Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP)

IEA OECD

Annex 40

Heat Pump Concepts for Nearly Zero-Energy Buildings

Task 2: Case studies on building system technologies for nZEB

Final Report

Operating Agent: Switzerland

2016

iea Energy Technology Network

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Preface

This project was carried out within the Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP) which is an Implementing agreement within the International Energy Agency, IEA.

The IEA

The IEA was established in 1974 within the framework of the Organization for Economic Cooperation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster cooperation among the IEA participating countries to increase energy security through energy conservation, development of alternative energy sources, new energy technology and research and development (R&D). This is achieved, in part, through a programme of energy technology and R&D collaboration, currently within the framework of over 40 Implementing Agreements.

The Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP)

The Technology Collaboration Programme on Heat Pumping Technologies (HPT TCP) forms the legal basis for the Heat Pumping Technologies Programme. Signatories of the TCP are either governments or organizations designated by their respective governments to conduct programmes in the field of energy conservation.

Under the TCP collaborative tasks or “Annexes” in the field of heat pumps are undertaken. These tasks are conducted on a cost-sharing and/or task-sharing basis by the participating countries. An Annex is in general coordinated by one country which acts as the Operating Agent (manager). Annexes have specific topics and work plans and operate for a specified period, usually several years. The objectives vary from information exchange to the development and implementation of technology. This report presents the results of one Annex. The Programme is governed by an Executive Committee, which monitors existing projects and identifies new areas where collaborative effort may be beneficial.

The IEA Heat Pump Centre

A central role within the HPT TCP is played by the Heat Pump Centre (HPC). Consistent with the overall objective of the HPT TCP the HPC seeks to advance and disseminate knowledge about heat pumps, and promote their use wherever appropriate. Activities of the HPC include the production of a quarterly newsletter and the webpage, the organization of workshops, an inquiry service and a promotion programme. The HPC also publishes selected results from other Annexes, and this publication is one result of this activity.

For further information about the IEA Heat Pumping Technologies Programme and for inquiries on heat pump issues in general contact the Heat Pump Centre at the following address:

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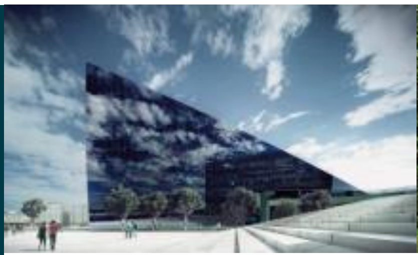
IEA HPT Annex 40

Heat pump concepts for Nearly Zero Energy Buildings

Final Report IEA HPT Annex 40 Task 2



Heat pump concepts for nearly Zero Energy Buildings

Case studies on building system technologies for nZEB

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Abstract

Since the mid of the 1990ties low energy buildings with a significantly reduced energy consumption down to ultra-low energy standard (typical space heating energy need of 15 kWh/(m^2a)) or even net zero energy consumption (on an annual basis by an integration of on-site renewable energy systems) have been realised. Based on the political strategies for the building sector in terms of meeting the climate protection targets, the building concepts are currently extended to derive a nearly zero energy balance, which requires on the one hand an energy-efficient building envelope and on the other hand energy-efficient building system technologies amended by an on-site renewable energy production.

IEA HPT Annex 40 is to investigate heat pumps for the application in nearly zero energy buildings. Due to the unique features of the heat pump, the application in nearly zero energy buildings can be particularly beneficial. Besides the high performance of the heat pump in combination with adapted systems of low supply temperatures, which can be installed in buildings with high performance building envelopes due to the low space heating loads, also the integration options of heat pumps with other building technologies can be an advantage of the heat pump application in these buildings.

Task 2 is dedicated to identify promising concepts for the further development of system configurations and analysis of technologies suitable for different applications in nZEB. Concepts shall be compared based on energy performance and cost to approve the feasibility of heat pump solutions for the application in nZEB.

For this evaluation case studies for the assessment of different system solutions for the energy supply of nZEB have been accomplished. Thereby, the following items are addressed:

- The role of heat pumps in nZEB under the boundary conditions of different countries
- Favourable technology combinations to reach the nZE balance cost-effectively
- Comparison of performance and cost for different system configurations in residential and office buildings
- Assessment and ranking of system solutions for different applications and boundary conditions
- Analysis of integration options of heat pumps with other building system technology, e.g. solar assisted heat pumps
- Development of design tools for the generator and emission system

This report on Task 2 covers the case studies performed under IEA HPT Annex 40. The results presented in this report summarise the:

- Case study in Canada for heat pump application in residential single- and multi-family as well as in office buildings
- Case studies in Finland and Sweden for residential single- and multi-family buildings
- Case study to reach nZEB in office buildings in Japan with pronounced air-conditioning loads
- Development of a design tool for heat pump and back-up system for optimised cost and minimised CO_2-emission at SINTEF Energy in Norway
- Development of a design tool for the comfort evaluation at the Center of Environmental Energy Engineering (CEEE) at the University of Maryland
- Case study in Switzerland of technology comparison for MINERGIE-A® residential and office buildings

Contents

1 Overview of case studies in Annex 40

In Task 2 of Annex 40 several case studies for the evaluation of heat pump application in nZEB have been performed. Tab. 1 gives an overview different boundary conditions used in the case studies.

The cases study cover different heat pump types as well as different boundary conditions. Case studies have been performed both for central European and Nordic climate conditions as well as in Japan, where pronounced air-conditioning needs occur. As building types both residential as single family and multi-family houses and office buildings have been considered. Thereby, the boundary conditions regarding the nZEB definition and economic aspects have been based on the current national state of definition and the present market conditions. Despite the partly different boundary conditions the case studies yield the results, that both in central European as in Nordic countries heat pumps are a favourable building system technology for the application in nearly Zero Energy Buildings both in terms of energy-efficiency and life-cycle cost. With the heat pump solutions, requirements of nZEB can be reached. Even though heat pumps may have higher investment cost compared to other heating systems on the national markets of the individual countries, heat pumps are among the systems with the lowest life-cycle cost.

Despite the differences in the nZEB balance definition, climate, and the economic boundary conditions, the resulting ranking of the different system solutions is quite similar, where heat pumps are among the most appropriate system solutions. This shows a certain robustness of the results regarding both the energy performance evaluation and the economic boundary conditions. For the energy evaluation heat pump solutions benefit from the high energy performance in nearly Zero Energy Buildings with good building envelopes enabling low supply temperatures. Thereby, the nZEB balance can be reached more cost-effectively, since less on-site generation is required for the compensation of the energy demand of the building. On the other hand, heat pumps may have higher investment cost on the national markets, but regarding the life-cycle cost, this initial disadvantage is compensated by less investment in generation technologies, e.g. PV systems can be designed smaller. Moreover, the higher energy performance of heat pumps reduces the operational cost, which is seen in the life-cycle consideration, too. According to these results, heat pumps are very favourable system technology for the application in future nZEB also for Nordic climate conditions.

Furthermore, two design tools have been developed in Task 2. In the USA, at the Center of Environmental Energy Engineering (CEEE) of the University of Maryland, a tool to evaluate the thermal comfort of surface heating and cooling systems for rooms has been developed, which is denoted as ThermCom software tool.

At SINTEF Energy in cooperation with the NTNU in Trondheim, Norway, the development of a design tool for heat pumps and back-up heating regarding cost and CO_2-eq.-emissions have been accomplished.

Tab. 1: Overview on case studies performed in Task 2 of Annex 40

Country	Building types			Building standard				HP Types				Other generators						Cost evaluation		Remarks
	SFH	MFH	OF	ULEH	LEH	SB	Retro	AW	GW	AA	EA	FB	DH	CHP	BB	ST	DE	Yes	No	
CA	X	X	X		X	X	X	X	X	X		X		X			X	X		• Different sites across Canada • Additional cold climate ASHP • Different energy prices depending on the site
CA	X				X												X	X		• Additional case study solar-assisted HP with ice-storage in Task 3
CH	X	X	X	X	X	X		X	X			X	X	X	X	X		X		• Standard weather of Zurich average year • Small and medium-sized office building
FI	X	X		X	X	X	X	X	X	X	X		X			X		X		• Case study for Finnish residential buildings taking into account FinZEB evaluation
JP			X	X	X	X			X	X									X	• Medium size and large office buildings • Including Air-conditioning and appliances • Different load situations and climates
NO	X				X			X								X			X	• Case studies on different internal refrigeration cycle of CO_2 heat pump
SE	X	X						X	X				X		X	X		X		• For MFH only HP and district heating
US			X		X			X		X									X	• Evaluation of comfort for different HP or AC systems, respectively: radiative panel heating/cooling, induction unit, ducted air conditioner, ductless room air conditioner

Legend: SFH – Single family house, MFH – multi-family house, OF – office, ULEH – ultra-low energy house, LEH – low energy house, SB – standard building, Retro – retrofitted building, AW – air-to-water heat pump, GW- ground-to-water heat pump, EA – exhaust air heat pump, AA – air-to-air heat pump, FB – fuel boiler, DH – district heating, CHP – combined heat and power, BB – biomass boiler, ST – solar thermal

2 Case studies of nZEB technologies in Canada

Task 2 of Canada comprises the analysis and optimisations of the most promising concepts out of the Canadian state-of-the-art analysis which can be successfully implemented in the Canadian nearly Zero Energy Buildings of the future. A techno-economic analysis of the integration of HPs and renewable energy systems to houses of three construction vintages are presented, including:

- A typical existing home (1980s)
- A typical newly constructed home
- A highly efficient low energy home

Some analyses of the integration of heat pumps with renewable energy systems are presented, as well. This approach will also be applied to commercial and multi-unit residential buildings. To date, archetype models of a large office and a multi-unit residential buildings have been developed to examine the implications of using heat pump systems on a larger scale and identify potential savings from scale economies.

2.1 Building energy models

The building energy models include high performance single-family homes, multi-unit residential buildings (MURB) and large office buildings.

2.1.1 High performance Multi-Unit Residential Building (MURB)

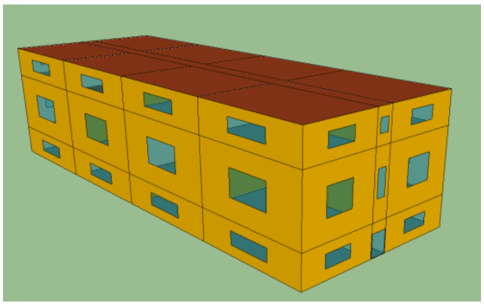


Abb. 1: Design of MURB

The shape and dimensions of the mid-rise apartment were taken from the Department of Energy (DOE) benchmark models (Torcellini et al., 2008). The total heated floor area of the building is 3,135 m² with a footprint area of 784 m². The building has a 15% fenestration to wall ratio. The apartment suites are maintained between 21 °C and 23 °C and relative humidity above 30%. Lighting, occupancy and receptacle schedules were assumed to follow those outlined in the National Energy Code of Canada for buildings (NECB) (NRC, 2010). Each apartment unit has its own 110 l domestic hot water (DHW) heater, with a hot water tapping profile assumed to follow that of the IEA Annex 42 DHW Load profile (Knight et al., 2007) for Canada. It is also estimated that the roof area can have a 172 m² PV array angled south at 45° such that there is no shading of the panels on one another.

2.1.2 High performance large office building

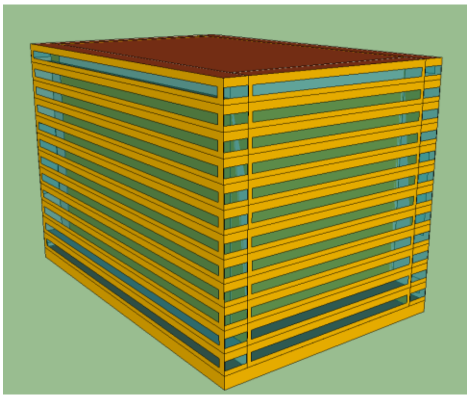


Abb. 2: Design of Large Office Building

For the large commercial office archetype, the physical geometry of the building was based on information from previous NRCan projects (Pope, 2013) and is representative of typical newly built offices in major Canadian centres. The office building has 10 floors with a total floor area of 13,396 m². The floor-floor height is 3.96 m and the glazing percentage is 40%.
In order to represent new constructions, building envelope and mechanical system performance were based on information provided in the NECB (2011). All building occupancy and operational schedules were also based on information in the NECB (2011).

2.1.3 High performance single family home

Housing models were developed for three distinct regions in Canada (Montreal, Toronto, and Vancouver) in order to assess the impact that various climates had on the performance of the system. The envelope of each home was based on the Canadian Center for Housing Technology test home in Ottawa, Ontario (Swinton et al., 2003).

For Montreal and Toronto boundary conditions, a house with an U-value of around 0.3 W/(m²K) and an infiltration of 0.75 ACH_{50}, 0.60 ACH_{50}, respectively, was taken. For Vancouver boundary conditions, a house with an U-value around 0.35 W/(m²K) and an infiltration of 1.0 ACH_{50} was taken. It has to be mentioned that the U-values in this form are not weighted with the areas.

2.2 System energy models

A variety of system models have been examined in each building.

2.2.1 Base cases

<u>High performance single family home</u>

The base mechanical system in the Montreal high performance home used electrical heating and DHW preparation in order to easily facilitate the future integration of on-site renewable energy generation (e.g. PV). Tab. 2 provides details of the system.

Tab. 2: Base mechanical system characteristics of high performance single family home

Region	Heating system	Cooling system	Ventilation	DHW
All	Electric Baseboard	Split System Rated COP = 3.45	HRV, 0.84 effectiveness	Electric Conventional Tank

In Toronto and Vancouver, the electric baseboards were replaced with a low capacity two stage natural gas furnace to make use of the lower gas rates available. Cooling and ventilation systems remained the same as above, while DHW was met using a natural gas boiler.

<u>Multi-Unit Residential Building (MURB)</u>

The base system for the MURB was derived from DOE benchmark energy models (Torcellini et al., 2008), and consists of a packaged terminal air conditioning (PTAC) unit with a hot water heating coil located in each apartment suite. This type of system gives each apartment suite owner the ability to choose between heating and cooling, which cannot be accomplished in two pipe fan coil systems where the heating and cooling season is set by the building operator. Key details are provided in Tab. 3.

Tab. 3: Mechanical system performance in MURB

Heating	Cooling	DHW
Central plant, Boiler, 83% efficient OA Reset: 82 °C WT at -16 °C OAT 60 °C WT at 0 °C	1 PTAC/suite, COP_{Rated}=2.94	Conventional gas boiler

The primary boiler pump are of continuous, constant flow type, while the secondary pump has variable flow and is considered to ride its pump curve as described in the NECB (2011) and NRC (2010). Cooling is accomplished by the apartment suite PTAC units turning on, when there is a demand for cooling.

<u>Large office building</u>

The base case system for the office building consisted of a central boiler and chiller, along with a variable air volume (VAV) air distribution system and hydronic baseboards in each zone. Key performance criteria are provided below in Tab. 4.

Tab. 4: Mechanical system performance of the base case system in large office buildings

Heating central Plant	Cooling central plant	Ventilation	DHW
Boiler, 83 % efficient	Water cooled electric chiller COP_{rated} = 5.17	VAV system, 1 AHU per floor	Conventional gas boiler

The primary boiler pump operates at continuous flow, while the secondary pump experiences variable flow and adjusts its energy use based on NECB curves (NRC, 2010).

2.2.2 Used heat pump systems

If not specially noted, the system has been examined in all three building types.

Tab. 5: Heat pump systems for system energy models

Conventional electrically-driven heat pump systems	Air-Source Heat Pump + Auxiliary
	Cold Climate Air-Source Heat Pump
	Ground-source Heat Pump
	Water-Loop Heat Pump (only in MURB)
Other heat pumps	Gas-fired Absorption Heat Pump (only in large office)
	Solar Assisted Ground-Source Heat Pump (only in MURB)
	Cogeneration with Ground-Source Heat Pump (only in MURB)

In Fig. 1 the system configuration of the two heat pump configurations of a solar assisted heat pump (left) and a water loop heat pump (right) are depicted.

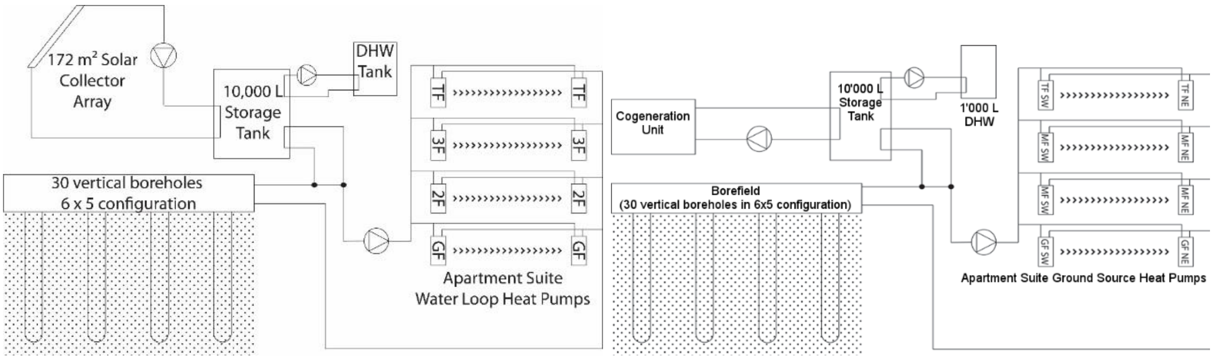


Fig. 1: Heat pump system configurations of solar-assisted heat pump and cogeneration with ground-source heat pump investigated in MURB

2.3 Techno-Economic Analysis

2.3.1 High Performance Home

Using the building and system models described above, a techno-economic analysis of several heat pump systems was performed for Montreal and Vancouver.

Tab. 6: Techno-Economic Results for High Performance Home in Montreal

Montréal	Base case: Electric baseboard	Standard ASHP with back-up electric heating	Cold climate ASHP	Closed loop vertical GSHP
Annual energy consumption (kWh)	19,441	16,897	15,100	13,877
Capital cost	$8,858	$10,803	$15,218	$14,017
Annual utility costs	$1,382	$1,193	$1,056	$964
Simple payback period (Years)	-	10.33	19.56	12.37

Utility rates were obtained based on 2011 pricing structures, while capital costs were derived from surveys of local heating ventilation and air-conditioning (HVAC) contractors. Full details of the study can be found in Kegel et al., 2012.
Tab. 6 summarises the results in Montreal with cold climate air-source heat pump (CC-ASHP) and Tab. 7 in Vancouver with high efficiency air-source heat pump (ASHP).

Tab. 7: Techno-Economic Results for High Performance Home in Vancouver

Vancouver	Base case: Electric baseboard	Standard ASHP with back-up electric heating	High efficiency ASHP	Closed loop vertical GSHP
Annual energy consumption (kWh)	19,496	15,107	14,606	14,261
Capital cost	$11,282	$11,780	$13,380	$14,148
Annual utility costs	$1,121	$1,074	$1,040	$1,006
Simple payback period (Years)	-	10.57	25.80	24.92

2.3.2 Pathways to "Net Zero Ready"

In addition to the standard techno-economic analysis provided above, different pathways for retrofitting a typical 1980s home to a "net-zero ready" (NZR) level have been examined from an energy and economic view in Montreal and Vancouver:

- Mechanical: installation of a high performance heat pump system
- Envelope: large scale building envelope modifications
- Mechanical + Envelope: conventional heat pump technology combined with smaller scale building envelope modifications

The required characteristics to reach NZR for the second (envelope) and third (mechanical + envelope) pathway are listed in Tab. 8.

Tab. 8: Characteristics for NZR with second and third pathway

Pathway	Envelope		Mechanical + Envelope	
Location	Montreal	Vancouver	Montreal	Vancouver
Wall R-Value	5.41 m²K/W	4.44 m²K/W	4.85 m²K/W	2.61 m²K/W
Roof R-Value	8.93 m²K/W	8.93 m²K/W	8.93 m²K/W	5.71 m²K/W
Basement Wall R-Value	4.93 m²K/W	4.93 m²K/W	4.93 m²K/W	1.21 m²K/W
Window U-value (incl. framing)	1.35 W/m²K	1.35 W/m²K		
Air Infiltration	0.75 ACH_{50}	1.0 ACH_{50}	1.5 ACH_{50}	1.5 ACH_{50}

Each system and pathway was modelled and simulated using the TRNSYS energy simulation program. For each pathway, it was also assumed that existing appliances were replaced with new ones, with total appliance energy set equal to the EnerGuide reference (6 kWh/d). An additional case is also presented in which the existing heating and cooling systems are replaced with new equipment operating with the same efficiency. It should be stressed that these modifications do not meet the NZR target, and are supplied to show the cost impact of maintaining current performance.
Tab. 9 summarises the total life-cycle cost for each city and pathway. The high performance heat pump option was clearly preferred for both climate regions, even when examining the case where the status quo for equipment performance was maintained.

Tab. 9: Summary of life-cycle cost

	Montreal	Vancouver
Maintain existing performance levels	$45,323	$28,225
High performance HP: GSHP	**$38,197**	$36,966
High performance HP: CCHP/ASHP	$31,470	**$27,801**
High performance building envelope	$70,931	$74,316
Combination ASHP/Envelope	$64,861	$48,993

2.3.3 High performance office building

For the office building, a selection of conventional heat pump systems has been examined in the Toronto region.

Tab. 10 summarises the energy performance of each heat pump system. It is evident that integrating electrically-driven heat pumps into the building results in a substantial decrease in total energy use, with the heat pump systems achieving total reductions between 45% and 53%. The lowest total energy use is achieved with the GSHP, primarily due to the higher COP of the water-source heat pump unit, and the stable source temperature provided to the heat pump from the ground heat exchanger.

Tab. 10: System Energy Performance (ASHP – air-source heat pump, CC ASHP – cold climate air source heat pump, GSHP – ground-source heat pump, GAHP – gas absorption heat pump)

	Case				
	Base Case	**ASHP+Boiler**	**CC ASHP**	**GSHP**	**GAHP**
Heating [MWh]	1,137	271	150	58	717
Cooling [MWh]	104	77	140	104	104
Fans and pumps [MWh]	256	48	47	67	256
DHW [MWh]	144	144	144	144	144
Lighting & appliances [MWh]	751	751	751	751	751
Annual natural gas use [MWh]	1,281	382	144	144	861
Annual electricity use [MWh]	1,111	909	1,089	980	1,111
Total [MWh]	2,392	1,292	1,233	1,124	1,972
Normalized energy use [kWh/m²]	179	96	92	84	147

Tab. 11 summarises the life-cycle economic performance of each system. All component costs used in the analysis included both material and labour costs. All material heat pump costs were obtained from manufacturer supplied pricing lists of Mitsubishi and Robur (2010), while labour costs were estimated from RS Means (2013). Ground heat exchanger costs were estimated to be $80/m (Kummert and Bernier, 2008) and assumed to include all necessary ground loop piping and connections. All remaining material and labour costs were obtained from RS Means.

Tab. 11: Techno-economic analysis

	Case				
	Base Case	**ASHP+Boiler**	**CC ASHP**	**GSHP**	**GAHP**
Capital Cost [k$]	$1,947	$2,368	$2,036	$2,276	$2,575
Life-cycle Utility Cost [k$]	$2,740	$2,052	$2,341	$2,122	$2,605
Life-cycle Maintenance Cost [k$]	$688	$675	$221	$221	$688
Life-cycle Cost [k$]	$5,376	$5,095	$4,599	$4,619	$5,868

From both, an energy and economic perspective, it is clear that heat pumps can play a significant role in meeting the heating and cooling loads of a typical office building.

2.3.4 High Performance MURB

A techno-economic analysis of several conventional and more innovative HP and cogeneration systems has been performed for both, the Montreal and Calgary regions. For the cogeneration unit, an approximate cost of $1,950 per kW installed electric capacity and $80/m borehole cost were estimated. Calgary and Montreal have similar cost indexes (RSMeans, 2013).

In both cities, the GSHP is the most energy-efficient system achieving 38% and 42% seconddary energy savings in Calgary and Montreal, respectively. The similarities of the results for both cities stop at the secondary energy efficiency. In Calgary, due to the electricity fuel source and utility costs, the cogeneration system is the most suitable system achieving 22% primary energy savings, 20% utility cost savings and 31% greenhouse gases (GHG) emission reductions.

In Montreal, the GSHP system is the most suitable achieving 43% primary energy savings, 18% utility cost savings and 76% GHG emission reductions in comparison to the base case. In Calgary, the only system that was able to achieve savings in all four categories (primary energy, secondary energy, utility costs and GHG emissions) was the cogeneration + GSHP system.

The 20-year life-cycle for each system is summarised in Fig. 2 for Calgary and Montreal.

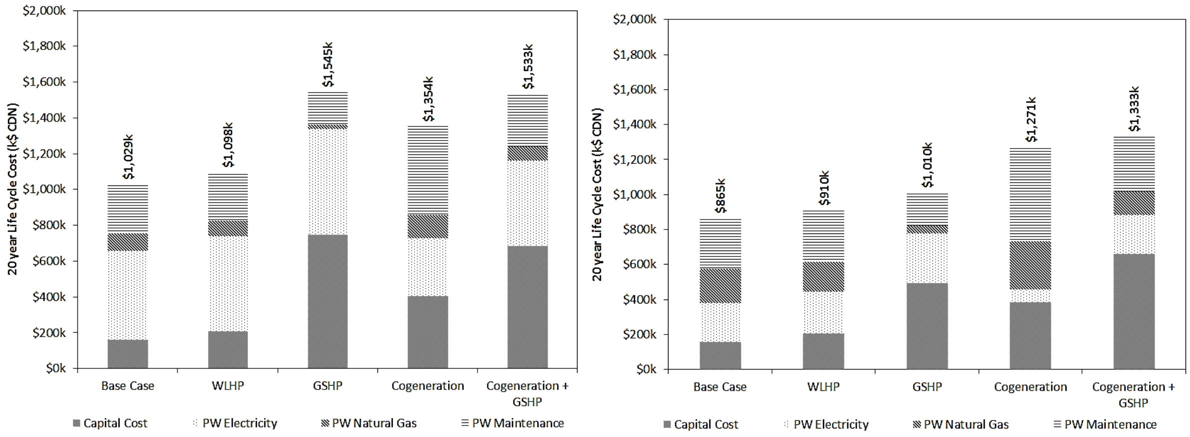


Fig. 2: Estimated 20-year life-cycle cost for systems in Calgary (left) and Montreal (right)

The results highlight that the base case system offers the lowest 20-year life-cycle cost of all systems evaluated in this study. This is not surprising as the systems evaluated addressed meeting the space heating and cooling loads more efficiently, which were already reduced by an improved building envelope. The 20-year life-cycle cost of the water-loop heat pump (WLHP) systems for both regions were approximately 5% higher than the base case system, attributed primarily due to the slightly higher capital cost of the system as the annual utility cost savings were minimal in both cases.

2.3.5 Pathways to Net Zero for a MURB

The MURB energy model has also formed the basis for a detailed examination of pathways to net zero energy in the Montreal region. The objective was to determine at what point it began becoming cost-effective to integrate on-site PV panels instead of implementing energy efficiency measures.

Detailed results of the case study on pathway to MURB are contained in Appendix A1.

<u>Standard Efficiency Measures</u>

To quantify the benefit to reducing the annual energy consumption of the mid-rise apartment compared to simply adding PV to make it net zero, several energy efficiency measures are investigated. These measures include:

- Reduction in base electrical loads through energy-efficient lighting and appliances
- Reduction of fan power using electronically commutated motors (ECM)
- Reduction in the space heating load by a highly efficient building envelope
- Reduction in the space heating load by the heat recovery from exhaust air

<u>Energy Efficient Systems</u>

The second level of energy efficiency measures was the integration of energy-efficient systems. The following systems have been included for this analysis:

- Conventional Air-Source Heat Pump (ASHP)
- Cold Climate Air-Source Heat Pump (CC-ASHP)
- Ground-Source Heat Pump (GSHP)
- Solar Assisted Ground-Source Heat Pump (SA-GSHP)

Analysis and Results

Energy models of the following energy efficiency measure combinations were developed to evaluate the energy saving potential versus cost over a PV system to achieve net zero energy.

Measure 1: Reduction of base electrical loads
Measure 2: Reduction of base electrical loads and use of ECM fan motors
Measure 3: Reduction of base electrical loads, use of ECM fan motors and building insulation improvement
Measure 4: Reduction of base electrical loads, use of ECM fan motors and heat recovery on the apartment unit exhaust air
Measure 5: Reduction of base electrical loads, use of ECM fan motors, heat recovery on exhaust air and building insulation improvement

From RS Means (2013) and contractor surveys the estimated energy efficiency measure cost was estimated and is summarised in Tab. 12.

Tab. 12: Estimated incremental costs for standard efficiency measure combinations

Energy efficiency measure	Estimated cost
Measure 1	$25,000 CDN
Measure 2	$32,000 CDN
Measure 3	$177,000 CDN
Measure 4	$115,000 CDN
Measure 5	$260,000 CDN

To evaluate how the energy-efficient systems compare in achieving energy savings with a PV system, each heat pump system was evaluated in the base case and the five standard energy efficiency measure combinations identified.
Tab. 13 summarises the estimated incremental costs for each heat pump system compared to the defined base case heating and cooling system (packaged terminal air conditioner with electric back-up heating).

Tab. 13: Estimated incremental costs for proposed heat pump systems

Heat pump system	Estimated incremental cost
1.5 ton ASHP	$102,500 CDN
CC ASHP	$166,250 CDN
GSHP	$396,500 CDN
Solar assisted GSHP	$414,250 CDN

For the GSHP and the solar assisted GSHP systems, the borehole fields were resized for energy efficiency measures 3, 4 and 5 as the building heating and cooling loads changed. The borehole field size and estimated incremental cost in comparison to the base case with the same energy efficiency measure is summarised in Tab. 14.

Tab. 14: Estimated borehole field size and incremental cost for the GSHP heat pump systems

Heat pump system	Borehole field	Estimated incremental cost
Measure 3 GSHP	5 x 5, 95 m	$343,250 CDN
Measure 4 GSHP	5 x 4, 93 m	$300,500 CDN
Measure 5 GSHP	4 x 4, 85 m	$259,000 CDN
Measure 3 Solar Assisted GSHP	4 x 4, 98 m	$382,250 CDN
Measure 4 Solar Assisted GSHP	4 x 4, 76 m	$352,000 CDN
Measure 5 Solar Assisted GSHP	4 x 3, 80 m	$351,225 CDN

Comparing the heat pump systems analysed, it is seen that the conventional ASHP system and the solar assisted (SA) GSHP system are the most favourable selections as both systems had similar cost differences between the efficiency measure and the avoided PV array cost.

The conventional ASHP system was found to be the most suitable for efficiency measure 1 through 3, with the SA GSHP being the most suitable selection for efficiency measures 4 and 5. Without the PV array, the conventional ASHP system has the lowest simple payback period for all efficiency measures ranging between 10 and 22 years. With the PV array to achieve Net Zero Energy, the simple payback period significantly increases due to the current low utility rates in Montreal. Here, the conventional ASHP has the lowest simple payback period for all efficiency measures with the exception of 4 and 5, where the solar assisted GSHP system is better. Simple payback periods range between 23 and 30 years for all systems.

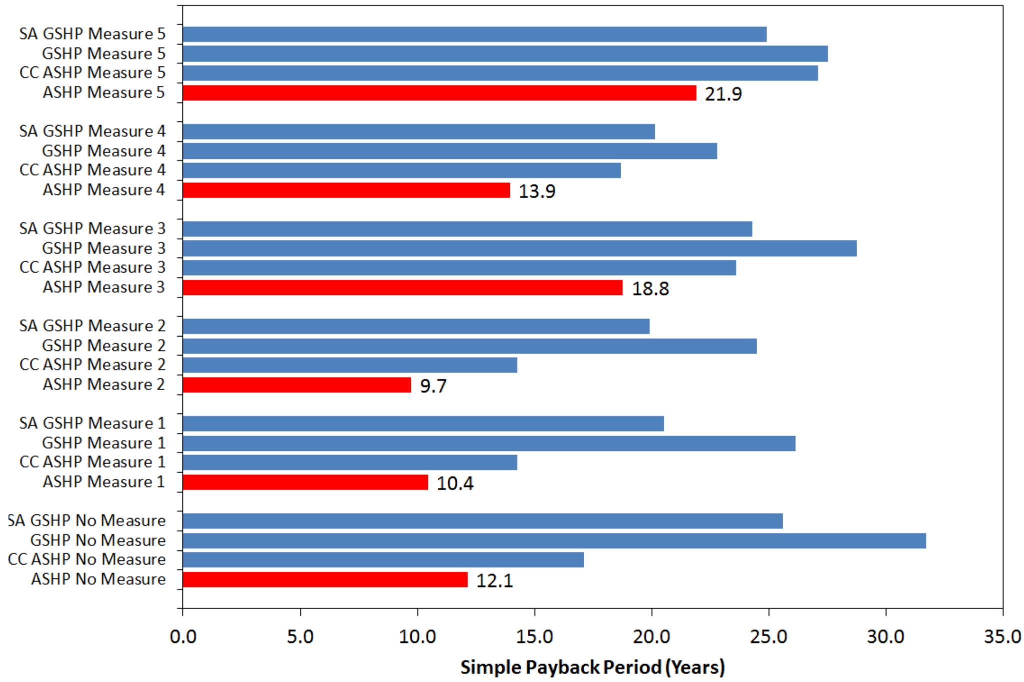


Fig. 3: Simple payback period for the heat pumps systems without a PV Array in comparison with the base case with no efficiency measures

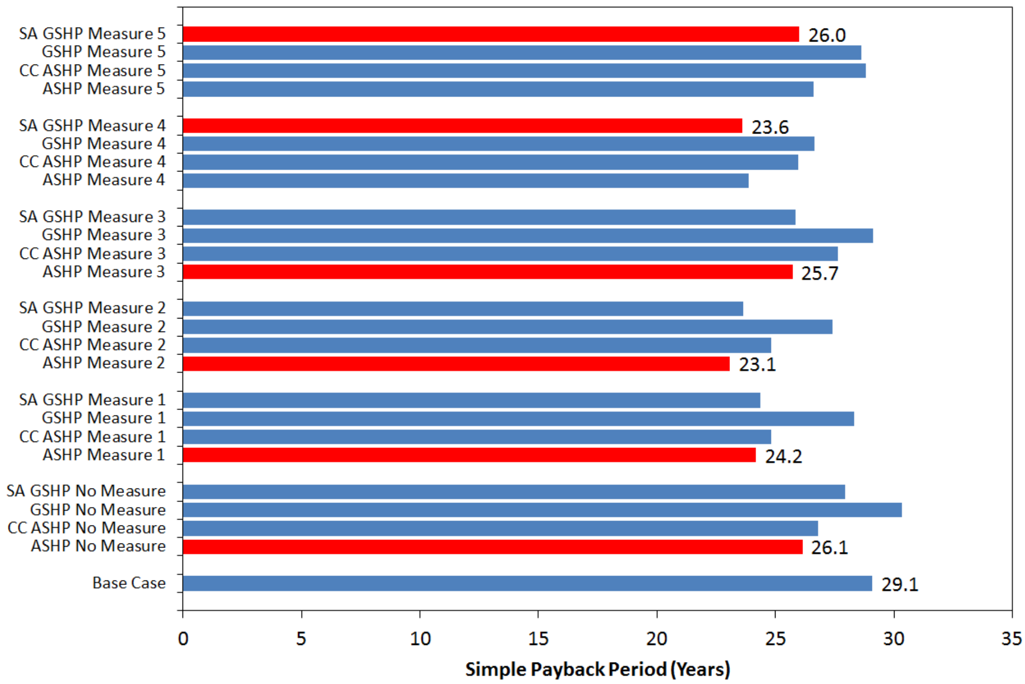


Fig. 4: Simple payback period for the heat pumps systems with a PV Array to achieve Net Zero Energy in comparison to the base case with no efficiency measures

Plotting the energy efficiency measure to PV array cost difference to achieve the same energy savings, it can be seen that the conventional ASHP generates the highest cost difference.

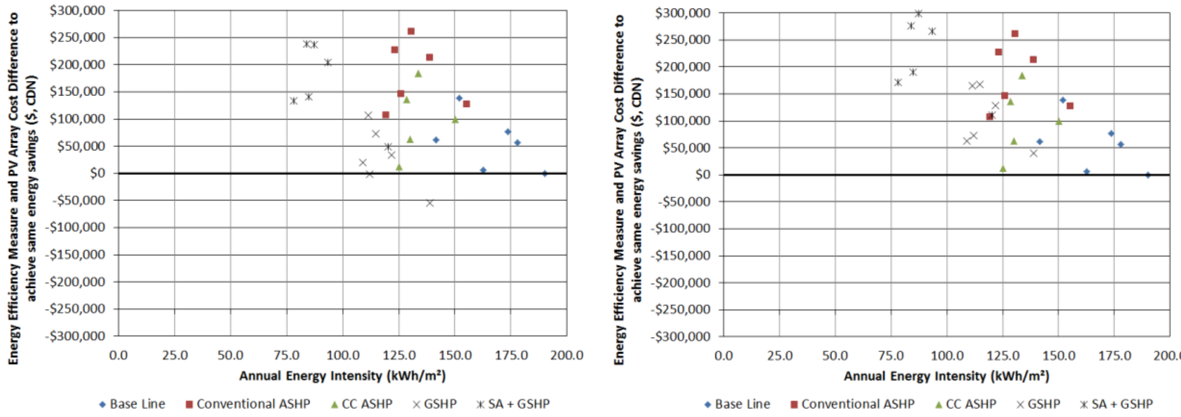


Fig. 5: Energy efficiency impact cost comparison left: $80/m and right: $50/m bore field drilling cost

Thus, implementing any efficiency measures evaluated in this study beyond this is not as beneficial as using PV. To evaluate the impact of borefield drilling costs on the most suitable system selection, the energy efficiency measure to PV cost array difference is plotted for a $50/m borefield cost, see Fig. 5. The results highlight that the solar assisted GSHP is the most suitable, with the highest cost difference.

Fig. 6: Energy efficiency impact cost comparison for left: $1/m² and right: $10/m² land value

In all measures, it was assumed that there is sufficient land space to install a PV array.

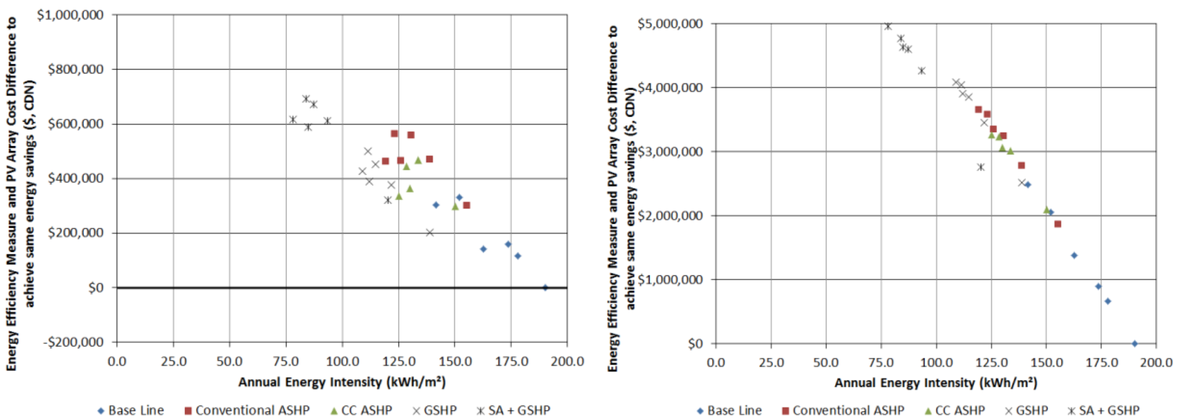


Fig. 7: Energy efficiency impact cost comparison for left: $100/m² and right: $1,000/m² land value

In the event that there is a land cost associated with installing a PV array, it can be seen in Fig. 6 and Fig. 7 that achieving energy efficiency by more efficient heating systems becomes beneficial – especially when the value of land approaches \$100/m².
Currently, in Montreal, the cost of land is approaching \$2,000/m² (Évaluation Foncière, 2014). Thus, when taking the land value into account, it can be seen that it becomes beneficial to reduce the annual energy consumption as much as possible, if net zero energy is the end target. The land cost breakeven point is estimated to be around \$15/m², where it becomes beneficial to implement a solar assisted ground source heat pump system.

3 Case studies in Finland

In Finland a case study of nZEB concepts for Finnish boundary conditions has been performed for three building types. One of the most important goals of this research project was to define and clarify the role of the heat pumps in nZEB building industry and to offer a realistic view what is the reasonable and cost-effective nZEB level in the Finnish climate. The investigated buildings are a new detached house, a new apartment building and an existing apartment building, built in the 1960s. The data of the new detached house and the 1960s apartment building are based on model buildings defined in a previous Cost optimal project (Vainio et al., 2012) that has already been completed. The data of the new apartment building is based on a real building that is under construction. The usage profiles and specific powers of the internal heat gains for lighting, persons and household equipment are based on the Finnish building code part D3 (2012). The usage profile of the DHW is based on measured DHW consumption of a Finnish apartment building.
For all systems different heat pumps solutions are investigated, and are additionally combined with solar energy systems. The solar based energy systems are designed in a way, that the total solar collector and large PV-panel area is not larger than 50% of the building's total roof area in order to prevent shading from each other in case of a total area larger than 50%.

3.1 Evaluations in the case study

3.1.1 Target levels for energy performance in different building types

The official Finnish definition of E-value is

$$E_{D3(2012)} = \frac{\sum_i E_{DE,i} \cdot f_{DE,i}}{A_{net}} \qquad \textit{(Eq. 1)}$$

where
$E_{DE,i}$ - delivered energy i (district heating (DH), electricity, fuels, district cooling (DC)) [kWh/a]
$f_{DE,i}$ - weighing factors (DH 0.7, electricity 1.7, fossil (renewable) fuels 1.0 (0.5), district cooling 0.4) [-]
A_{net} - heated net floor area of the building [m²]

The target level for energy performance ($E_{D3(2012)}$) was set to 0 kWh/(m²a) for all three building types of the project. The E-value is calculated with its current definition (Eq. 1). Tab. 15 shows proposals for the E-value elaborated in the FInZEB project, which were used in the Finnish case study.

Tab. 15: Proposal for E-values for nZEB according to the FInZEB project used in the case study

Building Type	E-value requirement D3/2012	Proposal for nZEB-E-value	Change
Small residential buildings (depending on the size of the building	160..204	120..204	
Apartment buildings	130	116	-11%
Office buildings	170	90	-47%
Schools	170	104	-39%
Day care centres	170	107	-37%
Retail and commercial buildings	240	143	-40%
Sports hall	170	115	-32%
Hotels	240	182	-24%
Hospitals	450	418	-7%

In the study also the definition by REHVA (Kurnitski, 2013) was evaluated, which takes exported energy into account (Eq. 2).

$$E_{EXP} = \frac{\sum_i E_{DE,i} \cdot f_{DE,i} - \sum_i E_{EXP,i} \cdot f_{EXP,i}}{A_{net}} \qquad \textit{(Eq. 2)}$$

where

$E_{EXP,i}$ - annual energy i, that is exported from the building, [kWh/a]

$f_{EXP,i}$ - weighing factors of exported energy form i, [-]

A_{net} - heated net floor area of the building [m²]

f_{EXP} is assumed to be same as the factor of delivered electricity f_{DE} in this study, because there is no national definition for the weighing factors of exported energy in Finland yet.

3.1.2 Life cycle cost analysis

The Life Cycle Cost (LCC) analysis takes the impact of both the capital cost and the life cycle cost into account by using the net present value. The resulting value will be the cost effect of different options. The life cycle calculations are carried out during the following 25-year period which is in line with the average of the calculation period used in FInZEB project (Reinikainen et al., 2015) (20 years and 30 years). In addition, a sensitivity analysis for life cycle cost of a 20-year period was accomplished.
The following issues have been taken into account in the calculation:

- The investment cost covering the design and construction cost
- Capital cost (= investment cost + financial cost – residual value)
- Service and maintenance cost
- Heating energy cost based on the average monthly tariffs and the average basic fees (Vantaan Energia Ltd., 3/2015). The selling price of electrical energy is estimated to be about 1/3 compared to the consumer price
- Electrical energy cost (based on the prices of Vantaan Energia Ltd., 6/2014)

The basic rate of interest is set to 0% in order to reflect the view of the property owner. An interest rate of 2% is included in the sensitivity analysis. Also possible taxation advantages and other supports for use of renewable energy resources and rise of energy prices have not been included, but are also addressed in the sensitivity analysis. This way the most important factors affecting the development of energy prices as the most difficult prediction factor have been noticed, as the ending the use of coal energy, investments on renewable energy, -rise of energy taxation and competition of energy companies.
The utilisation of maximal solar energy is also taken into account. The reason for this is that life cycle cost are almost the same at each building type and the respective heating system concepts. The benefits of solar energy and additional panel costs refute each other's positive effect. Therefore, it is justified to choose the system concepts with the best return of solar energy. This way it is also possible to have the lowest possible E-value.

