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Waste heat utilization and smart energy system of combined ice and swimming halls

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Aalto University
School of Engineering
Department of Mechanical Engineering

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Espoo, 13.03.2019
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Abstract of master's thesis

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The building sector consumes about 40 % of global primary energy and contributes to 30 % of global CO_2 emissions. The building sector has a great potential on reducing global energy use and emissions by improving overall energy efficiency and using more renewable energy sources instead of fossil fuels. Building sector in Finland includes about 220 ice halls and 280 swimming halls, which are easily overlooked as significant energy consumers and CO_2 emission producers. The Ministry of the Environment of Finland has set no limits for the energy consumption of these type of buildings. The main potential energy saving and emission reducing measures for these type of buildings are waste heat recovery and demand response. Waste heat recovery reduces purchased energy in ice and swimming halls, since it utilizes heat from ice refrigeration, sewage water, dehumidification and exhaust air. Demand response reduces the peak-load of electricity or district heat grid, which in turn reduces emissions, since the energy for peaks is produced with high-polluting plants.

The objectives of this thesis were to analyze the potential energy and cost saving potential of waste heat recovery and smart control of energy system in a combined energy system of ice and swimming halls. This thesis examines a case in Pirkkola (Helsinki), which includes an old existing medium-sized swimming hall and a new training ice hall. This study was carried out by dynamic building energy simulations and post-processing of the simulation results. The models of the ice and swimming halls and some specific component models of the energy systems were built. The waste heat recovery was complemented with short-time storing of heat. Measures for smart control of energy systems consisted of demand response of electricity and district heat and a smart exhaust air heat pump (EAHP). A rule-based demand response algorithm was introduced for control of electricity and district heat. A smart EAHP was built for the swimming hall, which adjusts the temperature set point of exhaust air to match the waste heat to the heat demands.

This study found out that 99 % of purchased district heat in the ice hall could be replaced by waste heat, while the total electricity demand in the ice hall increases only by 9 %. By transferring excess heat from the ice hall to the swimming hall and by utilizing waste heats of the swimming hall, the total purchased district heat in the swimming hall is reduced by 72 %, while purchased electricity in the swimming hall is increased by 37 %. Demand response of electricity for ice refrigeration decreases the total electricity costs of the ice hall by 1.9 %. Demand response of district heat for swimming pool water and pool space air temperatures decreases the average price of purchased district heat in the swimming hall by 2.8 % and the total district heat costs of the swimming hall are reduced by 1.1 %. With all previous measures of utilization of waste heat and smart control of energy system, the changes in the total annual energy consumptions with summer breaks for the ice and swimming halls are 83 % reduced purchased district heat, 23 % increased electricity and 45 % reduced total purchased energy.

Keywords Waste heat recovery, demand response, smart control of energy, combined energy system, ice hall, swimming hall, dynamic building energy modelling

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Maailmanlaajuisesti rakennussektori kuluttaa noin 40 % primäärienergiasta ja tuottaa noin 30 % hiilidioksidipäästöistä. Rakennussektorilla on potentiaali vähentää maailmanlaajuista energiakulutusta ja päästöjä parantamalla hyötysuhdetta ja käyttämällä enemmän uusiutuvia energialähteitä fossiilisten sijasta. Suomen rakennuskantaan kuuluu noin 220 jäähallia ja 280 uimahallia, joiden suuri energiakulutus ja päästöt jäävät helposti huomiotta. Ympäristöministeriö ei ole poikkeuksellisesti asettanut rajoja näiden rakennustyyppien energiakulutukselle. Näiden rakennustyyppien suurimmat mahdollisuudet energiakulutuksen ja päästöjen vähentämiseksi ovat hukkalämmön hyödyntäminen ja kysyntäjouston mukaan ohjatut järjestelmät. Hukkalämmön hyödyntäminen vähentää ostettua kaukolämpöä jää- ja uimahalleissa, sillä se hyödyntää hukkalämpöä jään jäähdytyksestä, jätevedestä, ilman kuivatuksesta ja poistoilmasta. Kysyntäjousto vähentää sähkö tai kaukolämpöverkon huippupiikkejä, joka puolestaan vähentää päästöjä, sillä huipputeho tuotetaan saastuttavammilla laitoksilla.

Tämän tutkimuksen tavoitteina oli analysoida yhdistetyn jää- ja uimahallin energiasysteemin energiakulutuksen pienentämistä ja kustannussäästöjä hyödyntämällä hukkalämpöjä ja älykästä energiaohjausta. Tarkastelukohteena on kohde Helsingin Pirkkolassa. Kohteessa on vanha keskikokoinen uimahalli ja uusi rakenteilla oleva harjoitusjäähalli. Tämä tutkimus toteutettiin dynaamisella rakennuksen energiasimuloinneilla ja simulointituloksina saatavien energiavirtojen jälkikäsittelynä. Mallit rakennettiin jää- ja uimahalleille, ilmanvaihtokoneille ja energiasysteemin komponenteille, joita ei ollut valmiiksi saatavilla. Hukkalämmön hyödynnyksessä käytettiin lyhyen aikavälin lämpövarastoja. Älykäs energiaohjaus sisälsi sähkön ja kaukolämmön kysyntäjouston ja poistoilmalämpöpumpun ohjauksen. Sääntöperustainen algoritmi luotiin tässä tutkimuksessa sähkön ja kaukolämmön kysyntäjoustoa varten. Lisäksi poistoilmalämpöpumpulle luotiin älykäs ohjaus, joka ohjaa poistoilman lämpötilaa asettaen tuotetun hukkalämpömäärän vastaamaan uimahallin ennustettua lämmöntarvetta.

Tämä tutkimus osoitti, että jäähallin ostetusta kaukolämmöstä 99 % voi korvata hukkalämmöllä, samalla kun jäähallin ostosähkö kasvaa vain 9 %. Siirtämällä jäähallin ylijäämähukkalämpö uimahallille ja hyödyntämällä uimahallin hukkalämpöjä, uimahallin ostettu kaukolämpö vähenee 72 % ja uimahallin sähkönkulutus kasvaa 37 %. Sähkön kysyntäjousto jään jäähdytyksessä vähentää jäähallin ostetun sähkön energiakustannuksia 1.9 %. Kaukolämmön kysyntäjousto uima-altaan ja allastilan lämmityksessä vähentää uimahallin ostetun kaukolämmön hintaa keskimäärin 2.8 % ja uimahallin kaukolämmön energiakustannuksia 1.1 %. Yhdistämällä kaikki mainitut hukkalämmön hyödyntämisen ja älykkään energiaohjauksen toimenpiteet, muutokset jää- ja uimahallin vuotuiseen energiakulutukseen kesätauot huomioiden ovat 83 % vähentynyt ostettu kaukolämpö, 23 % lisääntynyt ostosähkö ja 45 % vähentynyt kokonaisostoenergia.

Avainsanat Hukkalämpö, Kysyntäjousto, energian älykäs hallinta, yhdistetty energiasysteemi, jäähalli, uimahalli, rakennuksen dynaaminen energiamallinnus

Preface

I was delighted to have the opportunity to research the topics of waste heat and smart control, since energy efficiency and energy simulations have long been my interests. I greatly hope the results of this study aid on creating more energy efficient ice and swimming halls for us all to enjoy. This master's thesis was extensive and challenging and will surely grant knowledge aiding in career after long sought graduation.

This master's thesis is made for HUKATON–project. The goal of HUKATON–project is to develop measures for utilizing waste heat in five chosen cases. Aalto University and Turku University of Applied Sciences researched the cases. Other parties included in the project were Green Net Finland and Geological Survey of Finland. This thesis studies one of the cases, which is a closely located swimming hall and ice hall in Pirkkola (Helsinki). HUKATON–project is carried out from August 2018 to December 2020. I humbly thank European Regional Development Fund (ERDF), City of Helsinki and Helen for funding HUKATON–project and thus this master's thesis.

I want to thank people who aided on the making of this thesis. I deeply thank my thesis advisor D.Sc. Juha Jokisalo and supervisor professor Risto Kosonen for helping me every week with their advices and knowledge. I thank manager of IDA ICE Mika Vuolle for helping me with energy simulations. I want to thank Teijo Korva for helping to gather information from Pirkkola. I also thank everyone who was working in Aalto University's department of Mechanical Engineering.

In addition, sincerely I thank my girlfriend Olga Viholainen for supporting me at home during the making of this thesis. I also thank my parents for the supportive upbringing, which gave me the motivation to study. I also want to thank all of my friends for having a good time always when possible.

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Symbols

A_{ice}	[m²]	surface area of ice
A_w	[m²]	surface area of water
B_p	[m/h]	empirical evaporation coefficient
COP_M		coefficient of performance
E_{PHD}	[kWh]	predicted total heat demand of the hour
R	[J/(kg·K)]	gas constant of water vapor
R_{sp}	[J/(kg·K)]	specific gas constant for dry air
$R_{sp,da}$	[J/(kg·K)]	specific gas constant for dry air
S	[Pa]	pressure difference caused by stack effect
$S_{E,a}$	[€/a]	annual energy cost savings
S_{inv}	[€]	the maximum cost of profitable investments
T_{aw}	[K]	average temperature of air (on surface layer of water)
T_c	[K]	condensation temperature
T_{cw}	[°C]	cold water temperature
T_{DHW}	[°C]	domestic hot water temperature
T_e	[K]	evaporation temperature
T_{ea}	[°C]	set point temperature of exhaust air in EAHP
T_{IR}	[°C]	temperature of used water for ice resurfacing
T_{in}	[K]	indoor air temperature
T_{out}	[K]	outdoor air temperature
T_{sh}	[°C]	average shower temperature
V_{IR}	[m³]	water amount per one ice resurfacing
$V_{water,d}$	[m³/d]	average water usage per day
a''_n	[a]	total discount yield
$c_{p,j}$	[W·s/kg·K]	specific heat of ice
$c_{p,v}$	[W·s/kg·K]	specific heat capacity of water
f_e	[%]	annual inflation of energy prices
f_T		loss factor of exhaust air heat pump
h	[m]	height of space
g	[m/s²]	gravitational acceleration
i	[%]	annual nominal interest rate
l_f	[W·s/kg·K]	condensing heat of fusion from liquid to solid
n	[a]	repayment period
$n_{g,d}$	[1/d]	average number of games per day
n_{IR}		number of ice resurfacing uses
n_{occ}		number of occupants in swimming hall
$n_{sh,g}$		average number of showers per game
p_a	[Pa]	vapor pressure of air
p_{atm}	[Pa]	atmospheric pressure
p_{da}	[Pa]	dry air pressure
p_v	[Pa]	vapor pressure
$p_{v,sat}$	[Pa]	vapor pressure of saturated air at the temperature of pool water
$q_{DHW,d,SH}$	[m³/d]	average DHW usage per day in the swimming hall
q_{IR}	[W/m²]	average heat load to ice from ice resurfacing

q_{sh}	[m^3/s]	average shower flow
$q_{water,a}$	[m^3/a]	annual water usage in Pirkkola ice and swimming halls
$q_{water,a,IH}$	[m^3/a]	annual water usage in the Pirkkola ice hall
q_{wm}	[kg/h]	mass flow of evaporating water
r_e	[%]	annual real interest rate of energy price
$t_{days,open,SH}$	[d]	days the swimming hall is open during a year
t_{sh}	[s]	average time of a shower
x	[g_{water}/kg_{air}]	absolute moisture content of air
$\Delta T_{avg,use,SH}$	[K]	temperature difference between cold water and used water
ΔT_{DHW}	[K]	temperature difference between cold water and DHW
Δt	[h]	time period
ρ_{in}	[kg/m^3]	indoor air density
ρ_{out}	[kg/m^3]	outdoor air density
ρ_w	[kg/m^3]	density of water
ϕ	[%]	relative humidity

Abbreviations

AHU	air handling unit
COP	coefficient of performance
CAV	constant air volume
CEP	current energy price
DH	district heating
DHW	domestic hot water
DR	demand response
EAHP	exhaust air heat pump
EB	energy balance
EL	electricity
EPD	Environmental Product Declaration
FCBR	Finnish Code of Building Regulation
FINVAC	The Finnish Association of HVAC Societies
IH	ice hall
IIHF	International Ice Hockey Federation
SH	swimming hall
TES	thermal energy storage
TRY	test reference year (for weather)
VAV	Variable air volume
VAT	value-added tax
WHR	waste heat recovery

1 Introduction

1.1 Background

The European Union (EU) Commission has expressed its position on climate change. Three main goals of the EU Commission (2014) by 2030 are to reduce overall carbon dioxide (CO_2) emissions by 40 % compared to the levels of 1990, to increase the share of renewable energy sources to 27 % of final energy consumption and to increase energy efficiency by 27 % compared to the levels of 1990. EU commission (2018) has also proposed a climate neutral Europe by 2050. According to Costa et al. (2013:1) building sector consumes 40 % of global primary energy and contributes to a total of 30 % CO_2 emissions. Thus, building sector has potential to reduce a significant portion of the total energy consumption and emissions. This can be done by improving energy efficiency and using more renewable energy sources instead of fossil fuels.

Building sector in Finland includes about 220 ice halls and 280 swimming halls (Hemmilä and Laitinen, 2018:8). Ice and swimming halls are easily overlooked as significant energy consumers and CO_2 emission producers. They consume 1.2 % of total energy consumption of Finnish building sector (Appendix A). Ice and swimming halls are special building types due to their indoor air conditions and high specific energy consumption. The Ministry of the Environment (2017a:4) has set no limits for the energy consumption of these type of buildings. Do to the lack of regulations and relative low amount of these buildings, there has not been as much study and implementation of energy saving measures and systems compared to the rest of the building sector.

Ice and swimming halls consume lots of electricity and heat. Biggest electricity consumer in ice halls are ice refrigeration and dehumidification. In swimming halls the biggest electricity consumer are saunas and pool water pumping. The biggest heat consumer in ice halls is domestic hot water (DHW). In swimming halls biggest heat consumer are DHW, space heating and pool water heating.

The main potential energy saving measures for these type of buildings are waste heat recovery and demand based control of systems. Hemmilä and Laitinen (2018) recently studied methods to reduce energy consumption in ice and swimming halls. This thesis utilizes conclusions of Hemmilä and Laitinen, but focuses on waste heat recovery and demand response.

Possible sources of waste heat recovery in ice halls are ice refrigeration, gray water and condensing water from dehumidification. Heat recovery from ice refrigeration is recommended system to implement into new ice halls according to international ice hockey federation (IIHF, 2016:29). Heat can be recovered from sewage water, also called gray water, with heat exchangers. Sewage water is produced in significant amounts in both ice and swimming halls. In condensing heat recovery, the condensing heat is recovered from condensing water during air dehumidification.

Combining energy systems of closely located ice and swimming halls to utilize more waste heat is suggested in many studies (Kuyumcu et al., 2016; Linhartová and Jelínek, 2017; Nesbit, 2011). An example of a combined ice and swimming halls is located in Mänttä, Finland. Ice hall of Mänttä was built in 2015 and it utilizes waste heat from ice refrigeration and sells the excess heat to a nearby swimming hall (Lautiainen, 2018:12).

Possible sources of waste heat recovery in swimming halls are gray water and exhaust air. Exhaust air heat recovery utilizes heat from exhaust air, mainly as condensing water, but in a situation where dehumidification is not used. Exhaust air heat recovery could be utilized in big amounts in swimming halls due to high moisture load (Hemmilä & Laitinen, 2018:21).

Demand response is a smart control of energy system method for reducing energy demand and especially CO_2 emissions. Studies of demand response often focus only on electricity. Alimohammadisagvand et al. (2018:71) concluded that demand response can "prevent exposure of its infrastructure to critical strains" and "improve grid-wide load factor of the electric power system". Demand response is widely studied and adopted in modern smart energy grids (Cappers et al., 2010). However, demand response of district heat has been studied only in few publications, for example by Martin (2017) and Mäki (2018). This thesis studies both demand response of electricity and district heat.

Demand response can be implemented with different strategies. Strategies used in this thesis include short-term demand response by means or rule based control algorithms, which are controlled by dynamic energy prices. Energy demands are shifted from times of high dynamic energy price to times of low dynamic energy price or alternatively by simply limiting energy demands during times of high dynamic energy price. These strategies lower especially CO_2 emissions, since more polluting energy production from auxiliary power plants are used in times of high dynamic energy price. Demand response for a combined energy system of ice and swimming halls has not been studied previously.

1.2 Research questions and objectives

The objective of this thesis is to analyze the potential energy cost savings of waste heat recovery and demand response in the combined energy system of ice and swimming hall. This study was carried out by dynamic simulations and post-processing of the simulation results. The models of the ice and swimming halls and component models of ventilation and energy systems that were not available were built in this study.

This thesis examines a case in Pirkkola (Helsinki), which includes an old existing swimming hall and the new ice hall. The new ice hall is currently (1/2019) under preliminary design and will replace the old ice hall. This thesis analyzes reduction of energy consumption and energy cost savings for the new ice hall, the swimming hall and their combined energy system. The methods used to reduce energy consumption and energy costs are waste heat recovery and smart control of energy system. Other parties can use the calculated energy cost savings for evaluation of different investment options.

Research questions of this thesis are:

1. How much waste heat could be utilized in Pirkkola ice and swimming halls.
2. How much energy costs can be saved in Pirkkola ice and swimming halls with waste heat recovery and smart control of energy system.

To fulfill the first research question, this thesis evaluates the utilization of different waste heat sources with different heating systems. Heat demand of low temperature applications such as floor heating may be smaller than amount of available waste heat energy. Low temperature applications are to be utilized first and higher temperature applications alternatively with heat pumps. The full utilization potential of waste heat is examined. A mismatch of waste heat and heat demands is taken into account.

The second research question is to find out the electricity and heating costs savings by implementation of smart control of energy system in addition to waste heat recovery. Strategies for smart control of energy system in this thesis consist of demand response of electricity and district heat use, short-time storing of waste heat and a demand based control of exhaust air heat pump. The saved electricity and heat demand costs are to be evaluated for different combinations of smart control of energy system. In addition, the combined effect of waste heat recovery and demand response is evaluated.

1.3 Research restrictions

Ice halls are classified according to spectator stand capacity and swimming halls according to pool surface area (Hemmilä & Laitinen, 2018:8). This thesis studies a training ice hall and a middle-sized swimming hall.

The topic of this thesis is extensive, and thus it cannot observe all included theories in detail. The extensiveness is consequence by the large energy system. Crucial part of the energy systems are components in which the energy is processed, such as heat exchangers and heat pumps. These components are observed in a general theoretical level. The parameters of these components, such as efficiencies are defined according to regulations or technical specifications of actual recent components.

In this thesis, rule based demand response control algorithms were developed and used. Demand response control is used control heating systems and electricity systems of ice refrigeration, exhaust air heat pump and saunas. Hourly electricity price is available in Finland (Nordpool, 2019), but hourly district heat prices are not available for end user contracts in Finland. In this thesis, a contract with hourly changing heat price is used based on calculations from Rinne (2017). Thus, the possible future dynamic district heat price will not be similar to the price data used in this thesis.

The cost analysis gives savings from energy costs for fixed interest rates and payback periods. Profitability during life cycle of the systems is not analyzed, since the investment costs are not evaluated in this thesis. Conclusions of this thesis give investment recommendations based on energy cost savings.

2 Energy systems of ice and swimming halls

This Chapter presents energy systems of ice and swimming halls as well as different waste heat recovery and smart control of energy system. This Chapter also presents the latest scientific studies carried on energy saving measures of energy systems of ice and swimming halls.

2.1 Energy consumption

Energy consumptions of ice and swimming halls are high. Ice halls consume a lot of electricity on ice refrigeration and swimming halls consume a lot of electricity on pool water pumping and saunas. Swimming halls consume also a significant amount of heat on keeping the pool water and indoor air warm enough to maintain thermal comfort.

2.1.1 Energy regulations

Ice and swimming halls are classified according to building class 9 and do not have a limit for energy consumption, but an E-value is needed to be calculated (Ministry of the Environment, 2017a). E-value is calculated based on purchased energy per surface area, and by taking into account new energy type coefficients defined for year 2018 by Finnish Government (2017). E-value is supposed to make comparison of building energy consumption easier and thus simplify discovering buildings with high energy consumption. E-value and is required to be calculated for ice and swimming halls even though there are no limitations to be met (Ministry of Environment, 2017a). In addition, Compliance of heat loss requirement needs to be shown. It is seen that ice and swimming halls are not comparable in terms of energy consumption per area and thus would need their own energy certificate which there is none currently.

Heat loss is regulated for ice and swimming halls by comparing the total heat loss to a standard heat loss of the same building with defined parameters (Ministry of Environment, 2017a). The calculation is done according to Ministry of Environment (2017b). Total heat loss consists of heat losses of envelope, ventilation and infiltration air. The standard model allows some of heat losses to be over standard, as long as the total heat loss does not exceed the corresponding standard value.

2.1.2 Swimming halls

A swimming hall is defined in this thesis as an enclosed exercise hall that provides at least a 25 meters long pool of water to be used for swimming. The building also includes all necessary sanitary and bathroom units, saunas, locker rooms and technical spaces. The swimming pool spaces can also include different kinds of pools, such as smaller pools for children, whirlpool bath and cold- and hot water basin, and it can include water activities such as water slides and jump towers.

Energy consumption

Hemmilä and Laitinen (2018:8) divided swimming halls to groups by pool surface area and presented the average energy consumptions per gross floor area by group (Table 1). The specific surface areas are gross floor area of the whole building. Specific consumptions are relatively even between different size swimming halls. This means that the building size increases as the pool surface area and visitor count increases.

Table 1. Energy consumption of swimming halls classified by pool surface area (Hemmilä & Laitinen, 2018:8).

The swimming hall size	Small	Medium	Big	Stadium	unit
Pool surface area	under 300	300 - 500	500 -750	Over 750	[m^2]
Number of halls	34	62	26	21	
Specific heating energy	508	461	492	438	[kWh/m^2]
Specific electricity energy	257	248	267	230	[kWh/m^2]
Specific water consumption	3946	4538	5116	4727	[dm^3/m^2]

Building an energy efficient swimming hall is challenging in the cold climate of Finland because of the big temperature difference between indoor and outdoor air. In addition, swimming halls have a high moisture load in pool spaces, which can cause building structure to gather condensation moisture and mold. The condensation can be prevented by heating the indoor air over the temperature of pool water and by dehumidification of room air, but this consumes significant amount of energy.

Environmental Product Declaration (EPD) has made an official Finnish design regulation guide for swimming halls based on multiple standards that are more general (EPD, 2009). The guide is extensive in terms of a swimming hall design and thus referenced many times in this thesis.

The water temperature of the big swimming pool should be kept between 26 and 28 °C due to thermal comfort according to EPD (2009:3). According to EPD, the indoor air temperature should be maintained at 1.5 to 2.5 °C higher temperature than the indoor surface temperature of the structures. This temperature difference limits evaporating and improves thermal comfort. Indoor air relative humidity should be maintained below 60 % to prevent microorganism growth and over 45 % to improve energy efficiency due to decreased evaporation. (EPD, 2009:3)

Energy distribution

Hemmilä and Laitinen (2018) did a breakdown of energy use of Finnish swimming halls. Figure 1 shows a breakdown of heat energy use of a swimming hall. Domestic hot water (DHW) is divided into shower, washing and other usage.

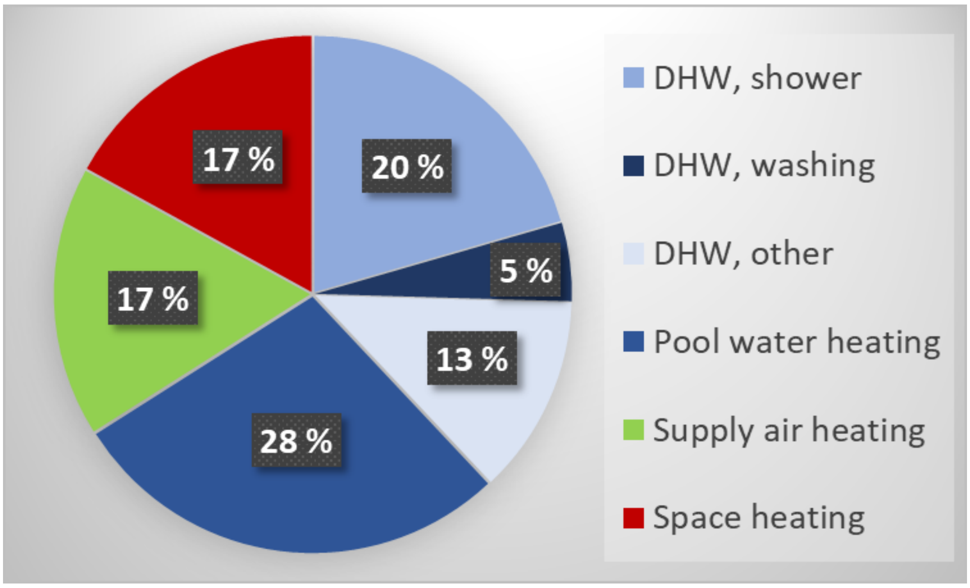

Figure 1. Typical breakdown of heat energy use of a Finnish swimming hall (Hemmilä & Laitinen, 2018:13, edited).

Shower water and washing water are drained into sewers. Other DHW usage consists of pool water changing, filter flushing and resupply pool water. Indoor air heating is implemented by space heating and supply air heating, which constitute of only about third of total heating (Figure 2). Space heating units consist of underfloor heaters and radiators. The significant amount of pool water and supply air heating demand is caused mainly by evaporation from pool to air. The evaporation consists of 77 % of heat losses from pool water (Kuyumcu, 2016:354). Evaporating water also transfers heat from air to evaporating water increasing heat demand of supply air. The evaporation can be decreased by increasing the temperature difference between pool water and indoor air.

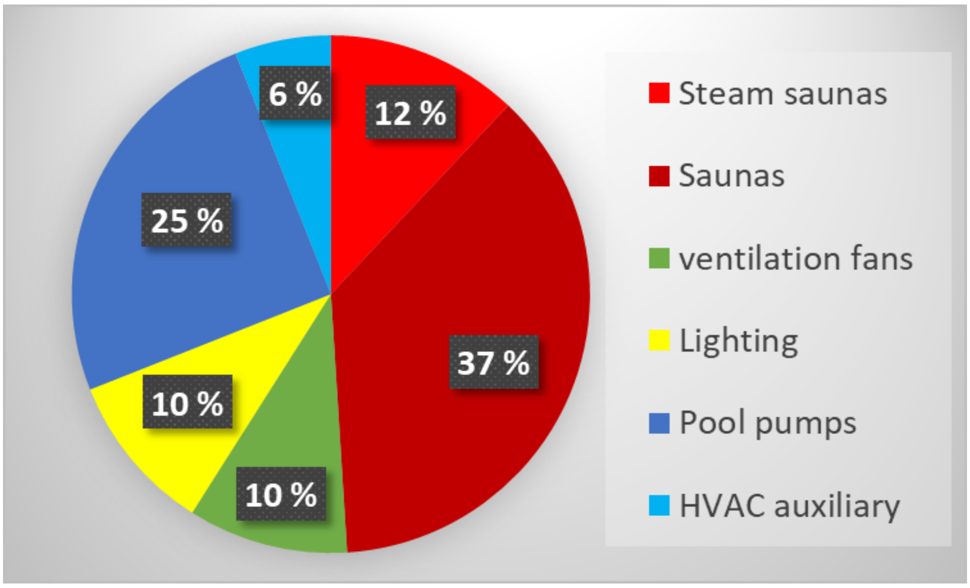

Figure 2. Typical breakdown of electricity use of a Finnish swimming hall (Hemmilä & Laitinen, 2018:13, edited).

Figure 2 shows a typical breakdown of electricity use of a Finnish swimming hall (Hemmilä & Laitinen, 2018:13). Even without the use of steam saunas, saunas are the biggest electricity user, which is because multiple big saunas are kept on during opening hours and used actively. The active use of saunas increases energy use since the sauna door is opened frequently and water is being thrown into the stove, which transfers heat from stove to exhaust air. Pool pumps use a significant amount of energy for recycling the pool water. The water is pumped to heat exchangers to keep the water warm and to filters to keep the water clean. Ventilation energy use is high because relative humidity of indoor air should be kept below 55 % and temperature over 26 to 28 °C (EPD, 2009:3). Figure 2 does not include air dehumidification, since it is not a compulsory in swimming halls as indoor air could also be dehumidified with outdoor air (EPD, 2009:31).

2.1.3 Ice halls

In this thesis, an ice hall is defined as an enclosed exercise hall that provides at least one EU full-sized 29 meters times 60 meters ice rink and separate spaces for dressing rooms, showers, lavatories and technical spaces. Ice halls usually have at least a small spectator stand on one side of the ice rink for spectators of ice hockey game.

Energy consumption

Specific energy consumption of an ice hall is highly dependent on the ice hall type, such as a training ice hall without spectator stand and a big ice arena with spectator stand capacity of tens of thousands of people. Hemmilä and Laitinen (2018:8) classified Finnish ice halls by number of seats in the spectator stand and presented the average energy consumptions per gross floor area by groups (Table 2). The specific surface areas are gross floor area of whole building. Specific heating and water consumption of ice halls are significantly smaller than in swimming halls, while specific electricity consumption stays about at the same level (Table 2). The specific consumptions of heating, electricity and water consumption of the groups is different because of the different kind of usage of the halls. For example, stadiums are not used in for training and they have a significant amount of auxiliary space reducing the specific consumption.

Table 2. Energy consumption of ice halls classified by spectator seat count (Hemmilä & Laitinen, 2018:8).

Ice hall size	Small	Medium	Big	Arena	unit
Spectator seat count	100 - 300	300 - 1500	1500 - 6000	over 6000	
Number of halls	16	28	6	3	
Specific heating energy	157	142	230	82	[kWh/m^2]
Specific electricity energy	258	197	347	110	[kWh/m^2]
Specific water consumption	822	681	938	586	[dm^3/m^2]

The biggest energy sink of an ice hall is the ice, which has to be cooled to maintain a temperature of -3 °C for training and -5 °C for a hockey game according to International ice hockey federation (IIHF, 2016:38). The ice is cooled by cooling pipes inside concrete, which is located below the ice and on top of thermal insulation layers. Below the ice and the insulation layer, there are heating pipes to prevent ground frost with circulating hot water illustrated in Figure 3. The insulation thickness can be increased to limit heat losses from heating pipes to cooling pipes. Ice rink spaces are cold spaces, since the indoor air is cooled by the cold ice sheet. The ice significantly increases the heat demand of spaces, since the ice temperature is lower even at room temperatures at around 6 to 12 °C. Cooling of the indoor air also causes condensation risk and thermal discomfort, which in turn requires dehumidification of indoor air.

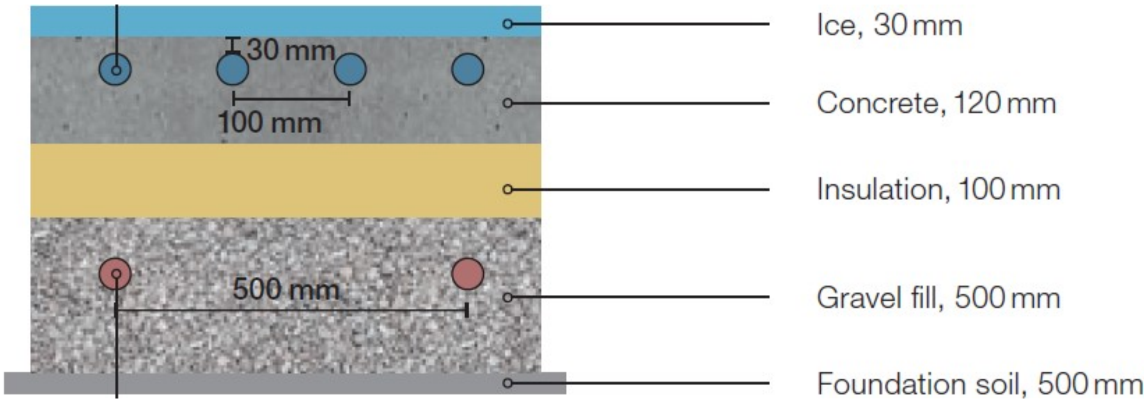

Figure 3: An example structure of an ice rink (IIHF, 2016:37).

Energy distribution

Laitinen and Kosonen (1994) developed a simulation program and evaluated the effect of different parameters on heat and electricity demand of ice halls. They calculated and presented the energy balance as a Sankey diagram for a training ice hall (Figure 4). The energy balance is for a year, the air temperature is +12 ºC and it includes refrigeration heat recovery. The values are scaled to match the percentage of total heat load on ice sheet, which equals 1680 MWh. The total heat, electricity and total energy usages are 1330, 920, 2250 MWh respectively. Excess condensation means the heat that can be transferred to other buildings, for example, to a swimming hall.

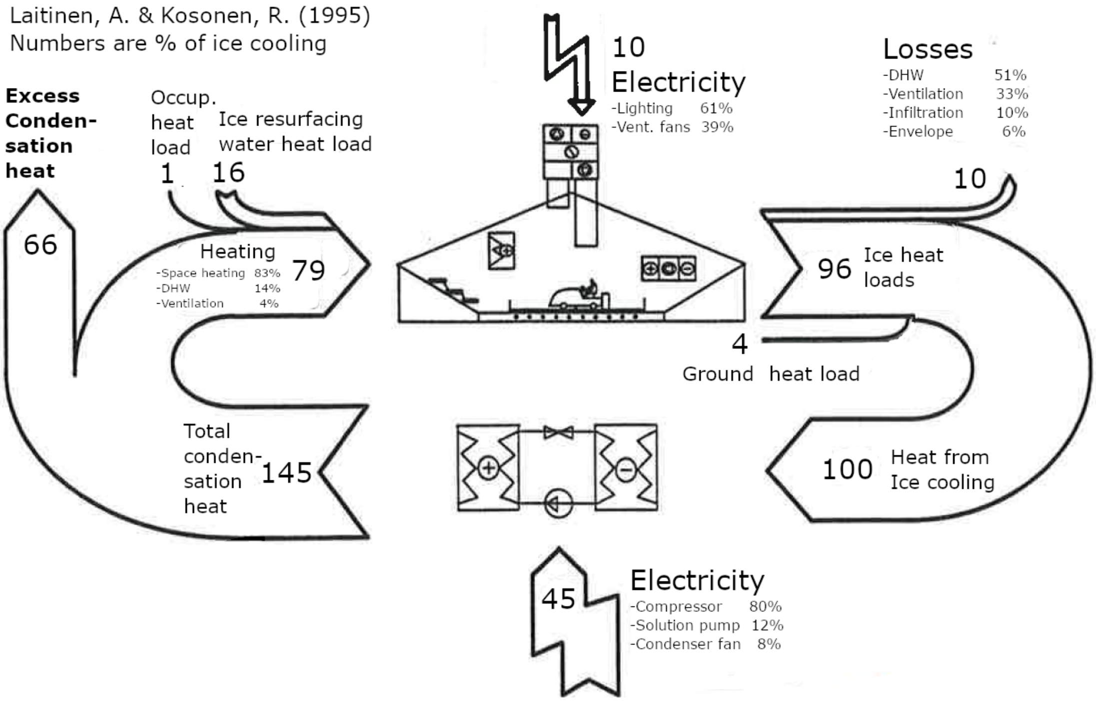

Figure 4. Energy balance of the ice hall with heat recovery from ice refrigeration (Laitinen & Kosonen, 1994:13, edited).

