apartment buildings?

- What is the effect of each HVAC parameter on the delivered energy demand?
    - What is the effect of ventilation, heat emitters and system scheduling on gross energy demand and thermal comfort?
    - Which energy supply technology is most efficient?
    - Which HVAC solution-set requires the least delivered energy?
    - Which HVAC solution-set has the lowest energy cost?

- What are the additional challenges/costs related to implementing these solution-sets?
    - Can these be solved/reduced?

- Is there a possibility for providing energy flexibility through the HVAC system?
    - How does the choice of system, system size and thermal energy storage capacity affect flexibility potential?
    - What is the financial benefit to the user?

The resulting data is intended for further study using pricing data for each system component to calculate the life-cycle cost of the solution-sets, providing a resource for the construction industry to improve the energy-efficiency of new construction economically.

1.3 Scope and Workflow

The research questions are investigated through a parametric analysis of HVAC solution-sets, based on common systems for a case study in Norway. The scope of the investigation is limited to apartment buildings in Nordic climates. An apartment building was defined as a residential building with multiple units over three or more stories sharing a common vertical circulation. Nordic climate refers to the typical climate found in Norway, Sweden and Finland, summarised as a high number of heating degree days (> 3 000) at a high latitude (> 55°). Although outside the scope of this thesis, similar conditions can be found in other Baltic countries, Canada and Russia. Although part of the Nordics, Denmark was judged as having too mild a climate. Similarily, Iceland also has a different climate, as well as abundent geothermal energy allowing for other heating solutions.

The content of the thesis is present according to the devised workflow, with the research questions split into two parts. First, common HVAC solutions for Norway, Sweden and Finland were found through a statistical and literature review, presented in Chapter 2. These formed the basis for the proposed solution-sets. Second, these solutions were compared through simulation for a case study of an affordable housing project in Norway.

The solutions were divided into parameters affecting the gross energy demand of the building and those affecting the delivered energy demand of the building. The former included ventilation type, heat emitter type and scheduling. The latter included energy supply systems and thermal energy storage. These parameters were simulated using *SIMIEN7*

(ProgramByggerne, 2021) and then analysed in *Microsoft Excel* according to the methodology outlined in Chapter 3.

The parameters affecting the gross energy demand were analysed for energy use and thermal comfort, as the primary aim of the HVAC system is to provide a good indoor environment. These energy demands were used as inputs for sizing the energy supply systems and thermal energy storage. For ground source heat pump systems, additional variables were analysed, such as total borehole length and the use of a solar thermal collect array. The flexibility potential of selected solutions through management of the thermal storage was also analysed using a *Microsoft Excel* macro developed as part of this thesis. The final solutions were then compared by delivered energy demand and energy cost. The results are presented in Chapter 4, including analysis of the effect of each parameter. A holistic assessment including other practical considerations is then discussed in Chapter 5.

1.4 Limitations

The results and conclusions of this study are subject to the following limitations:

- All results are based on simulation with inputs and assumptions based on the Norwegian market.

- The models used in the simulation software and in the created spreadsheets are a simplification of reality. Certain energy uses were not accounted for or simplified as these were not calculated in the used software. Therefore, the results are missing the energy demand from the fans in the fan convectors and the pump for the borehole system.

- The results are specific to the studied case and its location; however, aspects are applicable to similar projects in similar climates.

- The climate file was a typical meteorological year weather file for the municipality (Sørum) and so did not account for any local or yearly weather variations.

- Energy prices were calculated using historical price data. This was done for comparison of solutions and the results should not be used as an estimate of actual prices or savings. Analysis was done with 8 years of data to show the possible yearly variation.

2 Literature review and analysis

A literature review was undertaken to find common, commercial, HVAC systems in apartment buildings. Previous studies to find solution-sets for building systems were based on two approaches: either a survey of existing buildings/solutions or parametric simulations. IEA SHC subtask 40 produced solution-sets from a survey of 30 NZEBs. Of the 11 residential buildings, there was just one apartment building located in Kleehauser, Germany (Garde and Donn, 2014). A study of 32 NZEBs in Europe by Concerted Action EPB had two projects each from Norway, Finland and Sweden, of which only one was an apartment building (Erhorn and Erhorn-Kluttig, 2014). Paoletti et al. (2017) analysed 411 nZEBs in 17 European countries as part of the project EU IEE ZEBRA2020. Buildings were identified through reports and energy performance certificates. The buildings were divided into three climate groups, with 234 buildings in cold climates of which 31 were in Norway and 15 in Sweden. These buildings were mainly heated using heat pumps (31 %), boilers (21 %) or district heating (25 %). Photovoltaic solar panels (PV) were present in 15 %, solar thermal collectors in 18 % and a combination of both in 11 %. Mechanical ventilation with heat recovery was present in 84 % of all 411 buildings. Due to the large geographical area and multiple building typologies covered, the results of these studies are too general to form solution-sets.

Only two studies were found which specifically looked at residential buildings in northern latitudes. The NorthPass project presented 32 low-energy residential buildings, defined as having 25 % to 50 % less energy demand than the minimum requirement, in Denmark, Estonia, Finland, Latvia, Lithuania, Norway, Poland and Sweden (Blomsterberg et al., 2012). The report recommended the use of balanced ventilation with at least 80 % heat recovery and heat pumps with a coefficient of performance (COP) higher than 3. Renewable energy could be supplied by solar thermal collectors, PV or Biomass. In general, the low-energy buildings had a lower life cycle cost over 30 years than conventional buildings. The Canada Mortgage and Housing Corporation was the only report to focus on low-energy multi-unit residential buildings (CMHC, 2017). It included 27 buildings, many from Northpass, with other examples from Canada, Greenland and Alaska. There were 10 new constructions in Norway, Sweden or Finland. These were completed before 2012 and only 4 were multi-story apartment buildings. Typical systems included balanced ventilation with at least 75 % heat recovery and hydronic distribution (radiators (73 %) or underfloor heating (23 %)). Just over half of buildings used solar thermal collectors. Only 13 % used PV. Fossil fuels provided heating in the largest number of projects, however none of the Nordic project used fossil fuels. Both these surveys suffer from a lack of data both in quantity and recentness.

The surveys above present an atomised analysis of system components rather than full solution-sets. An article by Fabrizio et al. (2014) presented systems for providing space heating, cooling, DHW, ventilation and electricity production through an expansive literature review. The systems were primarily for single family houses, but some were suitable for apartment buildings. As the review was not location specific, only some of the systems were relevant for Nordic countries such as solar assisted heat pumps and compact HVAC units.

The policies and resources of a country determine which systems are viable. The EU Horizon 2020 project ConZEBS (Cost reduction of new Nearly Zero-Energy buildings) presented cost-effective solution-sets for multi-family houses in Denmark, Germany, Italy and Slovenia (Gutierrez et al., 2019). The solutions were identified through parametric simulation of a typical building for each country. Each country had its own unique set of solutions, demonstrating that there is not a silver bullet to achieve nZEBs. However, these solutions did share some common elements such as efficient heat recovery and use of solar energy. As the project name suggests, the solutions were optimised for life cycle cost whilst achieving the energy requirements. Harkouss et al. (2019) optimised six common solution-sets for energy production to achieve a net zero-energy balance in a 6 apartment building (432.6 m²) in three difference climates. Solutions were optimised for life cycle cost, CO_2-eq. emissions, total energy consumption, and grid interaction. The optimal solution was dependent on the weighting of these factors, demonstrating the additional decision complexity when multiple objectives are involved.

The lack of information about HVAC systems in Nordic apartment buildings was surprising considering that this building typology makes up an increasing share of new residential construction in Norway, Sweden and Finland, as shown in Figure 2.1. Apartment buildings accounted for over half of the new floor area in Sweden and Finland in the previous three years.

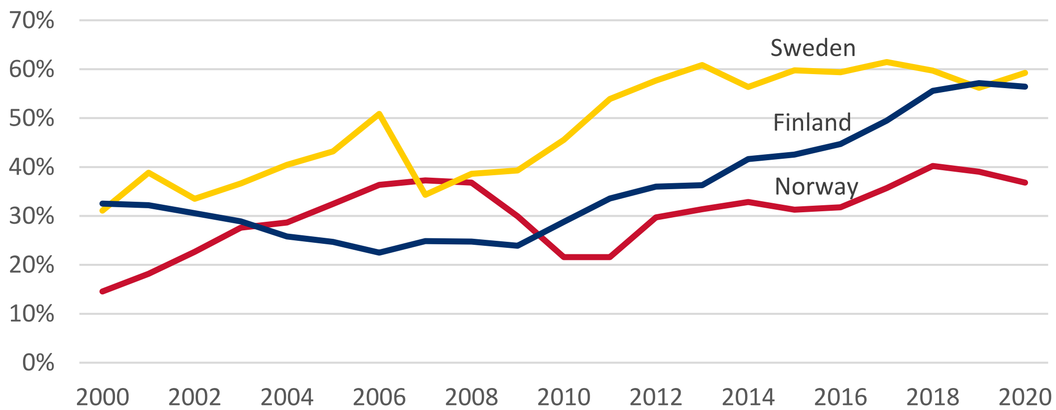

Figure 2.1. Percentage of new residential floor area accounted for by apartment buildings each year in Norway (Statistisk sentralbyrå, 2021), Sweden (Statistiska Centralbyrån, 2021) and Finland (Tilastokeskus, 2021) between 2000 and 2020.

Therefore, a statistical analysis of national energy performance certificates (EPCs) databases in Norway, Sweden and Finland was undertaken to define HVAC systems in low-energy apartment buildings from the past 20 years. The individual elements of the HVAC system were analysed, including heating systems, heating emitters, distribution, and ventilation. With a view to NZEB, the use of on-site renewable energy sources was also studied. These are discussed alongside studies and innovative systems from literature. Where possible, elements were analysed in combination to identify complimentary solutions. The intention of this study was to define potential systems for new apartment projects in Nordic countries. Therefore, the primary interest was in cost-competitive, commercially available systems.

2.1 Methodology for Statistical analysis

The review of building systems was investigated in two ways: first, through a statistical analysis of national EPC databases; and second, by examining exemplar projects with unique systems.

2.1.1 Statistical analysis

EPCs of buildings are an important instrument recommended since the original EPBD (EU, 2002). Their quality assurance was improved in the 2010 recast through stricter accreditation of assessors, independent control and penalties for noncompliance (EU, 2010). The primary aim of EPCs is to provide energy information for potential buyers and tenants in order to create market demand for energy-efficient buildings (Arcipowska et al., 2014); however, the databases can also be used to statistically analyse the building stock of a country. This has been used in a wide variety of analyses (Pasichnyi et al., 2019b) but few have investigated HVAC systems (Paoletti et al., 2017; Pasichnyi et al., 2019a; Streicher et al., 2018).

Although not mandated by the directive, Norway, Sweden and Finland have established detailed, centralised databases of these certificates. The nZEB definition was not clearly defined but open for each country to define (Sartori et al., 2012). Therefore, the requirements, inputs and primary energy factors (PEF) are different for the three countries (Kurnitski et al., 2018). As a starting point, all apartment buildings that had achieved a low-energy grade, in their respective country and construction year, were considered. The following section presents the background to these datasets, as well as the filtering techniques applied in the search of low-energy apartment buildings. As the dataset included entries comprising multiple buildings grouped under one development, results are presented in terms of heated floor area (A_{temp}). Broader context of each country's building stock and energy mix has been presented by Sirviö and Illikainen (2015).

Norway

EPC registration started in December 2009 and was mandatory for residential property sold or rented after July 2010 (Brekke et al., 2016). The rating is composed of a letter A through G, representing energy use, and five colours, which indicate the type of heating system used (Enova, 2009a). A building was classed as low-energy if it achieved the A grade. For apartment buildings, this equates to the delivered energy for heating, DHW, fans, pumps, lighting and equipment being less than 85 + (600/A_{temp}) kWh/(m²·year). The calculation is standardised to use an Oslo climate file, with fixed demands for DHW (29.8 kWh/(m²·year)), lighting (11.4 kWh/(m²·year)) and equipment (17.5 kWh/(m²·year)) (Standard Norge, 2014), with no PEFs applied.

The data from 2009 to September 2020 was anonymously registered by apartment. Buildings were created by grouping apartments registered with the same building year, region and similar upload times. The summation of delivered energy and A_{temp} was then used to determine if the building was low energy (A grade). Buildings less than 600 m² were excluded. Just under 60 % of all apartments have EPCs although potentially 10 % are duplicated entries (Enova, 2019), which could not be identified due to the anonymity. Furthermore, it is possible that some apartments were missed from buildings, and that large projects with multiple blocks were grouped together. All the entries used were simulated results with no measured data registered, although this is a possibility. Heating systems were defined using the data fields for percentage of heating demand delivered by a heating source. The highest proportion was assumed as the main heating source, with others treated as secondary sources. The DHW system was defined in a similar way. The systems were weighted by the related building's A_{temp}.

Sweden

EPCs were introduced in 2006 (Infrastrukturdepartementet RSED E, 2006) alongside the first energy requirements in the building regulations (Boverket, 2006). Registration was required for public buildings and buildings with rental units. It was made mandatory in 2012 for new buildings and those to be sold (Infrastrukturdepartementet RSED E, 2012). Energy classes were introduced in 2014. Therefore, almost all apartment buildings are registered but only 5 % have an energy class (Boverket, 2020a). The energy class is indicated by a letter A through G based on the energy delivered for heating, cooling, DHW, and building electricity, which covers fans, pumps, elevators and communal lighting in the building (Boverket, 2020b). The C grade is defined as the maximum allowed primary energy use for a new building and so changes with each update to the building regulations. B and A grades represent energy use that is at least 75 % and 50 % of the C grade, respectively. Buildings built before 1st July 2017 were assessed by energy use without any PEFs. The maximum energy use was dependent on the building's location and whether it was primarily heated by electricity or not. These requirements are presented in Table 2.1.

Table 2.1. Maximum allowed energy use for Swedish building regulations from 2012 to 2017.

Years in use	**2012 – 2015**	**2015 – 2017**
Building regulation	BBR19 (Boverket, 2011)	BBR22 (Boverket, 2015)
Zones	1 / 2 / 3	1 / 2 / 3 / 4
Non-Electric	130 / 110 / 90	115 / 100 / 80 / 75
Electric	95 / 75 / 55	85 / 65 / 50 / 45
Ratio Non-Electric to Electric	1.37 / 1.47 / 1.64	1.35 / 1.54 / 1.60 / 1.67

Since 1st July 2017 (Boverket, 2017a), the primary energy has been calculated using the following formula:

$$Primary\ Energy = \frac{\left(\frac{E_{heating}}{F_{geo}} + E_{cooling} + E_{DHW} + E_{building}\right) \times PEF}{A_{temp}}, \quad (1)$$

where F_{geo} is a geographical factor for each municipality ranging from 0.8 to 1.9. Stockholm has a geographical factor of 1. This simplified the energy requirement to a single number but introduced PEFs, shown in Table 2.2. Since 2017, apartments must have a primary energy less than 85 kWh/(m²·year). From 2020, the maximum value was reduced to 75 kWh/(m²·year) (Boverket, 2020c) and PEFs changed. The energy use can be simulated or measured (Boverket, 2017b). When measured, it is normalised to a standard year to discount variations in heating hours.

The Swedish database included data from 2012 until the end of 2020. It was the most detailed of the three, allowing for a high degree of filtering. Mixed use buildings that were at least 85 % residential were included. Duplicate entries and buildings with two floors or less were removed. The majority of these were terrace houses or low-rise projects, which were outside the scope of this study. Heating systems were defined using the data for the delivered energy for each heating source. The highest energy was assumed as the main heating source, with others treated as secondary sources. In the few cases where an entry used both a heat pump and district heating requiring similar energy, the heat pump was considered as the main heating source as it would deliver more energy to the building when accounting for its COP. DHW systems were defined in a similar way for data after 2017. Only the proportion of energy for DHW was available for data before 2017. The systems were weighted by the related building's A_{temp}.

Table 2.2. Primary energy factors for Swedish and Finnish EPCs.

	Sweden (2017) (Boverket, 2017a)	**Sweden (2020)** (Boverket, 2020c)	**Finland (2013)** (Ympäristöministeriö, 2018)	**Finland (2018)** (Ympäristöministeriö, 2018)
Electricity	1.6	1.8	1.7	1.2
District heating	1.0	0.7	0.7	0.5
District cooling	1.0	0.6	0.4	0.28
Fossil fuels	1.0	1.8	1.0	1.0
Biofuels	1.0	0.6	0.5	0.5

Finland

EPCs have been in use since 2008 but were significantly reformed in 2013 and then updated in 2018 (Ympäristöministeriö, 2018). Energy classes range from A to G with the same boundaries for both 2013 and 2018. An A building is one with a primary energy less than or equal to 75 kWh/(m²·year). This includes the primary energy for heating, DHW, ventilation, cooling, equipment, and lighting simulated for the climate of Helsinki, with fixed demands for DHW (35 kWh/(m²·year)), lighting (9.6 kWh/(m²·year) in 2013, 7.9 kWh/(m²·year) in 2018) and equipment (21.0 kWh/(m²·year)) (Ympäristöministeriö, 2017). The PEFs for 2013 and 2018 are shown in Table 2.2.

The dataset included data from 2013 to the end of 2020. Details about the heating system and ventilation were entered as text fields. These were converted into a select number of system descriptions. For 26 cases where the energy source was not specified, it was determined based on the supplied energy which was entered as either electricity or district heating. If a large portion of the energy came from district heating, this was deduced as the heat source. If the building was only provided by electricity, a ground source heat pump was specified. This corresponded with the pattern seen in the rest of the data. The systems were weighted by the related building's A_{temp}. The proportion of DHW and heating demand was not specified.

2.1.2 Exemplar buildings

Other environmental certificate schemes and organisations promoting low-energy buildings were examined for projects with unique solutions. These sources are outline in Table 2.4 and the selected projects are detailed in Table 2.3. There were no apartment projects in the three countries certified under Passive house (Passivhaus Institut, 2021a), Well building standard (International WELL Building Institute, 2021), Living building challenge (International Living Future Institute, 2021) or DGNB (DGNB System, 2021). Additional literature was found using the Scopus database, as well as looking at relevant technical journals and conferences. Several of the Swedish buildings appeared in multiple databases.

Table 2.3. Projects with unique systems referenced in this section.

Name in paper	Country	Location	A_{temp} / m²	Construction year	Source
Løvåshagen	Norway	Bergen	1 875	2008	(Blomsterberg et al., 2012; CMHC, 2017)
Kringsjå (student apartments)	Norway	Oslo	6 009	2019	(Futurebuilt, 2016a; Haugen, 2017)
Klosterenga	Norway	Oslo	1 300	2000	(Blomsterberg et al., 2012; CMHC, 2017)
Gullhaug torg (mixed used)	Norway	Oslo	N/A	TBC	(Futurebuilt, 2016b)
Sjögången	Sweden	Karlstad	8 179 in 4 blocks	2011	(Lågan, 2019; Sweden Green Building Council, 2021)
Vallastaden	Sweden	Linköping	1 451	2017	(Lågan, 2019)
Lärkträdet	Sweden	Vara	1 242	2010	(Andersson et al., 2012)
Brofästet	Sweden	Stockholm	N/A	2019	(Stockholms Stad, 2019)
Heka Kaljaasi	Finland	Helsinki	5 597	TBC	(Helsinki, 2020)
Kuopio	Finland	Kuopio	2 138	2011	(Pesola et al., 2016)

Table 2.4. Overview of searched certification schemes and organisations promoting low-energy buildings.

Certificate scheme / Organisation	Description	Online Database	Relevant countries	Level to be considered low-energy	Number of relevant buildings
LEED	International standard from USA, based on American regulations.	(Sweden Green Building Council, 2021)	All	Gold and above	3 gold in Sweden
BREEAM	International standard from UK. Adapted to local standards in Norway and Sweden.	(BREEAM, 2021)	All	Very good and above	4 very good and 1 excellent in Sweden; 7 very good in Norway
Futurebuilt	Pilot projects for achieving net zero-energy and net zero emissions buildings.	(Futurebuilt, 2016a)	Norway	N/A	8 residential projects of which only 1 is completed
Miljöbyggnad	Swedish certification scheme administered by the Swedish Green Building Council	(Sweden Green Building Council, 2021)	Sweden	Gold	20 projects accounting for 45 entries
FEBY	Swedish passive house standard	Some example projects (FEBY, 2018)	Sweden	N/A	35 reported with reports on 8 projects
Lågan	National initiative to increase rate of construction of low-energy buildings	(Lågan, 2019)	Sweden	N/A	63 projects. 9 are found in the Swedish EPC
ARA	Finnish housing association	Proect report (Pesola et al., 2016)	Finland	N/A	2

2.2 Results and Discussion of Statistical analysis

Due to the differences outlined in the methodology, the results of each country should be viewed independently. However, it was judged that the inputs and climates (Oslo, Stockholm and Helsinki) are close enough that a degree of comparison was possible. The Norwegian results are based on 347 entries, generated according to the methodology, with 1 070 007 m² A_{temp}. The Finnish results are based on 221 entries accounting for 831 445 m² A_{temp}. Only one entry was found under the 2013 regulations and was grouped with the other entries. The Swedish results are based on a total of 742 entries accounting for 2 392 538 m² A_{temp}. These were distributed over four different sets of regulations: 2012 (63 entries, 221 836 m²); 2015 (195 entries, 685 821 m²); 2017 (301 entries, 924 002 m²); and 2020 (183 entries, 560 879 m²). Of all 742 entries, 413 were measured and 329 were simulated.

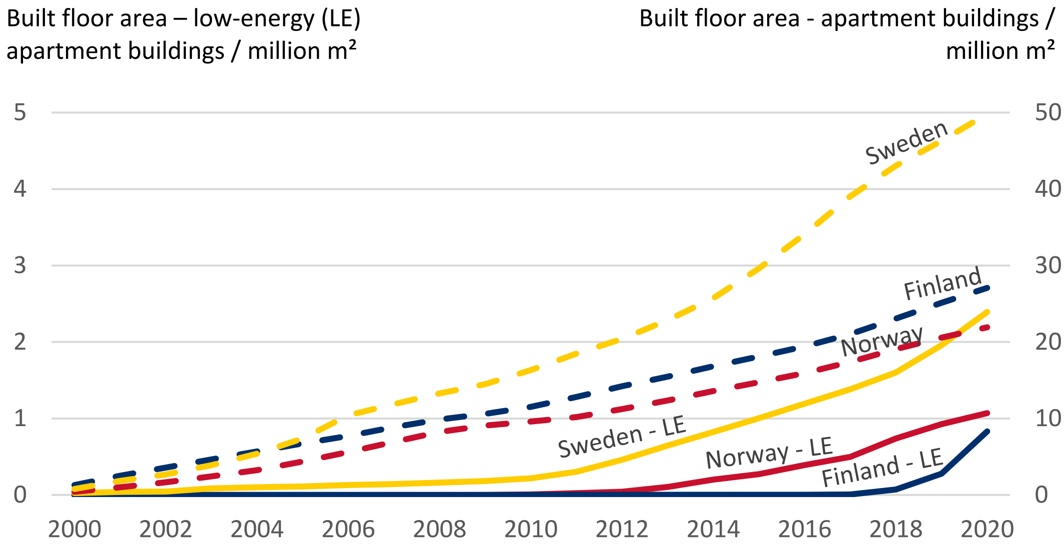

Figure 2.2. Cumulative built floor area of apartment buildings and low-energy apartment buildings since 2000.

In the last 20 years, low-energy apartment buildings have accounted for 4.9 % of built floor areas in Norway, 4.8 % in Sweden and 3.1 % in Finland. Most of this construction has taken place in the last 5 years at an exponential rate, shown in Figure 2.2. Since 2018, 8 % of new apartment buildings were low-energy in Finland (ARA, 2018a). The trend for Norway was slightly suppressed due to data only being available up till September 2020. Finland's growth was concentrated in the years since 2018, when the PEFs were reduced making an A grade more achievable. Annual new floor area for all apartment buildings has been relatively constant. Only Sweden showed an increased construction rate starting in 2014. Therefore, if these trends continue, low-energy apartment buildings will increase their market share. The fact that Sweden has around double the floor area of Norway or Finland was understandable in relation to the populations of each country.

2.2.1 Heating systems

The results are presented as the A_{temp} supplied by each type of system. The amount of energy used is not accounted for, in effect equalizing the system performance. For the Swedish and Finnish data, only delivered energy was available which would penalise heating systems with high working efficiencies such as heat pumps. Conversely, a direct count would penalise district heating as this was often the source for larger buildings. Therefore, the presented results indicate the popularity of a system, not its efficiency.

Norway

Figure 2.3 shows the distribution of heating systems in Norwegian low-energy apartment buildings. The main source for both heating and DHW was a heat pump (HP), often in combination with direct electrical heating (EL) to cover peak loads. It was not specified what type of heat pump was used. The average heat pump COP was 2.5, ranging from 1.8 to 5.0. District heating (DH) was the main source for 15 % of DHW demand and 20 % of heating demand, mostly with EL and HP as secondary sources. Direct electrical heating was the main source for 3 % of DHW demand and 6 % of heating demand. The majority of this was with a HP as a secondary source. Two entries used a combination of solar thermal collectors (SC), HP and gas to supply DHW. Two entries used a combination of biofuels (Bio), HP and EL for heating.

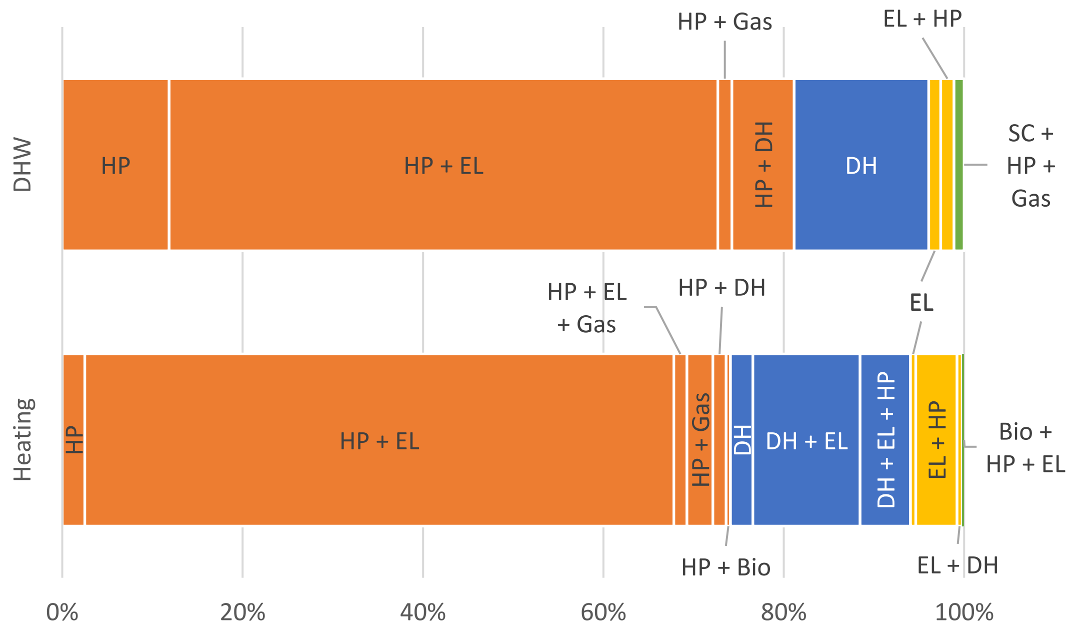

Figure 2.3. Proportion of heating and DHW systems in Norwegian low-energy apartment buildings, weighted by A_{temp}.

The systems used in low-energy apartment buildings can be roughly compared to all registered apartments by looking at the relative distribution of heating colour assigned in the EPC database. Figure 2.4 presents data from the entire database (Enova, 2018) and the typical systems for each colour. These typical systems are not strictly assigned, as 32 % of A-grade apartments have a dark green colour despite only 22 % utilising district heating as a primary or secondary source. The 3 % with a red colour matches with the 3 % to 5 % of systems with direct electric or gas. Such systems are the most common in the Norwegian housing stock. A-grade apartments make up a high proportion of apartments with a yellow colour, indicating the prevelance of heat pumps was particular to low-energy apartment buildings. It is likely that most of the heat pumps were ground source heat pumps (GSHPs) based on the typical systems for light green and yellow.

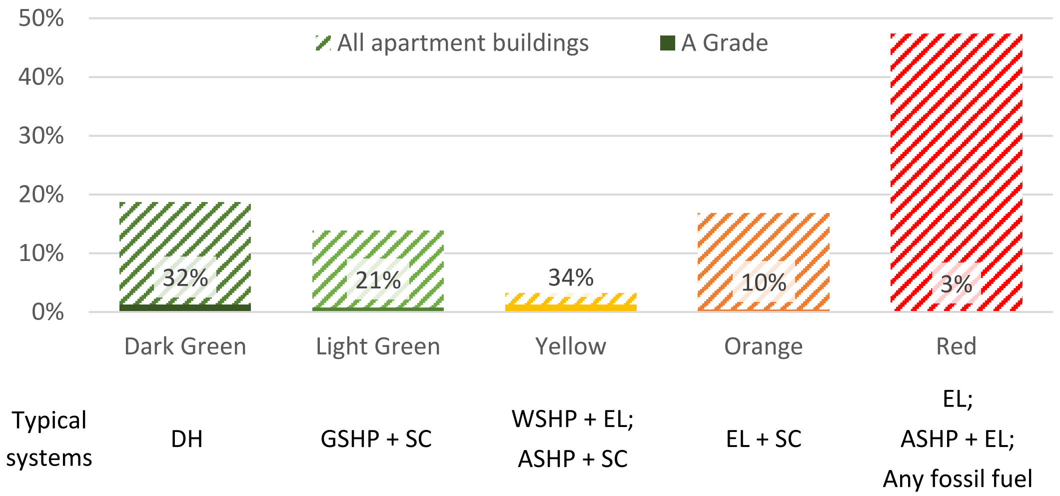

Figure 2.4. Energy performance colour, indicating typical heating systems, for all apartment buildings and low-energy apartment buildings in Norway. Percentage shows proportion of A-grade apartments in each colour.