3.2 New detached house

The building with a gross floor area of 233.6 m² and a heated net floor area of 180.0 m² is located at Helsinki in an urban environment.

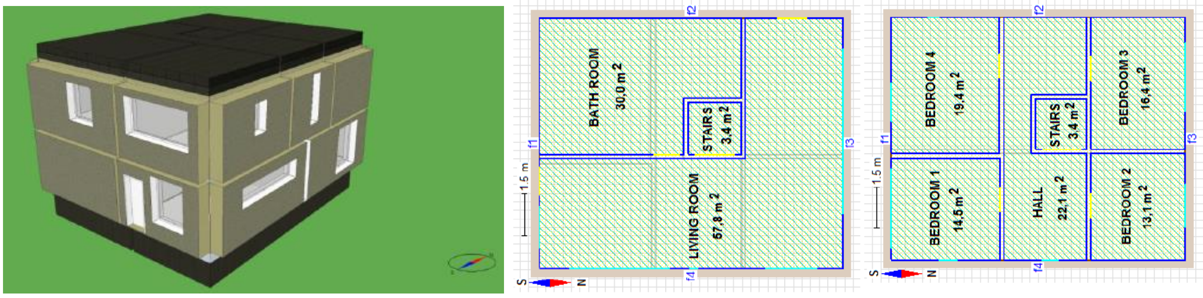


Fig. 8: Layout and floor plans of the new detached house for the Finnish case study

3.2.1 System concepts for the new detached house

The structures and the insulation thicknesses of the new detached house are kept constant in every calculation case. The exterior structures are at the Finnish passive house level (RIL 249, 2009) and the system concepts concentrate on different types of heat pumps and on the use of solar energy systems. The heat recovery unit of the ventilation system is also kept similar in each system concept, as it is reasonable to install a ventilation system with high supply air temperature efficiency in every new detached house.

The selected system concepts use heat pumps with different source and sink system, a ground source heat pump (GSHP), an outdoor air heat pump (ASHP), an exhaust air heat pump (EAHP), and an air-to-air heat pump (AAHP). The heat distribution system is a water-based floor heating system with design temperatures of 40 °C/30 °C in system concepts 1–3 and supply air heating system with electric floor heating in the bathroom in system concept 4. The system properties, which are based on the product information of the existing systems, are shown in Tab. 16.

Tab. 16: System concepts and properties used in the energy simulation of the new detached house

Properties	Concept 1	Concept 2	Concept 3	Concept 4
Main heating system	GSHP	AWHP	EAHP	AAHP
Heating power/COP of heat pump at test point	8.9 kW/COP 4.9 (0/35 °C)	8.0 kW/COP 4.4 (7/35 °C)	4.9 kW/COP 3.0 (20/35 °C)	1.9 kW/COP 3.8 (-10/37 °C)
Heat emission system	Floor heating (40/30 °C)	Floor heating (40/30 °C)	Floor heating (40/30 °C)	Supply air heating + el. floor heating in bathroom
Cooling system	Free cooling with boreholes	Floor heating with horizontal ground loop	Free cooling with horizontal ground loop	Main heating system in coo-ling mode and additional A/A heat pump for bedroom cooling
Cooling distribution system	B/W-based cooling system for supply air (10/15 °C) and a fan coil in a single bedroom	B/W-based cooling system for supply air (10/15 °C) and a fan coil in a single bedroom	B/W-based cooling system for supply air (10/15 °C) and a fan coil in a single bedroom	Cooling for supply air and a split unit in a bedroom
Ventilation system (temp. efficiency of heat recovery)	Mechanical supply-exhaust ventilation (80%)	Mechanical supply-exhaust ventilation (80%)	Mechanical supply-exhaust ventilation N/A	Mechanical supply-exhaust ventilation (80%)
Level of thermal insulation	Passive (RIL 249, 2009)	Passive (RIL 249, 2009)	Passive (RIL 249, 2009)	Passive (RIL 249, 2009)
On-site energy production	Solar collectors, PV-panels	Solar collectors, PV-panels	Solar collectors, PV-panels	Solar collectors, PV-panels

In total, 16 simulation cases are selected. There is also an extra simulation case with LED-based lighting. The extra LED-lighting simulation is performed to the most energy-efficient simulation case of the four system concepts. The four different concepts are listed below:

- Concepts without solar based energy systems
- Concepts with solar collectors (12 m²)
- Concepts with solar collectors (12 m²) and PV-panels, typical panel area (18 m²)
- Concepts with solar collectors (12 m²) and PV-panels, large panel area (36 m²)

3.2.2 Energy performance

The results show that the lowest total delivered energy consumption can be achieved with GSHP. Compared to the GSHP without the solar systems, the total delivered energy consumption is 11% (5 kWh/(m²a)) higher with AWHP, 55% (25 kWh/(m²a)) higher with EAHP or 62% (28 kWh/(m²a)) higher with AAHP.

The delivered energy saving by the solar systems depends on the energy efficiency of the main heating system. The achieved maximum saving with the solar systems is the lowest with GSHP (15 kWh/(m²a)) and highest (29 kWh/(m²a)) with AAHP. The saving is lower with GSHP due to the lower heating electricity consumption and DHW heating by the GSHP.

The E-value ($E_{D3(2012)}$) of all studied concepts fulfil the proposed Finnish nZEB level of the detached house. The proposed nZEB level of the studied detached house with 180 m² heated net floor area is 128 kWh/(m²a).

If the exported electricity is taken into account the E_{exp} is significantly lower than $E_{D3(2012)}$ with the studied PV areas. The E_{exp} reach even -2.5 kWh/(m²a) with GSHP and maximum studied PV area and LED lights. But, it should be noted, that the same weighting factors for the energy carrier were assumed for delivered and exported electricity.

3.2.3 Life cycle costs

For a more thorough analysis, four different heat pump types and GSHP with maximum amount of solar energy (GSHP/S) were chosen.

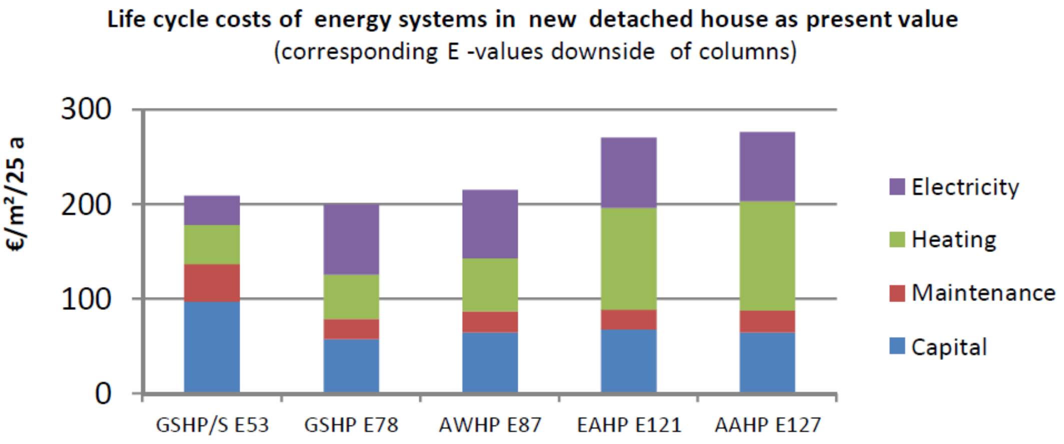


Fig. 9: Life cycle costs of energy systems in the new detached house

As seen in Fig. 9, the life cycle cost of the GSHP E78 is the cheapest. In this case it is also possible to reach a very low E-value. Even the AWHP E87 has a good energy and cost efficiency, too.

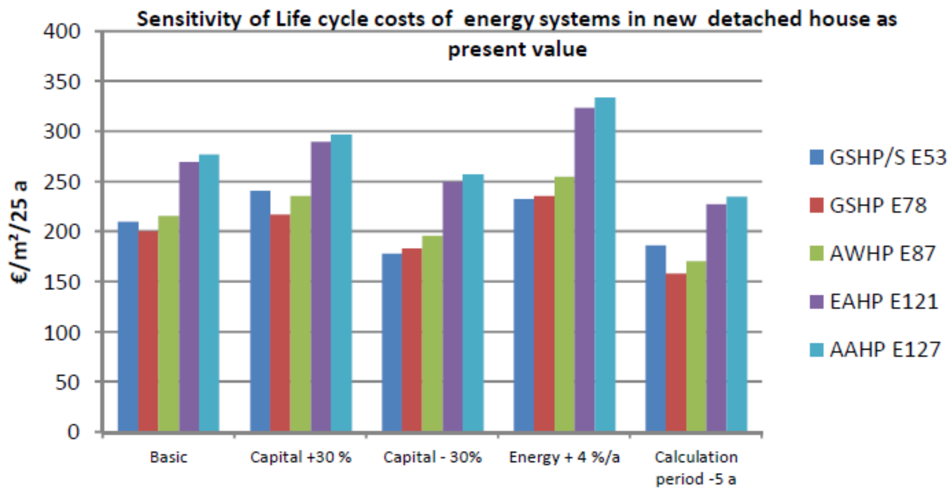


Fig. 10: Sensitivity of life cycle costs (€/m²/25 a) of the energy systems in the new detached house

The economic order of the depicted solutions does not change when the results are sensitized. The sensitivity analysis includes factors as capital cost growth +30%, capital cost reduction -30%, and annual energy cost rise +4.0% and a period of 20 a instead of 25 a (Fig. 10).

3.3 New Apartment Building

Fig. 11: Modelling of the New Apartment Building

The structures and the insulation thickness of the new apartment building are similar in each system concept. Even if the level of thermal insulation of the new apartment building almost fulfills the low energy level defined in RIL 249 (2009), the system concepts are simulated with the reference level of thermal insulation defined in D3 (2012). This lower level of thermal insulation was used because it has been shown - for example in Vinha et al. (2013) - that the better thermal insulation level brings negligible energy savings in apartment buildings in Southern Finland. The system concepts concentrate on different types of heat pumps and on the use of solar energy systems.

The heat distribution system is a water-based low-temperature radiator heating system with design temperatures of 45/35 °C in every system concept and in every simulation case. The low temperature heating system is beneficial for all heat pump types, increasing the COP-value of the heat pump. It is also beneficial for the district heating system by allowing lower return water temperatures in the heating system and in the district heating network. The selected system concepts and their properties are shown in Tab. 17, which are based on the product information of the existing systems.

Tab. 17: Properties of the system concepts for the energy simulation of a new apartment building

Properties	Concept 1	Concept 2	Concept 3
Main heating system	GSHP	AWHP	District Heating
Heating power and COP of the heat pump in a test point	61 kW COP 3.6 (0/45 °C)	64 kW COP 4.2 (7/45 °C)	-
Heat distribution system	Low temperature radiators (45/35 °C)	Low temperature radiators (45/35 °C)	Low temperature radiators (45/35 °C)
Cooling system	Free-cooling with boreholes	Mechanical water chiller system	Mechanical water chiller system
Cooling distribution system	Water-based cooling system for supply air (10/15 °C)	Water-based cooling system for supply air (10/15 °C)	Water-based cooling system for supply air (10/15 °C)
Ventilation system (temp. efficiency of heat recovery)	Mechanical supply and exhaust ventilation (80%) (Parking hall: 75%)	Mechanical supply and exhaust ventilation (80%) (Parking hall: 75%)	Mechanical supply and exhaust ventilation (80%) (Parking hall: 75%)
Level of thermal insulation	Reference level 2012 (D3, 2012)	Reference level 2012 (D3, 2012)	Reference level 2012 (D3, 2012)
On-site energy production	Solar collectors, PV-panels	Solar collectors, PV-panels	Solar collector, PV-panels

The heat pumps used in the concepts are selected as result of the current trend in the heat pump market, as well as on the experience and opinions of a group of professionals that are involved both in Finnish and in international heat pump markets.

According to this group's opinion, it is reasonable to choose both ground-source and air-to-water heat pumps as the main heating system in a new apartment building, rather than choosing an exhaust air heat pump, for example. The market and the technology is developing fast at the moment and there are new solutions and system concepts arising, especially in the air-source heat pump category, making the air-source heat pump a potential main heating system also in new apartment buildings in the near future. Ground-source heat pumps are already known to be a powerful main heating system in new apartment buildings, as well as in almost any type of building.
In total, 12 simulation cases are selected from the proposed concepts. There is also an extra simulation case with LED-based lighting. The extra LED-lighting simulation is performed with the most energy-efficient simulation case of the three system concepts. The four different concepts are listed below:

- Concepts without solar based energy systems
- Concepts with solar collectors, (typical collector area of 78 m² is selected due to a reasonable amount of DHW heating energy produced by the solar collectors based on the DHW consumption and the roof area of the building)
- Concepts with solar collectors (78 m²) and typical PV-panels (66 m²)
- Concepts with solar collectors (78 m²) and large PV-panels (200 m²)

The solar based energy systems are dimensioned in a way that the total solar collector and large PV-panel area is below 50% total roof area of the building, as the panels and collectors begin to shade each other, if the total area is increased beyond 50%.

3.3.1 Energy performance

The results show that the lowest total delivered energy consumption can be achieved with GSHP. Compared to the GSHP without the solar systems, the total delivered energy consumption is 7% (4 kWh/(m²a)) higher with AWHP or 63 % (35 kWh/(m²a)) higher with district heating.
The delivered energy saving by the solar systems depends on the main heating system. The achieved maximum saving with the solar systems is the lowest with GSHP (12 kWh/(m²a)) and highest (26 kWh/(m²a)) with district heating. The saving is lower with GSHP, especially due to DHW heating by the GSHP.
The E-value ($E_{D3(2012)}$) of all the studied concepts fulfill the proposed nZEB level of the new apartment building set to 116 kWh/(m²a).
According to the results, almost all the produced electricity can be used in the building, so the difference between the $E_{D3(2012)}$ and E_{exp} is small.

3.3.2 Life cycle costs

The life cycle cost of the GSHP E96 is the most inexpensive solution and the AWHP E103 is almost as good as the ground-source heat pump.

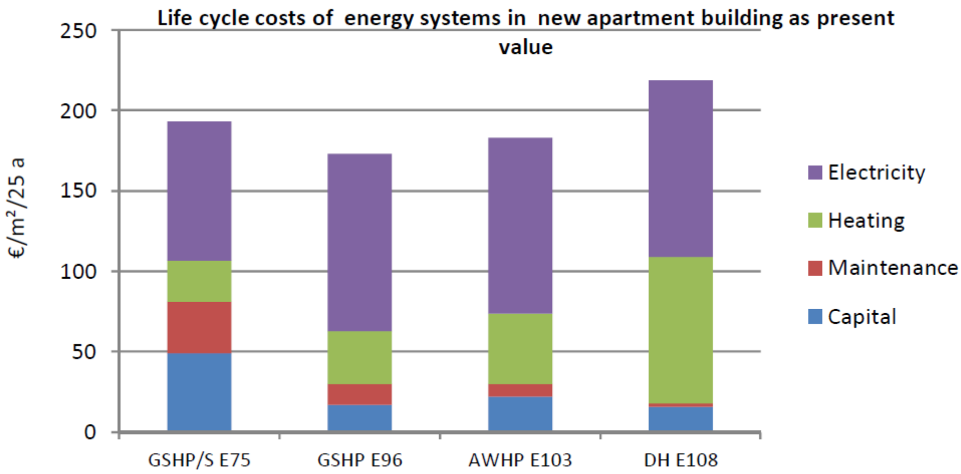


Fig. 12: Life cycle cost of the energy systems in the new apartment building

In addition, when solar energy is taken into account with the ground source heat pump (GSHP/S E53), it is also possible to reach a very low E-value.
As seen in Fig. 12, capital costs are approximately at the same level, excluding the solution where solar energy is included. Utilising solar energy also raises costs of maintenance and service. The district heating solution (DH E108) has low cost of maintenance and service compared to the other solutions. On the other hand, the energy cost of the district heating solution are much higher than with the heat pump solutions.
Fig. 13 shows the sensitivity calculations of the life cycle cost for the new apartment building. The economic order of the solutions depicted in Fig. 12 does not change when the results are sensitized. However, if the increase of electricity cost is high, the life cycle cost of AWHP become higher than those of district heating.

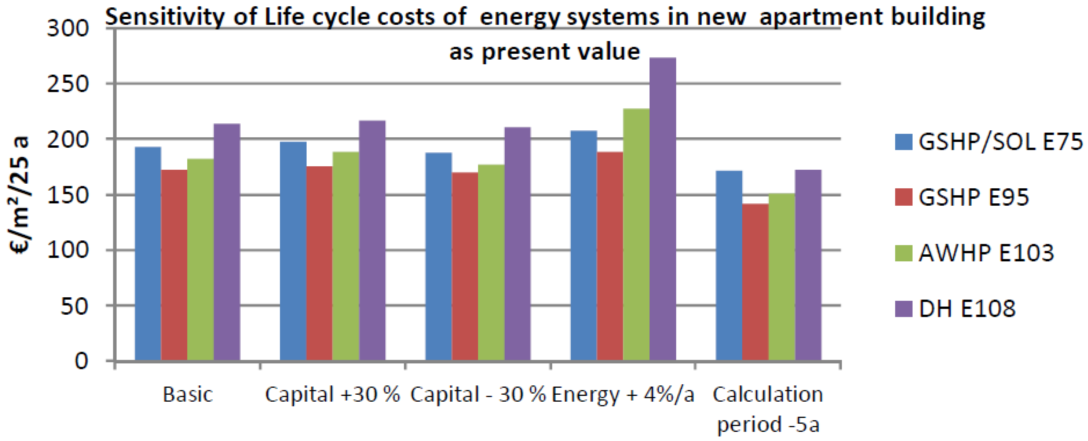


Fig. 13: Sensitivity of life cycle costs [€/(m²/25 a)] of the energy systems in the new apartment building

3.4 1960s apartment building

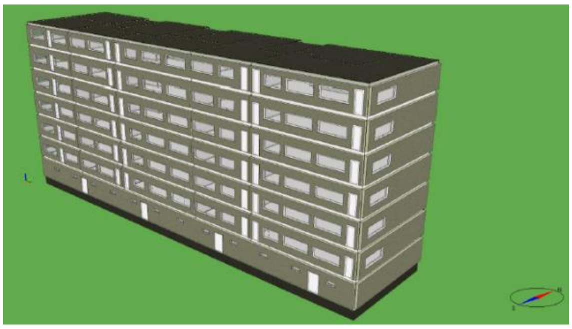


Fig. 14: Model of the 1960s apartment building

The structure and the insulation thickness of the 1960s apartment building are similar in every system concept. The heat distribution system will be renovated from the original high-temperature water-based radiator system to a water-based low-temperature radiator heating system with design temperatures of 45/35 °C in every system concept and in every simulation case, except for the initial base case simulation, where the heat distribution system is the original high temperature radiator heating system with design temperatures of 80/50 °C.

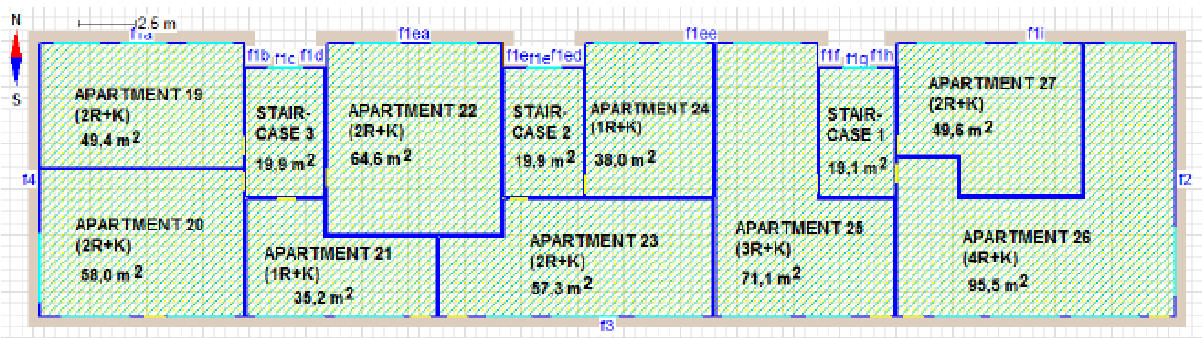


Fig. 15: Floor plan of the first floor of the 1960s apartment building

The selected system concepts and their properties are shown in Tab. 18 based on the product information of the existing systems. The selection of the concepts is based on the personal view and on the experience of the authors as well as on the results of the recent research projects that have been carried out in Sweden, considering the cost-optimal energy performance improvement measures in existing apartment buildings.

Tab. 18: Properties of the system concepts of the energy simulation for the 1960s apartment building

Properties	Concept 1	Concept 2	Concept 3	Concept 4
Main heating system	District heating	EAHP	GSHP	ASHP (air to water)
Heating power and COP of the heat pump in a test point	-	39 kW, COP 3.7 (0/45 °C)	156 kW, COP 3.7 (0/45 °C)	128 kW, COP 4.2 (7/45 °C)
Heat distribution system	Original radiators (80/50 °C)	Low temperature radiators (45/35 °C)	Low temperature radiators (45/35 °C)	Low temperature radiators (45/35 °C)
Ventilation system	Mechanical exhaust ventilation	Mechanical exhaust ventilation	Mechanical exhaust ventilation	Mechanical exhaust ventilation
Level of thermal insulation	Original	Original	Original	Original
On-site energy production	Solar collectors, PV-panels	Solar collectors, PV-panels	Solar collectors, PV-panels	Solar collectors, PV-panels

In total, 16 simulation cases are selected from the proposed concepts. There is also an extra simulation case with LED-based lighting. The extra LED-lighting simulation is performed to the most energy-efficient simulation case of the four system concepts. The four different concepts are listed below.

- Concepts without solar based energy systems
- Concepts with solar collectors, where typical collector area (90 m²) is selected due to a reasonable amount of DHW heating energy produced by the solar collectors, based on the DHW consumption and the roof area of the building
- Concepts with solar collectors and typical PV-panel area (66 m²)
- Concepts with solar collectors and large PV-panel area (200 m²)

The solar based energy systems are designed in a way that the total solar collector and large PV-panel area is not more than 50% of the building's total roof area, as the panels and collectors begin to shade each other, if the total area is increased beyond 50%.

3.4.1 Energy performance

The results show that the lowest total delivered energy consumption can be achieved with GSHP. Compared to the GSHP without the solar systems, the total delivered energy consumption is 16% (13 kWh/(m²a)) higher with AWHP, 59% (47 kWh/(m²a)) higher with EAHP or 129% (102 kWh/(m²a)) higher with district heating. The delivered energy saving by the solar systems depends on the main heating system.The achieved maximum saving with the solar systems is the lowest with GSHP (12 kWh/(m²a)) and highest (26 kWh/(m²a)) with district heating. It is notable that the E-value ($E_{D3(2012)}$) of the most energy-efficient concepts of GSHP and AWHP fulfill the proposed Finnish nZEB level (116 kWh/(m²a)) of the new apartment building. Most of the heat pump concepts and the district heating concept with the maximum studied ST and PV areas fulfill the current Finnish requirement of major renovation. According to the requirement, the E-value of the building after the major renovation should be ($E_{D3(2012)} \leq 0.85 \times E_{initial}$), where $E_{initial}$ is the E-value of the building before the renovation (Decree for the improvement of the energy efficiency of the building by repairs and modifications, 2013). The E-value of the studied building before the renovation is 162.4 kWh/(m²a). The concept fulfills the requirement, if it is less or equal than 138 kWh/(m²a) after the renovation. According to the results, the most of the produced electricity can be used in the building, so the difference between the $E_{D3(2012)}$ and E_{exp} is small.

3.4.2 Life cycle cost

The capital cost of district heating (DH E162) is low compared to the other solutions due to high residual value. It is also important to take into account the big differences between investment cost of the different solutions.
The AWHP E139 is more expensive than the other solutions, which is due to the low residual value of these heating methods. DH is used in every solution as a back-up system, which weakens the other solutions' competitive power. This includes the other solutions' extra costs, since the back-up system has to be fitted to match the maximum design power of the building.

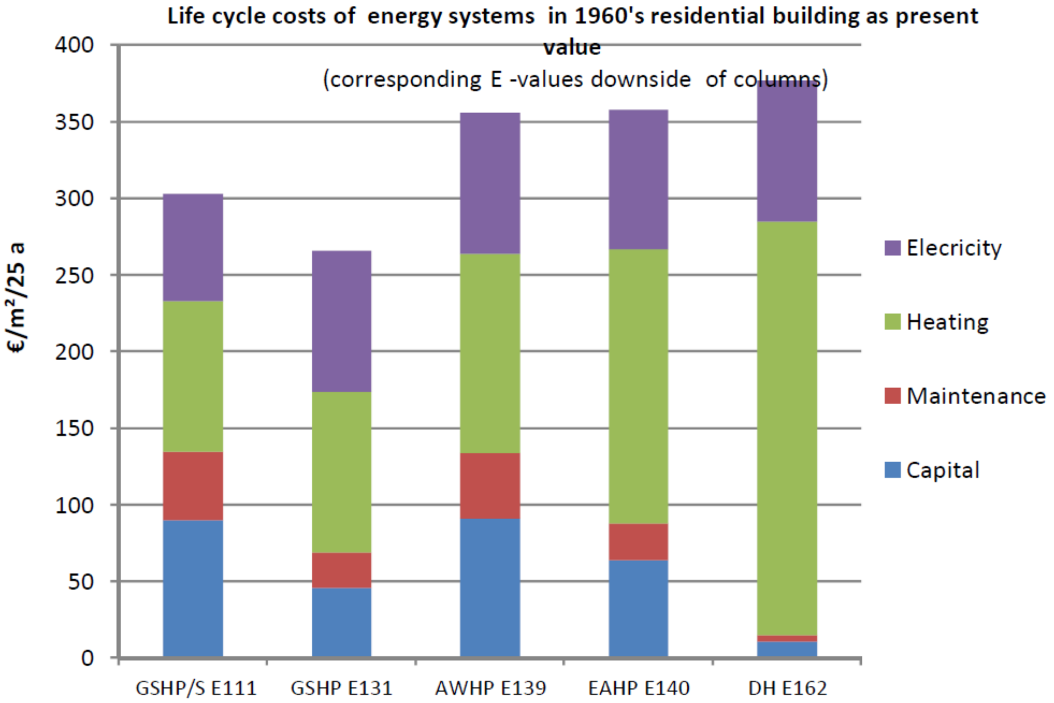


Fig. 16: Life cycle costs of energy systems in 1960s apartment building

The economic order of the solutions depicted in Fig. 16 does not change when the results are sensitized. The economic efficiency of the GSHP is even better in two cases: when energy costs rise and/or the GSHPs investment costs decrease at the same time when the investment costs of district heating stay the same. The results are reported in order of E-value starting with the lowest as depicted in Fig. 17.

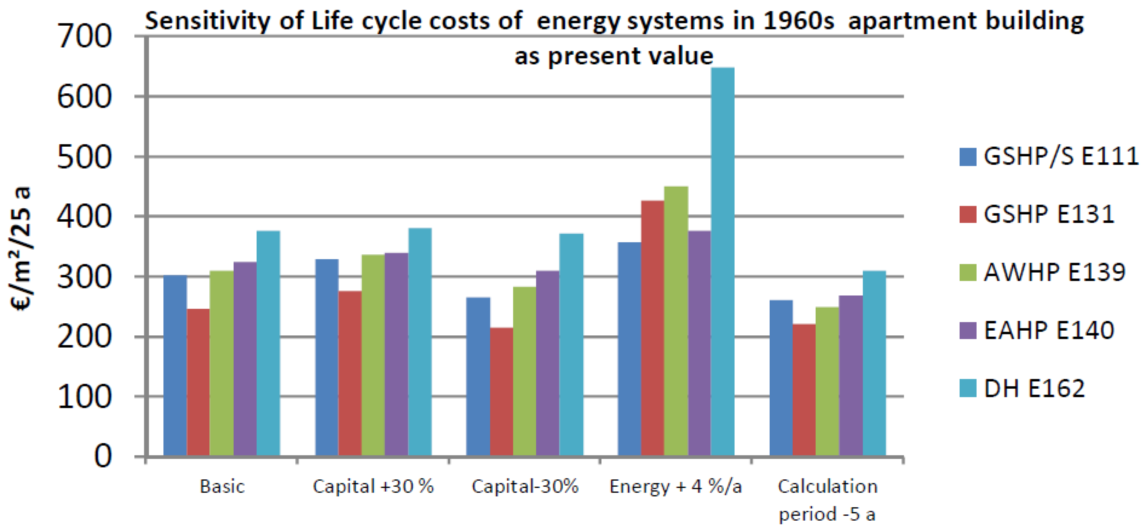


*Fig. 17: Sensitivity of life cycle costs (€/(m²*25 a)) of the energy systems in 1960s apartment building*

3.5 Conclusions

The results of the HP4NZEB project are surely flattering for the heat pump industry in Finland. Finnish nearly zero energy level for buildings can be achieved more cost-efficiently with concepts utilising heat pumps than district heating. All studied concepts utilising heat pumps reached the planned “nearly zero” level. In addition to lower life cycle cost, heat pumps can also cool the building and cooling can be done almost without any extra investment needed.
As a conclusion of the project it is safe to point out that investing in a heat pump can bring added value to the building and the user, such as:

- The cooling mode of the heat pump will increase comfort and well-being of the user
- The heat pump investment will increase the market value of the building
- The image of the building gets better
- Heat pumps have a positive impact on the environment
- Heat pumps increase energy independence and decrease the building owner’s dependency on energy pricing

HP4NZEB project has definitely promoted the know-how of the Finnish businesses. All the results of the project were excellent for the heat pump business, and they give valuable information for property owners, designers, construction companies and others considering what is the best way to heat your house.

4 Case study in Japanese office buildings

As part of Task 2, the Japanese team has studied the feasibility of developing the concept of a zero energy building (ZEB) for office buildings in Japan. In promoting ZEB, the team investigated the role of heat pump technologies and associated challenges.
The feasibility of a medium-scale ZEB office building (total floor area of about 5,000 m²) was studied, including both the performance of buildings and systems by modelling three cases with different energy efficiency levels: conventional, energy-efficient, and next-generation energy-efficient buildings.
Successively, a central heat source air-conditioning system and a multi-split air-conditioning system was modelled for a large-scale office building (total floor area of about 10’000 m²) aiming to find an effective way to realize ZEB and solve potential technical challenges. The development of ZEB was studied in two cases to make full use of the different characteristics of each system. Finally, the study was extended to evaluate the issues and potentials of heat pump technologies commonly used in a cold climate by investigating office buildings in Europe (Germany), where different climate conditions compared to Japan are applied.

4.1 Study on feasibility of ZEB in Japan

On the background of abating the global warming, the necessity of nearly or Net Zero Energy Buildings (nZEB, NZEB) has been declared as political target in Japan. However, feasibility of realizing it is not sufficiently demonstrated, yet. In particular for office buildings in hot and humid climate with large space cooling and dehumidification needs, it may get difficult to reach nZEB target.
For non-residential buildings the nZEB concept may be theoretically practicable with the current technology, if it is supplied with photovoltaic devices. Nevertheless, the PV technology requires a large installation area and has high installation cost, which pose impediments to the practical implementation. Among others, heat pump technology is considered indispensable in Japan, where a hot and humid climate governs summer due to high cooling energy reduction potentials.
The question, how to realize nZEB is discussed subject to practicable technology development in an assumed building. The building assumed is a five-storey reinforced concrete office building in Tokyo. The systems and operation conditions are assumed to be standard ones. For the air-conditioning system, individual decentralized air-conditioning with heat pumps is assumed following the prevalent style. Renewable energy is indispensable for the nZEB. In this case, the photovoltaic panels are assumed to be provided on the rooftop and walls, except for windows.

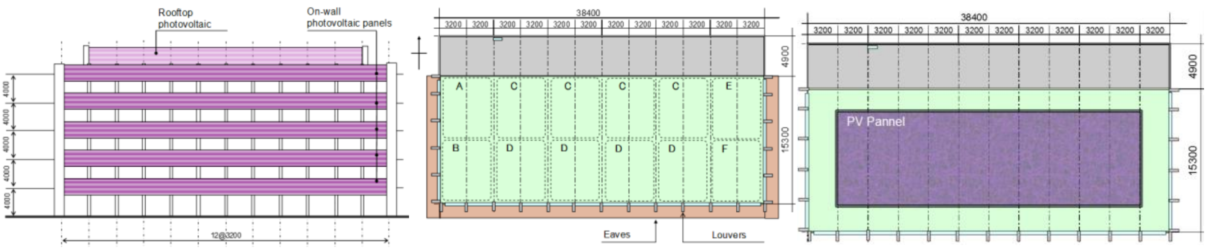


Fig. 18: Building front view (left), floor plan (middle) and roof plan (right) of the modelled building

For the purpose of the study, three cases are assumed, all of energy saving type:

- A reference building (common energy-efficient standard building)
- A high-performance building (using best available technologies (BAT))
- A ZEB-oriented building (using future technologies)

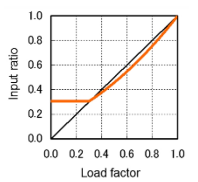
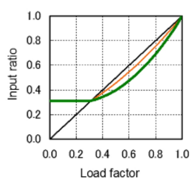
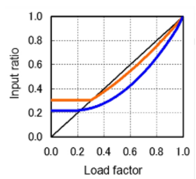
In comparison, the three cases differ in thermal insulation level, indoor heat generation rates, natural ventilation and outdoor air intake volumes. In addition, prospective technology development for the lighting (LED etc.) is applied in the ZEB-oriented building.

Tab. 19: Detailed boundary conditions for the three modelled building types

Case		1.Reference Building	2.High Performance Building	3. ZEB oriented Building
Location		Tokyo		
Usage		Office		
Scale		Five stories above ground; 3,878 m^2 in total floor area; 2,938 m^2 in air-conditioned floor space		
Indoor temperature and humidity		Summer (Jun to Sep) at 26°C/50%; Winter (Dec to Feb) at 22°C/40%; Intermediate seasons (others) at 24°		
Air-conditioner operation time		8:00 to 20:00 hours (Sunday, Saturday and legal holidays at halt)		
Outer skin	Outer walls	Concrete 150 mm thick Air layer Foamed styrene 25 mm thick Plaster board 12 mm thick	Sheet metal 1.2 mm + air layer Concrete 150 mm Foamed styrene 50 mm + air layer Plaster board 9+12 mm	Sheet metal 1.2 mm + air layer Foamed styrene 50 mm Concrete 150 mm Foamed styrene 50 mm + air layer Plaster board 9+12 mm
		Inner heat shielding K=1.03 W/m2•K	Inner heat shielding K=1.06 W/m2•K	Inner and outer heat shielding K=0.37 W/m2•K
	Window	Window ratio at 47% (window height 1.8 m)	Window ratio at 53% (window height 2.0 m)	Window ratio at 53% (window height 2.0 m)
		Single pane glass 6 mm K=4.8 W/m2•K Shade factor: 0.66 when rolling blind closed and 1.0 when open	Low-e glass K=2.6 W/m2•K Shade factor: 0.34 when rolling blind closed and 0.51 when open	AF+Low-e glass K=1.0 W/m2•K Shade factor: 0.12 when rolling blind closed and 0.3 when open
	Eaves	None	1.2m	1.8m
	Louvers	None	None	1.0 m (pitched at 1.6 m)
Indoor heat release	Lighting	13W/m2 750lx(HF)	9.5W/m2 500lx(HF)	6.5W/m2 350lx(LED+TAL+Dimer)
	Equip. heat	20W/m2 (Load factor 50% in daytime, 12.5% in night waiting)	15W/m2 (Load factor 50% in daytime, 12.5% in night waiting)	10W/m2 (Load factor 50% in daytime, 12.5% in night waiting)
	Personnel density	10 m^2/person, sensible heat 65W/person, latent heat 55 W/person		
Natural Ventilation		None	None	Effective

Across these cases, the corresponding items of equipment differ in their performance (such as rated performance, partial load performance), control methods, etc. The characteristics of the solar PV installation is a 945 m^2 area (320 m^2 on the rooftop with 30° inclination as well as 160 m^2 on the east and west façade and 320 m^2 on the south façade with 90° inclination, corresponding to 90% of the wall area without windows on). The power generation efficiency is set to 13% in the BAT building and 25% for the ZEB. An overview of the properties of the applied air-conditioning systems for the different buildings is given in Tab. 20.

Tab. 20: Properties of the air-conditioning systems

Case			1. Reference Building	2. High Performance Building	3. ZEB Oriented Building
Outdoor air conditioner	Rated operations Outdoor equipment performance coefficient	Cooling	**3.0**	**5.0**	**7.0**
		Heating	**3.5**	**5.5**	**7.5**
	Input rate characteristic against load factor during partial loading operation (in cooling)				
Outdoor air conditioner	Refrigerant control regarding evaporation temperature		constant	constant	variable
	Fan control		Not existing	Same as left	Same as left Fan at halt at thermostat off
	Fan consumption power ratio		**1.0**	**0.8**	**0.6**
Outdoor air treatment	System Total heat exchanger efficiency		Total heat exchanger	Same as left	Total heat exchanger + outdoor air treatment air conditioner
			60%	**70%**	**80%**

4.2 Result of study

For the simulation of air-conditioning, LCEM tools are used which have been opened for public use by the Ministry of Land, Infrastructure, Transport and Tourism. For the heat load calculation, HASP is used, which is the most typical heat load calculator in Japan. For the energy consumption for conveying and plumbing systems, same constant values are used.

Fig. 19 shows the primary energy consumption per air-conditioning area. The power generation by the solar PV system has been deducted from the total power consumption, assuming that the power is all consumed by battery filling up or reverse power flow. The high-performance building's energy consumption has been reduced by 26% and by 42% due to the solar PV generation. At the ZEB-oriented building of next generation type, the future expectant technology development has enabled 79% reduction, but could not arrive at a net zero-energy building. However, it has to be emphasised that as balance boundary, the entire electricity consumption (including plug loads and appliances) has been taken into account, which is an extreme boundary condition in medium rise office buildings in hot and humid climate with significant dehumidification loads. Thus, it is easily understandable that realizing the nZEB may be difficult; however, it has been demonstrated that there is a realistic hope on the premises of prospective technological advancement in the future.

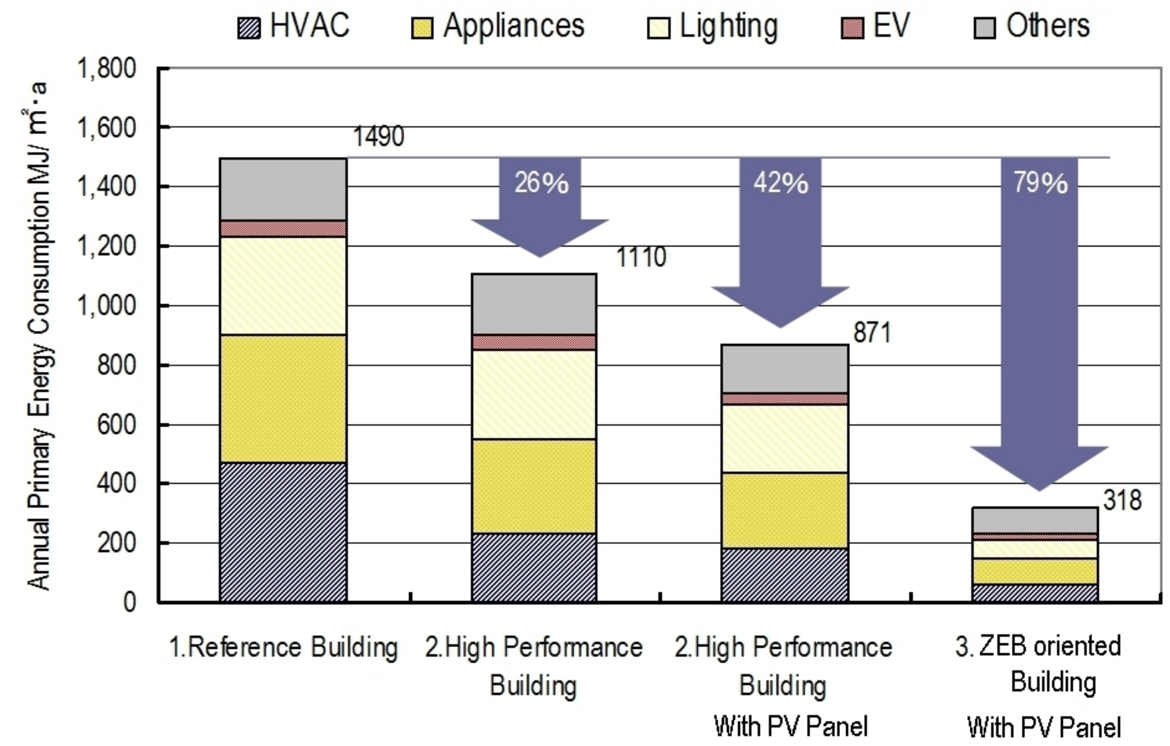


Fig. 19: Results of the medium size office building study as annual primary energy consumption

As conclusion, this study has calculated the primary energy consumption of the model buildings on a trial basis in order to verify the feasibility of the nZEB in a hot and humid climate. As a result, it has been shown that the current technological level enables 42% energy reduction and the prospective future technological advancement about 80%.

4.3 System solution towards realization of ZEB

Based on the results different technology options under development for an even larger office building of 10,000 m² of floor area have been investigated in follow-on case studies.

4.3.1 Building model

The annual heat load was calculated by modelling an office building located in Tokyo with a total floor area of about 10,000 m². Overview of the building model:

- Use of building: office building located in Tokyo
- Construction: reinforced concrete building
- Number of floors: 5, typical floor area 2,144 m²

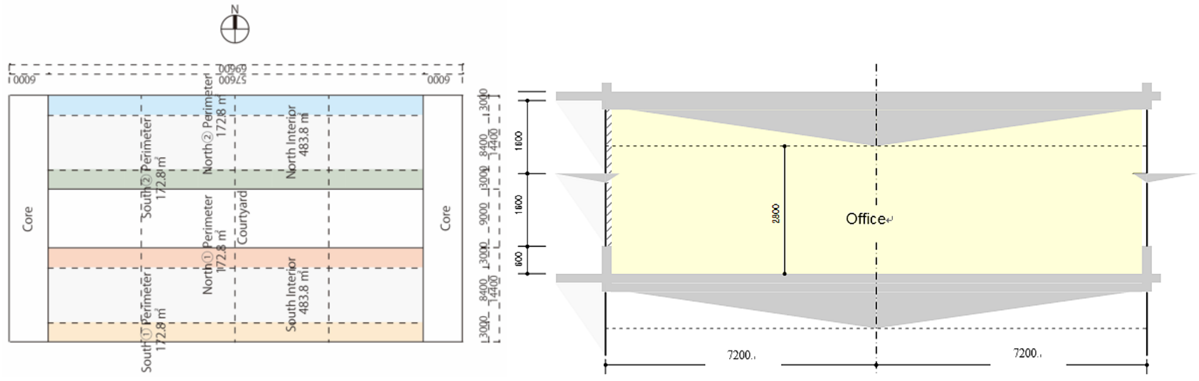


Fig. 20: Floor plan (left) and office (right) of the large office building used in the Japanese case study

Details on the large office construction and material properties are given in Appendix A.3.

4.3.2 Case study I: Installation of central heat source system

For the next-generation energy-efficient building, an air-conditioning system that consists of dedicated outdoor-air processing unit, radiation panels, and a reheating system has been constructed by NEDO. Fig. 21 shows the configuration of the considered air-conditioning system.

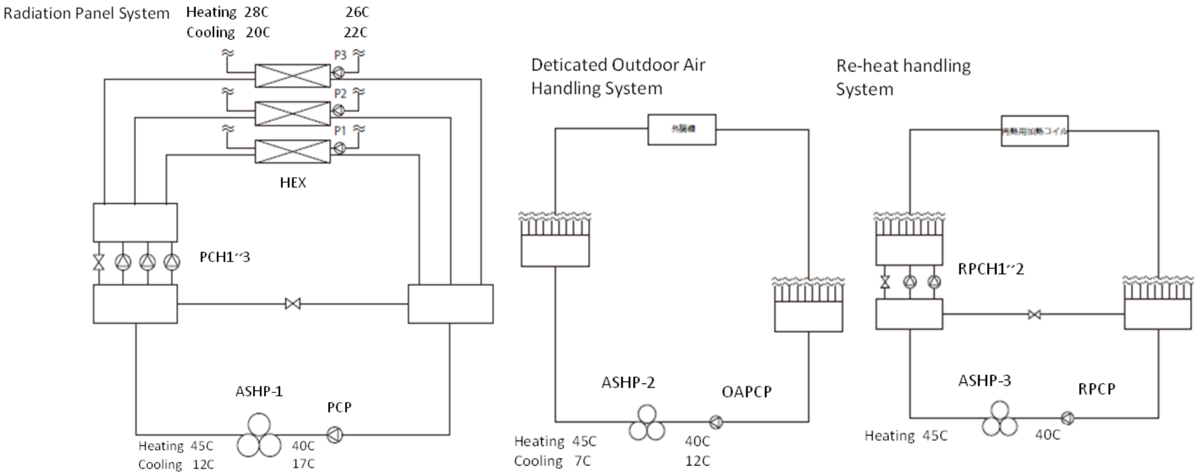


Fig. 21: Air-conditioning systems of the large office building used in the Japanese case study

The main components installed in the system are given in Tab. 21Tab. 20. For the outdoor-air processing unit, the design supply air temperature is set at 15.7 °C, so that at least the space latent heat load can be processed. To suppress excessive overcooling, this system is designed to control (raise) supply air temperature depending on the space latent heat load. How this system is actually controlled needs to be studied further in the future.