Figure 4 shows energy fluxes of the ice hall compared to percentage of condensation heat produced by ice cooling. Most of the heat losses are caused by the ice (96 %) compared to other heat losses (10 %). Space heating has the biggest heat demand (66%), but also a considerable amount of heat demands are caused by ice resurfacing water heating (16 %). Most of the electricity is used in the ice refrigeration compressor (36 %).

Nichols (2009:18) analyzed inefficient ice halls in Canada, which have an annual combined electricity and heating consumption of 1950 MWh. Figure 5 presents breakdown of energy consumption on the ice hall. The heat demands (1050 MWh) are colored red-brownish to separate them from electricity demands (900 MWh). Only the lighting seems to differ from the energy distribution by Laitinen and Kosonen. The lighting electricity portion is significantly bigger and it is a good example on how energy consumption can be reduced by upgrading to lighting with better efficiency.

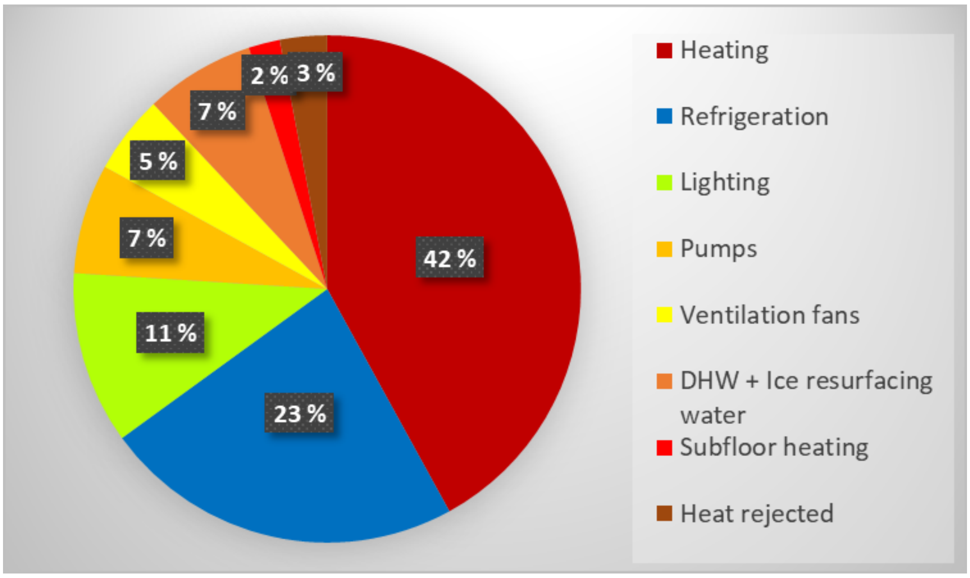

Figure 5. Breakdown of energy use of an inefficient ice hall in Canada (Nichols, 2009:18, edited).

Heating and ice refrigeration are the most significant part of energy consumption of ice rink. These are caused mainly by heat loads on ice from indoor air. IIHF (2016:13) states the heat loads on ice to be as follows:

> Ceiling radiation is generally the largest single component of the heat loads. Other ice heat load components are: the convective heat load of the ice rink air temperature, lighting, ice maintenance, ground heat, humidity condensing from the air onto the ice, and pump-work of the cooling pipe network.

Heat loads of ice according to Laitinen and Kosonen (1994:13) are presented in Figure 6. The order of magnitude is the same as stated by IIHF. Biggest single components of ice heat load are the amount of radiation and convection to ice. They are determined mainly by emissivity factor of the inner ceiling coating and the temperature difference between air and ice surface. The other considerable heat loads on ice is ice resurfacing. Ice resurfacing is done by first smoothing the ice surface by scraping and then spreading hot water on ice, which freezes as a new layer of ice. Other heat loads of ice are condensation from indoor air on ice surface, lighting and conduction heat load from ground frost protection to ice. Condensation heat load is affected by absolute moisture content of indoor air. Lighting heat load is affected by lighting power and orientation. Heat load of ground frost protection is affected by thermal insulation thickness between the cooling and heating pipes.

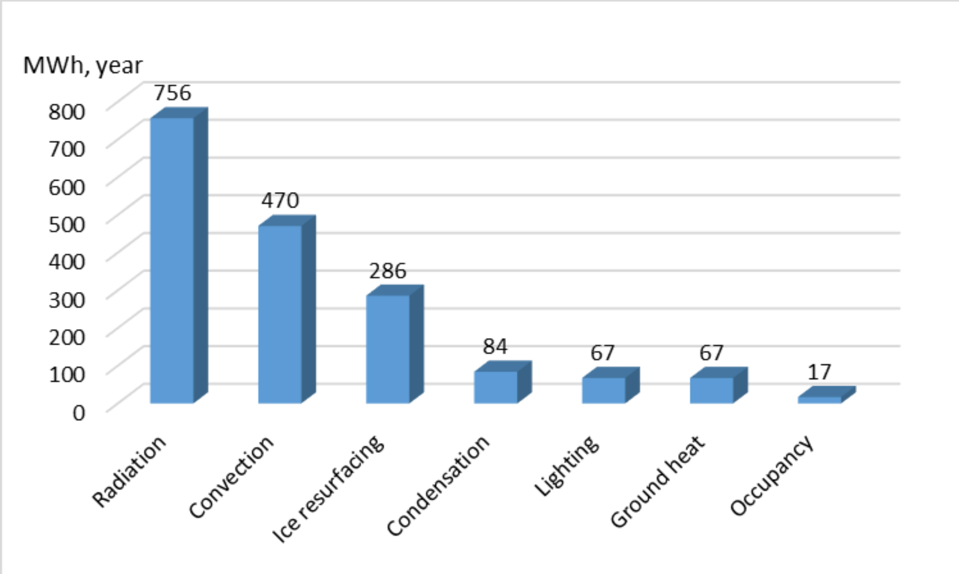

Figure 6. Heat load components of an ice rink (Laitinen & Kosonen, 1994:13, edited).

2.2 Ventilation

Ventilation has a significant impact on heating and electricity consumption of ice and swimming halls. Supply air heating is the main heating solution for pool and rink spaces, since it does not require any devices on floors. The proportion of heat demand of supply air heating compared to total space heating depends mainly on the used space heating systems, such as water radiators or underfloor heating. Supply air heating may need increased ventilation rate compared to space heated fully with other heating systems. The increased ventilation rate increases electricity consumption of ventilation fans.

Ventilation plays a crucial role in controlling of the indoor air quality. The Finnish Association of HVAC Societies (FINVAC) has set new guidelines in 2017 for ventilation of different types of Finnish buildings. FINVAC (2017:15) specifies exercise halls and swimming halls into own ventilation group. It is important to note that the dimensioning values of exercise spaces are based on occupancy or moisture load and not on floor surface area. Other warm space airflows are dimensioned normally on either occupancy or floor area (FINVAC, 2017:16).

2.2.1 Swimming halls

EPD (2009:3) interprets, that outdoor airflow rate of a swimming hall space must be kept at a minimum of value of 2 $dm^3/s/m^2$ (gross floor area). However, the new guidelines by FINVAC (2017:15) state that the ventilation is controlled only by moisture content of indoor air. The biggest moisture load and thus biggest dimensioning effect of airflow rate of ventilation comes from pool evaporation. Evaporation from pool happens due to vapor pressure difference of indoor air and saturated air at the temperature of pool water (Equation 1). EPD (2009:4) provides empirical Equation to define the mass flow of evaporated water from pools

$$q_{wm} = A_w \cdot \frac{B_p}{R \cdot T} \cdot (p_{v,sat} - p_a) \tag{1}$$

where

q_{wm}	mass flow of evaporating water [kg/h]
A_w	surface area of water [m^2]
B_p	empirical evaporation coefficient [m/h]. For example, 28 for a deep pool meant for swimming and 40 for a shallow pool meant for children
R	gas constant of water vapor = 461.52 [J/(kg·K)]
T_{aw}	average temperature of air (on surface layer of water) [K]
$p_{v,sat}$	vapor pressure of saturated air at the temperature of pool water [Pa]
p_a	vapor pressure of air [Pa].

According to EPD (2009:8), the spaces with different temperature set points should be divided into separate zones, which are supplied by, dedicated air handling units (AHU) and heat distribution systems. Spaces with swimming pools are recommended to be equipped with displacement ventilation supplied near the edge of pool (EPD, 2009:5).

Moisture control

Indoor air in humid pool spaces needs to be underpressured with ventilation to prevent humid indoor air going from inside through building envelope to outdoor air (EPD, 2009:6-11). Swimming halls are high spaces which causes the air pressure to be higher on top of the space compared to bottom of the space, this phenomenom is called a stack effect and is caused by temperature difference between indoor and outdoor air. Underpressure is implemented by dimensioning the ventilation to extract more air than supply air. This underpressure is used to compensate the overpressure in top part of pool space caused by stack effect, but it is not always enough, especially when outdoor air is at coldest or the ventilation is lowered during closing times. Even underpressure does not prevent water vapor from diffusing into the envelope, which has to be taken care of with a vapor barrier. EPD (2009:10) provides Equation to calculate the pressure difference caused by stack effect inside the space

$$S = (\rho_o - \rho_i) \cdot g \cdot h \tag{2}$$

which as a function of temperatures becomes

$$S = \left(\frac{1}{T_{out}} - \frac{1}{T_{in}}\right) \cdot \frac{p_{atm}}{R_{sp,da}} \cdot g \cdot h \tag{3}$$

where

S	pressure difference caused by stack effect [Pa]
ρ_{out}	outdoor air density [kg/m^3]
ρ_{in}	indoor air density [kg/m^3]
T_{out}	outdoor air temperature [K]
T_{in}	indoor air temperature [K]
p_{atm}	atmospheric pressure [Pa]
$R_{sp,da}$	specific gas constant for dry air [J/(kgK)]
g	gravitational acceleration [m/s^2]
h	height of space [m].

If the envelope were assumed to leak evenly, the pressure difference in top of pool space for the ventilation to compensate would be half of the given equation, and for the worst case, it would be equal to given Equations 2 and 3. The worst case would happen when the bottom part of building envelope leaks. Figure 7 visualizes the effect of ventilation on stack effect in swimming halls.

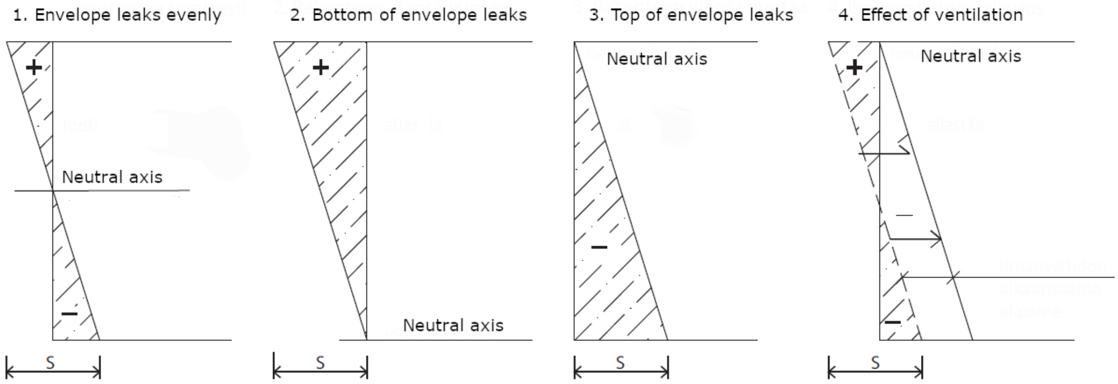

Figure 7. Stack effect on inside space with different envelope leak heights (EPD, 2009:9, edited).

Relative humidity of a swimming hall air has to be controlled. Air temperature of a swimming hall decreases when evaporation from pool happens, since the evaporating water absorbs heat from air. Higher relative humidity of indoor air reduces water evaporated from pool. Yli-Rosti (2012) found out that by raising relative humidity of the indoor air from 40% to 50% at outdoor temperature of 0 °C, the energy demand of air conditioning is halved. On the other hand, raising relative humidity too much may cause condensation and microorganism growth on cool surfaces of pool spaces (EPD, 2009:3). Overheated supply air can be directed into windows and other cool surfaces to heat them in order to prevent condensation by increasing the lowest indoor envelope temperature (EPD, 2009:5). Dehumidification is not compulsory in swimming halls (EPD, 2009:31).

For swimming halls in Helsinki, the supply air is dimensioned for a moisture content of 9 g_{water}/kg_{air} (EPD 2009:5). The dehumidification can be implemented by increasing outdoor airflow, as air can be dehumidified with less humid outdoor air (ME, 2007:83). An example annual outdoor air moisture content is compared to the dimensioned moisture content value in Figure 8. The absolute moisture content of air is calculated from Finnish Metrological Institute (FMI, 2012) test year temperature and relative humidity as follows

$$x = \frac{p_v}{p_{da}} \tag{4}$$

$$p_v = \phi \cdot p_{atm} \cdot e^{11.78 \cdot \frac{T_O - 372.79}{T_O - 43.15}} \tag{5}$$

$$p_{da} = R_{sp} \cdot \rho_{da} \cdot T_o \tag{6}$$

where

x	absolute moisture content of air [g_{water}/kg_{air}]
p_v	vapor pressure [Pa]
p_{da}	dry air pressure [Pa]
ϕ	relative humidity [%]

p_{atm} atmospheric pressure [Pa]
R_{sp} specific gas constant for dry air [J/(kg·K)]
ρ_{da} dry air density [kg/m^3]
T_{out} outdoor air temperature [K].

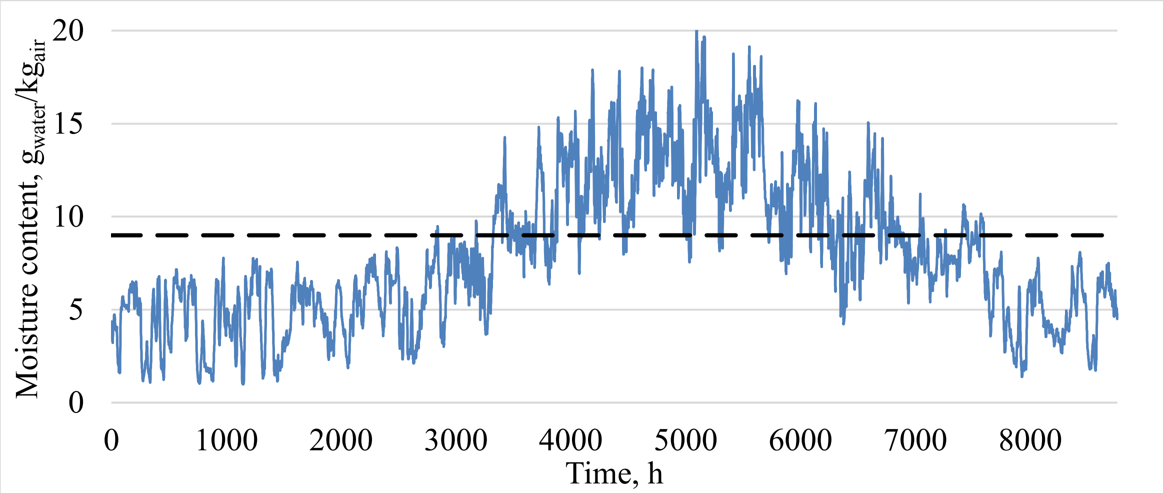

Figure 8. Dimensioned indoor air moisture content of a swimming hall (dashed line) and outdoor air moisture content (blue) calculated from test year data (FMI, 2012).

Figure 8 shows that dehumidification by increasing outdoor airflow rate works for most of the time, when outdoor air is dryer than the dimensioning value. However, during the summer the indoor air humidity will increase if dehumidification by cooling is not used.

2.2.2 Ice halls

FINVAC states (2017:15) that ventilation of exercise halls, such as ice rink space and spectator stand are dimensioned based on occupancy and metabolic rate. According to FINVAC (2017:16), the metabolic rate of a hockey player is 6 met requiring 30 dm^3/s per person and of a spectator is 1.2 met requiring 6 dm^3/s per person. This can be fulfilled with a variable air volume (VAV) system, which modulates airflow rate based on carbon dioxide (CO_2) concentration of indoor air.

An energy efficient ventilation requires heat recovery, air re-circulation and variable air volume control based on indoor carbon dioxide concentration (Toomla et al. 2018:3). Toomla et al. (2018) found out that during the normal ice hall use, carbon dioxide based set point was fulfilled with outdoor air entering the hall only from leakages and by infiltration air. Thus, during the normal ice hall use, there was no need to supply outdoor air inside, expect for dehumidification purposes.

Ice halls are challenging for ventilation, because the temperature of ice surface is much lower than temperature of indoor air. This temperature difference causes a stratification of air on top of the ice. This phenomenon was studied by Toomla et al. (2018) and they found that indoor air created a stagnant zone 3 meters on top of the ice. Toomla et al. (2018:15) concluded that indoor air in the case of studied ice hall was replaced inefficiently because of ceiling distribution, which was not able to blow supply air to the stagnant zone (Figure 9). The study did not consider ice-skaters on the mixing of indoor air but noted that movement on ice did not cause significant effect on the stagnant zone (Toomla et al. 2018:7). Thus, the air should be supplied at lower heights.

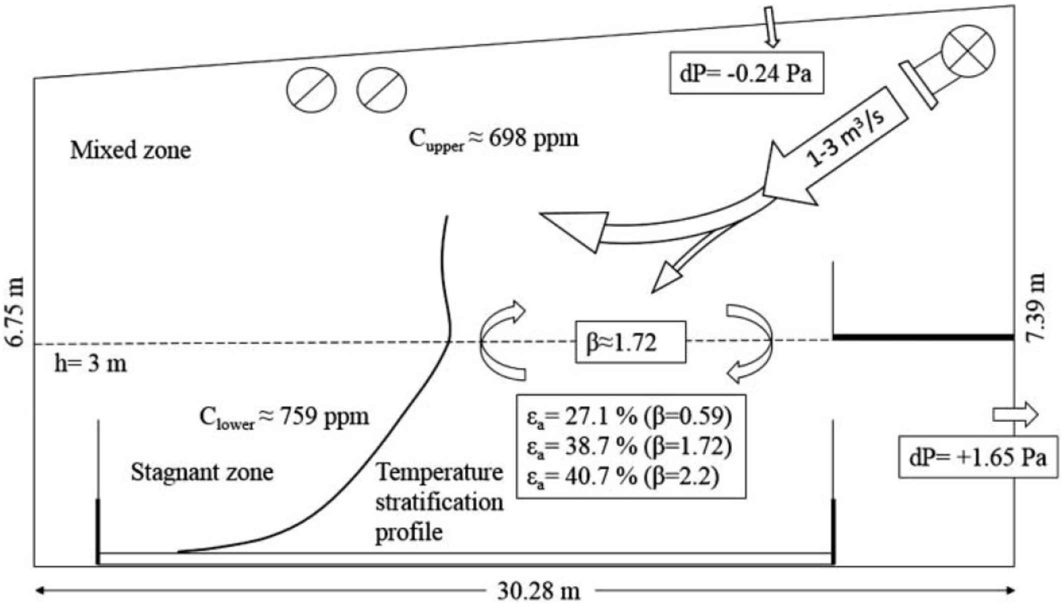

Figure 9. Stratification of an ice hall indoor air temperature with measured values (Toomla et al., 2018:14).

The biggest energy consumer in ice halls is space heating which is implemented by supply air heating. The heat losses of space heating are caused by the ice rink. The spectator stand is located in the same space as the ice rink and it can have an additional supply air heating to maintain spectator thermal comfort. The supply air heating causes the air rise towards ceiling increasing the ceiling temperature and thus increasing thermal radiation towards the ice (Figure 10). This results in a heat flux from spectator stand to the ice rink. Heat load from spectators also raises the temperature of spectator stand.

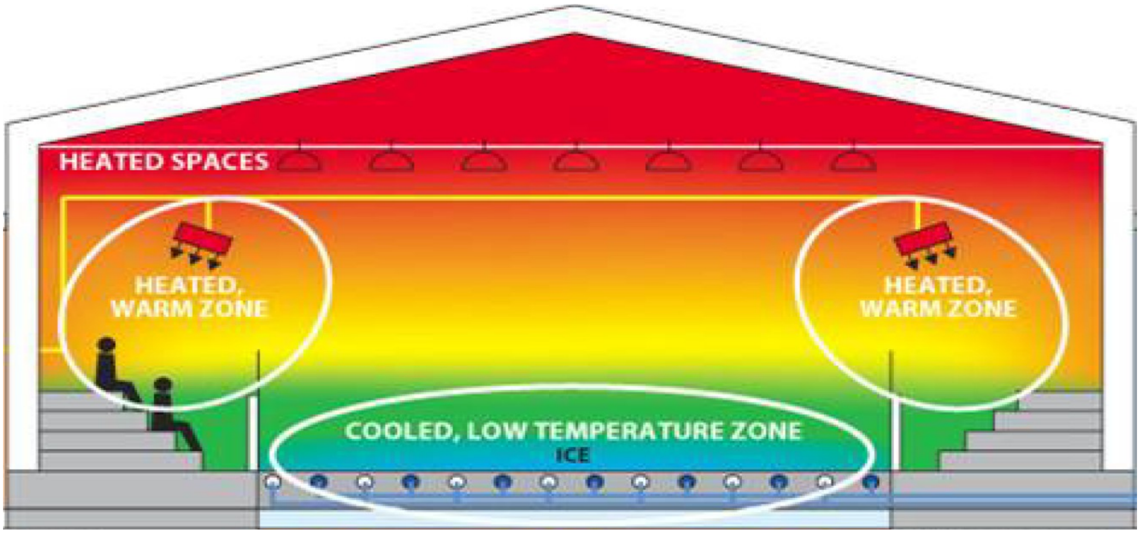

Figure 10. Heated supply air rises and increases heat loads to ice (Retscreen, 2005, edited).

Moisture control

The specific airflow rate per gross floor area ($dm^3/s/m^2$) of an ice hall is not as high as in swimming halls due to smaller moisture loads. Moisture loads of an ice hall consist of evaporation from occupants and moisture from outdoor air. Evaporation moisture load from occupants is especially high when spectator stand is fully occupied. Outdoor air causes a high moisture load during warm season when absolute moisture content of outdoor air is high. Thus, ice halls need a dehumidification of air to prevent condensation on indoor surfaces (IIHF, 2016:28). Ice temperature is below indoor air dew point and thus condensation happens on ice surface. IIHF suggests using 70 % as a design value for relative humidity (2016:38). However, the maximum relative humidity to avoid fog can be briefly 80% or 90% for indoor temperatures of +10 and +5 °C respectively (IIHF, 2016:44).

Outdoor infiltration air causes a moisture load. The infiltration air cannot be dehumidified, but it can be limited. Indoor air in an ice hall has low absolute moisture content of air due to low temperature. During times when outdoor air is warmer than indoor air, the stack effect causes an underpressure inside an ice hall. The air should be overpressured with ventilation to limit humid infiltration air. Overpressure is implemented by dimensioning the ventilation to supply more air than extract air.

In ice halls, dehumidification is implemented for most cases (Zhen et al. 2010) with condensing dehumidification method. In the method, supply air is cooled below its dew point causing the vapor to condense out of the air into cooling coil surface, from where the water is directed away. Other less commonly used dehumidification methods are using liquid desiccant or spray dehumidifiers. Liquid desiccant is used mainly in applications needing very dry air, such as in hot climates and in industries, such as pharmaceutical, food and chemical (Mujahid et al., 2015:182). Spray dehumidifiers cool the air down to dew point by spraying water with a temperature below dew point. Spray dehumidifiers remove pollutants from outdoor air due to water droplets binding small particles from air. However, the downside is the management of more generated water to direct away from air handling unit (AHU) than in electric cooling coils.

After dehumidification, outdoor air is heated with a heating coil to set point temperature of supply air. Dehumidification causes electricity demand for AHU cooling and the heating process causes heat demand for AHU heating.

For indoor air state of +6 ºC temperature and 70 % of relative humidity (RH), the corresponding absolute moisture content of indoor air is 4 g_{water}/kg_{air} and corresponding dew point is +2 ºC (IIHF, 2016:38). The cooling set point for supply air dehumidification has to be even lower, for example -3 ºC, to ensure the required relative humidity of indoor air (Kianta, 2018). An example of annual outdoor air moisture content is calculated according to Equations 4, 5 and 6 and is compared to the dimensioned value in Figure 11. Figure 11 shows why dehumidification is necessary in ice halls. The moisture content of outdoor air is almost always higher than the dimensioned indoor air.

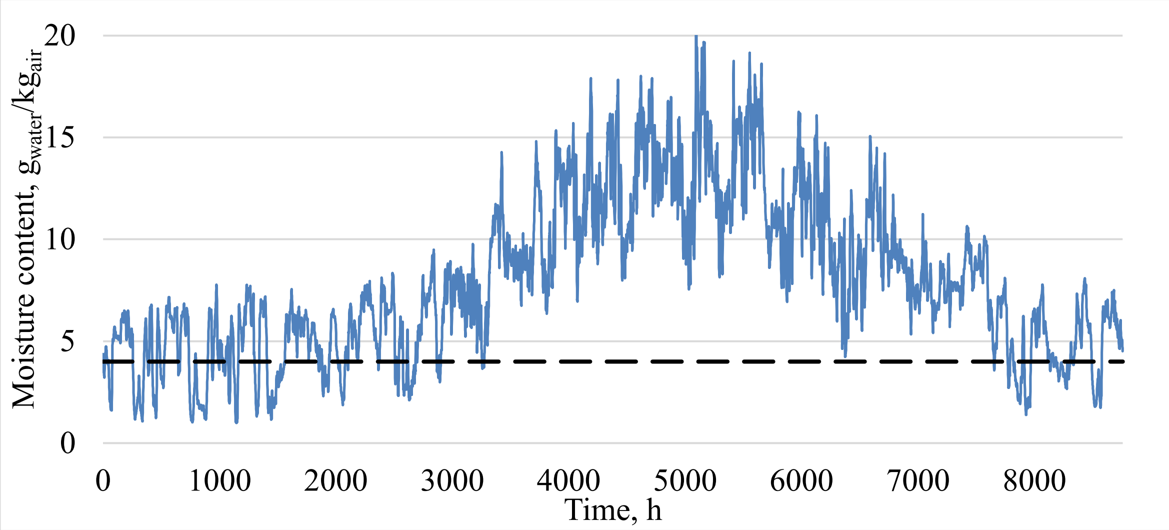

Figure 11. Dimensioned indoor air moisture content of an ice hall (dashed line) and outdoor air moisture content (blue) from Finnish Metrological Institute test year data (FMI, 2012).

2.3 Waste heat recovery

Waste heat recovery means utilizing excess heat, which would otherwise be wasted. It is implemented by transferring waste heat into a medium, which is usually water. The sources for waste heat in ice and swimming halls include ice, sewage water and air. According to Environmental Product Declaration (EPD) (2009:12), the waste heat fluxes are low temperature heat sources. Waste heat can be utilized straight only into low temperature heat demands, such as heating of underfloor heating and pool water heating (EPD, 2009:23). Possible space heating systems used in ice halls are supply air heating and convectors, and in swimming halls underfloor heating and supply air heating (EPD, 2009:12). Waste heat can be also used for pre-heating of higher temperature heat demands, such as DHW. Alternatively, a heat pump is required to raise the temperature of waste heat to be utilized for higher temperature applications. Heat pumps consume electricity, which consumption should be minimized.

2.3.1 Waste heat sources

Possible measures of waste heat recovery for ice and swimming halls are:

- Refrigeration heat recovery
- Condensing heat recovery
- Exhaust air heat recovery
- Gray water heat recovery

Waste heat recovery in this thesis is defined as the combination of refrigeration -, condensing -, exhaust air- and grey water heat recovery.

Refrigeration heat recovery

Refrigeration heat recovery utilizes heat received from ice rink cooling. The heat loads of ice and compressor electricity turned into heat are transferred from coolant into water in a condenser. Refrigeration heat recovery is recommended system in new ice halls according to international ice hockey federation (IIHF, 2016:29).

The ice refrigeration can be implemented with either an indirect or a direct refrigeration plant (IIHF, 2016:39). Indirect refrigeration works by pumping coolant from ice cooling piping to an evaporator. The energy efficiency would be better in a direct refrigeration but it is more expensive and thus indirect refrigeration is mainly used in ice rinks (IIHF, 2016:39). In an indirect cooling, the refrigeration heat pump has a separate cycling coolant inside. Figure 12 presents a schematic of a refrigeration heat pump. The coolant states are numbered in Figure 12, starting before compressor (1), before a condenser (2), before an expansion valve (3) and before an evaporator (4). In a direct refrigeration, the rink piping replaces the heat pump evaporator and thus only one cycle of coolant is needed.

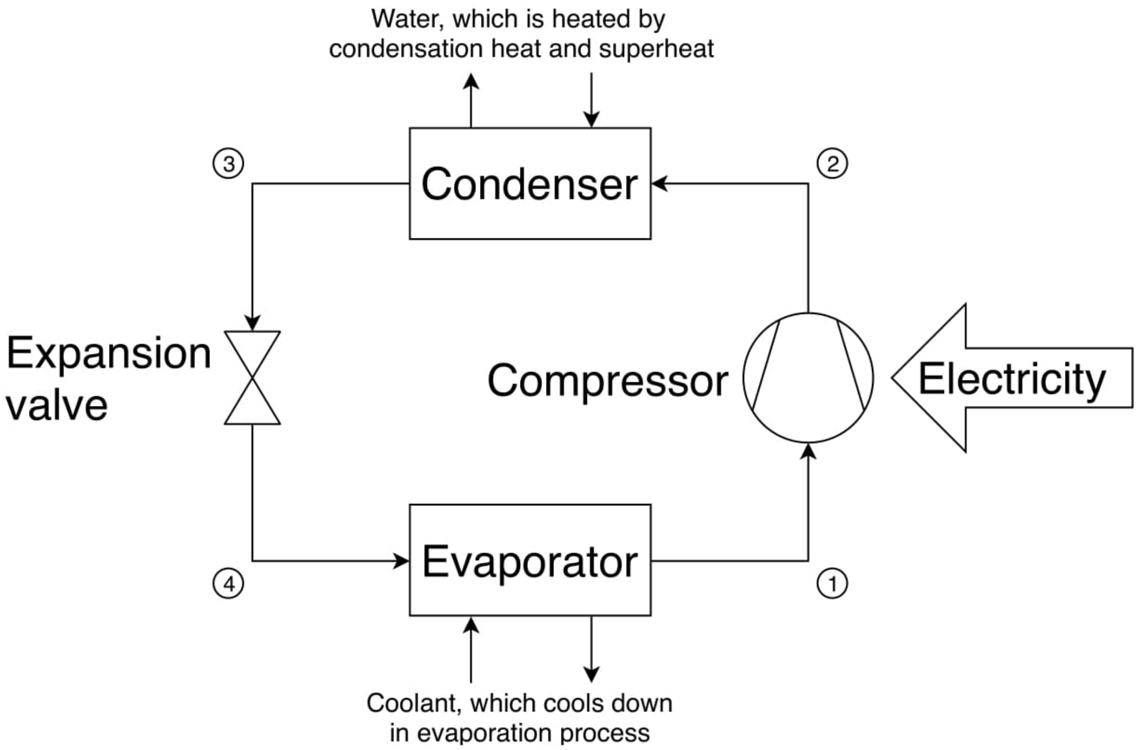

Figure 12. Schematic of refrigeration heat pump processes.

Figure 13 shows an example change of state of coolant for refrigerator during the cooling cycle. The pressure-enthalpy diagram can be used to analyze the four processes. Different quantities to analyze are enthalpy (heat per mass), condensing heat of coolant including superheat portion, temperatures and pressures of the processes. Coefficient of performance (COP) of the compressor defines the amount of heat transferred per electricity consumption. Electricity consumption of compressor is equal to the enthalpy change between coolant states 1 and 2. The amount of heat transferred from ice is equal to the enthalpy change between coolant states 4 and 1 (Figure 13).

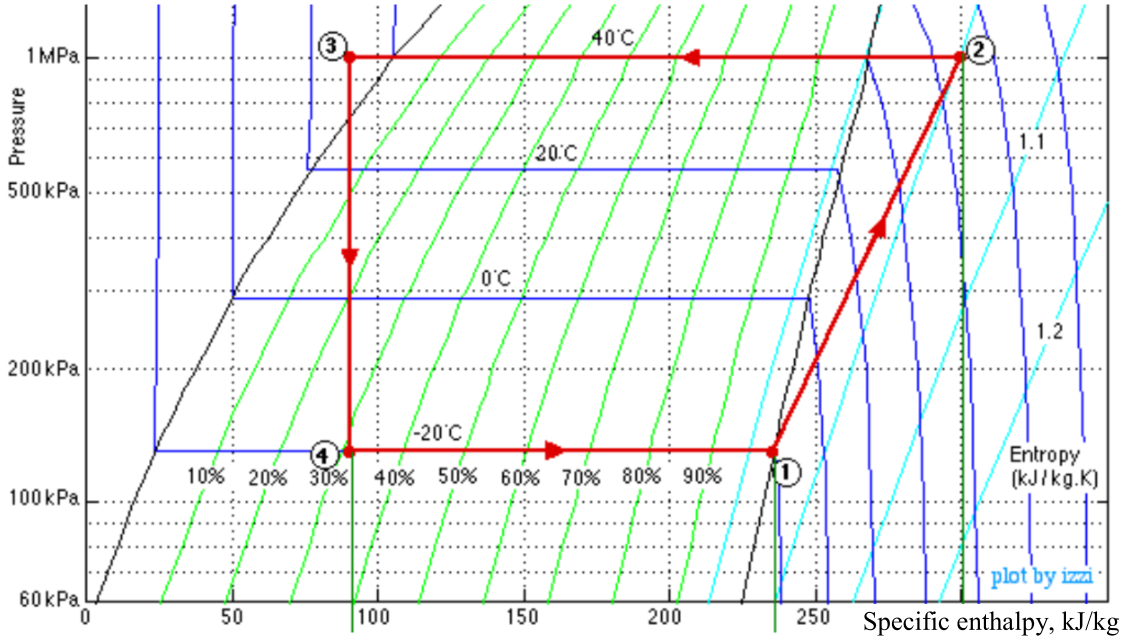

Figure 13. Pressure-enthalpy diagram of an example refrigerator coolant cycle (Ohio University, 2018).

Figure 14 illustrates an example how refrigeration heat recovery could be connected to a heating system of the building. Heating systems included are supply air-, floor-, DHW- and ground frost protection. Figure 14 shows also that DHW requires a hot water storage.