Sweden

Systems providing heating and DHW to Swedish low-energy apartment buildings are presented for the four regulation periods in Figure 2.5. In the 2017 and 2020 data, heating and DHW were entered as separate values. Systems with district heating as the primary energy source provided the most A_{temp}. Under 2012 regulations, 94 % used DH, with half of these utilizing an exhaust air heat pump (EAHP) as a secondary source. The proportion reduced to 86 % for 2015 regulations, with 45 % of that utilizing an exhaust air heat pump. At the same time, the proportion of GSHP systems increased. The results for 2017, which accounts for the largest share of A_{temp}, were 62 % DH systems, 33 % GSHP systems. The trend is reversed for 2020 with 76 % DH and 24 % GSHP, likely due to the change in PEFs. The systems for DHW closely match those for heating with a slightly higher proportion of DH. The dataset did not distinguish between EL or GSHP for DHW, however it is likely that most of the electricity is used for GSHPs. This was justified by a very strong correlation between the primary heating source for heating and DHW with over 94 % of entries matching. In the remaining cases, the primary heating source for DHW was the secondary source for heating or vice versa.

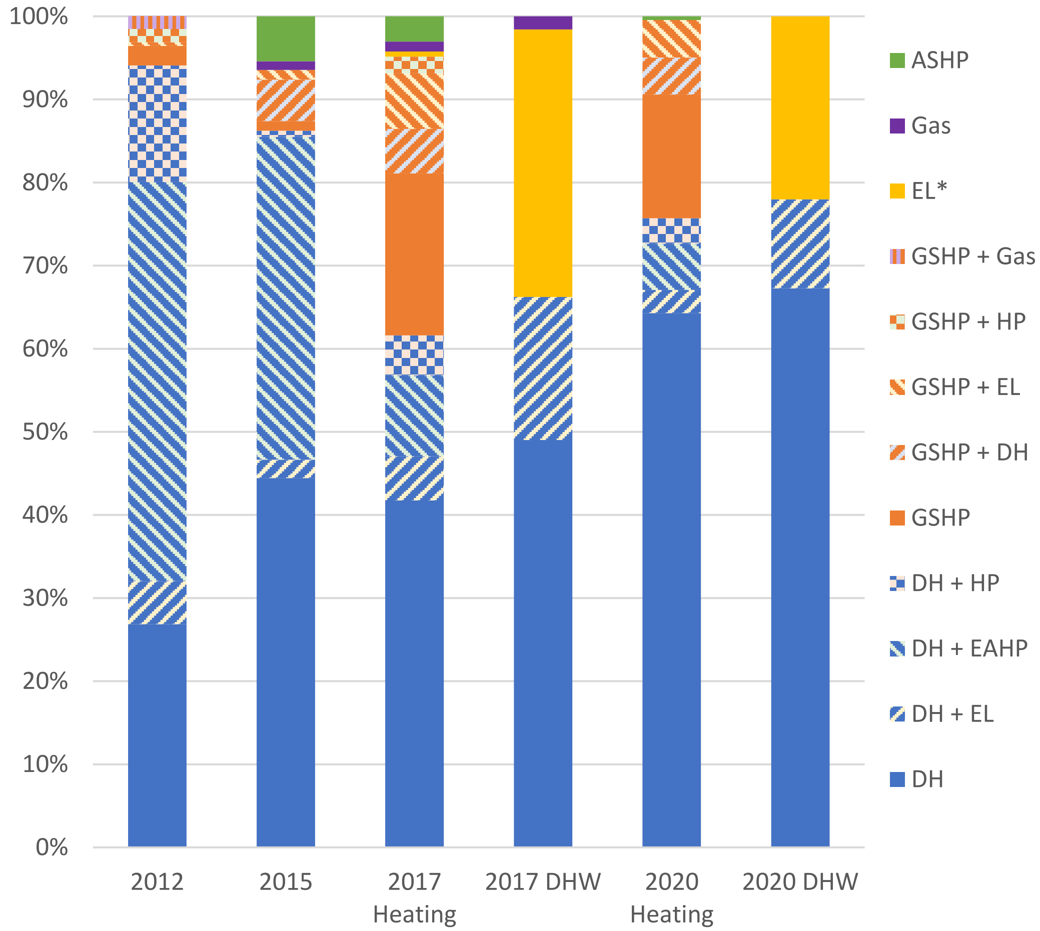

Figure 2.5. Distribution of heating systems in Swedish low-energy apartment buildings by regulation year, weighted by A_{temp}.
** The data does not differentiate between direct and indirect use of electricity for DHW. As such, heat pumps are included in this category for DHW.*

Finland

Heating and DHW for Finnish low-energy apartment buildings was divided between two primary sources: district heating (75 %) and GSHP (25 %). Secondary sources were only detailed for some entries.

Heating sources

All three countries have the same primary heating sources, DH and GSHP, just in different proportions. Direct electric systems were a common secondary source in Norway and Sweden likely due to their simplicity, instant start and stop and low construction costs (Medved et al., 2019). They can be applied as electric panel emitters, heating coils in the ventilation system or as water heating. Such systems are only viable as the primary heating source when the demand is very low, such as in a passive house. These were only present in Norwegian low-energy apartment buildings, likely due to the higher PEF in Sweden and Finland.

Heat pump systems overcome the electricity PEF handicap due to their ability to produce more work than the electricity put in, i.e. their coefficient of performance (COP), by extracting heat from an ambient source (Medved et al., 2019). This heat source can be air, water or ground. The highest proportion of heat pumps as a primary heat source was in Norway. Most of the heat pump systems were ground source systems which have higher COPs but also higher capital costs than other types due to need to drill boreholes. GSHPs can still be cost-effective for apartment buildings as the capital cost is shared between many units (Persson et al., 2014). An advantage of GSHPs is that the boreholes can be used in the summer for free cooling. Energy for cooling was delivered to only four Norwegian, four Finnish and sixteen Swedish projects demonstrating little need for dedicated systems. It was not possible to identify projects utilizing free cooling, as these would not require any delivered energy except to the pumps. Free cooling, available at almost no extra investment cost, improves further the advantages of GSHP for thermal comfort and property sales price (Karytsas and Choropanitis, 2017). It could even reduce costs as the better annual energy balancing can allow for a reduced borehole length (Javed et al., 2019). Exhaust air heat pumps were a common secondary source in Swedish low-energy apartment buildings. As an air source heat pump, EAHPs have lower capital costs but have improved efficiency due to using the warmer exhaust air as a heat source. However, they are limited by the airflow of the building.

District heating was a common solution, particularly in Finnish low-energy apartment buildings. District heating can utilise many energy sources at a large scale, benefitting from a higher efficiency and reduced emissions from any combustion-based sources through using flue scrubbers (Medved et al., 2019). District heating often has multiple heat sources providing greater energy security than local sources as the breakdown of one source can be compensated by another. Energy security is an important qualitative aspect in countries where lack of heating can be a serious problem. The grid network itself can act as a thermal energy store able to buffer peaks in demand (Balić et al., 2017). An emerging development is low temperature district heating (LTDH), which adds further advantages (Abugabbara et al., 2020). Heat losses are reduced while requiring lower costs for pipes and insulation (Buffa et al., 2019). Such networks can better utilise low temperature heat sources such as waste heat and solar thermal collectors. However, with grid temperatures below 50 °C, local heat pumps are required in each building substation to step up the temperature for DHW and for heating for grid temperatures below 35 °C. The heat pump would function similarly to a GSHP but with less capital cost as no borehole drilling is required.

Systems utilizing gas were present in 17 entries for Norway and 14 for Sweden. In Norway these were all buildings built before 2017 when fossil fuel heating for new buildings was banned (DiBK, 2017). Biomass boilers are an alternative, but only three examples were present in the results. Although favourable in terms of PEF, it is still a combustion boiler and so adds to urban air pollution (Williams, 2012).

Monovalent vs Bivalent

Bivalent systems were most common in Norwegian low-energy apartment buildings as shown in Figure 2.6. Norway also had the highest proportion of trivalent systems. Finland was 99 % monovalent systems, although some secondary sources may not have been included in the data. One of the multivalent entries had the lowest primary energy of the Finnish low-energy apartment buildings. This student dormitory with 47 apartments built in **Kuopio** was the first zero-energy apartment in Finland, utilising solar thermal collectors in combination with district heating and PV. Cooling was provided by free cooling from the ground, distributed by ventilation (Pesola et al., 2016). Another multivalent entry was **Heka Kaljaasi**, using a hybrid between GSHP, district heating and sewage heat recovery in Helsinki (Helsinki, 2020). This project is still in the planning phase. The data for Sweden showed a shift from multivalent to monovalent systems during the last decade. Where specified, DHW systems were more often monovalent than heating systems. As the DHW load varies less over the year, there is less need for an auxiliary source to cover a peak load.

Systems with more than one energy source offer increased efficiency through flexibility (Fabrizio et al., 2014). Often bivalent systems are comprised of a larger baseload system (GSHP) combined with a cheaper peaking system (EL). The cost of the additional system can be compensated by the reduced size of the baseload system compared to a monovalent setup. Alternatively, it can be cost optimal to work the system as a hybrid using the cheapest energy source at the point in time (Kensby et al., 2017). This works best when the heating sources purchase different types of energy, e.g. heat for district heating and electricity for heat pumps.

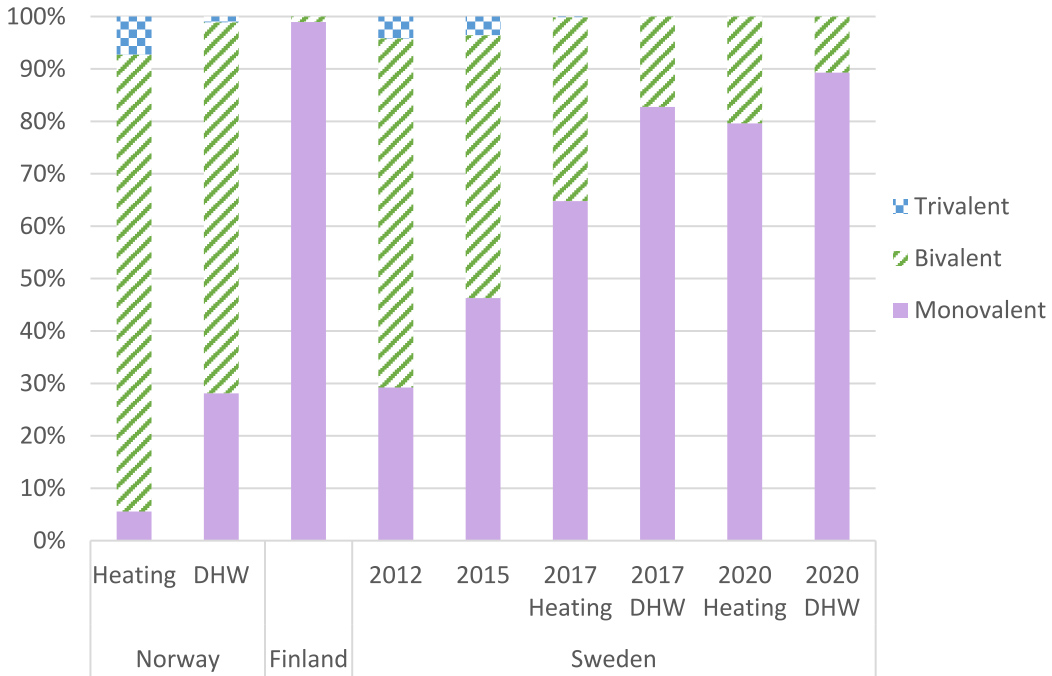

Figure 2.6. Proportion of A_{temp} supplied by monovalent, bivalent or trivalent systems in Norwegian, Swedish and Finnish low-energy apartment buildings.

DHW vs Heating

Data for the heating and DHW demand was available for Norwegian low-energy apartment buildings registered since 2016. Figure 2.7 shows that the portion of heating demands have slightly reduced in the past five years, meaning that DHW accounts for the majority of thermal energy demand. A similar trend can be seen for delivered energy for heating and DHW in Sweden in Figure 2.8, with DHW consistently accounting for 40 % or more of delivered energy in the last 6 years.

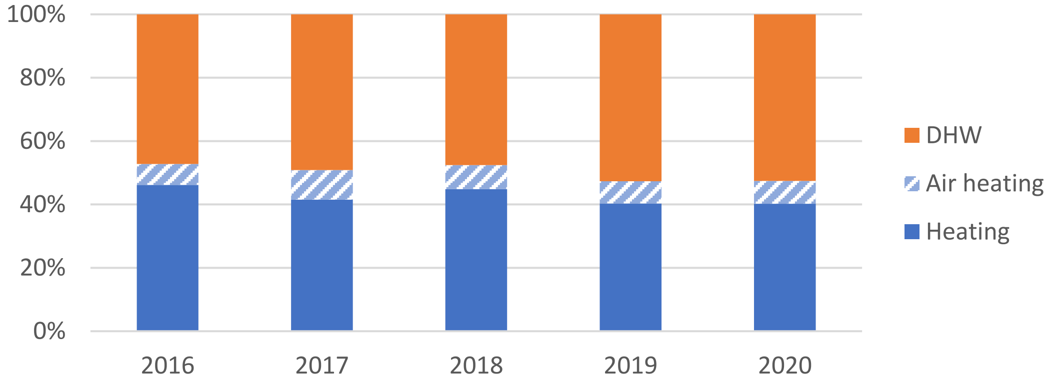

Figure 2.7. Proportion of heating demand for heating, ventilation heating and DHW for Norwegian low-energy apartment buildings by year of construction.

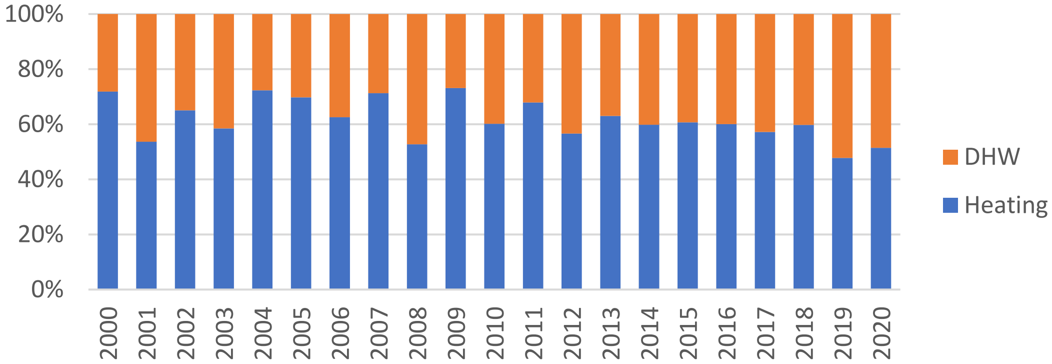

Figure 2.8. Proportion of delivered energy for heating and DHW for Swedish low-energy apartment buildings by year of construction.

This trend is likely due to improvements to the building envelope and heat recovery ventilation. This could mean that using heating sources optimised for DHW production become more relevant such as heat pumps with CO_2 as the working fluid. CO_2 heat pumps have a transcritical heater cycle, making them better suited to higher temperature differences and therefore ideal for heating DHW (Maratou et al., 2012). CO_2 is environmentally much better than standard refrigerants, with a global warming potential of just 1. When used for DHW and heating, a CO_2 heat pump outperforms the best conventional heat pump, if DHW share is between 45 % to 55 % of the energy need (Wemhoener, 2011).

Alternatively, DHW demand could be reduced using wastewater heat recovery. This still appears uncommon, with only four Finnish EPCs mentioning wastewater heat recovery. These systems preheat the cold water supply using a heat exchanger placed in the drainage system either locally or centrally, as a flow-through system or storage system (Medved et al., 2019). In optimal conditions it is possible to achieve 40 % to 60 % exchange efficiency, however the total system efficiency and cost effectiveness are dependent on the usage profile, energy cost and incoming water temperatures (Pomianowski et al., 2020). Kayo et al. (2019) analysed the exergy of a centralised wastewater recovery in a Stockholm apartment building and found that only 3 % to 24 % of the exergy was recovered when the electricity for the heat exchanger pump was included. Local heat recovery of shower waste water offers the best potential savings, as showering accounts for a significant portion of DHW demand and requires constant hot water supply for several minutes while draining (Bertrand et al., 2017). In a Norwegian exemplar project, **Kringsjå**, flow-through systems were installed in the shower of each apartment. This saved up to 30 % of the energy required for showering (Haugen, 2017). Alternatively, wastewater could be used to preheat ventilation air as a cost comparable alternative to free heating from a borehole system (Nourozi et al., 2019). A key issue for all of these systems is the fouling of the heat exchanger, reducing its efficiency and requiring regular maintenance (Pomianowski et al., 2020).

2.2.2 Heating emitters and Distribution

The Finnish data contained details of the heat emitters installed for most entries. These are presented in Figure 2.9 according to the heat source. There was a tendency to pair district heating with radiators and GSHP with underfloor heating, although all combinations were present in both. This was logical as the working temperatures of radiators pair better to that of district heating; similarly, underfloor heating to GSHP. A different type of emitter was used in the bathroom for 36 % of district heating systems and 28 % of GSHP systems. The reason for this is likely cost or desire for more control between the bathroom and the rest of the living space. Underfloor bathroom heating is a comfort issue with many using it year round (Berge and Mathisen, 2013).

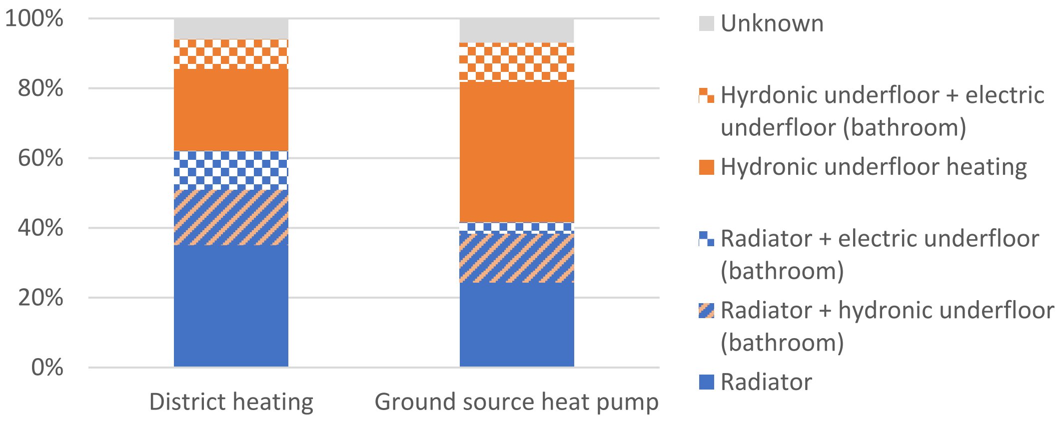

Figure 2.9. Proportion of A_{temp} using combinations of radiators and underfloor heating in Finnish low-energy apartment buildings, according to heat source.

The improved thermal comfort of underfloor heating is due to an ideal temperature gradient, warmer at the feet and colder at the head. By using the large floor surface, heating can be provided at lower setpoints, however this limits the responsiveness of the system. Radiators allow for faster response making their control more user friendly.

Hydronic distribution allows for the use of efficient heating sources but has a higher initial cost when compared to direct electric heating solutions (COWI, 2012). Electric radiators and underfloor heating are a common solution in Norway due to the plentiful supply of sustainable electricity, however these are now being discouraged out of concern for the electricity grid (Kipping and Trømborg, 2015). The low heat losses of low-energy apartment buildings, can allow for simplified hydronic solutions as it is not necessary to place a heat emitter in each room (Georges et al., 2016). One early Norwegian project, **Løvåshagen**, used a simplified system of underfloor heating in the bathroom and a single radiator on the outside of the bathroom wall that heats the rest of the apartment (Dokka and Amdahl, 2008). These are connected in series to an apartment-based accumulator tank and use the same working temperature. This simplified system is only slightly more expensive than a direct electric heating system.

Distribution of heat via the ventilation system was present in some older projects in Sweden. This simplifies the heating system but requires increased ventilation rates and higher supply air temperatures, resulting in an enlarged ventilation system. There has been renewed interest in air distribution due to the opportunities offered by highly insulated buildings with low heating demands (Javed et al., 2021). Even then, this is only possible in the milder Nordic climates and can result in uneven temperature distribution as the higher supply temperatures hinder air mixing (Georges et al., 2014). One interesting solution is **Lärkträdet** near Gothenburg. The four story passive house uses only air heating, supplied via the hollow concrete floor/ceiling element (Andersson et al., 2012). The concept utilises the thermal mass of the concrete to provide even heating to the space, similar to hydronic underfloor heating. The system also allows for night cooling of the slab in the summer which then cools air during the day. Therefore, the thermal mass regulates internal temperatures throughout the year. The system has been used in 20 residential projects in Sweden (TermoDeck, 2021a), two achieving the Passive house standard and two achieving a gold level in Miljöbyggnad certification (TermoDeck, 2021b).

Of the 113 Finish projects with radiators, 14 had details on the working temperatures: three had a supply / return of 60 °C / 30 °C, four had 50 °C / 30 °C, six had 45 °C / 30 °C and one had 45 °C / 35 °C. Of the 90 entries with hydronic underfloor heating, five had details on working temperatures. All had a temperature difference of 5 °C with four using a supply / return of 35 °C / 30 °C. Although not statistically significant, these numbers give an idea of the range of working temperatures. Supply temperatures of 50 °C to 60 °C are typical medium temperature systems matching with temperatures commonly supplied from district heating networks. Low temperature heating with supply temperatures at 45 °C or less are possible with lower heating demands. Low temperature heating reduces heat losses due to the smaller temperature difference with the ambient air. Heat losses from a 40 °C / 35 °C pipe system will be half of a 55 °C / 45 °C system (Kempe, 2013). The low temperatures can be supplied by renewable sources directly or a low temperature district heating network, which results in greater energy and exergy efficiency (Hesaraki et al., 2015b). Heat pumps have higher COPs due to the lower temperature rise required.

Low temperature heating can allow for a dual tank system with a low temperature storage tank for heating and high temperature storage tank for DHW. This is particularly useful for the storage of heating from solar thermal collectors in solar combi-systems. Two tank temperatures allow for greater efficiency through cascading, immersion or even decentralised distribution, with smaller high temperature tanks located close to demand which step up the water temperature from the low temperature tank. A system with decentralised electric reheating of DHW can result in 25 % final energy savings, but currently costs significantly more than an industry standard central system, resulting in a higher levelized cost of heat (Backes et al., 2018). Combining decentralised units with local wastewater heat recovery could improve cost-effectiveness (Pomianowski et al., 2020), but such systems would still require more maintenance (Hellgren and Olsson, 2012).

2.2.3 Ventilation

The distribution of ventilation systems in Swedish low-energy apartment buildings is shown in Figure 2.10 for the four regulation periods. There was a clear growth in balanced systems with heat recovery (HR) from 34 % in 2012 to 83 % in 2020. Buildings with multiple systems made up 7 %, 6 %, 13 % and 15 % in each of the respective periods. This could suggest that there is an increase in mixed-use buildings achieving a low-energy grade. All mixed systems had some HR, but it was not possible to determine the proportion from the dataset. Exhaust air systems, decreased from 58 % for 2012 to 6 % for 2020, with almost all including HR, except for 2017 where it was just over half of exhaust systems. These closely mirror the percentages for heating systems with EAHPs in Figure 2.5. When the results for 2012 and 2015 were filtered to just those that achieve the 2020 regulations, the percentage of balanced systems with HR is greatly increased, showing the efficiency of this type of system.

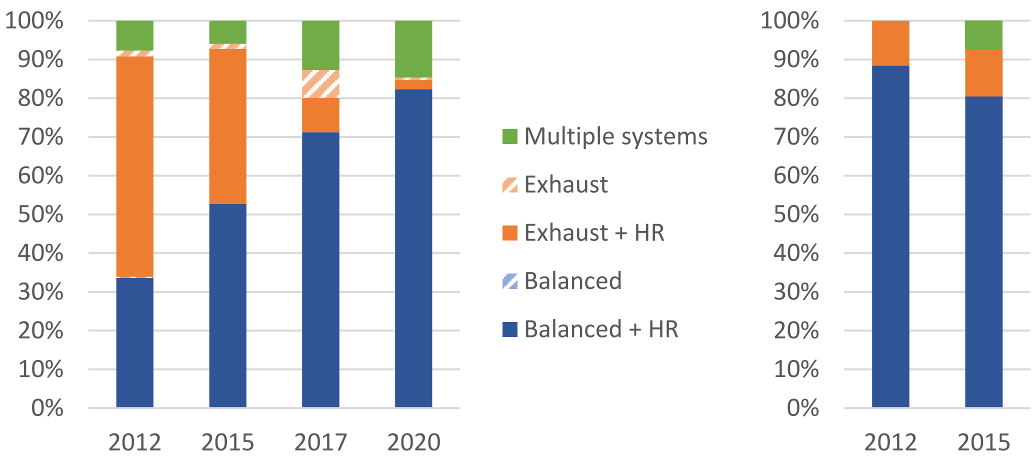

Figure 2.10. Proportion of A_{temp} served by different ventilation systems in Swedish low-energy apartment buildings, according to regulation period (left) and converted to 2020 regulations (right)

This was further supported by the data from the other two countries. Balanced ventilation systems were the only type mentioned in the Finnish data. The details for each entry varied as these were entered as text at the discretion of the assessor. It is likely that most had heat recovery with 76 % mentioning it explicitly. Only 24 Norwegian entries indicated a ventilation system, all of them balanced. Of the Finnish low-energy apartment buildings, 23 % were detailed as apartment-based and 26 % as centralised systems.

The CoNZEBs project identified apartment-based ventilation as a key technology because the shorter duct lengths compared to a centralised unit should reduce fan energy (Gutierrez et al., 2019). However, in the Finnish dataset the average energy use of apartment-based systems, 73.5 kWh/(m²·year), was slightly higher than centralised, 72.1 kWh/(m²·year). Correct placement of apartment-based systems and their inlets and outlets is critical to avoid the issues of noise and odour transfer between apartments. Apartment-based balanced systems can be combined with a heat pump between the supply and exhaust ducts. This solution, known as a compact unit, is common in individual passive houses. The heat pump can be used to heat water for heating and DHW, or it can operate between the two airflows as a heating coil (Ferrara et al., 2014). If reversible this can also allow for cooling. Such systems can struggle in very cold environments due to lower exhaust air temperatures after heat recovery, requiring a secondary heat source for peak loads.

Exhaust systems require half the fans and ducts of balanced systems and so are cheaper with lower fan energy use. However, the lack of heat recovery results in higher heat losses and heating demand. Part of these losses can be recovered by pairing the system with an EAHP which can provide hot water for heating and DHW, as is common in the Swedish low-energy apartment buildings. The COP of the heat pump is high due to the high air source temperature. One of the negatives of exhaust systems, is the lack of control over incoming air requiring larger heat emitters (Kempe, 2013). In cold weather this can lead to drafts. Ventilation windows and radiators are a potential solution as they provide a more controlled infiltration of air with heating.

Ventilation radiators simply allow the air to enter behind a radiator for instant heating. Although common in Sweden (Iivonen, 2019), no confirmed examples were found in low-energy apartment buildings. The airflow allows for better convective heat transfer requiring a smaller panel area or lower temperatures than a traditional radiator (Myhren and Holmberg, 2009). In lab measurements, ventilation radiators were found to use 17 % less energy compared to conventional radiators, due to the lower working temperature allowing for a higher heat pump COP (Hesaraki et al., 2015a). However, the heating output is dependent on the temperature difference between the outside air temperature and the radiator. The higher outputs only occur at the lowest temperatures, which only account for a portion of the heating season. Output can also be affected by window opening reducing the airflow through the ventilation radiators (Ploskić et al., 2019). Another drawback is the need to replace filters in many decentralised products.

Ventilation windows passively heat the incoming air as it passes between the glass of the window. In turn this reduces the heat losses through the windows which are the weakest part of the thermal envelope. This was calculated to reduce ventilation heat losses by 65 % in a Danish climate resulting in a 19 % energy saving (Heiselberg et al., 2013). McEvoy and

Southall (2005) measured such windows in Denmark achieving U-values of 0.69 W/(m²·K) at night and delivering average preheating of 41 % of the required load. In Poland, the same window achieved 0.52 W/(m²·K) and 35 % preheating due to a higher airflow rate. The disadvantage of these windows is that the incoming air temperature is still low enough to affect thermal comfort. Radiators positioned under the windows, due to the low demand of the building, were not powerful enough to overcome this effect. However, Hu et al. (2020) are investigating the addition of a phase changing material panel to help with air preheating. This solution also has a limited pre-cooling effect in the summer. No research was found using the windows in colder climates, but there are references to projects in Norway. One of these is a low-energy project but not an apartment building, where the windows preheat the outside air to between 10 °C to 14 °C (Førland-Larsen and Forsberg, 2018). The windows are typically 4000 DKK (approximately 540 Euros) more expensive than standard windows (Ventilationsvinduet, 2021).

The CoNZEBs project also recommended hybrid solutions where the mechanical ventilation is reduced during the summer by using natural ventilation. The planned **Gullhaug torg** in Oslo utilises an advanced hybrid ventilation as part of a simplified HVAC system (Myrup et al., 2018). However, this has required greater design work including computational fluid dynamic and laboratory studies.

2.2.4 Use of on-site renewable energy

The use of on-site renewable energy is not mandated by the regulations of the three countries (Erhorn-Kluttig and Erhorn, 2018). In the Swedish data, 11 entries used Solar thermal collectors (SC), 196 used Photovoltaic panels (PV) and 15 used both SC and PV. Figure 2.11 shows most systems were installed under the last two regulation periods and that the proportion of PV is increasing. A higher percentage of A_{temp} heated by GSHP utilises solar technologies than those heated by district heating (DH). The decreasing proportion of SC mirrors an international trend, due to market pressures from falling prices for heat pumps and PV (Weiss and Spörk-Dür, 2020).

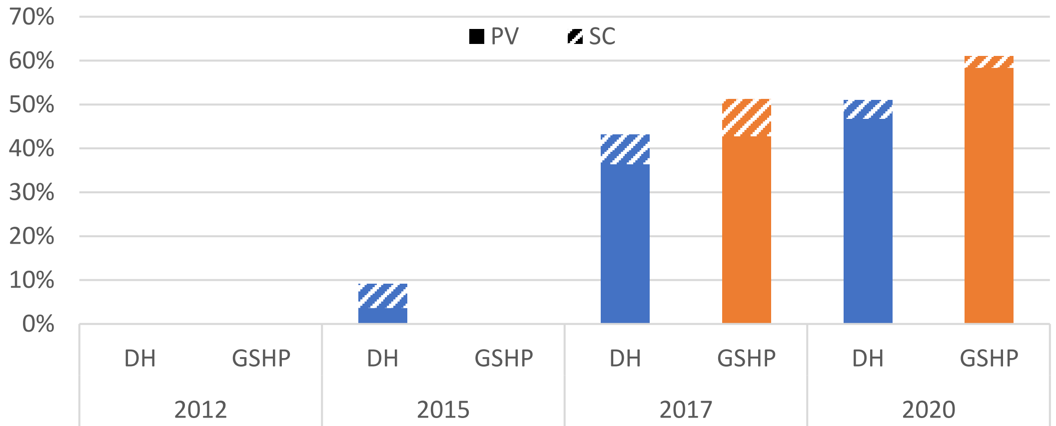

Figure 2.11. Proportion of A_{temp} using solar technologies in Swedish low-energy apartment buildings divided by regulation period and primary heating source.