Fig. 22 shows the simulation results of energy consumption as comparison of two occupancy schemes of the office space.

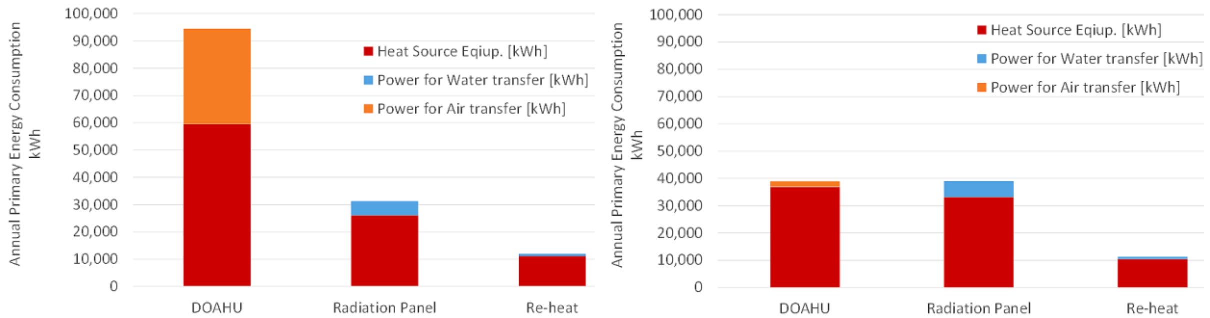


Fig. 22: Results for different air-conditioning systems of the large office building used in the Japanese case study

Tab. 21: Component characteristics of main parts of the air conditioning system in Fig. 21

Equipment	Specificationds		Phase−Volta (φ−V)	Power (kW)	Num.	Others
Air−Source Heat pump Chiller ASHP−1	Cooling Capacity 300kW Chilled Water vol. 860L/min Heating Capacity 355kW Hot water Vol. 1018L/min	(17−12℃) (40−45℃)	3−200	82.0 95.0	1	Heat Source for Radiation pannel (RR-XX1-310H_300)
Air−Source Heat pump Chiller ASHP−2	Cooling Capacity 85kW Chilled Water vol. 243L/min Heating Capacity 85kW Hot water Vol. 243L/min	(12−7℃) (40−45℃)	3−200	23.3 24.2	3	Heat Source for DOAHU (RR-XX2-310UH_85)
Air−Source Heat pump Chiller ASHP−3	Cooling Capacity 180kW Chilled Water vol. 517L/min Heating Capacity 180kW Hot water Vol. 517L/min	(12−7℃) (40−45℃)	3−200	48.5 57.1	1	Heat Source for Re-Heat (RR-XX1-310H_180)
1st Pump PCP	Wtr flow rate 860L/min	Lifting height 44.9kPa	3-200	0.9	1	For Radiation panel
1st Pump OAPCP	Wtr flow rate 730L/min	Lifting height 33.3kPa	3-200	0.4	1	For DOAHU
1st Pump RPCP	Wtr flow rate 517L/min	Lifting height 33.3kPa	3-200	0.7	1	For Re-heat
2nd Pump PCH	Wtr flow rate 282L/min	Lifting height 300kPa	3-200	2.3	3	For Radiation panel
2nd Pump OAPCH	Wtr flow rate 126L/min	Lifting height 339kPa	3-200	0.6	1	For DOAHU
2nd Pump RPCH	Wtr flow rate 216L/min	Lifting height 339kPa	3-200	2.1	2	For Re-heat
Wtr Pump P1	Wtr flow rate 760L/min	Lifting height 300kPa	3-200	5.4	1	For Interior zone
Wtr Pump P2	Wtr flow rate 778L/min	Lifting height 300kPa	3-200	5.8	1	For perimeter N①②＋1～4F Perimeter S②
Wtr Pump P3	Wtr flow rate 383L/min	Lifting height 300kPa	3-200	0.9	1	For perimeter S①＋5F perimeter S②
Radiation panel 1 (600mm×1800mm)	Sensible Cooling Cap 71W/㎡ sensible Heating Cap 53W/㎡	−	−	−	800	For interior zone
Radiation panel 1 (600mm×1800mm)	Sensible Cooling Cap 88W/㎡ sensible Heating Cap 60W/㎡	−	−	−	784	For perimeter N①②＋1～4F perimeter S②
Radiation panel 1 (600mm×1800mm)	Sensible Cooling Cap 88W/㎡ sensible Heating Cap 60W/㎡	−	−	−	378	For perimeter S①＋5F perimeter S②
DOAHU	Num. of row 6 rows Air volume 3733㎥/h Cooling ap. 47.1kW Heating Cap. 19.5kW	Num of Tubes 14 Static Press. 682Pa Humidification 7.85kg/h	3-200	2.2	2×5	For perimeter + latent heat ＋Fresh air load
Re−Heating coil	Num. of row 3列 Air volume 3733㎥/h Heating Cap. 30.5kW		3-200	−	2×5	For Re−heating
Air−to−air Heat exchanger	Exchange effi. Sensible 75% Latent 40%	−	3-200	—	2×5	

Fig. 22 left contains the results, when in-room headcount per unit is 0.15 persons/m² (corresponding to 6.66 m²/person). When the in-room headcount per unit decreased from 0.15 to 0.06 persons/m² (corresponding to 16.66 m²/person), a realistic value for typical office buildings is achieved. In this case, the specific primary energy consumption of this air-conditioning system is calculated to be 89 MJ/(m²a). Tab. 22 gives the numerical values of this case with relaxed density of persons.

Tab. 22: Energy consumption for each system

	DOAHU	Radiation panel	Re-heat	Whole system
Supply Heat rate [kWh]	91,027	85,598	32,637	209,262
Power for Water transfer [kWh]	121	5,788	811	6,719
Power for Air transfer [kWh]	2,012	0	0	2,012
Heat Source Equip. [kWh]	36,769	33,148	10,459	80,374
Total Amount [kWh]	38,903	38,935	11,268	89,105
System COP	0.84	0.79	1.05	0.85
Primary Energy Unit [MJ/(m²a)]	39	39	11	89

Tab. 23 summarises the study results of the case in which a water-source heat pump is installed in the radiation panel system, and another case in which hot- and cold-water outlet temperature requirements are relaxed. Assuming the use of a heat source from well water at 20 °C throughout the year, the heat source equipment in the radiation panel system was modified. The water-source heat pump has a cooling capacity (heating capacity) of 570 kW (539 kW) and a rated COP of 10.0, (5.16 respectively). One heat source water pump unit was added to the system to transfer well water.

Tab. 23: Study on installing water-source heat pump (HP) with the radiation panel system

	ASHP	WSHP (Well water)	Water-Source HP (Well water + mitigated chilled water temp.)
Supply heat rate [kWh]	85,598	85,598	85,598
Power for water transfer [kWh]	5,788	9,134	9,134
Power for air transfer [kWh]	0	0	0
Heat source equipment [kWh]	33,148	19,682	16,732
Total amount [kWh]	38,935	28,816	25,866
System COP [-]	0.79	1.07	1.19
Primary energy unit [MJ/(m²a)]	39	29	26

The change of the heat source equipment to a water-source heat pump reduced the energy consumption of the heat source itself by 40% compared to the use of an outdoor air-source heat pump.In the case in which the heat source equipment was changed to a water-source HP and hot- and cold-water temperature requirements were relaxed at the same time, the power consumed by the heat source equipment decreased to 16,732 kWh, raising the system COP to 1.19 and dropping specific primary energy consumption to 26 MJ/(m²a). Fig. 23 gives a summary on the simulated cases.

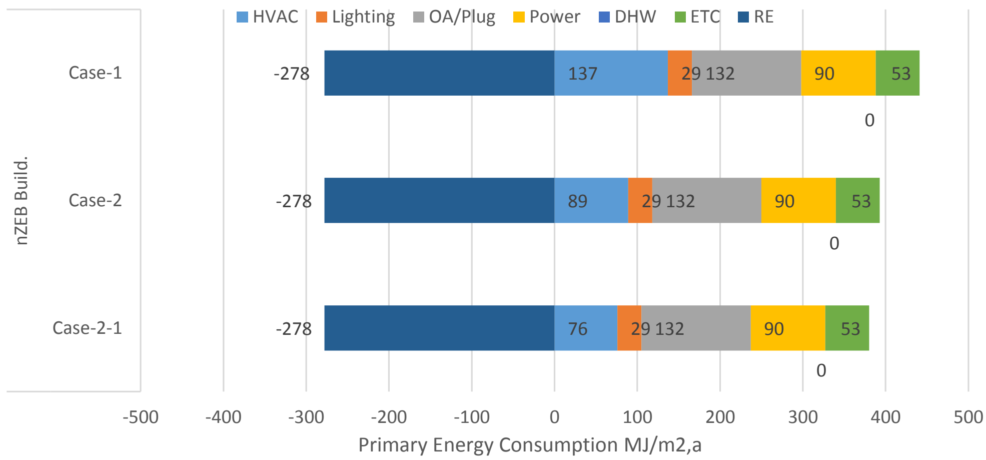


Fig. 23: Comparison of annual energy consumption between simulated cases

<u>Conclusion</u>

- In pursuing nZEB for a large-scale office building, it is necessary to maximize efforts to save the energy consumed by lighting, office automation equipment and the like. The same applies to the energy consumption for general power use, hot water supply, and other applications for unidentified purposes.
- As an air-conditioning system for an nZEB-oriented building, NEDO proposed a radiation panel system and studied its energy performance. As a result, the system can achieve an annual system COP of 1.04, indicating that high-efficiency air-conditioning operation is possible.
- As a step to further improve the efficiency of the radiation panel air-conditioning system, it is necessary to take additional measures including a reduction in air transfer power in the outdoor-air processing system, an improvement in the efficiency of the radiation panel system, and the use of solar heat in the reheating system.
- The use of unharnessed energy sources like well water heat could contribute to efficiency improvement, but it is absolutely necessary to save the power consumed by the pump to circulate heat source water.
- By optimised operation that allows the system to satisfy actual performance requirements (e.g. optimised outdoor air intake), more energy will be saved in the future. In this respect, priority should be given to such effort.

4.3.3 Case Study II: Installation of multi-split heat source

In Japan, multi-split heat source systems (multi-split air conditioners for buildings), which circulate the refrigerant directly through room spaces, are widely used in commercial buildings. This is why NEDO calculated energy consumption for air-conditioning with application of multi-split heat source system. This trial calculation was conducted for the same building and operating conditions that had been used to calculate the energy consumption for air-conditioning in the central heat source system. Just like the calculation conducted for the central heat source system, the trial calculation was conducted for two cases with different in-room headcount per unit area: Case 1 for 0.15 persons/m² and Case 2 for 0.06 persons/m².

Heat load processed by the air-conditioner

The sensible heat load in Case 2 drops more during the cooling period than in Case 1. At the same time, it rises more during the heating period. For both Case 1 and Case 2, the capacity of the air-conditioner was selected based on the peak load during the cooling period. In Case 1, however, latent heat is processed, while peak sensible heat is processed. This is why, the air conditioner in Case 1 has a 40% higher capacity than Case 2 after the ratio of peak processing sensible heat load is taken into account.
Due to its smaller outdoor air intake, the heat load processed in Case 2 is significantly smaller than in Case 1. The desiccant outdoor air processing unit characteristically processes mostly latent heat in the summertime and sensible heat in the wintertime. Details are given in Appendix A.3.

Calculation Results

Fig. 24 shows a comparison of the heating and cooling load (left) and the electrical consumption of the two investigated cases. During the cooling period the outdoor-air processing unit processes most of the latent heat load, and the air-conditioner processes mainly sensible heat load. In addition, as the internal heat load significantly decreased, the percentage of total heat load processed by the outdoor-air processing unit considerably increased.

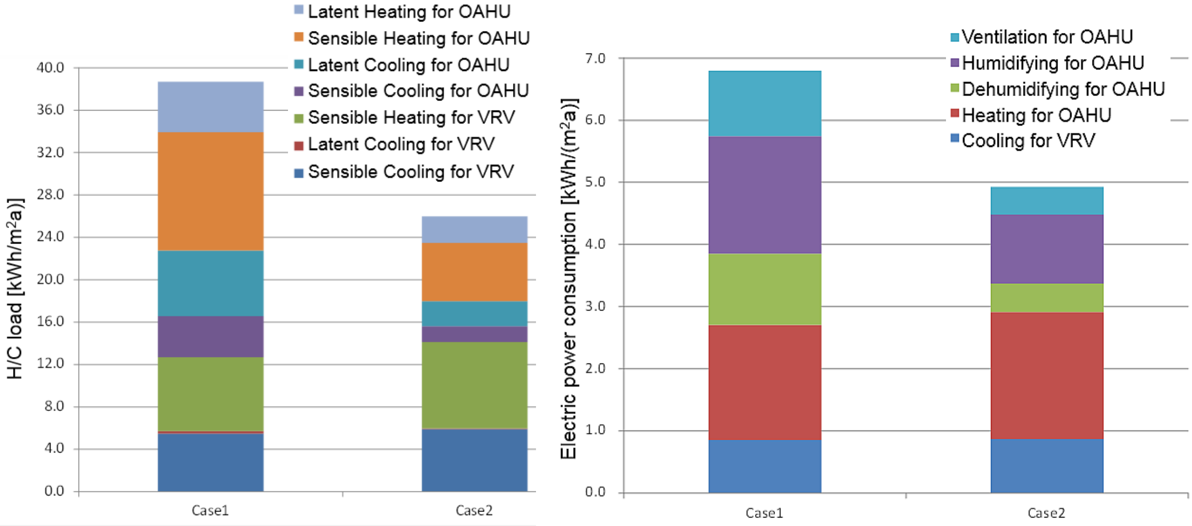


Fig. 24: Comparison of annual energy consumption between simulated cases

In Case 2, power consumption was about 28% less than that in Case 1, because of the reduction in the heat load processed by outdoor air as well as the decrease in the transfer power required for ventilation.
Particularly in Case 2, the power consumed by the multi-split air conditioner to heat the building accounts for 40% or more of the total because of the decrease in internal heat generation.
Power consumption values in both Case 1 and Case 2 are now below the tentative target for ZEB realization of 106 MJ/m².

Conclusion

- Because the space has a small heat load, outside air load should be processed efficiently through ventilation.
- As the effect of outside air load increases, the latent heat ratio of air-conditioning load becomes higher. It is therefore necessary to install a piece of equipment capable of processing latent heat more efficiently.
- The ratio of heating to cooling becomes higher. For this reason, it is necessary to make the heating operation more efficient. The efficiency of the heating operation falls during low load operation. It is necessary to select an adequate capacity for heating with the help of preheating and other operations.
- As the air-conditioning load is small, it is necessary to install smaller (other forms of) indoor equipment.

4.4 System solution toward realization of ZEB in cold climate (Europe)

The building has business offices on the north and south sides with the core zones at both ends as depicted in Fig. 25. This office building located in Germany is set to have a total floor area of 10,720 m² and a typical floor space of 2,144 m². The thermal insulation performance of the external walls and window panes are set at 0.6 W/(m²K) and 2.6 W/(m²K), respectively.

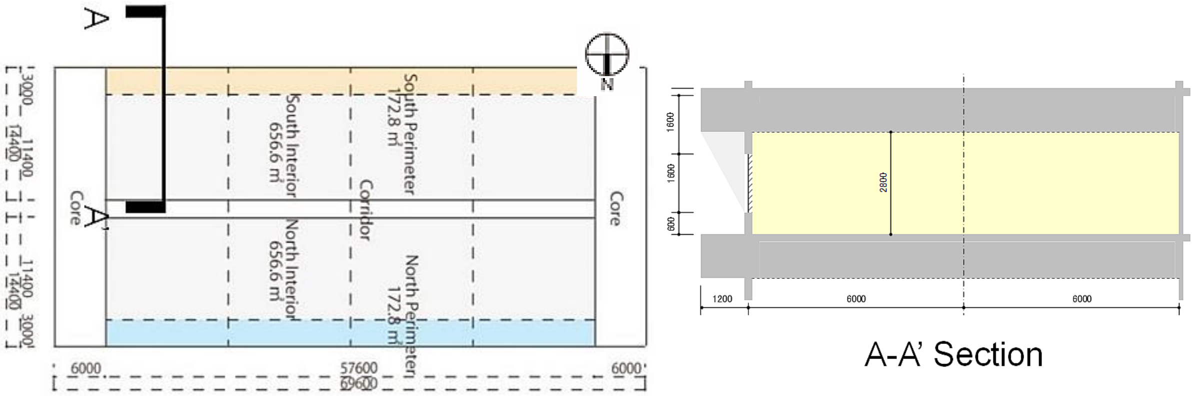


Fig. 25: Planned building model for the case study in cold climate

The temperature in the space is set to 22 °C throughout the year, with no humidity control involved. Outside air intake is set at 60 m³/(h·person) for 7 persons in the space (0.14 persons/m² or 7.1 m²/pers.). Outdoor air is processed by a heat exchanger (with a heat-exchange efficiency of 0.8).

4.4.1 Planning AC-system and study on system operation performance

Fig. 26 shows the equipment loads in each zone on the typical floor.

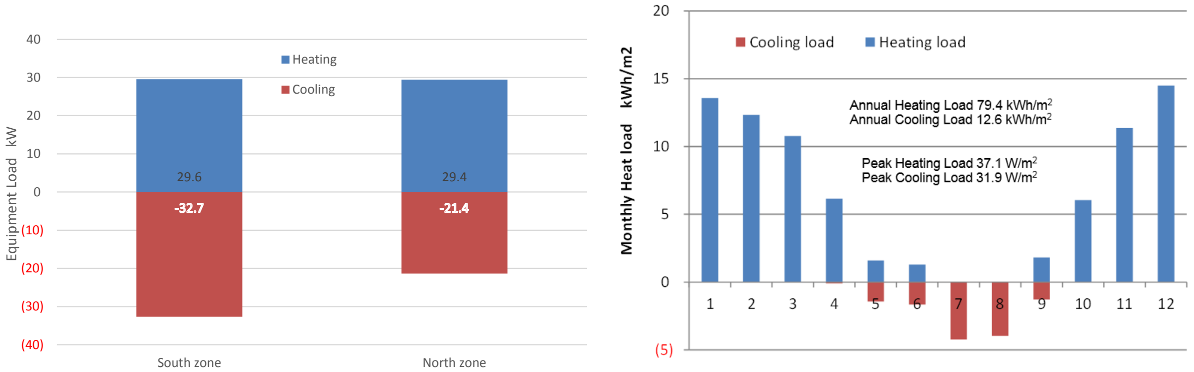


Fig. 26: Equipment loads in each zone on typical floor (left) and monthly energies, annual and peak loads (right)

Each equipment load is set at 95% of the heating and cooling load in the same zone. Note that the equipment load will be increased by 15% taking into account aging degradation and heat loss. The south-facing offices and north-facing offices have different heating and cooling equipment loads.

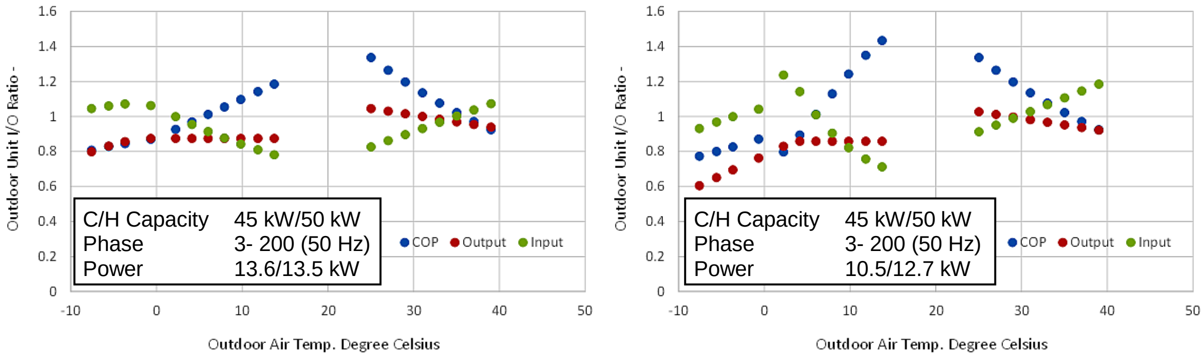


Fig. 27: Equipment characteristics of a standard type and high-efficiency type air-conditioner

The high-efficiency-type air-conditioner has a higher operation performance during both cooling and heating operation, particularly at an outdoor temperature of 5 °C or above during heating operation. The equipment characteristics are shown in Fig. 27.

Fig. 28 shows the performance characteristics of standard-type and high-efficiency-type package air-conditioners installed.

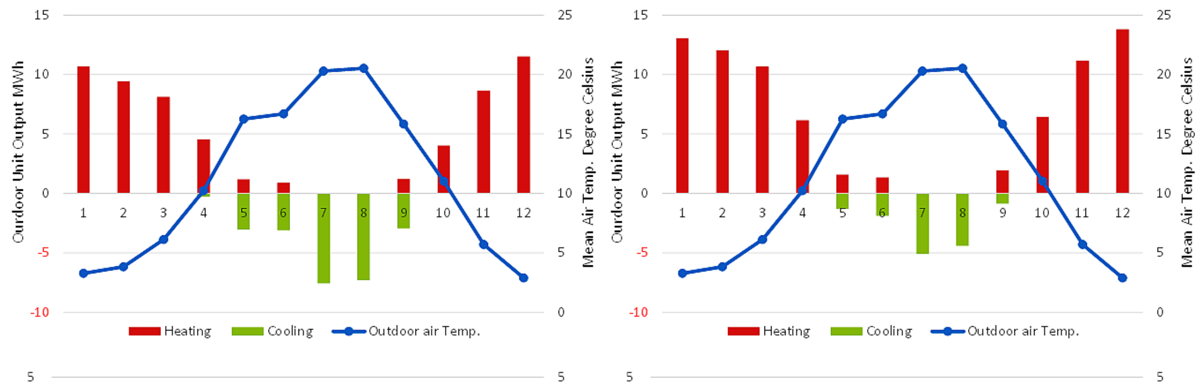


Fig. 28: Equipment characteristics of a standard type and high-efficiency type air-conditioner

The high-efficiency-type air-conditioner has a higher operation performance during both cooling and heating operation, particularly at an outdoor temperature of 5 ºC or above during heating operation.

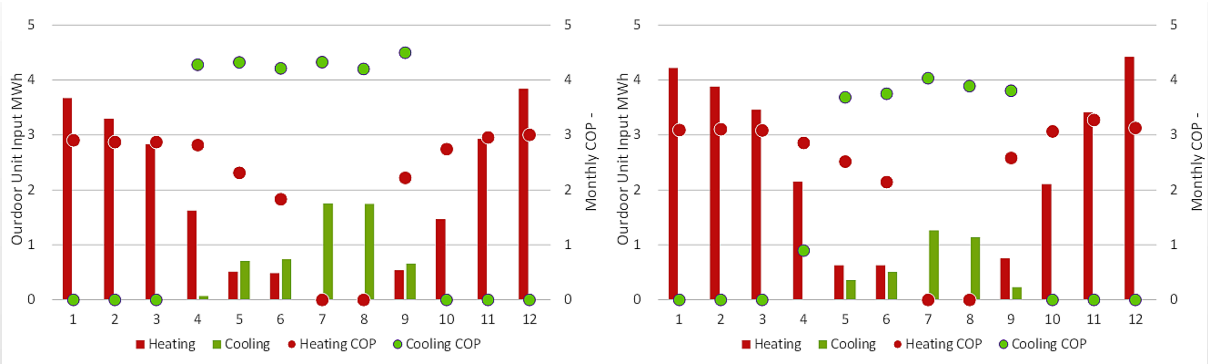


Fig. 29: Monthly performance factor of the standard (left) and high-efficiency-type (right) air-conditioner

The monthly mean COP of the heat pump during heating operation is about 3, but the value drops below that level from April to October. This is probably due to the fact that the heating load during the period is so small that the heat pump is operated at part load.
As a result, the mean COP of the heat pump during the heating period in the south-facing offices and north-facing offices is calculated to be 2.8 and 3.0, respectively.
In contrast, the monthly mean COP of the heat pump during the cooling period varies from month to month, but no sharp drop is seen, unlike during the heating period. The mean COP of the heat pumps on the south and north during the cooling period is calculated to be 4.3 and 3.9, respectively. Monthly energy values and performance factors of the standard systems and the high efficiency air-conditioner are shown in Fig. 29.
Fig. 30 shows the annual energies and annual performance factors. The mean COP of the heat pump is higher during the cooling period than the heating period. However, about 80% of the energy consumed by the heat pump is spent during heating operation, showing that the heat pump's energy consumption is very susceptible to the system's heating operation performance.

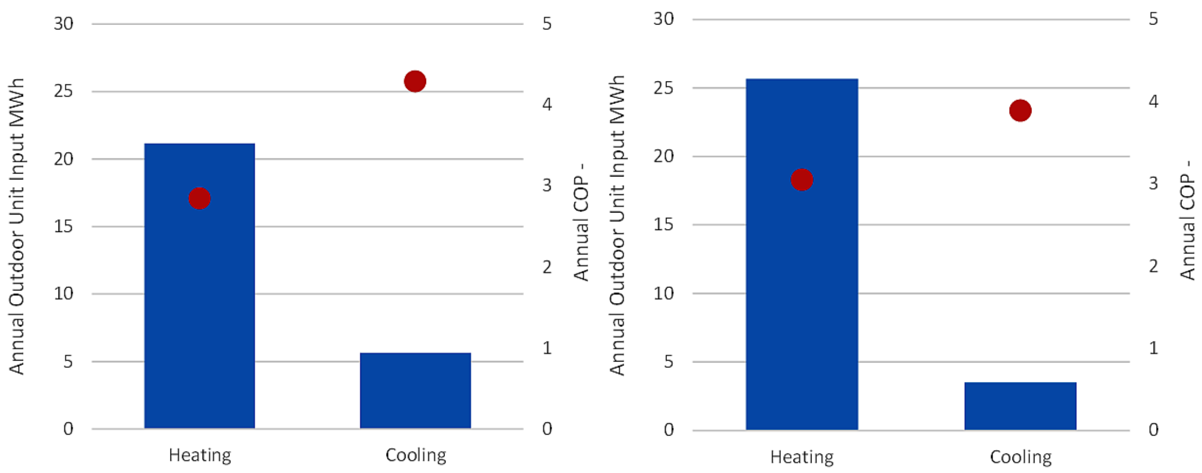


Fig. 30: Annual COP values of the standard type (left) and high-efficiency type (right) air-conditioner

Fig. 31 shows the overall energy and the seasonal performance of the outdoor unit of the two investigated variants.

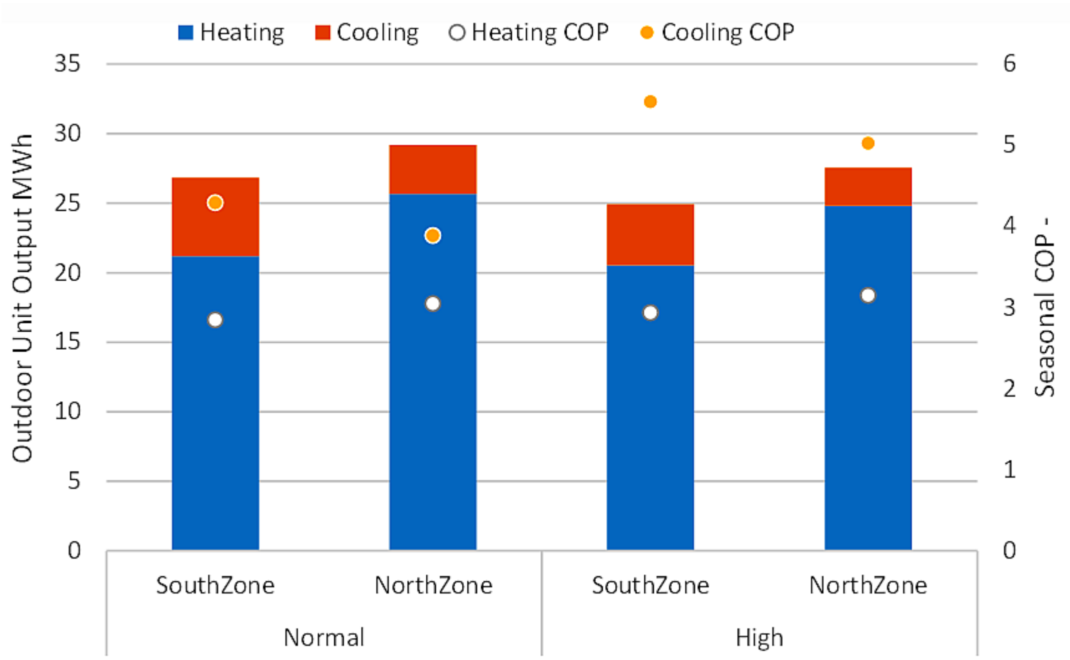


Fig. 31: Output of the Outdoor Unit

Conclusion

- Buildings in Europe are fundamentally different from their Japanese counterparts in the ratio of cooling to heating loads. With emphasis on heating load, those buildings need to improve efficiency in heating operation, and this is a major challenge. In addition, untapped energy such as geothermal heat will have to be proactively harnessed.
- Their cooling loads are generated when outdoor air temperature is low.
- It is therefore necessary to propose system solutions that proactively utilise outdoor air cooling.
- Due to equipment characteristics, the efficiency of their air-conditioners drops significantly when operated under partial load conditions. It is therefore necessary to take measures to improve the air-conditioning load factor of their buildings. It is also necessary to study energy storage technologies such as a building envelope heat storage air-conditioning system.

5 CO_2 heat pump and design tools for nZEB in Norway

In passive houses and ZEBs the demand for space heating and heating of ventilation air has been drastically reduced due to heavily insulated and airtight walls as well as the utilisation of high-efficiency heat recovery units (heat exchangers) in the ventilation system. In single-family houses, multi-family houses, row-houses, blocks of flats, apartment buildings, hotels, nursery homes, hospitals, commercial buildings and sport centres of passive house standard, the annual space heating demand is more or less equal to or lower than the annual energy demand for heating of domestic hot water (DHW).

The ratio between the annual energy demand for DHW heating and the total annual heating demand typically range from 0.5 to 0.8, i.e. 50 to 80%.

The ratio is to a large extent determined by the type of building, building shape, operating conditions as well as the climatic zone (latitude, coastal climate, inland climate).

5.1 Modelling of buildings incl. NZEB

Heating and cooling demands in selected passive house buildings have been simulated by means of the Norwegian software Simien (Programbyggerne, 2013). Simien is mainly used for simplified energy simulations and energy labelling of buildings, but can also be used for more detailed analysis.

Tab. 24 shows the simulated total gross specific thermal power demand at design outdoor temperature DOT ($P_{dim\text{-}tot}$), the total annual specific thermal heating demand ($q_{N\text{-}tot}$), and DHW share of the total annual heating demand for example-buildings (single-family-house, block of flats, and a nursey home) in the Norwegian cities Oslo (capital), Bergen (Western coastal climate), Trondheim, Røros (Eastern inland climate), and Tromsø (Northern coastal climate) in dependence of the DOT and the annual average ambient temperature (t_a).

Tab. 24: Simulated $P_{dim\text{-}tot}$, $q_{N\text{-}tot}$, and DHW share of the total annual heating demand for a single-family house, a block of flats, and a nursey home

Example	City	DOT (°C)	t_a (°C)	$P_{dim\text{-}tot}$ (W/m²)	$q_{N\text{-}tot}$ (kWh/m²)	DHW share (%)
Single-family house	Oslo (S)	-20	5.9	22	51	58
	Bergen (W)	-10	7.8	18	51	58
	Trondheim (NW)	-19	4.9	22	54	55
	Røros (E)	-40	0.5	31	66	45
	Tromsø (N)	-12	2.9	18	61	49
Block of flats	Oslo (S)	-20	5.9	18	44	68
	Bergen (W)	-10	7.8	15	40	74
	Trondheim (NW)	-19	4.9	19	46	64
	Røros (E)	-40	0.5	25	54	55
	Tromsø (N)	-12	2.9	15	49	61
Nursery home	Oslo (S)	-20	5.9	34	49	60
	Bergen (W)	-10	7.8	30	47	62
	Trondheim (NW)	-19	4.9	35	53	56
	Røros (E)	-40	0.5	51	72	42
	Tromsø (N)	-12	2.9	30	62	48

5.1.1 Conclusion

Heat pump and chiller systems for heating and cooling of different passive house buildings and ZEBs should be designed in accordance with the power duration curve of the building. An optimised design will lead to the lowest possible annual costs (€/year) for the heating and cooling system as well as long lifetime for the equipment.

Simien has proven to be a suitable tool for calculating gross/net power demands, annual energy demands as well as power duration curves for heating and cooling in buildings, which in turn can be used to find the optimum capacity for heat pump and chiller systems.

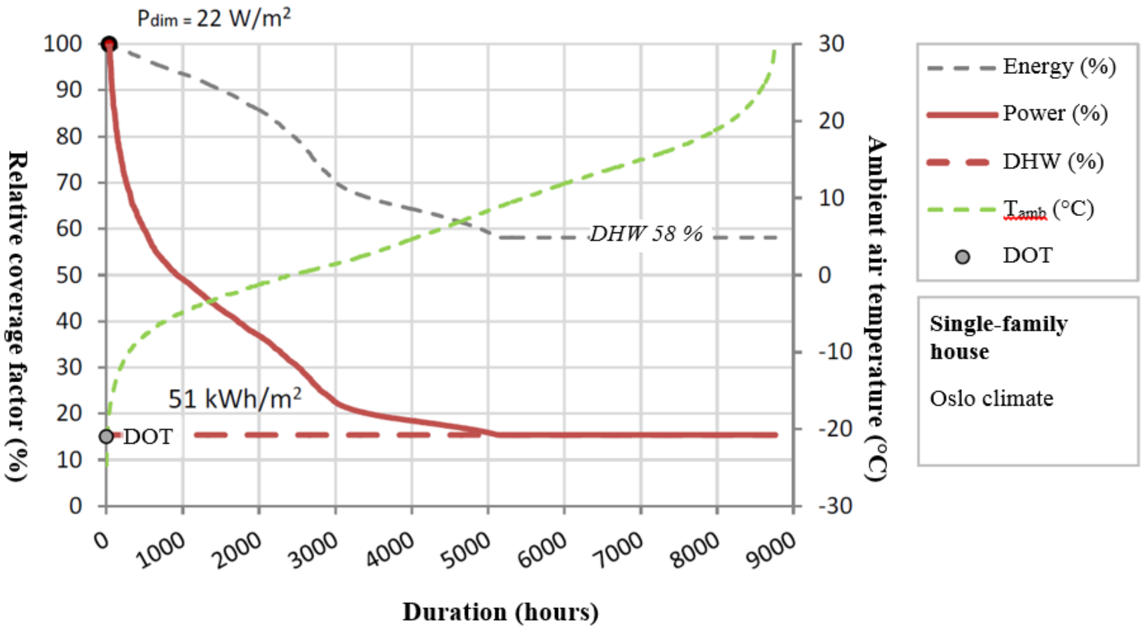


Fig. 32: Calculated thermal power duration curve for a 2-storey 128 m² single-family house of passive house standard, Oslo climate (Smedegård and Stene, 2013).

Tab. 25 gives a comparison of reduction potentials for passive houses regarding the current requirements of the building code TEK 10 (2010) in Oslo climate. In passive houses a significant reduction of the space heating demand by a factor 2.4-3.0 and the heat load by a factor 1.8-2.2 at design outdoor temperature prevails compared to the current standard buildings. Operation times are reduced by 25-35%.

Tab. 25: Annual specific heating demand (kWh/(m²a)) and the gross power demand (kW) at DOT for space heating and heating of ventilation air for a single-family house, a block of flats and a nursery home in Oslo climate (Smedegård and Stene, 2013)

Oslo climate	**Prevailing building code – TEK10**			**Passive house standard (NS3701)**		
	kWh/(m²a)	**W/m²**	**h/a**	**kWh/(m²a)**	**W/m²**	**h/a**
Single-family house	59	36	1,640	21 (-64%)	19 (-47%)	1,110
Block of flats	45	30	1,500	15 (-67%)	15 (-50%)	1,000
Nursery home	59	65	910	20 (-66%)	30 (-53%)	670

5.2 Modelling of CO_2 heat pumps for passive houses

Improvements on a residential air-to-water heat pump system have been simulated. The system utilises carbon dioxide (CO_2, R744) as the working fluid, and has been especially designed for the next generation of low-energy and passive houses.
The heat pump system is equipped with an unique tripartite gas cooler (Stene, 2004) – i.e. three gas coolers connected in series – which gives off heat for preheating of domestic hot water (DHW), floor heating and reheating of DHW. The tripartite design enables heating of DHW to a relatively high temperature while maintaining a high COP in space heating mode, DHW mode and so-called combined mode, i.e. simultaneous space heating and DHW heating. The heat pump system is also equipped with thermal energy storage (ice storage tank). The purpose of the thermal energy storage is to improve the efficiency of the system at low ambient temperatures by providing a stable temperature of 0 °C on the evaporator side of the system.

It was expected that the thermal energy storage for generation of ice (heat source) and melting of ice (heat sink) would improve the Seasonal Performance Factor (SPF) of the system compared to a conventional air-source heat pump. However, this was proven not to be right.

5.2.1 Modelling

The CO_2 heat pump system with a thermal energy (ice) storage tank has been modelled and simulated in Modelica (Modelica Association 2013).
The main goal of the simulations was to calculate the impact of the thermal energy storage on the overall Seasonal Performance Factor (SPF) of the heat pump system at varying ambient air temperatures (heat source temperature).

Heat Pump Design

For the simulations the following components/systems were used:

1. *Compressor* – Constant volumetric and isentropic efficiency of 0.7 with 0.017 litre displacement volume and a constant rotational speed
2. *Tripartite gas cooler* – 3 CO_2 gas coolers (water/CO_2) connected in series on the CO_2 high- pressure side (GC1, GC2, GC3). The capacity of the three CO_2 gas coolers depends on the number of plates for preheating DHW, space heating, reheating of DHW.
3. *Expansion valves* – Type PMV which can be controlled in 50 steps (semi-continuous).
4. *Evaporator* – Air-cooled type, not further specified. For the simulation it was assumed that the unit was sufficiently designed for the required capacity.
5. *Thermal energy storage* – Water/ice tank with integrated copper tubes (coil) and steel mantle, storage capacity not defined. The energy storage capacity was determined by the latent heat by freezing of water at constant temperature/pressure.
6. *Domestic hot water (DHW) system* – 60 to 70 °C water temperature
7. *Space heating system* – Floor heating system operating at 40/30 °C

Control Systems

- The expansion valve controls the high-side (gas cooler) pressure and with that the gas cooler performance and the required input power to the compressor. The valve is opened or closed in order to adjust the high-side pressure according to a set-point mainly determined by the ambient air (heat source) temperature.
- The variable speed compressor (inverter controlled) adapts the heating capacity to the momentary heating demand of the house – space heating and DHW heating.

Operating Modes

The heat pump system was operated in 3 different operating modes, depending on the heating demand during day and night operation as well as the availability of the heat source (ambient air). For each operating mode the following components were used in the simulations:

1. *Mode_A* – components:

- Evaporator – thermal energy storage (ice storage) on low-pressure CO_2 side
- Gas cooler (GC2) – floor heating on high-pressure CO_2 side
- Compressor
- Expansion valve

In "Mode_A" the hydronic floor heating system was heated by gas cooler no. 2. The evaporation energy was taken from the water in the energy storage which was cooled down and frozen. It was assumed that there was no heat transfer between the water and the house (i.e. zero heat loss). The supply/return water temperature of the floor heating system were 40°C/ 30 °C, and the flow was controlled by the mass flow controller for the water pump.

2. Mode_B – components:

- Evaporator – heat exchanger with ambient air on the low-pressure side
- Gas cooler (GC2) – floor heating on high-pressure CO_2 side
- Compressor
- Expansion valve

In "Mode_B" the hydronic floor heating system was heated by gas cooler no. 2 (same conditions as in "Mode_A"), but the evaporation energy was supplied from the ambient air via the evaporator. The ambient air conditions were assumed stable for all simulations.

3. "Day" – components:

- Gas coolers in series on high-pressure CO_2 side for:
  - Reheating of DHW (GC1)
  - Floor heating (GC2)
  - Preheating of DHW (GC3)
  - Energy storage for re-heating
- Evaporator – heat exchanger with ambient air on the low-pressure CO_2 side
- Compressor
- Expansion valve

In this mode the air-to-water CO_2 heat pump was used to supply heat for preheating of DHW (GC3, preheating tank) and reheating of DHW (GC1, main tank) as well as floor heating (Combined mode). When the energy storage was full of ice it was used as a gas-cooler in this mode in order to melt the ice. When all ice was melted the water temperature in the energy storage was increased until it reached equilibrium with the inlet of the preheating DHW (10 °C). When this equilibrium was reached the energy storage was defined as "thermodynamically inactive".

Basically the "Modes A and B" are used when the floor is heated. The "Day" mode is used when DHW and the floor are heated and the heat is taken from the ambient air. Principle sketches of the heat pump system with main components for the three operating modes are provided for the simulations.

Limitations

The simulations were based on several assumptions which were not further verified:

- The refrigerant (CO_2) charge of the system also depends on the tube volume between the different components. These volume elements can also act as storage units and CO_2 can be locked in certain tubes when the operating conditions require a different connection. By changing from "Day" mode to "Mode A and B" the system was physically changed regarding tube length etc. This effect was not included in the simulations.
- The thermodynamic behaviour of the ice tank was not included in the four first simulations. The growing ice layer thickness will change the overall heat transfer coefficient and thermal resistance, which will directly influence the amount of transferred heat and the evaporation temperature.
  For this special situation an appropriate tank model should be developed and included in the process simulation model in order to give a more detailed system analysis. In the current simulation an average (constant) heat load was used for the ice tank.
- The values for the PID controllers for the compressor and the expansion valve were only estimates. By choosing different controller values the dynamic behaviour of the heat pump system can be changed significantly.

5.2.2 Simulation results

For the simulation the models were stepwise applied, starting from a simplified model with only the floor heating (depicted in Fig. 33 a) as baseline model.

- The baseline model reached a COP=2 of the heat pump due to little cooling of the gas cooler gas by the temperature combination 40 °C/30 °C supply/return temperature for the floor heating. Moreover, a lot of liquid entered the CO_2 compressor which would not be possible in reality.
- As next refinement step a low pressure receiver (LPR) was integrated, which reduced the liquid in the compressor as shown in Fig. 33 b. The receiver acted as a controller on the low pressure side and secured pure vapour in the compressor. The simulated COP of the variant was about 3.5

- As next variant the DHW operation was simulated with the LPR, at almost full load conditions. The resulting COP was evaluated as 3.1
- Adding an internal heat exchanger as depicted in Fig. 33 c increased the COP in DHW operation mode to 3.4, since the gas cooler could reject more heat. At DHW temperatures of 75°C the COP was still as high as 3.1.