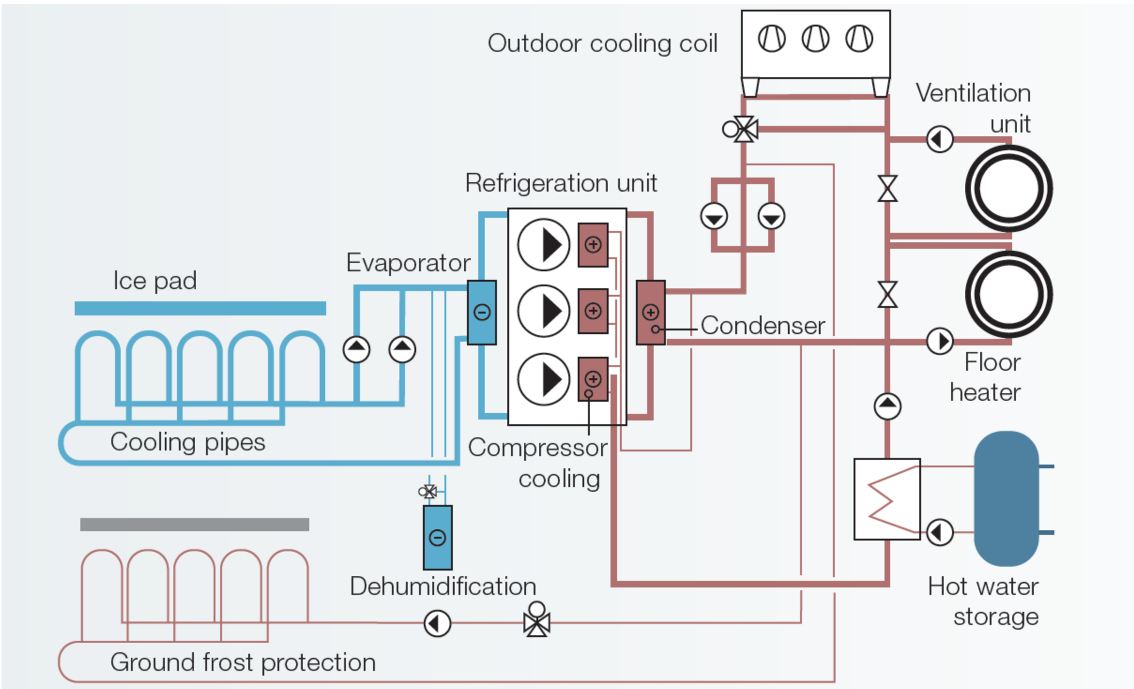

Figure 14. An example schematic for connecting refrigeration heat recovery to heating systems of an ice hall (IIHF, 2016:41).

Condensing heat recovery

Condensing heat can be recovered during condensing dehumidification. Dehumidification is compulsory in ice halls, but not in swimming halls (EPD, 2009:31; ME, 2007:83). This is because the dimensioning moisture content for supply air is lower in ice halls than it is in swimming halls.

The condensation happens, when the air is cooled below its dew point. The dehumidification process inside an air handling unit (AHU) can be used for outdoor and recirculation air. Heat is recovered mainly as condensing heat from condensing water, but heat is also received from dry air portion of the moist air. The condensing heat stands for heat released, when air moisture condensates from gas to liquid.

Exhaust air heat recovery

Recovering heat from exhaust air by cooling is called exhaust air heat recovery.. Exhaust air heat recovery requires its own heat pump, which increases the electricity consumption of the system. Exhaust air of a swimming hall has a bigger heat potential than an ice hall since it is warmer and contains more moisture.

Hemmilä and Laitinen (2018:76) stated that 80 to 90 % of heat demand of a swimming hall could be covered by heat recovery of exhaust air with a cooling temperature between 0 ºC and +5 ºC. They also state that even after an energy efficient heat recovery in AHU, the heating potential of exhaust air is still significant (Hemmilä & Laitinen, 2018:31). Lowering the temperature for exhaust air cooling increases the heat received but also increases electricity consumption. Thus, exhaust air heat recovery of swimming halls should be carefully analyzed to prevent increasing overall energy costs by increasing electricity consumption for unutilized waste heat.

Gray water heat recovery

In grey water heat recovery, heat is received from sewage water with a heat exchanger. Sewage water is produced in significant amounts in both ice and swimming halls, since almost all of the occupants of these building types take a shower after a visit. Shower water and washing water are drained into sewers and are sources of grey water heat recovery. Other DHW usage consists of ice resurfacing water, pool water changing, pool filter flushing and pool resupply water (Hemmilä & Laitinen, 2018:13). However, these other DHW usages are not drained into sewers and are thus not sources of grey water heat recovery.

The efficiency of heat exchangers for grey water heat recovery is about 30 % (Hemmilä & Laitinen, 2018:49). The total efficiency can be increased by combining multiple heat exchangers. The efficiency is defined as ratio of waste heat received from heat exchangers to heat used for DHW heating entering the sewers. The temperature level of heat from grey water heat recovery is too low to be utilized straight into high temperature heat demands. The temperature is still higher than in refrigeration- or condensing heat recovery and thus requires less electricity to be raised with a heat pump to a required temperature level. The temperature level can also be increased by, for example, constructing a separate sewage for hot shower waters and cold waters or by insulating the sewers.

2.3.2 Combined energy system

Swimming halls have a high specific heat demand and they could utilize significant amount of waste heat. Training ice halls have excess waste heat from refrigeration alone (Laitinen & Kosonen, 1994:13). The excess waste heat can be transferred to other buildings, such as swimming halls. Refrigeration heat recovery is correlated to the amount of ice rinks in the building. According to Ministry of Education (2007:47) in ice halls bigger than 4000 m^2 (gross floor area) per one ice rink, the total heat demand of an ice hall becomes bigger than the amount of refrigeration waste heat and thus no excess waste heat would be available. Small ice halls and especially halls with multiple ice rinks should have excess waste heat. Thus, there is no need for excess heat to be transferred from a swimming hall to a small ice hall.

According to Environmental Product Declaration (EPD) (2009:23), ice halls can utilize most of the waste heat that an ice hall produces and during the ice hall closing hours, refrigeration heat recovery drops to about half. Case parameters have a significant effect on the amount of excess heat from an ice hall. For example, the ice hall in Mänttä consumes 1009 MWh of district heat and produces 1210 MWh of waste heat, of which 254 MWh is sold to a nearby swimming hall (Lautiainen, 2018:53, 58).

Only the following two studies were found on utilizing waste heat from an ice hall to a swimming hall. Linhartová and Jelínek (2017) performed a study, where they analyzed how waste heat recovery and demand of different applications match each other in the combined energy system of ice and swimming halls. The study states that waste heat recovery needs to be evaluated on a case-by-case basis due to the differences in energy systems. The study was performed on an ice hall located in Czech and thus the energy amounts differ from corresponding values in ice halls in Finland.

Linhartová and Jelínek (2017:7) state that "only 19 % of the total amount of condensing heat (from ice rink) can be used in the ice arena" and that "the low percentage is caused by a time mismatch between the delivery and consumption". Which indicates that the full utilization in an ice hall needs thermal energy storages. For the heat utilization in swimming halls, Linhartová and Jelínek (2017:7) state, "In the swimming hall, the refrigeration heat may be used for space heating of cloakrooms, for domestic hot water heating and for pool-water heating". Linhartová and Jelínek also conclude (2017:8) that the system is profitable when the refrigeration waste heat from an ice hall is utilized in a nearby swimming hall. Having a thermal energy storage (TES) tank to even out the mismatch should also increase the profitability of the system.

Superheat is a high temperature heat available from heat pumps in amounts ranging from 10 % to 20 % of the total heat pump condenser heat depending on coolant and temperatures (Kianta, 2018; Keinänen 2018; Laitinen & Kosonen, 1994:12). For superheat, Linhartová and Jelínek (2017:5) decided on a design, where water is heated up by superheat and accumulated in a 4 m^3 tank, which sets to a temperature of +40 °C. Figure 15 shows their design as a scheme. Linhartová and Jelínek (2017:4) also found out, that the ammonia refrigerator condensing heat temperature level (from +25 to +33 °C) was too low to be used directly without heating up. Thus, an ammonia heat pump was added into the cycle to raise the temperature to +60 °C.

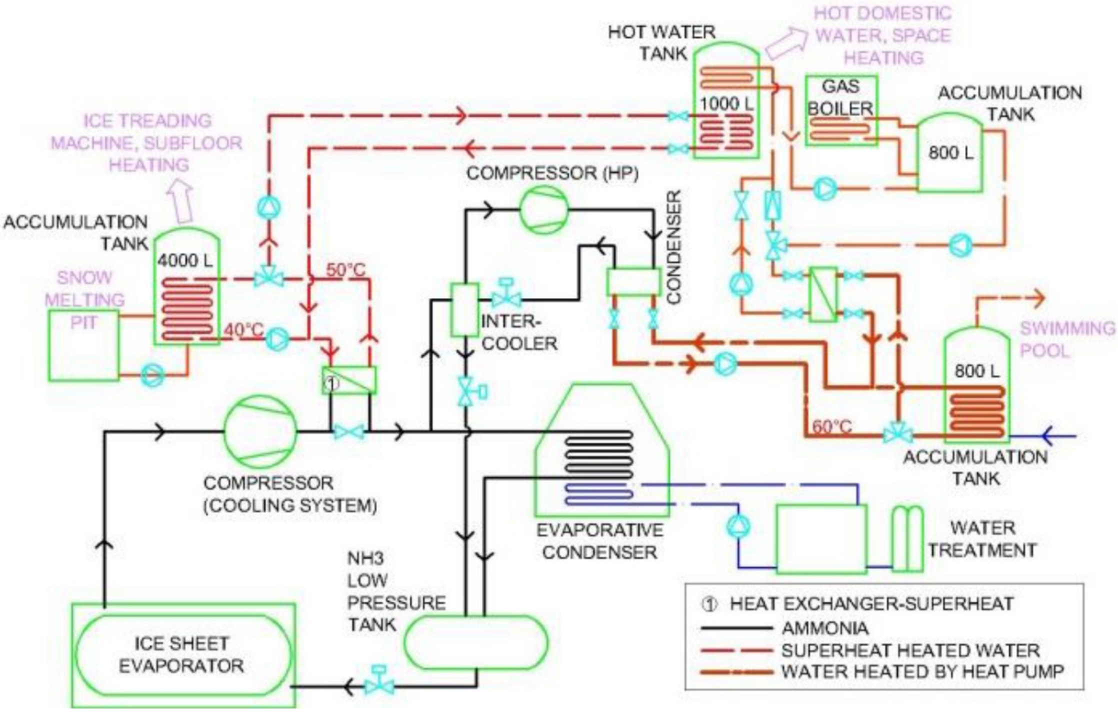

Figure 15. Condensing heat and superheat utilization from ice refrigeration to combined energy system of ice and swimming halls (Linhartová & Jelínek, 2017:5).

Kuyumcu (2016) optimized the performance of a swimming hall heating system in Turkey by utilizing refrigeration waste heat from an ice hall with an underground thermal energy storage (TES) tank (Figure 16). The TES tank was big enough (100…300 m^3) that no hourly mismatch was need to be taken into account between waste excess heat and heat demands. A TES tank of this size is enough to work as a season storage, thus the starting point of the study is different from this thesis. In addition, the ice hall of the study is located in Turkey and it is built in much warmer climate than in Finland.

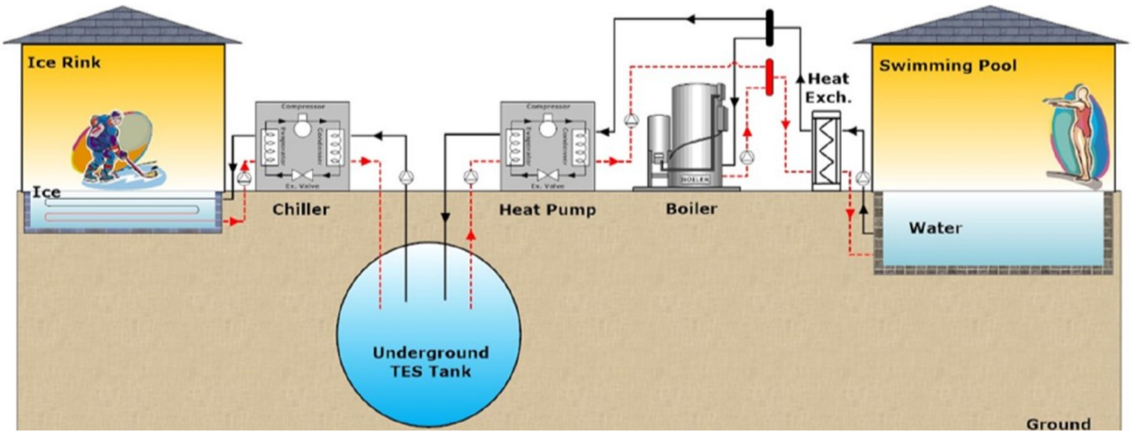

Figure 16. Ice rink heat utilization to a swimming hall energy system with a TES tank (Kuyumcu, 2016).

Kuyumcu (2016:357) concluded, "An ice rink with a size of 475 m^2 gives the optimum performance for a system with a semi-Olympic size swimming pool (625 m^2)". This indicates that Kuyumcu assumed all the heat from refrigeration heat recovery to be transferred to the TES tank and further into the swimming hall, which makes it an optimization problem for relation of ice rink and swimming pool sizes. Kuyumcu (2016:357) also concluded, “The temperature of the water in the TES tank has a great effect on the cooling performance of the chiller and the heating performance of the heat pump”. Meaning the temperature of a TES has a notable effect on the electricity consumption of the system.

To conclude the findings from the studies. Refrigeration heat recovery of an ice hall is beneficial to be used with a swimming hall. Temperature of the condensing heat has to be high enough to be utilized without heat pumps. In addition, at least TES with discharge time of 30 minutes are needed in the system. None of the studies found in the literature took simultaneously into account grey water heat recovery, condensing heat recovery or exhaust air heat recovery.

2.4 Smart control of energy system

Smart control of energy system in this thesis is limited to demand side management. Energy production is not included in this thesis. Different smart controls of energy system discussed in this thesis are demand response of district heat, demand response of electricity, utilizing short-term thermal energy storages and demand controlled energy systems. These strategies for smart control of energy system aim on reducing energy consumption or energy costs. Reducing energy consumption or costs reduces emissions created during energy production.

2.4.1 Demand response

Demand side management adjusts the energy demand of the system based on the strategy used (Figure 17). Demand side management can be divided to long-term and short-term strategies. The short-term strategies are also called demand response and shown in Figure 17. Demand response is widely studied and adopted in modern smart energy grids (Cappers et al., 2010). The two demand response concepts used in this thesis are peak clipping and load-shifting. The peak clipping can limit the load for example based on building load capacity or energy price.

Load-shifting specifically optimizes energy costs by shifting the load from times of high demand to times of lower demand, since demand and energy price correlate. The load-shifting needs a thermal energy storage (TES) to work. Swimming halls including pools and ice halls including layer of ice can act as a natural thermal energy storage (passive), alternatively load-shifting is combined with a TES (active). The sheer volume of the pool is a great asset for load-shifting implementation, even though the pool water temperature can be adjusted only by a few degrees. TES are an expensive option for storing a large amount of heat and thus passive TES should be utilized first.

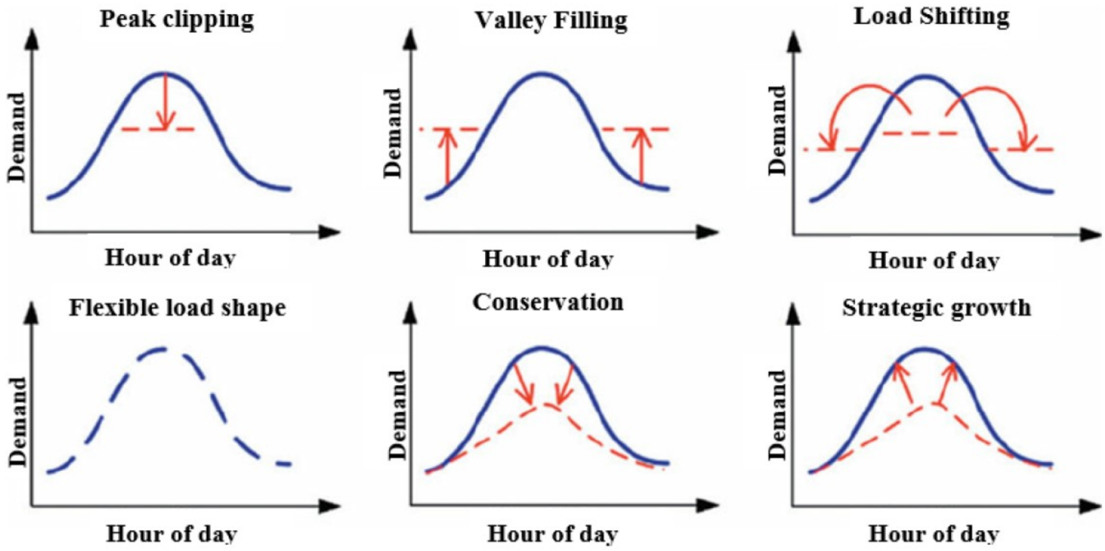

Figure 17. Effect of different strategies for demand side management on energy demand (Shan et al., 2014, edited by Martin, 2017:17).

Figure 18 shows the possible methods on implementing strategies for demand side management. Global temperature adjustment method is a control strategy, which adjusts temperature set points instead of loads. By adjusting temperature set point, the heat load of the heating system changes depending on design power and a control used. Schedule of equipment –method can be used for charging structures extending the time the structure can hold heat with lower heat load. Other methods include straight system adjustment and active thermal energy storages. Onsite renewable generation can also be integrated with a demand response system.

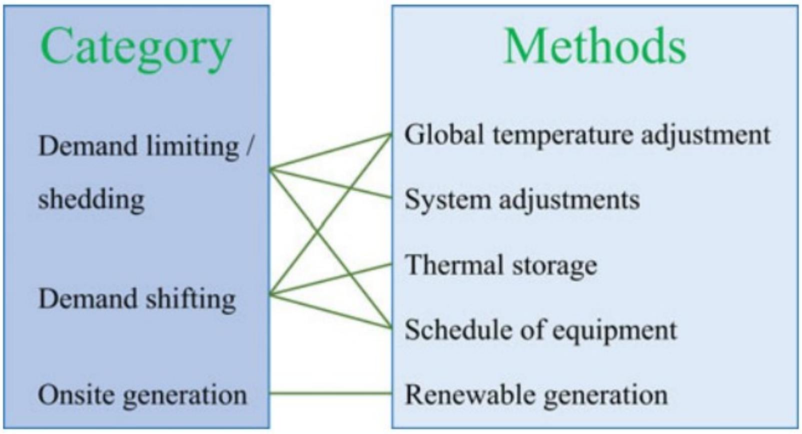

Figure 18. Methods on implementing strategies for demand side management (shan et al., 2016:696).

The possible energy cost savings achieved with demand response are significant. Le Dreau and Heiselberg (2016) decreased heating set point during periods of expensive electricity spot prices and increased heating set point during periods of cheap electricity spot prices. Le Dreau and Heiselberg (2016:1000) achieved energy cost savings between 3 to 10 % for both poorly and well-insulated residential buildings.

Alimohammadisagvand (2018) studied electric demand response in Finnish residential houses with geothermal heat pump and houses with direct electric heating system. Temperature set points controlled in the study were thermal energy storage water and space heating. Alimohammadisagvand (2018:71, 72) concluded that the heating electricity cost was reduced up to 15 %, while thermal comfort was maintained according to standard EN 15251.

District heating use can also be adjusted with demand response. Martin (2017) used dynamic district heat price based control algorithm in an energy simulation software to adjust temperature set points of ventilation supply air and space heating. Martin concluded (2017:116) that by using decentralized heating control, which allows for higher flexibility than centralized heating control, demand response could reduce annual heating costs by 6 %. With peak limiting, Martin (2017:116) was able to achieve even more significant energy cost savings due to reduced energy consumption and power charge. Although the power charge is highly dependent of the energy provider (Martin, 2017:112).

2.4.2 Other strategies for smart control of energy system

Thermal energy storages (TES) can be divided to short-term and long-term TES. TES are usually implemented as water tanks. A long-term TES can act as a seasonal TES, which stores heat during summer and releases it in winter. A short-term TES changes heat load only temporarily and does not require nearly as big TES as in long-term. A short-term TES is needed to avoid time mismatch between waste heat recovery and heat demands. Changes in available waste heat and heat demands happen based on opening hours, summer breaks, outdoor temperature and occupation.

Hemmilä and Laitinen (2018) presented multiple energy saving measures for building services in ice and swimming halls. These methods included energy efficient lighting, demand controlled ventilation and using inverter control for adjusting pump and fan speeds. Specifically for improving energy efficiency of swimming halls, they suggested keeping relative humidity under control with humidification and dehumidification, demand controlled saunas and increasing relative humidity in pool space during nights to reduce evaporation heat loss from pool. Specifically for improving energy efficiency of ice halls, Hemmilä and Laitinen suggested low emissivity coating for inner surface of ceiling to reduce radiation heat exchange between ceiling and ice, reflecting outer surface of ceiling to prevent overheating from sun radiation. In addition for ice halls, they suggested dehumidification of supply air to reduce condensation on ice and better thermal insulation between ice cooling and ground frost protection pipes. (Hemmilä & Laitinen, 2018:21-25)

3 Methodology

Methodology used in this thesis consisted of dynamic building energy simulations and post-processing in a spreadsheet. The simulation program used was IDA Indoor Climate and Energy (ICE) version 4.8. The input data for the simulation were data from Pirkkola, assumed parameters for the models, hourly weather data and hourly energy prices for district heat and electricity. Algorithms for smart control of energy system were implemented in the simulations. The spreadsheet computation program for post-processing was Microsoft Excel 2016. Utilization of waste heat was analyzed as a post-processing in Excel from hourly energy fluxes received from simulations. The resulted total energy needed to purchase was used to calculate the total savings from the investment. Figure 19 illustrates the methodology of this study.

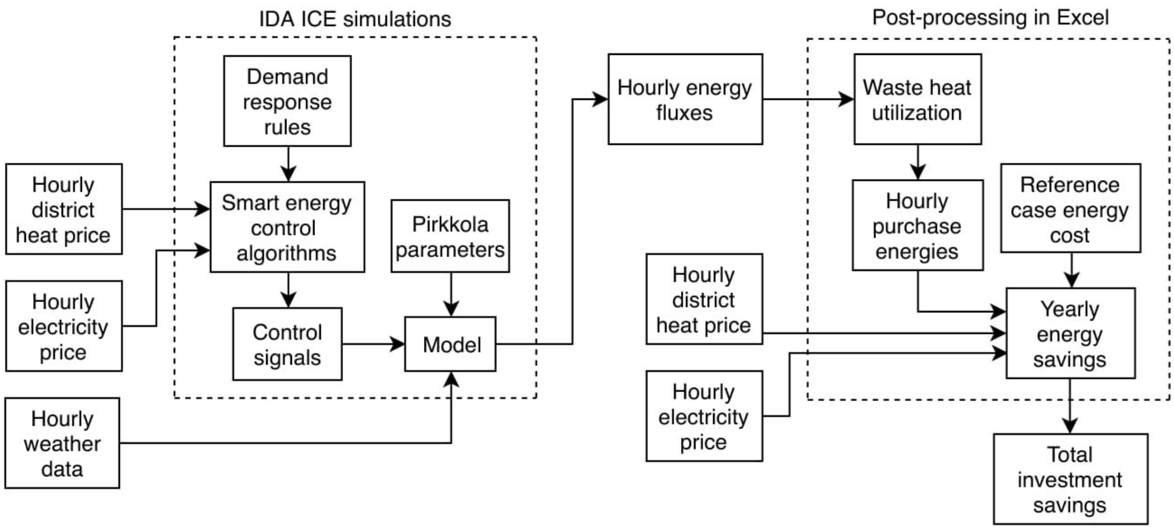

Figure 19. Methodology of the study as a logical diagram.

3.1 IDA ICE building energy simulation tool

IDA Indoor Climate and Energy (ICE) version 4.8 was used as the simulation program. IDA ICE is a dynamic multi-zone simulation program, which can be used to calculate the energy consumption of a building in depth. IDA ICE is a general-purpose simulation tool, which allows the user to simulate a wide range of system designs and configurations (IEA, 1999:3). The validation of IDA ICE simulation tool is performed in multiple studies successfully (EQUA Simulation AB, 2010; Achermann & Zweifel, 2003) and thus the validation of the tool is not needed in this thesis.

IDA ICE takes into account for example solar radiation, shading from other buildings and condensation in zones and inside cooling coils. Zone model fidelity in the simulation program is set on climate. The climate fidelity provides more detailed physical model of the building and its components (EQUA Simulation AB, 2013:54).

In this thesis, the advanced level of IDA ICE was used. In advanced level of IDA ICE, the building energy system components are interconnected by equalities between variables described in a mathematical sense with equations and parameters (EQUA Simulation AB, 2013:32). Appendix B shows two example views of IDA ICE advanced level interface.

Test reference year

A test reference year (TRY) is imported into IDA ICE and used in the simulations. The TRY is introduced by Finnish Meteorological institute (FMI), Aalto University and Tallinn University of Technology from a combination of earlier years to recreate a typical year in Helsinki (Kalamees et al., 2012). The used TRY describes the current climatic conditions of Southern Finland. The TRY defines temperature, relative humidity, wind direction, wind speed and solar radiation on direct normal surface and diffuse horizontal surface. Figure 20 shows temperature and relative humidity of the TRY. Temperature of the TRY ranges between -20 and +30 °C and relative humidity ranges mainly between 30 and 100 %.

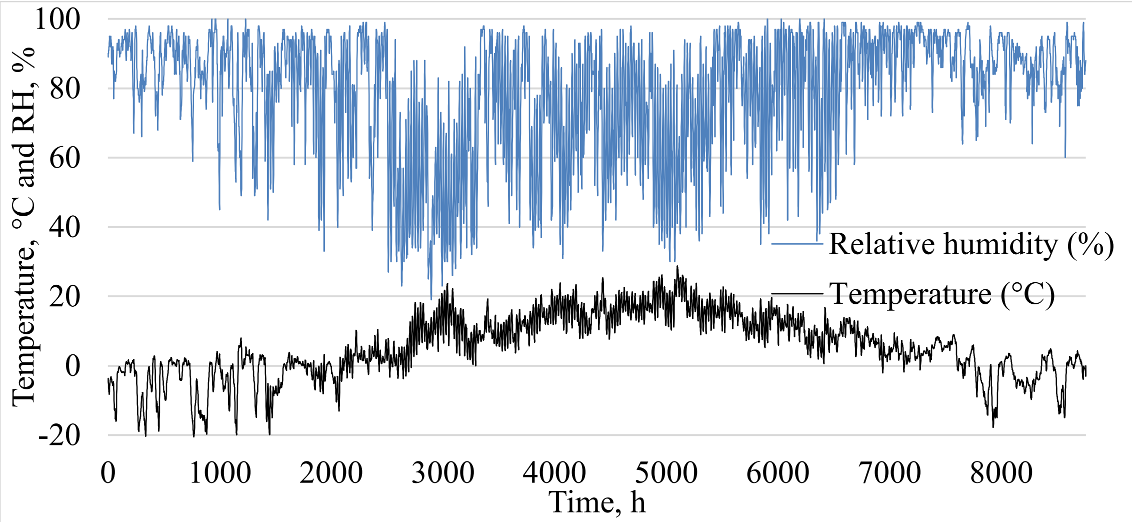

Figure 20. Temperature and relative humidity of the TRY (FMI, 2012).

3.2 The Pirkkola ice hall model

The parameters and assumptions for the Pirkkola ice hall model are presented in this Chapter. The old Pirkkola ice hall has been closed since spring 2018 due to moisture damages in the structures. The ice hall has been decided to demolish and rebuild tentatively during year 2020 with energy efficiency as one of the key priorities. The new ice hall is at the preliminary design phase and thus there are no design parameters available. In addition, energy consumption of the new ice hall has to be evaluated with simulations and references. (Priha, 2018)

The Pirkkola ice hall has two ice rinks and works mainly as an exercise hall and the heating set point is assumed to be at a relatively low temperature of +6 °C. The ice hall is operating from 13.8 to 31.4 (Korva, 2018). In this study, the ice hall is assumed to have the same amount of occupants as before the closing of the hall.

Energy consumption

The Pirkkola ice hall has a closely located swimming hall. The Pirkkola ice hall with a size of 6700 m^2 (gross floor area) and two ice rinks can be called either a training hall or a small ice hall (Hemmilä & Laitinen, 2018:8). The Pirkkola ice hall has about 1000 seats for spectators (Korva, 2018). For a corresponding size of ice halls compared to spectator stand seats in Pirkkola (300 – 1500), the average heat consumption per gross floor area is 142 kWh/m^2 (sample size of 28) and electricity consumption is 197 kWh/m^2 (sample size of 51) (Hemmilä & Laitinen, 2018:16). By multiplying these values by the Pirkkola ice hall gross floor area of 6741 m^2, the first approximation (1) for the total energy consumptions are 1000 MWh of district heat and 1300 MWh of electricity (Table 3).

Another approximation (2) is shown in Table 3 by comparing the Pirkkola ice hall to a similar the ice hall in Myllypuro (Helsinki). The ice hall of Myllypuro consumed between years 2013 to 2015 average of 500 MWh of district heat and 1500 MWh of electricity according to Jyväskylä University database (2018a). The small amount of district heat in the ice hall of Myllypuro indicates utilization of waste heat.

During year 2017, Pirkkola ice and swimming halls had a combined district heat consumption of 4850 MWh and electricity consumption of 3200 MWh, but the consumptions are not specified by buildings and only the total monthly heat and electricity consumption are known for Pirkkola (Appendix C). This allows for an another approximation on the ice hall energy consumption, since the latest months (September and October of 2018) do not include the ice hall consumption as it is closed. This approximation (3) for total annual average consumptions for the Pirkkola ice hall are for heating 2100 MWh and for electricity 1600 MWh (Appendix C) (Table 3).

The main design target of the Pirkkola ice hall is energy efficiency. The Pirkkola ice hall will thus assumable consume especially less electricity than these approximations, due to state-of-the art technologies of the building systems. These values are rough approximates. The energy consumptions of IDA ICE simulation were compared with the approximated energy consumptions (Table 3).

Table 3. Annual energy consumptions of three different approximations for the Pirkkola ice hall.

	District heat [MWh]	Electricity [MWh]
Approximation 1	957	1328
Approximation 2	481	1474
Approximation 3	2107	1568
Average	1182	1457

Envelope

The new ice hall will be the same size as the old one, with small changes in layout (Priha, 2018). The old the Pirkkola ice hall is shown in Figure 21. The geometry of the simulation model is built according to facade and floor plans of the old Pirkkola ice hall, which are received from online database held by City of Helsinki (2018) and from team manager of Pirkkola (Korva, 2018). The geometry of the model is built with AutoCAD MagiROOM extension made by AutoDesc. The rink spaces of the ice hall are simulated with flat ceilings instead of actual arched ceilings. The heights of the halls (6 and 10 meters) are chosen to have the same volume of the hall than before. Figure 22 visualizes the model used and Table 4 lists main values of the geometry.

Figure 21. Pirkkola ice hall before renovation (Google, 2019).

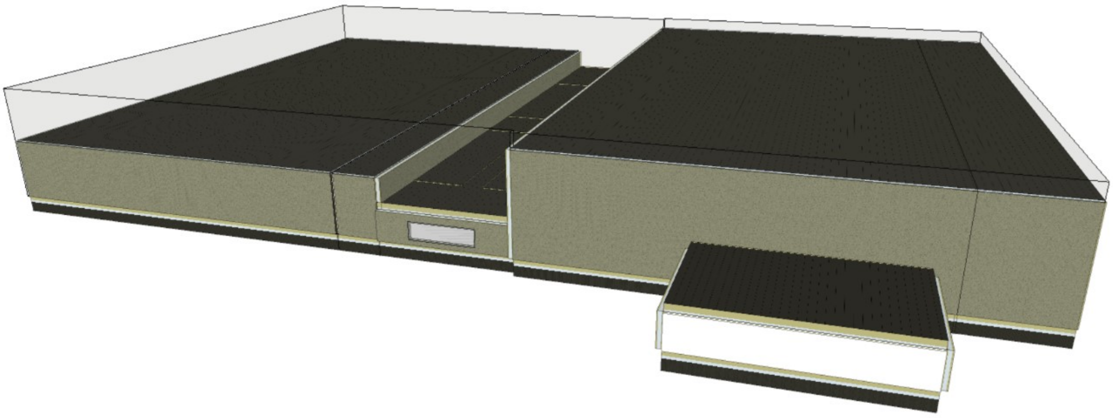

Figure 22. Model geometry of the simulated Pirkkola ice hall.

Table 4. Model geometry of the Pirkkola ice hall.

Model floor area	Model volume	Model ground area	Model envelope area	Window / Envelope area	Average U-value	Envelope area per Volume
[m²]	[m³]	[m²]	[m²]	[%]	[W/(m²K)]	[m²/m³]
6 674	48 780	6 741	16 671	0.10	0.1891	0.3417

Table 5 presents parameters for the envelope of the ice hall model. U-values for ceiling and walls of the ice hall model are chosen to be 0.28 W/(m^{2}K). According to study by Partanen (2014:92), these U-values do not have a significant effect on energy consumption of the ice hall. U-values for warm spaces are chosen according to Finnish code of building regulation (FCBR Part D3, 2012:13). Low-emissive coating with emissivity factor of 0.1 is used inside of the ice hall ceilings. The coating decreases electricity consumption by 15 to 20 % and heat consumption by 17 to 20 % (Hemmilä and Laitinen, 2018:68). Thermal bridge extra conductances are chosen according to the design values of Ministry of Environment (2017c:19). New ice halls have usually a tight envelope. Average annual infiltration rate of 0.03 1/h was used in the simulations based on the measurements by Toomla et al (2018). The used infiltration rate corresponds to air leakage rate of q_{50} = 2.1 m^3/h/m^2 based on the approximation defined by Ministry of Environment (2017a), when assuming a height factor of 20.

Table 5. Envelope parameters for the ice hall model.

Structure	location	U-value [W/(m²K)]	Layers from outside to inside: material, thickness [mm]
Base slab	Ice rink hall Warm space	0.16	Polystyrene, 230 Concrete, 40 Vinyl flooring, 5
Ceiling	Ice rink hall	0.28	Gypsum, 6 Steel, 6 Light insulation, 122 Steel, 6 Low-emissive coating (ε=0,1)
	Warm space	0.09	Bitumen felt, 10 Mineral wool, 49 Concrete, 15 Render, 10
External wall	Ice rink hall	0.28	Steel, 6 Light insulation, 122 Steel, 6 Gypsum, 6
	Warm space	0.17	Render, 10 Concrete, 10 Mineral wool 250 Concrete, 10 Render, 10
Internal wall	Ice rink / Warm space	0.4	Concrete, 10 Light insulation, 80 Concrete, 10
	Warm space / Warm space	0.62	Gypsum, 26 Air gap, 30 Light insulation, 30 Air gap, 30 Gypsum, 26

Air leakage rate q_{50} [$m^3/h/m^2$]	2.1	

Additional thermal bridge conductance [W/(mK)]		
Thermal bridges	Ceiling / External wall	0.08
	Base slab / External wall	0.24
	External wall / External wall	0.06
	External window and door	0.03

Ice rinks

The Pirkkola ice hall has two ice rinks. The ice rinks are identical in the model. The parameters of the rinks are presented in Table 6. According to International Ice Hockey

Federation (IIHF), the recommended ice thickness is 3.0 cm and ice temperature for a training hockey game is -3 °C (IIHF, 2016:38, 50). The coolant used in the model is Freezium, which is developed by Eastman and is used in indirect cooling systems and heat pumps (Arteco, 2019). The chiller cooling capacity is 400 kW to supply two ice rinks even during big heat load from spectator stand full of spectators. The schematic of the cooling system of ice rinks are shown in Figure 23. The cooling system of ice rinks includes PI-controlled condenser fan, two pumps, two expansion valves, two water tanks and PI-controlled refrigerator. In the model, the condenser fan (Figure 23) transfers the refrigerator waste heat to air. Instead of the condenser fan, the utilization of refrigerator waste heat is calculated as a post processing in a spreadsheet.