Solar thermal collectors were used by 1 Finnish and 2 Norwegian entries. It was unknown if the Norwegian or Finnish entries used PV as the datasets lacked this data, however it is likely that the proportion would be low based on national statistics (Ahola, 2019; Westgaard, 2018). In Finland, adoption of solar technologies has been hindered by: a vested interest in the current energy system; development of biomass generation; and a lack of faith in solar technologies meaning few subsidies, despite having similar irradiation to Germany (Haukkala, 2015). In Norway, the low-energy price from the high percentage of hydropower in the grid reduces the perceived need of solar technologies for financial or environmental reasons (Xue et al., 2021). Sweden's subsidies and tax incentives have increased the number of PV systems but apartment buildings still represent a small proportion of installed systems (Lindahl et al., 2019). Apartment buildings are hindered by limitations in electricity metering and tax laws, which make distribution of electricity to the individual apartments complicated. Any generated power can only be used for communal demands, putting a limit on the worthwhile size of the PV installations. Therefore, having centralised heat pump or direct electric systems for heating and DHW can increase the viability of PV. However, as renewable energy is not essential for meeting energy requirements, PV represents a large capital cost to the constructor which is not necessarily reflected in the sales price of the building (Brocklehurst, 2017).

If heat pumps are considered as an on-site renewable energy source, because they draw renewable thermal energy from the ambient environment, 65 % of Swedish, 84 % of Norwegian and 26 % of Finnish low-energy apartment buildings A_{temp} utilised renewable energy.

Renewable energy technologies

As building regulations continue to develop toward NZEB and beyond, the use of on-site renewable energy will become necessary. Two Swedish projects, **Vallastaden** and **Brofästet** already produce more energy than they consume. **Brofästet**, in the Royal Seaport sustainable urban development in Stockholm, has 43 apartments calculated to use 14.8 kWh/(m²·year) for heating, DHW and building electricity. The power demand is so low due to a combination of efficient centralised balanced ventilation, GSHP, and centralised wastewater heat recovery. This is more than compensated by the electricity produced by rooftop solar panels, of which 30 % is used directly in the building (Stockholms Stad, 2019).

The prime candidates for on-site renewable energy for low-energy apartment buildings are PV, solar thermal collectors and GSHP. Other options are less suitable at the building scale. Small building mounted wind turbines require detailed design and wind modelling to achieve worthwhile output and can have issues due to noise and vibration (Haase and Löfström, 2015). Also a few iconic buildings, that have architecturally integrated wind turbines, have had a long list of issues which has likely damped enthusiasm. Even when placed well, the economic viability is low (Deltenre and Runacres, 2019). Biomass boilers and Micro CHPs offer year-round energy generation but still produce emissions from combustion. The calculated CO_2 emission are lower than fossil fuel systems, due to accounting for the CO_2 absorbed by the biomass during growth and not that it releases less pollution during burning, potentially deteriorate air quality in the quest for lower carbon emissions (Williams, 2012). This air pollution makes such systems untenable in urban environments, where most apartment buildings are found. Hydrogen is an alternative fuel for CHPs with no negative emissions but currently is not cost-effective due to a high fuel price (Marszal et al., 2012). Furthermore, the majority of hydrogen is produced in gas processing and so is not yet a truly green option (Adam et al., 2015). Hydrogen fuel cells may also not be suitable for the higher latitudes as they produce a large quantity of electricity relative to heat, which would result in large quantities of electricity sold to the grid. Boilers and CHPs have the further disadvantages of short lifetimes and the additional space needed for the units and their fuel.

Solar thermal collector systems can be used alone to efficiently provide DHW (Biaou and Bernier, 2008; Fung and Gill, 2011) or heating and DHW in a combi-system. The **Klosterenga** project built in 2000 required 245 m² of SC and 13 000 L of tanks to supply heating and DHW for 35 apartments (CMHC, 2017). Large storage volumes are needed due to the imbalance between supply and demand, reducing the economic feasibility of these systems at high latitudes. An optimised system in Montreal had energy payback times between 5.8 and 6.6 year, but was never cost effective due to the low cost of electricity ($0.0754/kWh) (Cheng Hin and Zmeureanu, 2014).

Solar thermal collectors can be combined with a GSHP, known as a solar assisted ground source heat pump (SAGSHP). The solar energy reduces the amount of energy required from the ground allowing for shorter boreholes. Emmi et al. (2015) showed that the borehole length could be reduced by up to 50 %, achieving a higher seasonal COP, while maintain ground temperatures in Stockholm. The reduced length should mean that the additional cost of panels is offset by the reduced drilling, although this is dependent on local borehole drilling costs (Rad et al., 2013). The length reduction possible is dependent on the ratio of heating demand to cooling demand. The more unbalanced the demand, the smaller the possible reduction. Solar thermal collectors can be connected in series to the heat pump circuit or in parallel to a shared accumulator tank. Januševičius and Streckienė (2013) showed that a parallel connection has higher performance than series because the heat pump is required to produce less DHW (at a lower COP) with lower run times. However, the low temperatures generated by the solar thermal collectors in the winter are only useful in a series system. An optimal approach is a flexible system which provides direct heating of DHW in the summer and recharges the boreholes in the winter (Kjellsson et al., 2010); although the benefit of this recharging is dependent on the borehole field design. This is the control principle used by the hybrid solar system marketed in Norway (Free Energy, 2021). Although possible to combine solar thermal collectors with district heating, this is often less economical because at the point of maximal production (summer), district heating prices are low (Kempe, 2013).

Although not directly part of the HVAC system, electricity from PVs can be converted to heat via a heat pump or electric coil. Therefore, PV was examined as an alternative to solar thermal collectors, as they compete for roof space. When combined with a heat pump, PV has been shown to perform better than solar thermal collectors, in cost (Marszal and Heiselberg, 2011; Milan et al., 2012), life cycle impacts (Karunathilake et al., 2019) and the proportion of energy demand covered (Good et al., 2015). Due to the higher PEF assigned to electricity, PV systems help to reduce primary energy more than solar thermal collectors (Reda and Fatima, 2019). Photovoltaic Thermal collector (PVT) systems are a hybrid option which collect heat from the underside of PV panels, providing both thermal and electrical energy. Due to a small market share limiting development, PVT performance and cost is inferior to PV only systems (Good et al., 2015; Sommerfeldt and Madani, 2018).

The performance of PV in all these studies was reliant on connection with the electricity grid to balance the periods of under and over production, which is particularly acute at the high latitudes of the Nordics. In combination with the electrification of heating and transport, this will place added stress on the grid. This can be moderated through energy storage and load shifting (Lund et al., 2015). The economic feasibility of battery storage is highly dependent on energy price pattens and is lower than load shifting due to high initial costs (O'Shaughnessy et al., 2018). Electric vehicles could be utilised instead as they stand idle for long periods, so called vehicle-to-grid solutions (Lund et al., 2015). However, the pricing model and technology are still in development. The thermal energy system offers the best opportunities for these two techniques as thermal energy accounts for the majority of a building's energy demand (Economidou et al., 2011).

Thermal energy storage

Thermal energy storage (TES) is already widely used in buildings in the form of hot water tanks for buffering heating and DHW demand peaks. These reduce the required capacity of the heating source and let it operate more efficiently, as it operates for longer periods at the nominal power (Medved et al., 2019). TES is also essential for storing the unregulated supply from solar technologies. The main focus has been on solar thermal collector systems (Hadorn, 2006) but TES also offers the most cost-effective form of storage for excess electricity produced by PV, increasing self-consumption (Cao et al., 2013).

Recent research has explored ways of further utilising TES to allow greater flexibility through more intelligent control. By using the TES for load shifting, charging the TES at a different point in time than the demand, a better energy balance can be achieved. Where energy prices fluctuate, this can reduce costs. The total energy use is often higher due to the additional losses from the charged TES.

A flexible residential water tank system is already commercially available in the UK (Mixergy, 2021). Tank stratification allows for the tank to have a varied fullness without reducing temperatures at the tap. The thermal mass of the building can also be utilised as flexible TES. The amount of storage is influenced by the level of envelope insulation (Johra et al., 2019). Underfloor heating is also beneficial as it activates more of the buildings thermal mass. However water tanks outperform thermal mass as their utilisation is limited by thermal comfort requirements and higher heat losses (Romanchenko et al., 2018). It is possible to improve the storage capacity of both by using phase change materials. However, their use is still limited due to cost and durability concerns (Arteconi et al., 2012).

The control strategy for TES can vary depending on the control goal and forecasting information. Control strategies of hot water tank storage in Norway were found to provide worthwhile savings over a constant setpoint strategy (de Oliveira et al., 2016). Where the control strategy is reactive, simple strategies such as charging storage at night were shown to perform just as well as more complex strategies. The complex strategy becomes more relevant as hourly prices become more volatile. An ideal strategy with price forecasting resulted in the highest savings. This type of strategy is now more feasible with the development of model predictive control, a proactive control strategy. Unlike traditional rule-based controls, model predictive controls uses a mathematical model of the system to optimise for the available inputs (Jorissen et al., 2018). For TES this could be optimisation for price based on the predicted thermal energy demand and future energy prices.

2.3 Limitations of Statistical analysis

The statistical analysis used data from EPC databases. There are some limitations with this approach due to how EPCs are assessed and the different national regulations. First it is necessary to compare the systems to understand their limitations.

It was possible to express a number for the maximum HVAC energy use for each country, as the input values for lighting and equipment load are fixed for both Finland and Norway. This could be converted to a delivered energy based on the proportion of electricity and district heating used, shown in Table 2.5. Finland and Sweden 2020 have closely aligned delivered energies which can vary by nearly 50 kWh/(m²·year). This is further confirmed by Figure 2.12, with 50 % of Finland's EPCs being rated low-energy under Sweden's 2020 regulations compared to just over 30 % for Norway and Sweden 2017. The opposite can be seen for the Norwegian system, which favours buildings powered by electricity. Only 26 % are low-energy in Finland and 23 % in Sweden 2020. With a 50 % electricity and 50 % district heating mix, all three countries have a similar delivered energy requirement in the mid-fifties.

Table 2.5. Comparison of maximum primary and delivered energies for HVAC for Norway, Finland and Sweden (two regulations), for different ratios of electricity and district heating.

	Norway	**Finland**	**Sweden 2017**	**Sweden 2020**
Max primary energy for Low-energy HVAC / (kWh/(m²·year))	56.1	40.3	63.75	56.25
Max delivered energy when 100 % electric / (kWh/(m²·year))	56.1	33.6	39.8	31.25
Max delivered energy when 100 % district heat / (kWh/(m²·year))	56.1	80.6	63.75	80.4
Max delivered energy when 50 % electric and 50 % DH / (kWh/(m²·year))	56.1	57.1	51.8	55.8

The Swedish results, with their relative rating system, show how the regulations have helped progress low-energy buildings. Only 24 % and 26 % of A_{temp} under the 2012 system was regarded as low-energy under Finnish and Norwegian regulations. This improved to 55 % and 70 % for 2015, and 87 % and 93 % for 2017. The PEFs for the 2017 regulations (1.6 for electricity and 1 for all other sources) are close to the ratio of non-electric source to electric sources for the earlier regulations (1.35 to 1.67), shown in Table 2.1. The fact that buildings under these older regulations perform better under 2020 than 2017, reflects the results shown in the previous section, that many of these buildings are heated using district heating. For the

2020 results, where the PEF for district heating is below 1, 95 % are regarded as low-energy in Finland but only 61 % in Norway.

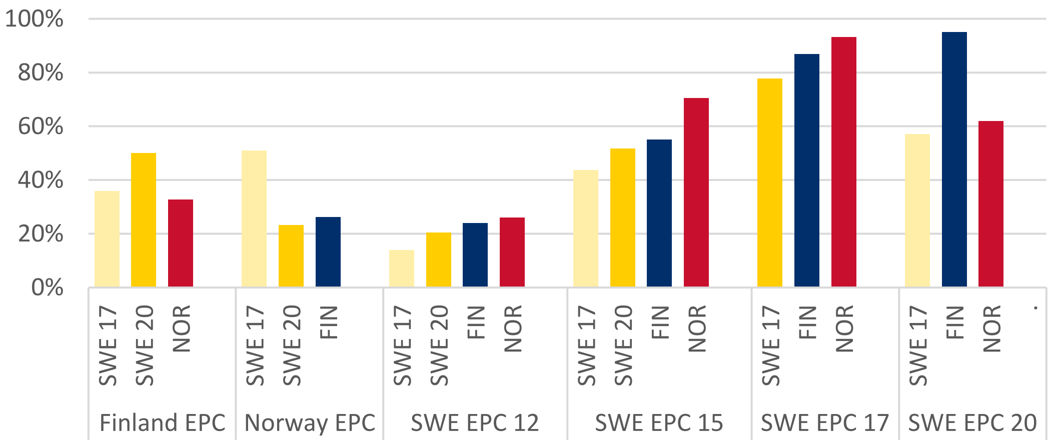

Figure 2.12. Percentage of A_{temp} that meets later EPC regulations and EPC regulations from other countries.

It should be stressed that these results do not indicate that one country's buildings are more energy-efficient than another. Rather that PEFs have a significant impact on the grade a building receives. A similar conclusion was reached by Kurnitski et al. (2018) through comparing the three countries and Estonia through simulations of a standardised building. In this analysis, neither Norway, Sweden or Finland had strict enough requirements to meet the EU nZEB recommendation which uses higher PEFs. The fact that buildings under the Sweden 2017 regulations performed the best under the alternatives, was mainly due to PEFs that sit in-between those of the other three regulations. When there is a mixture of energy sources, as shown in Table 2.5, the 2017 regulations have the lowest allowed delivered energy. At the extremes, the allowed delivered energy sits in the middle of the pack. There are small differences in the regulations for ventilation rates, setpoint temperatures, heat gain from occupants and DHW load, which make these comparisons less robust.

The results are influenced by the primary energy factors (PEF) of each country. For example, the low PEF for district heating in Finland was evident in the high proportion of A_{temp} heated by district heating. It is not possible to say if the favoured systems are actually more prevalent in the building stock because of the PEFs or just in the low-energy grade studied, where these systems are able to achieve lower primary energies. Further study using the complete database is required to reach solid conclusions. If the whole database was assessed under the different regulations, it is likely that many B and C grade buildings would be promoted. In the Swedish database for buildings under the 2012 and 2015 regulations, 13 additional entries would be low-energy under 2017 regulations and 108 additional entries under 2020 regulations. There were potentially 130 entries under the Finnish 2013 regulations, that could achieve an A grade in the latest requirements.

Although PEFs were originally intended as a method of accounting for the additional inputs and losses of converting an energy source for use in a building, their use has become more political (BPIE, 2017). This is often done with good intentions for the national interest. A low PEF for district heating encourages the use of an efficient thermal energy grid which already exists in all three countries. A high PEF for electricity discourages the use of direct electric heating systems which can put strain on electricity grids. Another example is that the reduction of all PEFs in Finland in 2018 was intended to encourage more builders to strive for better energy performance and achieve an A grade; a goal which seemed almost impossible to achieve in the 2013 regulations, meaning the EPC was viewed as not achieving its aim (ARA, 2018b). However, the use of PEFs can mean that less efficient solutions are chosen. Use of delivered energy as a metric would make the certificates more comparable. This is also a more meaningful number to residents as it reflects the likely energy cost (BPIE, 2017). The desired effects of PEFs can be achieved through other measures and regulations. Norway, which already uses delivered energy, uses regulations strongly encouraging connection to DH networks and the use of hydronic heating. This could be further strengthened by a proposal for an updated Norwegian EPC that includes a building's peak electricity load as part of its performance grade (Enova, 2019).

The validity of these results is dependent on the quality of the databases used. Due to the requirements to register new buildings it is likely that all apartments built in the last 5 years are included and so the statistics provide a good representation of the current state-of-the-art. The quality for data for each entry is dependent on the competence of the certifier. Quality assurance of data by the managing organisations is still relatively low and there is little enforcement for noncompliance (Arcipowska et al., 2014).

Furthermore, the performance of many entries is based on simulation rather than real world use. By not accounting for user variability, EPCs do not represent the actual energy demand (Pesola et al., 2016). It could be that some of these commonly used systems do not perform as well as simulated. Certain solutions which appear beneficial on paper, such as sewage heat exchangers or integrated wind turbines, have lower performance or higher costs than anticipated. The advantage of simulations with standardised loads and climate is the possibility to compare the performance of similar buildings, allowing for detailed analysis of energy-efficient solutions.

Further improvements to EPCs are already being proposed, offering more opportunities for the building performance sector. New indicators for smart readiness, indoor environmental quality, air pollution contribution and integration with district systems will allow for a more holistic evaluation of each building (Volt et al., 2020). Measured consumption data (as used in Swedish EPCs) allows for better quality control of certificates and feedback for building performance simulation. Improved recommendations for energy renovations combined with building finance will encourage faster and better energy renovations of the existing building stock. Therefore, the data collected through EPCs can be an important resource for improving the energy efficiency of the building stock.

2.4 Conclusions of Statistical analysis

Data for low-energy apartment buildings listed in the EPC databases of Norway (Grade A), Sweden (Grade A and B) and Finland (Grade A) was analysed in a statistical study. Due to the differences in assessment methods and databases, it was not possible to combine the data into a single dataset. However, the maximum allowed energy for the HVAC system was similar for all countries for a building with a 50:50 mixture of district heating and electricity, at around 56 kWh/($m^2 \cdot$year). Results were presented relative to floor area. The Swedish data was divided into four regulation periods, as each had different requirements or primary energy factors. The three most commonly used heating sources were district heating, ground source heat pumps (GSHP) and direct electric heating. The Norwegian low-energy apartment buildings mostly used bivalent systems, primarily heat pumps with direct electric heating for peak load. The Swedish low-energy apartment buildings mainly used GSHPs and district heating, often in combination with an exhaust air heat pump. Three quarters of Finnish low-energy apartment buildings were supplied by district heating. The rest were supplied using GSHPs.

DHW is shown to account for around 50 % of the energy demand in Norway and 40 % of the delivered energy in Sweden, allowing for new systems which prioritise DHW such as CO_2 heat pumps. The Finish dataset showed that heating was distributed using hydronic radiators or underfloor heating, with 34 % using a different emitter in the bathroom. Radiator systems were more often paired with district heating while underfloor heating was more often paired with GSHP. The Swedish datasets showed that most ventilation systems in low-energy buildings are balanced with heat recovery. This is supported by the limited data in the other countries' datasets. Sweden also had some exhaust air systems, but their use has decreased over time. The use of solar energy is limited with most systems found in Sweden. The use of PV has increased dramatically in the last 5 years while solar thermal collectors have declined. PV use is likely to increase in all three countries as regulations become stricter and PV prices fall.

From these results combined with the literature review, it is possible to say that low-energy solution-sets for the Nordic climate involve the use of district heating or heat pumps (primarily ground source) to provide hydronic heating with the peak load covered through electric radiators or electric water heating. The dominant ventilation strategy was balanced ventilation with heat recovery. An exhaust air system with an exhaust air heat pump combined with a ventilation window could be another option. The optimum solution-set will vary for each project dependent upon its precise location.

Based on these findings (or lack of), the following components were selected for study:

- Heat emitters:
    - Radiators
    - Passive convector
    - Fan convector
    - Underfloor heating

- Ventilation:
    - Balanced system with heat recovery
    - Exhaust system paired with exhaust air heat pump

- Primary heat sources
    - District heating
    - Ground source heat pump (option: combine with solar thermal collectors)
    - Exhaust air heat pump
    - Apartment-based compact unit.

3 Simulation Methodology

The systems identified through the statistical and literature review were tested through simulation on a proposed affordable housing project.

3.1 Description of the case project

BoKlok is a partnership between Skanska and IKEA to produce affordable housing. Costs are minimised by using prefabricated modules that can quickly be assembled on site (Boklok, 2021). The modules allow for five types of apartments, shown in Figure 3.1 and Figure 3.2. These are named after the local convention of the number of bedrooms plus the living room. The apartments are: small two room apartment (2S), large two room apartment (2L), small three room apartment (3S), large three room apartment (3L) and a four room apartment (4).

Figure 3.1. Apartment floor plans. From left to right: small two room apartment (2S), large two room apartment (2L), four rooms apartment (4). Image provided by BoKlok Norway.

Figure 3.2. Apartment floor plans. Left, large three room apartment (3L). Right, small three room apartment (3S). Two three room apartments must be placed together as they share a module. Image provided by BoKlok Norway.

These units can be stacked up to four floors high, with the vertical access, including the stairs and elevator, placed outside the building envelope. This report focuses on one of the proposed sites for BoKlok at Fossumjordet in Sørumsand (59°58'54" N, 11°14'25" E), Norway, roughly 20 km east from Oslo. Here it is proposed to build four blocks, four floors high providing 68 units and a total of 3 450 m² of heated living area. Figure 3.3 shows their arrangement. The blocks are built on top of an unheated basement which contains parking and the technical room for the buildings. The technical room is located in the northwest corner of the site, as this is closest to the district heating connection point. The details of the apartments are summarized in Table 3.1.

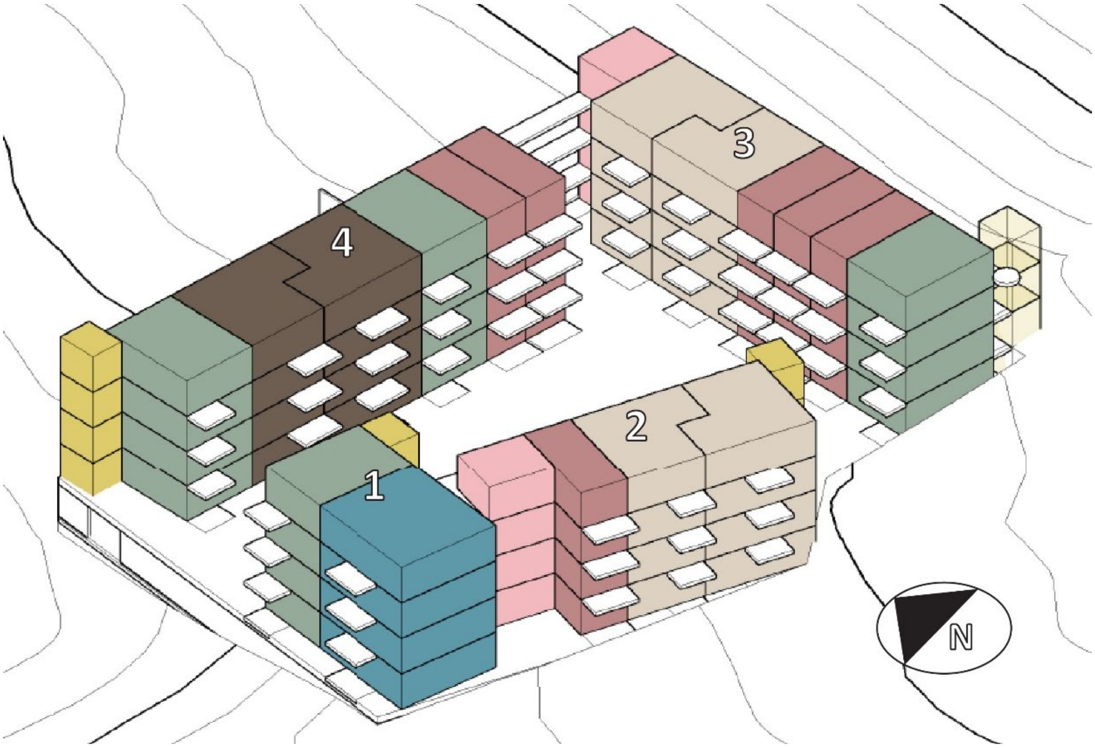

Figure 3.3. Perspective view of BoKlok Fossumjordet development from the southeast. Image provided by BoKlok Norway.

Table 3.1. Dimensions of the five BoKlok apartment types.

	2S	2L	3S	3L	4
Number of units	24	16	16	8	4
Internal floor area / m²	30.5	52.1	63.0	69.1	80.8
Floor to Ceiling height / m	2.5	2.5	2.5	2.5	2.5
Internal width / m	3.648	6.228	Long side: 8.948 Short side: 6.228	Long side: 9.668 Short side: 6.948	9.668
Internal Length / m	8.36	8.36	8.36	8.36	8.36
Door area / m²	4.2	4.2	4.2	4.2	4.2
Window area (incl. Frame) / m²	4.207 (4.858)	4.715 (5.637)	6.026 (7.172)	6.821 (8.217)	7.942 (9.753)
Glazing area (not including side walls)	26.6%	18.1%	18.9%	19.8%	20.2%

3.2 Base simulation inputs

The simulations were carried out in *SIMIEN7* (ProgramByggerne, 2021) in accordance with the Norwegian standard NS 3031:2020 (Standard Norge, 2020), using a 1-hour timestep. A climate file for Sørum, the municipality in which Sørumsand is located, was used.

3.2.1 Apartment modelling

Each apartment was modelled as a separate zone using the dimensions in Table 3.1. Where apartments shared a wall or floor/ceiling, the dimension was extended to the midpoint of the shared element, in accordance with the standard. The floor/ceiling was 614 mm thick. The partition wall was 292 mm thick. These shared elements were modelled as adiabatic. The effect of self-shading by the building was considered, otherwise a fixed horizon was used as the surrounding buildings were unknown. This was 9 degrees for the first three floors and 4 degrees for the top floor. Many of the windows are shaded by the balcony or external circulation above, which extends 2 m out. The top floor has a roof/pergola above the circulation/balcony and, therefore, has the same window shading. The properties of the thermal envelope are summarised in Table 3.2.

Table 3.2. Thermal envelope properties of BoKlok design against TEK17 requirements.

	BoKlok Design	**TEK17 Minimum requirements**
U-value / (W/(m²·K))		
- Wall	0.20	0.22
- Roof	0.13	0.18
- Floor	0.121	0.18
- Window	0.84	1.2
- Door	1	1.2
Window G-value	44%	n/a
Normalized Thermal Bridges / (W/(m²·K))	0.05	0.07
Air tightness / ACH at 50 Pa	0.8	1.5
Specific Fan Power / (kW/(m³/s))	1.5	1.5
Ventilation heat recovery	82%	80%

The thermal mass of the inside layer of each construction element was considered in the simulation. For the walls and ceiling, this was a 13 mm gypsum board, with a thermal capacity of 2.4 Wh/(m²·K). For the floor, this was 14 mm of parkette on top of 22 mm of chipboard, with a thermal capacity of 11.2 Wh/(m²·K).

3.2.2 Loads

Loads for domestic hot water, lighting, equipment, and people were according to NS 3031:2020 (Standard Norge, 2020) and are detailed in Table 3.3.

Table 3.3. Loads for apartment buildings according to NS 3031:2020

	Domestic hot water / (Wh/m^2)	**Lighting load** / (Wh/m^2)	**Equipment load** / (Wh/m^2)	**People load** / (Wh/m^2)
Time step				
00:00 – 01:00	0.3	0.3	1.0	1.5
01:00 – 02:00	0.3	0.3	1.0	1.5
02:00 – 03:00	0.3	0.3	1.0	1.5
03:00 – 04:00	0.1	0.3	1.0	1.5
04:00 – 05:00	0.1	0.3	1.0	1.5
05:00 – 06:00	1.8	0.3	1.0	1.5
06:00 – 07:00	3.8	1.7	1.0	1.5
07:00 – 08:00	7.2	1.7	1.9	1.5
08:00 – 09:00	5.5	1.7	1.9	1.5
09:00 – 10:00	3.9	1.7	1.0	1.5
10:00 – 11:00	2.2	1.7	1.0	1.5
11:00 – 12:00	2.2	1.7	1.0	1.5
12:00 – 13:00	2.5	1.7	1.0	1.5
13:00 – 14:00	2.5	1.7	1.0	1.5
14:00 – 15:00	2.2	1.7	1.9	1.5
15:00 – 16:00	1.9	1.7	3.4	1.5
16:00 – 17:00	3.6	1.7	4.3	1.5
17:00 – 18:00	6.9	1.7	4.3	1.5
18:00 – 19:00	7.2	1.7	4.3	1.5
19:00 – 20:00	4.8	1.7	3.9	1.5
20:00 – 21:00	3.1	1.7	3.9	1.5
21:00 – 22:00	2.7	1.7	3.4	1.5
22:00 – 23:00	2.2	1.7	2.4	1.5
23:00 – 00:00	1.2	0.3	1.0	1.5
Heat Energy sent to zone	0%	100%	60%	100%

3.3 HVAC parameters affecting gross demand

The parameters listed in this section determine the gross thermal energy required from the central supply system. A total of 58 solutions were tested based on five heat emitter options, three ventilation options and six heating schedules.

3.3.1 Ventilation

The following options for ventilation were simulated:

1. Apartment-based balanced system (as designed). A *Flexit K2.1* (Flexit, 2021a) unit was used in the 2S apartment type and a *Flexit Nordic S3* (Flexit, 2021b) in the other apartment types. Both systems had: 82 % heat recovery; a supply air setpoint of 19 °C to guarantee air mixing; and a specific fan power of 1.5 kW/(m³/s). No frost protection was applied, as practical experience in Norway has demonstrated that this is not an issue (Justo Alonso et al., 2015). Heating was provided by either:
   a. An electric coil
   b. A hot water coil connected to the heating distribution loop.

2. Central exhaust system with a specific fan power of 0.75 kW/(m³/s). This was only used for systems using an exhaust air heat pump.

The systems were simulated with constant airflows at the rate required by TEK 17, shown in Table 3.4. It is possible to optimise the energy use by using demand control strategies, such as an away button which reduces airflows when the apartment is unoccupied. However, these should not be assumed for sizing of the thermal system in order to avoid undersizing the system. Additional forced ventilation from the kitchen and bathroom was not considered. It is possible to achieve this without increasing the total airflow by reducing the draw from the bathroom when more air from the kitchen is required and vice versa, with a motorized damper. Such a system was implemented in the Løvåshagen project (Dokka and Helland, 2008).

Table 3.4. Required ventilation airflow for apartments according to TEK17.