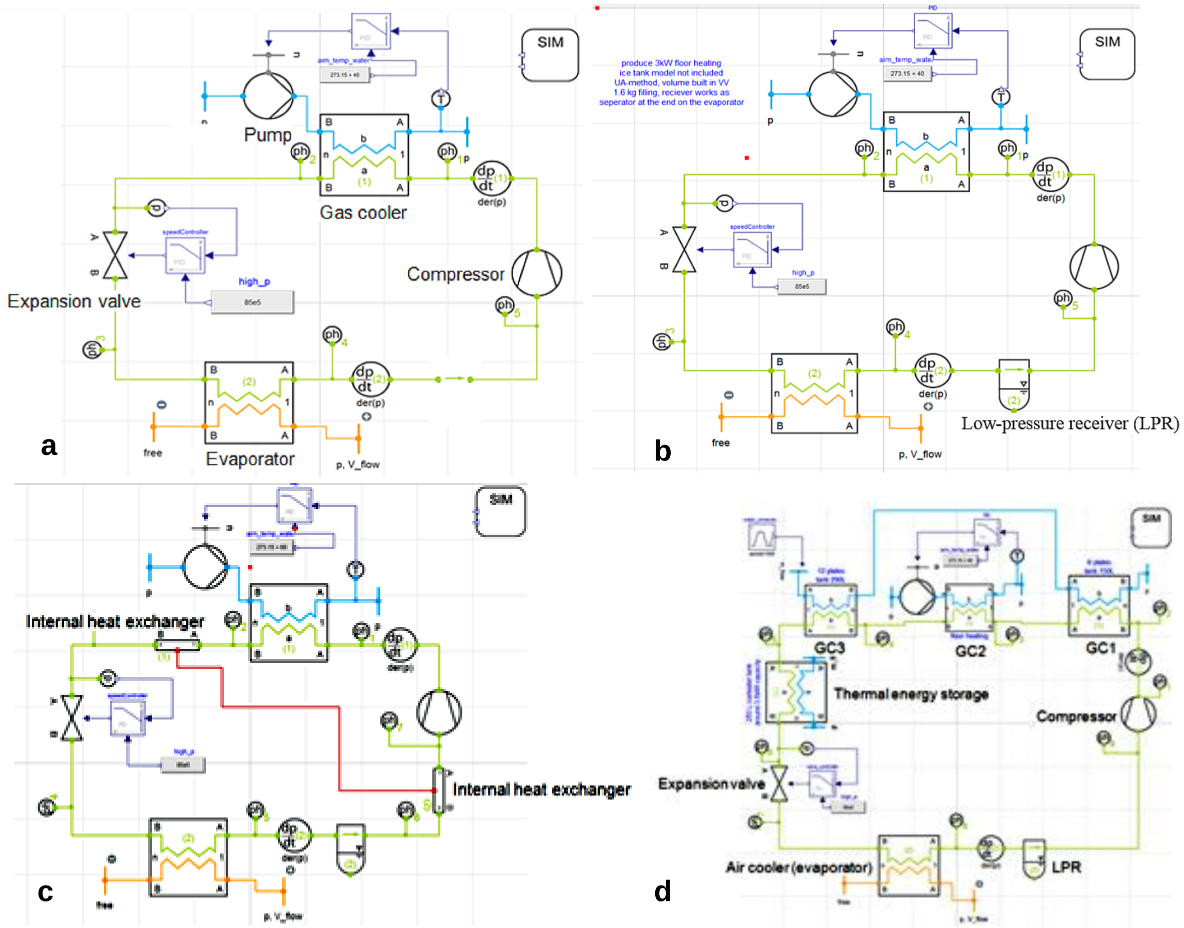


Fig. 33: Different variants of the simplified layout of the CO_2 heat pump model with all components including thermal energy storage and air cooler (evaporator) at simulation ("Day"-mode) with melting of ice in the storage (left) and internal heat exchanger (right)

- As next step the baseline models was taken and amended by the thermal energy storage, namely the ice storage which is shown in Figure Fig. 33 d. Adding an internal heat exchanger as depicted in Fig. 33 c increased the COP in DHW operation mode to 3.4, since the gas cooler could reject more heat.

<u>COP comparison</u>

When the gas cooler for melting of ice in the thermal energy storage is connected after the gas coolers for floor heating and preheating/reheating of DHW, the refrigerant is further cooled down in the energy-storage tank.
As a consequence, the evaporation temperature after expansion is lower than when using a thermal energy storage. This is illustrated in Fig. 34 where both CO_2 heat pump cycles are compared in a log-p-h diagram at an ambient air temperature of 0 °C. It can be seen that the cycle with thermal energy storage operates at a lower evaporation pressure and temperature. Hence, the compressor must operate at a higher pressure ratio which increases the input power demand to the compressor and reduces the COP of the heat pump system. As long as the ice in the energy storage is melted (during the first 6 hours) the heat pump cycle has a reduced efficiency. After 6 hours the energy storage is melted and no energy can be transferred from the refrigerant.

This means that the energy storage is no longer thermodynamically active and the both simulations reflect the same situation. This proves that the COP during the last 6 hours is almost identical for both cases.

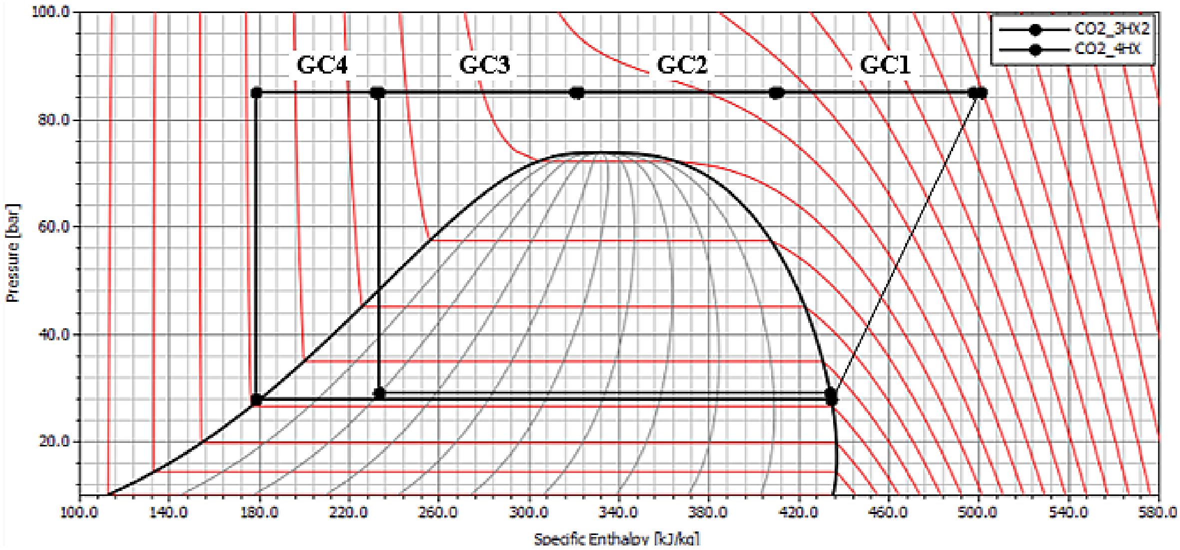


Fig. 34: Comparison of the CO_2 heat pump cycle in the log-p-h diagram with thermal energy storage (simulation 4, "CO2_4HX") and without thermal energy storage (simulation 5, "CO2_3HX"). Amb. air temperature of 0 °C.

5.2.3 Conclusion of simulation results

A unique CO_2 heat pump system comprising an air-cooled evaporator, a low-pressure receiver, an internal heat exchanger, a variable speed compressor, three CO_2 gas coolers for space heating and DHW heating, an expansion valve and a thermal energy storage (ice storage tank) with an integral gas cooler has been successfully modelled and simulated in Modelica.
The relatively complex CO_2 heat pump unit will have a higher investment cost, require a more advanced control/monitoring system and need more maintenance than standard CO_2 heat pumps. The Modelica simulations have shown that the heating capacity and COP of the CO_2 heat pump system drops when the temperature of the ambient air is reduced, due to the lower evaporation temperature and pressure. However, the relative drop in heating capacity and COP is less than that of heat pumps with HFCs (R410A, R407C, R134a) and HCs (R290, propane) refrigerants. Tab. 26 shows the COP in dependence on the ambient temperature.

Tab. 26: COP, heating capacity [W] and evaporation temperature [°C] for the modelled CO_2 heat pump dependent on ambient temperature (10°C to -20°C)

Ambient air temp. [°C]	COP	Heating capacity [W]	Evap. temp. [°C]
10	4.4	3,480	6.9
5	4.0	3,315	1.9
0	3.6	3,100	-2.8
-5	3.3	2,850	-7.7
-10	3.0	2,600	-12.5
-15	2.8	2,350	-17.3
-20	2.4	1,850	-21.7

The idea behind the application of a thermal energy storage is that it would enable a higher evaporation temperature at ambient temperatures below approximately -5°C, which should improve the overall COP of the system. It is important to notice that the thermal energy (heat) transferred into and from the thermal energy storage is never utilised in the heat pump or any other system. It is simply "pumped" from the high-pressure side to the low-pressure side and back again. The transferred heat is not utilised in a thermodynamic way by the surrounding system.

The simulations in Modelica have shown that the implementation of a thermal energy storage only gives a minor COP improvement at ambient air temperature below approximately -10°C. Above this temperature the system is more energy-efficient without a thermal energy storage. In Norway, the number of days where the ambient air temperature is below -10°C are quite limited, and *the thermal energy storage will therefore not improve the SPF of the system*.

The thermal energy storage poses a significant challenge from an engineering point of view – protection against high pressure up to 100 bar, several valves to be controlled, more tubing, larger receiver on the low-pressure side etc. Hence, the costs can be reduced if the system is built without a thermal energy storage without significant reduction of the SPF.

The compressor efficiency is essential for the COP of the CO_2 heat pump system. For the Modelica simulation constant isentropic and volumetric efficiencies were assumed. However, real compressors have an optimum operating point, and the system COP will drop at off-design operating conditions. In future investigations with Modelica the application of real efficiency data would lead to more precise simulations. However, for the comparison of different operating modes it is an advantage not to include this aspect in the simulations in order to outline more clearly the best system solution.

5.3 Modelling and simulation of heat pumps for ZEB

At the research centre "Zero Emission Buildings" *(*www.zeb.no*)* at NTNU-SINTEF, a software modelling tool for early-stage simulation of combined heat pump and cooling systems for non-residential ZEB has been developed.

Given that the design of heat pumps in the context of ZEB could be different from standard buildings, it is important to develop simulation tools to investigate the design procedure of such systems. Therefore, a tool has been developed to analyse design procedures as well as modelling procedures. This NZEP (simulation tool for *N*early *Z*ero *E*nergy buildings heat *P*ump installations.) has the following goals:

- Improve design know-how of components
- Improve selection and dimensioning of components
- Evaluate and optimise the hydraulic layout and the control strategies

Optimisation of the system can be done in terms of CO_2-emissions, primary energy use, delivered energy use, investment cost, length of operation cycles, etc. The tool gives the possibility to analyse every component in detail, but also their interaction, including the control.

In order to develop NZEP, three projects and three master works were initiated and carried out by Leif Småland, Thomas Murer, Simon Aldebert and Mikkel Ytterhus at the Norwegian University of Science and Technology, NTNU (Småland (2012) and Småland (2013), Murer (2014), Aldebert (2015), and Ytterhus (2014), Ytterhus (2015)). The objectives for the work were as follows:

- Identification of the different energy supply strategies based on heat pumps to cover the heating and cooling loads of a highly-insulated office building (so far only in Norway): distinction between heat sources (air-water or ground-source), emission and distribution of the heating/cooling (air or hydronic), possible combination with a peak-load system, or with free cooling.
- Definition of the quantitative and qualitative criteria that would affect the selection of strategy for heating and cooling of non-residential buildings and as a result to establish the specifications and requirements for a design simulation tool.
- Development and implementation of a beta-version (i.e. proof of concept) in a suitable programming language. It has been done using Matlab/Simulink and the Carnot toolbox. The main motivations to choose this language are its flexibility to quickly test different algorithms, and the fact that the pre- and post-processing can be directly performed in Matlab while user-friendly optimisation routines are already available, for example in the Matlab optimisation toolbox. Nevertheless, it was well understood that the computational cost will not be optimal.
- Definition of the default values for the mandatory input parameters.

- Demonstration of the early-stage decision making tool for heat pumps in ZEB. A consistent test case of a passive office building was used in order to test and illustrate the potential of the developed software. As the main assumption motivating this work was that the design of heat pumps in the context of ZEB could be different than with standard buildings, the test case was used to check this basis assumption including:
    - Influence of the ratio between the base and peak loads in ZEB
    - Influence of the peak load system selection in the context of ZEB
    - Influence of the accumulation tank sizing
    - Influence of the control systems
- Accomplishment of a sensitivity analysis, i.e. a robustness analysis, of the optimum ZEB concept with regard to uncertainties in the input parameters.

5.3.1 Model Specification

The main purpose of NZEP was to find the optimum design and combination of heat pump and peak power systems which gives the best economic performance (e.g. lowest annual costs [€/year]), highest energy efficiency, acceptable heat pump cycle duration and lowest environmental impact (e.g. here assessed using the net yearly CO_2 emissions for the system operation)

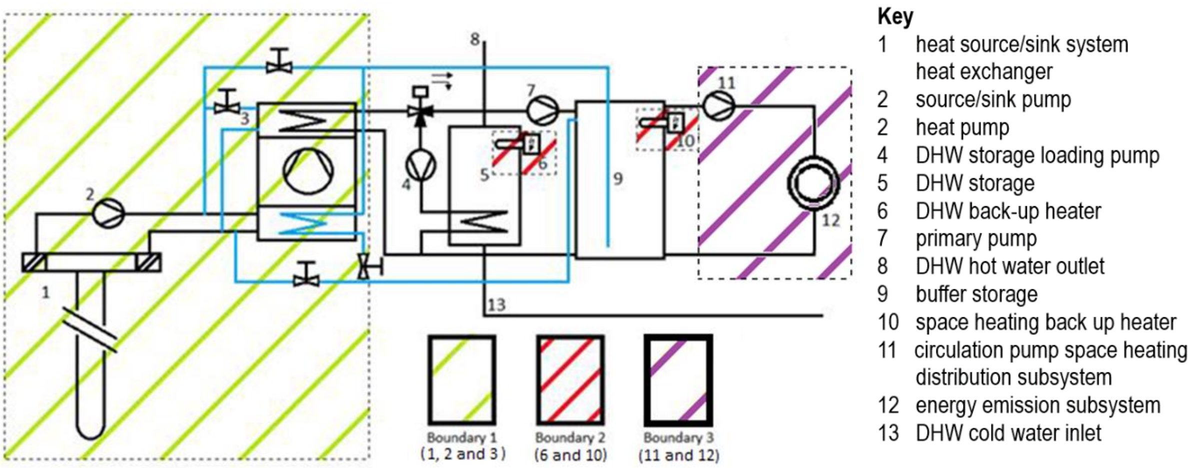


Fig. 35: Principle design of the modelled system: Boundary 1 – heat pump with heat source/sink, Boundary 2 – peak load system, Boundary 3 – DHW system and a heating/cooling system (Småland, 2013).

Fig. 35 shows the main layout of the modelled thermal system. The system comprises a heat pump unit connected to a heat source system, a peak load (heating) system, a space-heating and space cooling system as well as DHW storage tanks.
The model is currently limited to the following system design:

1. Heat pump: a) Air-source (ASHP) or b) Ground-source (GSHP)
2. Peak load: a) Electric boiler, b) Gas-fired boiler or c) Bio-fired boiler
3. Space heating, domestic hot water and cooling storage tanks.
4. In the future NZEP development, distribution and emission subsystem are to be improved regarding use of: a) high-temperature radiators, b) low-temperature radiators or c) floor heating system, d) ventilation cooling batteries, e) chilled beams or f) floor cooling system. With this version the distribution and emission systems are considered in the pre-processing that is done when the building is modelled using SIMIEN.

The configuration of each overall system layout was made by selecting and connecting the relevant subsystem, e.g. ASHP (1.a) + Gas-fired boiler (2.b) to cover the demands simulated in the building simulation program. The building thermal dynamics are not simulated in the software, but are rather taken as an input (i.e. using a so-called decoupled approach). It can be considered as an acceptable limitation at the current stage of the tool development as buffer tanks for space-heating, cooling and DHW are introduced. Furthermore, the idea is also to be able to combine the program with any building simulation software (typically SIMIEN, IDA-ICE or TRNSYS for the Norwegian context).

Input data and structure of NZEP

The main input data to the model are:

- Building simulation data: net space-heating and cooling nominal power and hourly energy demands (i.e. assuming an ideal heating system)
- Domestic hot water profile
- Building specific weather data,
- Default parameter values for the model components,
- Input parameters to evaluate the economic and environmental performances, such as the electricity price and the yearly CO_2-eq.-emission factor for the electricity delivered by the grid.

During the first step, the building envelope is simulated assuming idealized heating and cooling systems. The building under investigation can be designed according to the prevailing Norwegian building code (TEK10), the passive house standard (NS3701) or the ZEB-standard.

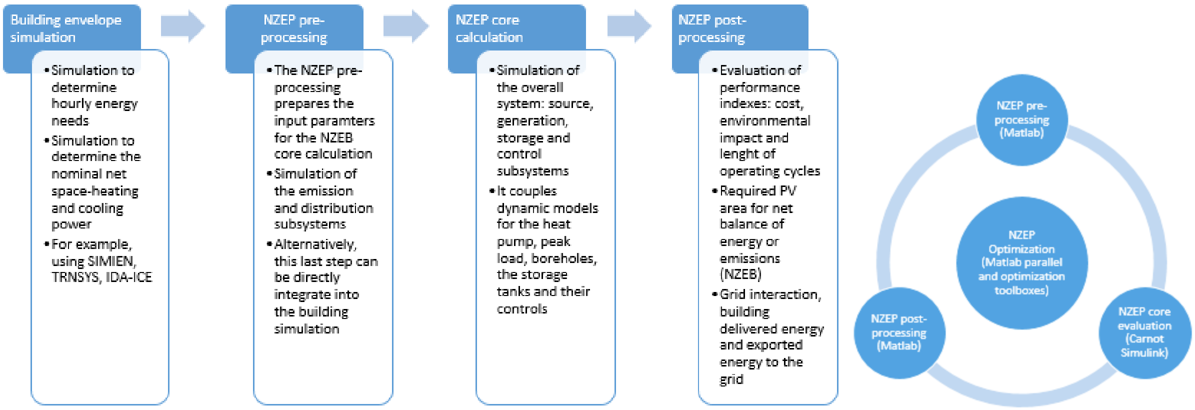


Fig. 36: The overall algorithm for the NZEP simulation tool and optimisation process by iterating on the three calculation phases: pre-processing, core evaluation and post-processing, of which each step can be done in Matlab environment

It is simulated in order to find the nominal power requirements at design conditions, as well as to evaluate hourly energy needs during a typical yearly operation (meaning 8760 hourly values). This includes the space heating, the space cooling, the heating and cooling of the ventilation air as well as the domestic hot water heating. So far, building envelope simulations have been done using the Norwegian software Simien (ProgramByggerne 2012).

Based on the demands of the building, as well as complementary user inputs (e.g. building specific and default values), the processing steps summarised in Fig. 36 are performed by NZEP:

- **Optimisation:** the optimum design for the selected system layout can be found by iterating over the main three phases of NZEP (pre-processing, core evaluation and post-processing).
  - Based on the current practice of industry, the default objective function is based on the lowest annual costs and lowest CO_2 emissions, while the typical optimisation variables are the power and energy coverage factors of the base-load heat pump system. So far, the tool has been essentially used for this type of optimisation. Although limited, it nonetheless makes sense in the context of the European NZEB definition based on cost-optimality.
    - The overall system can nonetheless be optimised using other objective functions and variables. Among other things, the heat pump control can be optimised to minimize the use of the peak-load system. For instance, domestic hot water may be produced when no space heating needs are present so that the use of the peak-load could be limited. Practically, the configuration and size of the borehole field could also be optimised. Alternatively, sensitivity and robustness analyses can also be performed. The control strategy can be also optimised to improve the building grid interaction (i.e. the flexibility to the grid).

- As the NZEP is essentially supported by Matlab, it is possible to directly use the optimisation routines readily available (within the Optimisation toolbox) and run several cases in parallel, for instance on the different cores a PC (using Parallel toolbox) (Murer, 2014).

5.3.2 Heat pump unit

As a first stage, the heat pump unit is modelled using performance curves defined by EN 14511 (2013). The heat pump is seen as a black-box: the thermodynamic cycles are not computed and the property of the heat pump components are not known (e.g. the compressor characteristics), the effect of the refrigerant choice is hidden being the heat pump performance curves. Nonetheless, the thermal mass of the heat pump is properly accounted for. This modelling is representative for many building simulation softwares.
Furthermore, it is well adapted to the limited technical data communicated by heat pump manufacturers and the data known during a design process.
By definition, the heat pump is thus modelled using performance data from standard rating conditions where the heat pump is operated at full load. In practice, the heat pump model can be run on part load, but its performance will not be adapted accordingly. At the current stage of the tool, the heat pump is operated at full load. Nevertheless, correction factors such as the *part load factor* (PLR) will be introduced in the future in order to investigate part load operation.
Finally, as regards ASHP, no proper modelling of the evaporator de-frosting process is done.
As a second step, the physics of heat pumps could be better captured by simulating the dynamics of the internal heat pump components (i.e. a grey-box model). Different approaches exists such as a coupling with the TIL library running on Modelica, coupled to Simulink using a Function Mock-up Unit (FMU). It could be useful if advanced heat pump cycles are considered, such as CO_2 heat pumps.

5.3.3 Conclusion

The tool already proves to be able to simulate the example building with specified cooling, heating and DHW demands by use of different heat pumps, different back up heaters and control systems. Optimisation of the system and their interaction can be done in terms of CO_2 emissions, primary energy use, energy use, cost, length of operation cycles, etc. The tool gives the possibility to analyse every component in detail but also their interaction, including the control.
The initial test simulations showed that the modelling concept represents a useful tool for calculation of optimum power and energy coverage factors, profitability, annual CO_2 emissions and PV requirements for a range of air-to-air, air-to-water and brine-to-water heat pump systems with different peak load systems. I.e. the modelling tool can be employed for early-stage overall design of heat pump and cooling systems in non-residential ZEB. This version of the tool decouples the building and the heating /cooling system. This proposes a limitation on the possibilities of controlling the building envelope as a function of the grid interaction.
Therefore, further development, refinement and optimisation of the computer tool is required before it becomes a reliable, fail-proof decision-making tool.

6 Case studies of nZEB technologies in Sweden

Heat pumps are considered as a promising building technology for nearly Zero Energy Buildings in Sweden. Nevertheless, in the beginning of Annex 40, no ground source heat pumps in the capacity range of single family houses with very low heat demands were available on the Swedish market. On the other hand, the heat pumps for multi-family buildings were mainly developed for high temperature heating and less optimised for the production of domestic hot water. The aim of the Swedish national project was therefore the development of competitive heat pumps for the Swedish and export market for the application in nZEB. Moreover, it should be shown by field-monitoring, how an nZEB in Sweden can be realised most energy-efficiently and cost-effectively using heat pumps in combination with other heat generators and energy-efficient building envelope technologies. The system boundary for the nZEB evaluation comprises both, energy for the HVAC technologies as well as plug loads for appliances.
As a part of the Swedish project a ground source heat pump adapted for very low heat loads of single family houses has been developed and put on the market. During the project time also two other small ground source heat pumps have been put on the market by Swedish heat pump manufacturers. In collaboration with a Swedish heat pump manufacturer an improved system for the production of hot tap water in multi-family buildings has been developed and tested in the laboratory. The design targets for the heat pump development have been to pass the requirements for nZEB, Fgas, Eco Design and RES directives. Another goal has been to make ground-source heat pumps a cost-effective alternative for low energy single-family houses. The developed heat pump for the single-family house has been tested in real houses in 2014-2015. Furthermore, calculation models for nZEB have been developed, and linked to the assessment process of the heat pump prototypes.
In another nZEB-related project initiated by the Swedish Energy Agency and the National Board of Housing, Building and Planning, monitoring has started in more than 20 Swedish nZE houses. Heat pumps are installed in several of these monitored houses.

6.1 Building and system configurations for the case study

The type houses were developed by the project group of SP together with the industry partners consisting of single-family house manufacturers, multi-family house manufacturers and heat pump manufacturers. The specification of the type houses is given in Appendix A.4.

6.1.1 Investigated system solutions

In an iterative process together with the whole project group the type houses were chosen to have the following system solutions for both the single-family house and the multi-family house.

- Heat source: vertical ground source (with borehole)
  Due to the requirement that the heat source should be applicable for the entire heat load and should work in all Swedish climate zones, the choice was made for a vertical borehole ground source.
- Heat recovery: mechanical supply and exhaust air with ventilation heat recovery by heat exchanger.
  Due to commonly high air tightness of new buildings, a mechanical ventilation was included to secure indoor air quality. Even though active heat recovery by exhaust air heat pumps are common in new Swedish houses, this option was not chosen for two reasons: On one hand, it is known from experience that the energy efficiency requirements are hard to meet in exhaust air ventilated houses. On the other hand, a balanced ventilation in house reacts with a higher inertia to intermittent heating which has advantages in future smart grid applications.
- Heat emission system: floor heating
  The floor heating emission system is chosen both, for efficiency reasons due to the low supply temperature and for aesthetical reasons since no visible heating surfaces in the room exist.

Tab. 27: Heat pump characteristic as input for the energy use assessment of the SFH and MFH

Heat pump single-family house		Heat pump multi-family house	
P HP heat, 0/35°C	4750 W	P HP heat, 0/35°C	90980 W
COP, heat, 0/35°C	4.24	COP, heat, 0/35°C	4.60
P HP heat, 0/45°C	4500 W	P HP heat, 0/45°C	87040 W
COP, heat, 0/45°C	3.28	COP, heat, 0/45°C	3.66
Desuperheater	No	Desuperheater	No
A-labelled brine pump	Yes	A-labelled brine pump	Yes
Standby power consumption	55 W	Standby power consumption	1201.6 W
Installed electric power	4500 W	Installed electric power	35880 W

The characteristics of the developed heat pump prototypes in the project are presented in Tab. 27 and were used as input to the assessment of specific energy use using the software TMF Energy. The type houses were determined to be situated in climate zone 3 in Stockholm according to the climate zones determined in the Swedish Building Regulations 2015 (BFS 2015:3 BBR 22). The outdoor design temperature used in the project are given in Tab. 28.

Tab. 28: Outdoor design temperature for different time constants of the building (SS-EN ISO 15927-5)

Stockholm	1 day	2 days	3 days	4 days
°C	-17.8	-17.0	-16.3	-15.8

The balance boundary for the nZEB prototype houses does not include the electricity for plug loads of appliances and limits are set for the delivered energy. Tab. 29 gives a proposal of the Swedish Energy Agency for limits on delivered energy for non-electrically and electrically heated houses, which also include heat pumps.

Tab. 29: Proposed maximum specific energy consumption (Swedish Energy Agency 2010)

	Non electrically heated [kWh/(m²a)]			Electrically heated [kWh/(m²a)]		
Building Type	**Climatic zone***					
	I	**II**	**III**	**I**	**II**	**III**
Residential**	75	65	55	50	40	30

*) In 2010 there were only three climatic zones in the Swedish building regulations. Climatic zone III consisted then of the present climatic zones III and IV.

**) In 2010 the same requirements are applied for single-family and multi-family residential building. In the present BBR 22 the requirements are slightly more stringent for multi-family buildings.

6.1.2 Energy calculations

The energy calculations in the project were accomplished with TMF Energy, which is an Excel based software developed by SP and TMF to calculate the energy performance of single-family dwellings according to the Swedish building regulations. It is used to make a preliminary energy declaration of specific energy use in newly built single-family houses. During the Annex 40 project a beta-version adapted for multi-family houses has been developed. The calculation method is based on the European standard EN ISO 13790:2008, but with national parameters for usage (Levin et al., 2007). It uses a stationary heat balance model and an approximate duration curve based on the mean average outdoor temperature as an input. The maximum electric power demand for “electrically heated houses” is calculated based on the winter outdoor design temperature corresponding to the time constant of the house, see Tab. 28.

6.1.3 LCC-analysis

An LCC-analysis was performed for the developed system prototypes together with the type houses developed in the project. Tab. 30 shows the calculation conditions for the LCC-analysis.

Tab. 30: Calculation conditions for the LCC-analysis

	Variable	Value	Unit
Calculation conditions	Discount rate	4.00	%
	Time of the cash flow	30.00	a
	Mean electricity price	1.46	SEK/kWh
	Mean price for DH	0.67	SEK/kWh
	Mean pellet price	0.55	SEK/kWh
	Increase in electricity price per year	3.00	%
	Increase in heat price per year	2.00	%
	Increase in cost for pellet per year	2.00	%

In Tab. 31 and Tab. 32, the assumed investment costs and lifetime expectancy used in the LCC calculations for single-family respectively for multi-family houses are listed.

Tab. 31: Investment cost and lifetime expectation for the single -family house

	Investment (what)	Investment [SEK]	Life expectancy [a]
Single Family House	GSHP	73,500	15
	Borehole	37,500	75
	ASHP	80,800	15
	District heating sub station	20,600	20
	Pellet boiler	43,750	20
	Solar thermal collector + accumulator tank	56,250	20
	Furnace	18,750	20
	Piping etc.	6,250	30
	PV-installation, total per kW_p	20,000	
	PV-module, per kW_p	12,000	25
	Inverter, per kW_p	2,000	15
	PV-modules: additional costs, per kW_p	6,000	25
	PV-installation: subsidy, per kW_p	-7,000	25

Tab. 32: Investment cost and lifetime expectation for the multi-family house

	Investment (what)	Investment [SEK]	Life expectancy [a]
Multi Family House	GSHP	440,000	15
	Borehole	308,000	75
	District heating sub station	60,000	20
	Solar thermal collector + accumulator tank	100,000	20
	PV-installation, total per kW_p	20,000	
	PV-module, per kW_p	12,000	25
	Inverter, per kW_p	2,000	15
	PV-modules: additional costs, per kW_p	6,000	25
	PV-installation: subsidy, per kW_p	7,000	25

6.1.4 LCC-analysis

An LCC-analysis was performed for the developed system prototypes together with the type houses developed in the project. Tab. 31 shows the calculation conditions for the LCC-analysis. Tab. 30 (row 4-6) gives mean prices of electricity, DH and pellet. The investment cost for the pellet boiler has been assumed without a storage tank since this is the most common case in Sweden.

6.2 Results of case studies of heat pumps in nZEB

6.2.1 Performance evaluation

TMF Energy gives the specific energy use for the single-family house and for the multi-family house. The specific energy use has been assessed for four different cases. In addition to the baseline with the type house and the heat pump prototypes, three different cases with the heat pump prototype and solar PV or solar thermal collectors as well as the combination of both have been assessed. For the single-family house a solar PV installation of about 3,000 kWh and 25% reduction of the heating demand according to the levels in the Building legislations by the Swedish board of Housing is considered. The solar thermal installation has been assumed to cover 40 % of the DHW demand and 10% of the heating demand for the type house.

For the multi-family house a solar PV installation of about 40,000 kWh and 25% reduction of the heating demand according to the levels in the Building legislations by the Swedish board of Housing is considered. For the single-family house it can be seen in Fig. 37 that the specific energy use in the baseline case without solar installations is below the proposed levels in Tab. 29. It should be noted that the alternative with solar PV gives a lower specific energy use than the alternative with solar thermal collectors.

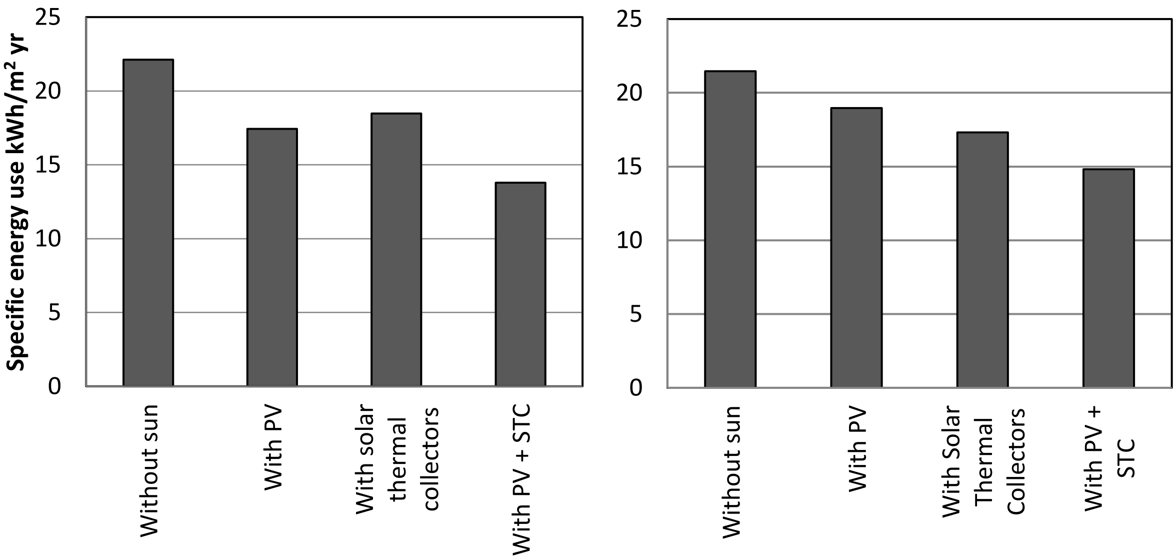


Fig. 37: LCC-analysis for the single-family house (left) and multi-family house (right) with various installations

6.2.2 Cost evaluation

From the LCC-analysis it can be seen that the net present value for the ground-source heat pump (GSHP) system prototype developed for the single family house in this project is slightly better than the air-source heat pump (ASHP) and the district heating (DH) alternatives as can be seen in Fig. 38. The alternative with pellet comes out a little bit better than the prototype GSHP alternative. For the cases with PV it has been assumed that the electricity that cannot be used in the house can be sold to the grid at the Nordpool spot electricity price plus the electricity certificate. The Nordpool spot electricity price has been calculated from statistical data from 2008 to 2012 with monthly variations. This ends up with a mean spot price at 0.43 SEK/kWh. The electricity certificate has been assumed at 0.20 SEK/kWh.

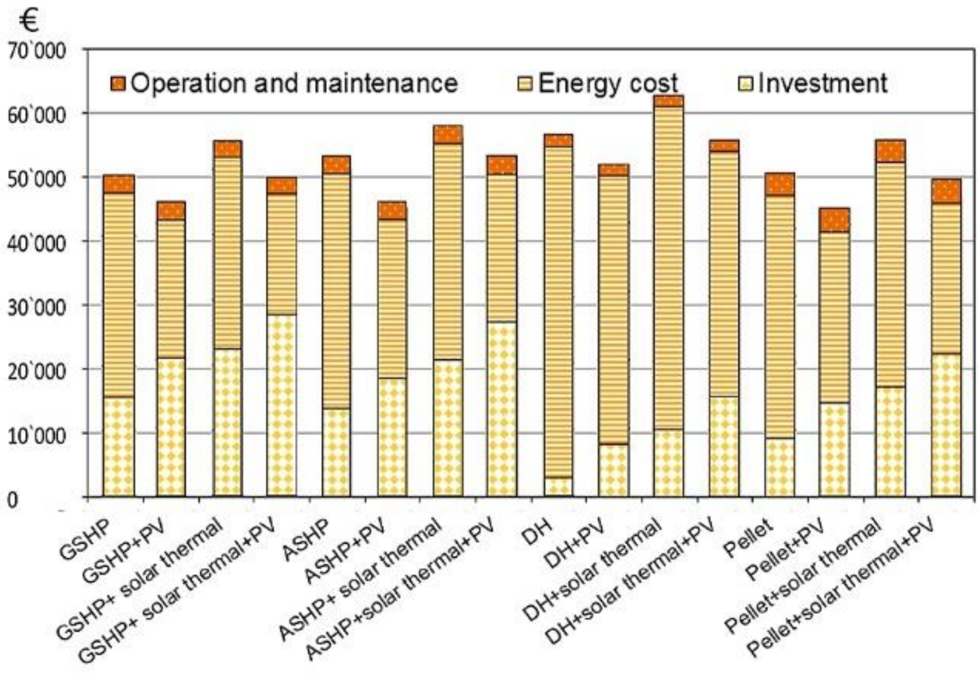


Fig. 38: LCC-analysis for the single-family house with various installations

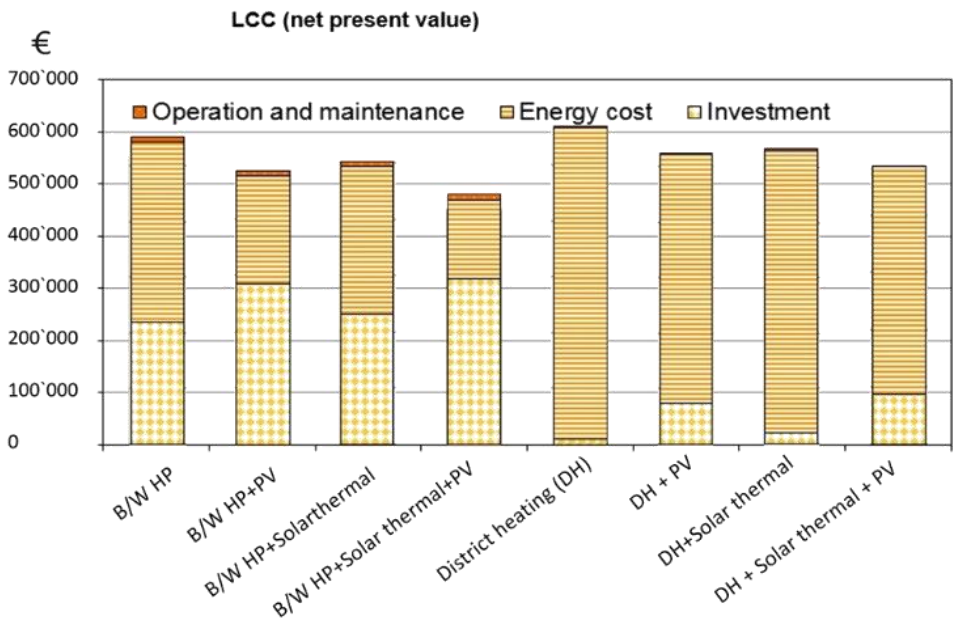


Fig. 39: LCC-analysis for the multi-family house with various installations

6.2.3 Special technologies evaluation – the solar PV-system

Household and operational electricity is not included in the system boundary of the energy requirements in the Swedish building regulations and only electricity produced by the solar PV-system that instantaneously can be used within the system boundary can be credited for the reduction of purchased energy. This means that only a rather small percentage of the electricity produced by the solar PV-system can contribute to reduce the specific energy use. For buildings with district heating, the contribution is very low as the use of electricity within the system boundary also is very low in the summertime, i.e. when most of the solar PV-electricity is produced. For houses with heat pumps the situation is somewhat better as electricity is used for domestic hot water production also in the summertime.

However, for the economic evaluation the household and operational electricity must be included in the evaluation.

6.2.4 Recommendation derived by the case studies

A PV-system can be used to reduce the specific energy use according to the Swedish building regulations, but it cannot be used to achieve a zero energy house. With a ground source heat pump and ventilation heat recovery, the proposed requirements for dwellings as near zero energy buildings in Sweden is technically possible to reach with a rather good margin.
According to the LCC-analysis there are rather small differences between different alternatives. Whether heat pumps or district heating are the most economical choice is very much dependent on differences in local energy prices, future energy prices and variations in investment costs. However, with district heating it might be more difficult to reach the proposed energy requirements.
The investment cost, including installation, of the building services system in the case study house is too high. As buyers tend to focus more on investment costs than life-cycle costs, only very stringent energy requirements and/or reduced investment costs will lead to an increased demand for houses with ground-source heat pump and ventilation heat recovery. A product where the heat pump and the ventilation system are integrated would probably facilitate introduction.

6.3 Outline of prototype developments

The new heat pump developed during the project is not very much different from what has been on the market for some time. It has energy-efficient circulation pumps and the latest control equipment. The only difference is that it has a smaller compressor. As it is still an on/off-controlled compressor it needs a buffer tank to minimise cyclic operation during spring and autumn heating season. The installation of the buffer tank on one side of the heat pump and the ventilation heat recovery unit on the other side takes a lot of space and floor area.

6.3.1 Achieved improvements

The price is slightly lower than for the larger versions of the heat pump. The main decrease in investment costs is due to the much shorter ground-source borehole. The active depth is only approximately 70 m. However, the cost for the buffer tank and installation of the system, including ventilation heat recovery, is still too high compared to the exhaust air heat pump system (that is more or less the standard in Swedish single-family houses today).
Connected to a very well insulated and airtight building equipped with ventilation heat recovery and a low temperature floor heating system on both floors, the energy performance is about 40% better, excluding reduction for electricity produced by the solar PV system. However, a standard house only would use approximately 7000 kWh/a. 40% less means only a reduction of about 2800 kWh/a. The additional investment cost for the improvement is therefore rather limited.

6.3.2 Experience of field monitoring in nZEB

The measurements on the “research house” at SP are still on-going and not yet fully evaluated. Preliminary data shows that the energy performance is close to the 22 kWh/(m^2a) that has been estimated in calculations.

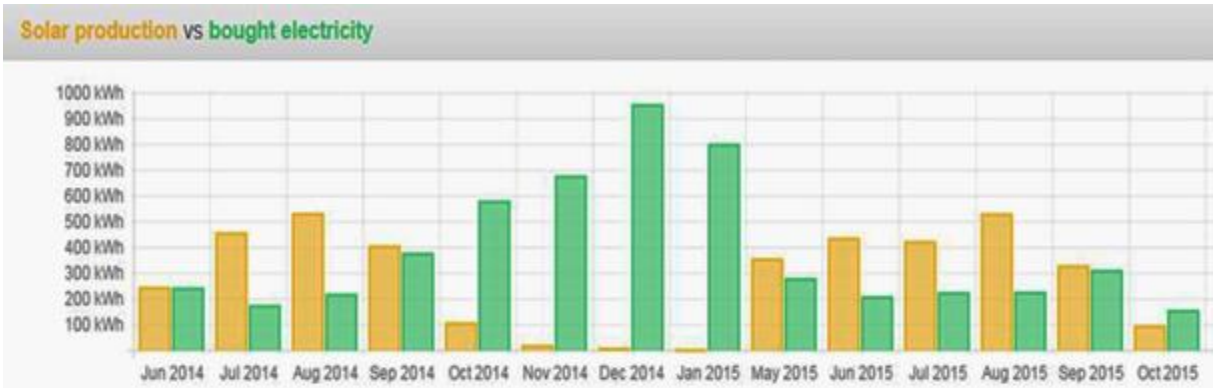


Fig. 40: PV Solar production vs. total bought energy for some evaluated months.

The solar PV system has produced about 3200 kWh/a, which is slightly more than the estimated 3100 kWh/a.
Fig. 40 shows the solar PV production vs. total bought energy for some evaluated months. On a yearly base the PV-system produces a large proportion of the total energy need. However, there is a great mismatch between production and energy need as can be seen.

6.4 Conclusions and outlook

The relatively low heating demand of single-family nZEB leads to relatively low power requirements of the heat pump. This means that an over-dimensioned heat pump will start and stop quite often and deliver unnecessarily high outlet water temperatures when applied in nZEB. The challenge is to construct a heat pump small enough to prevent cycling, but at the same time cheap enough to be cost-effective, when only delivering about 6000 kWh per year. One way of solving this is to go for a variable speed controlled compressor that can handle the low heating demand, but this might be too expensive leading to a solution that is not cost-effective.
Another solution is to combine a slightly larger heat pump with a large tank between the heat pump and the heat distribution system allowing longer running times of the heat pump for longer lifetime of the compressor. The drawback with an extra tank are additional costs and more required floor area to be taken into consideration when comparing the LCC for the various alternatives. Moreover, the storage also increases heat losses.
Regarding the multi-family house the heat demand is large enough to relatively easy produce a cost-effective system solution which has been shown in previous research projects. In this project the focus has rather been on the development of the dimensioning criteria for the heat pump system for the multi-family house. Little research has been done on dimension criteria and user behaviour for multi-family houses. Judging from common “rules of thumb” and design criteria used when dimensioning both DHW systems and HP systems for multi-family houses, many systems are too large. This leads to too large heat losses from DHW tanks, too expensive system solutions and to an overuse of material. When comparing the 3XL and 4XL tapping cycles in the Eco Design/Energy Labelling Directive (European Parliament et al., 2009) with user behaviour from field measurements in Sweden, it is quite clear that these tapping cycles are quite over-dimensioned. Of course, one should take into account the possibility of some residents in a multi-family house taking a shower at the same time at some heavily loaded time periods each day. However, to design a DHW system for all residents in a multi-family house taking a shower at exactly the same time does not make sense neither from an energy performance point of view nor from a cost-effective point of view. This is simply due to the fact that this situation statistically occurs very seldom which has also been shown by the field measurements used in this project. This needs to be further assessed with more field measurements of user behaviours in various types of multi-family houses with various numbers of residents.
Solar PV systems can only contribute to a rather small reduction of the specific energy use according to the Swedish building regulations. However, if the conditions for the tax reduction are met, solar PV systems still seem to be economically beneficial to install.