Table 6. Parameters for ice rinks cooling system.

Temperature set point for ice	3	[°C]
Chiller total cooling capacity	400	[kW]
Coolant	Freezium	
Coolant freezing point	-35	[°C]
Cooling power	450	[W/m²]
Supply coolant temperature	-12	[°C]
Return coolant temperature	-9	[°C]
Pump efficiency	0.8	
Pump max pressure head	3000	[Pa]

Ice layer thickness	0.03	[m]
Concrete thickness	0.4	[m]
Pipe depth in concrete	0.05	[m]
Supply line length	40	[m]
Supply line diameter	0.3	[m]
Length per coil	120	[m]
Coil inner diameter	0.06	[m]
Number of coils	145 x2	

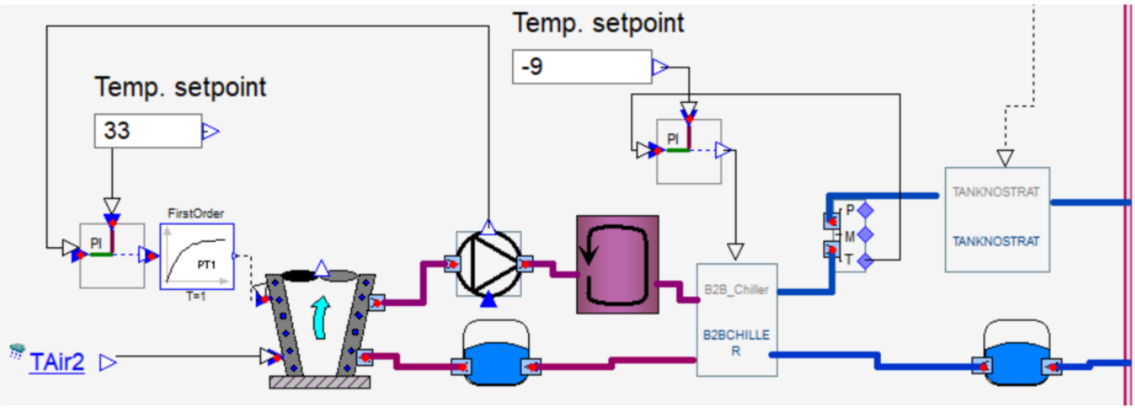

Figure 23. The schematic of ice rinks cooling system.

Heat loads to ice are simulated in IDA ICE with an exception of ice resurfacing heat load. Ice rinks in the Pirkkola ice hall are both resurfaced 40 times per week with 450 liters per run (Jyväskylä University, 2018a). Assuming a little higher use for the upcoming ice rinks, a value of 45 resurface runs per week is used. Ice resurfacing hot water is heated to an energy efficient temperature of +32 °C (Korva, 2018). The ice is assumable transferred outside and it does not need a separate ice-melting heat. Ice resurfacing heat load is calculated manually as a post-processing according to Laitinen et al. (2010:53):

$$q_{IR} = \frac{V_{IR} n_{IR} \rho_w}{3600 \cdot \Delta t \, A_{ice}} [c_{p,v}(T_{IR} - 0) + l_f + c_{p,j}(0 - T_{ice})] \qquad (7)$$

where

q_{IR} average heat load to ice from ice resurfacing [W/m²]

V_{IR} water amount per one ice resurfacing [m^3]
n_{IR} number of ice resurfacing uses
ρ_w density of water [kg/m^3]
Δt time period [h]
A_{ice} ice surface area [m^2]
$c_{p,v}$ specific heat capacity of water [4190 Ws/kgK]
T_{IR} temperature of used water for ice resurfacing [°C]
l_f condensing heat of fusion from liquid to solid [334 000 Ws/kgK]
$c_{p,j}$ specific heat of ice [1800 Ws/kgK]
T_{ice} average ice temperature [°C].

Schedules

The ice hall weekly opening times are as follows: from 7:00 to 22:30 except for Saturday from 13:00 to 22:30. DHW use is scheduled once every 1.5 hour for half an hour, which simulates the training session cycles in ice rinks. The locker room occupancy (24 people per locker room) also follows the same 1.5 hour cycle. Ice resurfacing machine is assumable filled at even rate during training sessions. Lighting is kept on during opening times.

Occupancy of the two ice rink spaces including spectators and players are shown in Table 7 (Korva, 2018The length of the spectator occupancy time per game is 2 hours. Three hockey games with different spectator amounts are held once a week. A game with full spectator stand is held once in every two months. The two ice rinks are occupied fully (48 people per ice rink) during opening times.

Table 7. Occupancy of the two ice rink spaces in the ice hall model.

Space	Occupants	Frequency	Time
Ice rink space with big spectator stand	48 players	Day	Opening times
	250 spectators	Week	Sunday 17-19
	750 spectators	2 months	Sunday 17-19

Ice rink space with small spectator stand	48 players	Day	Opening times
	62 spectators	Week	Saturday 13-15
	125 spectators	Week	Sunday 13-15

Air handling units (AHU)

The AHU:s are built specifically for this thesis. Ice rink spaces with spectator stands are supplied with a variable air volume (VAV) system based on CO_2 concentration. A minimum outdoor airflow per net floor area is chosen to be 0.5 m^3/h/m^2 according to Laitinen and Kosonen (1995:12), which corresponds in the ice hall model to a total airflow of 0.8 m^3/s. During maximum occupancy 875 spectators each need 6 dm^3/s of outdoor air, 12 players on both ice rinks each need 30 dm^3/s of outdoor air and 36 players resting on benches each need 12 dm^3/s outdoor air (FINVAC, 2017:16). This maximum occupancy requires 6.1 m^3/s of outdoor air, which is set to the maximum outdoor air of the AHU.

The Ice rink space AHU (Figure 24) is equipped with condensing dehumidification by cooling the supply air to a temperature of 0 °C. In addition, the AHU uses recycling air when needed to dehumidify and heat the space. During games with a full spectator stand, the VAV recycles up to 14 m^3/s to keep the relative humidity below 70 %. The heating set point of indoor air is +6 °C, which is achieved by heating the supply air with a PI-controller to a temperature between +6 and +25 °C. The heat exchanger efficiency is 0.75 (Nichols, 2009).

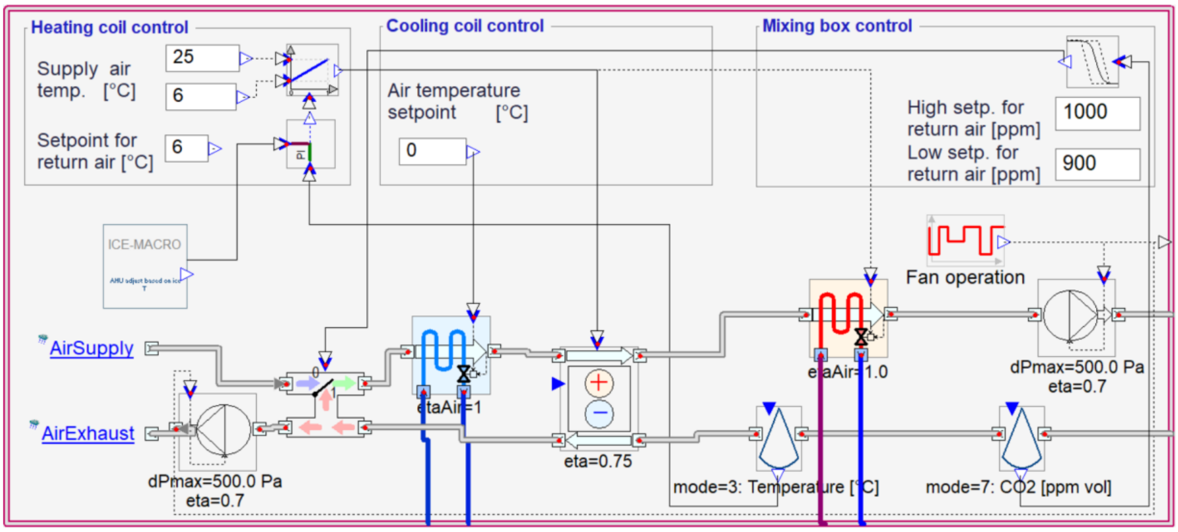

Figure 24. Schematic of AHU serving ice rinks and spectator stands in IDA ICE 4.8.

Warm spaces have a separate AHU, which has a constant supply air heating to a temperature of +17 °C. The airflow rate is controlled according to occupancy schedule. Maximum airflow is 3 $dm^3/s/m^2$ related to net floor area (FINVAC, 2018:16). The minimum airflow of 0.3 $dm^3/s/m^2$ is kept during times of no occupancy including closing times. The warm space AHU has a heat exchanger with an efficiency of 0.6.

In Table 8, different space groups of the ice hall model are listed with main average parameters, heat loads and set points. Ice rinks include spectator stands. Warm spaces include hallways and social spaces. The value of 1 Metabolic Equivalent of Task (MET) is equal to 104 W for an average human.

Table 8. Parameters of the ice hall model for different space groups.

Space group	Surface area	Heating set point	Minimum outdoor airflow	Maximum outdoor airflow	Lighting when used	Occupancy average	Average MET of occupant
	[m^2]	[°C]	[$dm^3/s/m^2$]	[$dm^3/s/m^3$]	[W/m^2]	[$1/m^2$]	
Ice rinks	5624	6	0.8	6.1	5	0.012	3.0
Locker rooms	426	21	0.3	3.0	12	0.047	2.0
Warm spaces	625	21	0.3	3.0	12	0.012	1.8

DHW usage

DHW heating set point in Pirkkola is +55 °C (Appendix D), which is also the minimum allowed temperature for DHW, according to Finnish Energy (FE, 2013:12). Incoming cold-water temperature is according to Finnish building code +5 °C when calculating energy consumption (FCBR part D5, 2017:69). The actual value is in practice higher, since the incoming water is heated up in warm underground tunnels, along which the cold water arrives. Finnish Energy defines in an example cold-water temperature to be +10 °C (FE, 2013:80). The chosen value for incoming cold-water in Pirkkola energy calculation is +8 °C.

Total DHW usage in the ice hall is approximated by calculating the average DHW usage per day and scaling it by DHW usage schedule. The average water usage per day is

$$V_{water,d} = q_{sh} \cdot t_{sh} \cdot n_{g,d} \cdot n_{sh,g} \\ = 0{,}15 \cdot 10^{-3} \frac{m^3}{s} \cdot 170\ s \cdot 19 \cdot 40 = 19 \frac{m^3}{day} \tag{8}$$

where

$V_{water,d}$	average water usage per day [m^3/d]
q_{sh}	average shower flow [m^3/s]
t_{sh}	average time of a shower [s]
$n_{g,d}$	average number of games per day [1/d]
$n_{sh,g}$	average number of showers per game

and the total DHW usage per day is

$$V_{DHW} = \frac{T_{sh} - T_{cw}}{T_{DHW} - T_{cw}} \cdot V_{water} \\ = \frac{38\ °C - 8\ °C}{55\ °C - 8\ °C} \cdot 19 \frac{m^3}{day} = 12 \frac{m^3}{day} \tag{9}$$

where

$V_{DHW,d}$	average domestic hot water usage per day [m^3/d]
T_{sh}	average shower temperature [°C]
T_{cw}	cold water temperature [°C]
T_{DHW}	domestic hot water temperature [°C]
$V_{water,d}$	average water usage per day [m^3/d].

Heating system temperatures

Different heating systems work at specific temperatures. The design values for these temperatures for the ice hall model are presented in Table 9. Before heating, supply air is cooled to a temperature of 0 °C, recovers heat in heat exchanger and then finally heated in a heating coil to a temperature between +14 and 29 °C depending on current heat demand. Water radiator heating is used in other warm spaces. Ground frost protection heats the ground below ice rinks to a temperature of +4 °C according to Sutherland (2015:79).

Table 9. Design temperatures for heating systems in the ice hall model.

Heating system	Working temperatures [+ °C]
Supply air heating	0…4 / 6…25
Water radiator heating	40 / 50
Ground heating	7 / 14
Domestic hot water heating	8 / 55
Ice resurfacing hot water heating	8 / 32

3.3 The Pirkkola swimming hall model

The parameters and assumptions for the Pirkkola swimming hall model are presented in this Chapter. The Pirkkola swimming hall is built in 1968 and renovated at 2002. The building U-values of structures are relatively poor compared to modern regulations. The windows are changed during the renovation (Korva, 2018). The building technical systems are assumed to match the modern systems installed in 2002. For this study, Pirkkola swimming hall has comprehensive data to build the model on, except energy consumption of the swimming hall is unknown and it has to be evaluated with simulations and references.

Energy consumption

Pool surface area in the Pirkkola swimming hall is 575 m^2. For a corresponding size of swimming halls (500 – 750 m^2 of pool surface area), the average heat consumption per gross floor area is 492 kWh/m^2, and electricity consumption is 267 kWh/m^2 (sample sizes 26) (Hemmilä & Laitinen, 2018:12,14). By multiplying these values by the Pirkkola swimming hall gross floor area of 7982 m^2, the first approximation (1) for the total energy consumptions are 4000 MWh of district heat and 2100 MWh of electricity (Table 10).

In year 2017, Pirkkola swimming and ice halls had a combined district heat consumption of 4850 MWh and electricity consumption of 3200 MWh, but the consumptions are not specified by building (Appendix C). The total monthly heat and electricity consumption are known for combined swimming and ice halls in Pirkkola. This allows for second (2) approximation on the swimming hall energy consumption, since the latest months (September and October of 2018) do not include the ice hall consumption as it is closed. The approximation (2) of total annual average consumption for the Pirkkola swimming hall are for heating 2700 MWh and for electricity 1600 MWh (Appendix C) (Table 10).

The approximation 2 in Table X can be held more valid than approximation 1 as Pirkkola keeps energy consumption as low as possible with relatively low heating set points (Appendices D and E). The approximated consumptions can be further divided to different systems according to energy distribution of swimming halls presented in Chapter 2.1.2 (Figure 1). IDA ICE simulations are compared with the approximated energy consumptions (Table 10).

Table 10. Annual approximated energy consumptions of Pirkkola swimming hall.

	District heat [MWh]	Electricity [MWh]
Approximation 1	3927	2131
Approximation 2	2743	1632
Average	3335	1882

Pools

The Pirkkola swimming hall has three swimming pools, a big pool for fitness swimming, a children pool for practicing swimming and a pool for young children. The big pool and the children pool are located in the same space and share an AHU (Figure 25). Young children pool is located in a separate space and has its dedicated AHU. Table 11 shows parameters for these pools. Evaporation coefficients for IDA ICE model are chosen according to ASHRAE (2003:4.6). The 10 meter long water slide in small pool increases water evaporation by 5 kg/h (EPD, 2009:5). Design power of the model pools is set to 200 W/m^2 per pool surface area. Design supply water temperature controlled by PI-controller is +37 °C. Pool water temperatures are set according to mean temperature from latest measurements from Pirkkola (Nikkola, 2018).

Table 11. The swimming hall parameters.

Pool	AHU	Pool surface area [m^2]	Average depth [m]	Tempe-rature [°C]	Evaporation coefficient	Evaporation and supply water flow [kg/s]
Big pool	Big pools	400	2.8	26.5	1	0.024
Children pool	Big pools	112	1.4	26.5	1.5	0.010
Young children	Small pool	63	0.75	28.0	1	0.006

Figure 25. The main pools in the Pirkkola swimming hall (City of Helsinki, 2019).

Envelope

The geometry of the swimming hall model is built according to facade and floor plans for the Pirkkola swimming hall, which are received from online database held by City of Helsinki and from the team manager of Pirkkola (City of Helsinki, 2018; Korva, 2018). The geometry of the model is built with AutoCAD MagiROOM extension made by AutoDesc. Figure 26 shows the geometry visually and Table 12 lists main values of the swimming hall model geometry.

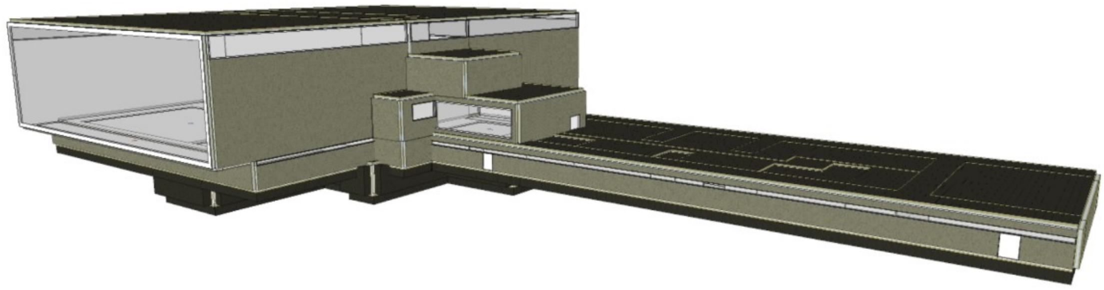

Figure 26. Model geometry of the Pirkkola swimming hall.

Table 12. Main values of the model geometry of the Pirkkola swimming hall.

Model floor area	Model volume	Model ground area	Model envelope area	Window /Envelope	Average U-value	Envelope area per Volume
[m^2]	[m^3]	[m^2]	[m^2]	[%]	[$W/(m^2K)$]	[m^2/m^3]
7 982	53 463	4 047	13 705	8.90	0.2696	0.2563

Table 13 presents parameters for the envelope of the swimming hall model. The U-values of the Pirkkola swimming hall are not known. The window U-value is the most significant parameter due to high window surface area in the big pool space. The windows are set to 1.0 W/m^2K according to Finnish code of building regulation (FCBR Part D3, 2012:13). The rest of the envelope U-values for the model are chosen to be between 0.20 and 0.24 W/m^2K. Pool spaces are set to have better U-values than rest of the internal walls, since pool spaces are significantly warmer than other spaces. Table 13 also shows structure layer materials and thicknesses. The layers in the model mostly consist of concrete and insulation. Saunas have a reduced concrete thickness to reduce excess heating of the structures.

Thermal bridge extra conductances are chosen according to design values by Ministry of Environment (2017c:19). Hemmilä and Laitinen (2017:45) used q_{50} =1.0 $m^3/h/m^2$ related to envelope surface area in their simulation of the swimming hall. Envelope of an old Pirkkola swimming hall building is assumable not nearly as tight. Average annual infiltration rate of 0.6 $m^3/h/m^2$ was used in the simulations corresponding to air leakage rate of q_{50} = 3.3 $m^3/h/m^2$ based on the approximation defined by Ministry of Environment (2017a), when assuming a height factor of 20.

Table 13. Parameters for the envelope of the swimming hall model.

Structure	location	U-value [W/(m²K)]	Layers from outside to inside: material, thickness [mm]
Base slab	Saunas	0.24	Light insulation, 140 Concrete, 10
	All other spaces		Light insulation, 140 Concrete, 20
Ceiling	All spaces	0.2	Bitumen felt, 9 Concrete, 150 Mineral wool, 210 Render, 10
External wall	All spaces	0.23	Render, 10 Concrete, 10 Mineral wool 180 Concrete, 10 Render, 10
Internal wall	Pool spaces	0.47	Concrete, 10 Light insulation, 50 Concrete, 10
	All other spaces	0.8	Wood, 20 Air gap, 20 Light insulation, 20 Air gap, 20 Gypsum, 20

Air leakage rate q_{50} [$m^3/h/m^2$]	3.3	

Additional thermal bridge conductance [W/(mK)]		
Thermal bridges	Ceiling / External wall	0.08
	Base slab / External wall	0.24
	External wall / External wall	0.06
	External window and door	0.03

Schedules

Annual opening schedule in the swimming hall is from 14.7 to 31.5. The swimming hall is open every weekday from 7:00 to 22:30. In the model, lighting is off during closing times, at half power during weekdays from 7 to 16 and at full power during other opening times. Weekly schedules for occupancy and DHW usage of the model are shown in Figure 27. Maximum number of occupants and DHW usage correspond to a value of 1.0 in Figure 27. The swimming hall spectator stand is fully occupied (300 people) once per 2 months from 18 to 21 on Sundays. The spectator stand is occupied at 10 % usage during other Sundays from 18 to 21 and not occupied during rest of the time.

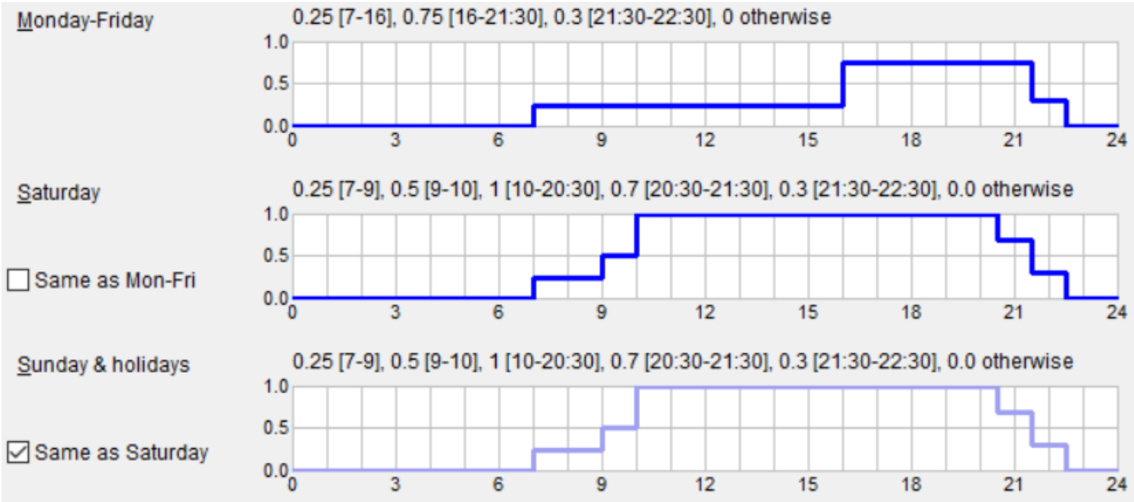

Figure 27. Weekly schedules for occupancy and DHW usage of the swimming hall model.

Saunas have a heating schedule according to opening times. Saunas are heated from a nighttime temperature of +18 °C to a temperature of +85 °C. A leak area of 0.08 m^2 is set to sauna doors. Sauna doors are also left open 10 % during opening hours to simulate people entering and exiting saunas. Each of the 6 saunas has an electric heater with a maximum heating power of 40 kW. The use of saunas and showers also emit liquid water droplets, which remove heat from air. Each sauna is set to emit maximum of 10 kg/h and each shower maximum of 2.6 kg/h, both related to occupancy schedule (Kokko et al., 1999:15).

Air handling units (AHU)

Existing AHU schematics for the Pirkkola swimming hall were provided from Pirkkola (Appendix E). The most significant are the AHU:s serving the two pool spaces, big pool space and the young children pool space. These AHU:s are built in IDA ICE accurately. To both pool spaces, supply airflow rates are supplied with own VAV AHU based on relative humidity (RH). The schematic of the AHU is shown in Figure 28. Moisture content is the dimensioning factor for airflow rate of pool space AHU (FINVAC, 2017:16). The supply airflow per net floor area changes from 2 $dm^3/s/m^2$ to 4 $dm^3/s/m^2$ for values of 50 to 57 % RH respectively. Recycling of indoor air is used to replace 0.5 to 0 portion of outdoor supply air to increase RH from low values of 40 % to 50 % respectively. A cooling coil of exhaust air is used with an exhaust air heat pump (EAHP) . The cooling coil also recycles exhaust air to replace 0 to 0.5 portion of outdoor air, when RH increases to values of 57 and 60 % respectively, which works as an emergency condensing dehumidification. The total outdoor air per occupant is set to 10.0 dm^3/s. The heating set point of indoor air in big pool space is +28 °C, which is achieved by heating the supply air between +25 and +37 °C. The corresponding temperature set points are 2 °C higher for AHU supplying young children pool space. The heat exchangers have an efficiency of 0.60 according to Hemmilä and Laitinen (2017:45).

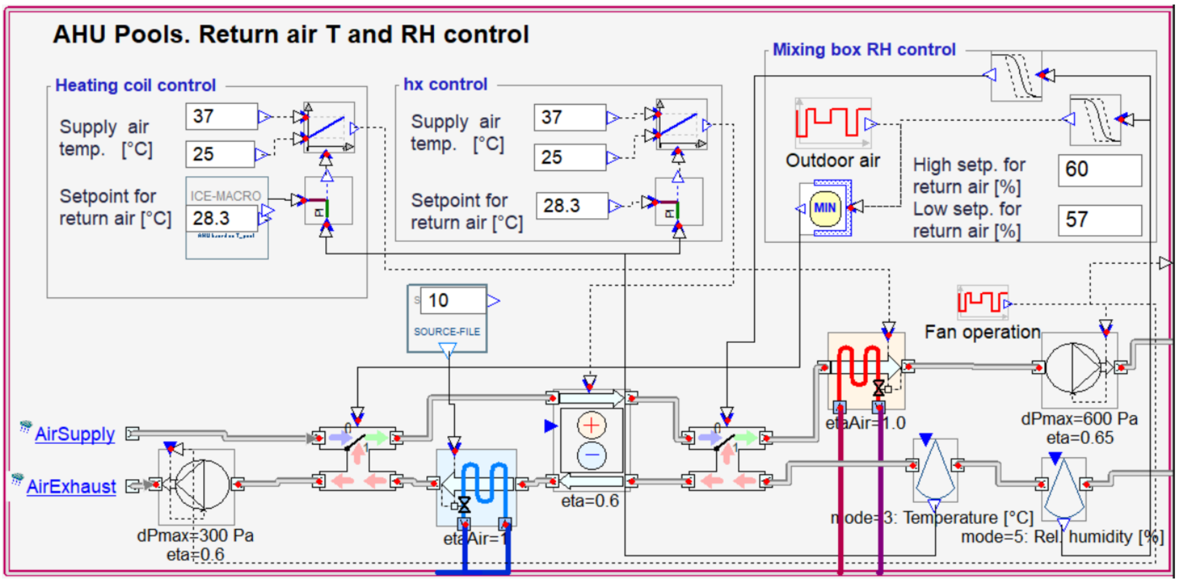

Figure 28. Schematic of the two AHU:s serving pool spaces of the swimming hall model.

Warm spaces and showers both have own VAV AHU, which supply airs are increased linearly based on CO_2 levels between 1000 and 1200 ppm. Warm space AHU has a constant supply air temperature of +16 °C and the supply air temperature of shower AHU varied between +13 and 30 °C based on the room air temperature. The smaller temperature limit of shower AHU is low because of heat loads from saunas. Both AHU:s have a heat exchanger with an efficiency of 0.60, but only shower AHU has an exhaust air heat pump (EAHP).

Table 14 lists average parameters of the four AHU serving areas. Spectator stand is located in same space as big pools. Shower AHU also serves saunas. Warm spaces include sports field, gym, dressing rooms, technical spaces and rest of the dry spaces.

Table 14. Average parameters of the four AHU serving areas in the swimming hall model.

Space group of AHU	Net surface area	Heating set point of air	Min. outdoor airflow	Max. outdoor airflow	Lighting when used	Occupancy average	Average MET of occupant	Equipment during opening times
	[m^2]	[°C]	[dm^3/s/m^2]	[dm^3/s/m^2]	[W/m^2]	[1/m^2]		[W/m^2]
Big pools	1 144	28	2.0	4.0	7.5	0.342	4.0	2.0
Young children pool	231	30	2.0	4.0	7.5	0.200	2.0	2.0
Showers	411	24	3.3	6.7	7.5	0.100	1.6	1.9
Other spaces	6 196	18	0.5	2.6	5.1	0.025	1.6	2.2

The amount of supplied outdoor airflow rate for sports halls and gym is 30 dm^3/s per occupant (FINVAC, 2017:16). Shower outdoor air per net floor area is dimensioned based on heavy usage needing minimum of 5 dm^3/s/m^2 and dressing room minimum of 3 dm^3/s/m^2 (FINVAC, 2017:16). Humid spaces are controlled to be under pressured by setting return airflow from 10 to 30 % lower than supply air according to EPD (2009:6-8). Big pools occupants are given a value of 4 MET, which is an average of swimming (6 MET) and resting at spectator stand (2 MET). Swimming small children are assumed to have a MET of 2. Sports field and gym occupants have a MET of 6 and occupants in rest of the spaces have a MET of 1.6.

DHW usage

DHW heating set point in Pirkkola is +55 °C (Appendix D). The selected value for incoming cold-water in Pirkkola energy calculations is +8 °C. Average water usage is set to the same as the consumption during years 2010 and 2016, which is 30700 m^3 according to database held by Jyväskylä University (2018b). Assuming average water temperatures (Table 15) and the usage portion according to Hemmilä and Laitinen (2018:13), the average used of the hot water temperature becomes +39 °C.

Table 15. Water usage breakdown in the swimming hall model.

Water type	Portion	Temperature [°C]	Portion times Temperature [°C]
Showers	0.54	38	20.5
Filter washing	0.23	45	10.4
Space washing	0.13	40	5.2
Pool resupply	0.08	28	2.2
Pool changing	0.02	28	0.6
Sum	1.00		38.9

Since the water meter is shared for swimming and ice halls (Korva, 2018), the ice hall hot water usage is reduced from the total (Equation 10). Now the average DHW usage per day can be calculated in the swimming hall, which is

$$q_{DHW,d,SH} = \frac{q_{water,a} - q_{water,a,IH}}{t_{days,open,SH}} \cdot \frac{\Delta T_{DHW}}{\Delta T_{avg,use,SH}} \tag{10}$$
$$= \frac{30700\ m^3 - 12.22\frac{m^3}{d} * 261\ d}{322\ d} \cdot \frac{(55-8)K}{(38{,}9-8)K} = 51\frac{m^3}{d}$$

where

$q_{DHW,d,SH}$ average DHW usage per day in the swimming hall [m^3/d]
$q_{water,a}$ annual water usage in Pirkkola ice and swimming halls [m^3/a]
$q_{water,a,IH}$ annual water usage in the Pirkkola ice hall [m^3/a]
$t_{days,open,SH}$ days the swimming hall is open during a year [d]
ΔT_{DHW} temperature difference between cold water and DHW [K]
$\Delta T_{avg,use,SH}$ temperature difference between cold water and average used water in the swimming hall [K].

Heating systems and temperatures

Heating system temperatures for the swimming hall model are presented in Table 16. Supply air recovers heat in a heat exchanger and afterwards is heated with a heating coil to a value between +13 and 37 °C depending on heat demand. Supply air heating is the main heating system for pool spaces. Water radiator heating is used in all spaces except shower rooms. Underfloor heating is used in pool spaces and shower areas as supportive heating for supply air heating. The design powers for underfloor heaters are for big pool space 40 kW, small pool space 6 kW and for showers 24 kW. The design heating powers for water radiators are for big and small pool spaces 23 kW and 5 kW respectively.

Table 16. Design temperatures for heating systems in the swimming hall model.

Heating system	Temperatures [+ °C]
Supply air heating	10…28 / 13…37
Water radiator heating	30 / 50
Underfloor heating	30 / 34
Domestic hot water heating	8 / 55
Pool water heating	27 / 40

3.4 Utilization of waste heat

Post-processing calculations were made in this thesis to analyze how building energy systems of ice and swimming halls can utilize waste heat. In the utilization of waste heat, there should be taken into account temperature levels, temperature differences between sub-systems and mismatch of heat demands and available excess heats.

The main sub-systems in the utilization of waste heat are heat pumps and thermal energy storages (TES). Figures 29 and 30 show heat fluxes between heat pumps, TES and ice and swimming halls. By combining the energy systems of ice and swimming halls, the utilization ratio of waste heat is increased. The merging of the two energy systems is implemented by connecting excess heat from the ice hall to a low temperature TES of the swimming hall. These heat fluxes are marked with red color in Figures 29 and 30. From the swimming hall, the excess heat is transferred out from the combined energy system.

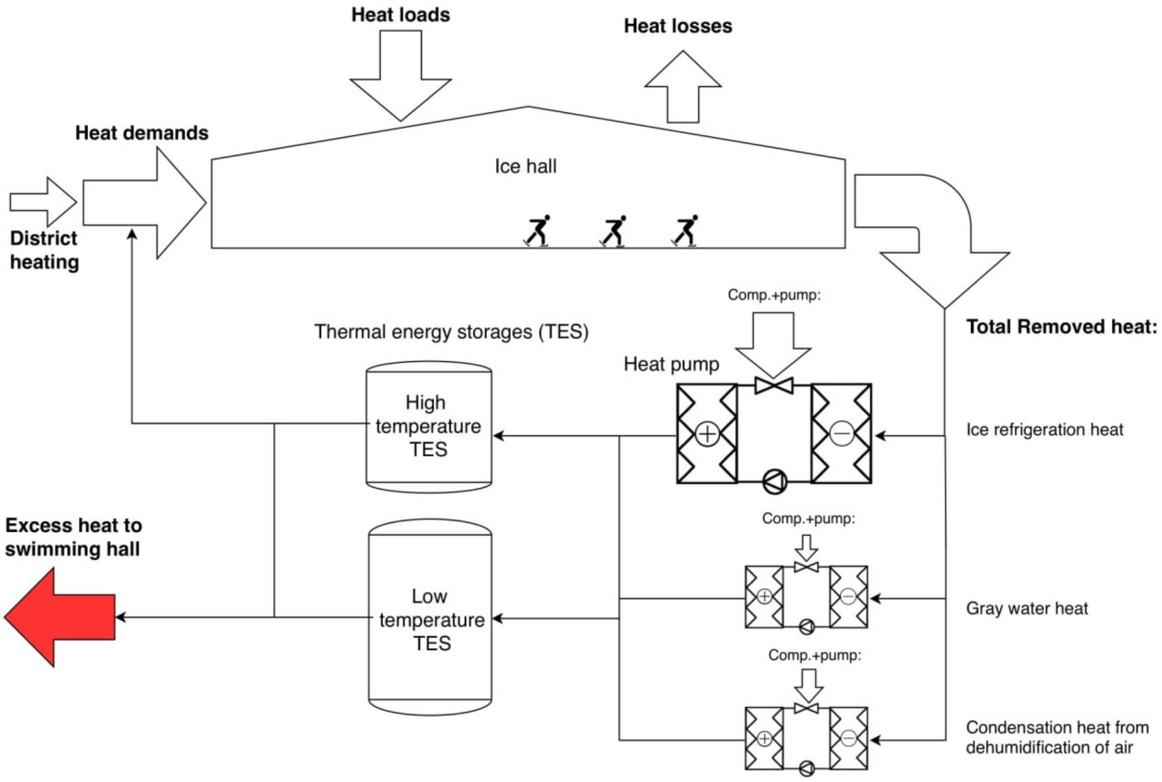

Figure 29. Heat fluxes analyzed in the ice hall.