Apartment type	Required constant airflow	
	m^3/h	$m^3/(h \cdot m^2)$
2S	90	2.95
2L	108	2.08
3S	116	1.84
3L	116	1.68
4	144	1.78

3.3.2 Heat emitters

Five solutions of heat emitter were simulated, detailed in Table 3.5. In each system, the panel emitters were sized to cover the peak heating demand in each apartment, using the nearest commercially available size. Losses from system regulation were not considered. Although the project is designed with electric underfloor heating in the bathroom, this was not considered in the simulation.

Table 3.5. Details of simulated heat emitters.

	Underfloor Heating	Medium temperature Radiator	Low temperature Radiator	Passive convector	Fan Convector
Supply temperature / °C	35	55	45	45	45
Return temperature / °C	30	35	35	35	35
Convective proportion (Oughton and Hodkinson, 2008)	40%	70%	70%	85%	100%

When cooling was possible, this was emitted using the same underfloor cooling or fan convectors, with the inputs shown in Table 3.6. The convective proportion for underfloor cooling is lower than heating because no natural convection is induced by the air over the floor (Pedersen et al., 1997).

Table 3.6. Details of simulated cooling emitters.

	Underfloor Heating	Fan Convector
Supply temperature / °C	19	16
Return temperature / °C	21	20
Convective proportion	10%	100%

3.3.3 Distribution

The central distribution systems for both heating and DHW were modelled as circulation systems. Domestic hot water was distributed at the required 65 °C (DiBK, 2017). Hot water for heating was distributed at the supply temperature specified for the heat emitter in Table 3.5. Where used, cooling was distributed using the same pipes for heating,

The primary circuit between the technical room and vertical shafts was placed in the underground parking garage, which was modelled with an average temperature of 10 °C. The length of piping for each circuit was calculated as 273 m. The hot water was distributed further to the manifold of each apartment via secondary circuits in the vertical shaft. The pipe length for these circuits totalled 220 m. This meant that each apartment had a specific pipe length of a 0.14 m/m² outside of the thermal envelope. A linear U-value of 0.2 W/K was used, corresponding to well insulated pipes (Standard Norge, 2020). Pumping power was based on specific pumping powers for well-designed systems: 0.2 kW/(L/s) for heat/cooling emitters and 0.15 kW/(L/s) for the hydronic ventilation coil.

The additional distribution from the shaft to the end uses within each apartment was not calculated directly. These small heat losses would be transferred to the zone and so reduce the heating demand by the same amount.

3.3.4 Setpoints and Schedules

Three annual schedules were tested with two options for daily schedules. Setpoints were based on NS 2020:3031 (Standard Norge, 2020). The operation dates were set based on the final cold period in May and the first cold period in September. A cold period was a daily mean temperature below 10 °C. The setpoint times were set according to NS 2014:3031 (Standard Norge, 2014). The options were:

1. Heating system operating all year
    a. Constant heating setpoint of 22 °C
    b. Heating setpoint of 22 °C (7:00 to 23:00) and setback of 20 °C (23:00 to 7:00)

2. Heating system operating from 9th September to 18th May
    a. Constant heating setpoint of 22 °C
    b. Heating setpoint of 22 °C (7:00 to 23:00) and setback of 20 °C (23:00 to 7:00)

3. Heating system operating from 9th September to 18th May and cooling system operating from 19th May to 8th September. The cooling setpoint was 24 °C
    a. Constant heating setpoint of 22 °C
    b. Heating setpoint of 22 °C (7:00 to 23:00) and setback of 20 °C (23:00 to 7:00)

Due to the lack of heat recovery, the exhaust system was only modelled using schedules with heating systems operating all year.

3.3.5 Evaluation of solutions

The systems were compared for energy use and compliance with thermal comfort requirements. Thermal comfort was achieved where the operative zone temperature was between 19 °C and 26 °C (DiBK, 2017). Up to 50 hours over 26 °C was deemed acceptable. Where a cooling system was not applied, operable windows were opened when the zone air temperature exceed 25 °C to prevent overheating. For zones which exceeded the limit, additional passive measures were applied to show the potential for maintaining thermal comfort.

3.4 Energy supply solutions

Relevant solutions from the previous section were then used as the demands for the proposed thermal energy supply systems, summarised in Table 3.7, accounting for energy use and practicality. The inputs and assumptions, including the modelling principle, of each heating source is detailed in this section. Where more than one hot water tank was used, the production of DHW was prioritised over heating. The systems were optimised and then compared for delivered energy demand and peak demand. As both electricity and thermal energy are imported, the energy cost was also considered.

Table 3.7. Studied energy supply solutions

System Name	Primary Heating	Secondary Heating	Thermal Storage	Ventilation	Heat emitter	Schedule
Base	Electric immersion heater	None	1 tank per unit	Balanced	FC	Heating all year
Compact	Compact HVAC unit integrating a reversible heat pump	Electric	1 tank per unit	Balanced	FC	Heating all year and cooling
DH	District Heating	None	None or 1 tank	Balanced	UFH, LTR, FC	Heating season only.
LTDH	District Heating with local heat pump	None	2 tanks	Balanced	UFH, FC	Heating season only
GSHP	Ground source heat pump	Electric for DHW peak	2 tanks	Balanced	UFH, FC	Heating and cooling season
SAGSHP	Solar thermal collector and GSHP	Electric for DHW peak	2 tanks	Balanced	UFH, FC	Heating and cooling season
EAHP	Exhaust air heat pump	Electric or DH	2 tanks	Exhaust	LTR	Heating all year

3.4.1 Base system

The standard solution uses an electric immersion heated water tank in each apartment providing DHW and heating via a fan coil. The *flexit* apartment-based balanced AHU provides ventilation. The water tank has a volume of 100 L with a setpoint of 75 °C, 10 °C higher than the central solutions. It has a 3 kW electric coil integrated. The tank loss was estimated at 2 W/K. An all-year heating schedule using a variable setpoint was assumed. The inputs for the ventilation system, fan coil and schedule are found in the previous section.

3.4.2 Compact unit

Compact units are all-in-one units composed of a balanced ventilation unit, reversible heat pump and hot water tank. The Nilan Compact P was most suitable for the airflow required by the different apartment sizes (Nilan AS, 2020). This has a 180 L tank and the heat pump characteristics shown in Table 3.8, based on testing by the Passivhaus Institut (2021). A 3 kW electric coil was added as a secondary heating source. The tank setpoint was set at 60 °C to improve the heat pump performance, while still protecting against legionella. This was 5 °C lower than the setpoint used in the central solutions. The tank heat loss is 1.63 W/K. Heating was provided by a fan coil unit with the same inputs as Section 3.3.2. A higher ventilation heat recovery of 92 % was used following the manufacturers specification.

Table 3.8. Maximum effect and COP of Nilan Compact P heat pump to provide 50°C for different outdoor temperatures and airflows, measured for Passive House certification. Values in brackets are intepolated or extrapolated from the other values

Outdoor air temperature	**-7°C**	**-4°C**	**2°C**	**7°C**	**20°C**
Values at 90 m³/h					
- Effect / kW	0.51	(0.57)	0.72	0.89	1.02
- COP	2.11	(2.13)	2.60	3.08	3.38
Values at 172 m³/h					
- Effect / kW	(0.55)	0.60	0.83	0.99	1.14
- COP	(2.10)	2.13	2.87	3.31	3.68

The heat pump is reversible so can cool the incoming air by up to 10 °C. The efficiency was calculated as 2.4. As hot water is still provided to the tank, it is possible to have heating and cooling all year. The heating was modelled using a variable schedule.

3.4.3 District heating

There is a district heating system in Sørumsand. The district heating is generated using a woodchip biomass boiler (90 %) and electricity (10 %) when electricity prices are comparably low (Akershus energi, 2020). The system was modelled to provide heat either directly or via a single accumulator tank for DHW and heating. The efficiency of the heat exchanger was 98 %.

A hypothetical low temperature district heating network was also tested. The grid temperature over the year was based on the Anergy system in Switzerland (ETH Zurich, 2018). The temperature varied between 8 °C at the start of May and 22 °C at the end of September. The heat pumps used to increase the temperature for heating and DHW are detailed in the next section.

3.4.4 Water source heat pumps

The same heat pump model was used for the GSHP, SAGSHP and LTDH systems. The heat pump performance was defined using heat pumps available from Alpha innotec (2021), due to the availability of detailed performance data. A general model was created, detailed in Table 3.9, which can be scaled to the required sizes. In reality the choice is limited by the possible combination of units. The units were assumed to run intermittently at part loads. They were modelled as a single large unit or multiple equally sized units in cascade. These were simulated using brine temperatures from the borehole simulations.

Table 3.9. Performance of water-source heat pumps to be scaled to required effect.

Brine Temperature	**0°C**	**5°C**	**10°C**	**15°C**	**20°C**
For supply of 35°C					
- Max effect / kW	1.15	1.35	1.53	1.65	1.77
- COP	4.7	5.2	6.0	6.4	6.7
For supply of 45°C					
- Max effect / kW	1.11	1.30	1.45	1.57	1.70
- COP	3.7	4.3	4.9	5.1	5.4
For supply of 55°C					
- Max effect / kW	1.08	1.25	1.40	1.53	1.62
- COP	2.9	3.4	3.8	4.0	4.2
For supply of 65°C					
- Max effect / kW	1.06	1.22	1.36	1.46	1.54
- COP	2.4	2.7	3.0	3.1	3.3

3.4.5 Solar thermal collectors

Solar thermal collectors would only be cost-effective where the reduction in cost of the other heating systems outweighs the capital cost and additional pumping energy of the solar thermal collector system. Their addition would not affect the sizing of the other heating sources, as they would produce limited energy during the peak sizing period. Therefore, solar thermal collectors are only relevant when combined with GSHP as they can reduce the required size of the borehole system.

Solar thermal collectors were modelled with a GSHP. This concept effectively utilises solar heat, changing its use depending on the temperature which the collectors can produce. When high temperatures are possible, the solar thermal collectors directly heat the DHW. When not possible, the solar thermal collectors heat the brine from the borehole, either supporting the heat pump or recharging the borehole system. This dual operation was not possible to simulate directly. Therefore. the operation was approximated by running two simulations in *SIMIEN7*

(ProgramByggerne, 2021) at a low (30 °C)[1] and high (65 °C) tank temperature to find the outputs including losses. A large tank size (10 000 L)[2] was used to minimise the fluctuation in inlet temperature flow through the panels, as the energy output reduces as the tank temperature increases. The hourly values were then compared with the DHW demand. The production profile at the high temperature was used to cover this demand. Where it was not possible or there was too much high temperature production, the remaining proportion was sized using the low temperature result. This was tested for flat plate and evacuated tube solar thermal collectors (without reflectors) in size increments of 24 m² gross area. The panel specifications were based on normal values from NS 3031:2020 (Standard Norge, 2020), shown in Table 3.10. An efficient system, with low pumping power (0.2 kW/(L/s)) and pipe losses, was assumed. The aperture percentage and gross dimensions reflect commercially available panels and allow for comparison.

Table 3.10. Characteristics of tested solar thermal collectors

	Flat Plate	**Vacuum Tube**
Aperture percentage	90%	55%
Gross dimensions	2 m x 1 m	2 m x 1 m
Optical efficiency	78%	72%
Linear heat loss	3.5 W/(m²·K)	1.8 W/(m²·K)
Quadratic heat loss	0.018 W/(m²·K²)	0.007 W/(m²·K²)

The panels were assumed to be placed on building 4 as this was the closest to the technical room. The panels were orientated to follow to orientation of the block (182°) with a 38° tilt. This tilt was found to have the most potential production from April to September, shown in Appendix A. It was possible to place 108 panels divided in three rows for which there would be minimum self-shading in these months. The length of the pipe connecting the technical room to the roof was estimated at 22 m. The outside piping was then estimated to be 6 m. This was increased by 4 m for each 24 m² increment. Detailed connection plans were not considered. The properties of the heat transfer fluid were modelled using program defaults.

3.4.6 Boreholes

The required borehole field was calculated using *GLHEPro 5.0* (Oklahoma State Univeristy, 2016) for the gross demand using fan convectors or underfloor heating, using either a variable or constant setpoint. Monthly loads were used for simulations lasting 25 years. Different sizes of TES, solar thermal collectors and peaking power were investigated for their effect on the required borehole depth and brine temperatures entering the heat pump.

[1] This was the lowest posible simulation value. The borehole temperature would likely be under 10°C. This means that the found values are likley an underestimate of the systems potential.

[2] The maximum possible size which could be simulated.

In addition, two minimum brine temperatures and two borehole configurations were studied. The minimum brine temperatures were 2 °C, representing a low exergy system, and 0 °C, representing a standard solution. The details of the single-U and double-U configuration are shown in Table 3.11. The values for the borehole thermal resistance are based on the works of Spitler et al. (2016) (single-U) and Javed (2018) (double-U). The brine was an ethanol-water mix with 12 percent concentration giving the properties shown in Table 3.12.

Table 3.11. Properties of single-U and double-U borehole configuration.

	Single U	**Double U**
Borehole diameter / mm	115	139
Shank spacing / mm	17	30
Tube inside diameter / mm	35.4	35.4
Tube outside diameter / mm	40	40
Volumetric flow rate/borehole / (L/s)	0.5	1
Fluid factor	1.1	1.1
Borehole thermal resistance / (K/(W/m))	0.09	0.06

Table 3.12. Properties of ethanol-water brine.

Freezing point	-5.3°C
Density	986.22 kg/m³
Volumetric heat capacity	4 273.2 kJ/(m³·K)
Conductivity	0.502 W/(m²·K)
Viscosity	0.002 87 Pa·s

The boreholes were arranged in a rectangular formation and could occupy a space 40 m by 60 m, dictated by the site. The number of boreholes was varied to achieve a borehole depth between 250 m and 350 m. The ground temperature profile used the nearest available location, Oslo-Gardemoen, roughly 25 km North of the case study site. The soil thermal conductivity was set at 3 W/(m²·K) and a volumetric heat capacity of 2 200 kJ/(m³·K).

3.4.7 Exhaust air heat pump

The exhaust air heat pump was only used in combination with the exhaust ventilation system. It was sized based on using four Nibe GreenMaster AHUs with heat pumps (NIBE AirSite, 2019), one for each building. These were sized using the manufacturers sizing tool. The resulting specifications are shown in Table 3.13. The performance of the heat pumps at different supply temperatures was interpolated from the manufacturers heat pump sizing software. An electric coil or district heating was used to cover the remaining load.

Table 3.13. Exhaust air heat pump performance calculated using manufactures software.

Building	**Airflow** / (m^3/h)	**Heat output supplying 55 °C** / kW	**COP supplying 55 °C**
1	1 008	8.4	4.1
2	1 296	10.9	3.9
3	2 448	20.1	4
4	2 520	20.7	4
		Interpolated heat output / kW 35 °C / 45 °C / 55 °C / 65 °C	**Interpolated COP** 35 °C / 45 °C / 55 °C / 65 °C
Total	7 272	63.8 / 62.8 / 60.1 / 57.2	4.7 / 4.4 / 4.0 / 3.6

3.4.8 Sizing of central system components

The central system was designed to cover the gross peak thermal energy demand, defined as the highest demand found for either the annual energy simulation or winter sizing simulation. This demand could be covered by numerous combinations of primary heating source, secondary (peaking) heating source and thermal energy storage.

The relationship between these three options was examined using the hourly simulated demand values through an optimisation tool created in *Microsoft Excel*. The tool calculated the required tank size to cover the energy demand (q_{demand}) which could not be covered by the maximum output of the heating sources (q_{max}). The size was based on the peak value of energy deficit ($q_{deficit}$) found for each timestep through the following formula:

$$q_{deficit} = q_{demand} - q_{max}, \qquad [kWh], \tag{2}$$

where q_{demand} included the hourly gross thermal demand from the simulations and the calculated heat loss from the tank, Equation (5). Heating and DHW were examined separately. Where q_{demand} was greater than q_{max} the deficit increased indicating energy being drawn from the tank. Where q_{max} was greater than q_{demand}, the deficit decreased indicating the tank being recharged back to its setpoint temperature. It was not possible to charge past this setpoint. In other words, $q_{deficit}$ could not be negative. By using hourly timesteps, it was possible to account for the effect of long periods of high heating demand. The resulting peak deficit was used to define the tank volume using the following formula:

$$V_{tank} = \frac{q_{deficit.PEAK}}{s_w \cdot (T_{Setpoint} - T_{supply})}, \qquad [m^3], \tag{3}$$

where the volumetric heat capacity of water (s_w) was defined as:

$$s_w = \frac{C_{p,w} \cdot \rho_w}{3600}, \qquad [kWh/(m^3K)]. \tag{4}$$

The heat capacity ($C_{p,w}$) and density (ρ_w) of water were set based on the setpoint temperature assuming constant pressure equivalent to atmospheric pressure at sea level. For a tank temperature of 50 °C, $C_{p,w}$ was 4.181 kJ/(kg·K) and ρ_w was 988.02 kg/m³. A table with further values can be found in Appendix B. The supply temperature (T_{supply}) was dependant on the heat emitter used. The setpoint temperature ($T_{setpoint}$) was varied in steps to a maximum of 95°C. The heat loss was calculated as:

$$q_{S,Loss} = H_S \cdot (T_{Tank} - T_a), \qquad [W]. \tag{5}$$

The ambient temperature (T_a) was set at 18 °C. The same value was used in the simulations. For modelling of heat losses to the surroundings the tank is assumed to be fully mixed for simplicity. This simplification is justified by the findings of (Steen et al., 2015) who showed there is little difference in energy losses between fully mixed and ideal stratification. The tank temperature was determined by the deficit of the tank:

$$T_{Tank} = \quad T_{Setpoint} - \left(\frac{q_{deficit}}{q_{deficit.PEAK}} \cdot \left(T_{Setpoint} - T_{supply}\right) \right), \qquad [°C]. \tag{6}$$

Here, the relationship between power and temperature was simplified to a linear corelation, shown in Appendix B. The specific heat loss (H_s) was calculated as:

$$H_s = A_{tank} \cdot k + \sum H_{s.connection}, \qquad [W/K]. \tag{7}$$

The surface area of the tank (A_{tank}) was calculated from the volume (V_{tank}). A cylindrical tank was assumed with a height equal to its diameter, giving the most efficient surface area to volume. The thermal conductivity of the tank and 100 mm of insulation was 0.37 W/(m²·K). Additional losses from each connection are estimated at 0.2 W/K based on the findings of Steinweg et al. (2014). A system would have a minimum of four connections, with more for each additional energy source. Where an electric heating source was used, the coils were assumed as being inside the water tank. The resulting sizing curves were then used for defining simulation inputs for the tested peak.

The size of the heating sources was defined in 10 kW steps. In all solutions using a heat pump, a minimum tank size of 500 L was used as this is required to allow the heat pump periods of continuous operation. Tank sizes were then rationalised up to the nearest 1 000 L.

3.4.9 Evaluation of systems

The systems were optimised and then compared for delivered energy demand and peak demand. As both electricity and thermal energy are imported, the energy cost is also considered.

The electricity price varies for each hour based on the Nordpool energy market. The end-user price consists of the spot price, grid tariff, electricity tax and VAT. In addition, there can be fees for the electricity provider, and electricity certificates. These are dependent on contract and so were not included in this analysis. The build-up of price is shown in Table 3.14. The grid tariff varies seasonably, and the spot price fluctuates hourly. In addition, there is a monthly cost comprised of a fixed part and a variable part based on the peak demand in that month, shown in Table 3.15.

The district heat price was based on the pricing model from the local supplier (Akershus Energi, 2021), shown in Table 3.14. In addition, there is a fixed monthly cost of 340 NOK. As there is no common price model for low temperature district heating, the same district heating price was use. Different percentages of this price were tested to find the breakeven point for the LTDH system.

Table 3.14. Cost per kWh of electricity and heat.

<table>
<tr><th></th><th>Spot price</th><th>Grid tariff</th><th>Energy tax</th><th>VAT</th></tr>
<tr><td>Electricity</td><td>Hourly spot price</td><td rowspan="2">0.070 NOK
(November through March)
0.039 NOK
(April through October)</td><td rowspan="2">0.1669 NOK</td><td rowspan="2">+ 25%</td></tr>
<tr><td>Heat</td><td>Monthly spot price</td></tr>
</table>

Table 3.15. Monthly fee for electricity (Elvia, 2021).

<table>
<tr><th>Period</th><th>Fixed cost / NOK</th><th>Peak cost (Max kW in the month) / NOK/kW</th></tr>
<tr><td>December through February</td><td rowspan="3">340</td><td>120</td></tr>
<tr><td>March and November</td><td>67</td></tr>
<tr><td>April through October</td><td>22</td></tr>
</table>

Electricity price data from 2019 was used as this was the most recent non-anomalous year, assumed to best represent the marked conditions in the coming years. A detailed comparison of yearly price data is presented as part of the flexibility analysis in Section 4.3.1.

3.5 Energy flexibility

As a final step, selected systems were further analysed for their potential for energy flexibility through thermal energy storage (TES). Energy flexibility allows for better use of the grid by shifting demand to reduce peaks or to match renewable generation. High energy prices often represent periods of higher demand than supply, which require the use of peaking power plants or importing electricity, often with higher CO_2 emissions (Claußet al., 2018). This has a particularly large effect on the average CO_2 emissions of the Norwegian grid. To be of interest to the consumer, it must also reduce energy cost, despite the increased energy use to cover losses in the storage system. As the peak capacity of a system is designed for the peak demand condition, there is extra capacity for the majority of the year which can be leveraged for energy flexibility.

A hypothetical control strategy to minimise cost was tested, which used the demand profiles from simulation and hourly electricity price as inputs. Response to the district heating grid was not considered as there is no short-term price variation. The strategy worked on a 34 hour period as the next-day Nordpool price information was assumed available to affect heating from 14:00 on the current day. The Nord Pool auction closes at 12:00 with price data available within the following hour. An additional hour was assumed to mitigate potential communication problems. It was assumed that the simulated energy use was a perfect prediction of reality and that an actual model predictive control would perfectly generate this profile based on the weather forecast.

3.5.1 Potential for energy flexibility

A simple analysis was first undertaken, which did not consider any system limits. Therefore, the results represent the maximum energy savings possible using the control. For each timestep the electricity spot price was compared to the spot prices in the preceding timesteps until 14:00 the preceding day. The additional heat loss of storing a kWh of energy in a TES was estimated at 0.265 W per hour based on a 1 000 L TES. This is approximately the same no matter the temperature of the tank. The full calculation is explained in Appendix C. The price for each preceding hour was calculated as:

$$Price_{0-n} = Price_n \cdot 1.0026^n, \qquad [NOK/kWh], \qquad (8)$$

where n is the number of hours preceding the current timestep. The potential saving was then found by subtracting the lowest found price from the timestep price. The process is repeated for each timestep. The resulting savings were then multiplied by the thermal energy demand profile for the solution using balanced ventilation, a fan coil and a variable heating setpoint for just the heating season. The last 8 years of electricity spot pricing for the Oslo region were analysed to find the possible year on year variation.

3.5.2 Optimisation of demand profile

An optimisation strategy was designed to create a cost optimal demand profile according to the available tank volume, tank temperature and capacity of the heating element. The tank was assumed to be fully mixed. In the strategy the tank was preloaded with the required energy. When it was depleted, it had a temperature equivalent to the required supply temperature of the heat emitter. Therefore, the tank should never have an energy deficit as this would reduce the temperature below the required supply temperature.

The approach in the previous section was used to define the price for each timestep and its preceding timesteps. These were multiplied by the timestep's demand and then ranked ($Rank_{timestep}$) by cost saving. This was repeated for all timesteps in a 24 hour period. These were also ranked ($Rank_{hour}$) by their maximum cost saving.

The demand for the highest $Rank_{hour}$ was then placed at the timestep which had the highest $Rank_{timestep}$ for that $Rank_{hour}$. If this exceeded the capacity of the heating power, the demand to fill that capacity was placed at the highest $Rank_{timestep}$ and the remaining demand was placed at the next highest $Rank_{timestep}$. This process was repeated for each hour in descending order of $Rank_{hour}$, as shown in Figure 3.4. The available heating power at each timestep was equal to the maximum capacity of the heating element minus any assigned demand to that timestep. This included the demands assigned in the previous 24 hour period. Where it was not possible to distribute all of an hour's demand within any of its preceding timesteps, this demand was added to the next proceeding timestep which did not create a deficit in the energy balance. This in essence moves a higher ranked hour's demand to a later point to make space for the lower ranked hour's demand.

	Demand from previous 24 hour period →	$Rank_{hour.1}$ Demand = 5 kW →	$Rank_{hour.2}$ Demand = 5 kW →	$Rank_{hour.3}$ Demand = 4 kW →	Demand profile
....	....	....	....	....	....
22:00	3 kW	$Rank_{timestep.2}$ (2 kW)	$Rank_{timestep.2}$	$Rank_{timestep.2}$	5 kW
23:00	2 kW	$Rank_{timestep.1}$ (3 kW)	$Rank_{timestep.1}$	$Rank_{timestep.1}$	5 kW
00:00	2 kW	$Rank_{timestep.3}$	$Rank_{timestep.3}$ (3 kW)	$Rank_{timestep.3}$	5 kW
01:00	n/a	$Rank_{timestep.4}$	$Rank_{timestep.4}$ (2 kW)	$Rank_{timestep.4}$ (3 kW)	5 kW
02:00	n/a	$Rank_{timestep.5}$	$Rank_{timestep.5}$	n/a (1 kW)	1 kW
03:00	n/a	$Rank_{timestep.8}$	$Rank_{timestep.8}$	n/a	0 kW
04:00	n/a	$Rank_{timestep.9}$	$Rank_{timestep.9}$	n/a	0 kW
....	....	....	....	....	....

Figure 3.4. Graphical representation of demand profile optimisation for a simple example with a heating element with a 5 kW capacity. Green shows placement of an hour's demand at that timestep. Red shows where demand placement is not possible as the capacity of that timestep has already been reached.

The resulting demand profile was then used to model the temperature and additional heat losses of the tank for each timestep. The energy storage capacity of the tank was defined in kWh (q_{max}) and the maximum temperature (T_{max}) was chosen depending on the energy supply system. The supply temperature (T_{supply}) was defined by the heat emitter used. The tank temperature for each timestep could then be calculated based on the energy stored in the tank at that timestep (q_{tank}) using the following formula:

$$T_{Tank} = T_{supply} + \left(\frac{q_{tank}}{q_{max}} \cdot (T_{max} - T_{supply})\right), \quad [°C]. \tag{9}$$

The required tank size and surface area were then calculated to meet q_{max} at T_{max}. The specific heat loss could then be calculated using Equation (7) and the heat loss for each hour using Equation (5). Where the demand for an hour was already at the maximum system size, the heat loss energy was placed at the next available hour. The final demand profile was then multiplied by the electricity price and compared to the unoptimised profile.

The optimisation model was limited to heat sources which had a constant system efficieny independant of the load on that system or outside conditions. Therefore, it was studied using a theoretical central system with electric elements and the base case. A maximum tank temperature of 90 °C was assumed.

4 Simulation Results

In line with the workflow, the results of the solutions affecting gross energy demand are presented first. The demand from these systems was then combined with the tested energy supply systems to find delivered energy use and peak demand. Finally, the potential for flexibility is analysed.

4.1 Solutions affecting gross demand

The simulated solutions and their variable parameters are presented in Table 4.1. These were simulated using an ideal energy source which always met the demand and a centralised distribution system. The cooling schedules were only used with the compatible emitters: underfloor heating and fan coils. Due to the lack of heat recovery, exhaust ventilation solutions were only simulated using schedules with heating operating all year. This resulted in 58 different solutions, analysed for energy efficiency and thermal comfort. Where individual apartments are examined, they are represented by 3 numbers for their position in the project and the apartment type, explained in Figure 4.1.

Table 4.1. Simulated solutions affecting gross demand

Ventilation	Heating emitter	Schedule
Balanced ventilation system with electric heating coil or hydronic heating coil	Underfloor heating (UF) Fan coil (FC)	Heating all year - constant setpoint (Con) - variable setpoint (Var) Only heating season - constant setpoint (Winter Con) - variable setpoint (Winter Var) Heating and cooling - constant setpoint (Cool Con) - variable setpoint (Cool Var)
	Medium temperature radiator (MTR) Low temperature radiator (LTR) Passive convector (PC)	Con Var Winter Con Winter Var
Exhaust ventilation	UF MTR LTR PC FC	Con Var

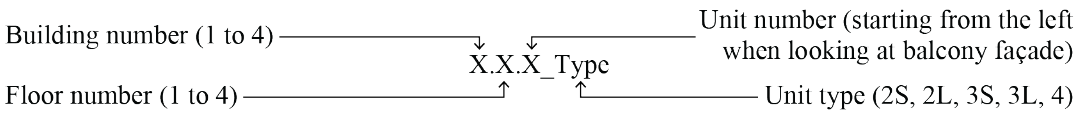

Figure 4.1. Guide to apartment labelling in results.

4.1.1 Energy efficiency

The gross thermal energy demand and peak for all 68 apartments are shown for solutions using balanced ventilation (with an electric heating coil) in Figure 4.2 and Figure 4.3; and using exhaust ventilation in Figure 4.4 and Figure 4.5. The gross thermal energy demand includes the demand and distribution losses from the heating emitters and DHW. The DHW use is the same for all cases using 86 180 kWh with distribution losses of 46 533 kWh. This represented between 39 % to 44 % of the thermal energy demand in balanced ventilation solutions and around 23 % in exhaust ventilation systems. The peak load for DHW was 30 kW. The electricity use of lighting and equipment was the same for all cases as it is standardised.

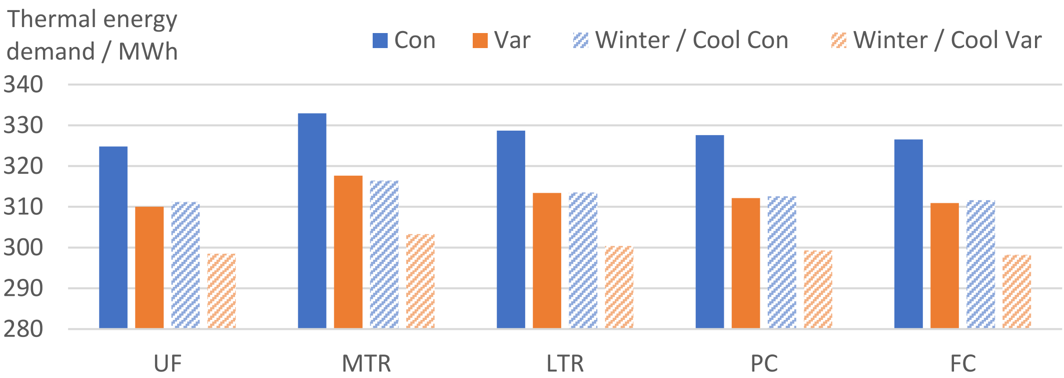

Figure 4.2. Thermal energy demand of solutions with balanced ventilation (with electric heating coil).