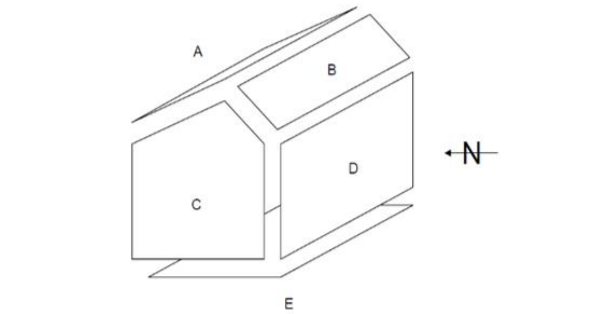
7 System assessment in Switzerland

In Switzerland, there is currently no uniform definition of a nearly zero energy building. However, in March 2011, the so-called MINERGIE-A®-label has been introduced as one implementation of the nZEB concept requiring a net zero energy balance in the balance boundary of the operational energy for the building technology. While initially only applied for the certification of residential buildings, an extension to non-residential buildings like offices and schools was introduced in May 2014.
Therefore, case studies for the MINERGIE-A® label certification have been carried out. Thereby, different standards of the building envelope (from ultra-low energy to low energy level) and different building uses (residential, office) have been considered. As heating system different heat pump types as well as biomass and fossil fuel generators and district heating have been considered. The objective of the case studies is an evaluation of adequate building technology to comply with the MINERGIE-A® label at lowest cost. The balance to reach MINERGIE-A® is met with the yield of an on-site solar PV installation. The design of the solar PV system is adapted in that way, that the MINERGIE-A® balance is just reached. In this way, it is evaluated, if the roof area is large enough to meet the requirements, or if additional installation area for the PV system in the façade is needed to generate enough energy to meet the balance criteria.
After an introduction to the boundary conditions of the different case studies and the MINERGIE-A® label, results are presented for the different building types and uses in terms of energy performance and life-cycle cost.

7.1 Systems and boundary conditions

7.1.1 Reference buildings

The single family house (SFH) is modified of a reference building used in IEA HPP Annex 38/ SHC Task 44 (Dott et al., 2013) as depicted in Fig. 41 (left). For the multi-family house (MFH) a typical layout of realised MFH in Switzerland is used. As a model for an office building, the company head office of “Marché Restaurants Switzerland” is used as example, which is described in detail in Naef and Stemmler (2010) and shown in Fig. 41 (right). The building is certified according to the MINERGIE-P-ECO®- standard.



	building	A =d*a	B =d*b	C =c*(e+r/2)	D =d*e	E =d*c
net (inside) area (m^2)	all	54.6	26.4	45.7	56.0	70.0

Fig. 41: SFH reference building of IEA HPP Annex38/SHC Task 44 (left) and Head office of Marché Restaurants Switzerland (right)

7.1.2 General Characteristics

In Europe there is a large variety of concepts and voluntary standards for highly energy efficient buildings or climate neutral buildings. Therefore, it is sometimes difficult to compare realized nZEB. In Switzerland, an nZEB definition is given for instance by the MINERGIE-A®-label. The MINERGIE-A® label considers only the balance boundary of the building technologies (i.e. the energy consumptions of equipment of space heating, DHW, ventilation and air-conditioning).

Thus, the MINERGIE-A® label is currently not considering plug loads. The weighting is accomplished by the so-called national Swiss weighting factors for the energy balancing, which are given in Tab. 33.

Tab. 33: Weighting factor of MINERGIE-A®

Energy carrier, energy source	Weighting factor
Solar and ambient heat	0
Biomass (wood and biogas)	0.7
Waste heat (incl. waste incineration and sewage treatment plants, industry)	0.6
Fossil fuels	1.0
Electricity	2.0

The balance for all concepts is evaluated for a whole year to balance the seasonal differences. For the evaluations different balance boundaries are used for residential (SFH, MFH) and office buildings. The balance for SFH and MFH only includes the building technology according to MINERGIE-A®. For the office buildings, the energy use of lighting and devices are included, as well. Tab. 34 shows the general characteristics of the different building types which are used for the simulation.

Tab. 34: General characteristics of SFH and MFH and two types of office buildings

	SFH	MFH	Office A	Office B
Energy reference area (ERA)	164 m²	1500 m²	960 m²	1680 m²
DHW	50 l/(pers.·d)	50 l/(pers.·d)	5 l/(pers.·d)	5 l/(pers.·d)
Roof area	54.6 m²	196 m²	400 m²	420 m²
Slope of roof	40°	40°	35°	35°
South façade for PV	-	312 m²	101 m²	168 m²
Orientation	south	south	south	south
Number of storey	2	5	3	5
Number of person	4	30	78	137

The energy data of the office buildings are calculated using the default values of SIA 2024 (2006). The electrical expense for the cooling demand is determined by typical EER values of SIA 382/2 (2011). In Tab. 35 the specific and annual energy demands are given.

Tab. 35: Energy requirements of the office buildings (ERA – energy reference area)

	per ERA	Office A	Office B
Air-conditioning	4.7 kWh/(m²a)	4,512 kWh/a	7,896 kWh/a
Electricity for Cooling	1.1 kWh/(m²a)	1,067 kWh/a	1,800 kWh/a
Electricity for lighting	16.3 kWh/(m²a)	15,677 kWh/a	27,434 kWh/a
Devices	12.9 kWh/(m²a)	12,384 kWh/a	21,672 kWh/a

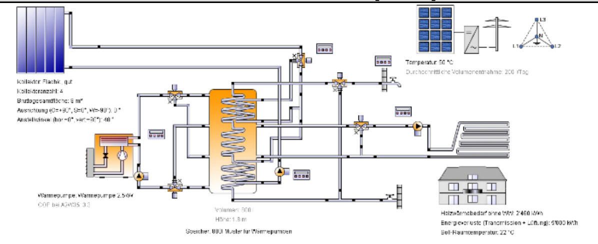
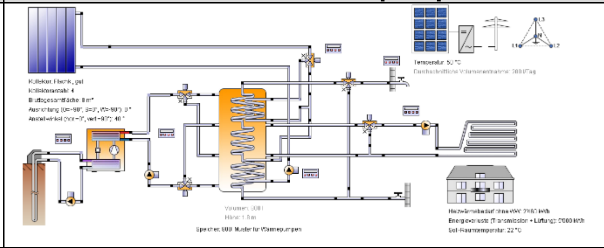
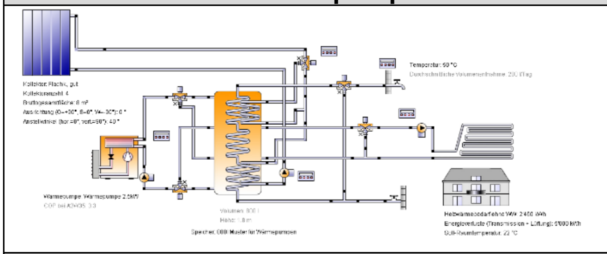
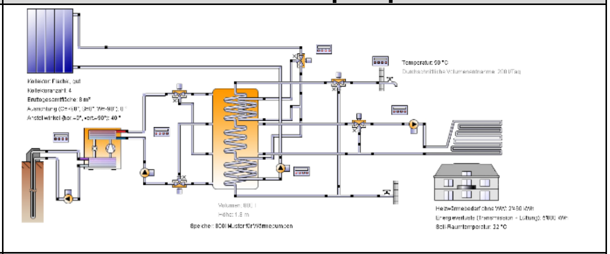
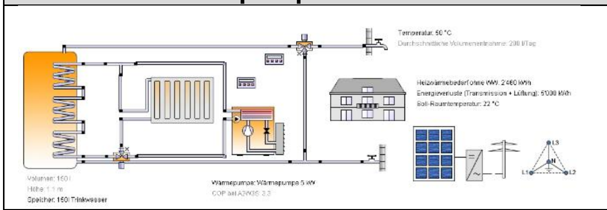
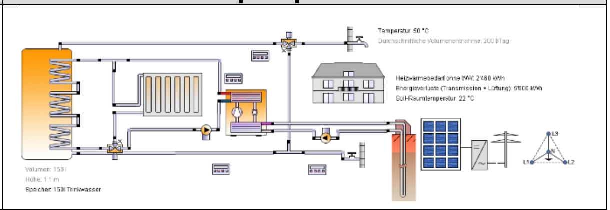
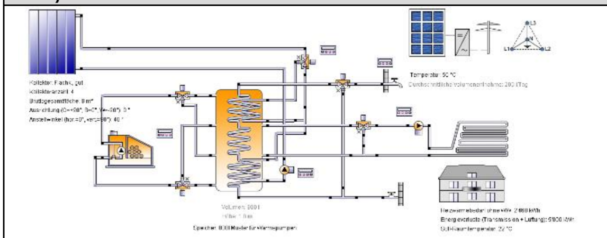
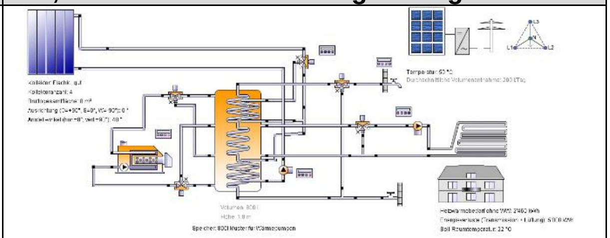
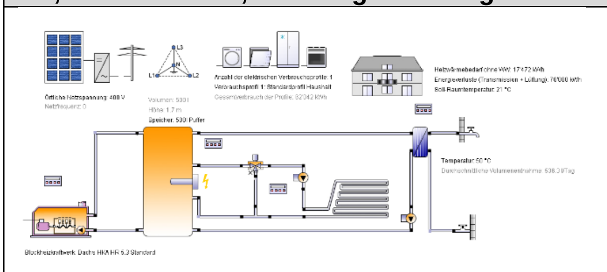
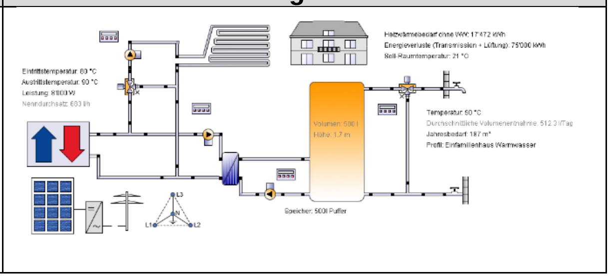
7.1.3 System simulations

The different concepts are simulated with the software Polysun® and are analysed regarding the criteria discounted annual cost [1000 €/a] and energy balance [kWh/a].

For the evaluation different insulation standards denoted as space heating energy needs of 15 kWh/(m²·a) for ultra-low energy house level denoted as MINERGIE-P® in Switzerland, 35 kWh/(m²·a) for the MINERGIE® and 55 kWh/(m²·a) for a standard new building are used to find limitations of the nZEB balance.

Different technology options and combinations for providing the space heating and domestic hot water (DHW) energy are used: air-to-water (A/W) heat pump, ground-source brine-to-water (B/W) heat pump, district heating, solar thermal collectors, co-generation of heat and power (CHP) with heating oil, biogas or natural gas and boilers with fuel oil, biomass (wood pellets), biogas or natural gas. Tab. 36 gives and overview on the simulated system configurations. In this study, no integrated system solutions are considered, but all systems are installed side-by-side.

Tab. 36: System configurations simulated with Polysun®

PV, solar heat and A/W-heat pump	PV, solar heat and B/W-heat pump
	
Solar heat and A/W-heat pump	**Solar heat and B/W-heat pump**
	
PV and A/W-heat pump	**PV and B/W-heat pump**
	
PV, solar heat and biomass	**PV, solar heat and natural gas/Biogas**
	
PV, CHP with fuel, natural gas or biogas	**PV and district heating**
	

For meeting the energy balance, photovoltaic panels are used on the roof area or in the building façade. The cost balance includes the investment cost, the operating cost and the interest rate for a uniform time period of 25 years.

7.2 Evaluation of performance

For the evaluation the weighted end energy according to the MINERGIE-A weighting factor is considered.

7.2.1 Single-family-house

The SFH concepts are simulated with a solar thermal collector area of 8 m² corresponding to a DHW design with higher solar fraction. The PV area is designed to use the remaining roof area. The concepts without PV panels installed achieve the zero energy balance with electricity purchase from renewable sources.

Fig. 42 shows the energy balance for the SFH of different insulation standards expressed as space heating needs 15/35/55 kWh/(m²·a). The energy balance depicts the surplus after energy weighting and confirms that the MINERGIE-A® balance can be reached by most of the concepts up to an insulation standard of a space heating energy need of 55 kWh/(m²·a) with only using the roof area for solar PV panels. The concepts with a B/W heat pump have the highest surplus of weighted energy.

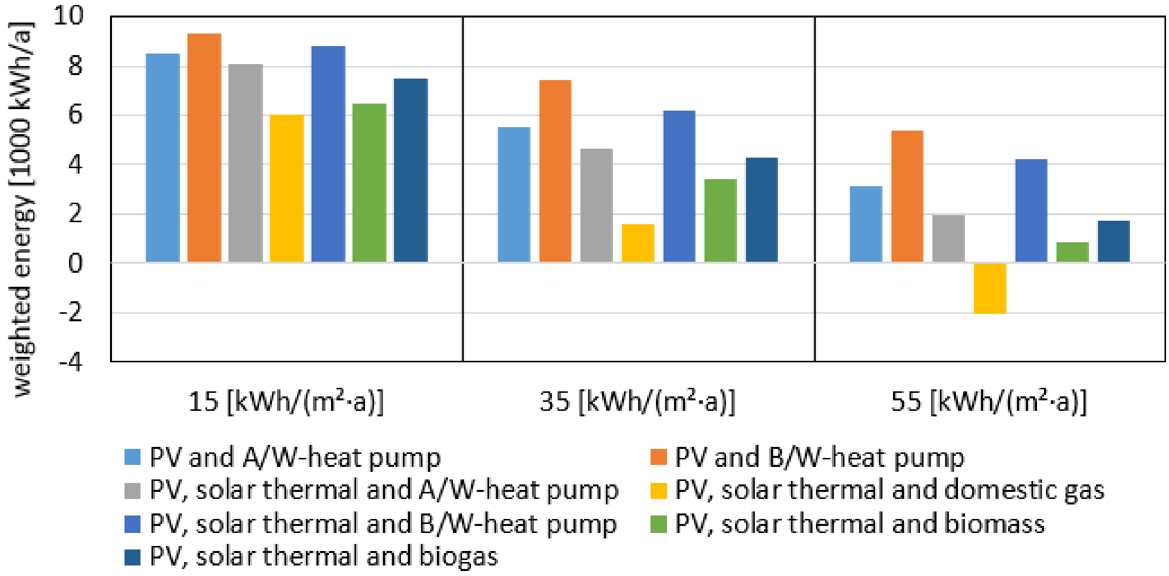


Fig. 42: Energy balance with weighting factors of MINERGIE-A® for SFH

Solar thermal collectors could not improve the energy balance of the concepts due to the competition with PV area on the restricted overall roof area. Moreover, PV has a weighting factor of 2, while solar thermal energy simply reduces the space heating or DHW needs according to the MINERGIE-A® factors. The concept with natural gas and solar thermal collectors has the worst energy balance, especially for increasing heating demands.

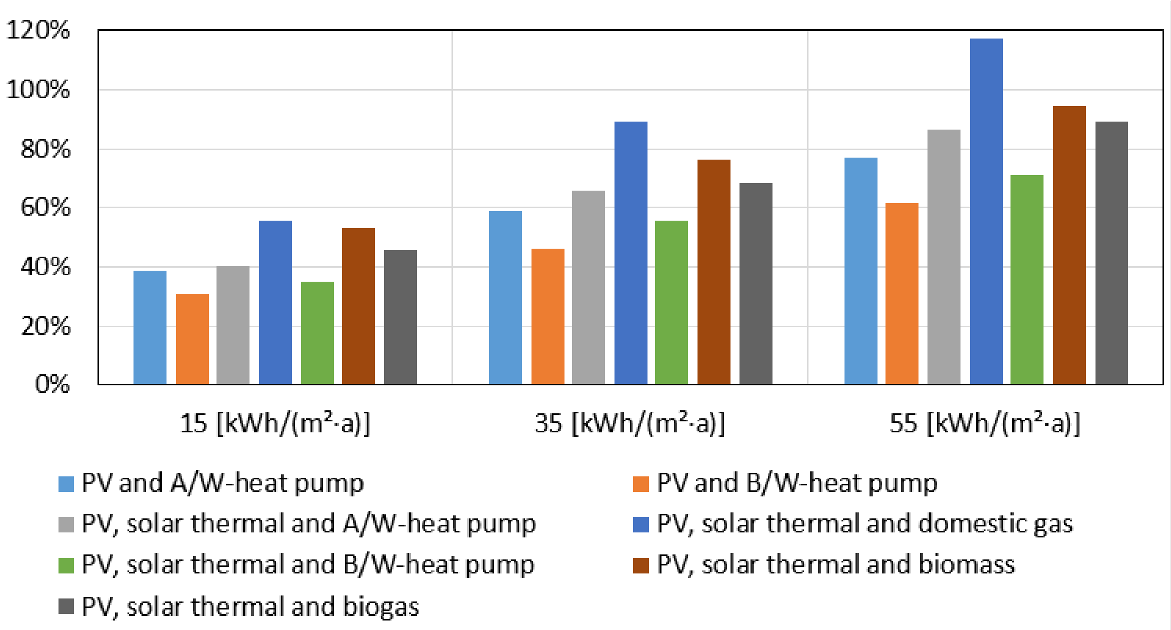


Fig. 43: Percentage of roof area to achieve MINERGIE-A®

Fig. 43 shows that with all concepts the MINERGIE-A® requirements can be achieved. Up to a heating demand of 35 kWh/(m²·a) all concepts can achieve MINERGIE-A® by only using the roof area for PV panels. For the heat pump concepts, it is still possible to reach a zero energy balance according to MINERGIE-A® with higher specific heating demand.

7.2.2 Multi-family house

In MFH for most concepts a solar thermal collector area of 50 m² is chosen corresponding to a system for DHW support. Only for the “PV-solar thermal-A/W heat pump” concept the whole roof area is used for solar thermal collectors and a seasonal storage of 50 m³ is installed (self-sufficiency for space heating and DHW). The PV area is sized to meet the MINERGIE-A® balance.

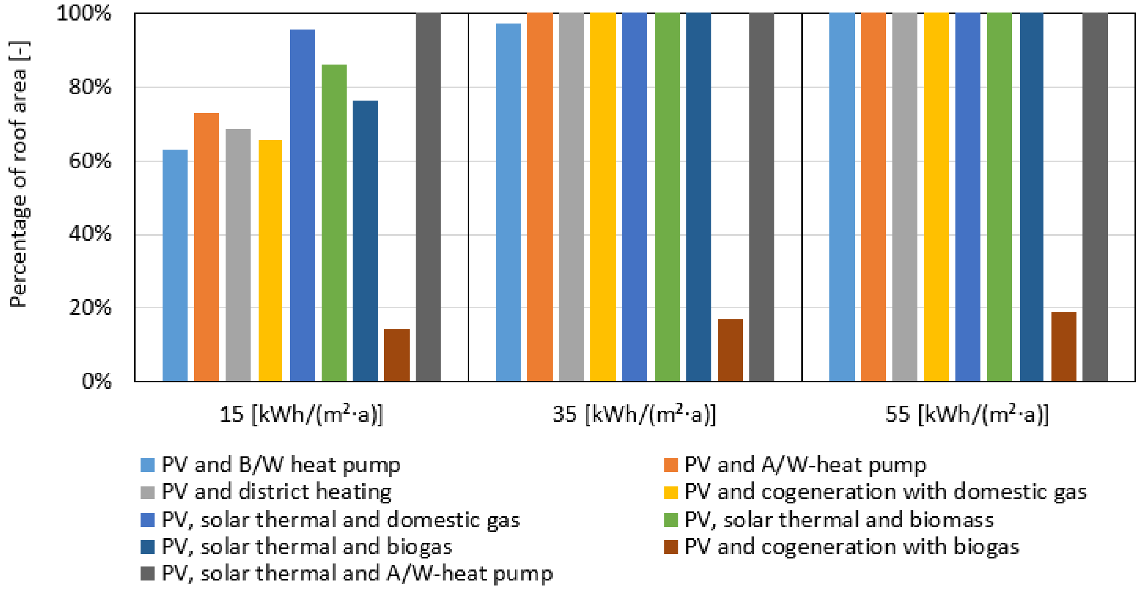


Fig. 44: Percentage of roof area to achieve MINERGIE-A®

Fig. 44 depicts the percentage of roof area needed to meet the balance. In buildings on ultra-low energy house level (15 kWh/(m²·a)) the MINERGIE-A® nZEB balance is reached by only using the roof area with all concepts. With higher specific space heating needs (35-55 kWh/(m²·a)) it is necessary to use the façade for additional PV panels. Only CHP with biogas can achieve an nZEB without using the façade area, but biogas has the highest energy cost.

7.2.3 Office buildings

For the office buildings two different buildings, one with three (Building A) and one with five (Building B) storeys were considered to find the limitations of nZEB concepts in office buildings.

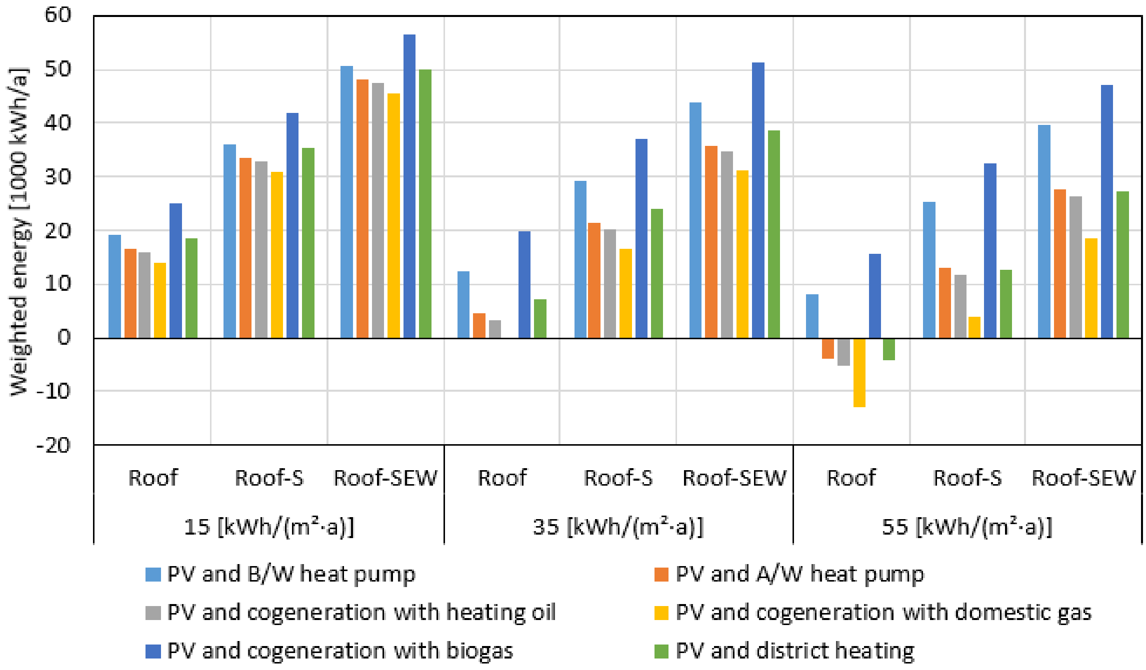


Fig. 45: Energy balance for office building A with three storeys

Besides the roof, also the façades (south (S), east (E) and west (W)) are used for the PV installations. Different to MINERGIE-A®, the building technologies and the energy demand for lighting and devices are included in the balance.
With an office building of three storeys it is possible to reach a zero energy balance with only using the roof area up to a specific heating demand of 35 kWh/(m^2a) for the considered boundary conditions. For a five-storey office building it is necessary to additionally use the façade area to reach the zero energy balance.
Fig. 45 and Fig. 46 illustrate that a limit for the nZEB balance is in the range of four storeys under the considered boundary conditions. For both office buildings, the same concepts (cogeneration with biogas and the B/W heat pump concept) have the best energy balance.

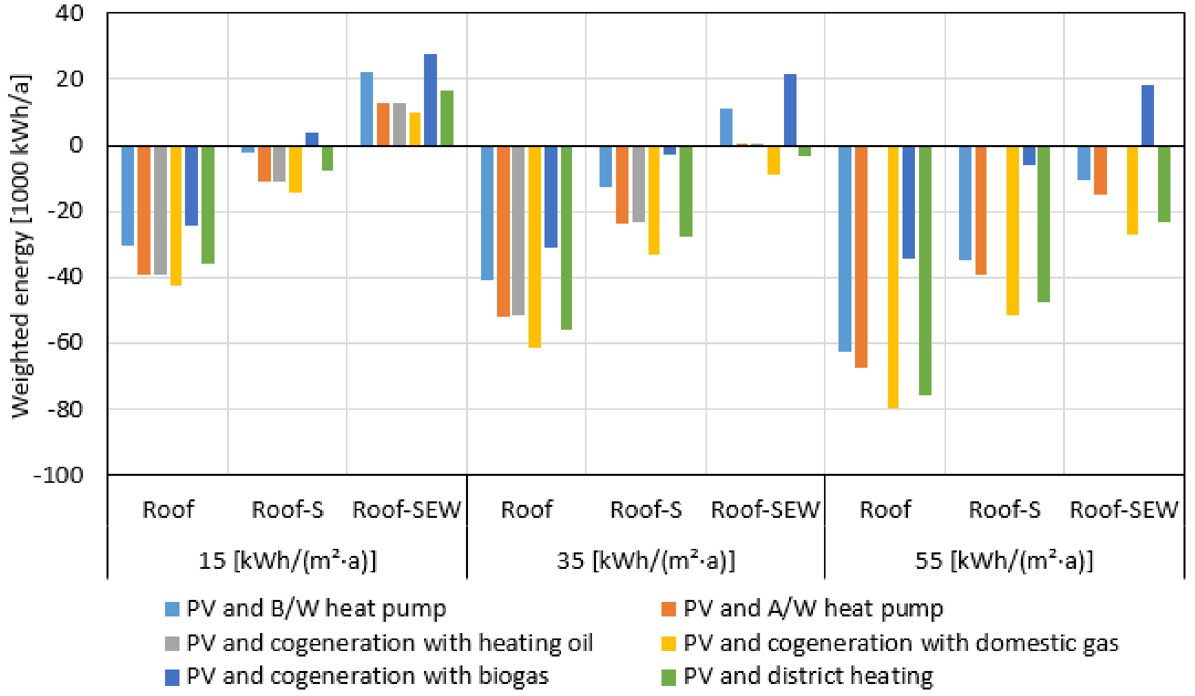


Fig. 46: Energy balance for office building B with five storeys

7.3 Evaluation of cost

The evaluation of the cost has been performed as annualised cost over a life-cycle of 25 years, considering investment, operation and maintenance.

7.3.1 Single-family-house

Fig. 47 shows the cost comparison of the system configurations for annualised cost over the life cycle of 25 years for SFH of the different insulation standards expressed as space heating needs 15/35/55 kWh/(m^2a) as described above.
In SFH buildings heat pump systems combined with PV show the lowest annual costs. In small buildings with low space heating needs A/W-heat pumps are cheaper than B/W-heat pumps. Systems with biomass or biogas generate higher annual costs due to higher investment and the higher fuel cost especially for biogas, which is at current prices about twice the price of natural gas.
Solar thermal collectors tend to increase the system cost, while not necessarily improving the performance substantially, which is due to higher cost for storage and piping. Moreover, a decrease of the seasonal performance factor (SPF) of the heat pump is observed, since the heat pump increasingly runs at worse operation conditions, by producing the higher water temperatures, as the solar system preheats the storage. Last but not least there may be a competition with the solar PV area in situations with scarce buildings surface areas.

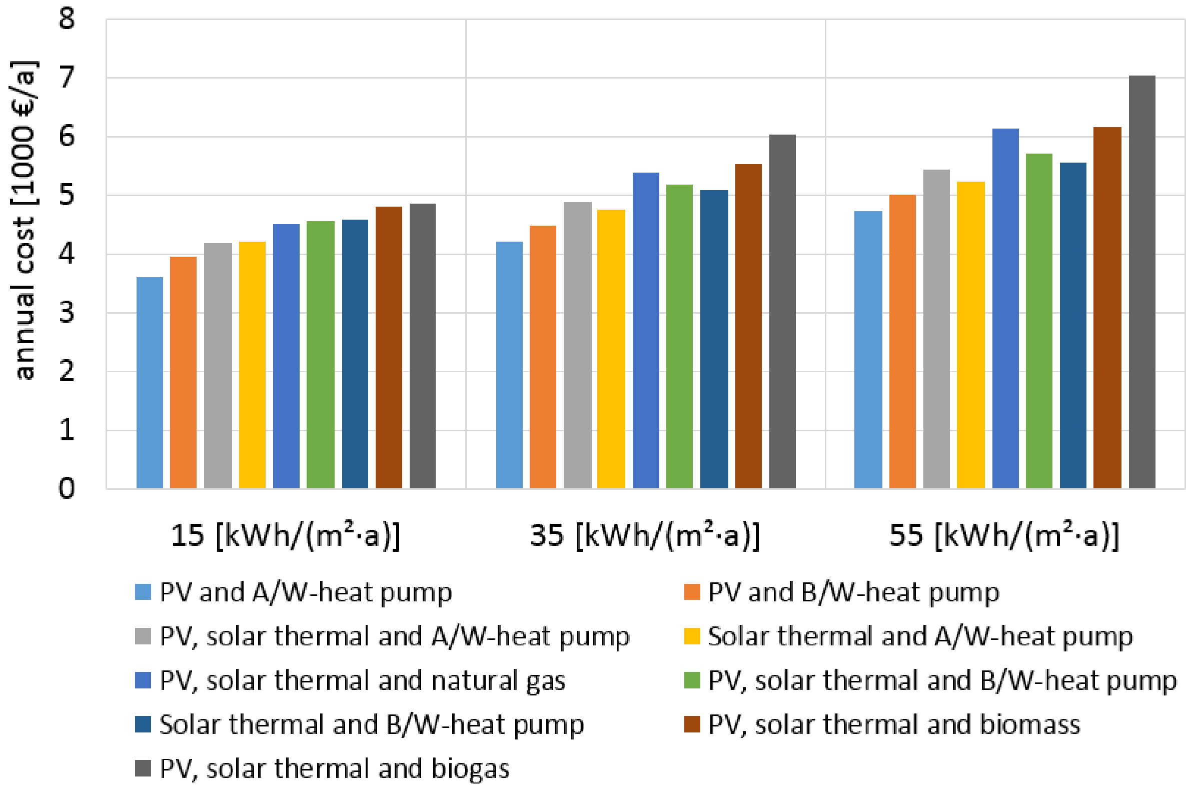


Fig. 47: Cost balance (annual cost) for SFH

7.3.2 Multi-family-house

The results for the cost balance of the MFH in Fig. 48 is similar to the results of the SFH. A difference, though, is that the B/W-heat pump get more cost-effective compared to the A/W-heat pump with the increase of the specific space heating needs.

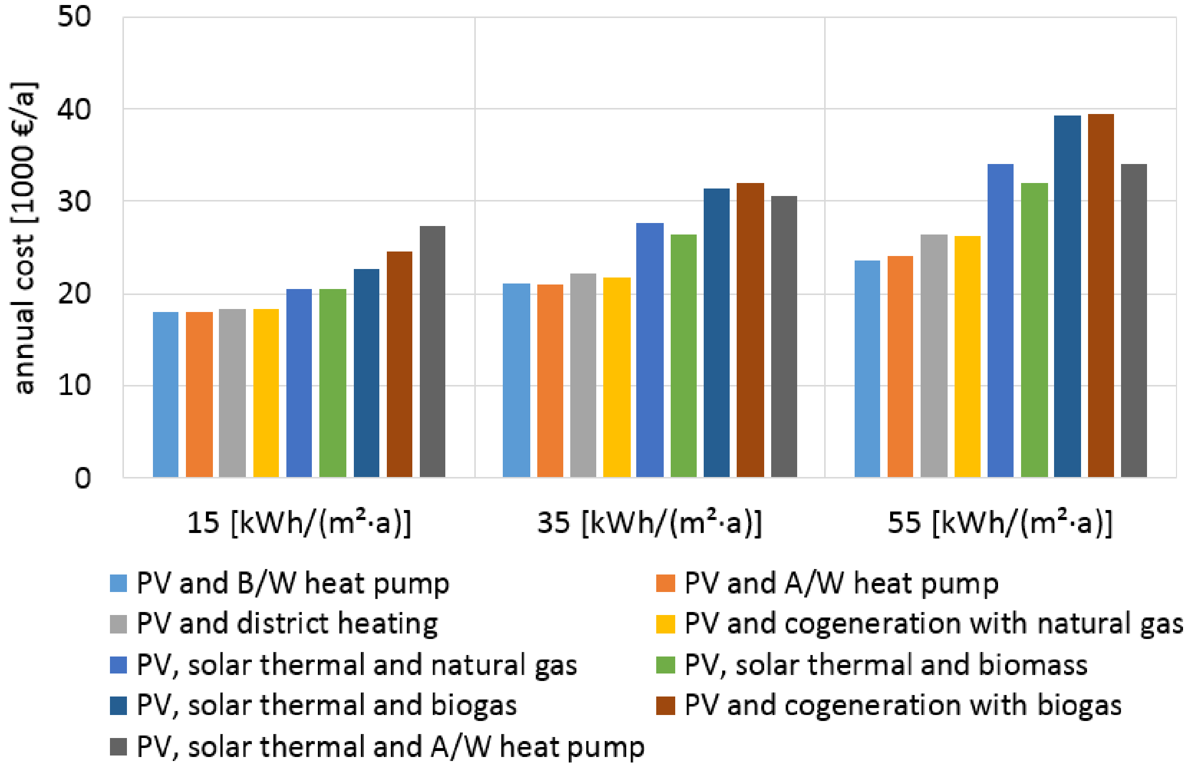


Fig. 48: Cost balance (annual cost) for MFH

Besides the heat pump concepts, district heating and combined heat and power (CHP) on the basis of natural gas are the solutions with the lowest cost with nearly the same costs as the heat pump systems for higher insulation standards (space heating needs of 15-35 kWh/(m²a)). For higher space heating needs the heat pump solutions tend to be more cost-effective. For low space heating needs, the seasonal storage concept has the highest cost, while with increasing space heating needs, the combinations with biogas get the highest cost.

7.3.3 Office buildings

The cost balance of building A and B are similar, and the number of storeys has only a minor impact on the system ranking regarding annual costs. Therefore, in Fig. 49 only the annual cost of the office building type A with 3 storey is depicted.

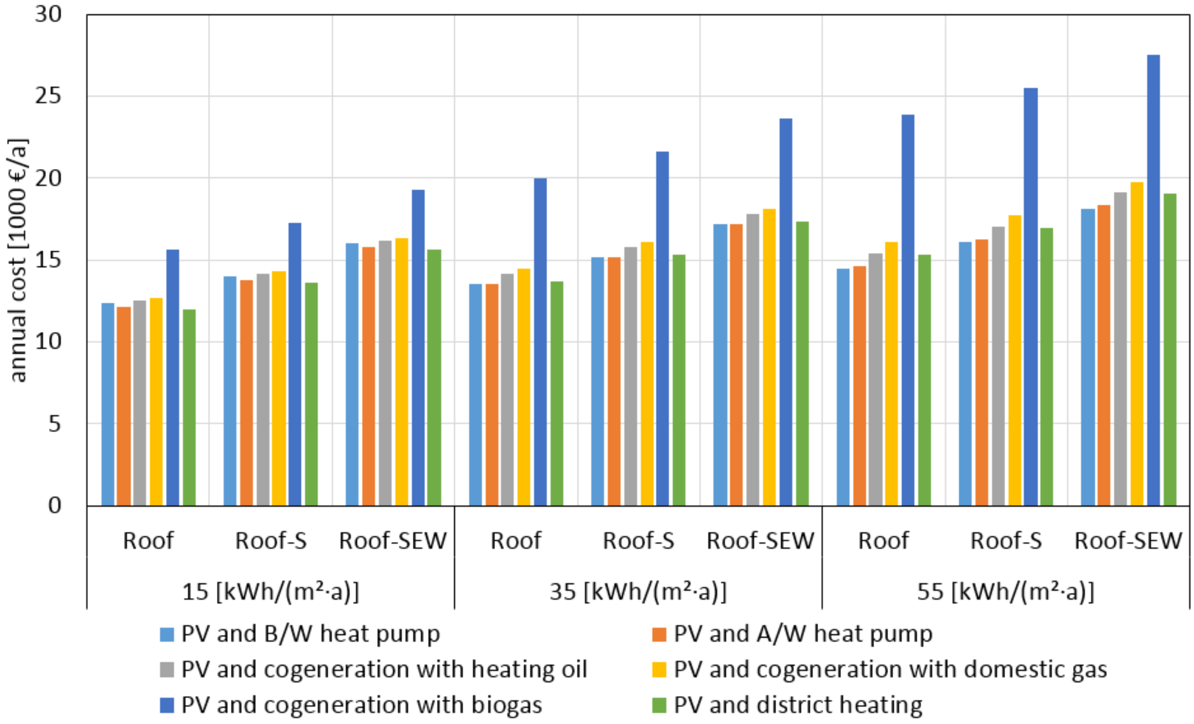


Fig. 49: Cost balance (annual cost) for office building A

With low space heating needs, district heating has slightly lower cost than the heat pump and CHP solutions. B/W and A/W heat pumps get cheaper with increasing energy needs, though. CHP with biogas has the highest cost as in the residential buildings.

7.4 Conclusions

Simulation results under the chosen boundary conditions confirm that heat pump concepts in combination with solar photovoltaic panels to balance the energy consumption have the lowest life-cycle cost to reach a nearly zero energy balance in buildings with moderate energy consumption up to 50 kWh/(m²a).

In SFH on ultra-low energy house level, all concepts can reach the MINERGIE-A® balance with PV on the roof area only. Heat pumps reach the lowest life cycle cost, while air-to-water (A/W)-heat pumps are slightly more cost-effective than brine-to-water (B/W) for low space heating needs.

In multi-family houses (MFH) heat pumps and district heating as well as combined heat and power (CHP) with biogas can reach an nZEB balance by just using the roof area. However, CHP with biogas has higher cost. The other concepts need an extended area by using the façade for PV generation. The most cost effective concepts are the heat pump systems as well as district heating and CHP with natural gas. B/W heat pumps get more cost-effective with increase of the specific heat demand.

For the office buildings possible concepts with the sole use of PV on the roof area are ground-source heat pumps, district heating and cogeneration. A limit for the nZEB balance is in the range of four storeys under the chosen boundary conditions. For low building energy demands, district heating is most cost-effective, while ground-source heat pumps get cheaper with increasing energy demands.

The heat pump systems are cost-effective with regard to the annual life-cycle cost and energy-efficient for all building types, even though SPF of the heat pump is rather conservative with 2.7 for A/W-heat pump and 3.3 for the B/W-heat pump. Only district heating and CHP can compete with the similar annual life-cycle cost for low space heating needs. With increasing heating needs, heat pumps are more cost-effective and ground-coupled heat pumps get better life-cycle cost than air-source heat pumps due to the better performance and less necessary PV area to meet the nZEB balance. With decreasing heating needs, though, higher investment cost cannot be compensated by the better SPF, so A/W-heat pumps have lowest life-cycle cost.

Side by side combinations with solar thermal collectors tend to increase the cost without substantial increase of energy performance due to additional components of storage and piping, competition with PV area on restricted overall roof area and worse operating conditions for the DHW operation of the heat pump. Moreover, PV is accounted with a weighting factor of 2, while solar thermal energy simply reduces the needs.

The only drawback of heat pumps is the limited load match due to the increase of winter electricity. In that sense, CHP has the better load match characteristic, since PV deficits in wintertime are compensated due to CHP electricity generation for winter heating needs resulting in a good load match. These aspects are topics for further investigations.

8 Comfort tool for surface heating systems in the USA

In the USA, case studies on the indoor thermal environment of different cooling or air-conditioning systems have been performed based on the development of the ThermCom design software tool for the evaluation of indoor thermal environment of rooms or automotive cabins of the size of a mid sedan.

8.1 Motivation and background

In a vapour compression system, the Coefficient of Performance (COP) of the system generally increases, if the temperature lift between heat source and heat sink decreases. This is due to a reduced pressure difference for the compressor leading to reduced power input caused by a lower temperature lift. In order to reduce the temperature lift, increase of the evaporating temperature or decrease of the condensing temperature or both are a promising option.
However, for conventional vapour compression air-conditioning systems, there is an upper limit for the evaporating temperature to maintain sufficient dehumidification. Use of separate sensible and latent cooling technology (SSLC) can eliminate this restriction (Ling, 2011). Despite the encouraging energy savings reported in the literature, the design of the indoor heat exchanger (HX) in the SSLC system remains a challenge. As the refrigerant evaporating temperature increases, the supply air temperature usually increases for a given HX. Consequently, the air mass flow rate has to increase to maintain the system capacity. The increased air flow rate leads to an increased fan power consumption and may offset the power savings by the compressor. In short, how to design an indoor HX that minimizes the fan power demand is the challenge.
There are various HX designs, which address such a challenge, and among those, chilled ceiling panels and heated floor systems are under the most consideration. For both products, the heat transfer mechanism between the working fluid and indoor air is natural convection and radiation, and therefore, the fan power demand is almost zero. Moreover, the enlarged heat transfer area of those products provides radiant cooling or heating to the occupants, so that both products may provide better thermal comfort than conventional indoor HXs. The ThermCom software is to evaluate the indoor thermal environment for different emission system solutions optimising the temperature lift of the heat pump or chiller, respectively.

8.1.1 Background on indoor thermal environment

ASHRAE Standard 55-2004 (2004) specifies the combinations of indoor thermal environmental factors and personal factors that will produce thermal environmental conditions acceptable to a majority of the occupants within the space.

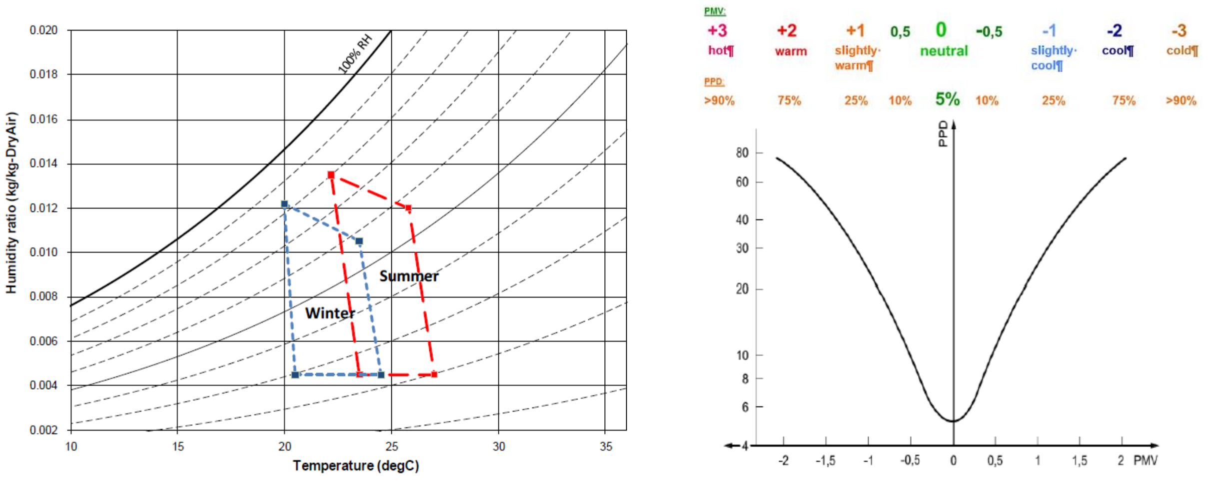


Fig. 50: Acceptable indoor thermal environment in psychrometric chart (left) and PMV scale with PPD as function of PMV (right, based on ISO 7730, 2005)

An index called the Predicted Mean Vote (PMV) is used to measure a large group of persons, thermal sensation on a seven-level scale which uses +3 to be hot and -3 to be cold. Fig. 50 lists the definition of the PMV index same as in the algorithm by Fanger standardised in ISO 7730 (2005). Six primary factors - metabolic rate (met), clothing insulation (clo), air dry-bulb temperature, mean radiant temperature, air velocity and humidity - are included in the standard when defining conditions for thermal comfort. With moderate air speed and humidity, the indoor thermal environment assessment can be expressed by the operative temperature, which can under these conditions be calculated by the average of the room air temperature and the mean radiant temperature of the room surfaces for moderate air velocity.