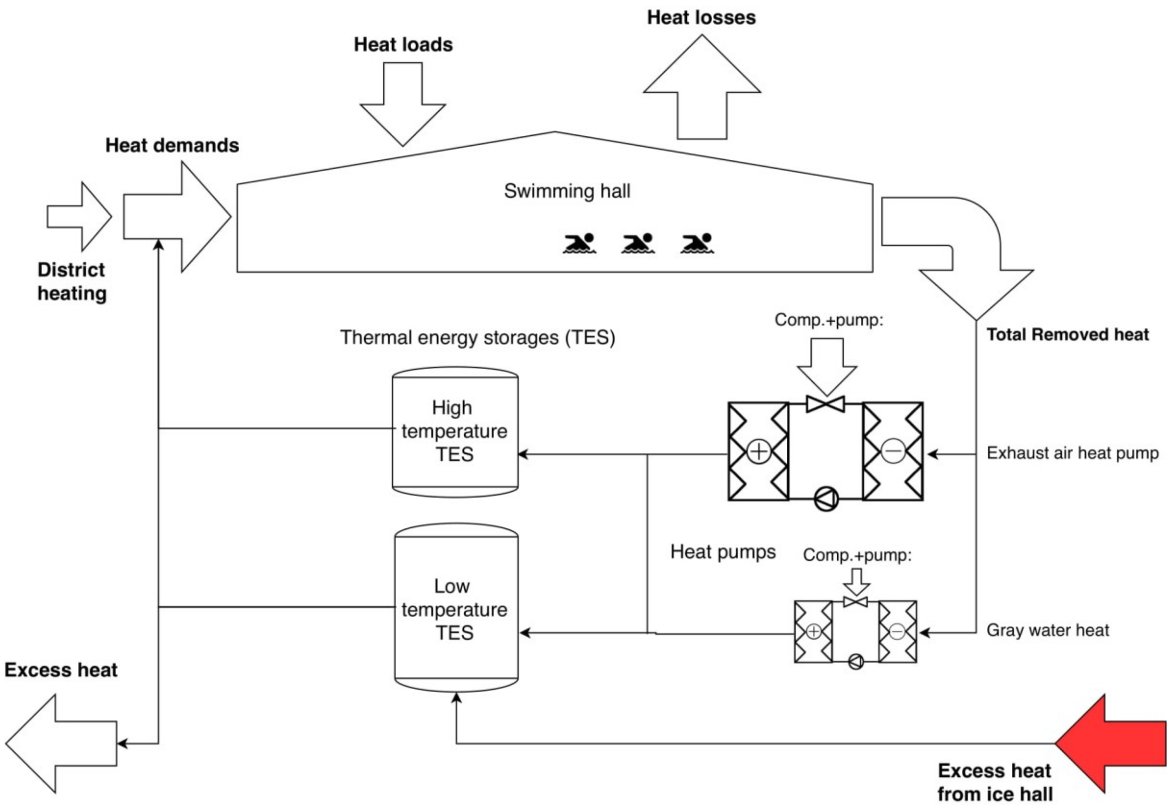

Figure 30. Heat fluxes analyzed in the swimming hall.

According to Finnish Energy (2013), the district heating network works at a high pressure of 1.6 MPa and temperature of +120 ºC. For the excess waste heat to match the state of district heat grid, a significant investment for a heat pump would be needed. Thus, excess waste heat is not sold to district heat grid in this thesis. The excess waste heat could be utilized nearby, for example to heat an outdoor turf football field or an outdoor swimming pool.

3.4.1 Short-term thermal energy storages

A short-term thermal energy storages (TES) are needed to avoid time mismatch between the available waste heat and heat demands. In this study, the post processing is conducted with an hour time step and with an assumption, that during each hour the recovered waste heat can be utilized. In practice, the mismatch between heat demands and waste heat fluxes would increase the amount of unutilized waste heat. With short-term TES, the mismatch can be eliminated and the assumption of full waste heat utilization inside a time step is valid.

The model for TES tanks used in the calculations of this thesis is a two zone model with a moving boundary. The moving boundary defines height, which divided the two zones with fixed temperatures. The schematic of the TES model is shown in Figure 31. The TES model is a tank, which is charged by heating cold water at bottom of the tank to a required temperature level. The heated water is located in the top of the tank. The boundary height of cold and heated water is calculated with energy balance including heat input and output (Dumont et al., 2016). In the two-zone model, full mixing conditions are assumed inside the zones. Ambient losses and mixing of water between the two zones are neglected.

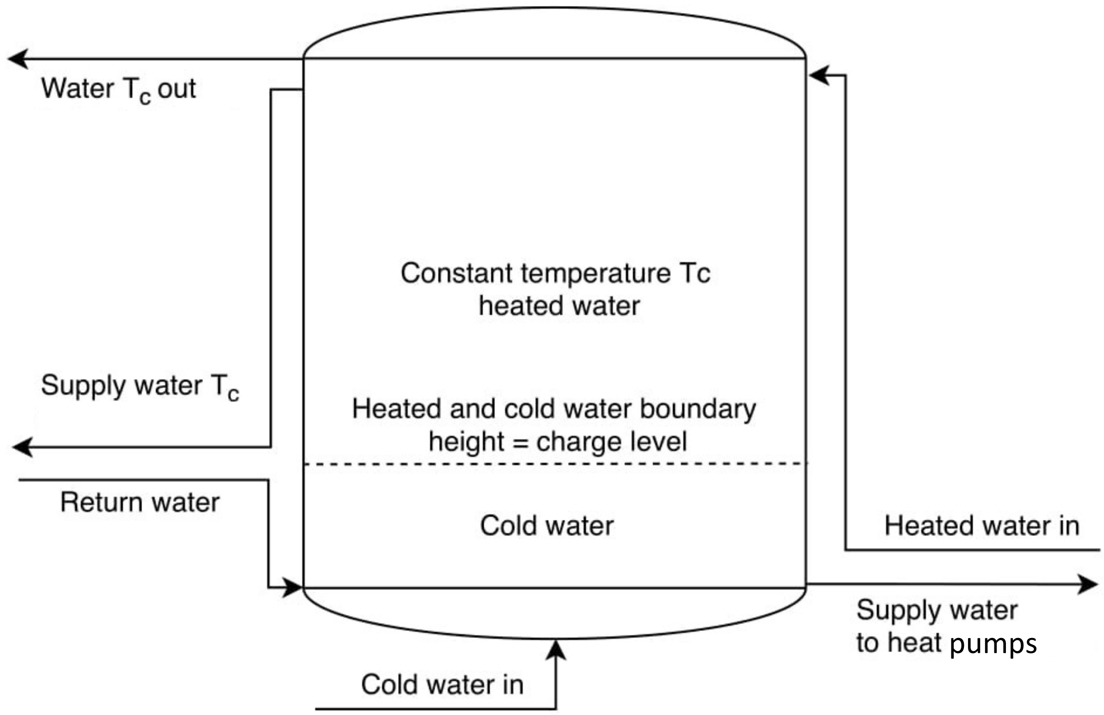

Figure 31. Schematic of a two-zone moving boundary model of a TES tank.

In Figure 31, the water flows to TES tank are depicted. The height of the boundary of the two zones follows the charge level of the tank. The tank is charged with heat from heat pumps, which recover the waste heats. The water mass flows are used straight without heat exchangers, thus supply waters to heat demands are at the same constant temperature (T_C) as the heated water. The tank is discharged by heat demands, which take heated supply water from the top of the tank and return it in the bottom of the tank. Some heat demands utilize water directly from the tank without returning the water; the removed water is replaced with cold water from the bottom of the tank.

Waste heats stored in the TES tanks are prioritized to cover heat demand of systems with high temperatures, such as DHW heating, and alternatively heat demands of systems with lower temperatures, such as underfloor heating. Heating of some of the heat demands are divided to preheating and heating. Preheating of heat demands is calculated first. Excess heat after utilization of waste heat is stored into TES tanks to be utilized during the next time step.

Each of TES tanks has a possibility for two capacities. The capacity is the same as the volume of the tank, which includes both of the zones. The smaller capacity is defined as a capacity, which discharges within 30 minutes and the bigger capacity discharges within 2 hours. The discharge times are calculated with average heat demands of the systems, which the corresponding TES tank serves. Table 17 lists the four TES tanks constant heated water temperatures, sizes and capacities for the two different discharge times.

Table 17. TES tank sizes, temperatures and capacities for two different discharge times.

		Temperature	Capacity	Size
		[°C]	[kWh]	[m^3]
Discharge time of 30 minutes	Swimming hall TES1	+34	75	2.5
	Swimming hall TES2	+55	100	4.2
	The ice hall TES1	+33	51	1.8
	The ice hall TES2	+55	39	1.5
	Total		265	9.9
Discharge time of 2 hours	Swimming hall TES1	+34	313	10.5
	Swimming hall TES2	+55	462	19.1
	The ice hall TES1	+33	204	7.1
	The ice hall TES2	+55	154	6.1
	Total		1133	42.7

3.4.2 Heat pumps for different waste heat sources

Waste heat is utilized for all heating systems of ice and swimming halls. Heat pumps are used in utilization of waste heat to increase the temperature of waste heat to the suitable level. Heat pumps consume electricity and COP of the heat pump depends on the temperature difference of heat pump evaporation and condensation temperatures. Thus, the heat pump condensation temperature has to be set to a level, which produces temperature high enough for the utilization of waste heat, but not too high to use excess electricity.

Preheating can be used if the temperature of waste heat water is too low for utilization. Preheating raises the temperature of the waste heat into some temperature between the system working temperatures. The complete heating demand is not met with preheating.

Preheating also creates requirement for additional technical appliances, which increases investment costs and space requirements.

Superheat of heat pump is acquired at higher temperatures than condensing heat and is suitable even for high temperature heat demands, such as DHW (Keinänen, 2018; Linhartová & Jelínek, 2017:3). The benefits of the higher temperature should be fully utilized. For example, superheat usage can be reduced by preheating DHW first with condensing heat and then heating it further with superheat compared to heating DHW fully with superheat. Superheat is extracted with its own heat exchanger before condenser in a heat pump (Keinänen, 2018). In this study, refrigeration heat recovery in the ice hall and exhaust air heat pump in the swimming hall are equipped with a superheat heat exchanger.

Superheat temperature depends on the used coolant, condensing temperature and pressure. According to different references superheat temperature ranges between +70…100 °C (ME, 2007:55), +80…90 °C (Laitinen & Kosonen, 1994:12), up to 110 °C (Kianta, 2018) and even up to +127 °C (Linhartová & Jelínek, 2017:3). In this thesis, superheat temperature is assumed to be at +100 °C. According to different references, the portion of the superheat varied between 10-20 % (Kosonen and Laitinen, 1994:12), 10-28 % (Linhartová and Jelínek, 2017:3) and 20 % (Keinänen, 2018; Kianta, 2018). In this thesis, the portion of 15 % is used.

Two TES tanks with different temperature set points are used for both the ice hall and the swimming halls. The two low temperature TES tanks have a set point temperature of about +34 °C and the two high temperature TES tanks have a set point temperature of +55 °C. The lower temperature level of the two TES tanks is selected to be able to cover majority of the low temperature heat demands. The higher set point temperature level of the other two TES tanks is selected to match the heat demand of DHW. Flow chart (Figure 32) illustrates the logic of storing strategy ($T_{TES,low}$ or $T_{TES,high}$) and the control strategies of the heat pump for each of the waste heat sources. The different control strategies of heat pumps are with a superheat heat exchanger, a heat pump with a high condensation temperature of +60 °C, a normal heat pump and without heat pump. The deciding conditions are shown in sharp quadrangles and the five outcomes are shown in rounded quadrangles.

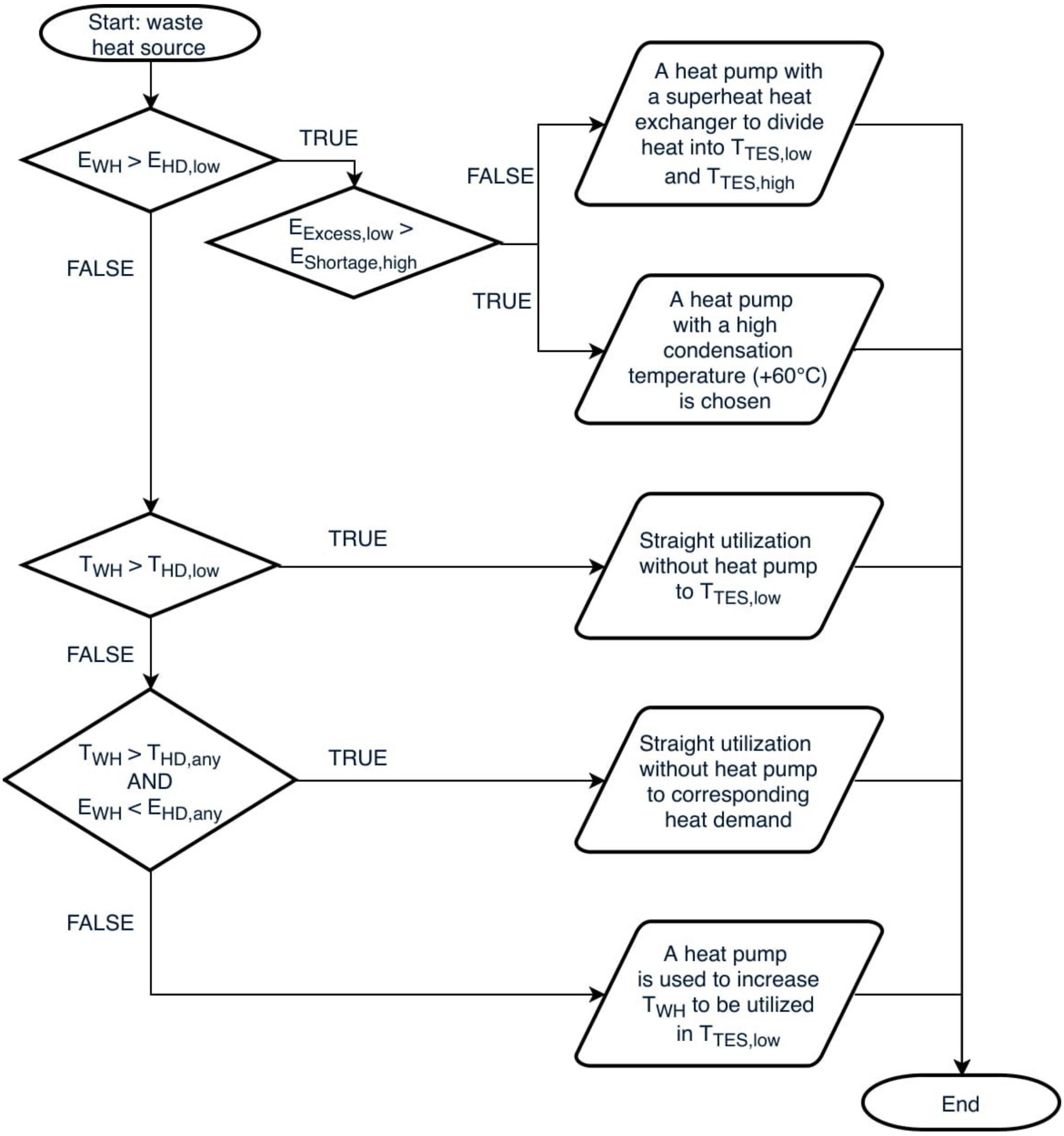

Figure 32. Logical chart of the control strategy of temperature of TES tank and of heat pump operations.

The symbols used in Figure 32 are

E_{WH}	energy of waste heat (annual)
$E_{HD,low}$	energy of low temperature heat demands (annual)
$E_{HD,any}$	energy of any heat demand (annual)
$E_{Excess,low}$	excess energy after utilization to low temperature heat demands (annual)
$E_{Shortage,high}$	shortage of energy for high temperature heat demands (annual)
T_{WH}	temperature of waste heat
$T_{HD,low}$	temperature of all low temperature heat demands = $T_{TES,low}$
$T_{HD,any}$	temperature of any (lowest) heat demand
$T_{TES,low}$	temperature of low temperature TES
$T_{TES,high}$	temperature of high temperature TES.

Coefficient of performance (COP) of heat pumps in the ice hall

COP is defined as the relation between heat amount raised from lower to higher temperature and electricity consumed. The electricity consumed of heat pumps is depending on the operation temperatures. The loss factor of heat pump f_T is defined as relation between the actual coefficient of performance COP_M and Carnot efficiency COP_C (Verley et al., 2014). The loss factor can be calculated (Eskola et al., 2012:20) as the relation

$$f_T = \frac{COP_M}{COP_C}. \tag{11}$$

Carnot efficiency of a refrigeration corresponds to a reversed Carnot cycle (Cengel & Boles, 1998:10-3). The Carnot efficiency as a function of evaporation temperature T_e and condensing temperature T_c is

$$COP_C = \frac{T_e}{T_c - T_e}. \tag{12}$$

This thesis utilizes study by Hemmilä and Laitinen, who used COP value of 2.7 for a refrigerator system of an ice rink (2018:63). Evaporation temperature is -14 °C and condensing temperature is +40 °C according to simulations done with IDA ICE, the loss factor can be calculated from equations 11 and 12 to be

$$f_T = \frac{COP_M}{\left(\frac{T_e}{T_c - T_e}\right)} = \frac{2.7}{\left(\frac{259K}{313K - 259K}\right)} = 0.56. \tag{13}$$

This loss factor is used to calculate COP of condensing heat pump and gray water heat pump, which are used in the ice hall, from equation 13 as follows

$$COP_M = f_T \cdot \frac{T_e}{T_c - T_e}. \tag{14}$$

Condensing heat recovery utilizes heat from the ice hall dehumidification, in which supply air is cooled to a temperature of 0 °C. Evaporation temperature of heat pump is 5 °C lower than the cooling coil supply temperature, which is also 5 °C lower than the cooling set point temperature. This results in an evaporation temperature of -10 °C. The heat pump of condensing heat recovery increases the temperature of waste heat to the same condensing temperature of +40 °C as in ice refrigerator condenser. The waste heats can be stored to the same TES tank when the waste heat temperatures are at least the same as the temperature of the heated TES tank. Using the loss factor f_T calculated in Equation 13 and calculating with Equation 14, the COP of condensing dehumidification is

$$COP_M = 0.56 \cdot \frac{263K}{313K - 263K} = 3.0. \tag{15}$$

Grey water heat recovery utilizes heat from hot water used in showers. In the ice hall, all DHW is utilized in grey water heat recovery. The gray water flow in the ice hall is assumed to be periodical during the opening times, since the showers are typically taken after each hockey trainings. Heat exchangers for grey water heat recovery have a wide range of efficiencies depending on temperatures, water and coolant flows. By assuming that the shower water enters the sewage pipes at a temperature of +38 °C, the water entering the heat exchanger could be 10 °C lower due to heat losses along the sewage pipes and mixed cold water. Thus, the grey water entering heat exchanger is at a temperature of +28 °C. Heat received from gray water in the ice hall could be estimated with the product data (Ecowec, 2018) to be 35 % (Appendix F). The gray water heat exchanger is combined with a heat pump. The heat pump evaporation temperature is set to 0 °C when coolant entering the heat pump is +5 °C and condensing temperature is +40 °C. Using the loss factor f_T calculated in Equation 13 and calculating with Equation 14, the coefficient of performance of gray water heat pump is

$$COP_M = 0.56 \cdot \frac{273K}{313K - 273K} = 3.9. \tag{16}$$

Coefficient of performance (COP) of heat pumps in the swimming hall

Exhaust air heat recovery (EAHP) in the swimming hall is recovered from the AHU serving the pool spaces and showers, but not from AHU supplying other spaces, since the extract air of the other spaces contain only a low amount of heat. EAHP is equipped with a superheat heat exchanger due to high amount of heat in exhaust air. In this thesis, COP of EAHP is varied, since it depends on continuously changing evaporation temperature. The loss factor of EAHP is assumed constant. According to an energy consultant Keinänen (2018), the actual coefficient of performance COP_M of modern heat pumps is 3.5 in those conditions, when exhaust air is cooled to +5 °C and condensing temperature T_c of heat pump is +40 °C. Assuming 10 °C lower evaporation temperature T_e of -5 °C, the loss factor for EAHP is calculated using Equation 13

$$f_T = \frac{COP_M}{\left(\frac{T_e}{T_c - T_e}\right)} = \frac{3.5}{\left(\frac{268K}{313K - 268K}\right)} = 0.60. \tag{17}$$

Grey water heat recovery in the swimming hall utilizes heat from hot water used in the showers and washing water of floors. These two hot water usages consist of 67 % of the total DHW usage a swimming hall (Hemmilä and Laitinen, 2018:13). In swimming hall, gray water flow is assumed to follow occupancy schedule. Due to high gray water flow during the operation hours, two heat exchangers are used in swimming hall instead of one to recover more heat. Heat received from the two heat exchangers (Ecowec, 2018) is 18 % of the heat used for the total DHW heating (Appendix F). The COP of gray water heat exchanger is 3.9 according to Equation 16.

3.5 Smart control of energy system

Smart control of energy systems in this thesis includes demand response systems controlled by dynamic energy prices of electricity and district heat, and exhaust air heat pump controlled by predicted heat demand of swimming hall. This Chapter presents the studied systems dynamic energy prices and the predicted heat demands. The last part of the Chapter presents the developed rule-based control algorithms of the systems.

3.5.1 Systems included

In ice and swimming halls, demand response can utilize heat capacities of the water of swimming pools and ice layer of rinks in addition to the thermal mass of building structures. Global temperature adjustment is a control strategy, which is used in the simulations. The global temperature adjustment adjusts temperature set points based the on global control signal. The temperature set points have to be set at ranges ensuring that thermal comfort in the swimming hall and ice quality in ice rink is acceptable.

Temperature of the two ice rinks is controlled by demand response of electricity with the peak clipping strategy. The normal and minimum set point temperatures of the ice surface are -3 °C and -6 °C (Table 18). The temperature set point for room air is always set to 9 °C warmer than the ice temperature to prevent increasing heat losses of ice rinks. In the ice hall, demand response of district heat is not included, since the ice hall is almost fully independent of district heating due to high amount of waste heat available.

In the swimming hall, temperature of the big pool and the children pool water is controlled by demand response of district heat with the load-shifting strategy. The temperature set points of the two swimming pools are minimum of 26 °C, normal of 26.5 °C and maximum of 30 °C (Table 18). The temperature set point for indoor air is controlled to be 2 °C warmer than the pool water temperature in all cases to prevent condensation. In practice, this would require a continuously measuring temperature sensor in the pool return water to set the air heating set point depending on water temperature.

The saunas in swimming hall are controlled by demand response of electricity with peak clipping strategy. The normal and minimum temperature set points for sauna air are +90 °C and +75 °C (Table 18). The normal temperature of sauna is increased from +85 °C to +90 °C to maintain average temperature and electricity consumption close to the reference case.

Cooling the swimming hall exhaust air to a constant temperature of 0 °C or +5 °C according to Hemmilä and Laitinen (2018:77) produces continuously excess heat. This happens, because the exhaust air heat pump (EAHP) produces more heat than the heat demands of the swimming hall. Thus, the optimal would be to match EAHP waste heat production to upcoming heat demand of the swimming hall. In this thesis, smart control of EAHP means adjusting the evaporation temperature based on predicted heat demands of the swimming hall. The evaporation temperature is adjusted between values of 0 °C and +30 °C (Table 18). Smart control of EAHP also enables the possibility of demand response of electricity of the EAHP. The heating of pool-space water and indoor air heating can be done partially with heat produced from the smart EAHP.

Table 18. Energy systems with smart control and temperature ranges.

System	Hall	Energy type	Adjusted temperature	Minimum temperature [°C]	Maximum temperature [°C]
Swimming pools	Swimming hall	District heat & electricity	Water	+26	+30
Ice rinks	Ice hall	Electricity	Ice	-6	-3
Saunas	Swimming hall	Electricity	Indoor air	+75	+90
Exhaust air heat pump (EAHP)	Swimming hall	Electricity	Exhaust air after EAHP	0	+30

3.5.2 Dynamic energy prices

This thesis uses hourly energy prices for both electricity and district heat. Hourly electricity price called spot price is available in Finland and is announced every day at 12:42 (CET) for the upcoming day (Nordpool, 2019a). District heat prices are currently constant during seasons for end user contracts in Finland. In this thesis, hourly district heat price is used based on the previous study (Rinne 2017).

The latest full year hourly electricity spot price from year 2017 is used in this thesis (Nordpool, 2019b). Transfer cost and electricity tax are added to spot price according to Helen (2019). Transfer contract with medium voltage power transfer is used. Base monthly transfer fee is 175 €/m and monthly maximum power fee is 3.68 €/kW calculated with maximum power of 700 kW making the base fee for transfer costs 2750 €/kk (Helen, 2019). The actual transfer fee is normally 6.30 €/MWh with an exception of 14.10 €/MWh for winter days from December to February during weekdays between times of 7 and 21. A taxation class of I is assumed with a price of 22.53 €/MWh (Helen, 2019). A value-added tax (VAT) of 24% is added to all of the mentioned prices. Also a seller marginal of 2.40 €/MWh is added to the total.

The hourly total electricity price is shown in Figure 33 for the year 2017. The average annual electricity price is 86.29 €/MWh and standard deviation is 12.78 €/MWh. Since the standard deviation is 15 % of the average price, the price is relatively stable. Excluding around 5 % of the most expensive price peaks, which stand out from the average price.

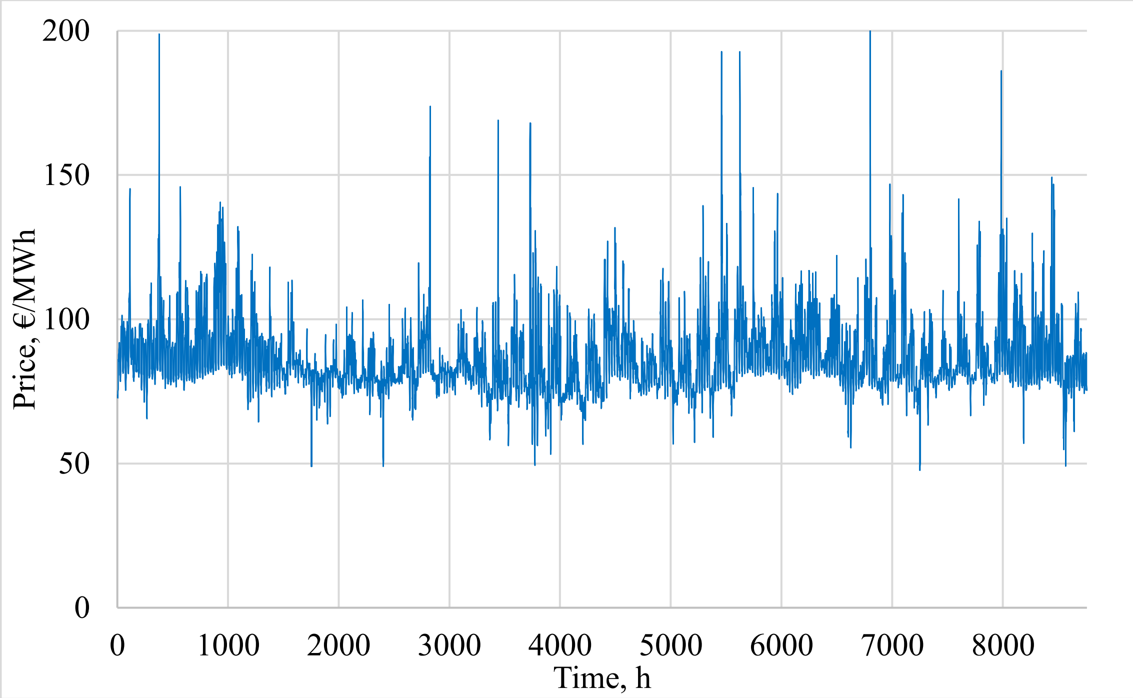

Figure 33. Hourly total electricity prices for year 2017.

Hourly district heat price for year 2012 is used (Rinne, 2017). A value-added tax (VAT) of 24% is added to the price. The price is generated based on hourly fuel price data. It is important to note, that the price may not describe accurately a possible hourly district heat price in the future. Hourly district heat price (Rinne, 2017) is compared to latest (2017-2018) seasonal district heat contract from Helen with all the fees (Finnish energy, 2019) in Figure 34.

The average annual hourly district heat price is 50.63 €/MWh and standard deviation is 23.24 €/MWh. Since the standard deviation is 46 % of the average price, the price is alternating significantly and thus gives a bigger potential for demand response than hourly electricity price. Especially around 20 % of the most expensive prices stand out from average price. All of the price peaks happen during the winter season.

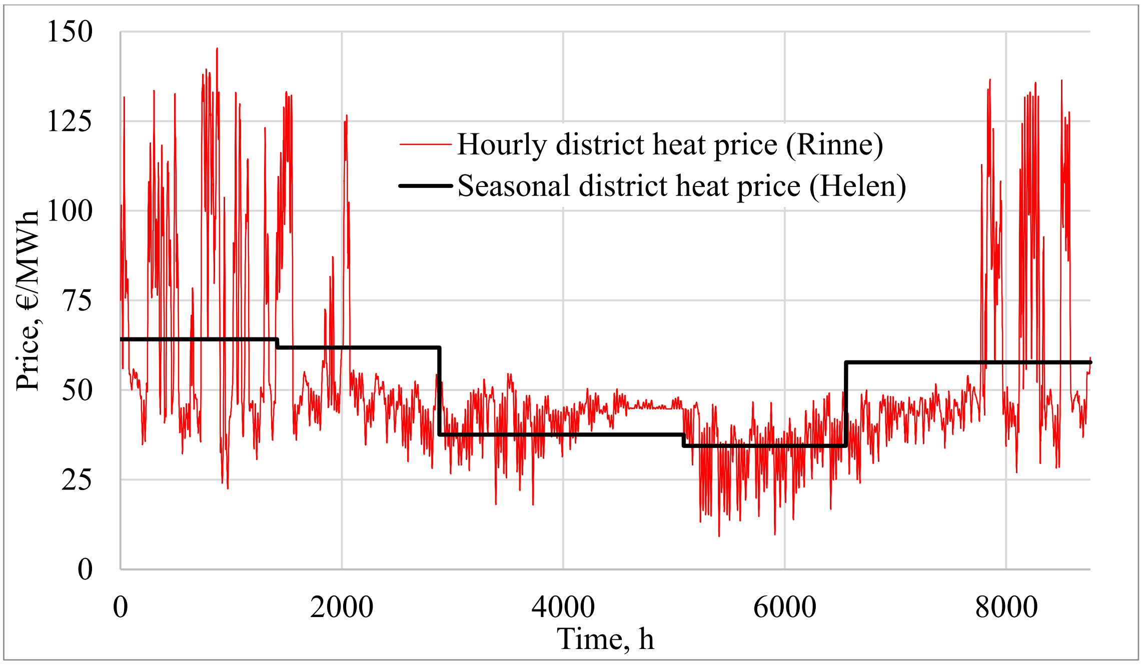

Figure 34. Hourly district heat price for year 2012 (Rinne) and seasonal district heat price for year 2017 (Helen).

3.5.3 Predicted heat demands

Smart control of EAHP in swimming hall means adjusting the evaporation temperature based on upcoming total heat demands, since waste heat is always utilized for the upcoming hour. The total heat demand of the swimming hall for upcoming hour is predicted with linear correlation using swimming hall occupancy and outdoor temperature as parameters as follows

$$E_{PHD} = 346.5 + 1.336 \cdot n_{occ} - 16.07 \cdot T_{out} \qquad (18)$$

where

E_{PHD}	predicted total heat demand of the hour [kWh]
n_{occ}	number of occupants in swimming hall
T_{out}	Outdoor temperature [°C].

Swimming hall occupancy and outdoor temperature are known upfront. Figure 35 compares IDA ICE simulated total heat demand of the Pirkkola swimming hall to predicted heat demands (Equation 18). The average annual simulated and predicted total heat demands are both 401 kW. The average error of predicted hourly heat demand against simulated heat demand is 49 kW. Thus with a relative error of 12 %, the predicted heat demands are sufficient for smart control of EAHP.

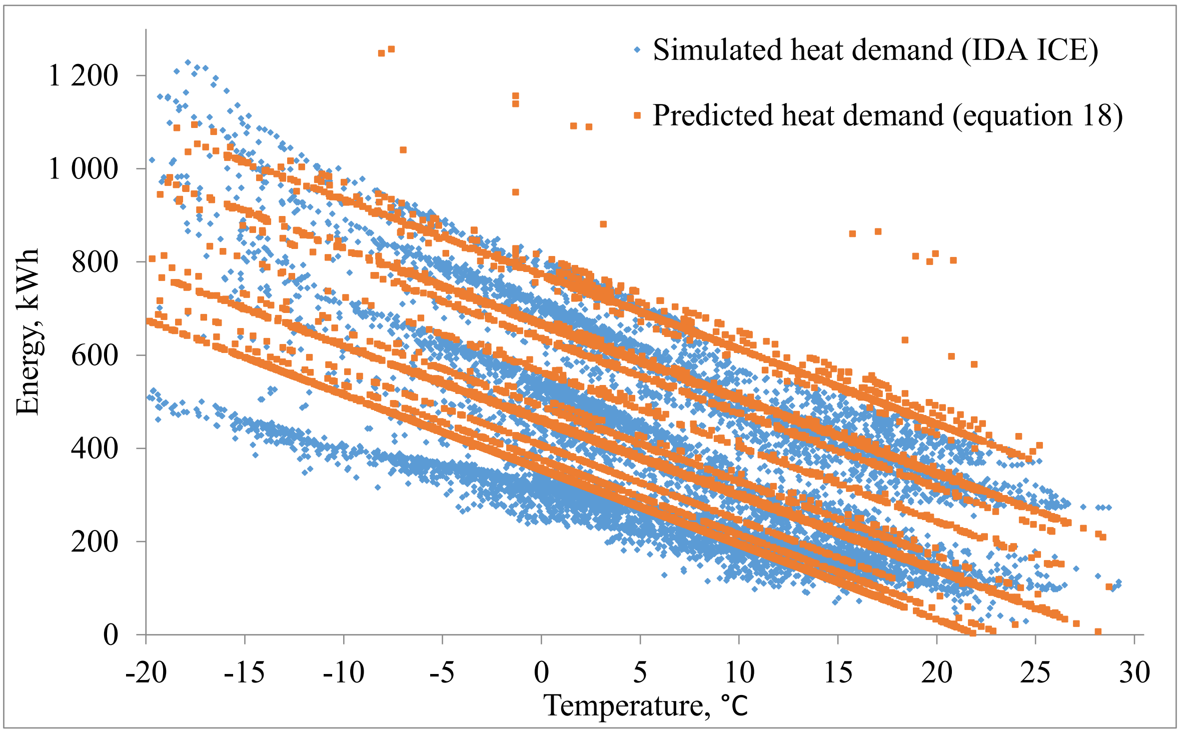

Figure 35. Simulated heat demands (blue) compared to predicted heat demands (orange).

3.5.4 Control algorithms

The used control algorithms are rule-based control algorithms, which either uses dynamic energy prices or predicted heat demand as parameters. The goal of the control algorithms is to give control signal for the energy systems, which reduces the overall energy consumption or energy cost. The control signals are set to adjust temperature set points of the energy systems, which affect either the heating or electricity power of the system.

Dynamic energy price algorithm

Demand response systems are controlled by dynamic energy prices. The system temperature set points are decreased during times of expensive energy prices and increased during times of cheap energy prices. Thus, it is important to decide the percentage of times for cheap and expensive energy prices, since the system temperature set points are changed according to these percentages. Figures 36 and 37 show the annual dynamic prices of electricity and district heat as a duration curve with dotted lines presenting the excluding percentages of 8, 20 and 30 % for both cheap and expensive energy prices. The percentages are chosen to divide the energy prices to different price classes according to the shape of the duration curves.