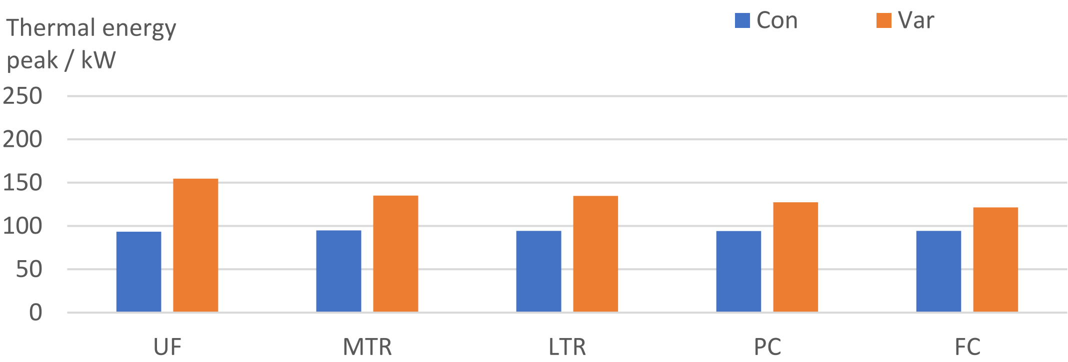

Figure 4.3. Thermal energy peak load of solutions with balanced ventilation (with electric heating coil).

Effect of ventilation system

The balanced systems required 26 467 kWh for fans. The exhaust required half, 13 233 kWh. However, the lack of heat recovery meant that exhaust systems required between 272 253 kWh to 281 153 kWh more thermal energy. This resulted in the pumping energy more than doubling, seen in Table 4.3. The exhaust systems had peak demands between 81 kW and 85 kW higher than the balanced systems. For the balanced system, there was no difference in peak demand between electric and hydronic ventilation coils, shown in Table 4.2. The choice is still important as it decides if this peak is covered directly by electricity or the central system. Likewise, energy demand was the same where the heating system was operating the entire year. In the other schedules, the hydronic coil was stopped with the heating system. Due to the small heating demand in the summer, this only resulted in 286 kWh to 289 kWh difference. Again, the importance is in the allocation of the energy demand.

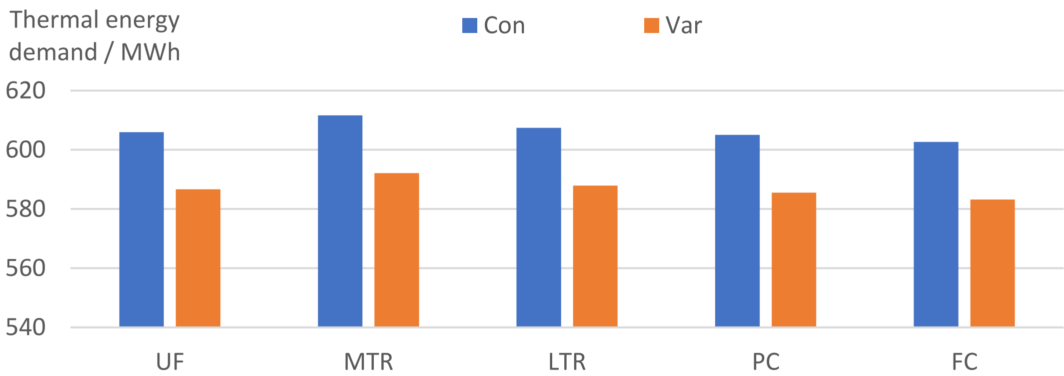

Figure 4.4. Thermal energy demand of solutions with exhaust ventilation

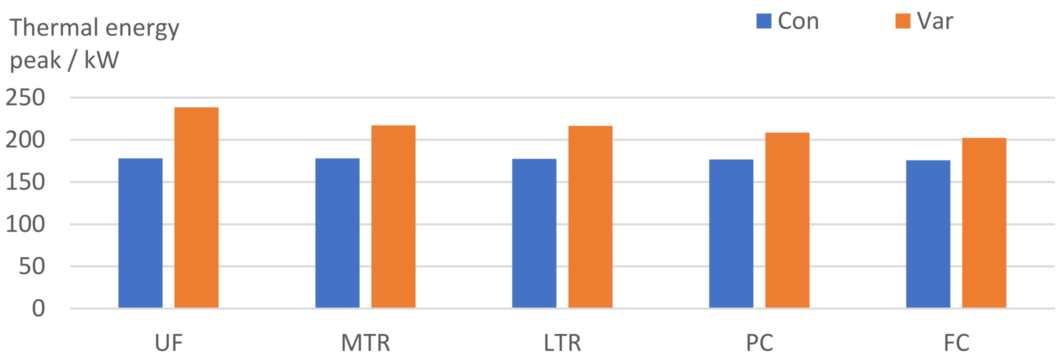

Figure 4.5. Thermal energy peak load of solutions with exhaust ventilation

Table 4.2. Energy demand and peak for a hydronic coil or electric coil in the balanced ventilation system. Where a range is stated, this is due to differences between heat emitters

	Con	**Var**	**Winter Con**	**Winter Var**
Electric coil demand / kWh	9 576 to 9 594	14 832 to 15 267	9 862 to 9 882	15 053 to 15 479
Hydronic coil demand / kWh	9 576 to 9 594	14 832 to 15 267	9 576 to 9 594	14 765 to 15 190
Electric coil peak / kW	9.12	12.70 to 12.78	9.12	12.70 to 12.78
Hydronic coil peak / kW	9.12	12.70 to 12.78	9.12	12.70 to 12.78

Effect of heat emitters

Underfloor heating (UF) had the lowest thermal demand and medium temperature radiators (MTR) the highest. This is primarily due to the distribution losses associated with the supply temperatures to the emitters. The other three emitters have similar values as they use the same supply temperature. Fan coils (FC) had the lowest demand, followed by passive convectors (PC) and low temperature radiators (LTR). The higher convective heating proportion leads to reduced energy demand as it has more affect on the air temperature in the zone which controls the thermostat in the simulations. An opposite pattern was seen in the ventilation coil energy, although the differences here were much smaller, shown by the range of values in Table 4.2.

The supply temperatures also affected the required pumping powers, shown in Table 4.3. LTR, PC and FC had similar energy use, HTR the lowest and UF the highest. When the pumping powers are considered, FC had a lower energy use than UF. There were small differences in peak demands under the constant schedule, with these following a similar patten to the energy demand. Under a variable schedule, the peak increases as the convective proportion of the heat emitter decreases. This results in underfloor heating having the largest difference in peak demand between constant and variable schedules.

Table 4.3. Energy use for pumps for the different solutions.

	Balanced / kWh				**Exhaust** / kWh	
	Con	Var	Winter Con	Winter Var	Con	Var
UF	1795	1633	1712	1574	4512	4317
MTR	406	370	389	357	1070	1023
LTR	812	740	778	715	2140	2047
PC	807	734	773	710	2128	2035
FC	802	728	769	705	2117	2024

Effect of schedules

The effect of the schedules on energy demand was logical. Those with a setback temperature and/or a reduced heating period have lower demands. The reduction was relative to the energy demand with higher initial demands having larger reductions. This can be seen clearest when comparing balanced ventilation systems with exhaust ventilation systems. The difference between a constant and variable setpoint was around 15 200 kWh for balanced systems and 19 500 kWh for exhaust systems.

Schedules with variable setpoints have higher peak loads than constant setpoints due to the warmup from setback to the setpoint. This can be seen clearly in the daily load profile shown in Figure 4.6. The difference in peak demand between variable and constant setpoints was almost the same for each heat emitter combined with either balanced or exhaust ventilation. Therefore, this difference was relatively larger for the balanced air system where the peak demand is smaller.

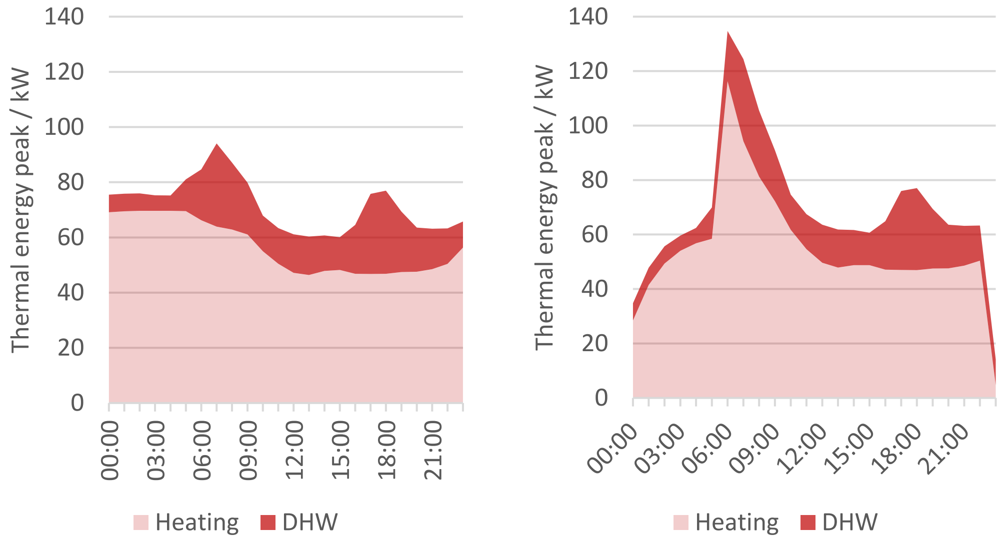

Figure 4.6. Demand profile over the peak load day (13.01) for a constant setpoint (left) and variable setpoint (right).

Schedules which utilised cooling had additional energy demands. The cooling demands and peaks were very similar for the 4 simulated solutions. The energy demand was 8 261 kWh and the peak in July was 72.44 kW. As the proposed solution involving cooling utilised free cooling from the ground, this energy use costs only the additional 1 000 kWh of pumping energy required.

4.1.2 Thermal comfort

Thermal comfort analysis is presented using the variable summer schedule as this had the worst thermal comfort performance. Figure 4.7 shows the thermal comfort for each apartment in building 4 using a balanced ventilation system and low temperature radiators. This building had the apartment with the highest number of hours over 26 °C and the apartment with the highest number of hours under 19 °C. The results for all apartments can be found in Appendix D.

Overheating problems were most present in the largest apartments, particularly those with fewer external surfaces (e.g. 4.2.3_3L and 4.3.3_3L). Underheating problems were most present in the smallest apartments, particularly those with many external surfaces and with low solar exposure (e.g. 4.1.6_2S and 4.4.6_2S).

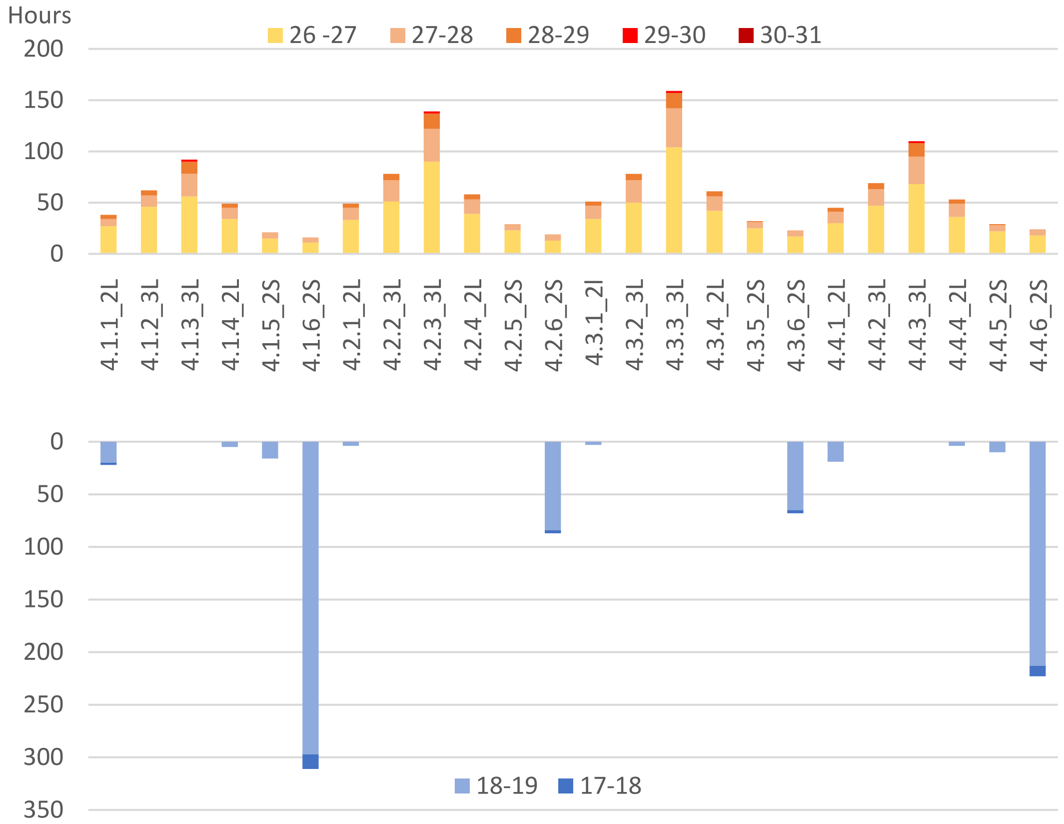

Figure 4.7. Hours above and below the thermal comfort limits for the solution with balance ventilation, low temperature radiator and variable schedule for just the heating season

Effect of schedules

Schedules with heating all year round had the same overheating hours but no underheating hours. Schedules with cooling had the same underheating hours but no overheating hours. There was a negligible variation between schedules using constant setpoints and variable setpoints.

Effect of ventilation

Exhaust ventilation required schedules with all year round schedules to achieve thermal comfort at the lower end. The choice of ventilation heating coil in balanced ventilation systems had a small effect on thermal comfort of schedules with no heating during the summer. As the hydronic coil was inactive in the summer period, the number of hours of underheating was greater than those shown in Figure 4.7 by up to 164 hours in the worst performing apartment (4.1.6_2S). The increase was proportional to the number of hours shown in the figure. Of the 68 apartments, 43 had increases less than 10 hours with 15 seeing no increase.

Effect of heat emitter

The choice of heat emitter had little effect on the total number of hours outside the thermal comfort range. However, it did affect the distribution of the operative temperatures, as shown in Figure 4.8. Heat emitters with higher radiant heat transfer, such as UF, had higher operative temperatures. This resulted in more overheating but less under heating. The opposite was true for highly convective heat emitters. The variation in the number of hours of overheating and underheating was still small. In the worst overheating case, the variation was 16 hours. In the worst underheating case, the variation was only 2 hours. When viewed with a stricter thermal comfort range, UF clearly provides the best thermal comfort with the highest number of hours with operative temperatures between 21 °C to 24 °C.

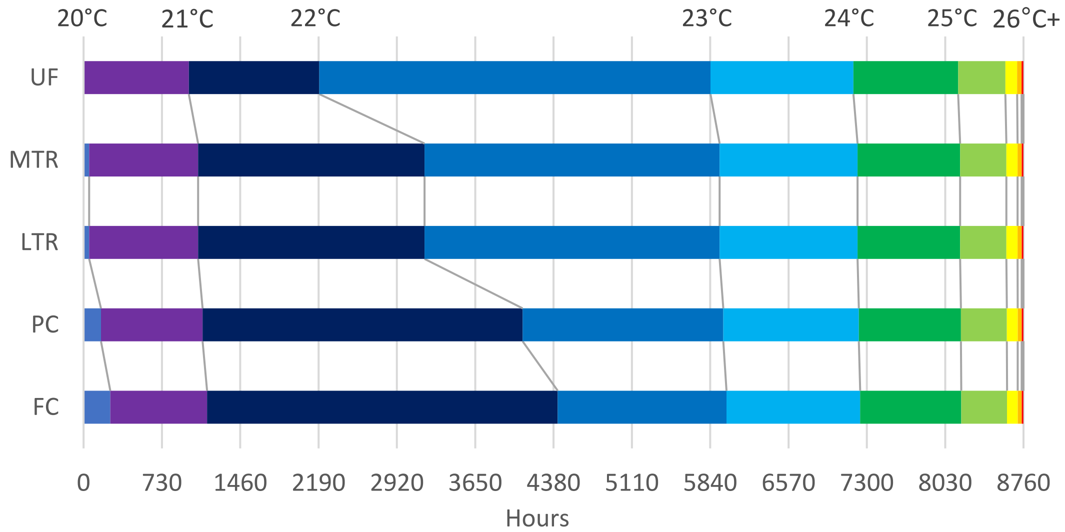

Figure 4.8. Distribution of operative temperatures in apartment 4.3.3_3L for each heat emitter using balanced ventilation and a variable schedule for just the heating season.

Possible solutions to thermal comfort

The number of apartments with more than 50 hours outside the thermal comfort range was 58. This was mainly due to the light application of passive cooling measures which were difficult to realistically apply in the simulation software without creating heating demand spikes. Furthermore, the same strategy was applied equally to all the apartments for the interest of energy use comparison. By tailoring the control of the systems and using passive measures it was possible to achieve good thermal comfort in almost all apartments.

For the apartments which suffer from underheating, a higher supply air temperature and direct electric radiators could be used. The low installation cost makes this more cost effective than leaving the central heating system on all year.

For the apartments which suffer from overheating, more passive measures can be used. The effect of some common measures on the worst performing apartment in each block are shown in Figure 4.9. The combination of internal window shading and more windows airing is enough to bring all but the 16 worst performing apartments under the limit. Even with additional external shading applied, 6 apartments still exceeded the limit. These apartments would benefit the most from an HVAC system with cooling.

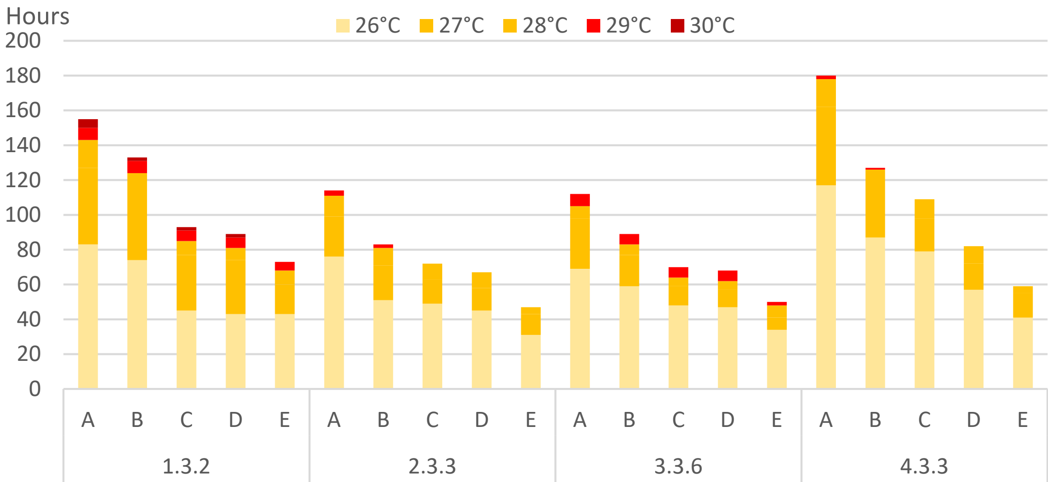

Figure 4.9. Effect of passive solutions on the apartment with the worst thermal comfort from each building. A = As simulated. B = A + Internal window shading. C = B + additional window airing. D = C + Variable supply air temperature in ventilation system. E = D + External shading.

4.1.3 Summary

Balanced ventilation solutions outperformed exhaust ventilation solutions for both energy demand, energy peak and thermal comfort. The additional 13 233 kWh of fan energy was dwarfed by the difference in heating demand. The higher heating demand of exhaust systems required larger heat emitters and more energy, likely making an exhaust solution more costly. There is also the potential for drafts due to the uncontrolled supply of air, although this was not examined in detail. This difference was compounded by the possibility for balanced systems to use schedules where the heating was stopped in the summer, further reducing energy use by 5 %. Although the simulations of these schedules resulted in periods under 19 °C in some apartments, this could be remedied in practice by using small electric heaters and control of the ventilation system to cover this small demand. In the project, the apartments at the gable ends will have additional direct electric radiators to cover the higher demand. For other apartments with only a few hours of underheating, increasing the supply temperature from the ventilation systems could provide thermal comfort during this period. This requires an electric heating coil. The use of the electric coil during the heating season could be minimised by reducing the ventilation setpoint; thus, increasing the proportion of heat provided by a more efficient central system. In solutions utilising heat pumps, a hydronic heating coil would be more efficient but lacks the flexibility for when heating is not operation. By stopping heating during the summer, the distribution system is free to be used for cooling improving the thermal comfort. The additional pumping cost required for free cooling was negligible.

The choice of heat emitter had a limited effect on energy demand, with the required supply temperature being the critical factor. This was balanced by the pumping power, with lower supply temperatures requiring more pumping power. This resulted in underfloor heating and fan coils having a similar energy demand. The high radiative proportion of underfloor heating gave better thermal comfort, however UF also had the highest difference in peak demand when comparing constant and variable setpoints. The opposite was true of FC.

4.2 Energy supply systems

Based on the findings of the previous section, only certain solutions were investigated further. As the performance of radiators, passive convectors and fan coils was similar, only results for fan coils and underfloor heating are presented in the rest of the thesis. The thermal energy demands of these solutions were analysed in order to find the best configuration of heating system and thermal energy storage. Several parameters of the GSHP and SAGSHP systems were also investigated in detail, including: studying the output of a solar thermal collector system; the borehole field design; and the number of heat pumps used in the system. Finally, all the possible systems are compared for the delivered energy and energy cost.

4.2.1 Sizing of system components

The central systems were defined in three parts: a primary energy source covering most of the demand; an optional secondary energy source to cover the peak load; and thermal energy storage (TES). The sizing of the total system had to cover the peak load. The size of the primary energy source could be reduced by increasing the size of the secondary source and/or the TES. This is particularly relevant for the systems using GSHPs as reduction of the peak demand often reduces the required borehole length.

The potential to reduce the peak was dependent on the load profile. The constant setpoint profile had less variation than the variable setpoint and so a lower peak demand, shown in Figure 4.10. If the demands for heating and DHW are viewed separately, shown in Figure 4.11, it is clear that heating accounts for most of the variation in load. The variation in DHW demand is low as the demand follows the same profile every day. The lack of a peak in DHW demand means that there was little need for secondary sources or TES. As this profile is based on averages, it is possible that the DHW demand could fluctuate more. However, such profiles have been shown to be closer to reality the more apartments are connected to the system (Ahmed et al., 2016). The DHW demand accounted for 18 % to 32 % of the peak thermal load in balanced ventilation solutions and 13 % to 17 % in exhaust ventilation systems. The constant setpoint profile was similar for the underfloor heating and fan coil. The variable setpoint profile differed at the peak, with underfloor heating peaking over 30 kW higher.

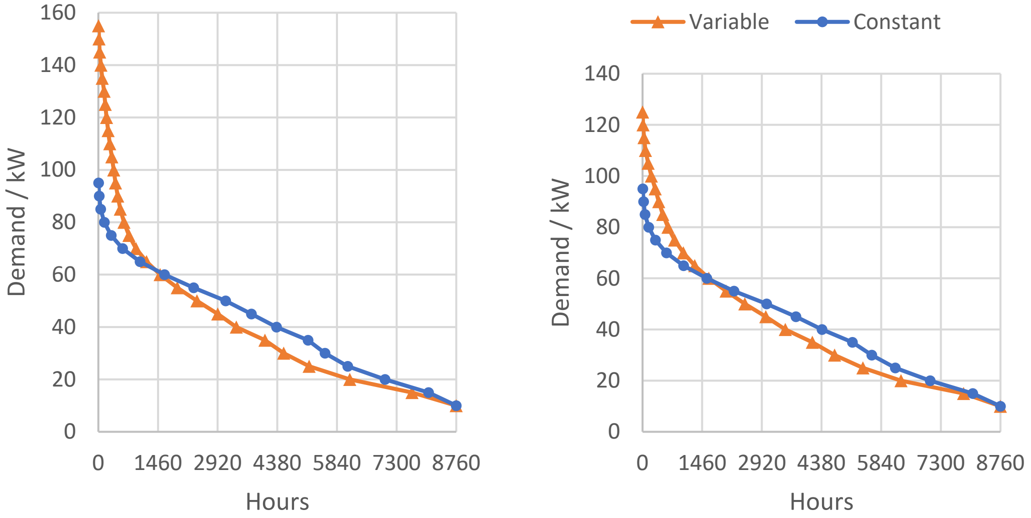

Figure 4.10. Distribution of thermal energy demand profile for underfloor heating (left) and fan coil (right) for a balanced ventilation system with heating only supplied in the heating season.

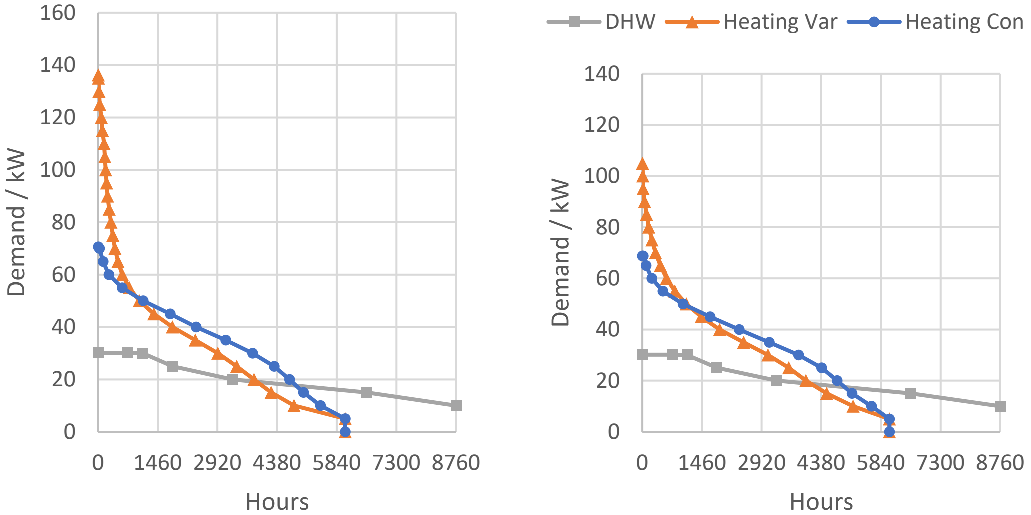

Figure 4.11. Distribution of demand profile for heating and DHW for underfloor heating (left) and fan coil (right) for a balanced ventilation system with heating only supplied in the heating season.

A secondary source is a direct replacement of the primary source: a 10 kW secondary source reduces the size of the primary source by 10 kW. This does not affect the demand curve, unlike TES. With TES, the peak is capped at the specified capacity of the primary and secondary sources. The peak is then redistributed to hours with lower demand. This in effect extends the number of hours at the system's designed capacity, shown in Figure 4.12, which is better for energy sources less able to modulate their output. As TES only redistributes the load, the minimum possible size of the heating sources is equal to the average annual demand. However, this is unfeasible due to the enormous tank size required and increased energy use due to the additional heat losses.

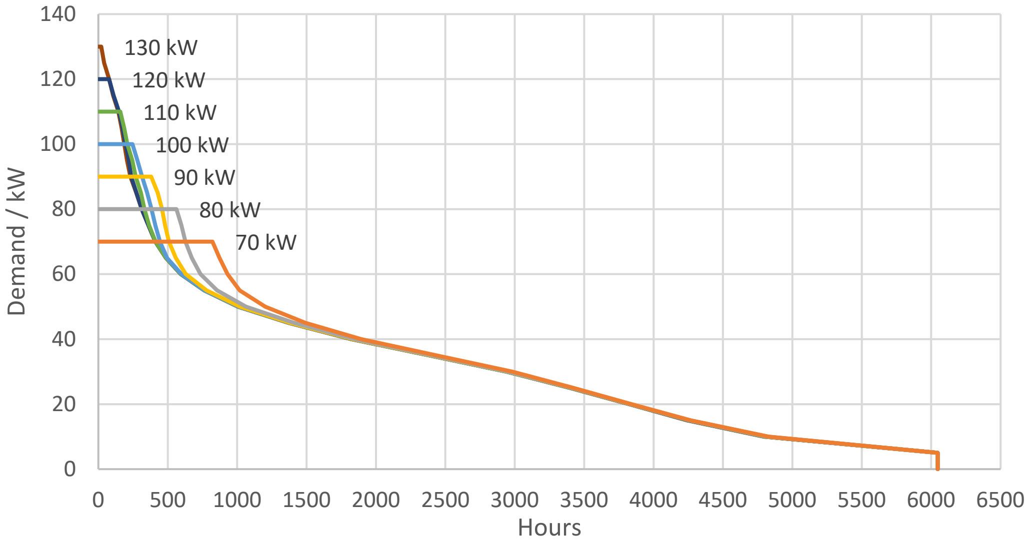

Figure 4.12. Effect of TES on heating demand profile for underfloor heating with a variable setpoint. Curves correspond to the maximum output of the heating sources with TES covering the peaks.

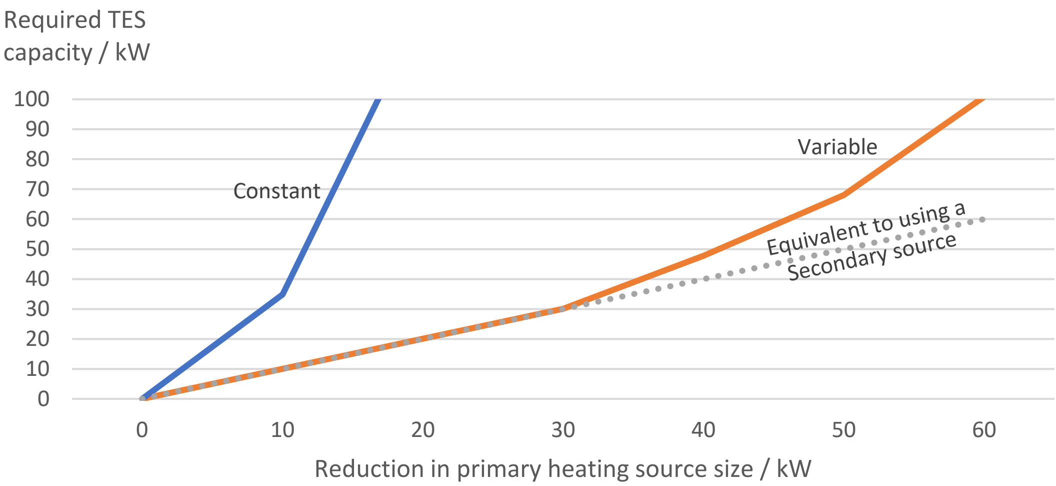

Figure 4.13. Required TES capacity to reduce the heating source size for underfloor heating.