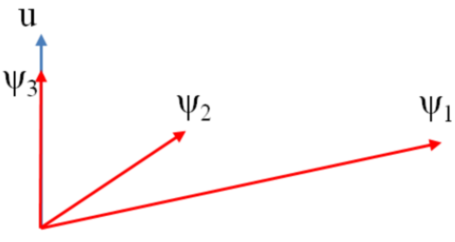
8.1.2 Reduced order method for indoor air flow field prediction

In order to effectively and accurately model indoor thermal environment, one has to be able to simulate both the mean radiant temperature (MRT) field and the air temperature field. MRT field modelling requires every surface temperature and view factors between the surfaces and object as inputs. In case the room can be simplified to regular walls with no obstacles between walls and occupants, the simulation of the radiant temperature field is very straightforward.
However, the air temperature field modelling can be computationally intense. In order to obtain the air temperature field, computational fluid dynamic (CFD) simulations are typically used, solving the equation set of continuity, momentum and energy. But due to the relatively large room scale, the CFD model typically results in a fairly large mesh size, taking hours for CFD to render a solution.
Proper orthogonal decomposition (POD) was chosen as the reduced-order simulation methodology. The POD method, which was first introduced by John Lumley (Berkooz et al., 1993) is also called Karhunen-Loève decomposition or principal components analysis. It has several advantages as pointed out by Berkooz et al. (1993) since:

- it is statistically based; extracting data from experiments and simulations
- its analytical foundations supply a clear understanding of its capabilities and limitations.
- it permits the extraction of the results.

In short, the method seeks to decompose a large degree of freedom system into a series expansion:

$$v(x,t)=\sum_{i=1}^{m} a_i(t)\varphi_i(x) \qquad \text{(Eq. 3)}$$



The search for a good bases is the first step of constructing the expansion. A good set of bases makes the expansion efficient in terms of minimizing m. In order to make the basis to be optimal, it is equal to the problem of finding a set of bases that is “most similar” to v. This can be explained from the figure aside. Assuming u is the vector to be decomposed while ψ_1, ψ_2 and ψ_3 are the different candidates of POD basis φ, neither ψ_1 or ψ_2 can represent u unless additional bases were introduced to offset their horizontal components. For ψ_3, it is most similar to u in the sense that no additional basis is necessary. Only a coefficient is needed to adjust the magnitude of ψ_3 to match u.
Thus, it is the same problem as to solve the following equations and to find φ:

$$max\frac{|(v,\psi)|^2}{(\psi,\psi)}=\frac{\langle|(v,\varphi)|^2\rangle}{(\varphi,\varphi)} \qquad \text{(Eq. 4)}$$

The parentheses in the equation represent inner products. When v and ψ are in the same direction, the inner product of the two reaches the maximum. If those two are perpendicular to each other, the inner product reduces to zero.

In some literatures, it is also common to use the following expression. One of the advantage of this expression is to produce normalized bases:

$$max(|(v,\varphi)|^2)-\lambda(|\varphi|^2-1 \qquad \text{(Eq. 5)}$$

It is essential to have a restriction on the normalization in this extreme calculation. In the figure above, ψ_3 is the worst basis of the three, however, without the normalization, it still can be the maximum of inner product due to its large magnitude.
For simplicity, several steps of extreme computing are skipped. For those who are interested in them, please refer to the spectral theory (Riesz and Nagy, 1990).
It turns out that the base functions (φ) are the eigenfunctions of the integral equation:

$$\int C(x,x')\varphi(x')dx'=\lambda\varphi(x) \qquad \text{(Eq. 6)}$$

where the kernel C is given by:

$$C(x,x')=\frac{1}{N}\sum_{i=1}^{N}v_i(x)v_i(x') \qquad \text{(Eq. 7)}$$

Therefore, in short, the search for POD bases can be divided into two steps:
construction of the kernel C and calculation of eigenvectors of the C.
Before we start to apply the POD to our air temperature simulation, two questions remain to be answered:

- How can one guarantee that the series of expansion covers the entire span of u?
- How can one prove that the series of expansion is the optimal one?

By observing the kernel C, it is clear that the matrix is non-negative. Therefore, all the eigenvalues must be non-zero, i.e., $\lambda_i \geq 0$. Those eigenvectors corresponding to zero eigenvalues do not contribute to the entire kinetic energy of the space. The entire system space is hence reduced to be only formed by eigenvectors corresponding to eigenvalues. There is no piece of information of u missing from the process. More detailed explanation can be obtained from the propositions 2.1 and 2.2 in Berkooz et al. (1993).

Optimality

The POD basis set is optimal for modelling or reconstructing v. Proposition 2.3 in Berkooz et al. (1993) was considered as the basis to the claim.

Proposition 2.3: Let $v(x,t)$ be an ensemble member square integrable on Ω for almost every t and $\{\varphi_i,\lambda_i\}$ be the POD orthonormal basis set with associated eigenvalues. Let

$$\mathrm{v(x,t)}=\sum_{\mathrm{i}}\mathrm{a_i(t)\varphi_i(x)} \qquad \text{(Eq. 8)}$$

be the decomposition with respect to this basis, where equality is almost everywhere. Let $\{\psi_i\}$ be an arbitrary orthonormal set such that

$$\mathrm{v(x,t)}=\sum_{\mathrm{i}}\mathrm{b_i(t)\psi_i(x)} \qquad \text{(Eq. 9)}$$

Then the following holds:

1. $\langle a_i(t)a_j^*(t)\rangle=\delta_{ij}\lambda_i$, i.e. the POD coefficients are uncorrelated.
2. For every n we have $\sum_i^n\langle a_i(t)a_i^*(t)\rangle=\sum_i^n\lambda_i\geq\sum_i^n\langle b_i(t)b_i^*(t)\rangle$

This implies, that, among all linear decompositions, this is the most efficient in the sense that, for a given number of modes the projection on the subspace used for modelling will contain the most kinetic energy possible in an average sense. In addition, the time series of the coefficient $ai(t)$ are uncorrelated.

8.2 Case study I: Radiative panel heat exchanger system

In order to reduce the temperature lift, a novel radiative heat exchanger (HX) has been investigated. The radiative HX has a similar structure as a chilled ceiling panel, which include serpentine-shaped tubes fixed on metal sheets. The tubes circulate a working fluid such as water that serves as heating/cooling source and conducts heat to/from the metal sheets. The metal sheets condition the space air by both, natural convection and radiation. An attractive feature of the radiative HXs is that instead of installing them over the ceiling or under the floor, the HX can be simply installed against walls. This installation method has a two-fold benefit: easier installation and capability of providing both heating and cooling.
The objective of the application of the simulation tool development is to evaluate the indoor thermal environment conditioned by a radiative HX system. In the modelling, the radiative HX is assumed to control the temperature of the entire wall being installed. The modelling includes two major efforts: the simulation of the operative temperature (OT, or t_o) field and the evaluation of ASHRAE's thermal comfort criteria. In the case of applying the radiative HX, operative temperature calculation can be simplified as the arithmetic mean of MRT and air temperature. The indices PMV and predicted percentage of dissatisfied (PPD) are used to quantify the thermal comfort criteria.
The air temperature simulation uses the technique of computational fluid dynamic (CFD) for air temperature field inside the space. A commercially-available CFD package (ANSYS, 2006) was chosen for the modelling.
As an example to facilitate initial research, a 2D square was assumed to represent the vertical middle intersection plane of the space. It has one cold side (left side) of 20°C which can be assumed as a case of the radiative HX filled with cold water to provide cooling, and one hot side (right side) of 35 °C which can be assumed as a case of hot window by direct solar radiation. The square has a mesh of 240 by 240 quad cells with enhanced mesh density in the boundary layer to capture the complicated flow characteristics. To be specific, the boundary layer has the first row of 1 mm and the growth of 1.15, i.e., the entire depth of the boundary layer is 20 mm. The turbulence model used in the model is k-ω Shear Stress Transport (SST) model. The velocity (streamline) and temperature field are demonstrated in Fig. 51.

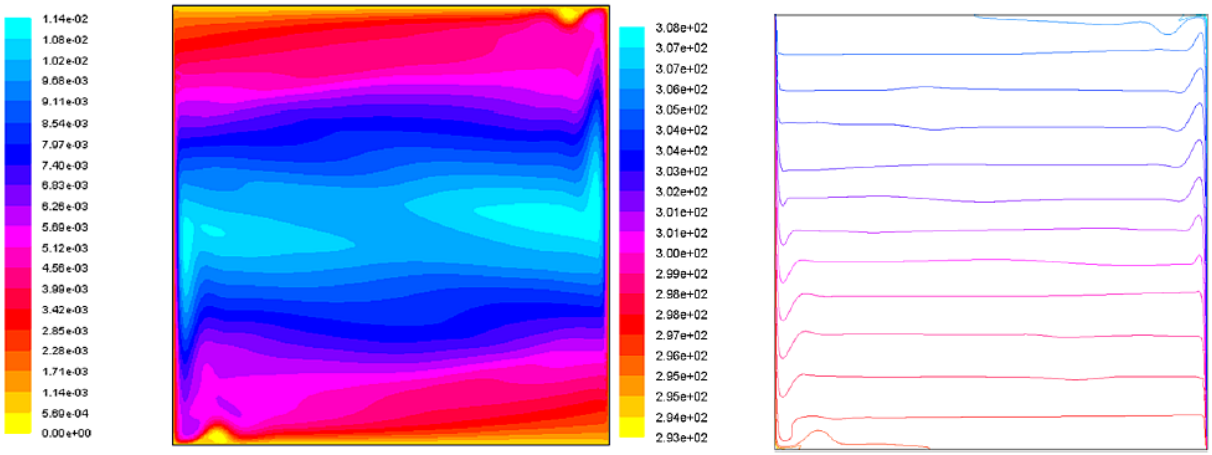


Fig. 51: Streamline velocity field (left) and temperature field (right)

The steps to be performed for the POD methods are shown in Fig. 52 and (Eq. 10).

$$v(x,t) = \sum_{i=1}^{m} a_i(t)\varphi_i(x) \qquad \text{(Eq. 10)}$$

The basis functions (φ) are the eigenfunctions of the integral equation (Ly and Hein, 2001):

$$\int C(x,x')\varphi(x')\,dx' = \lambda\varphi(x) \quad \text{(Eq. 11)}$$

where the kernel C is given by:

$$C(x,x') = \frac{1}{N}\sum_{i=1}^{N} V_i(x)V_i(x') \quad \text{(Eq. 12)}$$

For the current problem, in order to obtain the POD modes for the air temperature field, a technique called “snapshot” was applied to form the matrix C by utilising the existing CFD simulation results. The single steps are shown in Fig. 52. Nine sets of temperature fields and velocity fields were chosen to form the snapshot. The difference among each snapshot is the different Rayleigh numbers. The CFD software package is used to calculate the eigenvalues of the kernel matrix C. The built-in singular value decomposition (SVD) function was applied to return the eigenvalues and the corresponding eigenvectors.

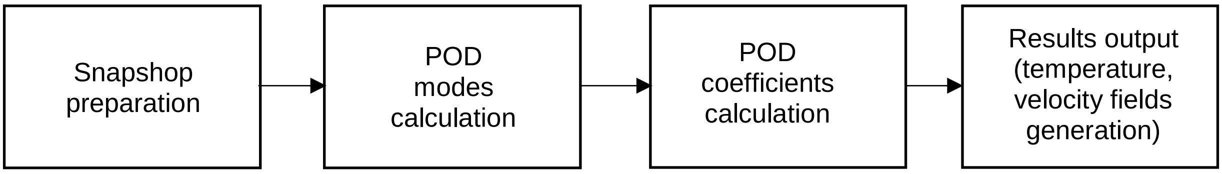


Fig. 52: Steps of the snapshot technique to perform the POD method

If the eigenvalue is zero meaning that it poses no impact on the system, its corresponding eigenvector is neglected. The POD provides the basis of the expansion series. The next step is to find the coefficients in the expansion. A method called Galerkin projection is considered to be a standard approach to obtain the coefficients. The method projects the governing equations on the modal subspace and then solves the governing equations, usually in the form of ordinary differential equations to obtain the coefficients.

<u>PMV field simulation</u>

The previous subsection discusses the simulation of both MRT and air temperature. By simply averaging those two, one can obtain the OT field. However, it is not straightforward to use OT to describe thermal comfort. Therefore, the PMV and PPD fields are also simulated by the software tool. As noted earlier the PMV is a seven-scale system to describe occupant’s thermal sensation from very hot (+3) to very cold (-3). To calculate the PMV, the calculation in ISO standard 7730 (2005) is used. Fig. 53 right show sample outputs of PMV and PPD from the tool. As shown in the figures, both the left and right sides of the space are affected by the hot wall and hot window as well as the larger vertical air velocity than that in the middle, so the PMV indices near the walls are between slightly warm and warm.

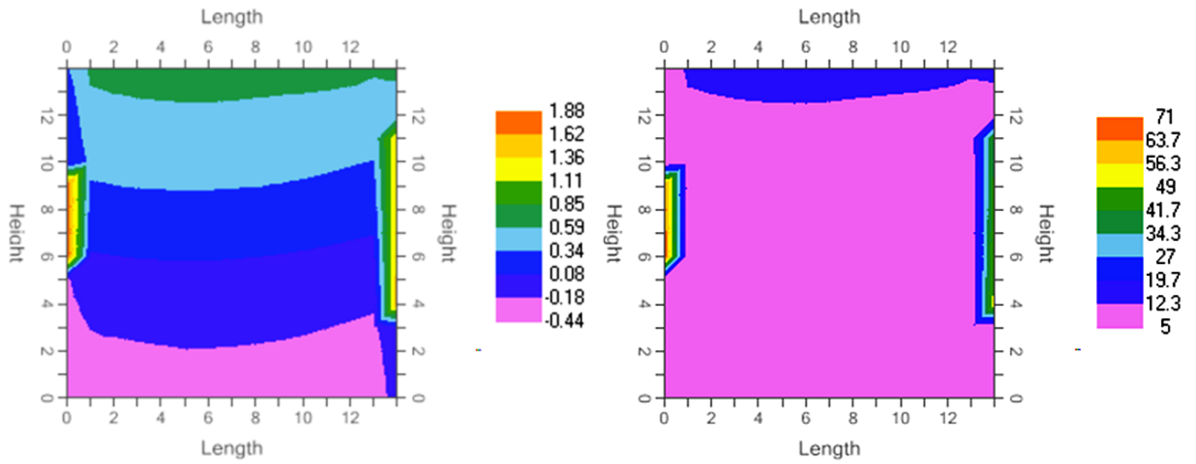


Fig. 53: PMV (left) and PPD (right)

8.3 Case study II: Induction supply air unit

A novel heat pump system using induced-air supply units capable of maintaining the same dehumidification without increasing supply air fan power is introduced. The schematic of an induced-air unit is shown in Fig. 54.
Fresh air from an air handling unit is sent to the induced-air supply unit. The fresh air has a temperature of 13 °C which is lower than that of traditional units, but because of careful design of heat exchangers the evaporating temperature is maintained at 10 °C. As the fresh air is received from the top, the unit entrains room air through the bottom inlets located at the two sides by induction. Although the fresh air flow is low, the total mixed supply air flow rate is actually higher than that of conventional units. Since the induction does not require any power input, the larger air flow rate does not increase fan power consumption. The amount of induced air is proportional to that of the fresh air supplied and the total air flow is typically made up of 60% room air induced with 40% fresh air supply. The total supply air is sent to the space through the middle part at the bottom of the unit. The surface area of the supply unit is relatively larger than that of conventional outlets (for a rated 300 m³/h supply air flow rate, the unit surface area is 0.3 m²) and the supply air velocity is between 0.2 m/s to 0.9 m/s. The relatively lower air velocity brings the benefit of minimizing the possibility of draught sensations and hence better thermal comfort.

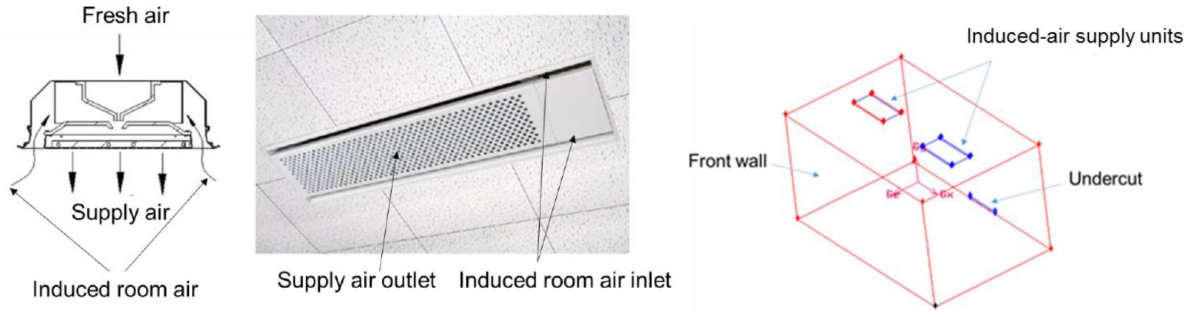


Fig. 54: The supply air unit (left) and computational domain for a room with two air supply units (right).

Moreover, due to the mixing with room air, the supply air temperature is around 19 °C requiring no reheat, and consequently provides occupants with comfortable supply air at no extra fan power penalty. Concluding an induction can enhance comfort both in space heating and cooling operation at lower fan power than traditional units. In the following, their indoor thermal environment is evaluated.

8.3.1 Indoor air temperature calculation

Several room models with multiple induced-air units are modelled in ANSYS (2006) using 3D double precision option. The size of rooms varies depending on the number of supply units. For one- and two-supply cases, the room dimension is 3 m long, 1.5 m wide and 2.7 m high. For three and four supply cases, the room is modelled as 4.5 m long, 3 m wide and 2.7 m high. The supply units are uniformly distributed at the top of the room. Each supply is around 1.1 m in length and 0.6 m in width. The CFD mesh size for the smaller room is 500,000 and 1,260,000 for the larger room. Fig. 54 right shows the computational domain for a room with two outlets. The supply unit is modelled as a face with two return air inlets at two sides and supply air outlet in the middle just as shown in Fig. 54 left. The Boussinesq assumption (Boussinesq, 1897) is applied to enable the simulation of natural convection of indoor air.
According to the characteristic length and temperature difference of the model, the Rayleigh number exceeds 10^9, and therefore the k-ω SST model is enabled as the turbulent viscous model. The conditioned space also receives solar radiation through the outside (window) wall. The solar heat flux is assumed 800 W/m².
Fig. 55 and Fig. 56 show sample results of indoor air temperature profiles and velocity profiles of one supply case at the middle plane of the room in both heating and cooling conditions, respectively.

The air temperature profile shows a clear stratification due to air density difference, however, the temperature difference from floor to a point 1.5 m high is within 2 K. The velocity profile shows a maximum velocity of 0.9 m/s at the supply duct outlet.

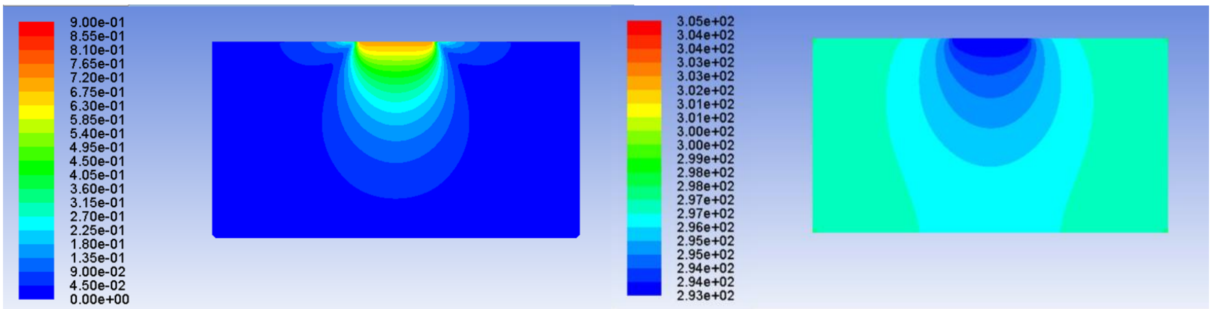


Fig. 55: Air velocity [m/s] profile of one induced air supply unit (cooling case, middle pane, left) and air temperature [K] profile of one induced air supply unit (cooling case, middle pane, right)

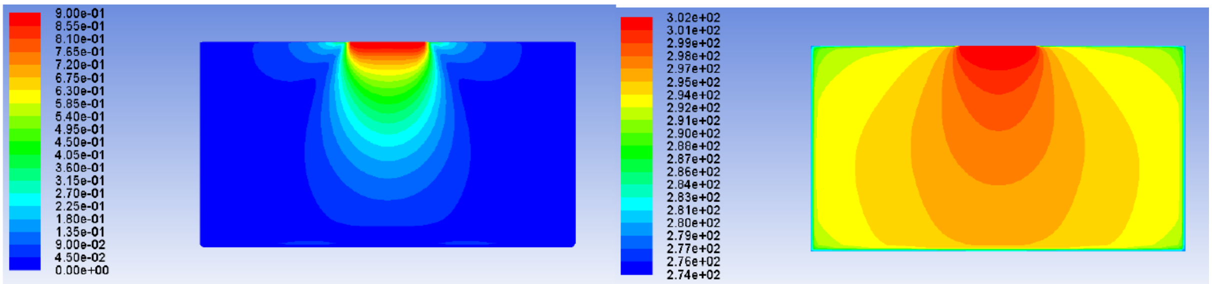


Fig. 56: Air velocity [m/s] profile of one induced air supply unit (heating case, middle pane, left) and air temperature [K] profile of one induced air supply unit (heating case, middle pane, right)

The bulk air region has a negligible air velocity showing minimal signs of draught. It can also be found that the return air flows back to the supply unit to represent the induction effect.
Fig. 57and Fig. 58 show the indoor air temperature and velocity profiles in the case of one-supply room. Since it is a reduced-order model, the resolution was downgraded to 16 by 16 by 8. The POD calculation took only ~2 minutes compared to more than 8 hours for a 3D CFD simulation. The significant speed improvement is due to POD conducts interpolation based on CFD snapshots. Based on the POD outputs of temperature and velocity distributions, the average air temperature and velocity are calculated. The MRT is calculated by using room surface temperatures as well as view angles of the centre point in the room to individual surfaces.

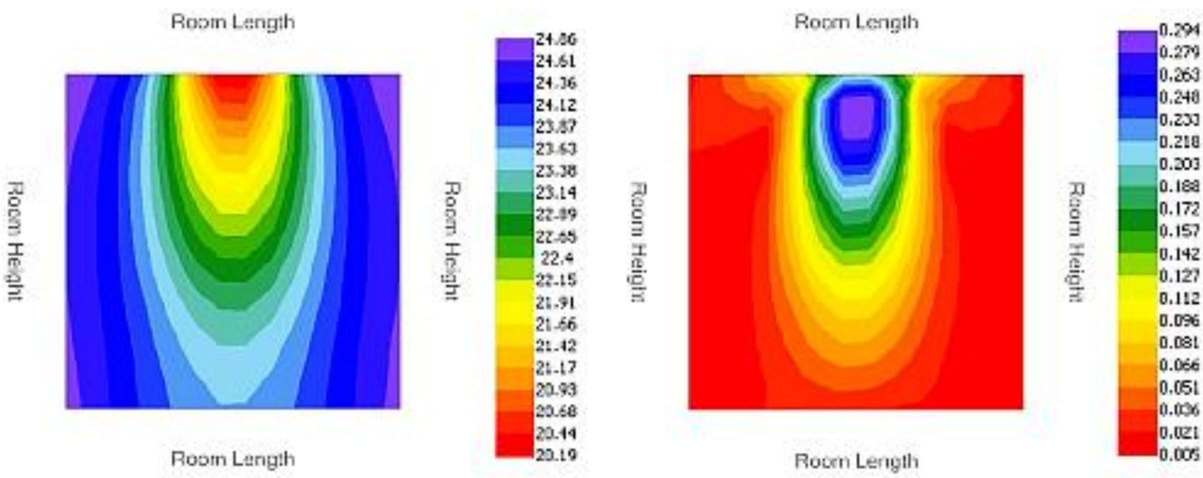


Fig. 57: Sample POD results on temperature and velocity profile for the mid plane, temperature (left) and velocity (right) in the cooling case

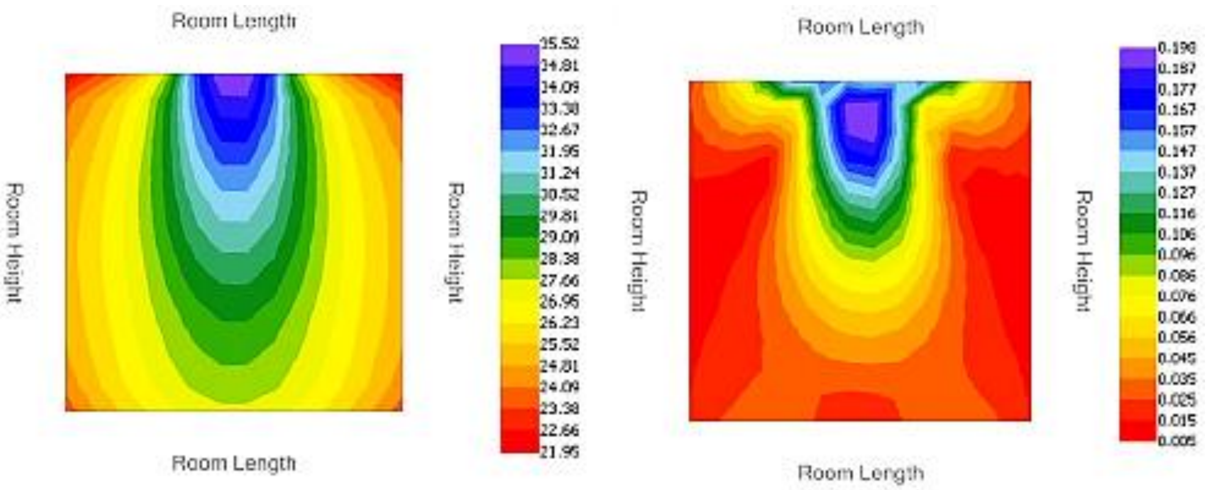


Fig. 58: Sample POD results on temperature and velocity profile for the mid plane, temperature (left) and velocity (right) in the heating case

Fig. 59 shows the PMV and PPD prediction for the room based on one typical weather day in Beijing, China. The typical weather days are defined based on the coldest and hottest days in one year.

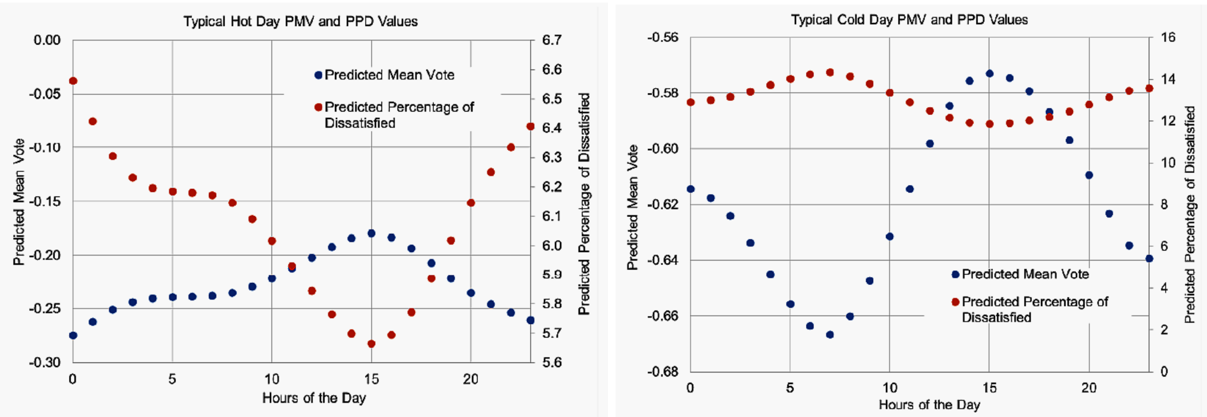


Fig. 59: Evaluation of PMV and PPD for typical hot day (left) and cold day (right) in Beijing climate

8.4 Case study III: Comparison of ducted and ductless system

CFD models were established for both ducted and ductless heat pump systems.

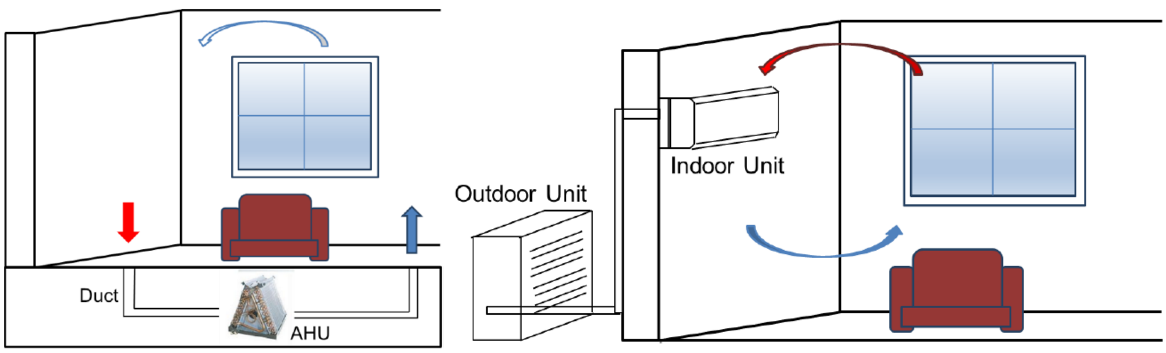


Fig. 60: Principle of ducted (left) and ductless (right) air conditioning or heat pump system, respectively

For both systems, a fixed room dimension is selected as length (L) x width (W) x height (H) = 5 m x 3 m x 2.4 m. Since the focus of the study is indoor thermal comfort, only indoor units are relevant for the simulation. The indoor unit dimension of the ductless system was obtained from the measurement of a field test system. The dimension is L x W x H = 0.9 x 0.3 m x 0.15 m.

To maintain a mass balance in the room, the room air will be discharged through an undercut with a dimension of L x H = 1 m x 0.006 m. The ducted system is assumed to have a floor outlet. The air outlet size is chosen from common grills on the market to be L x W = 0.4 m x 0.4 m. The undercut dimension is the same as the one in the ductless system.

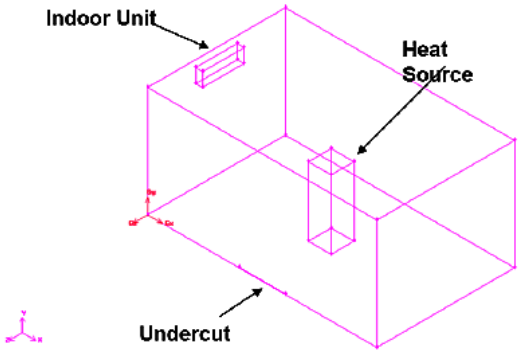


Fig. 61: Room model with the ductless system

Both heating season and cooling season simulations were conducted with various supply air temperatures. In heating, the supply air temperature varies from 35 °C to 45 °C while in cooling, the supply air temperature varies from 12 °C to 18 °C. To maintain the same cooling capacity into the space, the supply air velocity is adjusted according to the supply temperature. Fig. 62 shows the air temperature and Fig. 63 the velocity fields of ducted system and ductless system.

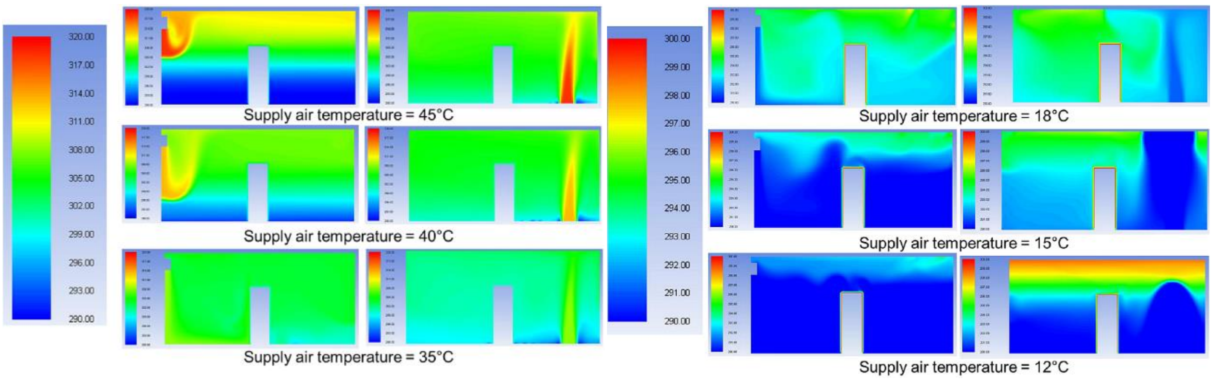


Fig. 62: Comparison of temperature fields in heating case (left) and cooling case (right) for ductless systems (left in each figure) and ducted systems (right in each figure)

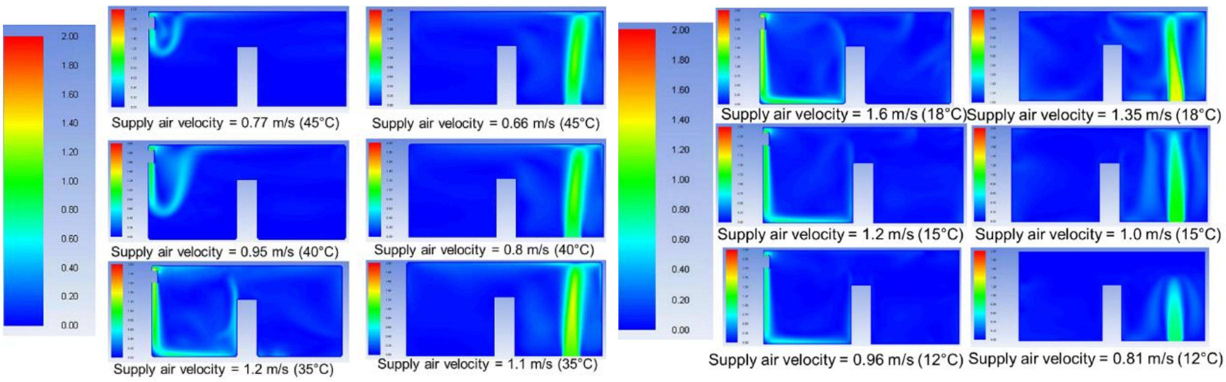


Fig. 63: Comparison of velocity fields in heating case (left) and cooling case (right) for ductless systems (left in each figure) and ducted systems (right in each figure)

It is found that in heating the ductless system shows a greater degree of temperature stratification than does the ducted system. The ductless system with smaller supply air velocity performs better in the sense that the bottom half of the space has a comfortable temperature range, but it also shows that less supply air velocity leads to larger temperature stratification. However, for these analyses, the ductless system is assumed to have a fixed air supply direction of vertical downward. In the real case, the outlet vane can be adjusted to guide the supply air flow to different angles. This may improve the thermal comfort for ductless systems.

It is also applied to ducted system. The grill can be adjusted so that the air plume is not vertically upward (or the grill can be placed in the ceiling or on a side wall).
It is found that in the cooling case, the ducted system with the smallest supply air velocity (lowest supply air temperature) shows a high degree of room temperature stratification. Overall, the ductless system provides more effective cooling than does the ducted system. Again, the different supply angle is not considered.
The ASHRAE PMV equation is then applied by using the temperature and velocity fields as inputs. Generally speaking, the ducted system with large supply air velocity shows an overall best PMV distribution in heating, while for cooling, the ductless system is better. Fig. 64 shows the calculated PMV for heating (left) and cooling (right) operation of the ductless system.

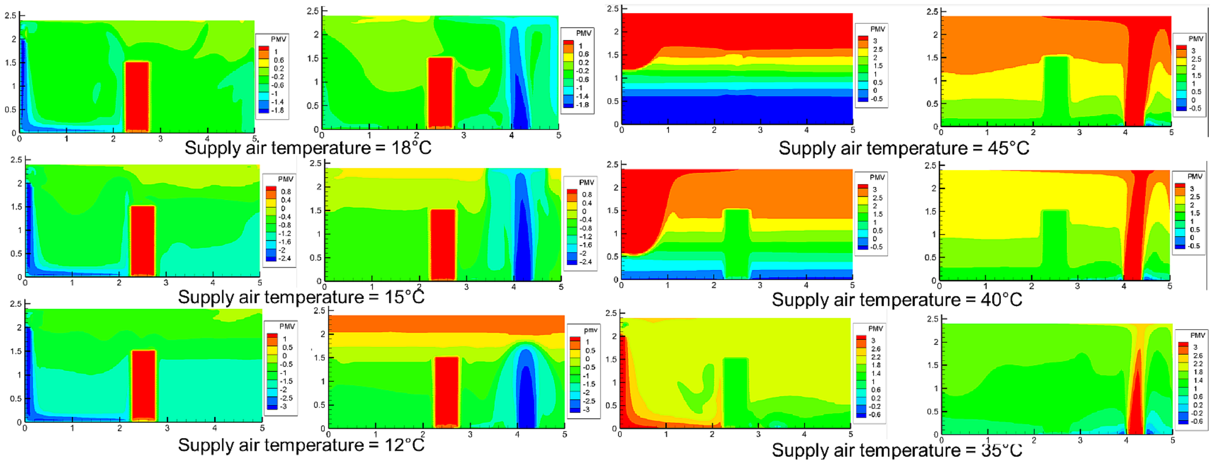


Fig. 64: Comparison of PMV fields in cooling case (left) and heating case (right) for ductless systems (left in each figure) and ducted systems (right in each figure). Effects of supply air angles for ductless system not considered.

In the discussion above, the supply air angle of the ductless system was assumed to be vertically downward. In the real case, it can actually be changed by adjusting an air discharge vane angle. Additional CFD simulations were carried out by assuming different vane angle settings and the impacts on thermal comfort were evaluated. Fig. 65 shows the ductless system temperature and velocity fields with three vane angle settings, namely 0° (vertically downward), 45° and 80°.

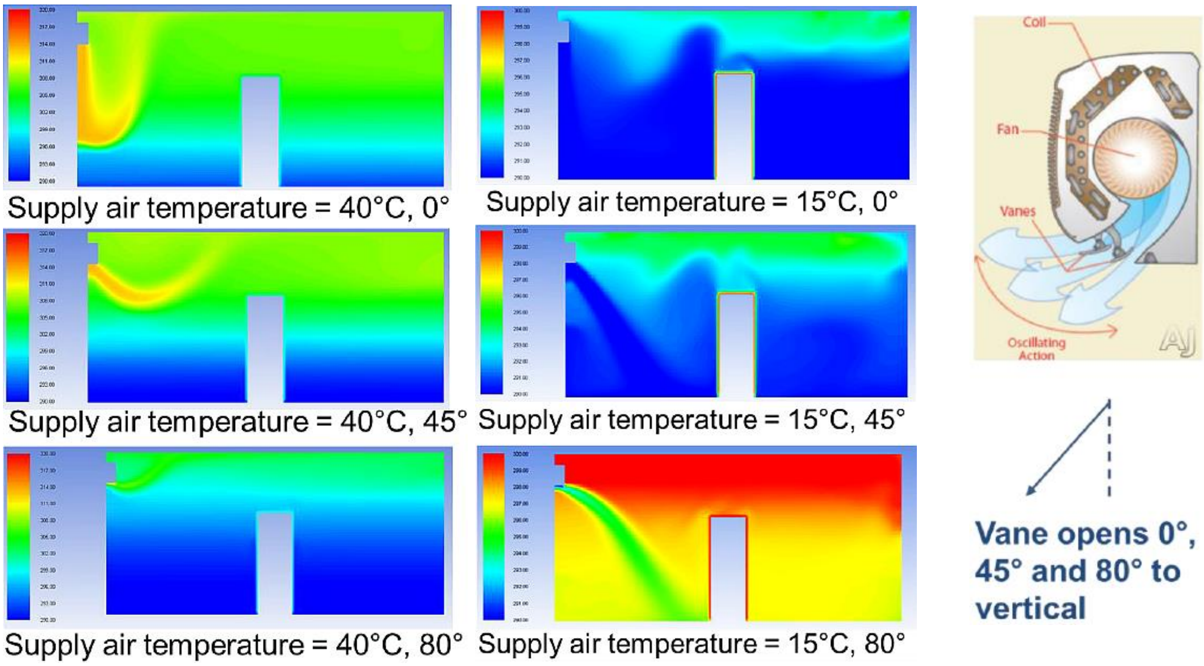


Fig. 65: Comparison of temperature fields for ductless systems in heating case (left) and cooling case (right) with different vane position angles

For heating, the 45° vane setting helps sending the hot air farther into the room. For cooling the 0° and 45° vane position settings have a good thermal comfort in terms of PMV. Fig. 66 gives the velocity field for the comparison between the ducted and ductless system and Fig. 67 illustrates the PMV field the two situations, each for the different vane angles.

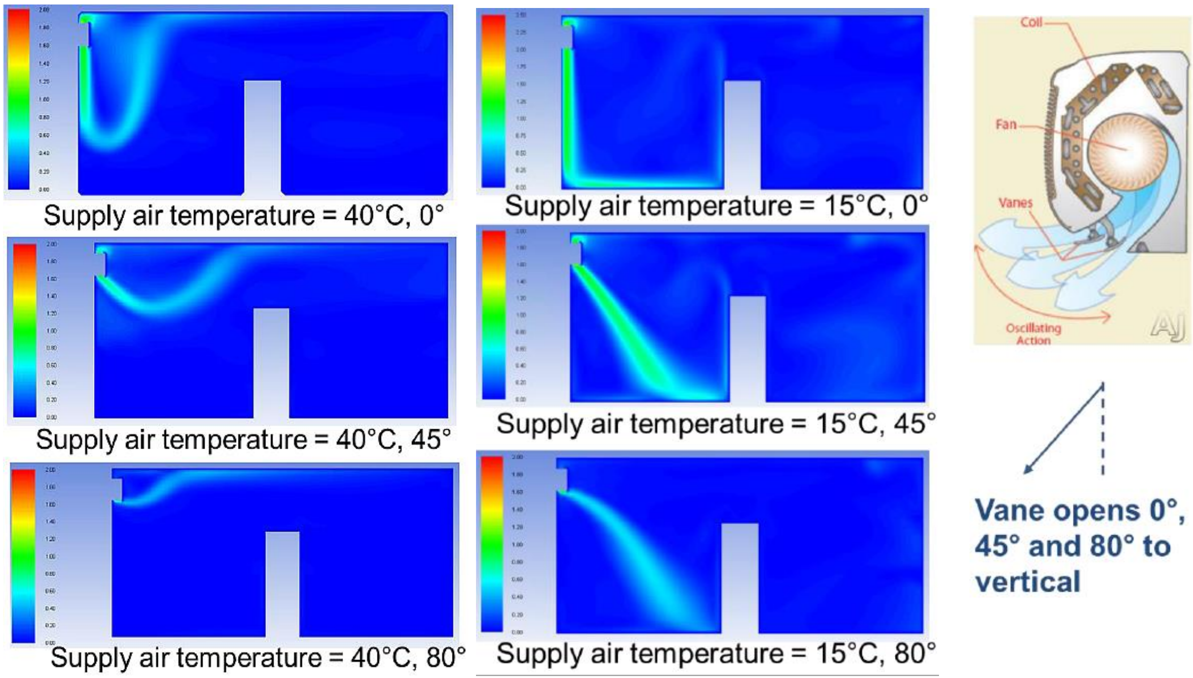


Fig. 66: Comparison of velocity fields for ductless systems in heating case (left) and cooling case (right) with different vane position angles

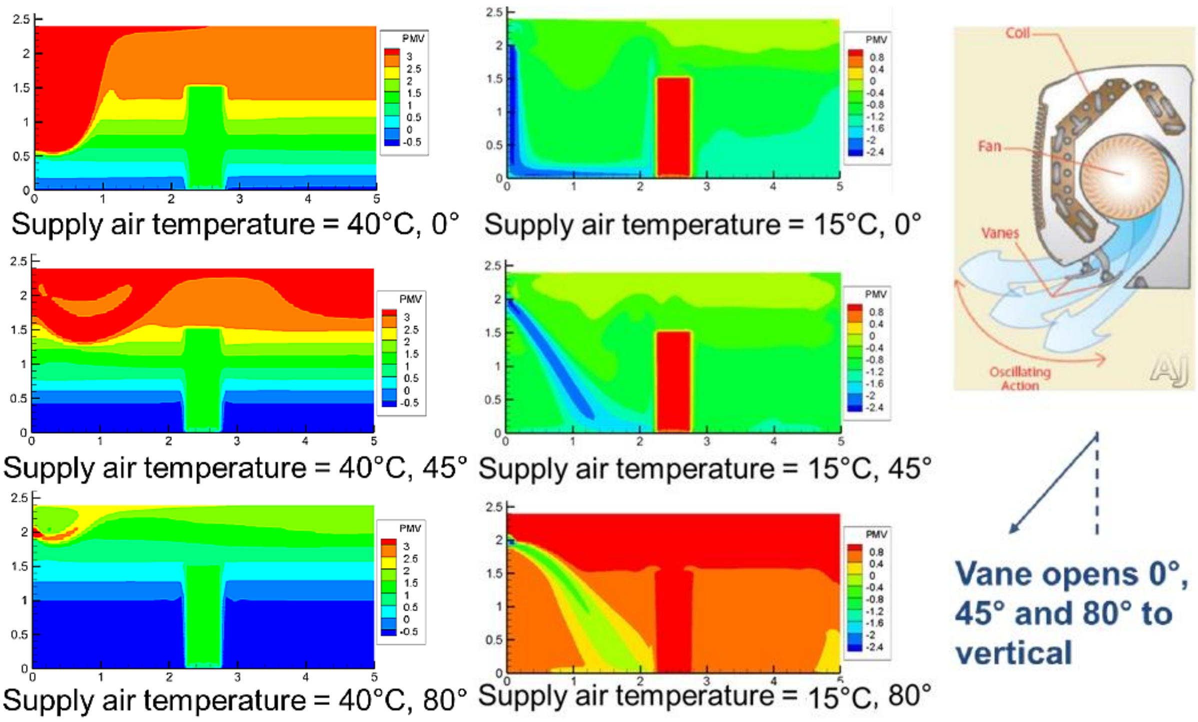


Fig. 67: Comparison of PMV fields for ductless systems (right) in heating case (left) and cooling case (right)

8.5 Recommendations on system layout in nZEB

Based on the three case studies, it is found that the radiant heat exchanger and induction-based supply unit systems (Cases I and II) are capable of providing improved thermal comfort at low energy consumptions. The major reason is, that for both systems the room air circulation is driven either by natural convection or pressure gradient. The bulk air circulates at relatively low air velocities and therefore minimizes the chance of draft.