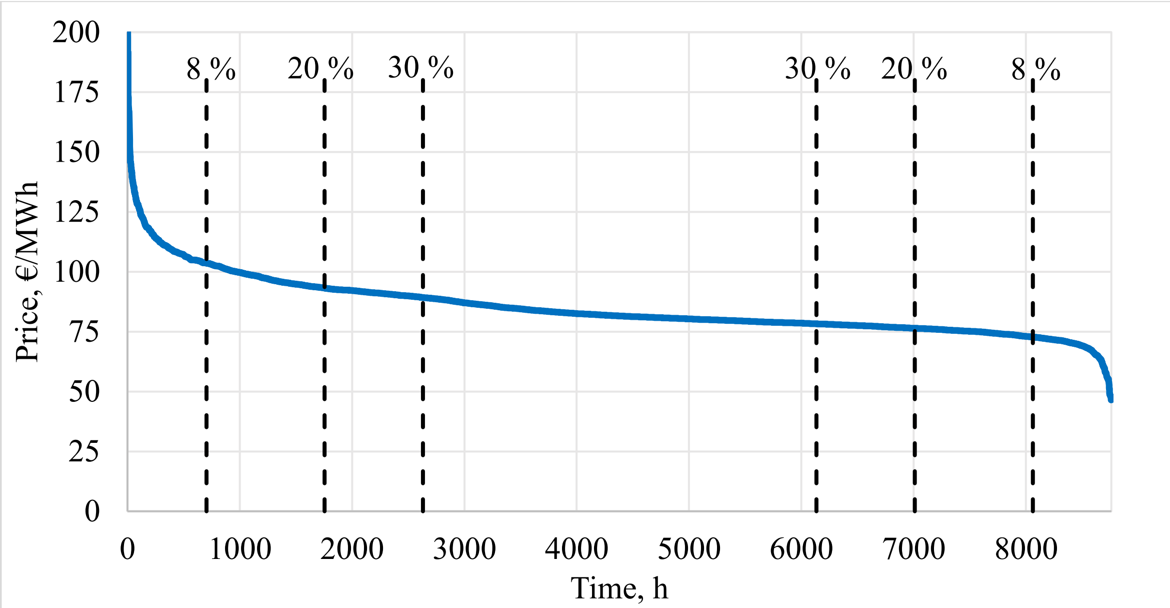

Figure 36. Duration curve for hourly total electricity prices of year 2017.

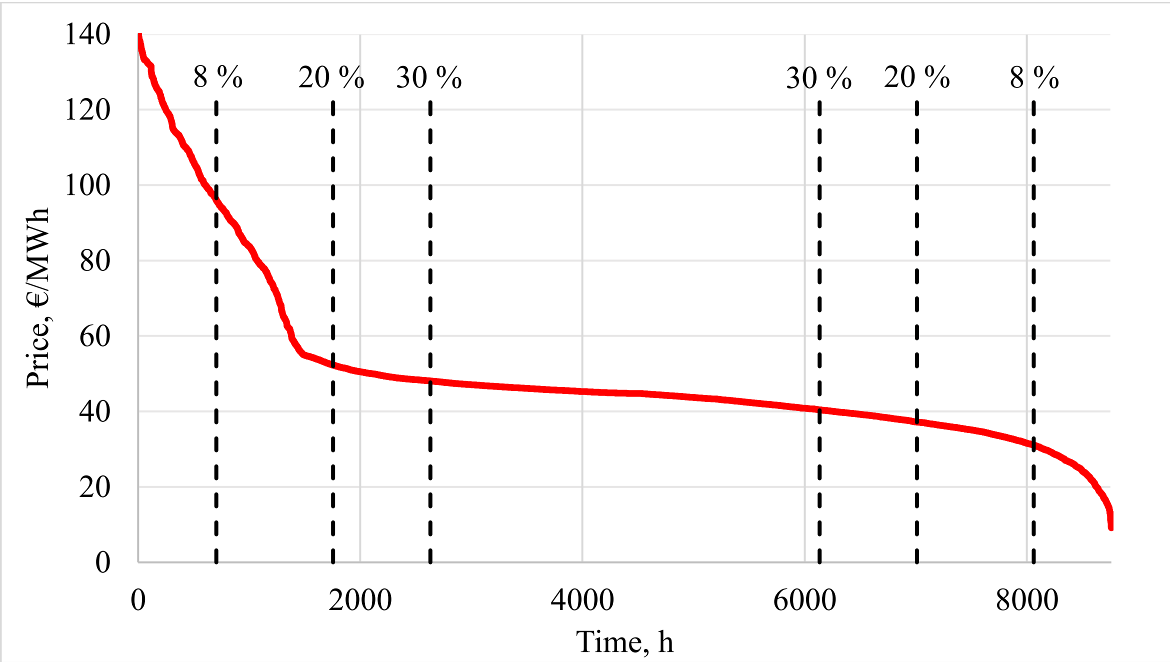

Figure 37. Duration curve for hourly district heat price of year 2012.

Algorithm developed in this thesis defines if the current energy price (CEP) is expensive, normal or cheap. The algorithm is the same for electricity and district heating. The algorithm uses as input the given percentages of 8, 20 or 30 % for the minimum amount of cheap and expensive prices, last 2 weeks of energy prices and the following 12-hour energy prices. The algorithm gives as an output a control signal of -1, 0 or +1 whether the CEP is expensive, normal or cheap respectively. The algorithm aims on classifying the corresponding minimum percentage of 8, 20 or 30 % for cheap and expensive prices. Figure 38 shows dynamic energy price algorithm as a flowchart with input data, output data and decision making.

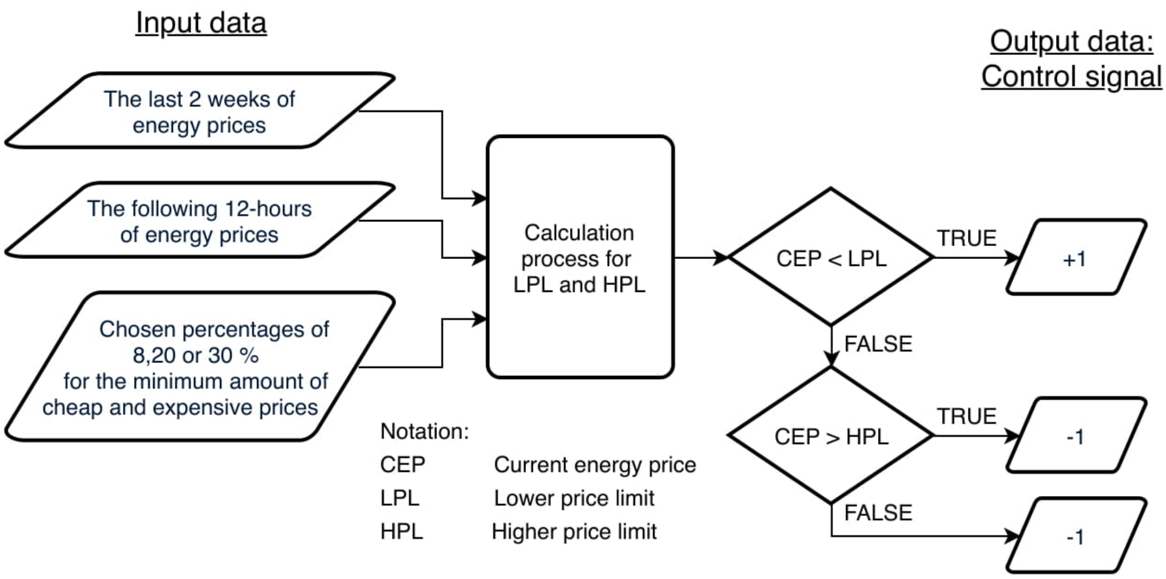

Figure 38. Dynamic energy price algorithm decision making, input data and output data.

The algorithm decides the price class for CEP by comparing it to higher price limit (HPL) and lower price limit (LPL). The calculation process for HPL and LPL based on input data is explained in detail in Appendix G. If energy price goes below LPL, the price is classified as cheap. If energy price goes over HPL, the price is classified as expensive. When energy price is between LPL and HPL, the price is classified as normal. The control signals of -1, 0 and +1 correspond to the classified expensive, normal and cheap prices respectively. Figure 39 shows an example price data for the last two weeks and the following 12-hours, CEP, current HPL and current LPL. In Figure 39, the resulting control signal would be +1.

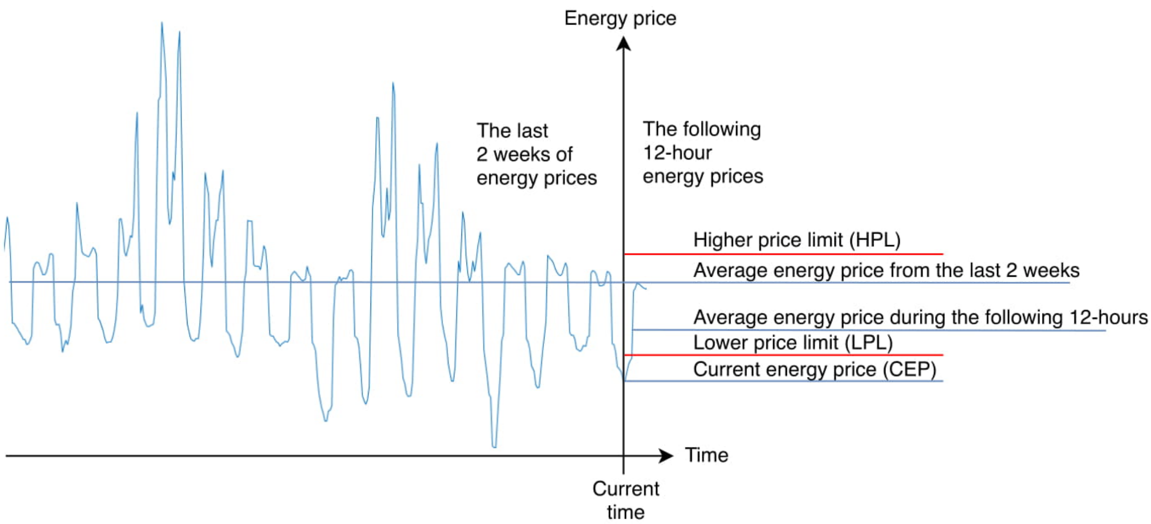

Figure 39. Example price data of the last two weeks and the following 12-hours, CEP, current HPL and current LPL.

Chosen excluding energy price percentages for the systems included in demand response are shown in Table 19. For heating of swimming pool water, 20 % is chosen as the HPL percentage, since about 20 % of the most expensive district heat prices are clearly higher than the rest of the prices (Figure 37). The LPL percentage for swimming pool water is chosen to be the same 20 %, which is pretested with IDA ICE simulation to result in a water temperature, which stays within the acceptable limits. For cooling of ice rink, 30 % is chosen as the LPL percentage, which should include most of cheap night electricity. Saunas are chosen to have HPL percentage of 30 % to limit about half of electricity prices during the day. Exhaust air heat pump (EAHP) is important part of the swimming hall energy system and is turned off only during the most expensive electricity prices (8%).

Table 19. Chosen excluding energy price percentages for the systems included in demand response.

System	Excluding minimum percentage for HPL	Excluding minimum percentage for LPL
Swimming pool water (district heat)	20 %	20 %
Ice rink ice (electricity)		30 %
Sauna air (electricity)	30 %	
Exhaust air heat pump exhaust air (electricity)	8 %	

Figure 40 shows an example period (2 last weeks in December) for hourly electricity price. HPL excludes the expensive prices while lower price limit (LPL) excludes the cheapest prices according to Table 19. The example period shows that the algorithm performs well with the defined price limits. The fast daily changes in price limits are caused by data from the following 12-hour prices. The values between which the price limits oscillate are limited by data from the last 2 weeks, which can be seen as a flat period for the price limit.

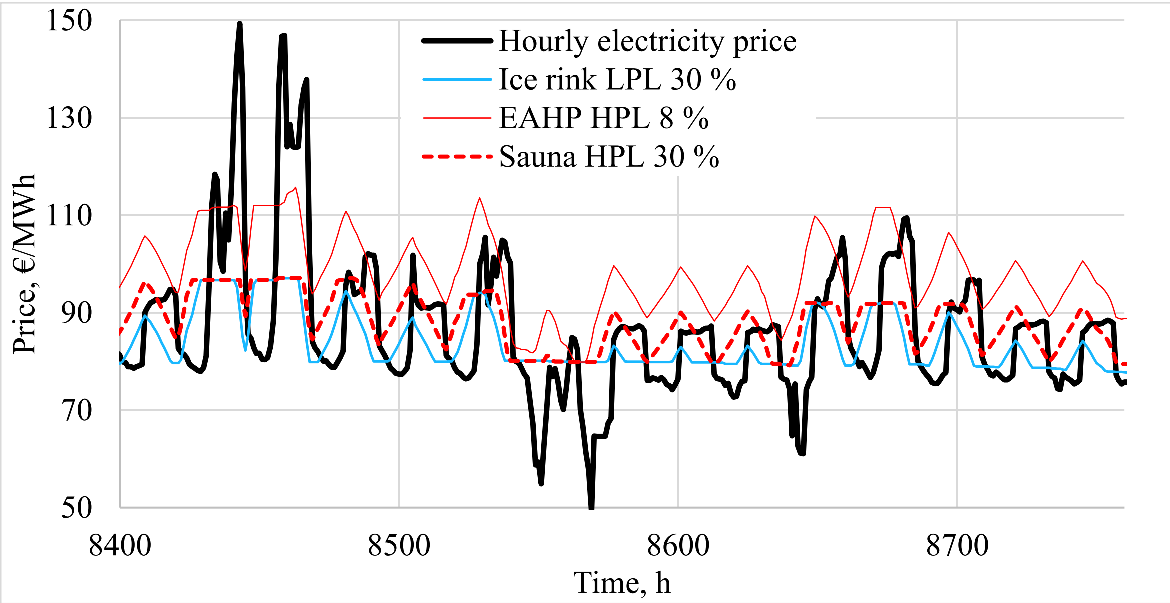

Figure 40. Hourly electricity price with price limits in December.

Figure 41 shows an example period (December) for hourly district heat price with price limits excluding energy prices according to Table 19. The algorithm performs well during big changes in the price, since it uses as parameters both the last two weeks of prices and the following 12-hour prices. During some periods, LPL and HPL are limited by each other and result in the same price. During these kind of periods, the price can only be classified as expensive or cheap.

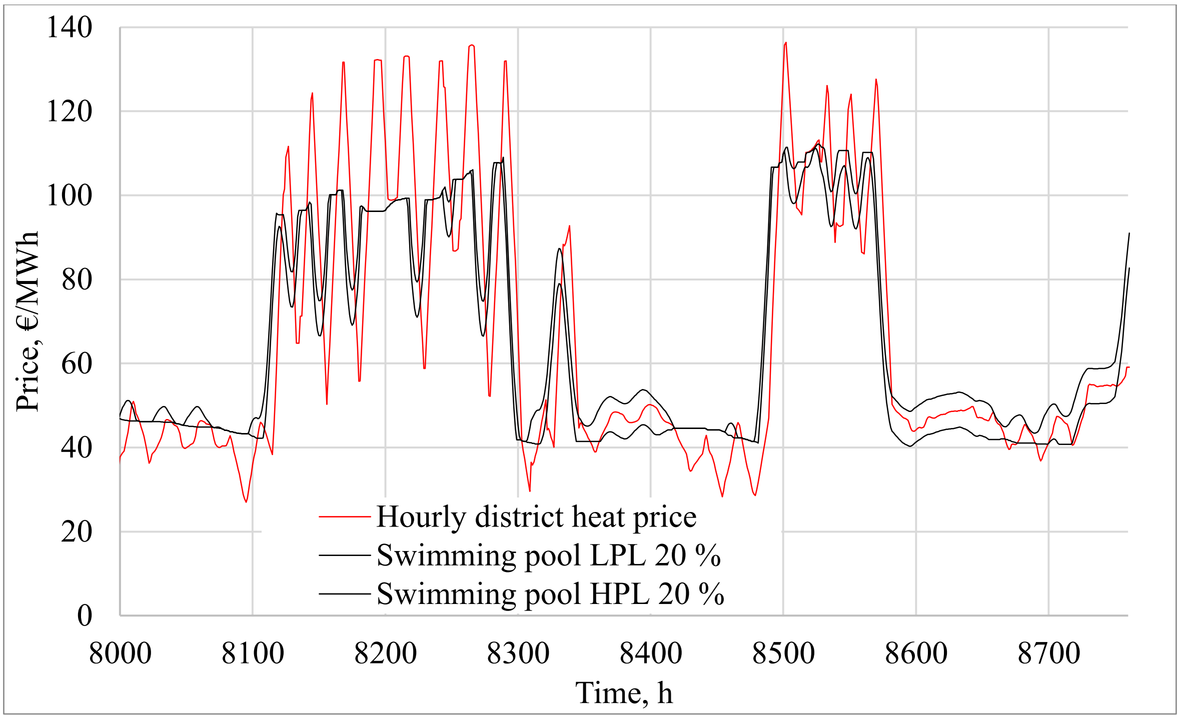

Figure 41. Hourly district heat price with price limits in December.

Smart exhaust air heat pump (EAHP) algorithm

Smart EAHP algorithm gives the set point temperature of exhaust air after the heat pump as an output. The set point temperature is controlled based on the predicted total heat demand of the upcoming hour. The acquired waste heat power from EAHP as a function of temperature set point of exhaust air is pretested with multiple simulation and shown in Figure 42. A trend line of second-degree polyline is placed according to acquired waste heats from EAHP.

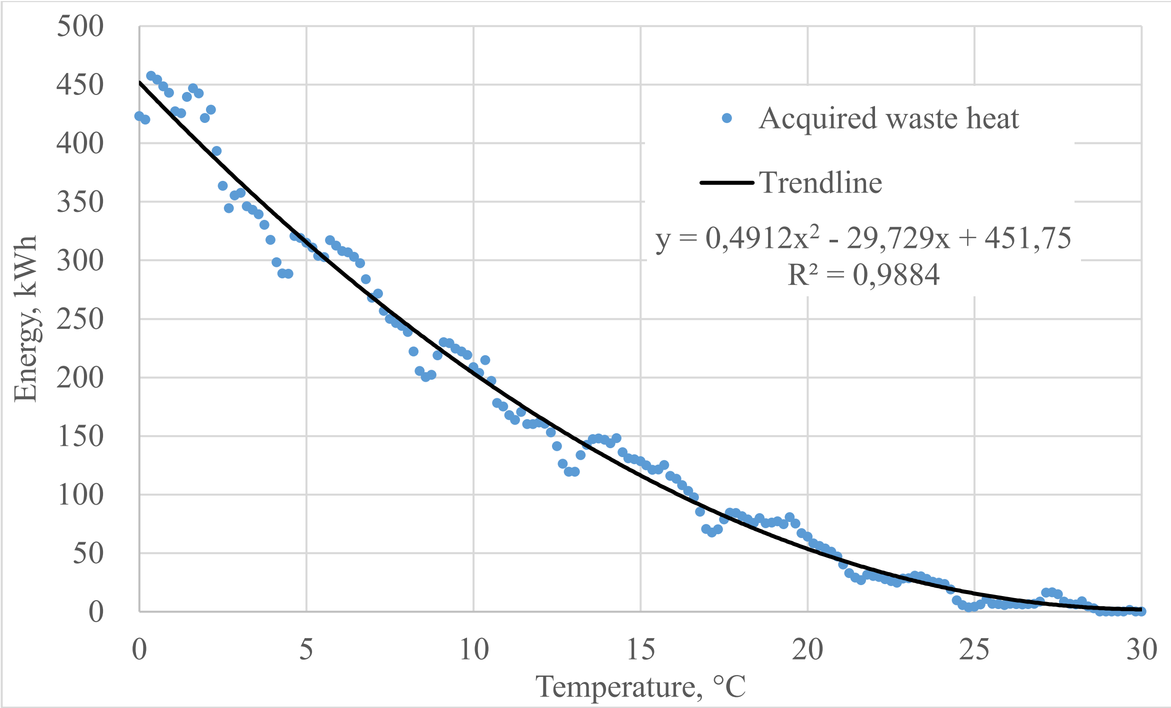

Figure 42. Available waste heat energy for one hour from EAHP as a function of set point temperature of exhaust air.

The evaporation temperature is assumed 10 °C lower than set point temperature of exhaust air. The coefficient of performance (COP) for EAHP is calculated dynamically for each hour depending on evaporation temperature according to Equation 16 presented in Chapter 3.4.2, where loss factor of exhaust air heat pump is 0.60 and condensation temperature is +60 °C.

The temperature set point of exhaust air as a function of acquired waste heat energy from EAHP, which is set to be equal with the predicted heat demand of the upcoming hour, obeys function

$$T_{ea} = 30.262 - \sqrt{2.036 \cdot E_{PHD} - 3.924} \tag{19}$$

where

E_{PHD} predicted heat demand of the hour [kWh]
T_{ea} set point temperature of exhaust air [°C].

Controlling the smart EAHP also with demand response of electricity allows demand response of electricity of pool water heating, since the pool water can be heated with heat from EAHP. Pool water heating with smart EAHP can be controlled with demand response of electricity only, or with demand response of both electricity and district heat. Control signals for combined demand response of electricity and district heat of pool water are listed in Table 20. A contradiction between electric and district heat control signals of -1 and +1 results in combined control signal of 0.

Table 20. Control signal for demand response of combined electricity and district heat for pool water heating.

Demand response of electricity control signal	-1	+1	-1	+1	0	0	-1	0	+1
District heat demand response control signal	+1	-1	0	0	-1	+1	-1	0	+1
Combined control signal	0	0	-1	+1	-1	+1	-1	0	+1

Figure 43 shows flowchart of the smart EAHP algorithm with input data, output data and calculation process. First, the set point temperature of exhaust air is controlled based on the predicted total heat demand of the upcoming hour (Equation 19). This algorithm uses control signals from dynamic energy price algorithm as input. A control signal of +1 from dynamic energy price algorithm for swimming pool increases the smart EAHP power or a control signal of -1 decreases the power. The change in EAHP power is defined as a 5 °C increase or decrease in temperature set point of exhaust air, which is tested with simulations to match the change in EAHP waste heat to the change in swimming pool heating. The smart EAHP is also controlled with peak clipping strategy of demand response of electricity. The EAHP is set to turn off during 8 % of most expensive electricity price periods. Finally, EAHP exhaust air cooling set point is limited to a minimum temperature of 0 °C.

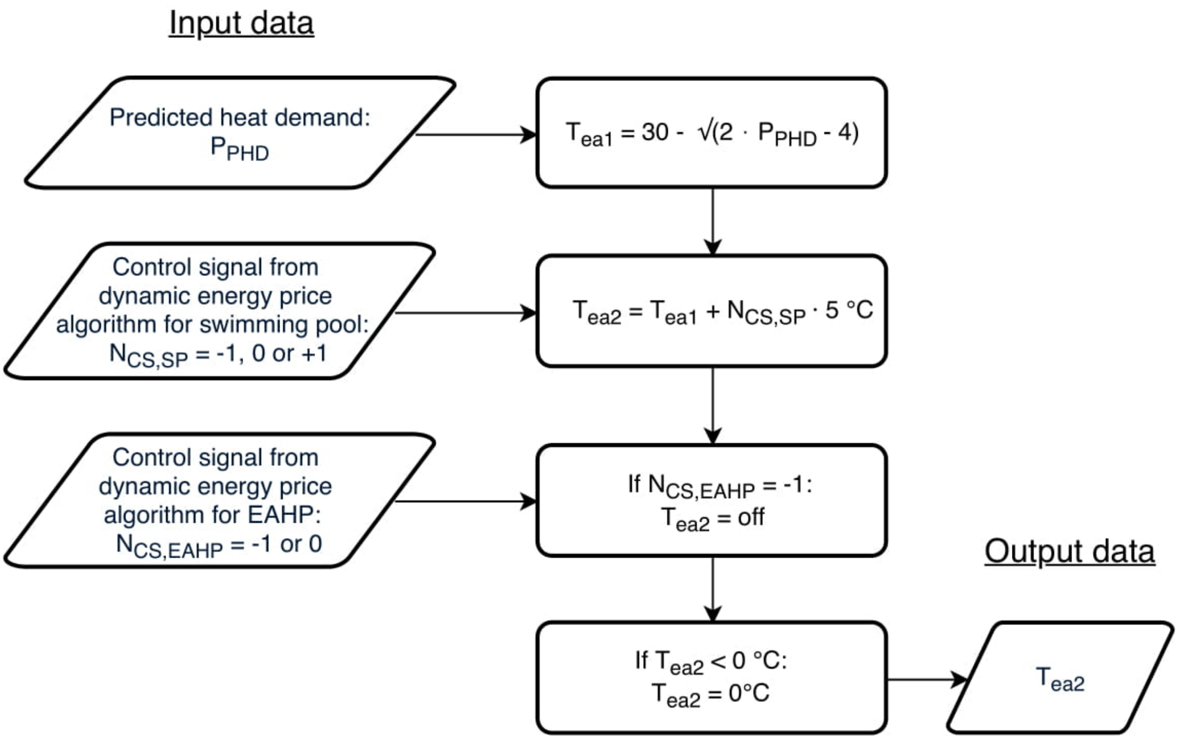

Figure 43. Smart EAHP algorithm input data, calculation process and set point temperature of exhaust air as an output data.

Figure 44 shows the EAHP cooling set point temperature of exhaust air for one year. The cooling set point temperature is lower during winter season due to bigger heat demands. The periods, in which EAHP is turned off do not show in Figure 44, because the periods are relatively short (1 to 2 hours) compared to the timescale.

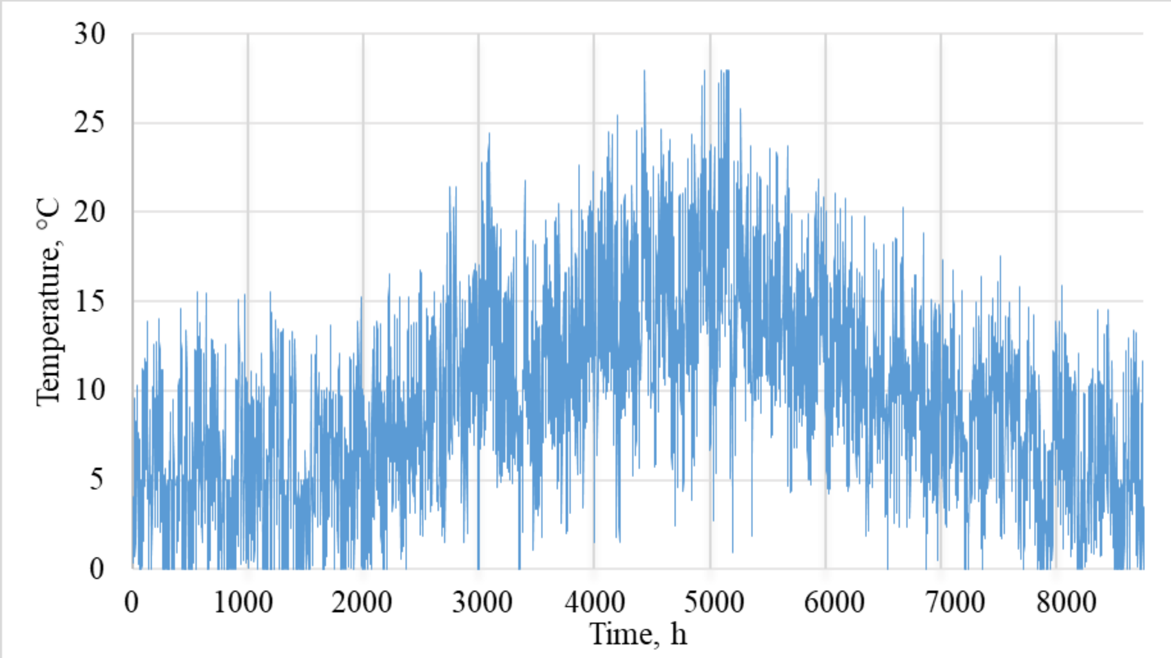

Figure 44. Smart EAHP temperature set point of exhaust air for one year.

Control of the energy systems based on demand response

Demand response control strategy used in this thesis is global temperature adjustment. In the strategy, a control signal of -1 lowers the temperature set point and control signal of +1 increases the temperature set point. The two demand response concepts used in this thesis are load-shifting and peak clipping. Using load-shifting strategy requires control signals of at least 0 and +1. The peak clipping strategy requires -1 and 0 control signals.

Table 21 shows temperature set points according to control signals for different systems and percentage the set points are active. Systems included are swimming pools, ice rinks, saunas and EAHP. Only swimming pools are controlled with demand response of district heating. All of the systems are controlled with demand response of electricity.

Table 21. Temperature set points and percentage the set points are active for different systems and demand response cases.

System and demand response case	Conservation -1	Normal 0	Loading +1
Swimming pool set point temperature (district heat)	+26 °C 32 %	+26.5 °C 38 %	+30 °C 30 %
Swimming pool set point temperature (electricity)	+26 °C 27 %	+26.5 °C 43 %	+30 °C 30 %
Swimming pool set point temperature (district heat and electricity)	+26 °C 36 %	+26.5 °C 35 %	+30 °C 29 %
Ice rink set point temperature (electricity)		-3 °C 59 %	-6 °C 41 %
Sauna set point temperature (electricity)	+75 °C 38 %	+90 °C 62 %	
Smart exhaust air heat pump (EAHP) (electricity)	off 12 %	0…+30 °C 88 %	

Figure 45 shows an example period (December) for swimming pool set point temperature controlled with demand response of district heating. The length of even temperature set point periods differ from only few hours to multiple days. This is caused by the unpredictable nature of the dynamic district heating price. The swimming pool has a significant heat capacity and requires long time to be heated from lower to higher temperature set point. Thus, the short periods of higher temperature increase the actual swimming pool temperature only by little.

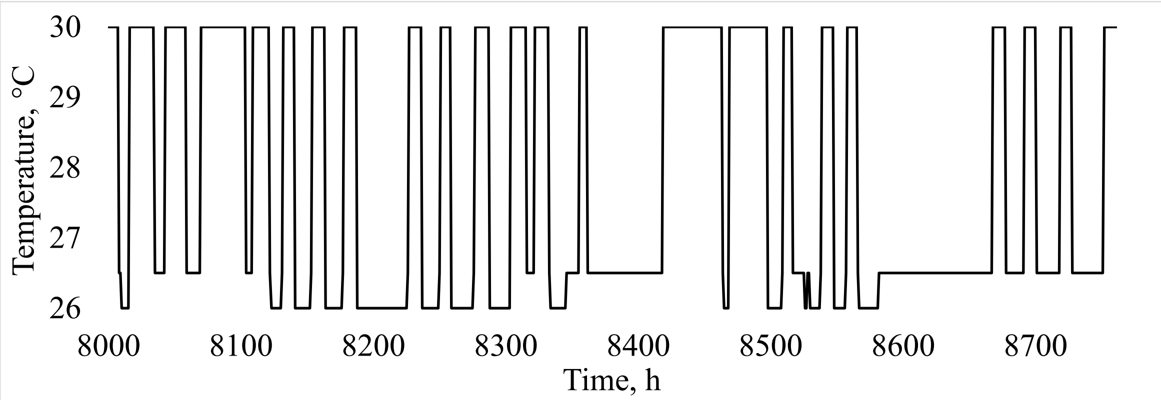

Figure 45. Pool water temperature set point during December.

Figure 46 shows an example period (2 last weeks of December) for temperature set point of ice rink. The lower temperature of -6 °C is shown on top of the chart as it corresponds to control signal +1. The relatively even alternation between low and normal temperature set point is caused by electricity being cheaper during nights than days. Longer periods of normal temperature set point (-3 °C) happen when electricity is classified as expensive even during nights. Longer periods of loading with lowered temperature set point (-6 °C) happen when electricity is classified as cheap even during days. The sauna temperature set point is similar to ice rink with only two temperature set points. However, saunas are controlled with conservation of electricity during times of expensive electricity instead of loading during times of cheap electricity.

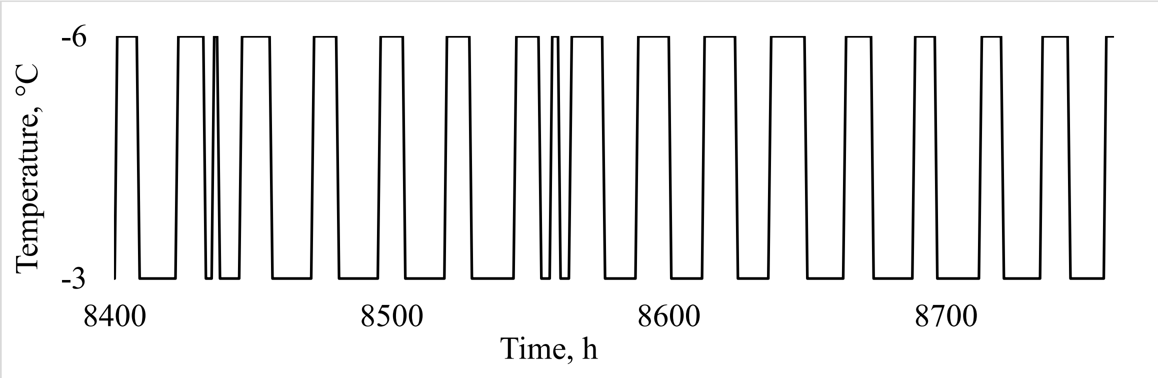

Figure 46. Ice rink temperature set point during 2 last weeks of December.

The heating and cooling systems in the simulation model use PI-controllers to match the temperature to the given heating set point. The proportional part of the controller (P) adjusts the load more if temperature set point is more off from the measured value, and integral part of the controller (I) continuously adjusts the load to reduce the error without overcompensating the difference between measured temperature and temperature set point. Other important parameters for global temperature adjustment are design powers. Design powers of the heating and cooling systems affect the rate, at which the system is able to approach the temperature set point during loading.

3.6 Cost investment analysis

The cost investment analysis in this thesis calculates the maximum costs of profitable investments, which equals the total energy cost savings achieved with the investment. The calculation takes into account the energy cost savings for each case, inflation of energy prices, nominal interest, and the assumed minimum lifecycle of the implemented system, called repayment period. The cost investment analysis is done for the ice hall only and for the combined energy system of ice and swimming halls.

The general increase in expenses is called inflation, which lowers the purchasing power of money. Inflation of energy prices focuses specifically to energy price. Nominal interest rate expresses the change in value of money in terms of time. The effect of inflation of energy prices and nominal interest rate can be combined to real interest rate of energy price as follows (Sirén, 2015:20)

$$r_e = \frac{i - f_e}{1 + f_e} \tag{20}$$

where

r_e	annual real interest rate of energy price [%]
i	annual nominal interest rate [%]
f_e	annual inflation of energy prices [%].

This thesis assumes the value for escalation of energy price of 3 %, which corresponds to nominal interest of 5.1 % and inflation of energy prices of 2 % (Equation 20). Three repayment periods of 7, 10 and 15 years are used in this thesis. Discount yield calculates the return on investment and is calculated as follows (Sirén, 2015:21)

$$a_n'' = \frac{1 - (1 + r_e)^{-n}}{r_e} \tag{21}$$

where

a_n''	total discount yield [a]
r_e	annual real interest rate of energy price [%]
n	repayment period [a].

The maximum cost of profitable investments equals the total energy cost savings, which is calculated by multiplying the total discount yield by the annual energy cost savings as follows

$$S_{inv} = a_n'' \cdot S_{E,a} \tag{22}$$

where

S_{inv}	the maximum cost of profitable investments [€]
a_n''	total discount yield [a]
$S_{E,a}$	annual energy cost savings [€/a].