The required TES capacity was influenced by the daily demand curve. The variable setpoint system gives a window for the tank to recharge, whereas there was less opportunity under the constant setpoint. This meant that much larger TES was required to reduce the heat source size, shown in Figure 4.13. For underfloor heating using a variable setpoint, a 30 kW reduction using TES is almost equivalent to direct replacement with a secondary heating source. For fan coils, a 10 kW reduction was possible. Beyond this point the required capacity increases exponentially. Although not feasible to effectively reduce its peak with TES, the constant setpoint still had a lower peak than the variable setpoint with optimal TES.

The size of tank required to deliver the required capacity depends on the tank temperature relative to the supply temperature, shown in Figure 4.14. A larger temperature difference requires a smaller tank and therefore a smaller initial cost. It was more efficient to achieve the temperature difference through using a lower supply temperature than increasing the tank temperature, favouring heating systems with lower supply temperatures. Furthermore, the maximum tank temperature is limited by the maximum output of the heat source or 95 °C to prevent boiling of the water. As the tank temperature increases, the higher heat loss per m² counters the reduction in heat loss due to the decreased tank size, resulting in an optimal tank temperature (and resulting tank size) with the lowest heat loss. This optimum temperature increased as the required capacity increased with lower supply temperatures having greater sensitivity. As the tank surface area was calculated based on one optimally dimensioned tank, the heat losses will likely be worse than modelled for the larger tank volumes, meaning that higher tank temperatures would be favoured. This optimum temperature is further complicated for the heat pump systems as the working efficiency decreases as the required tank temperature increases.

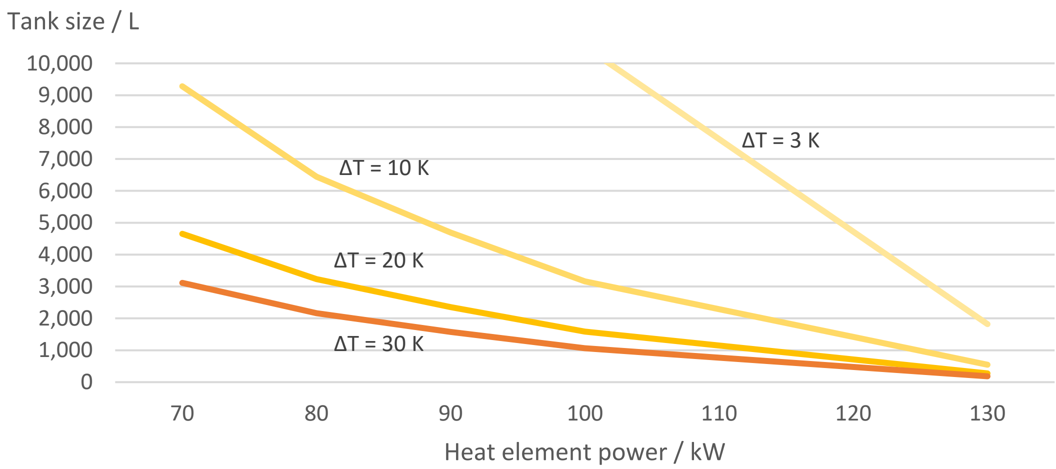

Figure 4.14. Effect of temperature difference on the required tank size to reduce the required heating source power.

For GSHP, an optimum tank size and temperature can be found by placing a cost on the tank, power source and energy cost. An example for a GSHP is shown in Figure 4.15, using estimated costs and the system efficiencies in Section 3.4.4. The example shows that the optimum TES size and temperature is not intuitive and dependent on many factors. The optimum point is constrained by reality as the required tank size was often not standard. Similarly, the cost curves would not be as smooth if using available sizes for the tank and GSHP.

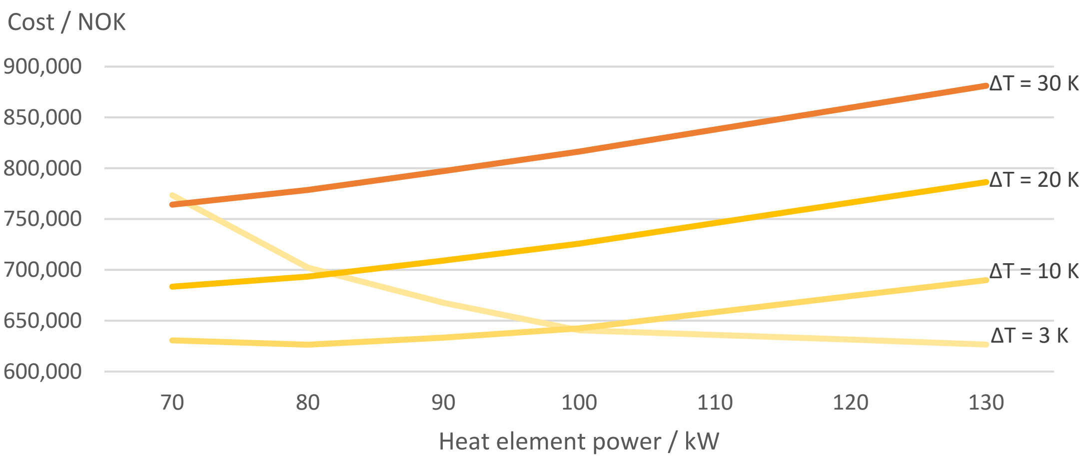

Figure 4.15. Example cost curve for TES sizing of an underfloor heating solution with a variable setpoint supplied by GSHP for 10 years. Tank = 10 NOK/L, GSHP = 2500 NOK/kW, energy cost = 0.8 NOK/kW.

The results suggest that TES at a fixed temperature was best used to buffer the top of the peak. The more the TES affected the demand profile, the less feasible it was. An electric coil for the rest of the peak would likely be cheaper than additional TES capacity. Using a constant setpoint was the most effective way to reduce the peak. Although this required around 15 000 kWh more energy, most of this could be covered by the primary source. Where the same size heat pump is used, the resulting energy cost favours the constant setpoint, shown in Table 4.4. As the heat pump covers all the load, the system efficiency is higher for the constant setpoint system. The higher proportion of direct electric heating used in the variable setpoint system resulted in a higher unit and monthly energy cost.

Table 4.4. Comparison of energy cost for supplying underfloor heating solution combined with a 70 kW heat pump for a constant and variable setpoint.

	Constant	**Variable**
System	70 kW heat pump with buffer TES	70 kW heat pump, 10 kW TES, 50 kW electric coil
Energy supplied by heat pump	325 346 kWh	292 241 kWh
Unit Energy cost of heat pump	81 940 NOK	76 060 NOK
Energy supplied by electric coil		16 922 kWh
Unit Energy cost of electric coil		14 551 NOK
Monthly energy cost	15 056 NOK	44 054 NOK
SCOP	3.13	2.72
Total energy cost	96 996 NOK	134 665 NOK

The cost curve for a district heating system resembles Figure 4.14, as the incremental sizing cost of the district heating heat exchanger was assumed small and the efficiency does not change for the different tank temperatures. The result suggests that these systems should not utilise TES. Likewise, the energy cost difference between constant and variable setpoints was similar to the energy difference. Instantaneous heating is a common solution for district heating with different building loads evened out across the network (Frederiksen and Werner, 2013). Space is saved by not having a tank, however addition control is required for managing a constant supply temperature.

4.2.2 Solar thermal collector system

The energy delivered from the two types of solar thermal collectors is shown in Figure 4.16. The total energy delivered by the vacuum tube type (VT) is lower than the flat plate type (FP) due to the smaller aperture area relative to gross area. The two types delivered a similar amount of energy for DHW at gross areas above 120 m². FP delivered far more energy to the borehole than VT. At the smaller sizes, this was mainly because there are more periods when FP was unable to reach the temperature required for DHW. At larger sizes, it was mainly because the peak output of DHW was higher than demand and so was sent to the borehole. The electric pumping power required was around 2 % of the total delivered energy for all tested solutions. This percentage decreased slightly as the system became larger.

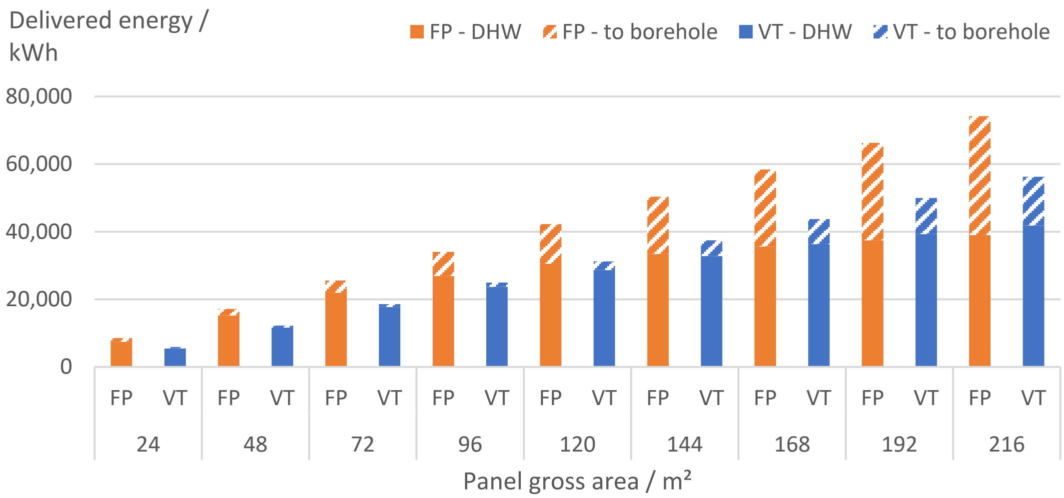

Figure 4.16. Energy delivered to DHW tank and borehole for different sizes and types of solar thermal collectors.

When scaled to aperture area, VT was more efficient, producing more DHW water per square meter, shown in Figure 4.17. This would be even more evident if a smaller water tank was used, which would lead to periods of higher tank temperatures. The high inlet temperature to the panel favours the better insulated vacuum tubes.

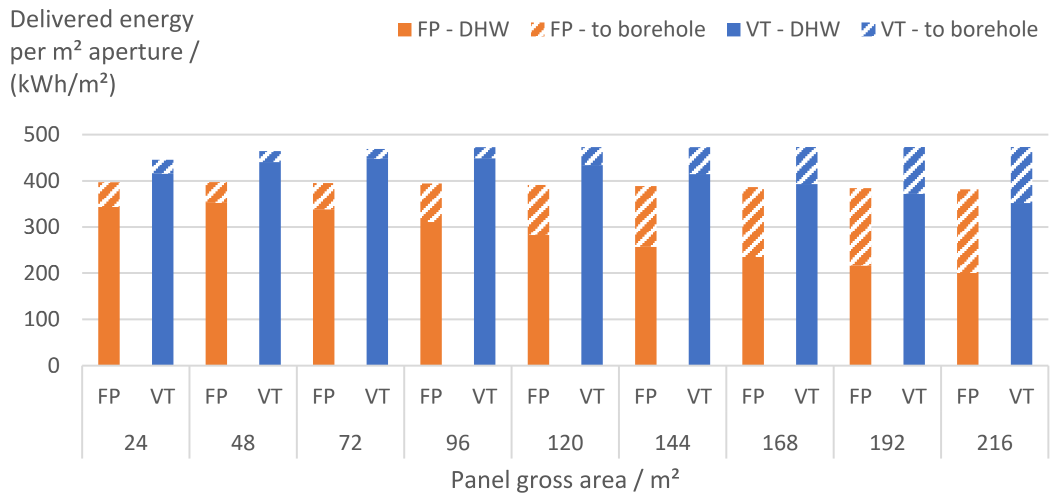

Figure 4.17. Energy delivered to DHW tank and borehole per m² aperture area, for different sizes and type of solar thermal collectors.

The total efficiency is similar for each gross size step, although the proportion of energy for DHW and the borehole changes. Both types decreased in efficiency as the area increases due to higher heat losses from the longer pipes. FP decreased at a faster rate than VT. As the solar panels were investigated for utilisation with the SAGSHP system, the quality of energy is less important than the quantity because any low temperature output can be connected to the water entering the heat pump. This can be used by the heat pump or circulated back into the boreholes. Therefore, flat panel collectors were deemed more suitable.

4.2.3 Effect of parameters on borehole sizing

The borehole cost represents the largest share of the initial cost of GSHP systems (Shah et al., 2020). The following methods to reduce the required borehole depth were examined: reduction of peak load through TES; reduction of energy demand and peak by using secondary energy source; and better energy balance by cooling and adding of solar thermal collectors (SAGSHP). The solutions were compared for different heat emitters, borehole configurations and minimum brine temperatures entering the heat pump. For the interest of comparison, the borehole field was standardised to a 4 x 4 rectangular grid for achieving minimum temperature of 0 °C and 5 x 6 for a minimum temperature of 2 °C. Both used 10m spacing between boreholes. Reducing the number of boreholes, increased the depth and temperature entering the heat pump. Increasing the number of boreholes did the opposite. If the spacing was decreased, the borehole depth increased for all scenarios.

The required borehole depth to cover the demand for solutions using underfloor heating and fan coils, with and without cooling are shown in Figure 4.18 and Figure 4.19. The borehole depth was less for solutions with cooling. The difference was more significant for the solutions achieving a minimum temperature of 2 °C. There was only a small difference in the borehole depths for single-U and double-U configurations. Again, the difference was more pronounced in the 2 °C cases. The results for underfloor heating and fan coil solutions were also similar with fan coil systems requiring less borehole depth. This was due to the higher supply temperature of the fan coils, which means the COP of the heat pump was less and so less heat was extracted from the ground.

The resulting maximum temperatures entering the heat pump in the 25th year of the simulation are shown in Table 4.5. In all the cases the minimum temperature was either 0 °C or 2 °C at the end of January. There was little difference between single-U and double-U configurations, with a maximum difference of 0.06 °C. The temperature increased with the addition of cooling. The increase was greater for the cases with a minimum temperature of 0 °C. Among the cases without cooling, those with the deeper borehole depths had the higher maximum temperature. The same was true among the cases with cooling.

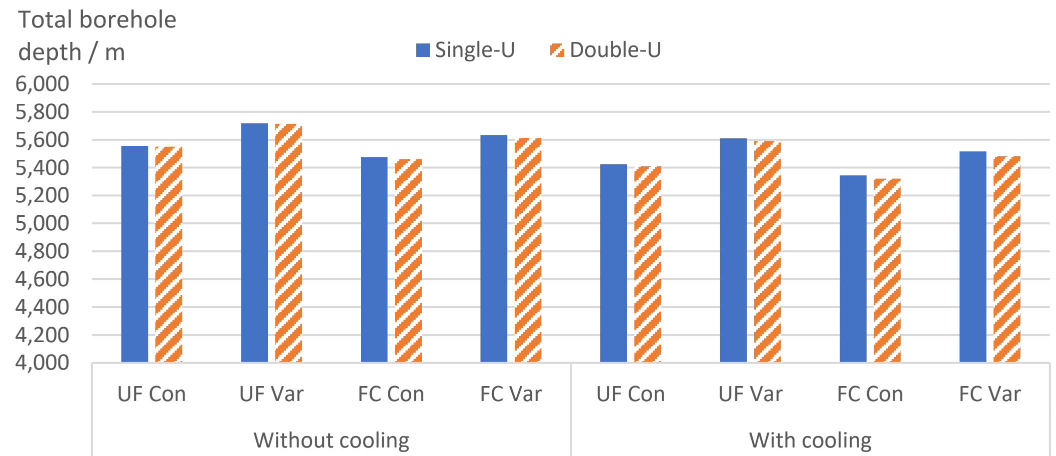

Figure 4.18. Total borehole depth required to meet a minimum temperature of 0°C entering the heat pump, comparing underfloor heating and fan coils, constant and variable setpoint, with and without cooling.

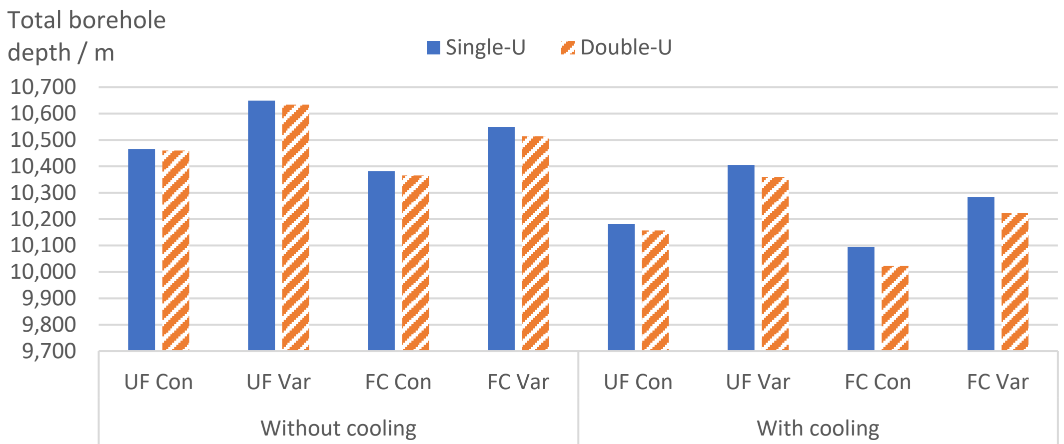

Figure 4.19. Total borehole depth required to meet a minimum temperature of 2°C entering the heat pump, comparing underfloor heating and fan coils, constant and variable setpoint, with and without cooling.

Table 4.5. Maximum temperatures entering the heat pump, comparing underfloor heating and fan coils, constant and variable setpoint, with and without cooling.

	Max temperature without cooling / °C				**Max temperature with cooling** / °C			
	UF Con	UF Var	FC Con	FC Var	UF Con	UF Var	FC Con	FC Var
Min 0 °C								
Single-U	2.07	2.33	1.96	2.22	3.73	3.93	3.68	3.89
Double-U	2.07	2.32	1.94	2.21	3.68	3.9	3.62	3.84
Min 2 °C								
Single-U	3.11	3.27	3.05	3.21	4.11	4.15	3.95	4.11
Double-U	3.11	3.26	3.04	3.2	3.97	4.12	3.93	4.08

Using the ground for cooling both reduced the borehole depth and increased the maximum and average temperature, improving the efficiency of the heat pump. The reduced drilling cost easily compensated for the additional energy cost for the pumps. There was little difference between single-U and double-U configurations or between underfloor heating and fan coils. Therefore, the following investigations are presented using the double-U configuration, underfloor heating and summer cooling. This effect could be further enhanced at a relatively small cost by changing the building design to increase the cooling demand and therefore increase the amount of thermal energy sent to the boreholes, as shown by Javed et al., (2019).

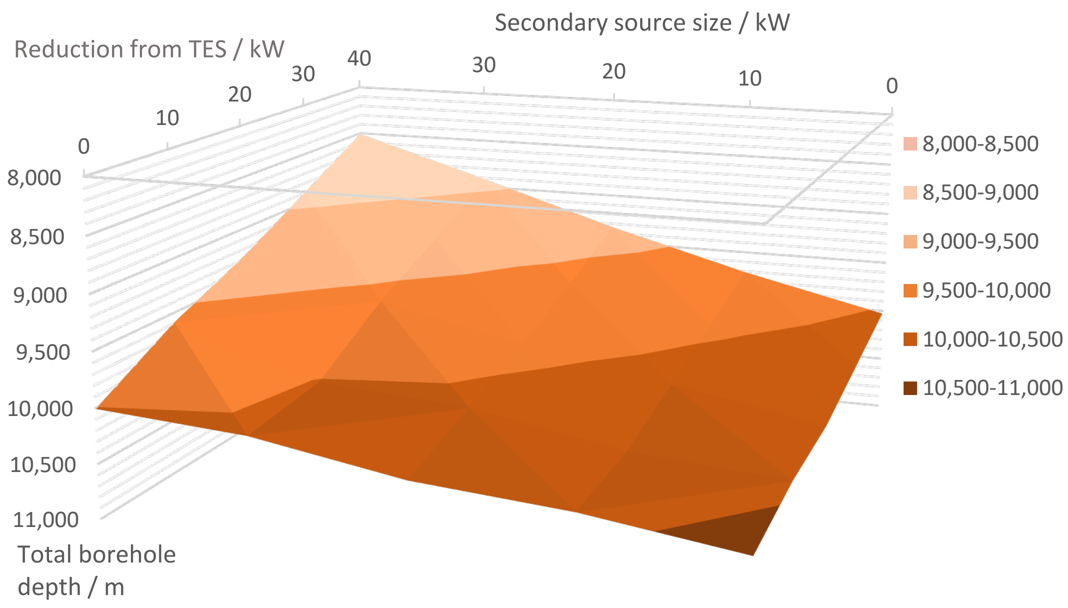

Figure 4.20. Effect of reduction of GSHP size by secondary heating source and TES on borehole depth.

The effect on the borehole depth of using TES was similar to using a secondary source, presented in Figure 4.20. Increasing the size of TES or the secondary source reduces the borehole depth by a similar amount. The two appeared compatible and so can directly replace each other. The temperature reaching the heat pump did not change significantly with a variation of just 0.08 °C. It should be noted that the reduction from TES is not equal to its capacity. In Section 4.2.1, it was shown that past a certain point the capacity increased faster than the reduction of the required primary source. As there was no difference in the achieved borehole depth, this further suggests that using a secondary source is preferable. TES should only be used where its capacity matches the reduction.

The effect of solar thermal collectors on the borehole depth is shown in Figure 4.21. No secondary sources or TES were used in the cases presented. The larger the area of solar panels, the higher the quantity of energy inserted into the boreholes. This reduced the required borehole depth and increased the maximum temperature entering the heat pump. The change in depth showed a roughly linear relationship to the area of solar panels. The reduction was greater for the cases requiring a minimum temperature of 2 °C than those requiring 0 °C. The increase in maximum temperature was greater for the 0 °C cases due to the shallower borehole depth, however the average temperature was still higher for the 2 °C cases. The trend was the same for variable and constant schedules, with a slight convergence of maximum temperature as the area of solar thermal collectors increased.

The benefits of adding solar thermal collectors were independent of utilising TES and secondary sources. Adding 216 m² of solar thermal collectors to the cases shown in Figure 4.20 did not change the overall shape of the surface. The depth of all points was reduced by around 1 200 m. The points with shallower depths had slightly less reduction and those with deeper depths, slightly more. The maximum temperatures were increased by up to 1.8 °C. As shown in Figure 4.21, the temperature increases were higher in the shallower boreholes.

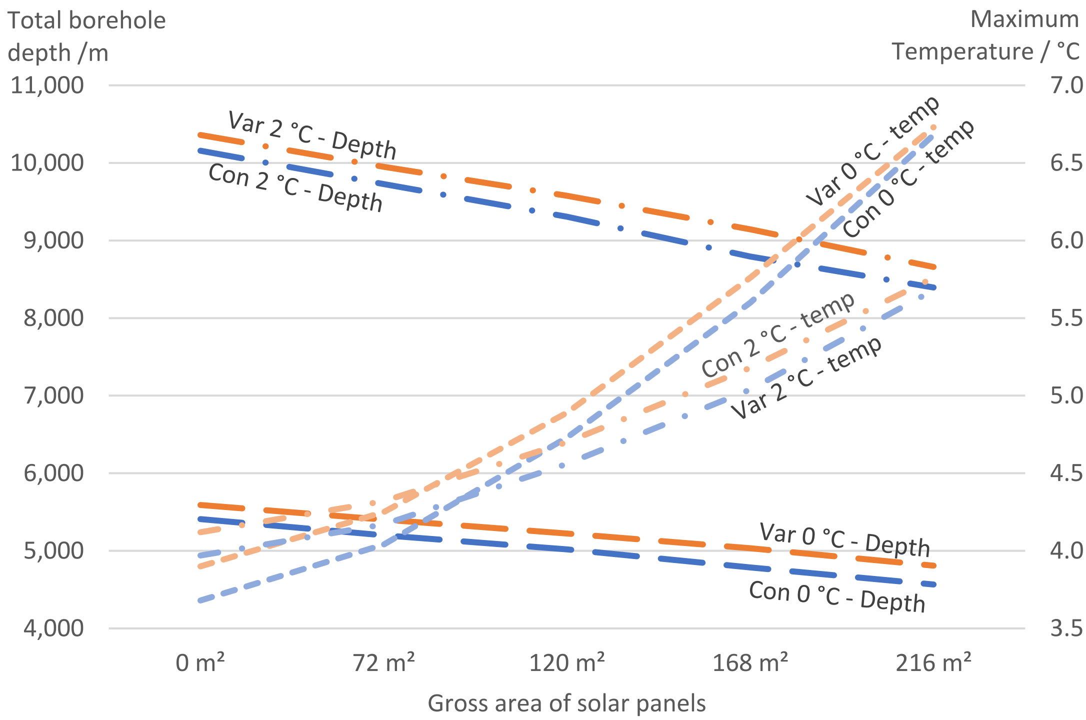

Figure 4.21. Effect of solar thermal collectors on borehole depth and maximum temperature.

More optimal configurations are possible for each of the presented results. For example, the boreholes could be arranged in a way that each borehole has a larger exposure to the ground. These optimisations would likely exaggerate the trends discussed in this section. Another important factor to consider for the life cycle costing is the required pump size and energy use, which can be significant at the depths used here (Gehlin et al., 2016). Further optimisation of the borehole field and calculation of the borehole pump were outside the scope of this thesis.

4.2.4 Heat pump system

The number of heat pumps and their cumulative size affected the efficiency and required energy cost. The seasonal coefficient of performance (SCOP) for the system is the combined efficiency of the heat pumps and the secondary electrical heating. This is shown for a fan coil solution with a variable setpoint in Figure 4.22 and a constant setpoint in Figure 4.23. Each 10 kW reduction in heat pump sizing was matched by a 10 kW increase in direct electric heating. For both variable and constant setpoints, it was more efficient to use multiple smaller heat pumps than one large heat pump, as there was less intermittent use. The efficiency difference between the number of heat pumps reduced as more direct electric heating was used. The benefit of three heat pumps over two heat pumps was small.

When multiple heat pumps were used, the highest system efficiency was achieved with either a 0 kW or 10 kW size reduction. Single heat pumps required a 20 kW size reduction. This small reduction improved efficiency as the intermittent use of the heat pumps was decreased. As the size of the heat pump(s) was further reduced, the system efficiency reduced due to the higher proportion of energy provided by direct electric heating. The reduction was more gradual for the variable setpoint schedule. The higher minimum brine temperature of 2 °C gave an increased efficiency of between 0.12 to 0.18 for a variable setpoint and 0.13 to 0.22 for a constant setpoint. The improvement in efficiency increased as the heat pump size decreased.

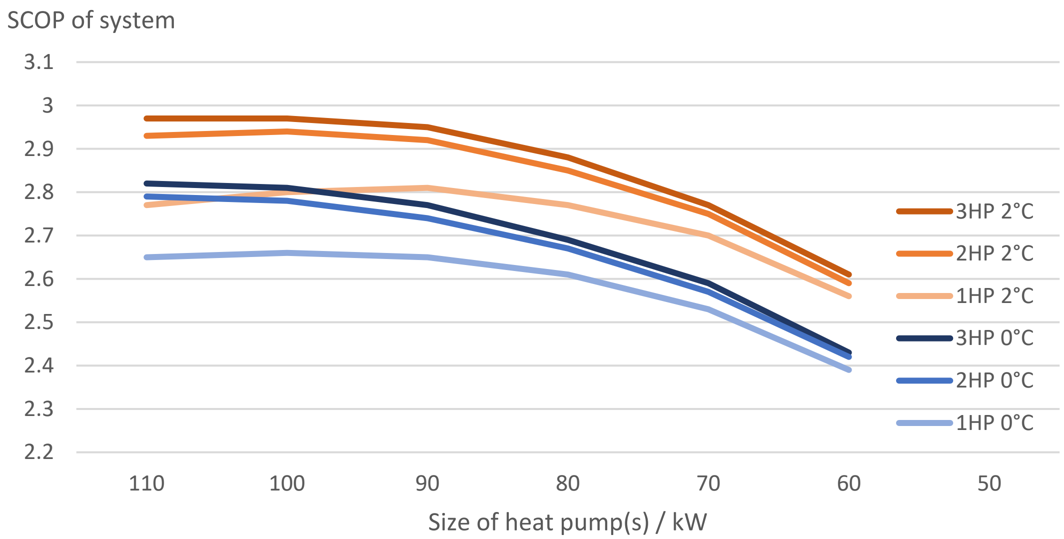

Figure 4.22. System SCOP of different heat pump system arrangements and minimum brine temperatures for a solution using fan coils and a variable setpoint schedule.

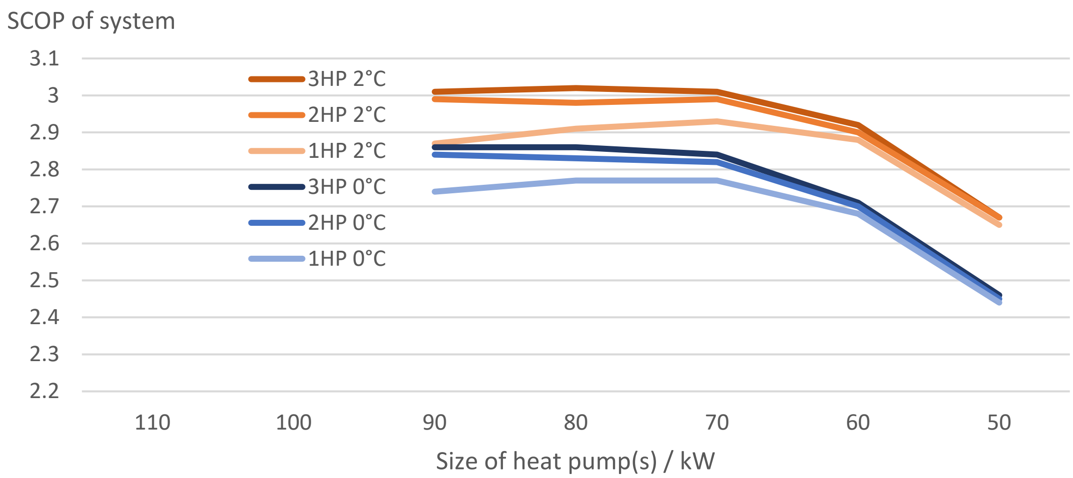

Figure 4.23. System SCOP of different heat pump system arrangements and minimum brine temperatures for a solution using fan coils and a constant setpoint schedule.