To achieve better thermal comfort, it is also important to control the MRT. Conventional systems cannot control the MRT effectively due to the small surface area and view factor between supply air discharge grills or units and occupants. Radiant panels and induction-based supply units have a relatively larger surface area and therefore a larger view factor to occupants. It is proved that with a better control of MRT, the PMV and PPD indices are improved.

With regards to conventional centrally ducted air distribution systems and new emerging ductless systems (Case study III), the thermal comfort evaluation showed significant temperature stratification in the room. For the ductless system, it is important to adjust the supply air vane angle according to supply air temperature. For a ducted system with air outlets in the room floor, the vertical plume of supply air causes draft sensations and therefore affects thermal comfort. However, the ducted system can provide good thermal comfort in the heating season because the hot supply air flows naturally from the bottom to the top of the room.

9 Conclusions

In Task 2 of the IEA HPT Annex 40, case studies in order to compare system solutions in nZEB have been performed for different boundary conditions both for central European and Nordic climate conditions, as well as in Japan for climate with pronounced air-conditioning needs. As building types, both residential buildings and office buildings have been considered. Thereby, the boundary conditions regarding the nZEB definition and economic boundary conditions have been based on the current national state of definition and markets. Despite the partly different boundary conditions the case studies yield the results that both in central European as in Nordic countries heat pumps are a favourable candidate for the application in nearly Zero Energy Buildings both in terms of energy-efficiency and cost. With the heat pump solutions, requirements of nZEB can be reached. Even though heat pumps have partly higher investment cost compared to other heating systems on the national markets of the single countries, heat pumps range among the systems with the lowest life-cycle cost.

Despite the differences in the nZEB balance definition, the economic boundary and climate conditions, the resulting ranking of the different system solutions is quite similar and heat pumps are among the most energy-efficient and cost-effective system solutions, showing a certain robustness of the results both regarding the energy performance evaluation and the economic boundary conditions. For the energy evaluation heat pump solutions benefit from the high energy performance in nearly zero energy buildings, since the good building envelopes enable low supply temperatures. Thereby, the nZEB balance can be reached more cost-effective, since less on-site generation is required for the compensation of the energy demand of the building. As stated above heat pumps may have higher investment cost on the national markets, but this initial disadvantage is compensated by less investment in generation technologies, e.g. PV systems may be designed smaller, and the higher energy performance of heat pumps reduces the operation cost, which is seen in the life-cycle consideration.

According to these results heat pumps are very suitable for the application in nZEB also for Nordic climate conditions.

Moreover, the development of two design tools has been started in Task 2 of the Annex.

One of the tools developed at the University of Maryland in the USA is dedicated to provide low supply temperature for the heat pump in space heating as well as high temperatures in space cooling without taking a risk of violating thermal comfort in the room zone. Based on a model reduction technique of CFD simulation, the tool can evaluate the indoor thermal environment of rooms in high spatial resolution. Typical comfort parameters like PMV and PPD can thereby be evaluated with less computational expense. Even though radiative systems used to be in focus, the tool can also be applied for convective systems of ducted air-handling units or ductless room air conditioners and for induction units. For these systems case studies are contributed.

The other tool developed at SINTEF Energy in cooperation with NTNU in Trondheim, Norway, deals with an optimised design of the heat pump system for the objective function of CO_2-eq.-emissions and cost. Based on preprocessing of the building data, an iterative optimisation of the system solutions regarding a cost-effective zero emission system design can be performed. Since the different models have recently been gathered in Matlab-Simulink, a future development perspective is the automatic optimisation of the system with included optimisation algorithms.

Summarising, both tools can help to create even more energy-efficient and cost-effective system designs for the application of heat pumps in nZEB.

10 Acknowledgements

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Abbreviations and Nomenclature

A/W .... *Air/Water (Heat Pump)*
AAHP .... *Air-Source Heat Pump (Air to Air)*
ACH .... *Air Exchange per Hour*
AHU .... *Air Handling Unit*
ASHP .... *Air-Source Heat Pump*
AWHP .... *Air-Source Heat Pump (Air to Water)*
B/W .... *Brine/Water (Heat Pump)*
BAT .... *Best Available Technology*
CC ASHP .... *Cold Climate Air-Source Heat Pump*
CFD........ *Computational Fluid Dynamics*
CHP .... *Combined Heat and Power*
COP .... *Coefficient of Performance)*
DH........ *District Heating*
DHW .... *Domestic Hot Water*
DOE .... *Department of Energy*
DOT .... *Design Outdoor Temperature*
e.g. .... *exempli gratia*
EAHP .... *Exhaust Air Heat Pump*
ECM .... *Electrically Commutated Motors*
EER........ *Energy efficiency Ratio*
ERA........ *Energy Reference Area*
FMU .... *Function Mock-up Unit*
GAHP........ *Ground-Air Heat Pump*
GC .... *Gas cooler*
GHG........ *Greenhouse Gases*
GSHP........ *Ground-Source Heat Pump*
GSHP/S .... *Ground-Source Heat Pump with Solar*
HC........ *Hydrocarbons*
HEX........ *Heat Exchanger*
HFC........ *Hydrofluorcarbons*
HP........ *Heat Pump*
HPP........ *Heat Pump Programme*
HPT........ *Heat Pump Technologies*
HRV .... *Heat Recovery Ventilation*
HVAC .... *Heating Ventilation and Air Conditioning*
HX........ *Heat Exchanger*
IEA .... *International Energy Agency*
LCC........ *Life Cycle Cost*
LCEM .... *Life Cycle Energy Management*
LED .... *Light Emitting Diode*
MFH .... *Multi-Family House*
MRT .... *Mean Radiant Temperature*
MURB .... *Multi-Unit Residential Building*
NECB .... *National Energy Code of Canada*
NEDO........ *New Energy and Industrial Technology Development Organization*
NRCan .... *National Research Council Canada*
nZEB .... *nearly Zero Energy Building*
NZEB .... *Net Zero Energy Building*
NZEP .... *Nearly Zero Energy buildings heat Pump installations*
NZR........ *Net-Zero Ready*
OA........ *Outdoor Air*
OAT........ *Outside Air Temperature*

OT .... *Operative Temperature*
PID .... *Proportional Integral Differential Controller*
PLR .... *Part Load Factor*
PMV .... *Predicted Mean Vote*
POD .... *Proper and Orthogonal Decomposition*
PPD .... *Predicted Percentage of Dissatisfied*
PTAC .... *Packaged Terminal Air Conditioning*
PV .... *Photovoltaics*
RC building .... *Reinforced Concrete building*
SA XXHP .... *Solar Assisted XX Heat Pump*
SFH .... *Single-Family House*
SPF .... *Seasonal Performance Factor*
SSLC .... *Separate Sensible and Latent Cooling*
SST .... *Shear Stress Transport*
ST .... *Solar Thermal*
SVD .... *Singular Value Decomposition*
VAV .... *Variable Air Volume*
WLHP .... *Water-Loop Heat Pump*
WT .... *Water Temperature*
WWHP .... *Water-Source Heat Pump (Water to Water)*
ZEB .... *Zero Energy Building*

Nomenclature

A_{net}	Heat net floor area	m²
$E_{D3(2012)}$	E-value	kWh/(m²a)
$E_{DE,i}$	Delivered energy i	kWh/a
E_{EXP}	Weighted exported energy	kWh/(m²a)
$E_{EXP,i}$	Exported energy i	kWh/a
$f_{DE,i}$	Weighing factor delivered energy	-
$f_{EXP,i}$	Weighing factor exported energy	-

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A. Appendix

A.1 Detailed results of the Canadian pathway the NZEB for MURB

<u>Standard Efficiency Measures</u>
To quantify the benefit to reducing the annual energy consumption of the mid-rise apartment compared to simply adding PV to make it net zero, several energy efficiency measures are investigated. From an examination of energy end use graph, space heating, lighting and receptacle loads, fans and domestic hot water heating make up just over 85% of the energy end use. Thus energy efficiency measures addressing these end uses are selected to reduce the energy consumption. These measures include:

- *Reduction in base electrical loads through energy efficient lighting and appliances*
  The base electrical loads were modelled based on the recommended appliance and lighting loads. Energy Star® packages are often offered and can have substantial energy savings compared to standard equipment in new housing developments. A 60% reduction in lighting power and a 25% reduction in appliance electrical consumption were assumed for this case (Energy Star®, 2014). The estimated incremental costs for the efficient base electrical loads are $25,000 CDN.

- *Reduction of fan power using electronically commutated motors (ECM)*
  Each apartment unit has a central heating and cooling system with a two speed indoor blower assumed to draw 1 W per L/s of airflow. ECM motors are variable speed and can have significant savings compared to PSC motors. For the analysis, the ECM motors are assumed to be two speed and conservatively draw 0.5 W per L/s of airflow (Gusdorf et al., 2002). The estimated incremental cost for the ECM motors is $7,000 CDN total.

- *Reduction in the space heating load through a high efficient building envelope*
  The building envelope insulation levels were set to the minimum levels outlined by the NECB for the Montreal region. This efficiency measure evaluates whether there is a benefit to heavily insulating the building using realistic construction practices to further reduce the space heating loads. The estimated incremental cost to improve the insulation level of the building envelope is $145,000 CDN. Tab. A 1 summarizes the proposed changes.

Tab. A 1: Proposed building envelope insulation improvement

Characteristic	Baseline	Proposed
Roof u-value	0.181 W/m²°C	0.125 W/m²°C
Wall u-value	0.248 W/m²°C	0.188 W/m²°C
Window u-value	1.98 W/m²°C Double Glazed	1.31 W/m²°C Triple Glazed

- *Reduction in the space heating load through the heat recovery from exhaust air*
  Due to the improved air tightness of the building, fresh air must be brought in to each apartment unit. Heat recovery ventilators are only required however in colder Canadian regions (NECB Region 7A and higher). This measure evaluates the benefit of adding in suite heat recovery ventilators to recover the exhaust heat. Approximately 33 L/s of fresh air is brought into each apartment unit. The anticipated total cost to install heat recovery ventilators in all apartment suites is $84,000 CDN.

Energy Efficient Systems

The second level of energy efficiency measures was the integration of energy efficient systems. These systems have been described in previous sections, and for this analysis included:

i. Conventional Air-Source Heat Pump
ii. Cold Climate Air-Source Heat Pump
iii. Ground-Source Heat Pump
iv. Solar Assisted Ground-Source Heat Pump

Analysis and Results

Energy models of the following energy efficiency measure combinations were developed to evaluate the energy saving potential versus cost over a PV system to achieve net zero energy.

- Measure 1: Reduction of base electrical loads
- Measure 2: Reduction of base electrical loads and use of ECM fan motors
- Measure 3: Reduction of base electrical loads, use of ECM fan motors and building insulation improvement
- Measure 4: Reduction of base electrical loads, use of ECM fan motors and heat recovery on the apartment unit exhaust air
- Measure 5: Reduction of base electrical loads, use of ECM fan motors, heat recovery on exhaust air and building insulation improvement

From RSMeans (2013) and contractor surveys the estimated energy efficiency measure cost was estimated and summarized in Tab. A 2.

Tab. A 2: Estimated incremental costs for standard efficiency measure combinations

Energy Efficiency Measure	Estimated cost
Measure 1	$25,000 CDN
Measure 2	$32,000 CDN
Measure 3	$177,000 CDN
Measure 4	$115,000 CDN
Measure 5	$260,000 CDN

Fig. A 1 presents the results of standard energy efficiency measure impact. Fig. A 1 a summarizes the final energy consumption intensity for each measure, Fig. A 1 b presents the annual utility costs with no PV system, Fig. A 1 c compares the PV array cost difference to achieve net zero energy (NZE) over the baseline energy consumption and Fig. A 1 d presents the cost comparison between using a PV array and the efficiency measure to achieve the same level of energy savings. A negative cost indicates that it is economically more beneficial to install a larger PV system than implement the proposed energy efficiency measure. A positive cost indicates that it is more beneficial to implement the energy efficiency measure to save energy rather than using a PV array.

As can be seen in Fig. A 1, all energy efficiency measures are beneficial in comparison to using photovoltaics to offset the same amount of energy. However, comparing the energy efficiency measures among one another, it is seen that the addition of building insulation is not beneficial as the cost difference between the energy efficiency measure and PV array to achieve the same energy savings significantly decreases (Measure 3 and Measure 5).

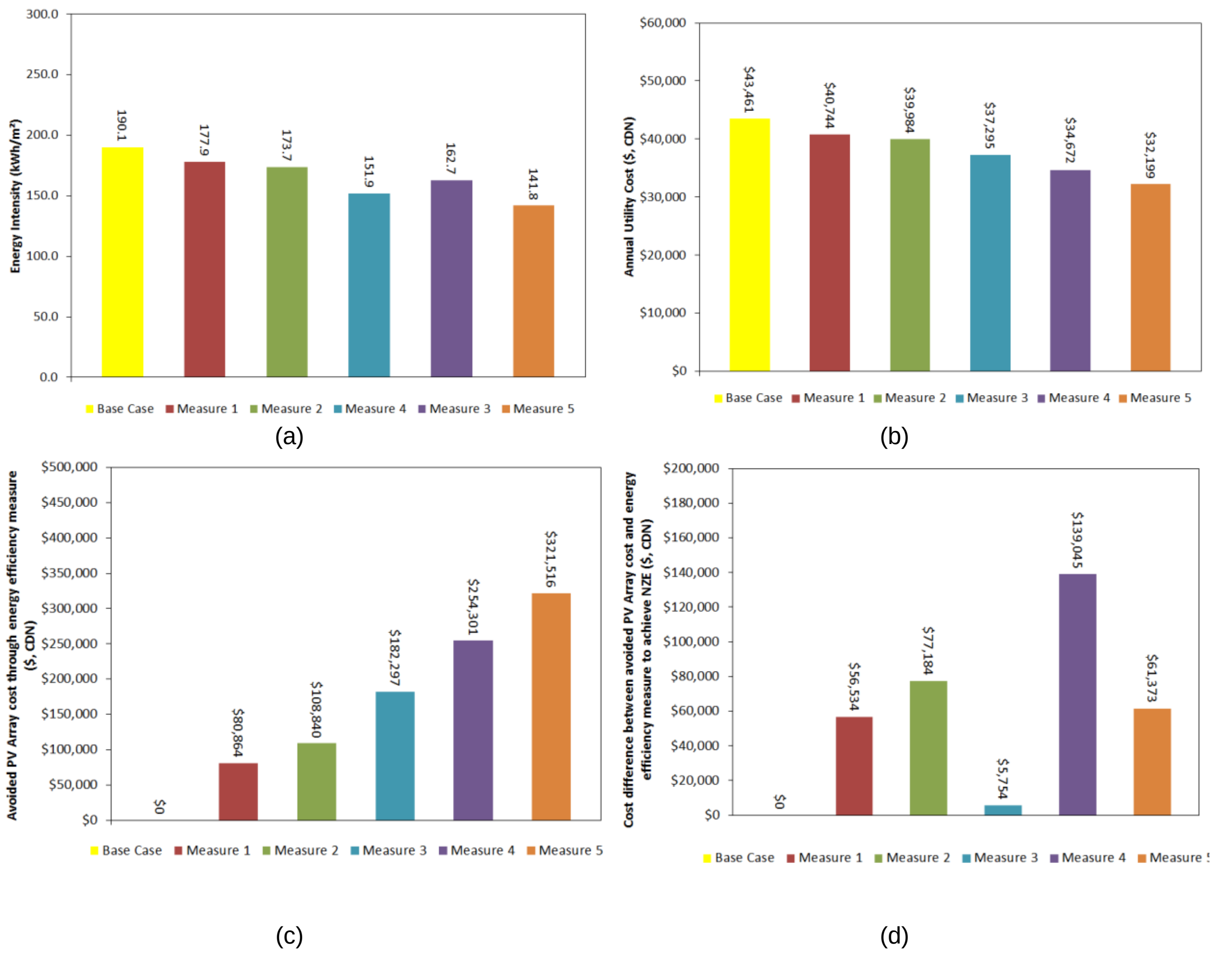


Fig. A 1: Comparison of the base case and energy efficiency measure (a) annual energy intensity (b) annual utility cost (c) PV array cost to achieve NZE and (d) cost difference between the efficiency measure and using PV to achieve the same energy savings

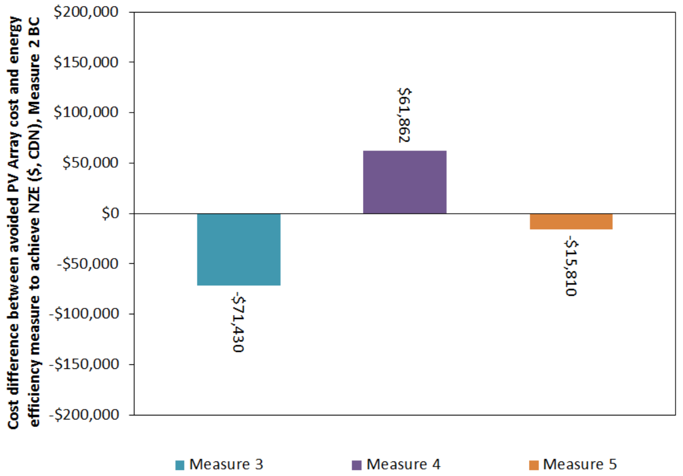


Fig. A 2: Cost difference between the efficiency measure and using PV to achieve the same energy savings in comparison to energy efficiency measure 2

This is further highlighted in Fig. A 2 , where the cost difference between the avoided PV array cost and the energy efficiency measure cost for efficiency measures 3, 4 and 5 in comparison to the efficiency measure 2 is shown. A negative cost difference highlights that a PV array is a more suitable economic choice than the energy efficiency measure.

Due to the already decent building envelope, it becomes difficult to economically justify further improving the building insulation. The building owner is better off in this case to use a PV system to offset the building energy consumption. There is a benefit however to installing heat recovery ventilators as opposed to using a PV array. Future work will investigate the breakeven point of adding insulation versus the incremental PV array cost.

To evaluate how the energy efficient systems compare in achieving energy savings with a PV system, each heat pump system was evaluated in the base case and five standard energy efficiency measures combinations identified above. From RSMeans (2013) and contractor surveys, Tab. A 3 summarizes the estimated incremental costs for each heat pump system compared to the defined base case heating and cooling system (packaged terminal air conditioner with electric back-up heating).

Tab. A 3: Estimated incremental costs for proposed heat pump systems

Heat pump system	Estimated incremental cost
1.5 ton ASHP	$102,500 CDN
CC ASHP	$166,250 CDN
GSHP	$396,500 CDN
Solar assisted GSHP	$414,250 CDN

For the GSHP and the solar assisted GSHP systems, the borefields were resized for energy efficiency measures 3, 4 and 5 as the building heating and cooling loads changed. The borefield size and estimated incremental cost in comparison to the base case with the same energy efficiency measure is summarized in Tab. A 4.

Tab. A 4: Estimated borefield size and incremental cost for the GSHP heat pump systems

Heat pump system	Borefield	Estimated incremental cost
Measure 3 GSHP	5 x 5, 95 m	$343,250 CDN
Measure 4 GSHP	5 x 4, 93 m	$300,500 CDN
Measure 5 GSHP	4 x 4, 85 m	$259,000 CDN
Measure 3 Solar Assisted GSHP	4 x 4, 98 m	$382,250 CDN
Measure 4 Solar Assisted GSHP	4 x 4, 76 m	$352,000 CDN
Measure 5 Solar Assisted GSHP	4 x 3, 80 m	$351,225 CDN

<u>Conventional air source heat pump system</u>

Fig. A 3 presents the results of the conventional air source heat pump evaluated in each of the standard energy efficiency models outlined above. Fig. A 3 a compares the annual energy intensity for each case, Fig. A 3 b compares the annual utility cost with no PV system, Fig. A 3 c compares the PV array cost difference to achieve NZE over the baseline energy consumption and Fig. A 3 d presents the cost comparison between using a PV array and the efficiency measure to achieve the same level of energy savings. A comparison to the base case with no energy efficiency measures is also provided.

From the results it can be seen that the use of an air source heat pump is profitable with an increased heating load on the building. By reducing the internal gains through Energy Star® lighting, appliances and the ECM fan motors, the heating load increases and the air source heat pump has a larger beneficial impact compared to using PV to offset the energy consumption. Similar to the base case, all efficiency measures are beneficial over using a PV array to achieve the same energy savings. However, when comparing the results of the energy efficiency measure 3 with energy efficiency measure 2, it can be seen that the benefit of adding insulation to reduce the space heating load is still difficult to economically justify compared to a PV array. While the energy efficiency measure as a whole is economically beneficial in comparison to using a PV array, the incremental benefit of improving the building envelope has no benefit as the cost difference decreases. A similar conclusion can be drawn with the use of heat recovery ventilators (measure 4) and the combination of improved building envelope and use of heat recovery ventilators (measure 5).

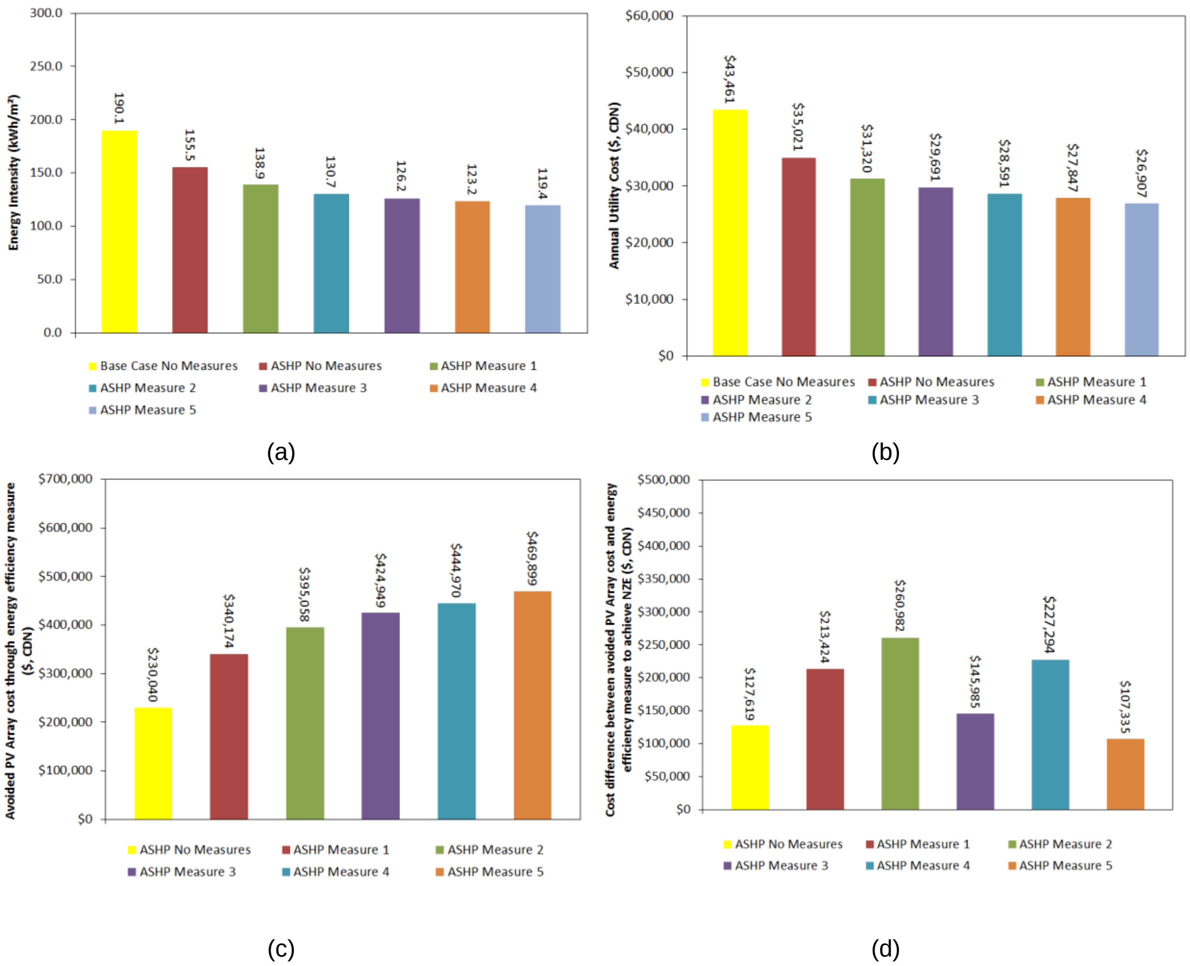


Fig. A 3: Comparison of the air-source heat pump (a) annual energy intensity (b) annual utility cost (c) PV array cost to achieve NZE and (d) cost difference between the efficiency measure and using PV to achieve the same energy savings

<u>Cold climate air source heat pump system</u>

Fig. A 4 presents the results of the CC ASHP evaluated in each of the standard energy efficiency models outlined above. Fig. A 4 a compares the annual energy intensity for each case, Fig. A 4 b compares the annual utility cost with no PV system, Fig. A 4 c compares the PV array cost difference to achieve net zero energy over the baseline energy consumption and Fig. A 4 d presents the cost comparison between using a PV array and the efficiency measure to achieve the same level of energy savings. A comparison to the base case with no energy efficiency measure is also provided. It should also be noted that the cold climate air source heat pumps already have ECM fan motors and thus there is no difference between energy efficiency measure 1 and 2.

A similar trend to the conventional air source heat pump system is seen with the CC ASHP. A cost benefit is seen for all efficiency measures, especially at higher heating loads, where there is a benefit towards implementing the CC ASHP system to reduce the annual energy consumption compared to using a PV system. However, similar to the conventional ASHP, as the building heating load is reduced through building insulation or heat recovery, the energy saving benefit of the CC ASHP is unable to overcome the cost of the incremental PV system required.

Comparing the cold climate ASHP results to the conventional ASHP, it can be seen that the conventional ASHP system has a lower annual energy intensity for the energy efficiency measures 2 through 5. Although the cold climate ASHP has improved performance at lower ambient temperatures and a variable speed compressor, the system is still significantly oversized for the requirements of each apartment unit and thus operates at low part loads (below 40%). Future work will investigate the optimum size for a cold climate ASHP.

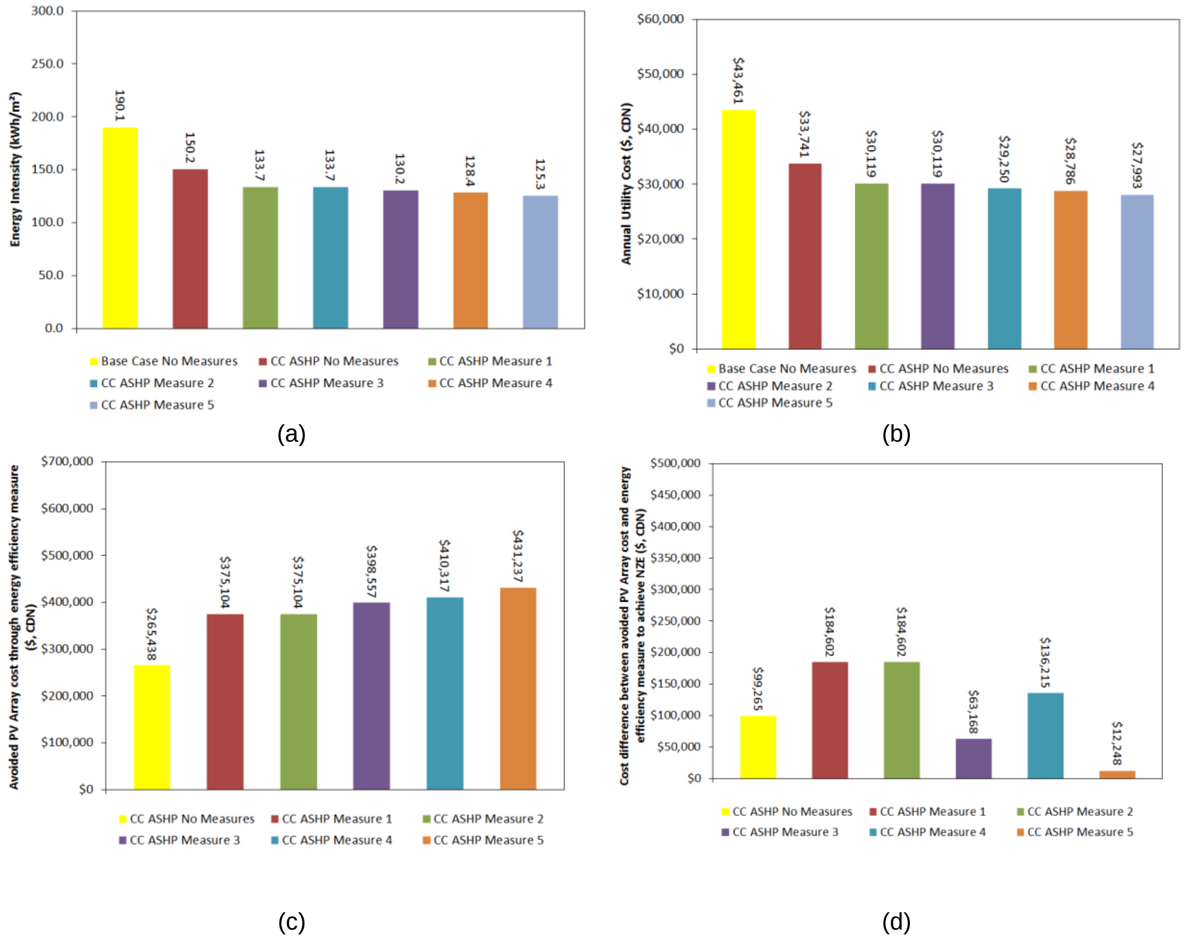


Fig. A 4: Comparison of the cold climate air-source heat pump (a) annual energy intensity (b) annual utility cost (c) PV array cost to achieve NZE and (d) cost difference between the efficiency measure and using PV to achieve the same energy savings

<u>Ground source heat pump system</u>

Fig. A 5 presents the results of the ground source heat pump evaluated in each of the standard energy efficiency models outlined above. Fig. A 5 a compares the annual energy intensity for each case, Fig. A 5 b compares the annual utility cost with no PV system, Fig. A 5 c compares the PV array cost difference to achieve net zero energy over the baseline energy consumption and Fig. A 5 d presents the cost comparison between using a PV array and the efficiency measure to achieve the same level of energy savings. A negative cost difference indicates the PV array is more cost beneficial than the efficiency measure, while a positive cost indicates the efficiency measure is worthwhile implementing to offset the energy consumption to achieve net zero energy. A comparison to the base case with no energy efficiency measures is also provided.

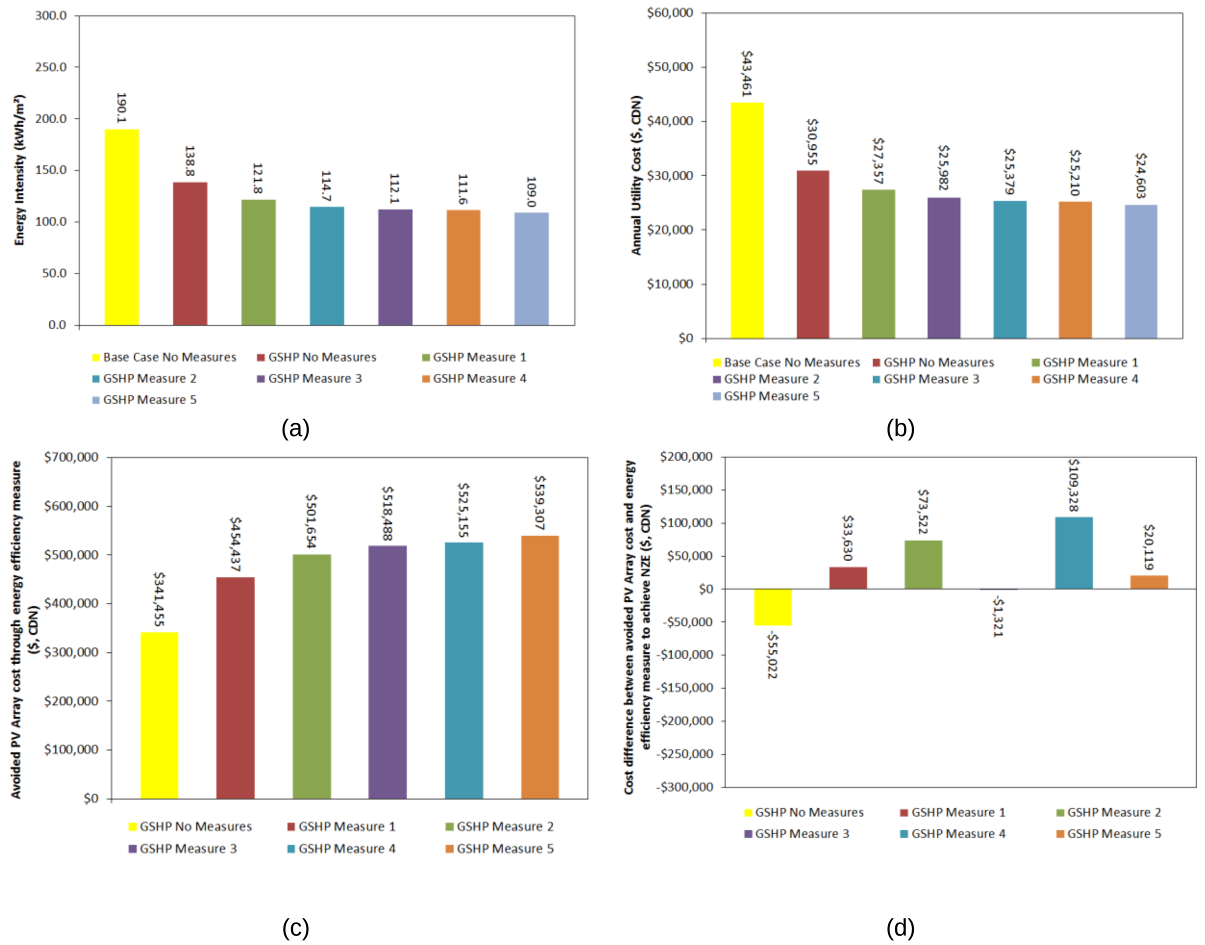


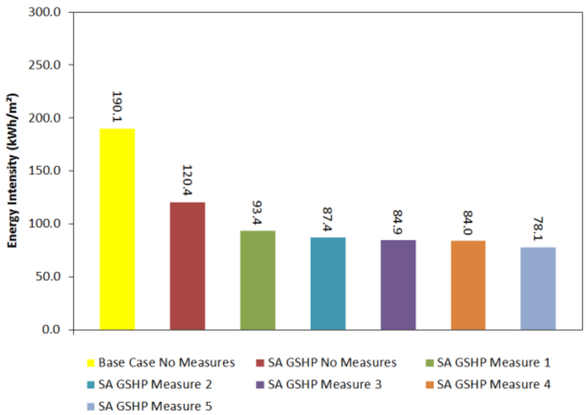
Fig. A 5: Comparison of the ground-source heat pump (a) annual energy intensity (b) annual utility cost (c) PV array cost to achieve NZE and (d) cost difference between the efficiency measure and using PV to achieve the same energy savings

In this scenario, the GSHP alone did not demonstrate a benefit over using a PV array system to meet the same energy saving target. This is attributed to the cost associated with a larger borefield requirement to meet the space heating load. As the space heating load increases through a reduction of interior gains, the GSHP system becomes more cost competitive. Similar to the air to air heat pump systems evaluated the benefit of adding insulation to reduce the building heating load does not have benefit in comparison to using a PV array even with a reduction in the borefield size. However, contrary to the air to air heat pump systems, the use of heat recovery ventilators is beneficial as the lower cost of the energy efficiency measure combined with a reduction in the borefield size, ultimately makes the GSHP system cost beneficial over using a PV array.

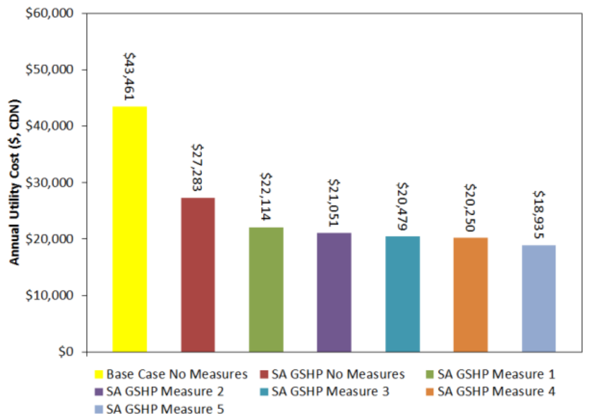
Solar assisted ground source heat pump system

Fig. A 6 presents the results of the ground source heat pump evaluated in each of the standard energy efficiency models outlined above. Fig. A 6 a compares the annual energy intensity for each case, Fig. A 6 b compares the annual utility cost with no PV system, Fig. A 6 c compares the PV array cost difference to achieve net zero energy over the baseline energy consumption and Fig. A 6 d presents the cost comparison between using a PV array and the efficiency measure to achieve the same level of energy savings. A comparison to the base case with no energy efficiency measures is also provided.

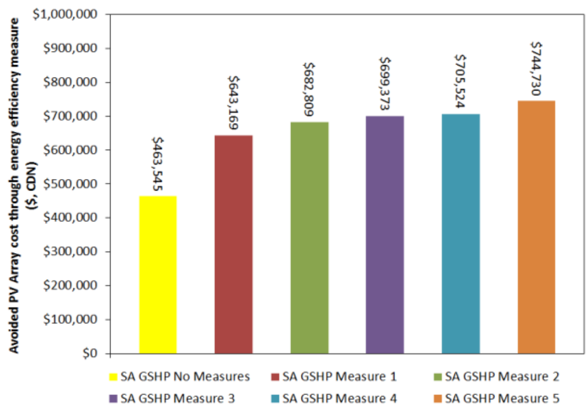
The use of solar energy to reduce the borefield size and meet a portion of the domestic hot water load is beneficial in all cases in comparison to using a PV system to meet the same electricity savings. Similar to the GSHP system, by reducing the borefield size through the use of heat recovery ventilators to meet a portion of the heating load, the greatest cost difference savings is achieved. The high cost of improving the building envelope insulation is unable to compete with the reduced cost of the borefield and use of PV.
Comparing the results of the GSHP system only to the SA GSHP system, it is interesting to see that the use of solar thermal energy has a benefit. By reducing the borefield size and meeting a portion of the domestic hot water load, the incremental cost of the solar thermal system is able to outweigh the cost of a PV array.