4 Results

The results for waste heat utilization and smart control of energy system for the Pirkkola ice and swimming halls are presented in this Chapter. The results include breakdown of annual energy fluxes for 13 analyzed cases for the Pirkkola energy systems. Most relevant annual energy fluxes are presented as dynamic graphs and duration curves for corresponding energy power. The energy prices are always dynamic.

This Chapter first explains the analyzed cases and presents energy balances of the ice and swimming halls. The results are presented as 5 different energy balances. The reference cases are presented first, after that utilization of waste heat is analyzed as the first measure and smart control of energy system as the second measure. Finally increasing thermal energy storage sizes is analyzed. Summary of annual energies and cost investment analysis compare the analyzed cases collectively.

4.1 Analyzed cases

Altogether 13 different cases are analyzed. The analyzed cases are chosen to clarify individual effects of different measures for the energy consumption and to find out the effect of different combinations of those measures. The parameters and the measures chosen as variables, which define the cases, are the summer brake schedule of Pirkkola (with and without), waste heat recovery (WHR) usage, demand response (DR) usage and thermal energy storage (TES) discharge time. Figure 47 shows the 13 analyzed cases as paths defined by previous four variables. The variables are shown on top of the columns, the case paths are numbered and the reference case path is marked with yellow color (Figure 47). Case numbers 8, 9 and 10 are used twice for different TES discharge times. The variable values for the reference case are with summer break, no waste heat recovery, no demand response and TES discharge time of 30 minutes.

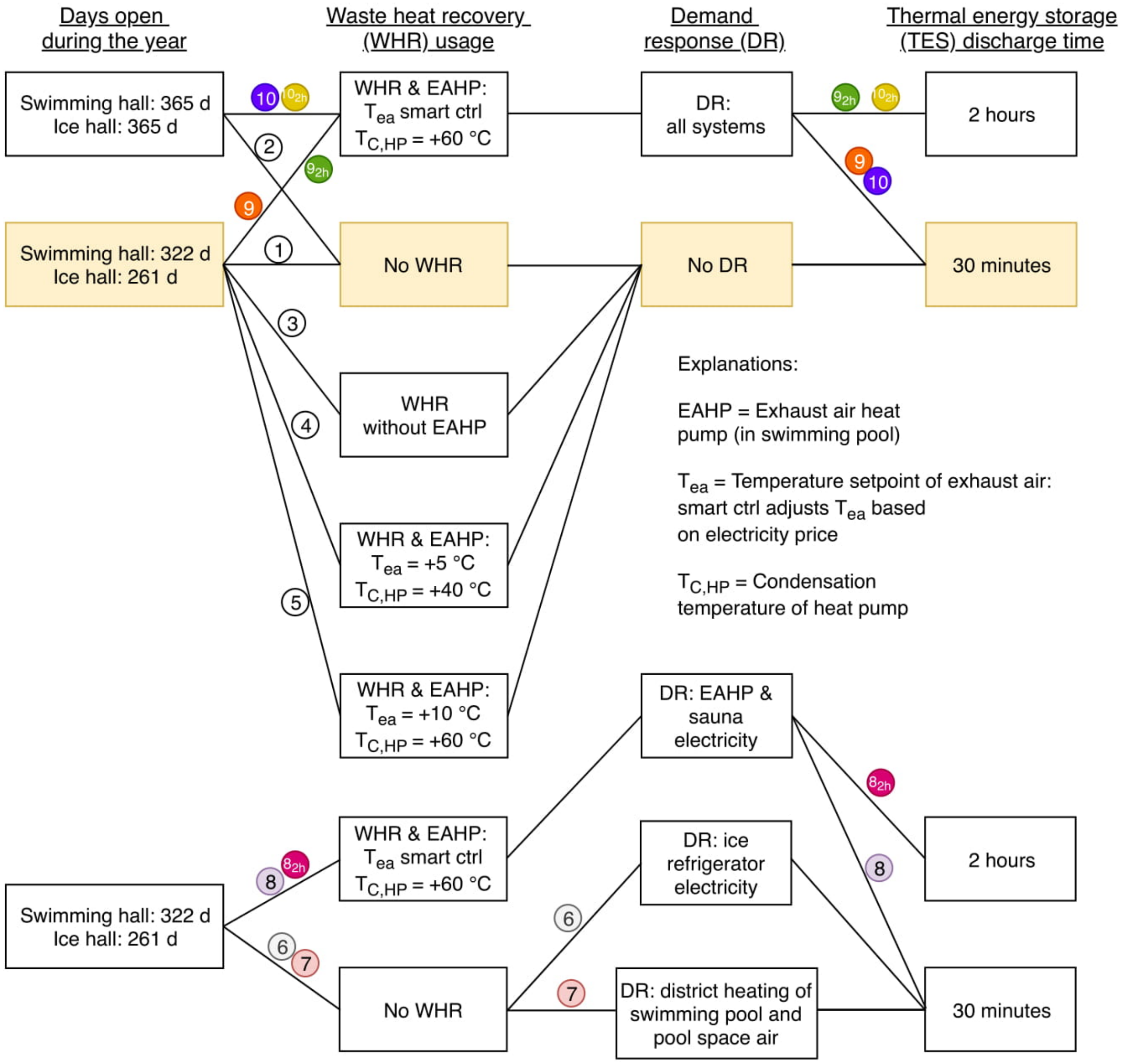

Figure 47. Analyzed cases as paths.

Figure 47 is separated into two brackets. The top bracket separates the different waste heat recovery (WHR) cases and the bottom bracket separates cases with demand response (DR) of only one system per case. WHR means all the measures of waste heat recovery including Exhaust air heat pump (EAHP) are used.

The summer breaks are held for the swimming pool from 1.6 to 13.7 and for the ice rink from 1.5 to 12.8. Cases with different EAHP temperature set point for exhaust air cooling and EAHP condensation temperature are included. In Figure 47, EAHP temperature T_{ea} means the temperature set point for cooling of exhaust air, which can be +5 °C, +10 °C or smart control. Smart control means that the temperature of exhaust air is controlled based on predicted heat demands. $T_{c,HP}$ for EAHP means the chosen condensation temperature in the heat pump, which is either +40 °C or +60 °C. One case with WHR but without EAHP is also analyzed. DR consists of controlling electricity in swimming pools, saunas and ice rinks, and controlling district heat in swimming pools. TES discharge time is defined as the duration time of TES from full to empty for average heat demands.

A breakdown of the analyzed cases is shown in Table 22. The cases, which do not have utilization of waste heat, are divided to cases, which separate the ice hall (IH) and the swimming hall (SH).

Table 22. Breakdown of the analyzed cases.

Case number	Presenting Chapter	IH	SH	Summer break during the year	Utiliza-tion of waste heat	EAHP and temperatures for exhaust air cooling / condensation	Demand response	Discharge time of thermal energy storage
1_{IH}	4.3	X	-	Yes	-	-	-	30 minutes
1_{SH}	4.3	-	X	Yes	-	-		30 minutes
2_{IH}	4.3	X	-	-	-	-	-	30 minutes
2_{SH}	4.3	-	X	-	-	-		30 minutes
3	4.4	X	X	Yes	Yes	-	-	30 minutes
4	4.4	X	X	Yes	Yes	5 °C / 40 °C	-	30 minutes
5	4.4	X	X	Yes	Yes	10 °C / 60 °C	-	30 minutes
6_{IH}	4.5	X	-	Yes	-	-	Only IR EL	30 minutes
7_{SH}	4.5	-	X	Yes	-	-	Only SH DH	30 minutes
8	4.5	X	X	Yes	Yes	SC / 60 °C	Only SH EL	30 minutes
9	4.5	X	X	Yes	Yes	SC / 60 °C	All systems	30 minutes
10	4.5	X	X	-	Yes	SC / 60 °C	All systems	30 minutes
8_{2h}	4.6	X	X	Yes	Yes	SC / 60 °C	Only SH EL	2 hours
9_{2h}	4.6	X	X	Yes	Yes	SC / 60 °C	All systems	2 hours
10_{2h}	4.6	X	X	-	Yes	SC / 60 °C	All systems	2 hours
Notation: EAHP = Exhaust Air Heat Pump, SC = Smart Control, IR = ice refrigerator, IH = ice hall, SH = Swimming Hall, EL = Electricity, DH = District Heat								

4.2 Energy balances of the ice and swimming halls

The results of this thesis are presented as annual energy fluxes in energy balances. Connection scheme for the ice and swimming hall energy system was designed in this thesis. The energy balances and annual energy fluxes are presented for that designed connection scheme. Energy balances for separately both the ice and swimming halls are defined for building level and for processing of waste heat of the combined energy system. The corresponding annual energy fluxes for connection schemes are presented in table format. The tables in this chapter presenting the annual energy fluxes for the 13 cases are separated for the ice hall, the swimming hall and the combined ice and swimming halls.

Figure 48 shows connection scheme for the energy system of the ice hall with energy balance of the building (EB1), energy balance of processing of waste heat (EB2) and energy flux indices. Energy balance boundaries are marked as red dashed lines where energy fluxes out of the set boundaries are shown with arrows. Cold water enters the system into the low temperature TES. Each waste heat source has a separate heat pump. The sizes of energy flux arrows, TES tanks and heat pumps indicate the order of the magnitude of the flux. Energy fluxes are also transferred as cycling water or coolant streams between the sub-systems inside energy balance 2. The shortage of heat demand is covered with purchased district heat. Excess heat flux to the swimming hall is shown in Figure 48. All different heat loads of the ice hall are marked with indices with prefix of capital I.

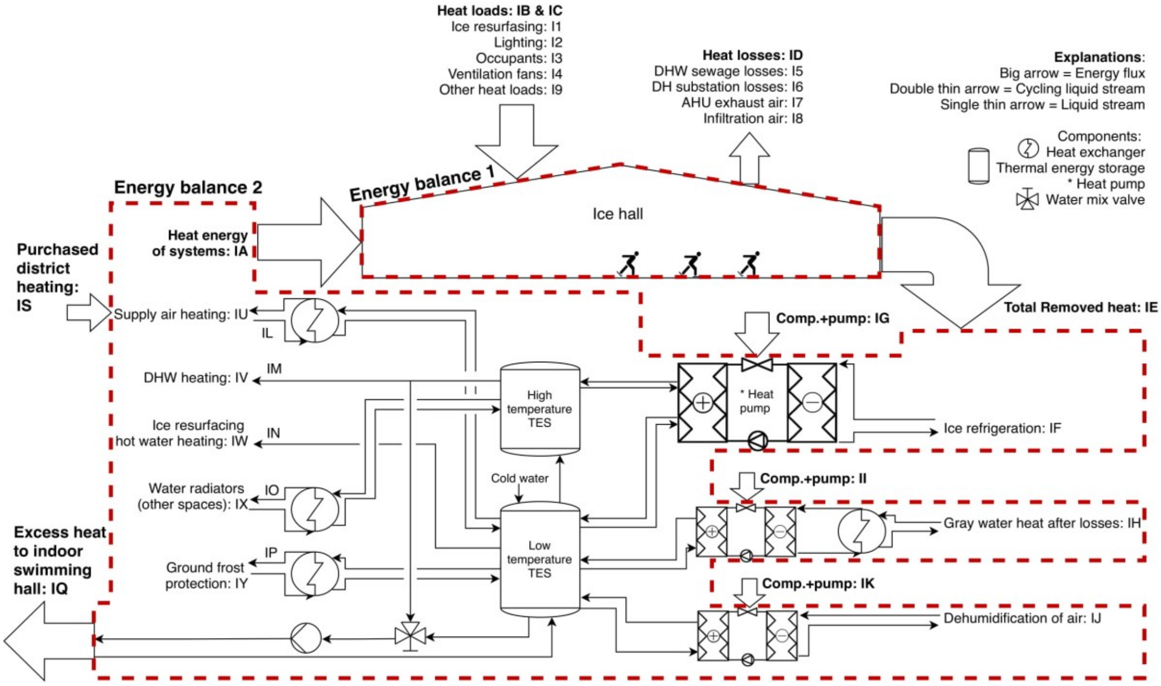

Figure 48. Connection scheme for energy system of the ice hall with energy balance of building (EB1), energy balance of processing of waste heat (EB2) and energy flux indices.

The energy fluxes of the ice hall marked with indices in Figure 48 are presented in Table 23. Table 23 divides energy fluxes to sections of the two energy balances, total energies, breakdown of heat energy of systems and breakdown of heat loads and losses. Utilized waste heat and purchased electricity do not have indices, as they are combination of multiple energy fluxes.

Table 23. Energy fluxes of the ice hall with indices corresponding to Figure 48.

Annual energy [MWh/a]. Ice hall floor area 6674 m²		
Energy balance 1: Building	Heat energy of systems	IA
	Heat loads from electricity	IB
	Other heat loads	IC
	Heat losses	ID
	Total removed heat for use	IE
Energy balance 2: Waste heat	Ice refrigeration heat	IF
	Ice refrigeration HP electricity	IG
	Gray water heat	IH
	Gray water HP electricity	II
	Dehumidification heat	IJ
	Dehumidification electricity	IK
	Supply air heating	IL
	DHW heating	IM
	Ice resurfacing water	IN
	Water radiator heating	IO
	Ground frost protection	IP
	Excess heat to the swimming hall	IQ
Notation: HP = Heat Pump		

Total energies	Utilized waste heat Purchased district heat Purchased electricity	IS
Breakdown of heat energy of systems	Supply air heating	IU
	DHW heating	IV
	Ice resurfacing water freezing	IW
	Water radiator heating	IX
	Ground frost protection	IY
Breakdown of heat loads (+) and losses (-)	Ice resurfacing water freezing	I1
	Lighting	I2
	Occupants	I3
	Ventilation fans	I4
	DHW sewage losses	I5
	DH substation losses	I6
	Infiltration air	I7
	Envelope	I8
	Other heat loads(+) / losses(-)	I9

Figure 49 shows the connection scheme for the energy system of the swimming hall with energy balance of the building (EB3), energy balance of processing of waste heat (EB4) and energy flux indices.

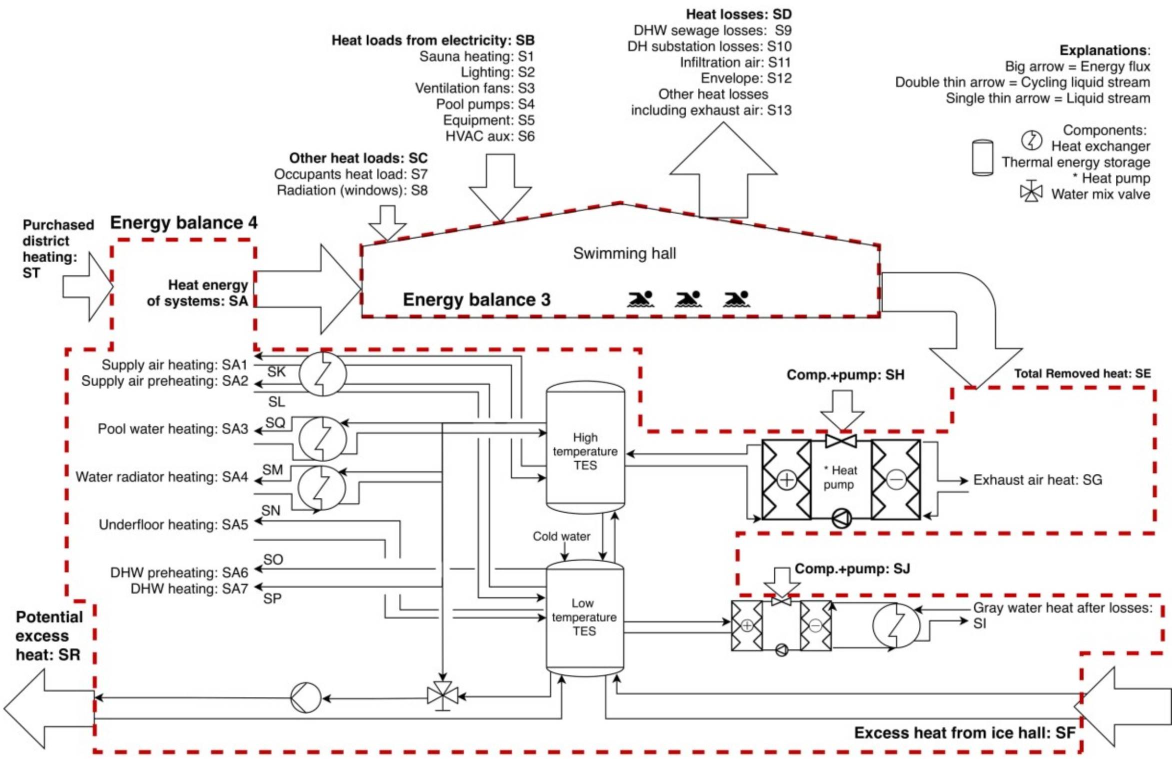

Figure 49. Connection scheme for energy system of the swimming hall with energy balance of the building (EB3), energy balance of processing of waste heat (EB4) and energy flux indices.

The energy balances, energy fluxes and components are shown in Figure 49 similarly to the connection scheme of the ice hall. Excess heat flux from the ice hall and excess heat flux out from swimming hall are shown in Figure 49. The relevant energy fluxes of the swimming hall are marked with indices with prefix of capital S. The indices are presented in Table 24. Table 24 divides energy fluxes to similar sections than in the ice hall with an addition of breakdown of waste heat, which separates the amounts utilized from the ice hall and the swimming hall. Energies in Table 24, which do not have an index, are a combination of multiple energy fluxes.

Table 24. Energy fluxes of the swimming hall with indices corresponding to Figure 49.

Annual energy [MWh/a]. The swimming hall floor area 7982 m²

Energy balance 3: Building building	Heat energy of systems	SA
	Heat loads from electricity	SB
	Other heat loads	SC
	Heat losses	SD
	Total removed heat for use	SE
Energy balance 4: Waste heat	Heat from the ice hall	SF
	Exhaust air HP heat	SG
	Exhaust air electricity	SH
	Gray water heat	SI
	Gray water HP electricity	SJ
	Supply air preheating	SK
	Supply air heating	SL
	Water radiator heating	SM
	Underfloor heating	SN
	DHW heating	SO
	DHW preheating	SP
	Pool water heating	SQ
	Total excess heat	SR
Total values	Utilized waste heat	SS
	Purchased district heat	ST
	Purchased electricity	SU

Breakdown of waste heat	Utilized the ice hall waste heat	SV
	Utilized SH waste heat	SW
	Excess heat from TES 1	SX
	Excess heat from TES 2	SY
Breakdown of heat energy of systems	Supply air heating	SA1
	Supply air preheating	SA2
	Pool water heating	SA3
	Water radiator heating	SA4
	Underfloor heating	SA5
	DHW preheating	SA6
	DHW heating	SA7
Breakdown of heat loads (+) and losses (-)	Sauna	S1
	Lighting	S2
	Ventilation fans	S3
	Pool pumps	S4
	Equipment	S5
	HVAC aux.	S6
	Occupants	S7
	Radiation through windows	S8
	DHW sewage losses	S9
	DH substation losses	S10
	Infiltration air	S11
	Envelope	S12
	Other heat loads(+) / losses(-)	S13

Notation: SH = swimming hall, HP = Heat Pump

Figure 50 shows simplified scheme of the combined energy system of the ice and swimming halls with total energy balance (EB5) and energy flux indices. Energy flux arrows are directionally pointed to the swimming hall, the ice hall or the waste heat processing systems. Energy inflows are colored blue and energy outflows are colored red. The potential excess heat flux from the combined energy system is shown in Figure 50, which is not included in the heat losses, since it could be utilized elsewhere. The energy fluxes are marked with indices with prefix of capital C.

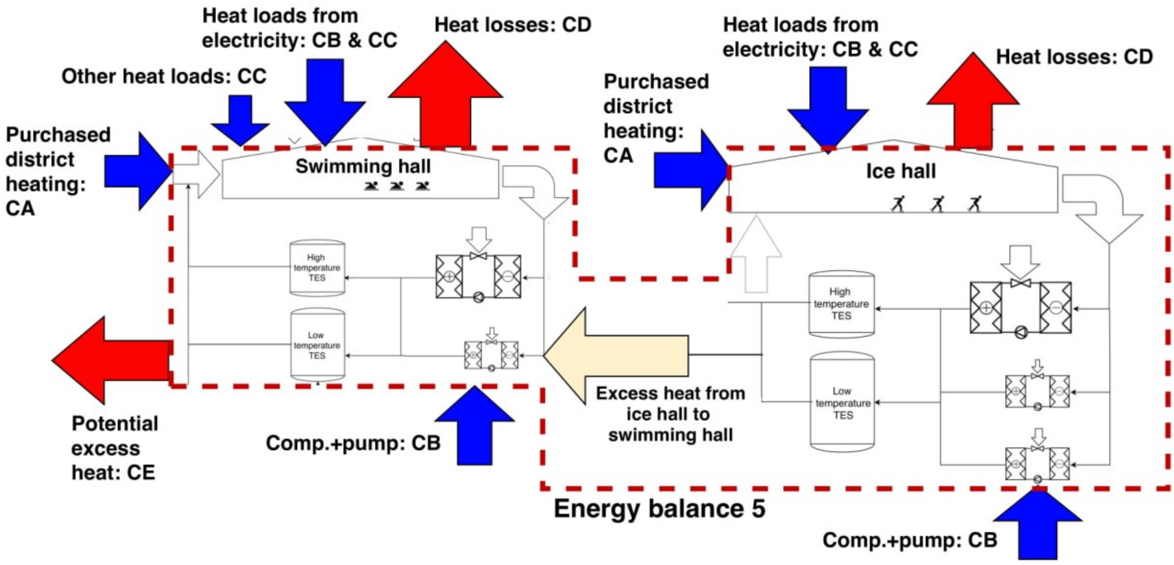

Figure 50. Simplified scheme of the combined energy system of ice and swimming halls with total energy balance (EB5) and energy flux indices.

Energy fluxes of the total energy balance of the ice and swimming halls with indices in Figure 50 are presented in Table 25. Energies in Table 25 are a combination of multiple energy fluxes shown in Figure 50 as recurring indices.

Table 25. Energy fluxes of the total energy balance of ice and swimming halls (EB5) with indices corresponding to Figure 50.

Annual combined energy [MWh/a]. Total floor area 14656 m²		
Energy balance 5: Total	Total purchased district heat	CA
	Total purchased electricity	CB
	Total heat loads	CC
	Total heat losses	CD
	Total potential excess heat	CE

4.3 Annual energies of the reference cases

Two reference cases are used as a comparison for utilization of waste heat and smart control of energy systems. The first reference case simulates year with summer breaks and the second reference case simulates full year operation without the summer breaks. Altogether 10 cases are compared to the reference case with summer breaks and 2 cases are compared to the reference case without summer breaks. The reference case energies should correspond to energy consumptions of a new ice hall and an old swimming hall.

4.3.1 The ice hall

Table 26 shows the simulated annual energies for the ice hall in the reference case with summer break. Chapter 3.2 presents approximations for an average consumption of an ice hall with two rinks to be for district heat 1200 MWh/a and for electricity 1500 MWh/a. The corresponding simulated consumption for the Pirkkola ice hall are for district heat 1600 MWh/a and for electricity 1000 MWh/a. The higher district heat consumption is within acceptable range, because the input data for approximations is quite uncertain and direct comparison with the measurement data is not possible. The lower electricity consumption simulated is explained by new technical systems.

Energy balance 1 of the building shows that heat energy of systems (1600 MWh/a) constitutes 73 % of heat losses of the ice hall (2200 MWh/a). Energy balance 2 of the technical systems shows zeros as utilization of waste heat is not included in the reference cases. Supply air heating (1300 MWh/a) dominates heat energy of systems (1600 MWh/a) with an 81 % portion. Ice refrigeration heat pump is the biggest electricity consumer with a 57 % portion. Biggest heat loads are coming from ice resurfacing water (150 MWh/a) and lighting (170 MWh/a). Heat losses are dominated by ice refrigeration heat (1600 MWh/a) with a portion of 72 % of total heat losses. Other notable heat losses are caused by DHW sewage losses (190 MWh/a). (Table 26)

Table 26. Simulated annual energies for the ice hall in the reference case with summer break (Case 1_{IH}).

Energy flows [MWh/a]. The ice hall floor area 6674 m²					
Energy balance 1: Building	Heat energy of systems	1 614	Total values	Utilized waste heat	0
	Heat loads from electricity	244		Purchased district heat	1 614
	Other heat loads	355		Purchased electricity	1 007
	Heat losses	-2 212	Breakdown of heat energy of systems	Supply air heating	1 304
	Total removed heat for use	0		DHW heating	191
Energy balance 2: Technical systems	Ice refrigeration heat	1 593		Ice resurfacing water freezing	42
	Ice refrigeration HP electricity	578		Water radiator heating	23
	Gray water heat	0		Ground frost protection	54
	Gray water HP electricity	0	Breakdown of heat loads (+) and losses (-)	Ice resurfacing water freezing	155
	Dehumidification heat	221		Lighting	174
	Dehumidification electricity	73		Occupants	90
	Supply air heating	0		Ventilation fans	69
	DHW heating	0		DHW sewage losses	-191
	Ice resurfacing water	0		DH substation losses	-48
	Water radiator heating	0		Infiltration air	-37
	Ground frost protection	0		Envelope	-56
	Excess heat to swimming hall	0		Other heat loads(+) / losses(-)	-1 842
Notation: HP = Heat Pump					

Figure 51 shows hourly heat power of systems (1600 MWh/a) and electricity power (1000 MWh/a) of the ice hall during the simulated year. In Figure 51, the gap in the middle of the year is caused by the summer break. The heat power of systems is caused mainly by supply air heating, which is higher during winter. The demand of the electricity power is quite stable thorough the year. This is because the electricity demand of ice refrigeration heat pump, ice rink coolant pumping, dehumidification, lighting and ventilation fans are stable over the whole year. Only dehumidification load is dependent on outdoor weather conditions.

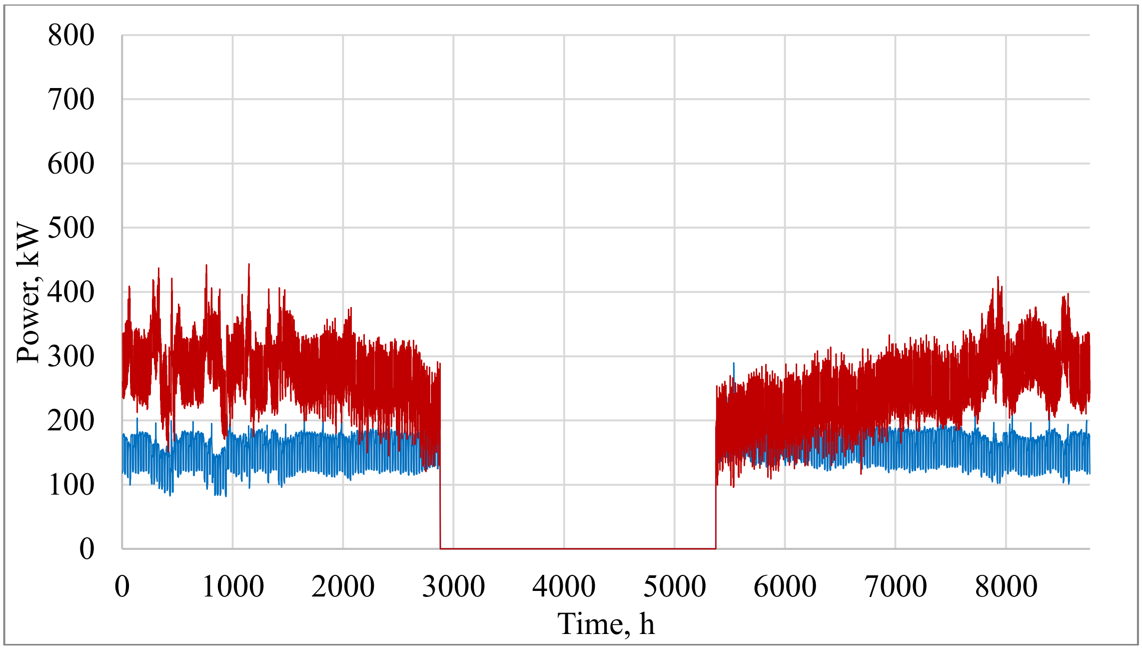

Figure 51. Heat power of systems (Red) and electricity power (Blue) during the simulated year with summer break for the ice hall (Case 1$_{IH}$).

Table 27 shows annual energies for the full year reference case of the ice hall. In the full year simulation, heat power of systems and electricity power are identical to Figure 51, except the powers continue during the summer break cap. The operation time of the full year simulation is 40 % longer than the case with summer break. Ice refrigeration heat and electricity are increased a corresponding amount of 41 %. Heat energy of systems is increased only 31 %, since heat demand is lower during the summer. The lowered heat demands are caused by increased heat loads, which are increased by 90 % when excluding electricity heat loads. Dehumidification demand is increased by 82 %, since absolute humidity of air is higher during summer period. Purchased electricity is increased by 46 % due to increased dehumidification.

Table 27. Simulated annual energies for the full year reference case of the ice hall (Case 2_{IH}).

Energy flows [MWh/a]. The ice hall floor area 6674 m²					
Energy balance 1: Building	Heat energy of systems	2 109	Total values	Utilized waste heat	0
	Heat loads from electricity	341		Purchased district heat	2 109
	Other heat loads	676		Purchased electricity	1 466
	Heat losses	-3 124	Breakdown of heat energy of systems	Supply air heating	1 683
	Total removed heat for use	0		DHW heating	267
Energy balance 2: Technical systems	Ice refrigeration heat	2 250		Ice resurfacing water freezing	58
	Ice refrigeration HP electricity	818		Water radiator heating	26
	Gray water heat	0		Ground frost protection	76
	Gray water HP electricity	0	Breakdown of heat loads (+) and losses (-)	Ice resurfacing water freezing	217
	Dehumidification heat	403		Lighting	244
	Dehumidification electricity	133		Occupants	126
	Supply air heating	0		Ventilation fans	97
	DHW heating	0		DHW sewage losses	-267
	Ice resurfacing water	0		DH substation losses	-63
	Water radiator heating	0		Infiltration air	-18
	Ground frost protection	0		Envelope	30
	Excess heat to swimming hall	0		Other heat loads(+) / losses(-)	-2 575
Notation: HP = Heat Pump					

4.3.2 The swimming hall

Table 28 shows the simulated annual energies for the swimming hall in the reference case with summer break. Chapter 3.3 presents approximations for an average consumption of a swimming hall to be for district heat 3300 MWh/a and for electricity 1900 MWh/a. The corresponding simulated consumption for the Pirkkola swimming hall are for district heat 2700 MWh/a and for electricity 1400 MWh/a. The lower consumptions are acceptable, since the Pirkkola swimming pool surface area is smaller than in the compared swimming halls used in the approximation and the water temperature set point (26.5 °C) is low compared to guidelines (EPD, 2009:3).

Energy balance 3 of the building shows that heat energy of systems (2700 MWh/a) constitutes 60 % of heat losses of the swimming hall (4500 MWh/a). Energy balance 4 of technical systems shows zeros as utilization of waste heat is not included in the reference cases. Biggest heat energy of system is DHW heating (980 MWh/a), seconded by pool water heating (670 MWh/a). Saunas (530 MWh/a) and pool pumps (440 MWh/a) are biggest electricity consumers and internal heat loads of the building. Biggest heat losses are caused by exhaust air of ventilation. Heat losses of exhaust air are not separated in the simulations, but they constitute for the rest of the heat losses of the swimming hall (3300 MWh/a). Adding an exhaust air heat pump (EAHP) would greatly reduce heat losses of exhaust air. DHW sewage losses (980 MWh) are another significant heat loss, especially in the reference cases, since no heat is recovered from sewage water. (Table 28)

Table 28. Simulated annual energies for the swimming hall in the reference case with summer break (Case 1_{SH}).

Energy flows [MWh/a]. The swimming hall floor area 7982 m²

Energy balance 3: Building	Heat energy of systems	2 717
	Heat loads from electricity	1 445
	Other heat loads	365
	Heat losses	-4 487
	Total removed heat for use	0
Energy balance 4: Technical systems	Heat from the ice hall	0
	Exhaust air HP heat	0
	Exhaust air electricity	0
	Gray water heat	0
	Gray water HP electricity	0
	Supply air preheating	0
	Supply air heating	0
	Water radiator heating	0
	Underfloor heating	0
	DHW heating	0
	DHW preheating	0
	Pool water heating	0
	Potential excess heat	0
Total values	Utilized waste heat	0
	Purchased district heat	2 717
	Purchased electricity	1 404

Breakdown of waste heat	Utilized ice hall waste heat	0
	Utilized SH waste heat	0
	Excess heat from TES 1	0
	Excess heat from TES 2	0
Breakdown of heat energy of systems	Supply air heating	322
	Supply air preheating	292
	Pool water heating	669
	Water radiator heating	348
	Underfloor heating	110
	DHW preheating	540
	DHW heating	436
Breakdown of heat loads (+) and losses (-)	Sauna	527
	Lighting	177
	Ventilation fans	119
	Pool pumps	437
	Equipment	89
	HVAC aux.	96
	Occupants	300
	Radiation through windows	65
	DHW sewage losses	-976
	DH substation losses	-82
	Infiltration air	-92
	Envelope	-335
	Other heat losses (exhaust air)	-3 263

Notation: SH = swimming hall, HP = Heat Pump

Figure 52 shows hourly heat power of systems (2700 MWh/a) and electricity power (1400 MWh/a) of the swimming hall during the simulated year. In Figure 52, the gap in the middle of the year is caused by the summer break. The heat power of space and pool water heating is reduced during summer due to bigger heat loads to the building. Fluctuation range of electricity power stays almost the same thorough the year, since the outdoor temperature and humidity affect only the electricity consumption of ventilation fans in the swimming hall simulation.

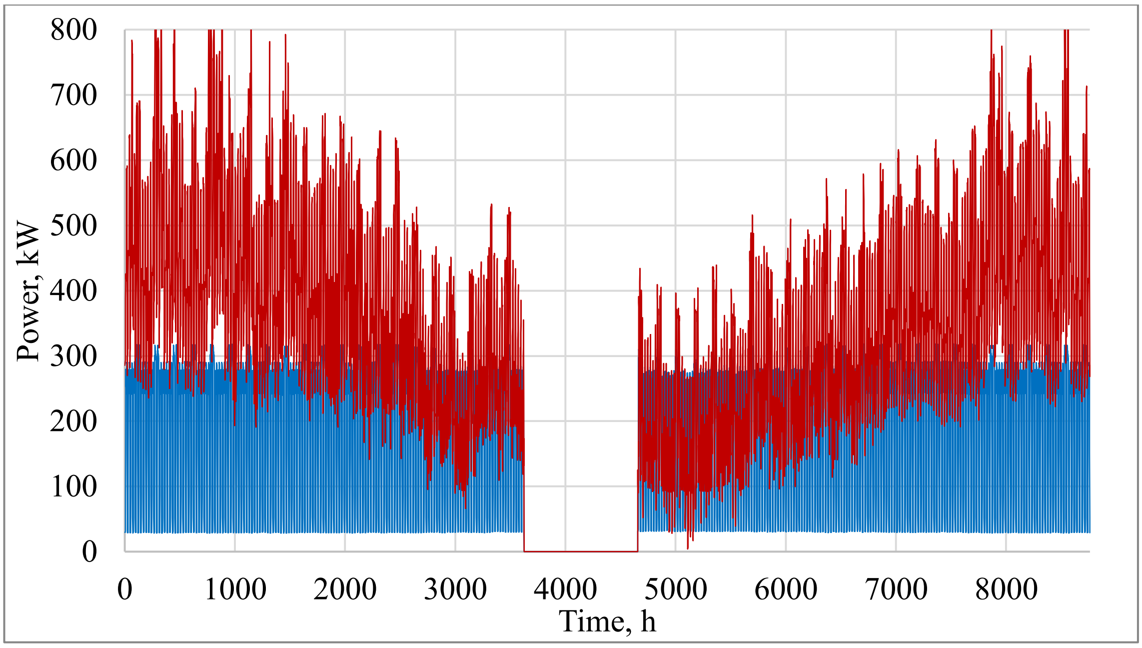

Figure 52. Hourly heat power of systems (Red) and electricity power (Blue) during the simulated year with summer break for the swimming hall (Case 1_{SH}).