The constant setpoint schedule allowed for a 20 kW smaller heat pump size. The SCOP was also slightly higher, likely due to the more constant demand over a 24 hour period shown in Figure 4.6.

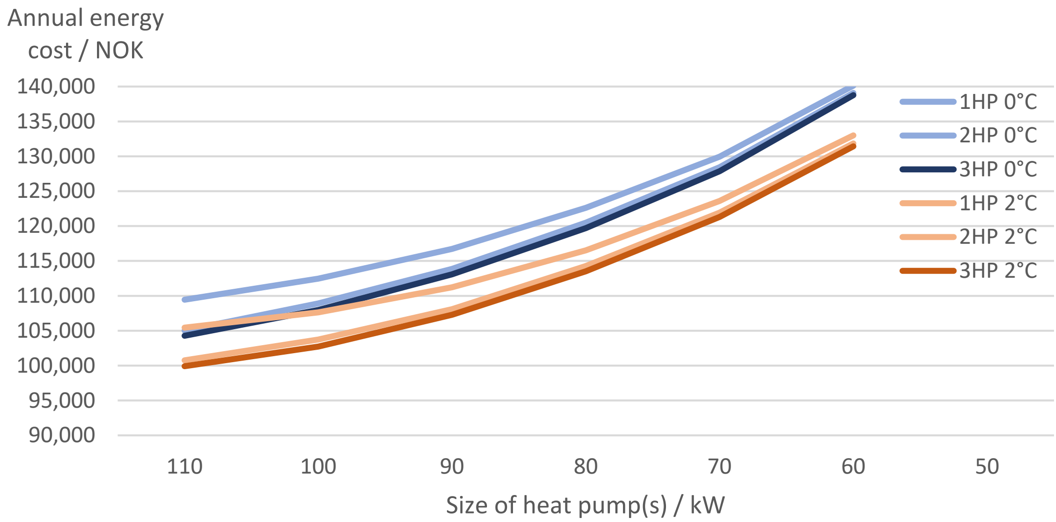

Figure 4.24. Annual energy cost of different heat pump system arrangements and minimum brine temperatures for a solution using fan coils and a variable setpoint schedule.

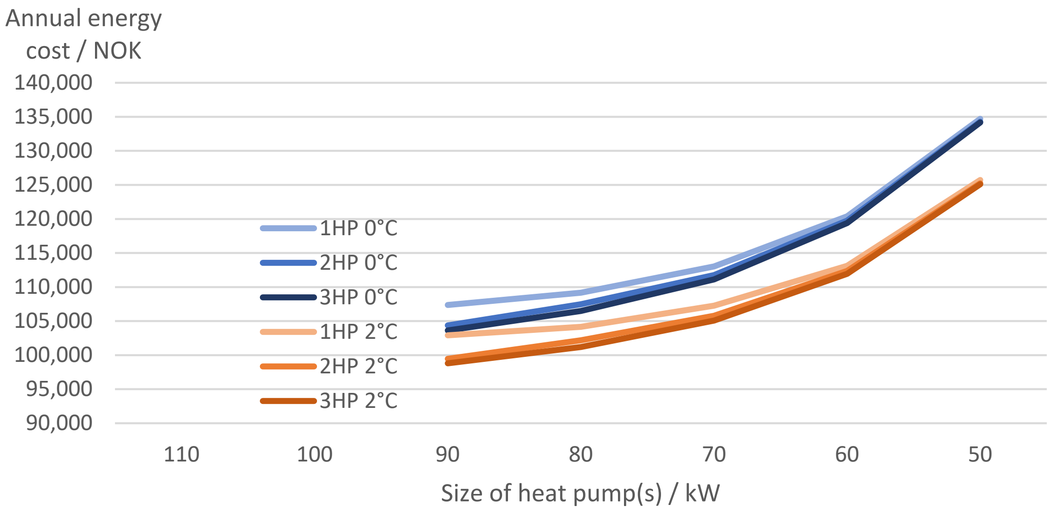

Figure 4.25. Annual energy cost of different heat pump system arrangements and minimum brine temperatures for a solution using fan coils and a constant setpoint schedule.

The resulting energy costs, shown in Figure 4.24 and Figure 4.25, were lowest when all the load was covered by heat pumps. The cost increased exponentially as the size of the heat pump(s) reduced, as the lower efficiency of direct electrical heating increased both the unit cost and monthly cost due to a higher peak electricity demand. Using more than two heat pumps gave a negligible reduction in cost. The constant setpoint gave the lowest energy cost. The higher brine temperature resulted in a saving of around 5 000 NOK; not enough to justify the extra drilling required for the deeper borehole depth shown in the previous section. If a borehole drilling cost of 350 NOK/m is assumed (Norsk Varmepumpeforening, 2021), the simple payback time is over 300 years.

In the presented results for multiple heat pumps, similar sized units were used. For example, a three heat pump system sized to 90 kW was comprised of three 30 kW heat pumps. Varying these sizes had a negligible effect on efficiency and cost, likely because all heat pump data was scaled. Nevertheless, using multiple similar sized units has the advantage of better redundancy and the ability to repair a unit without impacting building occupants, particularly when demand is low.

These patterns were similar for the other systems utilizing heat pumps, SAGSHP and LTDH, shown in Figure 4.26. SAGSHP had a slightly higher SCOP due to the higher brine temperatures from the charged boreholes. In addition, the solar thermal collectors supplied a proportion of the DHW load, reducing the demand from the heat pump to produce these higher temperatures at a lower COP. LTDH required a smaller heat pump size as it could achieve a higher SCOP due to the higher temperatures provided by the district heating grid compared to boreholes.

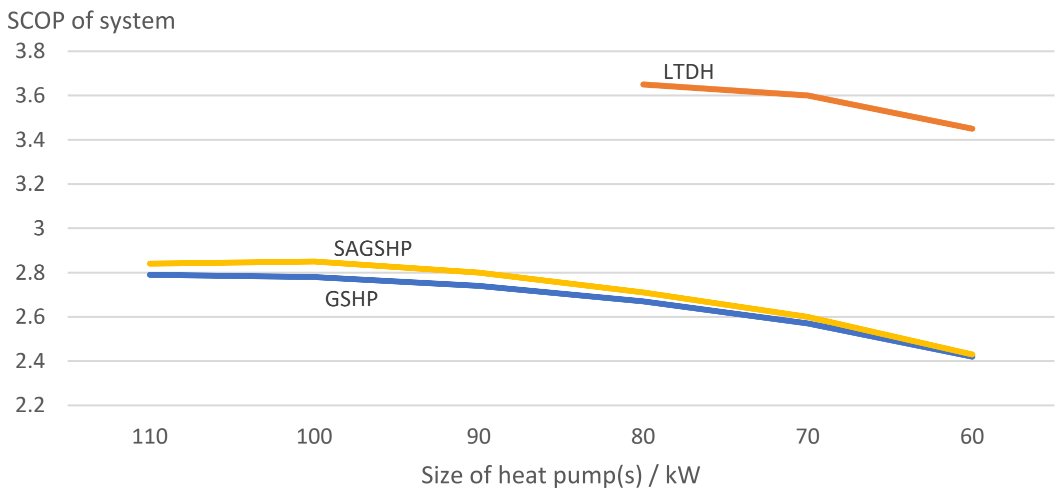

Figure 4.26. System SCOP for different heat pump energy supply systems, using two heat pumps.

4.2.5 Comparison of Energy supply systems.

All the simulated systems are shown in Figure 4.27 for solutions utilising fan coils and a variable schedule for comparison. The presented GSHP, SAGSHP and LTDH systems use two heat pumps to cover the entire building load. The unit energy cost is the total cost for each kWh of delivered energy. The monthly energy cost includes the fixed monthly costs and the cost for the peak electricity demand in each month. The delivered energy and energy cost of other system combinations follow the trends presented in the previous sections.

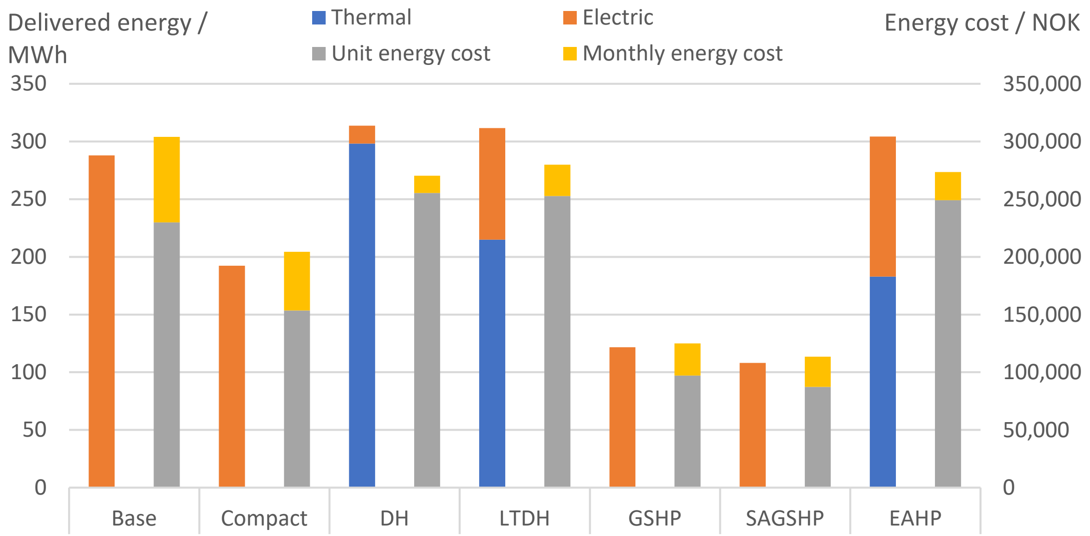

Figure 4.27. Annual delivered energy and energy cost for each energy supply system, using a fan coil and a variable schedule.

The decentralised base case and compact units had lower gross energy demand due to fewer distribution losses. The base case had the highest energy cost due to having the highest electric peak load of all the systems. The integrated heat pump of the compact unit greatly reduces the required electric energy and peaks resulting in a lower delivered energy and energy cost.

The energy demands for the central systems were higher due to the higher distribution losses. District heating (DH) had the highest delivered energy and unit energy cost. However, as most of the energy was supplied by district heating, the monthly energy cost was the lowest of all the solutions. The LTDH had similar delivered energy to the DH but a higher proportion of this was electricity used by the heat pump. This resulted in a lower unit energy cost, due to electricity prices often being under the thermal energy price, which for the case study was based on the monthly electricity spot price. However, the higher monthly cost meant that LTDH had a higher energy cost than DH. The cost of thermal energy for LTDH would have to be 5.5 % less than current costs to be competitive.

The systems integrating GSHPs had the lowest delivered energy use and energy cost as a large proportion of the energy demand was sourced from the ground. The addition of solar collectors (SAGSHP) provided even more “free” energy, which reduced the delivered energy and energy cost.

Despite the use of a heat pump, the higher gross energy demand of the exhaust ventilation solution meant that the EAHP case had a high delivered energy. As the total capacity of the exhaust air heat pumps, limited by the airflow, was not enough to meet all the energy demand, 30% of the thermal energy had to be provided by a secondary source. This resulted in a SCOP of 2,08. Using an EAHP with district heating as the secondary source, as shown in Figure 4.27, gave an energy cost of 273 574 NOK. Despite using 3657 kWh less energy due to reduced system losses, using direct electric heating instead cost 47 745 NOK more. This was the most expensive system solution in terms of energy cost.

4.3 Control strategies

Thermal energy storage has been shown in Section 4.2.1 to have limited application in reducing the size of the primary heating source. Using a secondary source for peak loads was a better option in most cases. The performance of TES can be improved by using a model predictive control to leverage the storage optimally, by fluctuating the setpoint temperature to charge and discharge as required. This can reduce the required tank size as the energy capacity can be achieved using temperature instead of volume. The control only increases the tank temperature when necessary. Otherwise, a low tank temperature is kept, reducing heat losses. Alternatively, the control can improve the flexibility of a building by utilising the extra capacity to shift peaks in demand to hours when there is less demand on the electricity grid and so lower prices. The potential for increasing flexibility is examined in this section.

4.3.1 Electricity prices

The electricity price varies hourly based on a market price dictated by supply and demand. As both elements are influenced by weather and vary year on year, it was important to define a “standard year” in the same way that a standard meteorological year is used for energy simulations. The past 8 years of electricity price data was analysed via price distribution over the year, shown in Figure 4.28 and price variation in a 24 hour period, shown in Figure 4.29. Prices are given in øre, with 1 NOK equal to 100 øre.

Both 2018 and 2020 were regarded as anomalous years. Electricity prices were extremely low, sometimes below zero, in 2020 due to the depressed demand caused by the coronavirus pandemic. Conversely, prices were unusually high in 2018 due to a lack of precipitation constricting the supply of hydropower combined with increased prices for imported energy due to higher CO_2 taxes (NVE, 2019). This year also had significantly more price variation in a 24 period than the other 7 years. Prices were also volatile in 2016, although this was due to a few extreme 24 hour periods, of which three were over 100 øre in difference (and so not visible in Figure 4.29). The interquartile range and mean spot price for 2016 were similar to the other “normal” years. The mean spot price was around 25 øre/kWh for most years. It was slightly less in 2015 and slightly more in 2019, due to the CO_2 taxes. 2019 had slightly more variation than the previous years. This could be due to the increasing number of grid interconnections (Energifakta Norge, 2021). When prices are high in other countries, it is appealing for Norwegian energy producers to sell their electricity abroad, in turn raising prices nationally. As both CO_2 taxes and grid interconnections will exist in the coming years, 2019 was considered a good example of a typical year for the coming years. The general variation of prices was less than 10 øre in all years except 2018. This flat profile is due to the high proportion of hydropower in the Norwegian electricity mix, as it is possible to regulate the generation capacity of hydropower without effecting efficiency.

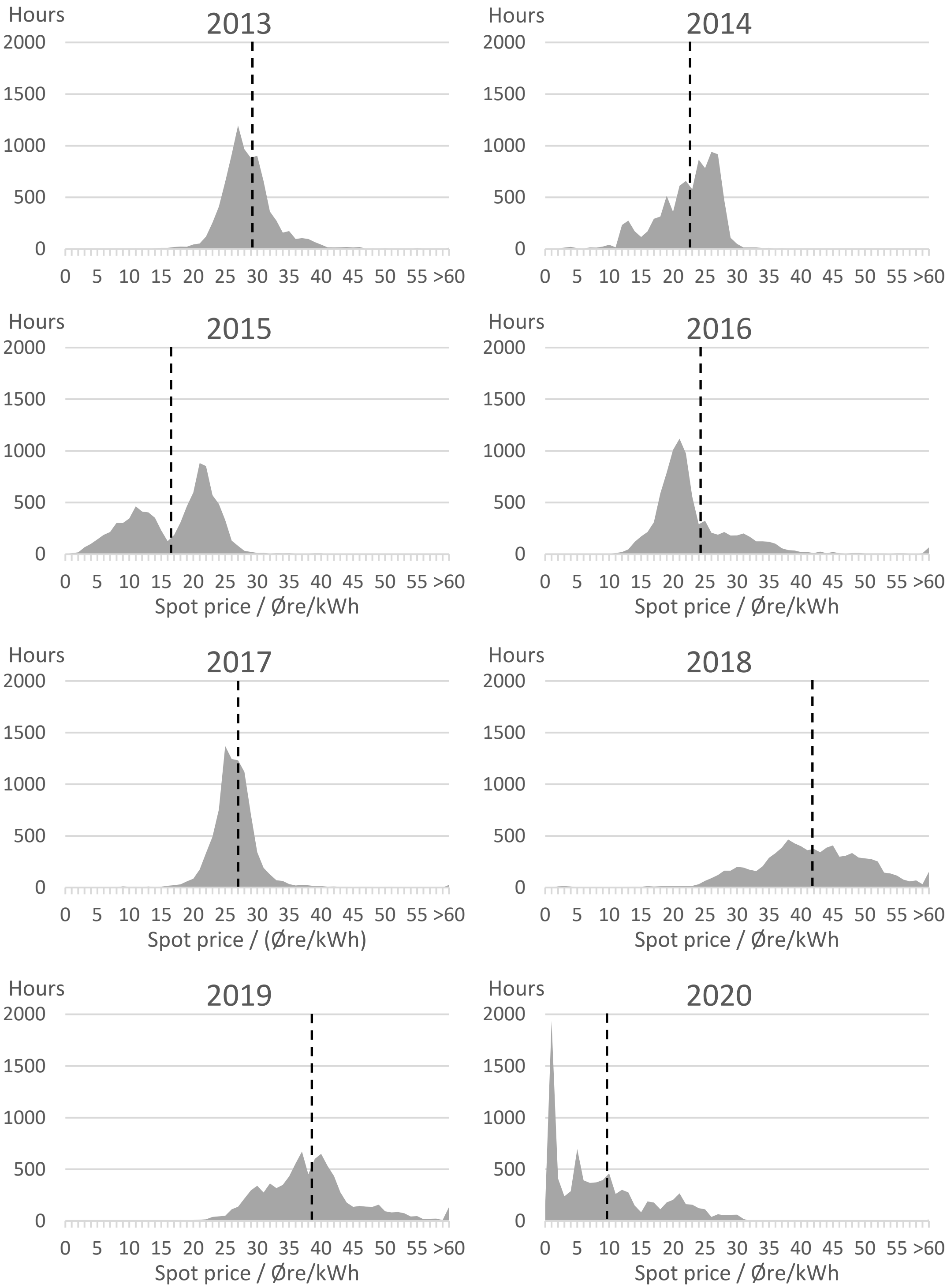

Figure 4.28. Distribution of spot prices for the last 8 years. Average spot price indicated by the dashed line.

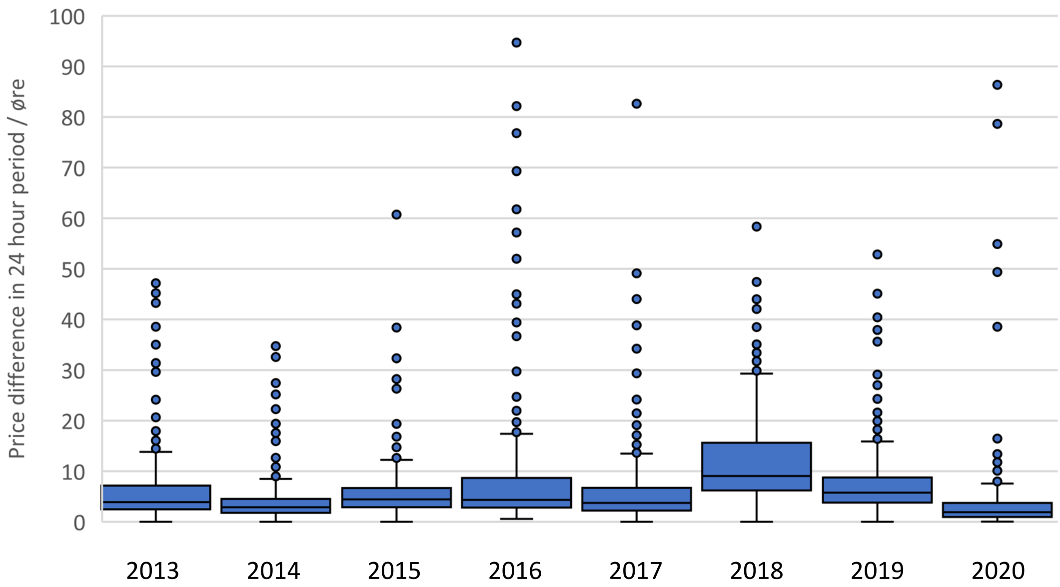

Figure 4.29. Distribution of the price difference between the highest and lowest spot price in each 24 hour period in a year, for the last 8 years of spot price data.

The potential unit energy cost saving from shifting the energy demand of the BoKlok project was analysed for all 8 years shown in Table 4.6. Demand shifting was possible within each 24 hour period, reflecting the availability of price forecasting from Nordpool. The effect of different system efficiencies was also included to reflect the use of heat pumps in the solution-sets.

Table 4.6. Annual savings in unit energy cost from shifting demand to lowest price within a 24 hour period. Prices in NOK.

	2013	2014	2015	2016	2017	2018	2019	2020
El replacing EL / NOK	13 029	10 321	13 496	20 311	12 962	30 074	16 177	9 371
El replacing a HP with a COP of 4 / NOK	0	0	0	26	0	6	0	0
HP replacing a HP with the same COP / NOK	3 257	2 580	3 374	5 078	3 240	7 519	4 044	2 343
HP with a COP of 3 replacing a HP with a COP of 4 / NOK	272	298	209	1 252	369	1 340	291	247

The potential savings were greatest when electricity replaced electricity, ranging from 9 371 NOK to 30 074 NOK. The yearly results reflect the pattern shown in Figure 4.29. The high price variation in 2018, resulted in the largest savings. The low variation in 2020 resulted in the lowest savings. When 2016, 2018 and 2020 are excluded, the variation in annual savings was less than 6 000 NOK. Where a heat pump is used for demand shifting, maintaining its efficiency, the possible savings were divided by that efficiency. In Table 4.6 the savings are a quarter of the electricity savings as the COP is 4. The more efficient the heat pump, the lower the potential savings from load shifting. This was even more pronounce when the efficiency at the shifted time was lower than the original time of the demand. As the energy has to be stored, it will likely require a higher temperature than at the point of demand, reducing the efficiency of the heat pump. With a COP difference of 1, the savings are reduced to a few hundred NOK. Where the shifted load was supplied using an electric heater instead of a heat pump, there were no savings in 6 of the 8 years. This indicates that shifting of the DHW load would not be worthwhile in solutions utilising heat pumps.

The savings mirror the potential benefit to the electricity grid. As the COP of heat pumps already reduces the load on the grid, shifting it to another point in time has less benefit. Flexibility control should therefore be focused on direct electrical systems. These are well suited to control as they can be started and stopped nearly instantly with little energy loss.

4.3.2 Demand profile optimisation

The potential for electrical systems was further analysed, accounting for the capacity of the electrical heating source and the size of the thermal energy storage. This model was then applied to the base case. The monthly cost for the electrical peak power was also considered.

The unit cost saving increased with TES until the required storage point to achieve the maximum possible cost saving was reached, as shown in Figure 4.30 and Figure 4.31. There was a diminishing rate of return, as adding TES increased the energy use, with each additional water tank adding further heat losses. The maximum possible cost saving was determined by the coil size, with a smaller coil resulting in a smaller saving. However, the increase with each increase in coil size was also subject to diminishing returns. The savings are less when using a constant schedule than when using a variable schedule.

The optimisation resulted in the peak load of each month equalling the capacity of the selected coil. This meant that the peak cost was higher for the optimised case than the unoptimised case, when the same coil size was used. For the variable setpoint schedule using a 120 kW coil, shown in Figure 4.30, the peak cost was an extra 9835 NOK, outweighing the possible unit cost saving. For the constant setpoint schedule using a 90 kW coil, shown in Figure 4.31, the peak cost was an extra 6288 NOK, also outweighing the unit cost saving.

For each 10 kW coil increase, the peak electricity cost increased 6480 NOK. The difference in peak cost was larger than the unit cost savings. The optimisation would only be worthwhile if the peak coil capacity is lower than used without optimisation. This greatly reduces the potential return from investing in additional TES.

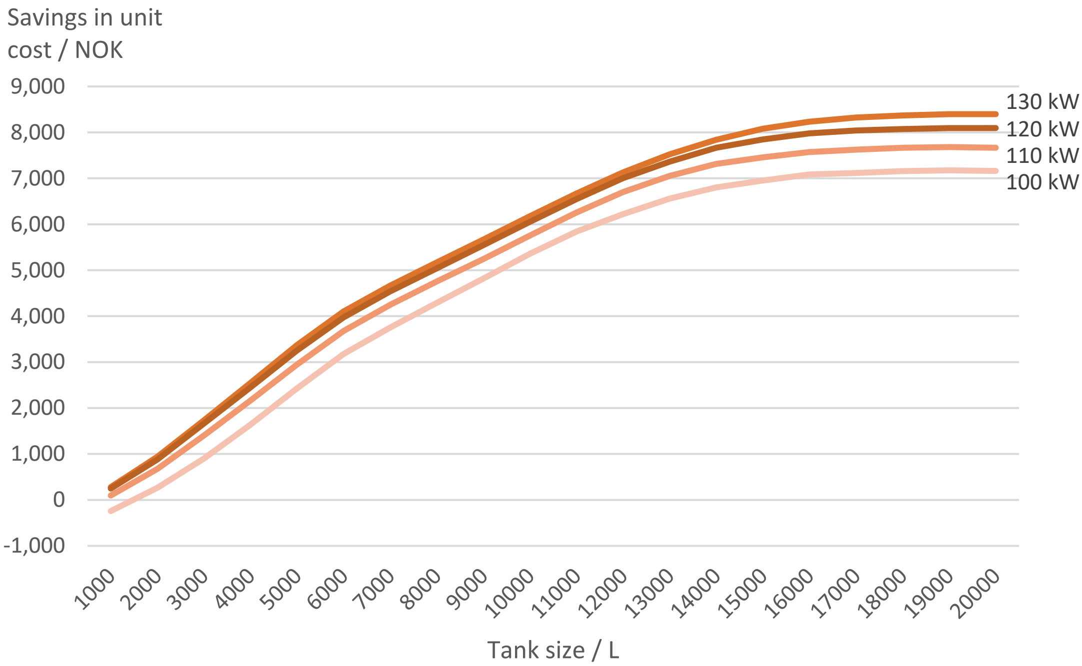

Figure 4.30. Unit cost savings from TES control for different tank sizes and heating system capacities for a variable schedule. The peak demand without optimisation was 120 kW.

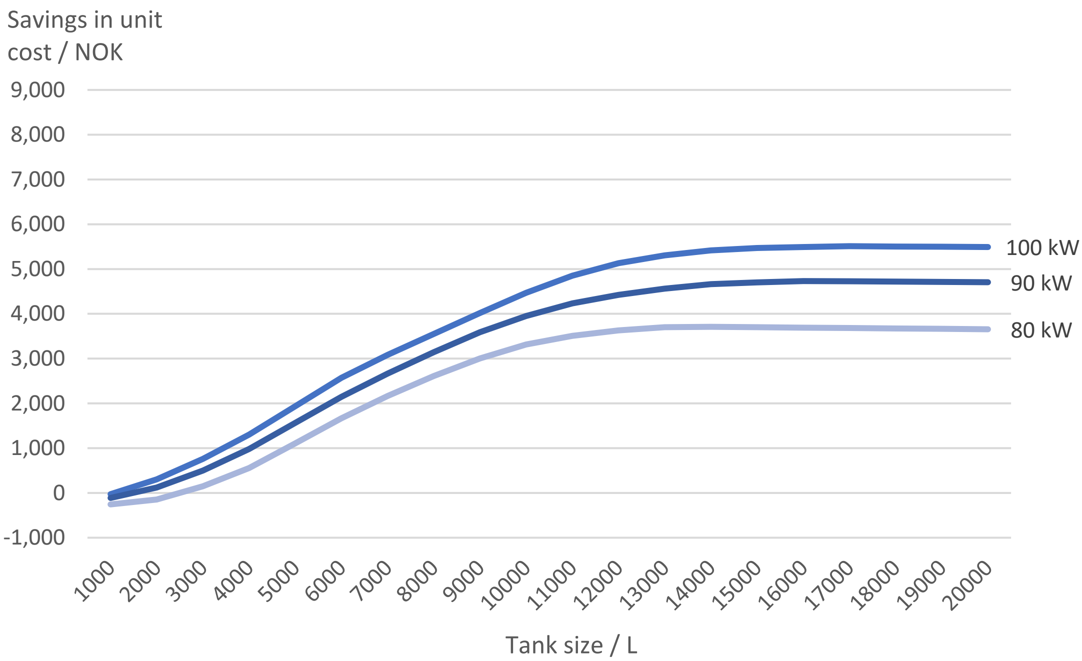

Figure 4.31. Unit cost savings from TES control for different tank sizes and heating system capacities for a constant schedule. The peak demand without optimisation was 90 kW.

There is still potential where there is already a large volume of storage. In the base case, 100 L water tanks are placed in each apartment, giving 6800 L of potential TES. With each tank equipped with a 3 kW heating element, there is also a large potential heating capacity of 204 kW. The optimisation results are shown in Table 4.7. The result is an approximation as the tanks and heaters in each apartment were considered as one unit. The peak demand without optimisation was 124 kW.

Table 4.7. Cost savings from optimised used of TES in the base case. Prices in NOK. Negative values represent savings.

Coil size	**100 kW**	**124 kW**	**204 kW**
Unit cost difference. Min 65°C / NOK	-1752	-2222	-2661
Unit cost difference. Min 55°C / NOK	- 10090	-10547	-11010
Peak cost difference / NOK	-5265	10260	62127

The general findings hold true in the applied case. The savings in unit energy cost are outweighed by the peak energy cost when a minimum tank temperature of 65°C was used. The increase in peak cost was greater than the increase in unit cost savings, for each increase in coil size. Reducing the coil size below the peak demand of the unoptimised case, resulted in a net saving.

Unlike in the general case, it was possible to decrease the minimum tank temperature. The tank temperature was allowed to drop to 55°C, which was assumed enough to meet DHW and legionella requirements as the tank temperature was regularly raised above 65°C in a 24 hour period. The lower temperature resulted in lower energy use than the unoptimised case due to the reduction in heat loss. It also increased the energy capacity of the TES, allowing for more load shifting. The combination increased the unit cost savings for all the coil sizes. The difference between the coils was similar to the 65°C minimum temperature. The best method for achieving significant savings was still to reduce the coil size.

The annual saving from implementing the control strategy with a minimum tank temperature of 55°C and coil capacity of 100 kW was 15 355 NOK, a 5% reduction of the unoptimised case. However, this requires each apartment's tank to have a control unit. This needs to be cheap as the saving per apartment was only 226 NOK per year.