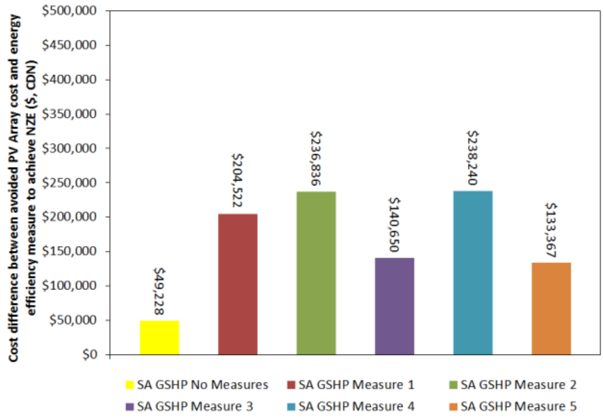
(a)



(b)



(c)



(d)

Fig. A 6: Comparison of the solar assisted ground-source heat pump (a) annual energy intensity (b) annual utility cost (c) PV array cost to achieve NZE and (d) cost difference between the efficiency measure and using PV to achieve the same energy savings

A.2 Detailed results of the Finnish case study

Tab. A 5: Detailed results of the Finnish case study for new detached house

Case	Solar system energy, kWh/m²a			Delivered energy, kWh/m²a						E-value, kWh/m²,a	
	ST[1]	PV		Electricity						$E_{D3(2012)}$	E_{esp}
	Utilized	Produced	Utilized	Heat pump	Aux. heater	Cooling	HVAC-aux. devices	Lighting+ equip.	Total[2]		
Ground source heat pump											
No solar systems	0	0	0	16.7	0.4	0.1	6.1	22.8	46	78.2	78.2
+ST (12m2)	19.0	0	0	14	0.4	0.1	6.1	22.8	43.3	73.7	73.7
+ST (12m2)+PV (18m2)	19.0	20.9	9.8	14	0.4	0.1	6.1	22.8	33.6	57.1	38.1
+ST (12m2)+PV (36m2)	19.0	41.9	12.1	14	0.4	0.1	6.1	22.8	31.2	53.1	2.5
+ST (12m2)+PV (36m2)+LED	19.0	41.9	11.1	13.8	0.4	0.1	6.1	20	29.3	49.9	-2.5
Air-to-water heat pump											
No solar systems	0	0	0	21	1.1	0.1	6.1	22.8	51	86.7	86.7
+ST (12m2)	19.0	0	0	14.6	1.1	0.1	6.1	22.8	44.6	75.9	75.9
+ST (12m2)+PV (18m2)	19.0	20.9	9.6	14.6	1.1	0.1	6.1	22.8	35	59.5	40.3
+ST (12m2)+PV (36m2)	19.0	41.9	11.8	14.6	1.1	0.1	6.1	22.8	32.8	55.8	4.7
Exhaust air heat pump											
No solar systems	0	0	0	39.7	2.7	0.1	6.1	22.8	71.4	121.3	121.3
+ST (12m2)	19.0	0	0	33.6	2.7	0.1	6.1	22.8	65.3	111	111
+ST (12m2)+PV (18m2)	19.0	20.9	10.8	33.6	2.7	0.1	6.1	22.8	54.5	92.6	75.4
+ST (12m2)+PV (36m2)	19.0	41.9	14	33.6	2.7	0.1	6.1	22.8	51.3	87.2	39.8
Air-to-air heat pump											
No solar systems	0	0	0	6.7	38.9	0.1	6	22.8	74.5	126.6	126.6
+ST (12m2)	19.0	0	0	6.7	23.1	0.1	6	22.8	58.6	99.7	99.7
+ST (12m2)+PV (18m2)	19.0	20.9	10.4	6.7	23.1	0.1	6	22.8	48.2	82	64.1
+ST (12m2)+PV (36m2)	19.0	41.9	13.4	6.7	23.1	0.1	6	22.8	45.2	76.8	28.4

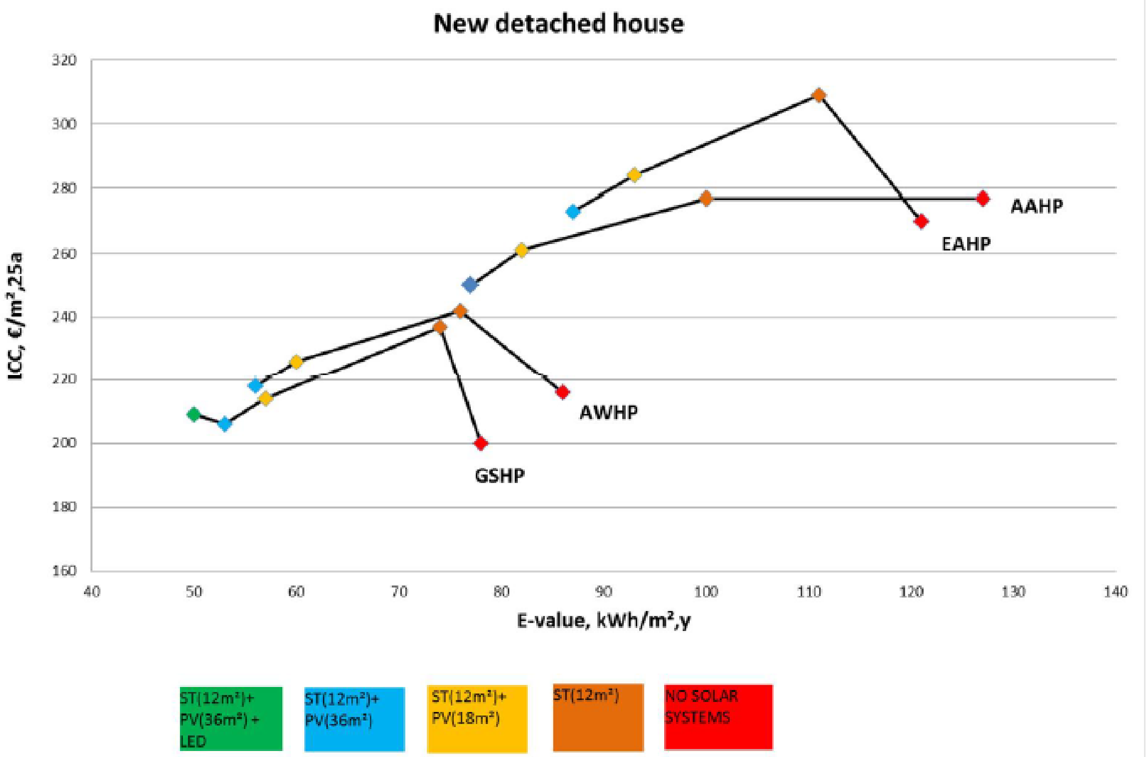


Fig. A 7: Life cycle cost of the variants for new detached house

Tab. A 6: Detailed results of the Finnish case study for new detached house: life cycle costs and E-values

Life cycle costs and E -values of new detached house									
Heated netto area: 180 m2									
	Acquisition cost and life cycle cost €/m²/25 y								E-value
	Acquisition cost	Renewing	Residual value	Capital	Maintenance	Heating energy	Electrical energy	Summary	kWh/m²,v
Ground source heat pump									
No solar systems	86.0	22.0	50.0	58.0	22.0	46.0	74.0	200.0	**78.2**
+ST (12m2)	110.0	46.0	69.0	87.0	35.0	41.0	74.0	237.0	**73.7**
+ST(12m2)+PV (18m2)	114.0	46.0	69.0	91.0	37.0	41.0	47.0	214.0	**57.1**
+ST (12m2)+PV (36m2)	118.0	46.0	69.0	95.0	39.0	41.0	31.0	206.0	**53.1**
+ST(12m2)+PV (36m2)+LED	124.0	46.0	69.0	101.0	39.0	42.0	27.0	209.0	**49.9**
Air-water heat pump									
No solar systems	60.0	16.0	11.0	65.0	22.0	56.0	73.0	216.0	**86.0**
+ST (12m2)	84.0	40.0	30.0	94.0	35.0	40.0	73.0	242.0	**75.9**
+ST(12m2)+PV (18m2)	89.0	40.0	30.0	99.0	38.0	40.0	49.0	226.0	**59.5**
+ST(12m2)+PV (36m2)	94.0	40.0	30.0	104.0	41.0	40.0	33.0	218.0	**55.8**
Exhaust-air heat pump									
No solar systems	55.0	55.0	42.0	68.0	21.0	108.0	73.0	270.0	**121.3**
+ST(12m2)	84.0	79.0	61.0	102.0	34.0	93.0	73.0	309.0	**111.0**
+ST(12m2)+PV (18m2)	89.0	79.0	61.0	107.0	37.0	93.0	47.0	284.0	**92.6**
+ST(12m2)+PV (36m2)	94.0	79.0	61.0	112.0	40.0	93.0	28.0	273.0	**87.2**
Air-air heat pump									
No solar systems	60.0	17.0	12.0	65.0	23.0	116.0	73.0	277.0	**126.6**
+ST(12m2)	89.0	36.0	31.0	94.0	36.0	78.0	69.0	277.0	**99.7**
+ST(12m2)+PV (18m2)	94.00	36.0	31.0	99.0	39.0	78.0	44.0	261.0	**82.0**
+ST(12m2)+PV (36m2)	99.0	36.0	31.0	104.0	42.0	78.0	26.0	250.0	**76.8**

Tab. A 7: Detailed results of the Finnish case study for new apartment building

Case	Solar system energy, kWh/m²,a			Delivered energy, kWh/m²,a								E-number, kWh/m²,a	
	ST[1]	PV		District	Electricity						Total	$E_{D3(2012)}$	E_{exp}
	Utilized	Produced	Utilized	heating	Heat pump	Aux. heater	Cooling	HVAC-aux. devices	Lighting+ equip.	Total[2]	delivered energy		
Ground source heat pump													
No solar systems	0	0	0	0	13.1	0.5	0.1	12	30.6	56.3	56.3	95.7	95.7
+ST (78m2)	15.7	0	0	0	10.2	0.5	0.1	12	30.6	53.3	53.3	90.7	90.7
+ST (78m2)+PV (66m2)	15.7	3.9	3.9	0	10.2	0.5	0.1	12	30.6	49.5	49.5	84.1	84.1
+ST (78m2)+PV (200m2)	15.7	11.7	9.5	0	10.2	0.5	0.1	12	30.6	43.9	43.9	74.6	70.7
+ST (78m2)+PV (200m2)+LED	15.7	11.7	8.9	0	10.9	0.5	0.1	12	25.2	39.9	39.9	67.8	62.9
Air-to-water heat pump													
No solar systems	0	0	0	0	15.3	1.8	0.7	12	30.6	60.4	60.4	102.6	102.6
+ST (78m2)	15.7	0	0	0	11.3	1.8	0.7	12	30.6	56.5	56.5	96	96
+ST (78m2)+PV (66m2)	15.7	3.9	3.9	0	11.3	1.8	0.7	12	30.6	52.6	52.6	89.4	89.4
+ST (78m2)+PV (200m2)	15.7	11.7	9.8	0	11.3	1.8	0.7	12	30.6	46.7	46.7	79.4	76.1
Distric heating													
No solar systems	0	0	0	48.4	0	0	0.7	12	30.6	43.3	91.8	107.6	107.6
+ST (78m2)	15.7	0	0	32.2	0	0	0.7	12	30.6	43.3	75.6	96.2	96.2
+ST (78m2)+PV (66m2)	15.7	3.9	3.9	32.2	0	0	0.7	12	30.6	39.5	71.7	89.7	89.7
+ST (78m2)+PV (200m2)	15.7	11.7	9.5	32.2	0	0	0.7	12	30.6	33.8	66.1	80.1	76.3

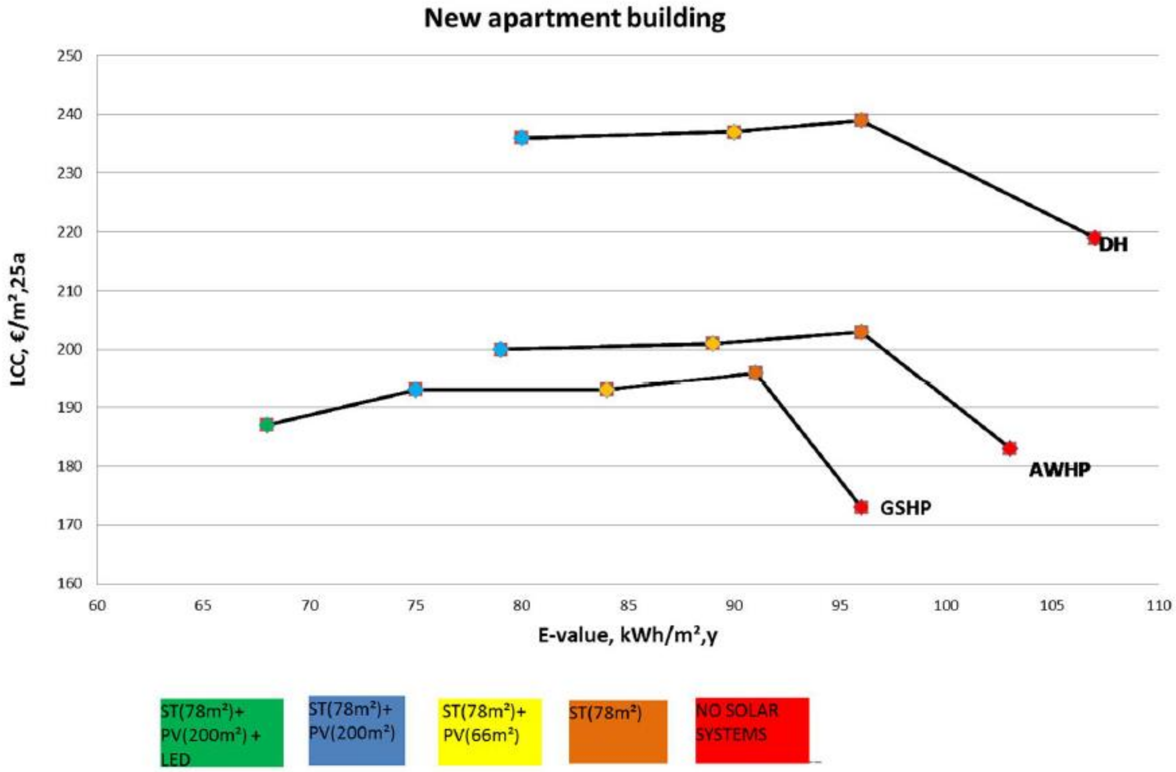


Fig. A 8: Life cycle cost of the variants for new apartment building

Tab. A 8: Detailed life-cycle cost calculation of the Finnish case study for the new apartment building: life cycle costs and E-values

Life cycle costs and E -values of new apartment building

Heated netto area: 3571 m2

	Acquisition cost and life cycle cost €/m²/25 y								E-value
	Acquisition cost	Renewing	Residual value	Capital	Maintenance	Heating energy	Electrical energy	Summary	kWh/m²,v
Ground source heat pump									
No solar systems	25.0	7.0	15.0	17.0	13.0	33.0	110.0	173.0	**95.7**
+ST(78m2)	41.0	23.0	27.0	37.0	23.0	26.0	110.0	196.0	**90.7**
+ST(78m2)+PV (66m2)	45.0	23.0	27.0	41.0	26.0	26.0	100.0	193.0	**84.1**
+ST(78m2)+PV (200m2)	53.0	23.0	27.0	49.0	32.0	26.0	86.0	193.0	**74.6**
+ST(78m2)+PV (200m2)+LED	60.0	23.0	27.0	56.0	32.0	26.0	74.0	187.0	**67.8**
Air-water heat pump									
No solar systems	20.0	7.0	5.0	22.0	8.0	44.0	109.0	183.0	**102.6**
+ST(78m2)	36.0	23.0	25.0	34.0	18.0	34.0	109.0	203.0	**96.0**
+ST(78m2)+PV (66m2)	40.0	23.0	25.0	38.0	21.0	34.0	100.0	201.0	**89.4**
+ST(78m2)+PV (200m2)	48.0	23.0	25.0	46.0	27.0	34.0	85.0	200.0	**79.4**
District heating									
No solar systems	27.0	3.0	14.0	16.0	2.0	91.0	110.0	219.0	**107.6**
+ST(78m2)	47.0	23.0	23.0	47.0	14.0	60.0	110.0	239.0	**96.2**
+ST(78m2)+PV (66m2)	51.0	23.0	23.0	51.0	17.0	60.0	101.0	237.0	**89.7**
+ST(78m2)+PV (200m2)	59.0	23.0	23.0	59.0	23.0	60.0	86.0	236.0	**80.1**

Tab. A 9: Detailed results of the Finnish case study for 1960s apartment building

Case	Solar system energy, kWh/m²,a			Delivered energy, kWh/m²,a						E-number, kWh/m²,a	
	ST[1]	PV		District	Electricity				Total	$E_{D3(2012)}$	E_{esp}
	Utilized	Produced	Utilized	heating (aux. heating)	Heat pump	HVAC-aux. devices	Lighting + equipment	Total[2]	delivered energy		
Distric heating											
No solar systems	0	0	0	144.3	0	5.5	30.6	36.1	180.4	162.4	162.4
+ST (90m2)	17.2	0	0	126.6	0	5.5	30.6	36.1	162.7	150	150
+ST (90m2)+PV (66m2)	17.2	3.7	3.7	126.6	0	5.5	30.6	32.5	159	143.8	143.7
+ST (90m2)+PV (200m2)	17.2	11.3	8	126.6	0	5.5	30.6	28.1	154.7	136.4	130.7
Exhaust air heat pump											
No solar systems	0	0	0	74.3	15.5	5.3	30.6	51.5	125.8	139.5	139.5
+ST (90m2)	17.2	0	0	74.3	11.8	5.4	30.6	47.8	122.1	133.2	133.2
+ST (90m2)+PV (66m2)	17.2	3.7	3.7	74.3	11.8	5.4	30.6	44	118.4	126.9	126.9
+ST (90m2)+PV (200m2)	17.2	11.3	8.4	74.3	11.8	5.4	30.6	39.4	113.7	119	114
Ground source heat pump											
No solar systems	0	0	0	2.7	39.9	5.6	30.6	76.1	78.8	131.2	131.2
No solar systems+LED	0	0	0	3.1	40.8	5.6	25.2	71.6	74.6	123.8	123.8
+ST (90m2)	17.2	0	0	2.7	36.7	5.6	30.6	72.9	75.6	125.8	125.8
+ST (90m2)+PV (66m2)	17.2	3.7	3.7	2.7	36.7	5.6	30.6	69.2	71.8	119.4	119.4
+ST (90m2)+PV (200m2)	17.2	11.3	8.6	2.7	36.7	5.6	30.6	64.3	67	111.2	106.5
Air-to-water heat pump											
No solar systems	0	0	0	17.1	38.5	5.5	30.6	74.6	91.7	138.8	138.8
+ST (90m2)	17.2	0	0	17.1	32.9	5.6	30.6	69.1	86.1	129.4	129.4
+ST (90m2)+PV (66m2)	17.2	3.7	3.7	17.1	32.9	5.6	30.6	65.3	82.4	123	123
+ST (90m2)+PV (200m2)	17.2	11.3	8.6	17.1	32.9	5.6	30.6	60.5	77.5	114.8	110.1
+ST (90m2)+PV (200m2)+LED	17.2	11.3	8	18	33.4	5.6	25.2	56.2	74.2	108.2	102.5

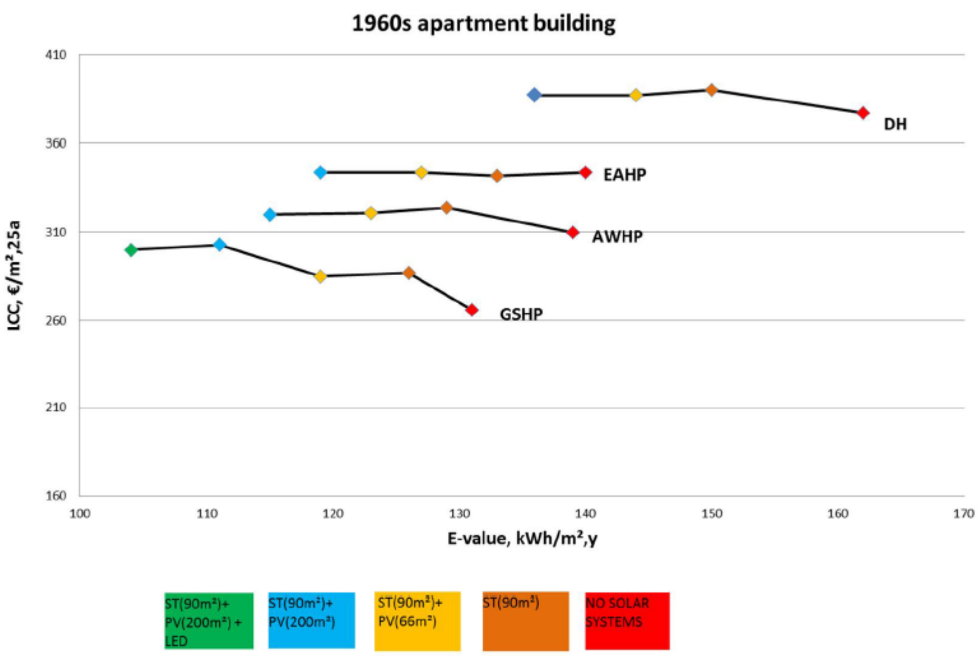


Fig. A 9: Life cycle cost of the variants for 1960s apartment building

Tab. A 10: Detailed results of the Finnish case study for 1960s apartment building

Life cycle costs and E -values1960s apartment building									
Heated netto area: 3697 m2									
				Capital = acquisition + renewing - residual value					
	Acquisition cost and life cycle cost €/m²/25 y								E-value
	Acquisition cost	Renewing	Residual value	Capital	Maintenance	Heating energy	Electrical energy	Summary	kWh/m²,v
District heating									
No solar systems	22.0	0.0	11.0	11.0	4.0	270.0	92.0	377.0	**162.4**
+ST(90m2)	47.0	25.0	31.0	41.0	20.0	237.0	92.0	390.0	**150.0**
+ST(90m2)+PV (66m2)	51.0	25.0	31.0	45.0	22.0	237.0	83.0	387.0	**143.8**
+ST(90m2)+PV (200m2)	59.0	25.0	31.0	53.0	26.0	237.0	71.0	387.0	**136.4**
Exhaust-air heat pump									
No solar systems	61.0	37.0	33.5	64.5	9.0	179.0	91.0	343.5	**139.5**
+ST(90m2)	86.0	62.0	53.5	94.5	25.0	169.0	91.0	341.5	**133.2**
+ST(90m2)+PV (66m2)	90.0	62.0	53.5	98.5	27.0	169.0	82.0	343.5	**126.9**
+ST(90m2)+PV (200m2)	98.0	62.0	53.5	96.5	31.0	169.0	70.0	343.5	**119.0**
Ground source heat pump									
No solar systems	81.0	9.0	44.0	46.0	23.0	105.0	92.0	266.0	**131.2**
No solar systems+LED	87.0	9.0	44.0	52.0	23.0	108.0	80.0	263.0	**123.8**
+ST(90m2)	101.0	20.0	60.0	61.0	36.0	98.0	92.0	287.0	**125.8**
+ST(90m2)+PV (66m2)	106.0	20.0	60.0	66.0	38.0	98.0	83.0	285.0	**119.4**
+ST(90m2)+PV (200m2)	116.0	34.0	60.0	90.0	45.0	98.0	70.0	303.0	**111.2**
Air-water heat pump									
No solar systems	64.0	8.0	8.0	64.0	24.0	130.0	92.0	310.0	**138.8**
+ST (90m2)	84.0	19.0	24.0	79.0	37.0	116.0	92.0	324.0	**129.4**
+ST(90m2)+PV (66m2)	88.0	19.0	24.0	83.0	39.0	116.0	83.0	321.0	**123.0**
+ST(90m2)+PV (200m2)	96.0	19.0	24.0	91.0	43.0	116.0	70.0	320.0	**114.8**
+ST(90m2)+PV (200m2)+LED	102.0	19.0	24.0	97.0	43.0	119.0	58.0	317.0	**108.2**

A.3 Boundary conditions for large office in Japanese case study

Tab. A 11: Simulation condition set to calculate air-conditioning loads of zero-energy-oriented building

		Next-generation energy-efficient building
Envelope performance	Exterior walls (typical member)	**Heat transmission coefficient: 0.4 W/m²K** Aluminum 1mm Unsealed air layer Polyethylene foam plate 30 mm Concrete 150 mm Polyethylene foam plate 60 mm Air layer Aluminum 1 mm
	Roof (typical member)	**Heat transmission coefficient: 0.55 W/m²K** Concrete 150 mm Polyethylene foam plate 60 mm Air layer Aluminum 1 mm **Roof solar absorptance: 0.5 (except Dec. to Mar.)** Roof greening
	Interior walls	**Heat transmission coefficient: 1.06 W/m²K** Gypsum board 16 mm Polyethylene foam plate 20 mm Unsealed air layer Gypsum board 16 mm
	Window (typical member)	**Shading coefficient: 0.24** **Heat transmission coefficient: 1.6 W/m²K** AFW Double glass window with integral light colour blind Airflow: 11 l/(m・s) **Shading coefficient: 0.34** **Heat transmission coefficient: 2.6 W/m²K** Top side window (above light shelf) Highly-reflective insulating multilayer glass (low-E) Light colour blind
	Eaves	1.2 m
	Air filtration	0.1 times/h
	Lighting	2 W/m² 500 lx (LED＋Lighting control＋Blind control)
Heat generation of equipment		Daytime: 5 W/m² Nighttime: 1.25 W/m²
Natural ventilation		Available*1
Room temperature/humidity settings		Cooling: 23-28 ℃, 40%~60% Heating: 19-26 ℃, 40%~60%
Outside air handling		Total heat exchanger: Available (Efficiency: Sensible heat 75%, Latent heat 40%) Outside air intake variable control*2: Available

*1 This is only for the cooling period (June–September). Outdoor air temperature shall be at least 18 ºC. Space air enthalpy is greater than outdoor air enthalpy (the number of air ventilations per hour shall be 1 time/h for every 1 m/s increase in outdoor air velocity, but not exceeding 6 times/h in total). If the amount of natural ventilation is less than that required, mechanical ventilation (under CO_2 concentration control) is supposed to make up for the shortfall.

*2 The amount of outdoor air intake is variably controlled depending on in-room headcount per unit area (under CO_2 concentration control).

On the south side of its North Tower, the next-generation energy-efficient building receives the benefit of the eave effect from the South Tower. The heat load of the building is calculated, based on solar altitude and the eave length at the time of the year. Tab. A 12 shows this relationship.

Tab. A 12: Condition for eave length on south side of North Tower

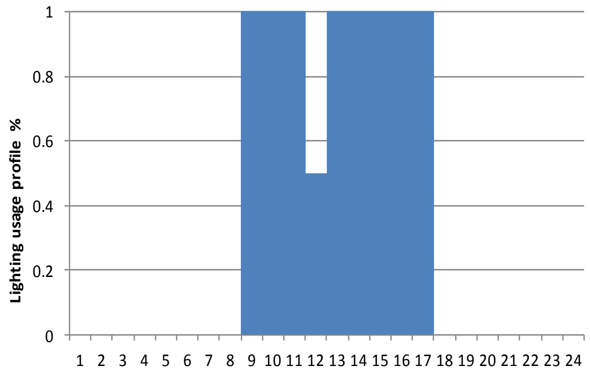
	June and July (cooling)	August and September (cooling)	December to March (heating)
5F			
4F	1.2 m		
3F			
2F		9 m	
1F			

2) Other common simulation conditions for annual heat load calculations

(1) Computation tool: Micro-HASP/TES ver2.6
(2) Weather data: HASP standard annual weather data (Tokyo)
(3) Periods and time slots during which air conditioning is turned on

Summertime: June 1 through September 31, wintertime: December 1 through March 31
Intermediate periods (during which the heat source is not taken in): April 1 through May 31, October 1 through November 30
*Note that the next-generation energy-efficient building is air-conditioned even during these intermediate periods (at a room temperature between 19 ºC and 28 ºC and at a relative humidity between 40% and 60%).
Air conditioning is turned on from 09:00 to 17:59 on weekdays (precooling starts at 08:00 and preheating starts at 07:00). This is not applied to weekends and holidays.

(4) Outdoor air intake: 30 m^3/h-person (no outdoor air is taken in during the time slots for pre-cooling or preheating)
(5) Infiltration draft: Taken into account
(6) Scheduled heat generation from lighting and office automation equipment; scheduled in-room headcount per unit area



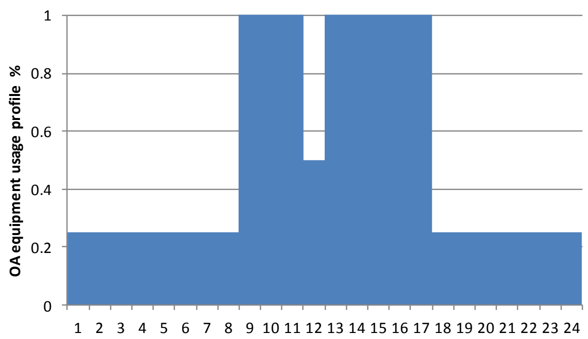


Fig. A 10: Various schedules for heat load calculations (a) scheduled lighting utilisation rates (b) Scheduled office automation equipment capacity utilisation rates

Fig. A 10 shows various schedules for heat load calculations. The schedules for in-room headcount per unit area are similar to the schedules for lighting utilisation rates.

3) Results of annual heat load calculations

(1) Peak heat load; Unit: kW

Precooling for 1 h, preheating for 3 h, the risk level of room conditions is 5 %, and the load factor for internal heat generation is 100 % for cooling and 30% for heating.

Tab. A 13: Peak equipment load (space load + outdoor air load) Unit: kW (Part 1)

Zone	Area	Heat load for room			Heat load for fresh air			Heat load, total		
	m²	kW	kW	kW	kW	kW	kW	kW	kW	kW
Perimeter S①	172.8	8.18	2.05	10.23	.52	3.68	4.2	10.65	2.93	13.58
Interior S	656.6	7.31	2.01	9.32	0.7	3.48	4.19	7.12	5.76	12.89
Perimeter N①	172.8	8.18	2.05	10.23	0.52	3.68	4.2	10.65	2.93	13.58
Perimeter S②1F	172.8	8.49	5.2	13.68	1.97	9.74	11.71	6.69	17.13	23.82
Perimeter S②2F	172.8	8.49	5.2	13.68	1.97	9.74	11.71	6.69	17.13	23.82
Perimeter S②3F	172.8	8.41	1.99	10.40	0.7	3.48	4.19	9.02	4.86	13.88
Perimeter S②4F	172.8	7.61	2.03	9.64	0.7	3.48	4.19	9.02	4.86	13.26
Interior N	656.6	7.61	2.03	9.64	0.7	3.48	4.19	9.02	4.86	13.26
Perimeter N②	172.8	8.41	1.99	10.40	0.7	3.48	4.19	9.02	4.86	13.88
Perimeter S①, Top floor	172.8	9.84	5.2	15.04	1.97	9.74	11.71	12.32	12.21	24.53
Interior S, Top floor	656.6	9.84	5.2	15.04	1.97	9.74	11.71	12.32	12.21	24.53
Perimeter N①, Top floor	172.8	4.28	2.03	6.31	0.7	3.48	4.19	5.17	4.79	9.96
Perimeter S②, Top floor	172.8	4.28	2.03	6.31	0.7	3.48	4.19	5.17	4.79	9.96
Interior N, Top floor	656.6	4.28	2.03	6.31	0.7	3.48	4.19	5.17	4.79	9.96
Perimeter N②, Top floor	172.8	7.31	2.01	9.32	0.7	3.48	4.19	7.12	5.76	12.89

Tab. 37: Peak equipment load (space load + outdoor air load) Unit: kW (Part 2)

Zone	Area	Heat load for room			Heat load for fresh air			Heat load, total		
	m²	kW	kW	kW	kW	kW	kW	kW	kW	kW
Perimeter S①	172.8	-5.65	-0.27	-5.92				-8.97	-3.28	-12.25
Interior S	656.6	-8.18	-0.41	-8.59				-10.89	-3.29	-14.18
Perimeter N①	172.8	-5.65	-0.27	-5.92				-8.97	-3.28	-12.25
Perimeter S②1F	172.8	-5.47	-0.58	-6.05				-12.46	-8.65	-21.11
Perimeter S②2F	172.8	-5.47	-0.58	-6.05				-12.46	-8.65	-21.11
Perimeter S②3F	172.8	-7.91	-0.27	-8.18				-10.99	-3.29	-14.28
Perimeter S②4F	172.8	-10.49	-0.41	-10.90				-13.23	-3.29	-16.52
Interior N	656.6	-10.49	-0.41	-10.90				-13.23	-3.29	-16.52
Perimeter N②	172.8	-7.91	-0.27	-8.18				-10.99	-3.29	-14.28
Perimeter S①, Top floor	172.8	-12.76	-0.58	-13.34				-21.0	-8.65	-29.66
Interior S, Top floor	656.6	-12.76	-0.58	-13.34				-21.0	-8.65	-29.66
Perimeter N①, Top floor	172.8	-11.39	-0.41	-11.79				-15.13	-3.29	-18.42
Perimeter S②, Top floor	172.8	-11.39	-0.41	-11.79				-15.13	-3.29	-18.42
Interior N, Top floor	656.6	-11.39	-0.41	-11.79				-15.13	-3.29	-18.42
Perimeter N②, Top floor	172.8	-8.18	-0.41	-8.59				-10.89	-3.29	-14.18

Tab. A 14: Peak equipment load (space load + outdoor air load); Unit: kW

	Room heat load	Fresh air load	Total heat load	Outdoor air	
				Temp.	Humid.
Cooling	329.7	165.2	494.9	30.6	16.5
Heating	-116.8	-335.1	-451.9	4.1	2.0

*The peak cooling load took place at 11:00 on August 14, and the peak heating load at 10:00 on January 17

(2) Annual equipment load fluctuations

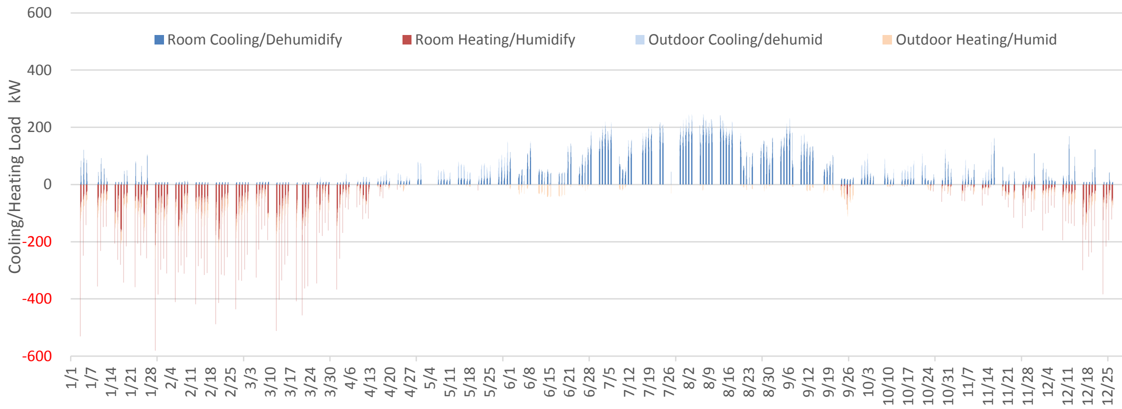


Fig. A 11: Annual heat load fluctuations

During the cooling period, the system is involved only in cooling and dehumidifying the building, not in processing heating and humidification loads. It is considered that internal heat generation and other internal heat sources can offset heating load during the cooling period. The cumulative humidification load over the period is estimated to be 4,044 kWh.
During the heating period, the system is involved only in heating and humidifying the building, but not in processing cooling and dehumidification loads. It is assumed that outdoor air cooling can offset cooling load during the heating period.
Air-conditioning operation during the intermediate periods generates cooling load that is included in the space load (cooling requirement) as well as heating load that is included in the outdoor air load (heating and humidification requirements).
It is assumed that outdoor air cooling can process cooling load that is included in the space load. For this reason, this study ignores the cooling load. But the system should process heating load that is included in the space load (heating requirement only). It is considered that the outdoor air heating requirement should be satisfied. During the intermediate periods, the outdoor air load (heating and humidification) comes to 65,177 kWh, and the space load (heating requirement) comes to 6,245 kWh.

Tab. A 15: Summary of annual cumulative equipment loads (space load + outdoor air load) Unit: kWh

Season	Room Load				Fresh Air Load			
	Cooling	Heating	Dehumid	Humid	Cooling	Heating	Dehumid.	Humid.
Cooling	64,415	-734	17,549	0	613	-1,877	5,487	-4,339
Heating	4,965	-68,327	4,333	-1,375	0	-8,632	214	-12,039
Mid	4,629	-9,009	10,321	-281	962	-2,285	12,479	-1,650
Year	64,415	-77,336	17,549	-1,375	1,575	-10,917	5,487	-13,689

*If cooling and heating loads coexist in each zone, heat loads are listed separately without being added up.

Tab. A 16: Summary of annual cumulative equipment loads Unit: kWh

Season	Equipment Load				Total
	Cooling	Dehumid	Heating	Humid	
Cooling	65,028	23,036	0	0	88,064
Heating	0	0	76,958	13,414	90,373
Mid	0	0	11,294	1,650	12,944
Year	65,028	23,036	88,253	15,064	191,381

Total annual cumulative equipment load comes to 400,498 kWh (1,441.8 GJ).

4) Forecasts for annual energy consumption and balance in energy creation

(1) Annual energy consumption

Tab. A 17: Calculated annual energy consumption (with some assumptions) Unit: MJ/(m²·a)

	HVAC*1	Lighting*2	OA/Plug*3	Power*4	DW	etc.	Total amount*5
General Build.	804	572	507	179	17	106	2,185(1,678)
Energy-efficient building	290	191	444	179	17	106	1,227(783)
Next-generation energy-efficient building	106	29	132	90	0	53	410(278)

*1 This is calculated from annual cumulative heat source load. It is assumed that the three types of buildings have a specific coefficient of performance (SCOP) of 0.5, 0.9, and 1.2, respectively.
*2 Conventional (18 W/m² in the daytime and 2 W/m² in the night-time), energy-efficient (6 W/m² and 1 W/m²), and next-generation energy-efficient (1.4 W/m² and 0 W/m²)
*3 Conventional (18 W/m² in the daytime and 2 W/m² in the night-time), energy-efficient (15 W/m² and 2 W/m²), and next-generation energy-efficient (3.5 W/m² and 0.9 W/m²)
*4 This includes energy consumption for ventilation, water supply and drainage, and elevators
*5 Numbers in parentheses represent the total power consumption less power consumption through electric outlets for office automation equipment.

(2) Forecasted annual energy creation

- Installation area (on the south-facing roof): Approximately 1,140 m² (about 51% of the total roof surface)
- 20% generation efficiency: Targeted for 2017 in the NEDO roadmap.
- Annual power generation: 279 MWh per year (278 MJ/m², a)

*3.73 kWh/m² per day (monthly mean quantity of solar radiation, NEDO), 10% energy loss through power conditioners and the like

Tab. A 18: Estimated energy consumption by application. Source: past reference

	HVAC	Lighting	OA/Plug	Power*1	DW	etc	Total amount
Percentage for each usage purpose*2	43.1%	21.3%	21.1%	8.6%	0.8%	5.1%	100.0 %
Energy consumption intensity (MJ/(m²·a))	899	445	440	179	17	106	2,087*3

*1 This includes energy consumption for ventilation, water supply and drainage, and elevators
*2 Source: Energy Conservation Center, Japan (ECCJ)
*3 DECC's average value for office buildings in region G (2006–2010) with a total building floor area 10,000 m² or larger and smaller than 30,000 m²)

A.4 Detailed results of the Swedish case study

Tab. A 19: Type houses developed as a basis for the heat pump prototypes

	Single family nZEB	**Multi family nZEB**
Nr of houses/apartments	1	40
Nr of residents	4	80
Household electricity	4,800 kWh/a	112 MWh/a
Specific, household electricity	30 kWh/(a·m²·A_{temp})	28 kWh/(a·m²·A_{temp})
Other building electricity	-	8 MWh/a
Specific other building electricity	-	2 kWh/(a·m²·A_{temp})
Tempered floor area, A_{temp}	160 m²	4,000 m²*
Inner area building envelope, A_{om}	390 m²	6,000 m²
Nominal airflow; 0,35 dm³/(s m² A_{temp})	56 dm³/s	1,4 m³/s
Average heat transfer coefficient (building), U_m	0,2 W/(K·m²·A_{om})	0,3 W/(K·m²·A_{om})
Air tightness (at ±50 Pa)	0,2 dm³/(s·m²·A_{om})	0,3 dm³ /(s·m²·A_{om})
Spec. heat loss from ventilation (at dimensioning winter outdoor temperature)	0,14 W/(K·m²·A_{temp})	0,14 W/(K·m²·A_{temp})
Heat loss from ventilation	22,4 W/K	0,56 kW/K
Temperature efficiency, mechanical supply and exhaust air with heat exchange	about 80%	about 80%
Specific fan power (SFP) **	≤ 1,5 W/(dm³/s)	≤ 1,3 kW/(m³/s)
Fan power	≤ 85 W	≤ 1,8 kW
Specific energy to fan	≤ 4,5 kWh/(a·m²·A_{temp})	≤ 4,0 kWh/(a·m²·A_{temp})
Energy to fan	≤ 700 kWh/a	≤ 16 MWh/a
Specific pump power for heat- and DHW distribution ***	≤ 0,3 W/(m² A_{temp})	≤ 0,3 W/(m² A_{temp})
Pump power for heat- and DHW distribution	≤ 50 W	≤ 1,2 kW
Specific pump energy for heat- and DHW distribution	≤ 1,5 kWh/(a·m²·A_{temp})	≤ 1,3 kWh/(a·m²·A_{temp})
Pump energy for heat- and DHW distribution	≤ 250 kWh/a	≤ 5 MWh/a
Specific heat loss DHW/Circulation System at stand-by	≤ 0,5 W/(m²·A_{temp})	≤ 0,3 W/(m²·A_{temp})
Heat loss DHW/Circulation System at stand-by	≤ 80 W	≤ 1,2 kW
Specific DHW demand/Circulation System at stand-by	≤ 4,5 kWh/(a·m²·A_{temp})	≤ 2,5 kWh/(a·m²·A_{temp})
DHW demand /Circulation System at stand-by	≤ 700 kWh/a	≤ 10 MWh/a
Specific heat demand in climate zone III	36,5 kWh/(a·m²·A_{temp})	34,5 kWh/(a·m²·A_{temp})
Heat demand in climate zone III	5,800 kWh/a	138 MWh/a
Maximum power demand, heating system	4 kW	75 kW
Specific DHW demand (excl. heat losses)	21,5 kWh/(a·m²·A_{temp})	17,5 kWh/(a·m²·A_{temp})
DHW demand (excl. heat losses)	3,400 kWh/a	70 MWh/a
Net specific energy demand, incl. household electricity	100 kWh/(a·m²·A_{temp})	90 kWh/(a·m²·A_{temp})
Net energy demand, incl. household electricity	16,000 kWh/a	360 MWh/a
Specific energy use (bought energy), incl. household el. ****	60 kWh/(a·m²·A_{temp})	55 kWh/(a·m²·A_{temp})
Energy use (bought energy), incl. household el.	9,600 kWh/a	220 MWh/a
Specific energy use (bought energy), excl. household el.	30 kWh/(a·m²·A_{temp})	27 kWh/(a·m²·A_{temp})
Energy use (bought energy), excl. household el.	4,800 kWh/a	108 MWh/a

*) incl. stairs etc. (i.e. about 80 m²/apartment)
**) excl. fan in outdoor part if air to water heat pump
***) excl. pump for brine
****) excl. on site produced electricity from PV

A.5 Cost overview Swiss case studies of MINERGIE-A®-buildings

Costs SFH				
	15 kWh/(m²*a	35 kWh/(m²*a	55 kWh/(m²*a)	
building technology				
HP a/w	17'600	19'800	22'000	[Fr.]
HP s/w	24'000	27'000	30'000	[Fr.]
Pelletheizung	21'600	24'300	27'000	[Fr.]
Biogasheizung	17'600	19'800	22'000	[Fr.]
Erdgasheizung	17'600	19'800	22'000	[Fr.]
WW-Speicher	3'000	3'000	3'000	[Fr.]
Komfortlüftung	18'000	18'000	18'000	[Fr.]
solar thermal and PV				
PV pro m2	500	500	500	[Fr./m²]
ST + Speicher	10'000	10'000	10'000	[Fr.]
ST pro m2	500	500	500	[Fr./m²]
energy costs				
electricity purchase	0.18	0.18	0.18	[Fr./kWh]
electricity sale	0.18	0.18	0.18	[Fr./kWh]
electricity green	0.25	0.25	0.25	[Fr./kWh]
pellets	0.075	0.075	0.075	[Fr./kWh]
heating oil	0.095	0.095	0.095	[Fr./kWh]
domestic gas	0.095	0.095	0.095	[Fr./kWh]
biogas	0.185	0.185	0.185	[Fr./kWh]
others				
interest	2.5	2.5	2.5	[%]
lifetime	25	25	25	[a]
maintenance	0.02	0.02	0.02	[-]
storage				
150 l	200 l	400 l	800 l	
3'000	3'500	4'500	5'500	[Fr.]

Costs MFH				
	15 kWh/(m²*a)	35 kWh/(m²*a)	55 kWh/(m²*a)	
building technolgy				
HP a/w	24'000	32'000	40'000	[Fr.]
HP s/w	33'000	44'000	55'000	[Fr.]
pellet-heating	27'000	36'000	45'000	[Fr.]
biogas-heating	18'000	24'000	30'000	[Fr.]
gas-heating	18'000	24'000	30'000	[Fr.]
storage hot w ater	9'000	9'000	9'000	[Fr.]
ventilation	150'000	150'000	150'000	[Fr.]
district heating	15'000	20'000	25'000	[Fr.]
CHP	30'000	40'000	50'000	[Fr.]
solar thermal and PV				
PV	450	450	450	[Fr./m²]
solar thermal fix costs	10'000	10'000	10'000	[Fr.]
solar thermal	500	500	500	[Fr./m²]
energy costs				
electricity purchase	0.18	0.18	0.18	[Fr./kWh]
electricity sale	0.18	0.18	0.18	[Fr./kWh]
electricity green	0.25	0.25	0.25	[Fr./kWh]
pellets	0.075	0.075	0.075	[Fr./kWh]
heating oil	0.095	0.095	0.095	[Fr./kWh]
domestic gas	0.095	0.095	0.095	[Fr./kWh]
biogas	0.185	0.185	0.185	[Fr./kWh]
domestic gas	0.082	0.082	0.082	[Fr./kWh]
others				
interest	2.5	2.5	2.5	[%]
lifetime	25	25	25	[a]
maintenance	0.02	0.02	0.02	[-]
storage				
300 l	500 l	1500 l	4000 l	
4'000	5'000	7'000	10'000	[Fr.]
50 m^3	100 m^3	200 m^3	300 m^3	
60'000	100'000	140'000	180'000	[Fr.]

Office Building A				
	15 kWh/(m²*a)	35 kWh/(m²*a)	55 kWh/(m²*a)	
building technology				
HP a/w	19'200	25'600	32'000	[Fr.]
HP s/w	26'400	35'200	44'000	[Fr.]
CHP heating oil	21'600	28'800	36'000	[Fr.]
CHP biogas	21'600	28'800	36'000	[Fr.]
CHP domestic gas	21'600	28'800	36'000	[Fr.]
district heating	12'000	16'000	20'000	[Fr.]
ventilation	0	0	0	[Fr.]
PV	450	450	450	[Fr./m²]
storage	4'000	4'000	4'000	[Fr.]
energy costs				
electricity purchase	0.18	0.18	0.18	[Fr./kWh]
electricity sale	0.18	0.18	0.18	[Fr./kWh]
electricity green	0.25	0.25	0.25	[Fr./kWh]
heating oil	0.095	0.095	0.095	[Fr./kWh]
domestic gas	0.095	0.095	0.095	[Fr./kWh]
biogas	0.185	0.185	0.185	[Fr./kWh]
district heating	0.082	0.082	0.082	[Fr./kWh]
others				
interest	2.5	2.5	2.5	[%]
lifetime	25	25	25	[a]
maintenance	0.02	0.02	0.02	[-]

Office Building B				
	15 kWh/(m²*a)	35 kWh/(m²*a)	55 kWh/(m²*a)	
building technology				
HP a/w	24'000	32'000	40'000	[Fr.]
HP s/w	33'000	44'000	55'000	[Fr.]
CHP heating oil	27'000	36'000	45'000	[Fr.]
CHP biogas	27'000	36'000	45'000	[Fr.]
CHP domestic gas	27'000	36'000	45'000	[Fr.]
district heating	15'000	20'000	25'000	[Fr.]
ventilation	0	0	0	[Fr.]
PV	450	450	450	[Fr./m²]
storage	5'000	5'000	5'000	[Fr.]
energy costs				
electricity purchase	0.18	0.18	0.18	[Fr./kWh]
electricity sale	0.18	0.18	0.18	[Fr./kWh]
electricity green	0.25	0.25	0.25	[Fr./kWh]
heating oil	0.095	0.095	0.095	[Fr./kWh]
domestic gas	0.095	0.095	0.095	[Fr./kWh]
biogas	0.185	0.185	0.185	[Fr./kWh]
district heating	0.082	0.082	0.082	[Fr./kWh]
others				
interest	2.5	2.5	2.5	[%]
lifetime	25	25	25	[a]
maintenance	0.02	0.02	0.02	[-]

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