5 Discussion

The composition of the energy-efficient solutions found through simulation closely matches the results of the statistical review, especially in the Norwegian context. Ground source heat pump solutions combined with balanced ventilation offer the lowest delivered energy and energy cost. These systems were most efficient with multiple heat pumps and small secondary electric heating to cover the peak. The difference in energy cost of the other simulated energy supply systems was small. In this section, key features of the results are contextualised by additional factors which could influence the choice of solution.

5.1 Constant or Variable setpoint

Maintaining a constant setpoint or allowing it to setback 2°C during the night affected the energy demand and peak demand without altering thermal comfort. The difference in demand appears to be proportional to the energy demand. If the exhaust ventilation system and balanced ventilation system are used to represent a building with a high and a low demand, the difference is larger for the high demand than the low demand. However, the difference between the peak load of constant and variable schedules was almost the same for the high and low demand. The lower peak demand of the low demand system means that this absolute difference is relatively larger. If the trends for the difference in energy demand and peak loads are extrapolated, the difference in peak load per kWh saved increases. Therefore, there is a point at which it becomes more sensible to use a constant schedule as the saving from the reduced peak sizing outweighs the additional energy use. The lower peak demand when using a constant setpoint would allow for smaller emitters and energy supply equipment reducing initial costs. Ground source heat pump systems especially benefit, as the constant setpoint allows for smaller heat pump sizes and borehole depths, each a significant cost saving.

The resulting cost of the difference in energy demand depended on the heat source. For heat pump systems, the cost was reduced by the COP and was minor when the monthly cost for peak electricity was factored in. Furthermore, the lower daily variation of the constant schedule allows for the heat pump to work for longer at its nominal working range. The improved COP could outweigh the additional energy demand. The cost difference was greater for the direct electric and district heating systems. The combination of a higher energy cost and smaller savings from the reduction in peak mean a variable setpoint would likely be better for these systems.

A middle way is possible by using a ramp up time to reduce the peak of the variable setpoint schedule. This would result in an energy use and peak demand between the two studied schedules. The optimum point on this spectrum depends on the value assigned to saving energy or reducing the system peak.

5.2 System cost

Initial cost is an important factor, especially for affordable apartments, as the development cost directly impacts the purchasing cost of the apartments. District heating and direct electric solutions have low initial costs but were shown to have the highest ongoing energy costs. The competitiveness of district heating is sensitive to the price model used. In the studied case, the heat cost was tied to the monthly electricity spot price. In Sweden, district heating is priced competitively against other heating sources and is cheaper than electricity (European Commission, 2018).

Ground source heat pump systems are relatively expensive, however most of the cost is for the borehole drilling. Several possibilities for reducing the borehole depth were analysed, including TES, secondary heating sources and solar thermal collectors. These options are worth implementing if the savings from the reduced borehole drilling is more than the cost of these systems. Using a low temperature district heating network instead of boreholes saves this large initial cost, but again increases the ongoing energy cost. Furthermore, such a solution is dependent on the local availability and highly sensitive to the price of the supplied heat. The compact unit would likely be the most expensive option, despite saving some costs from fewer distribution pipes. The costs could be reduced by sharing the units between several apartments, better utilising the capacity of the unit. It would be likely that the electrical heater will be used more in this arrangement increasing the cost of energy. There are also the practical problems of where to place the unit and how to measure each apartment's energy use.

To achieve a clearer picture of which solutions would be cost-effective over the long term, it is necessary to carry out a full life cycle costing accounting for both the initial and ongoing energy costs.

5.3 Thermal comfort and Occupant behaviour

Each apartment was modelled as a single zone and therefore the thermal comfort of the entire apartment was assessed as one uniform space. In reality, there is a desire for different temperatures in different rooms. There is a tendency in Norway to have a higher setpoint in bathrooms and a lower setpoint in bedrooms (Berge and Mathisen, 2013). The bathrooms are planned with electric underfloor heating in order to have the possibility of heated bathroom floors all year round, as this is a comfort issue for many Norwegians. The bathrooms accounted for 10% of the heated floor. It could be assumed that at least 10% of the heating would be provided by these floors, increasing the energy cost of the heat pump systems. The energy use would likely be higher than simulated if this underfloor heating was on all year at the desired higher setpoint. The preference for colder bedrooms can be difficult to achieve in highly insulated dwellings as there is a reduced ability to differentiate room temperatures (Georges et al., 2016). The room temperature is limited by the supply air temperature of the ventilation, which is often not adjusted lower by the occupant. Instead, occupants have been documented opening windows, even in winter, to achieve the desired temperature. This would again result in a higher energy use than simulated.

Occupant behaviour can also drastically affect the loads, for which standard values were used in the simulations. This greatly influences the overall performance as DHW, equipment and lighting account for a high percentage of the energy demand in low-energy buildings. In the Løvåshagen project mentioned in the literature review, actual demand was 25% higher than calculated for the low-energy apartments and 70% higher for the passive house apartments due to additional window ventilation and year-round bathroom floor heating (Berge and Mathisen, 2013). Detailed measurements of 9 apartments (6 built to a low-energy standard and 3 to passive house) showed that this variation is present in heating, DHW, ventilation and equipment use (Nesland, 2010). Therefore, there is not a single identifiable cause. The variation in low-energy apartments was between 59.5 to 88.4 kWh/m²a and 46.6 to 89.7 kWh/m²a in the passive house apartments. It was also suggested that there was a rebound effect, with the high DHW use possibly due to the thought that the water is delivered "free" from the solar thermal collectors.

5.4 Practicality of system solutions

The practical considerations of installation and maintenance in the modular construction of BoKlok impacts the system choice. Decentralised systems such as the base case and compact unit pass best in a modular construction as they require the least work at the site, although there is not enough space for a compact unit in the 2S apartment type. Most of the plumbing and services are contained in one module with small connections to other modules where needed. These connections are difficult to place in the floor making hydronic underfloor heating less practical. Underfloor heating would also require a change to the current floor finish to improve heat transfer. Although possible, it is the most expensive heat emitter option. Of the panel emitters, a fan coil was chosen because it provided the best distribution of heat. Due to the low net energy demand, only one unit is required which must heat the space evenly.

Central energy systems have a single point for maintenance with easy access; however, they require extra space particularly if large TES is used. The current technical room is only 20 m², so any large system would reduce the number of parking spaces in the basement. Certain systems can be favoured by zoning regulations. The case study lies within the concession zone for the district heating network, where it is encouraged to connect to the network. A low temperature district heating network would only be optimal if a low temperature energy source was available nearby, such as waste heat from another building. This is unlikely in this purely residential area. Alternative systems also have practical considerations. The placement of the exhaust air heat pump would require either additional ducting or distribution piping to fit with the current design, adding losses. Furthermore, the use of exhaust ventilation does not meet the heat recovery requirement of the building regulations. The practicality of the GSHP systems is highly dependent on the properties of the ground.

5.5 Energy Flexibility

Energy flexibility was assessed for the potential cost saving which could be achieved by shifting thermal loads. This is necessary for consumer driven demand side management. Significant unit cost savings were only possible for direct electric systems although these were moderated by increased peak costs. Where savings were possible, they were relatively small for each apartment. The potential for savings could be increased through increasing the thermal demand by installing hot-fill dishwashers and washing machines. Alternatively, the control can be used for increased self-consumption of energy produced by onsite solar thermal collectors or photovoltaics

The model assumed a perfectly functioning control which met perfectly predicted demands based on a perfect forecast of the weather. In reality, the control would require a degree of safety to cover unexpected demands or weather changes, further reducing savings. It is then questionable, if the risk of the system malfunctioning is worth the small cost saving.

If grid operators want to encourage demand side management, then further price incentives are required. One solution is to change the structure of peak pricing so that it also can be used as a price signal. For example, the peak pricing could be lower at night to encourage greater charging of the TES in this period. Increasing the variation in unit price would make load shifting relevant for heat pump systems.

6 Conclusion

This thesis investigated energy-efficient HVAC solution-sets for multi-family residential buildings in Nordic climates through a statistical review and simulation of a case study. The case study was an affordable housing project in Sørum, Norway to be built using a modular construction.

Common systems were identified through a statistical and literature review, using the EPC databases of Norway, Sweden and Finland. The three most used heating sources were district heating, ground source heat pumps (GSHP) and direct electric heating. The Norwegian low-energy apartment buildings mostly used bivalent systems, primarily heat pumps with direct electric heating for peak load. The Finish dataset showed that heating was distributed using hydronic radiators or underfloor heating, with 34% using a different emitter in the bathroom. Radiator systems were more often paired with district heating while underfloor heating was more often paired with GSHP. The Swedish datasets showed that most ventilation systems in low-energy buildings are balanced with heat recovery. This was supported by the limited data in the other countries' datasets. Sweden also had some exhaust air systems paired with an exhaust air heat pump, but their use has decreased over time. The system elements inspired by the statistical review, included five heat emitter options, three ventilation options, six schedules and seven energy supply systems. Thermal energy storage was also identified as an important element in the sizing of the energy supply system.

The solutions were divided into parameters affecting the gross energy demand of the building and those affecting the delivered energy demand of the building. The effect of each of these parameters was analysed through simulation, with optimal solutions used in solution-sets. The solution-sets were compared for delivered energy use and energy cost. The potential for load shifting using thermal energy storage was also examined.

Underfloor heating and fan coils were the most efficient heat emitters. It was also possible to provide cooling with these emitters for systems with boreholes. This reduced the required borehole depth and improved the thermal comfort, particularly in the large apartments. Although more efficient, underfloor heating was hard to implement in the modular construction. As only one emitter was required in each apartment, fan coils provided better distribution of heat than the other panel emitters. It would likely be a cheaper option than underfloor heating.

Solutions using a balanced ventilation system were more efficient than those with exhaust ventilation, even when combined with an exhaust air heat pump. Using a hydronic heating coil in the balanced ventilation unit was more efficient, especially in systems with heat pumps. However, it lacked the flexibility of an electric coil. With an electric coil it was possible to turn off the hydronic heating system between 18th May to the 9th September, while maintaining good thermal comfort. The energy saved from reduced distribution losses was greater than the difference in energy use of the two coil types.

Maintaining a constant setpoint or allowing it to setback 2°C during the night affected both the energy demand and peak demand. The constant setpoint required more energy but had a lower peak demand, allowing for a smaller system. The optimal setpoint strategy depended on the heat emitter and energy supply system used. Underfloor heating had the largest difference in peak demand between the two schedules, favouring the constant setpoint. Fan coils had a small difference and so the energy savings of the variable setpoint are more appealing. For systems with a fixed size, such as the two apartment-based systems, the variable system was a better option. This was also the case for the district heating system because increasing the system size has a low cost relative to the annual energy cost. The opposite was true for the heat pump systems especially the ground source systems. The use of a constant schedule not only reduced the size of the heat pump but the borehole system as well. The COP of the heat pump meant that the additional energy cost of the constant schedule was a fraction of the energy difference. If the energy demand of the project was reduced by improving the thermal envelope, the constant setpoint schedule would become a more attractive option for all cases. This was because the difference in energy use decreases more than the difference in peak energy.

All balanced ventilation solution-sets had lower energy costs than the base system. Ground source heat pumps systems had the lowest delivered energy and energy costs. Using solar thermal collectors to provide DHW and recharge the borehole field had multiple benefits. The borehole depth was reduced, offsetting the cost of the solar thermal collectors. The borehole temperature was increased improving the COP of the heat pump. The SCOP of the heat pump was further increased because the heat pump had to provide less DHW at lower COPs. Together this resulted in a further reduction of the delivered energy and energy cost. Whether the benefits offered by the solar thermal collectors outweigh their costs requires further investigation.

District heating systems had the highest delivered energy but not the highest energy cost, as they avoided the large costs related to the peak electricity demand. Low temperature district heating systems required the same quantity of delivered energy but used a higher proportion of electricity due to the heat pump. To be competitive with a traditional district heating system the cost of heat would need to be 5.5% lower. This type of heating network would be more suitable in an area with a mix of building types as it can be used to exchange thermal energy between buildings.

Compact systems offered the best thermal comfort with the possibility of heating and cooling all year round. The lower distribution losses compared to the central systems and heat pump, meant that the energy cost was the lowest after the GSHP and SAGSHP system. The unit would replace the ventilation unit in each apartment, except in the 2S type where the apartment design would need to change to accommodate the unit. However, this heating system would likely have the highest initial cost. This could be reduced by sharing a unit between several apartments, but this creates further issues that need to be resolved.

It was possible to achieve a lower energy cost than the base case using an exhaust ventilation system with an exhaust air heat pump. However, this was only when the energy not covered by the heat pump was provided using district heating. Implementing an exhaust system in the case study would require significant changes to the design and could lead to reduced thermal comfort due to drafts.

An analysis of hourly electricity prices showed that only direct electric systems would give significant cost savings from shifting the thermal load. Variable setpoint schedules had higher savings potential than constant setpoint schedules. A detailed optimisation of the demand profile for each 24 hour period for different heating system and TES capacities, highlighted the importance of the peak load pricing in the flexibility assessment. The peak load cost outweighed the potential savings in hourly electricity cost. Also, large TES capacities around 20 000 L were required to achieve the maximum possible savings. Therefore, optimisations with the smallest possible coil had the best results. The optimisation was performed on the base case, a 100 L electric water tank in each apartment. If the tank temperature was allowed to drop to 55°C, a 5% energy cost saving was possible.

The potential of consumer driven energy flexibility was shown to be limited due to the small incentives to shift demand. The optimisation control may still be relevant for increasing the self-consumption of energy produced by solar thermal collectors or photovoltaics. If grid operators wish to encourage energy flexibility, incentives need to be increased through changes to electricity pricing or credits.

The most energy-efficient solution-set depends on the definition of energy-efficient. A SAGSHP solution with balanced ventilation, fan coils and a variable setpoint schedule had the lowest delivered energy. However, the same system with a constant setpoint schedule had the lowest energy cost due to the reduced peaks. The most energy-efficient solution-set that is actually used in a building is highly dependent on the practically and capital cost. In order to improve the performance of the building stock, finding this practical solution is of great importance. The examined solutions in this thesis will be analysed further to assess their cost-effectiveness.

7 Further research

This thesis was conducted as part of a larger project to find cost-effective energy-efficient solution-sets. The results and tools developed during this work will be used as the basis for the life cycle cost calculations. There is scope for further optimisation of some of the solutions, notably the borehole field design.

This thesis presented an investigation of a single case study through simulation using standardised values. The applicability of the results to new apartment buildings in the Nordic region can be examined further through the following suggested research:

- Using a similar approach on another case study
- Adjusting elements in the thermal envelope
- Using varying loads for each apartment to be closer to reality.
- Assessing the environmental impact of each of the options.

The system study could be developed further to identify the most cost-effective way to achieve net zero-energy buildings through including solar photovoltaics and battery storage. This is particularly relevant for systems utilising solar thermal collectors as photovoltaics compete for roof space. Solar systems are now more relevant in Norway as the price paid for sold electricity has increased from the spot price to 1 NOK per kWh from some suppliers (Ishavskraft, 2021). This on-site system could then be compared to using imported electricity for cost and environmental impact.

The optimisation of the demand profile for energy flexibility through TES could be applied to other building typologies and systems. For example, the charging strategy of undersized borehole fields or increasing the self-consumption of onsite photovoltaics.

Due to limitations of the simulation program or lack of detailed product data, several potential system options were unable to be tested. These included:

- Heat pumps with CO_2 as the refrigerant to produce DHW.
- Ventilation windows in combination with the exhaust ventilation system.
- Domestic hot water tanks immersed in the heating hot water tank
- Arranging hot water tanks for energy cascading
- Satellite water heaters. Each apartment would have an independent DHW tank which would use the heating loop and an auxiliary heating element such as an air source heat pump or direct electric element. The use of one hot water circulation system would reduce heat losses.

- Domestic hot water provided instantaneously through a heat exchanger. This could allow for DHW and heating to be distributed using one high temperature central loop. The use of one hot water circulation system would reduce heat losses.

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Appendices

A. Optimal tilt for solar thermal collector.

Initial simulations using the default values in *SIMIEN7* for 1 m² effective area are shown in Table A.1. The output between March to September was assumed the most important.

Table A.1. Energy output by month of solar thermal collector at tilts between 35° to 41°.

Tilt	**35°**	**36°**	**37°**	**38°**	**39°**	**40°**	**41°**
Output / kWh							
January	0.00	0.00	0.00	0.00	0.00	0.00	0.00
February	2.79	2.95	3.10	3.26	3.43	3.58	3.71
March	20.38	20.92	21.34	21.76	22.08	22.52	22.92
April	39.28	39.53	39.76	40.05	40.18	40.27	40.38
May	61.59	61.52	61.38	61.25	60.98	60.79	60.56
June	72.59	72.23	71.96	71.52	71.10	70.69	70.15
July	78.22	77.97	77.71	77.44	77.11	76.84	76.36
August	43.02	43.04	43.03	43.04	43.00	42.96	42.92
September	25.49	25.75	25.99	26.16	26.41	26.62	26.78
October	10.10	10.42	10.78	11.10	11.40	11.69	12.02
November	0.00	0.00	0.02	0.02	0.03	0.03	0.05
December	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Sum Mar - Sep	340.57	340.96	341.17	341.22	340.86	340.69	340.07

B. Properties of water at atmospheric pressure at sea level

Table B.1. Properties of water at sea level.

Temperature	Density (ρ_w)	Specific heat ($C_{p,w}$)	Volume heat capacity	
°C	kg/m³	kj/(kg·K)	kj/m³	kWh/m³
35.00	994.08	4.178	4 153.51	1.153 75
40.00	992.25	4.179	4 146.28	1.151 74
45.00	990.22	4.180	4 138.75	1.149 65
50.00	988.02	4.181	4 130.87	1.147 46
55.00	985.65	4.183	4 122.63	1.145 18
60.00	983.13	4.185	4 114.05	1.142 79
65.00	980.45	4.187	4 105.17	1.140 33
70.00	977.63	4.190	4 096.03	1.137 79
75.00	974.68	4.193	4 086.69	1.135 19
80.00	971.60	4.196	4 077.20	1.132 56
85.00	968.39	4.200	4 067.62	1.129 89
90.00	965.06	4.205	4 058.00	1.127 22
95.00	961.62	4.210	4 048.39	1.124 55

Using these values to plot the storage capacity against tank temperature for a 1m³ tank, it is possible to assume a linear relationship between kWh and temperature, as shown in Figure B.1.

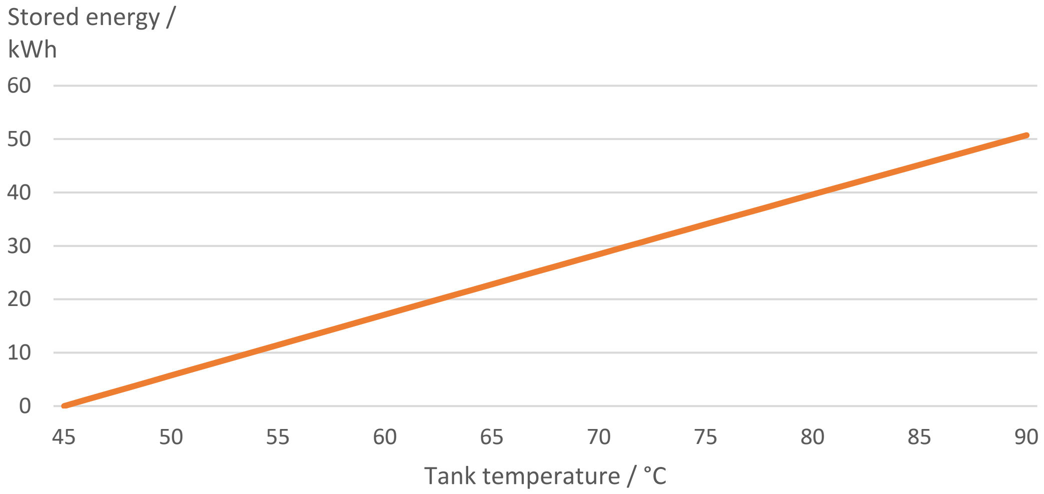

Figure B.1. Energy stored per degree Celsius for a 1 m³ tank supply 45 °C water.

C. Calculation of heat loss per additional kWh

The additional heat loss of storing an additional kWh of thermal energy was estimated based on a 1 000 L tank. Based on commercially available tanks this was modelled as a cylinder 2.2 m high and 1 m in diameter. It was assumed to have 100 mm of insulation with a thermal conductivity of 0.037 W/(m·K). The internal dimensions were thus 2 m high and 0.8 m in diameter. The resulting surface area is 6.03 m². An additional 0.2 W/K loss was assumed for each of the 4 connection points. The specific heat loss is therefore 3.03 W/K. The heat loss per additional kWh was therefore calculated as:

$$Heat\ loss\ per\ kWh = \frac{H_s \cdot (T_s + 1 - T_a) - H_s \cdot (T_s - T_a)}{C_{p,w} \cdot \rho_w / 3.6} = \frac{H_s}{C_{p,w} \cdot \rho_w / 3.6} \quad [W]$$

The heat capacity ($C_{p,w}$) and density (ρ_w) of water do vary slightly with temperature as shown in Appendix B. The range of heat loss within the working range of 35 °C to 95 °C was 0.263 W to 0.270 W, shown in Figure C.1. As it was probable that the tank temperatures would be primarily in the lower half of this range, the value of 2.65 W (equivalent to 57 °C) was deemed a reasonable assumption.

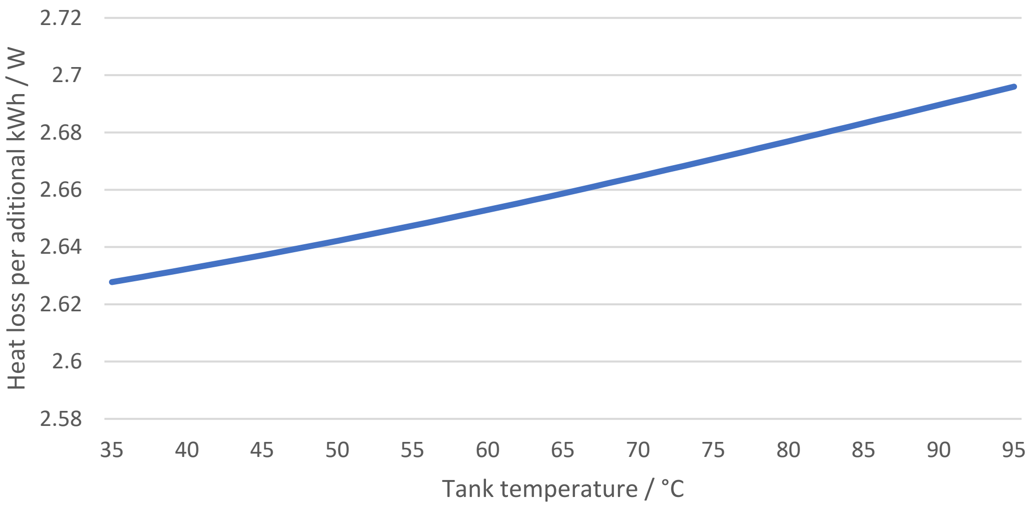

Figure C.1. Heat loss per additional kWh of store energy as a function of tank temperature.

D. Thermal comfort of all apartments

The hours which exceed the thermal comfort are shown for all 68 apartments in Figures , D.2 and D.3. They are for a system using a low temperature radiator for just the heating season with a variable setpoint schedule. This arrangement gives the highest number of hours outside the thermal comfort range of 19 °C to 26 °C.

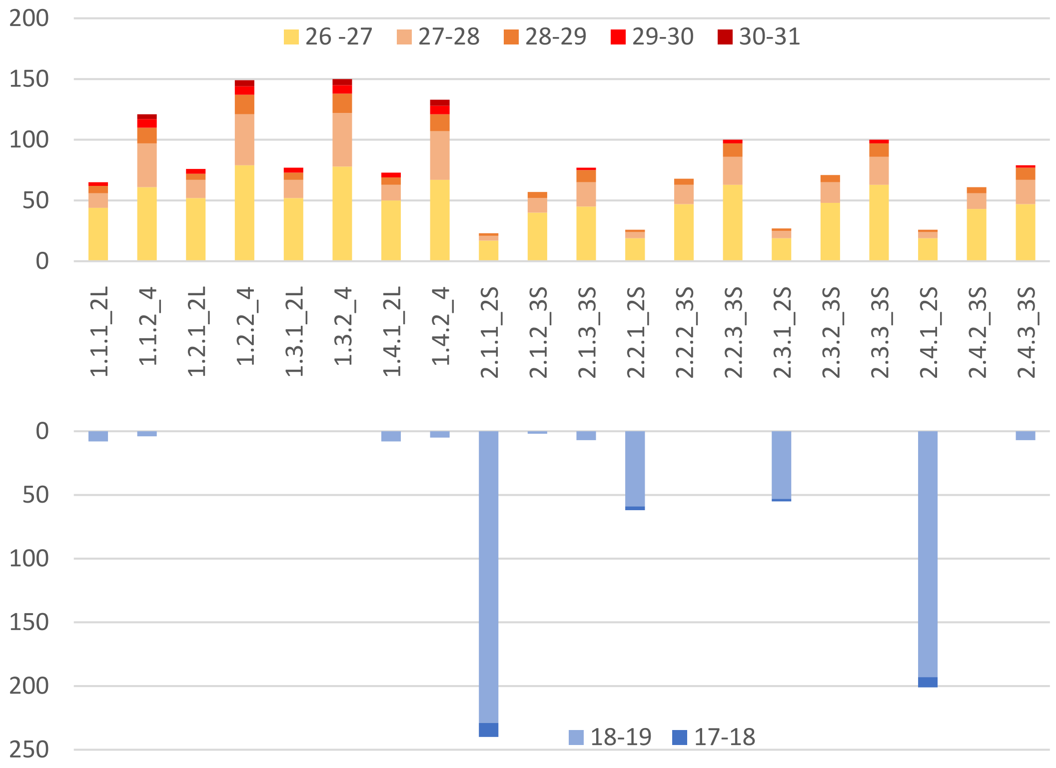

Figure D.1. Hours above and below the thermal comfort limits in building 1 and 2 for the solution with balance ventilation, low temperature radiator and variable setpoint schedule for just the heating season.

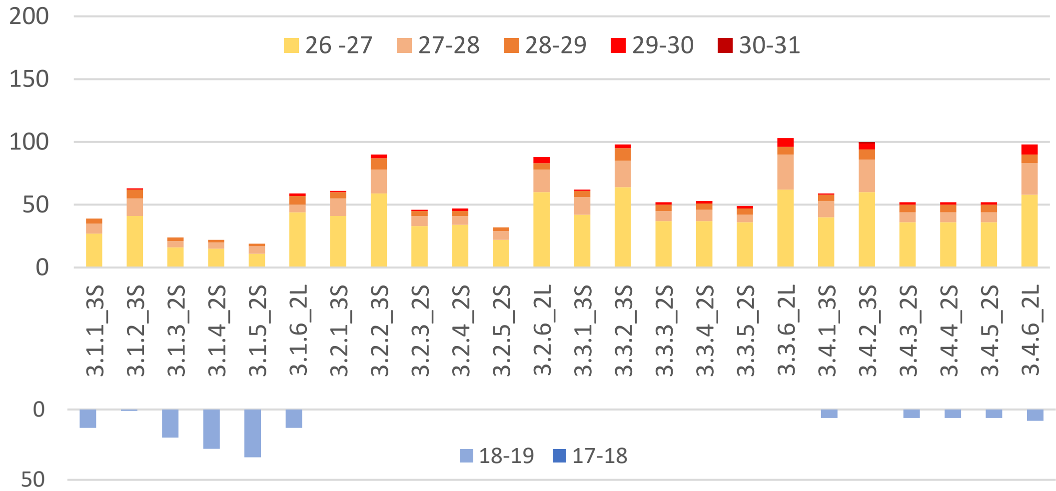

Figure D.2. Hours above and below the thermal comfort limits in building 3 for the solution with balance ventilation, low temperature radiator and variable setpoint schedule for just the heating season.

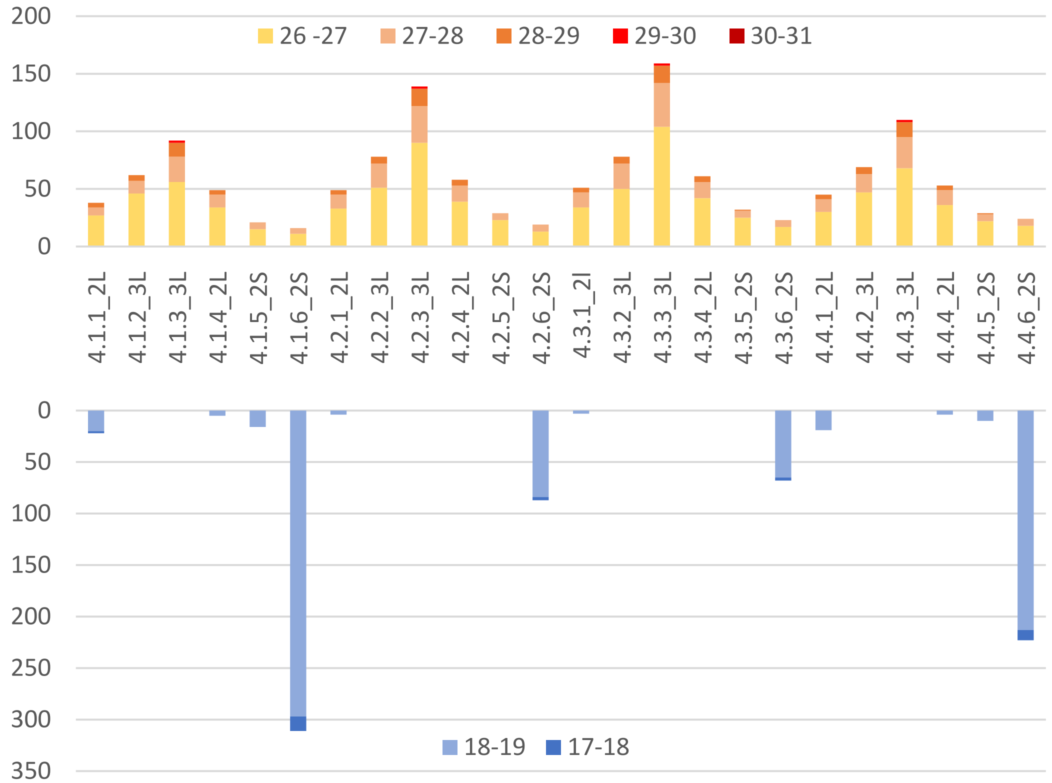

Figure D.3. Hours above and below the thermal comfort limits in building 4 for the solution with balance ventilation, low temperature radiator and variable setpoint schedule for just the heating season

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