Table 29 shows simulated annual energies for the full year reference case of the swimming hall. In the full year simulation, heat power of systems and electricity power are identical to Figure 52, except the powers continue during the summer break cap. The full year simulation is 13 % longer than the simulation with summer break. Heat energy of systems is increased only 9 %, since heat demands are lower during the summer period. Pool water heating is increased only 10 %. The lowered heat demands are caused by increased heat loads, which are increased by 25 % when excluding electricity heat loads. Electricity demands are increased 13 % as expected.

Table 29. Annual energies for the full year reference case of the swimming hall (Case 2_{SH}).

Energy flows [MWh/a]. The swimming hall floor area 7982 m²					
Energy balance 3: Building	Heat energy of systems	2 951	Breakdown of waste heat	Utilized ice hall waste heat	0
	Heat loads from electricity	1 636		Utilized SH waste heat	0
	Other heat loads	458		Excess heat from TES 1	0
	Heat losses	-4 997		Excess heat from TES 2	0
	Total removed heat for use	0	Breakdown of heat energy of systems	Supply air heating	337
Energy balance 4: Technical systems	Heat from the ice hall	0		Supply air preheating	293
	Exhaust air HP heat	0		Pool water heating	734
	Exhaust air electricity	0		Water radiator heating	365
	Gray water heat	0		Underfloor heating	116
	Gray water HP electricity	0		DHW preheating	612
	Supply air preheating	0		DHW heating	494
	Supply air heating	0	Breakdown of heat loads (+) and losses (-)	Sauna	594
	Water radiator heating	0		Lighting	201
	Underfloor heating	0		Ventilation fans	135
	DHW heating	0		Pool pumps	495
	DHW preheating	0		Equipment	101
	Pool water heating	0		HVAC aux.	109
	Potential excess heat	0		Occupants	340
Total values	Utilized waste heat	0		Radiation through windows	118
	Purchased district heat	2 951		DHW sewage losses	-1 106
	Purchased electricity	1 590		DH substation losses	-89
				Infiltration air	-96
Notation: SH = swimming hall, HP = Heat Pump				Envelope	-361
				Other heat loads(+) / losses(-)	-3 642

4.3.3 Total energy balance

The energy balance 5 defined in Chapter 4.2 of total ice and swimming hall energy flows is shown in Table 30. The total values are a sum of corresponding ice hall and swimming hall values.

Table 30. Simulated combined annual energies for ice and swimming halls in the reference case with summer breaks (Case 1_{IH} + Case 1_{SH}).

Combined energy flows [MWh/a]. Total floor area 14656 m²		
Energy balance 5: Total	Total purchased district heat	4 331
	Total purchased electricity	2 411
	Total heat loads	721
	Total heat losses	-7 365
	Total potential excess heat	0

Figure 53 shows hourly heat power of systems (4300 MWh/a) and electricity power (2400 MWh/a) during the simulated year with summer breaks for the total of ice and swimming hall. The fast changes in the powers between the hours of 2900 h and 5300 h are caused by summer break of the ice hall, while the swimming hall is still open.

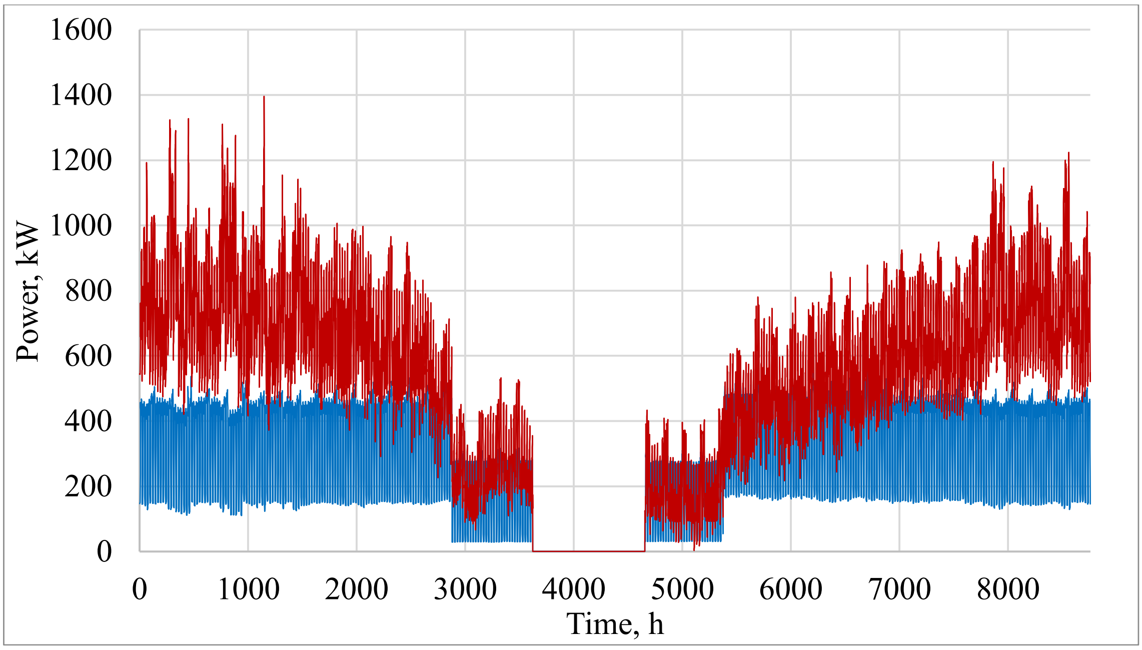

Figure 53. Hourly heat power of systems (Red) and electricity power (Blue) during the simulated year with summer breaks for the total of ice and swimming halls (Case 1_{IH} + Case 1_{SH}).

Table 31 shows annual energies of the full year reference case for the total of ice and swimming halls energy flows. Without summer breaks, the increases in the energies are for purchased district heat +17 %, for purchased electricity +27 %, for heat loads +57 % and for heat losses +23 %.

Table 31. Simulated annual energies of the full year reference case ice and swimming halls (Case 2_{IH} + Case 2_{SH}).

Combined energy flows [MWh/a]. Total floor area 14656 m²		
Energy balance 5: Total	Total purchased district heat	5 060
	Total purchased electricity	3 056
	Total heat loads	1 134
	Total heat losses	-9 094
	Total potential excess heat	0

4.4 Annual energies with utilization of waste heat

This Chapter presents the results for how much waste heat could be utilized in Pirkkola ice and swimming halls. Possible sources of waste heat recovery are ice refrigeration, gray water, condensing water from dehumidification and exhaust air. Three cases are presented in this Chapter, one without exhaust air heat pump (EAHP) in the swimming hall and two cases with different EAHP temperatures in the swimming hall. All of the cases include all rest of the sources of waste heat recovery. The cases assume summer breaks. Two cases are presented for the ice hall and the swimming hall as a connection scheme with annual energies in addition to the Table format. The methodology for these cases is presented in Chapter 3.4.

4.4.1 The ice hall

Table 32 shows the annual energies for the ice hall with utilization of waste heat. Purchased district heat is decreased by 99 %, which means the ice hall is self-sufficient for heat energy. Total electricity demand increases only by 9 %. The total purchased energy of the ice hall is decreased by 45 %. The comparison is done to reference (Case 1_{IH}). The energy of heat demand in the ice hall stays the same, since no improvements are made on demand side. The cases 3, 4 and 5 are identical between each other for the ice hall.

Also heat losses stay the same, except for DHW sewage losses, which are reduced by 38 % due to gray water heat recovery. Utilized waste heat in the ice hall (1600 MWh/s) still leaves 950 MWh/a of excess heat to be transferred for the swimming hall. Energy balance 2 of technical systems shows that ice refrigeration heat dominates as a waste heat source by being 8 times bigger than condensing heat from dehumidification and 25 times bigger than gray water heat. (Table 32)

Table 32. Annual energies for the ice hall with utilization of waste heat (Cases 3, 4 & 5).

Energy flows [MWh/a]. The ice hall floor area 6674 m²					
Energy balance 1: Building	Heat energy of systems	1 614	Total values	Utilized waste heat	1 603
	Heat loads from electricity	244		Purchased district heat	22
	Other heat loads	355		Purchased electricity	1 093
	Heat losses	-259	Breakdown of heat energy of systems	Supply air heating	1 304
	Total removed heat for use	-1 953		DHW heating	191
Energy balance 2: Technical systems	Ice refrigeration heat	1 593		Ice resurfacing water freezing	42
	Ice refrigeration HP electricity	578		Water radiator heating	23
	Gray water heat	64		Ground frost protection	54
	Gray water HP electricity	16	Breakdown of heat loads (+) and losses (-)	Ice resurfacing water freezing	155
	Dehumidification heat	221		Lighting	174
	Dehumidification electricity	73		Occupants	90
	Supply air heating	-1 266		Ventilation fans	69
	DHW heating	-182		DHW sewage losses	-118
	Ice resurfacing water	-42		DH substation losses	-48
	Water radiator heating	-23		Infiltration air	-37
	Ground frost protection	-54		Envelope	-56
	Excess heat to swimming hall	-950		Other heat loads(+) / losses(-)	110
Notation: HP = Heat Pump					

Figure 54 presents the annual energies of the ice hall presented in Table 32 as a connection scheme with temperatures of all of the heating and waste heat processing systems. The arrow sizes of the energy fluxes indicate the amount of energy.

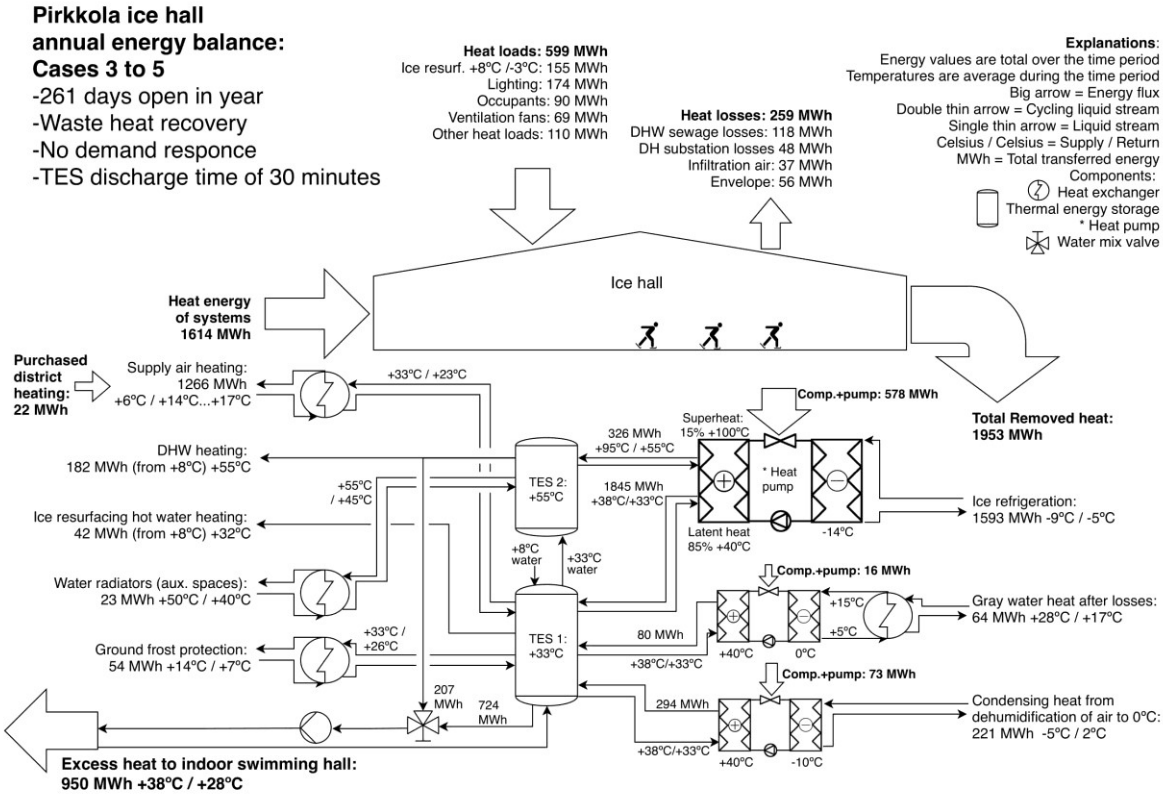

Figure 54. Annual energies of the ice hall with utilization of waste heat as a connection scheme of waste heat processing systems (Cases 3, 4 & 5).

Figure 54 is depicting the location and direction of all heat fluxes. The heat is transferred either as big energy flux arrows or as thin arrows indicating liquid flow rates transferring heat. The temperatures of all heating and waste heat processing systems are shown in Figure 54. Heat from ice refrigeration is divided into high temperature superheat and low temperature latent heat. This superheat transferred to water from low temperature TES creates the +55 °C high temperature water. All three waste heat sources produce +38 °C water for the low temperature TES. Only heating system requiring high temperature water is DHW heating, which is fully supplied by high temperature TES. Heating systems requiring heat exchangers are shown in Figure 54. Only heating systems requiring district heat is supply air heating. The excess heat from both TES tanks is mixed and transferred to the swimming hall, in this case as a 38 °C water.

Figure 55 shows a stacked chart of total heat in the ice hall (2600 MWh), divided to purchased district heat (red, 1 %), waste heat utilized in the ice hall (orange, 67 %) and excess heat in the ice hall (purple, 32 %). The chart shows visually the portion of total heating power of the ice hall (red + yellow) compared to total waste heat produced in the ice hall (yellow + purple). After summer break, the ice hall has a period of two months, when the produced waste heat is exceptionally high. This is caused by the warmer season. The utilization of the ice hall excess heat in the swimming hall may encounter limitation, since the excess heat is clustered into the warm season.

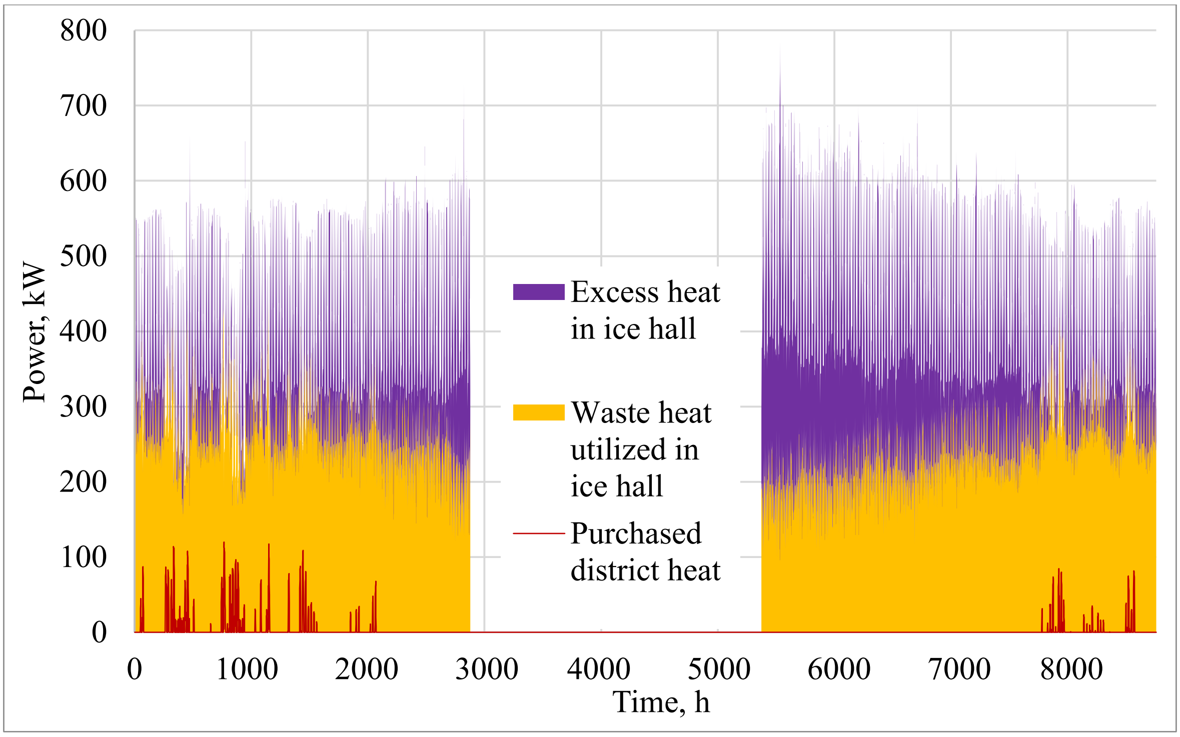

Figure 55. Total heat in the ice hall divided to purchase district heat, waste heat utilized in the ice hall and excess heat out from the ice hall (Cases 3, 4 & 5).

4.4.2 The swimming hall

Waste heat of ice hall in this thesis is transferred to the swimming hall, which is assumed in all cases with waste heat recovery. The utilization of waste heat in the swimming hall is separated for waste heat from the ice hall and for waste heat from the swimming hall. The calculations were done with an assumption, that the swimming hall utilizes its own available waste heats before it utilizes the waste heat from the ice hall. This way it is possible to observe, how much of the ice hall waste heat is used in the swimming hall compared to a case where the swimming hall only utilizes its own waste heats.

Case 3, utilization of waste heat in the swimming hall

Table 33 shows the annual energies for the swimming hall with utilization of waste heat. The case does not include exhaust air heat pump (EAHP). Thus, only waste heat source from the swimming hall is gray water (240 MWh/a) in addition to excess waste heat from the ice hall (950 MWh/a). Purchased district heat is decreased by 26 %, while electricity demand is increases only by 4 %. The heat energy of heating systems in the swimming hall stays the same, since no changes are made on demand side. Total of utilized waste heat is 730 MWh/a, of which 77 % is excess heat from the ice hall. All of the waste heat is from low temperature TES, which limits the utilization, since most of the swimming hall heat demands require high temperature heat. Heat losses are reduced by 4 % due to gray water heat recovery, which reduces DHW sewage losses by 27 %. (Table 33)

Table 33. Annual energies for the swimming hall with utilization of waste heat but without exhaust air heat pump (Case 3).

Energy flows [MWh/a]. The swimming hall floor area 7982 m²		
Energy balance 3: Building	Heat energy of systems	2 717
	Heat loads from electricity	1 445
	Other heat loads	365
	Heat losses	-4 299
	Total removed heat for use	-188
Energy balance 4: Technical systems	Heat from the ice hall	950
	Exhaust air HP heat	0
	Exhaust air electricity	0
	Gray water heat	188
	Gray water HP electricity	49
	Supply air preheating	-232
	Supply air heating	0
	Water radiator heating	0
	Underfloor heating	-71
	DHW heating	0
	DHW preheating	-417
	Pool water heating	0
	Potential excess heat	-458
Total values	Utilized waste heat	720
	Purchased district heat	1 997
	Purchased electricity	1 462

Breakdown of waste heat	Utilized ice hall waste heat	557
	Utilized SH waste heat	162
	Excess heat from TES 1	512
	Excess heat from TES 2	0
Breakdown of heat energy of systems	Supply air heating	322
	Supply air preheating	292
	Pool water heating	669
	Water radiator heating	348
	Underfloor heating	110
	DHW preheating	540
	DHW heating	436
Breakdown of heat loads (+) and losses (-)	Sauna	530
	Lighting	177
	Ventilation fans	118
	Pool pumps	437
	Equipment	89
	HVAC aux.	108
	Occupants	300
	Radiation through windows	65
	DHW sewage losses	-715
	DH substation losses	-83
	Infiltration air	-90
	Envelope	-334
	Other heat loads(+) / losses(-)	-3 077

Notation: SH = swimming hall, HP = Heat Pump

Figure 56 presents the annual energies of the swimming hall presented in Table 33 as a connection scheme with temperatures of all of the heating and technical systems. The arrow sizes of the energy fluxes indicate the amount of energy.

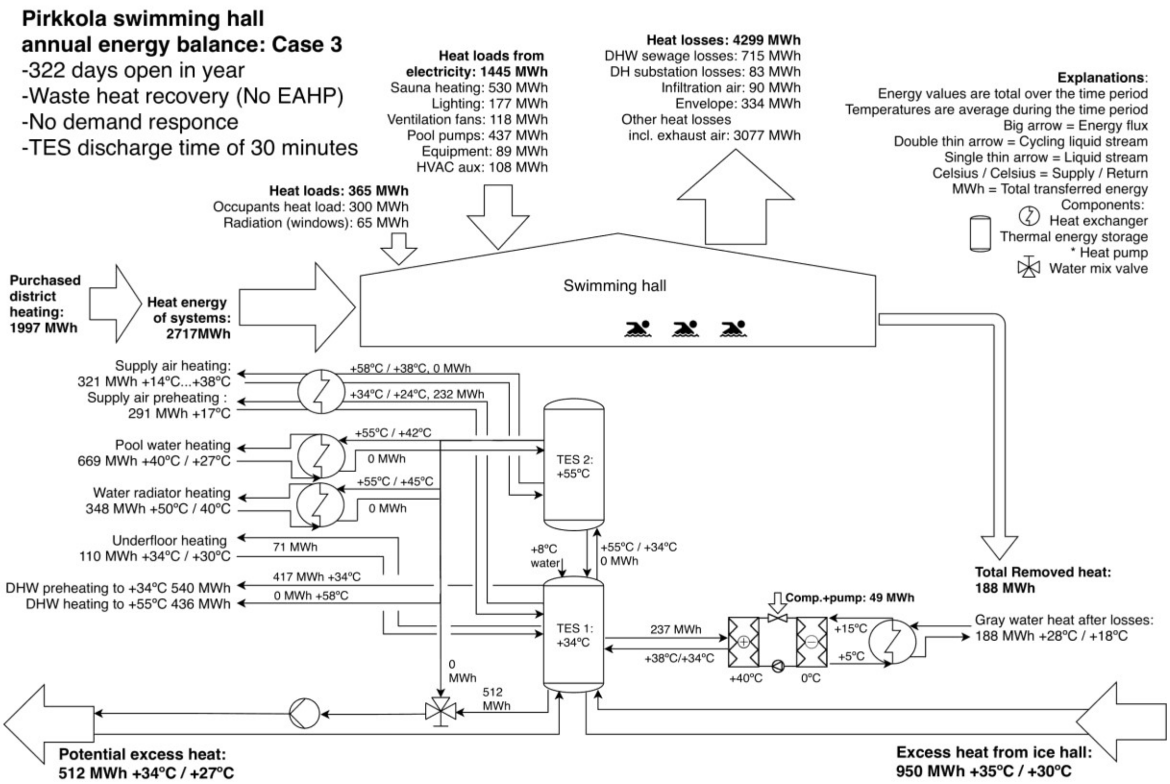

Figure 56. Annual energies of the swimming hall with utilization of waste heat as a connection scheme of waste heat processing systems (Case 3).

Figure 56 shows how waste heat from the ice hall (bottom right) is transferred to low temperature TES tank. If the TES tank is full, the waste heat from the ice hall becomes potential excess heat out from the total system. All the swimming hall heat demands, except for underfloor heating, require high temperature heat.

Figure 57 shows a stacked chart of the total heat in the swimming hall (3200 MWh), divided to purchased district heat (red, 63 %), waste heat utilized in the swimming hall (orange, 623 %) and potential excess heat from the swimming hall (purple, 14 %). The chart shows visually the portion of total heating power of the swimming hall (red + yellow) compared to total waste heat produced in the total system (yellow + purple).

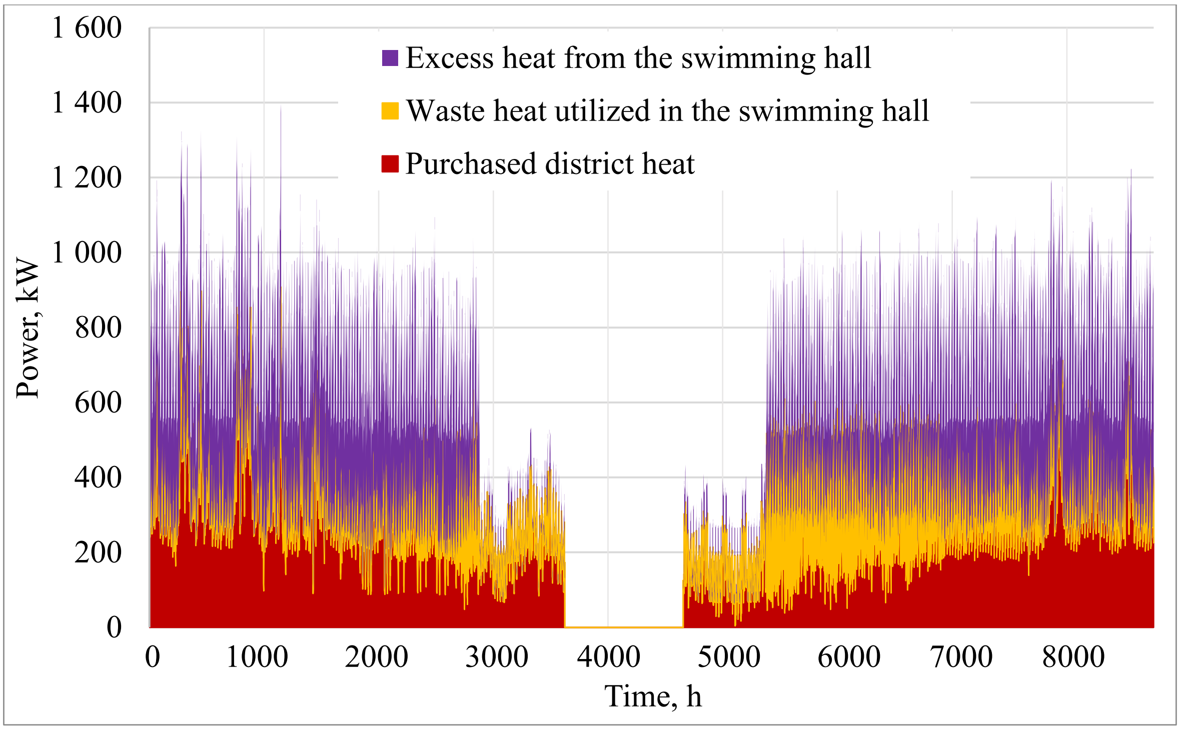

Figure 57. Total heat in the swimming hall divided to purchase district heat, waste heat utilized in the swimming hall and potential excess heat out from the swimming hall (Case 3).

Figure 57 shows well, that the waste heat remains unused while district heat is purchased. This is consequence from the temperature mismatch between the low temperature waste heat and high temperature heat demands. Thought potential excess heat from the swimming hall (purple) may seem in the chart bigger than its portion (14 %) due to small spikes coloring large areas.

Case 4, utilization of waste heat in the swimming hall with low temperature EAHP

The comparison for energies is done against the reference case. Table 34 shows the annual energies for the swimming hall with utilization of waste heat and with low temperature EAHP. The EAHP cools the exhaust air to a temperature of +5 °C and raises the condensation heat to a temperature of +40 °C. Purchased district heat is decreased in this case by 44 %, while electricity demand is increases greatly by 50 %. Total of utilized waste heat is 1200 MWh/a, of which 30 % is excess heat from the ice hall. Heat losses are reduced by 38 % due to EAHP. (Table 34) EAHP produces a huge amount of waste heat (2100 MWh/a) but increases the amount of utilized waste heat by only 500 MWh/a compared to case 3 where EAHP is not used. Thus, only 29 % of EAHP is utilized. All of the waste heat is from low temperature TES, which greatly limits the utilization, since most of the swimming hall heat demands require high temperature heat. Thus, the rest of the cases are done with an EAHP with high condensation temperature, which is able to supply all of the swimming hall heat demands.

Table 34. Annual energies for the swimming hall with utilization of waste heat with low temperature EAHP (Case 4).

Energy flows [MWh/a]. The swimming hall floor area 7982 m²

Energy balance 3: Building	Heat energy of systems	2 749
	Heat loads from electricity	1 459
	Other heat loads	365
	Heat losses	-2 769
	Total removed heat for use	-1 764
Energy balance 4: Technical systems	Heat from the ice hall	950
	Exhaust air HP heat	1 576
	Exhaust air electricity	450
	Gray water heat	188
	Gray water HP electricity	49
	Supply air preheating	-281
	Supply air heating	-43
	Water radiator heating	-225
	Underfloor heating	-106
	DHW heating	-49
	DHW preheating	-523
	Pool water heating	-13
	Potential excess heat	-2 267
Total values	Utilized waste heat	1 240
	Purchased district heat	1 509
	Purchased electricity	2 107

Notation: SH = swimming hall, HP = Heat Pump

Breakdown of waste heat	Utilized ice hall waste heat	378
	Utilized SH waste heat	794
	Excess heat from TES 1	2 266
	Excess heat from TES 2	0
Breakdown of heat energy of systems	Supply air heating	332
	Supply air preheating	292
	Pool water heating	688
	Water radiator heating	349
	Underfloor heating	112
	DHW preheating	540
	DHW heating	436
Breakdown of heat loads (+) and losses (-)	Sauna	530
	Lighting	177
	Ventilation fans	118
	Pool pumps	437
	Equipment	89
	HVAC aux.	108
	Occupants	300
	Radiation through windows	65
	DHW sewage losses	-715
	DH substation losses	-83
	Infiltration air	-90
	Envelope	-334
	Other heat loads(+) / losses(-)	-1 546

Figure 58 shows a stacked chart of total heat in the swimming hall (5000 MWh), divided to purchased district heat (red, 30 %), waste heat utilized in the swimming hall (orange, 25 %) and potential excess heat from the swimming hall (purple, 45 %). Figure 58 shows well the ratio of the unused waste heat. Thought excess heat from the swimming hall (purple) may seem in the chart bigger than its portion (45 %) due to small spikes coloring large areas.

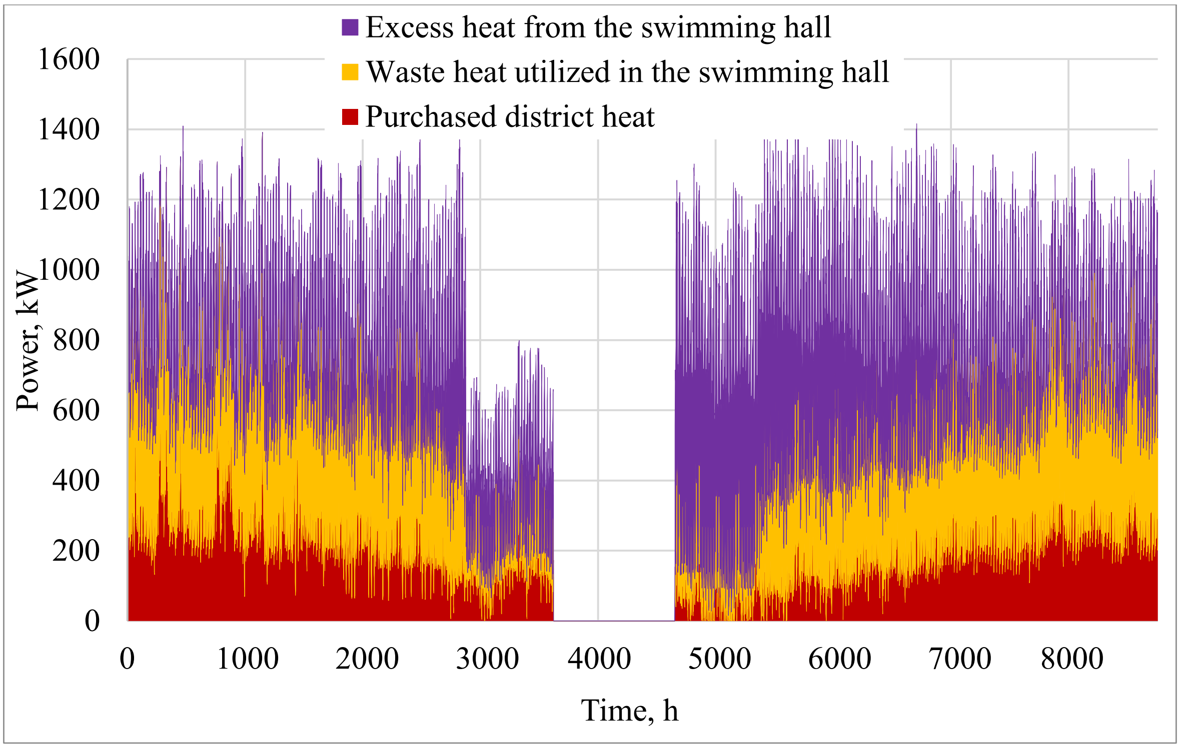

Figure 58. Total heat in the swimming hall divided to purchase district heat, waste heat utilized in the swimming hall and potential excess heat out from the swimming hall (Case 4).

Case 5, utilization of waste heat in the swimming hall with high temperature EAHP

In this case, the total purchased energy of swimming hall is decreased by 18 % compared to reference (Case 1_{SH}). An EAHP with high condensation temperature of +60 °C is able to supply all of the swimming hall heat demands. In this case, also the temperature of exhaust air cooling is raised to +10 °C to limit huge amount of excess heat.

Table 35 shows the annual energies for the swimming hall. EAHP produces 1400 MWh/a of high temperature heat in addition to low temperature heat produced by gray water heat pump (240 MWh/a) and excess waste heat from the ice hall (950 MWh/a). Portion of waste heat utilized (1100 MWh/a) from EAHP is 78 %. Purchased district heat is decreased by huge amount of 72 %, while electricity demand is increases only by 34 %. Total of utilized waste heat is 2000 MWh/a, of which only 28 % is excess heat from the ice hall. Heat losses are reduced by 28 % mainly due to EAHP. The heat energy of heating systems in the swimming hall stays the same. (Table 35)

Table 35. Annual energies for the swimming hall with utilization of waste heat and high temperature EAHP (Case 5).

Energy flows [MWh/a]. The swimming hall floor area 7982 m²		
Energy balance 3: Building	Heat energy of systems	2 746
	Heat loads from electricity	1 455
	Other heat loads	365
	Heat losses	-3 218
	Total removed heat for use	-1 308
Energy balance 4: Technical systems	Heat from the ice hall	950
	Exhaust air HP heat	1 121
	Exhaust air electricity	320
	Gray water heat	188
	Gray water HP electricity	49
	Supply air preheating	-232
	Supply air heating	-263
	Water radiator heating	-331
	Underfloor heating	-72
	DHW heating	-332
	DHW preheating	-417
	Pool water heating	-280
	Potential excess heat	-827
Total values	Utilized waste heat	1 973
	Purchased district heat	773
	Purchased electricity	1 921

Breakdown of waste heat	Utilized ice hall waste heat	558
	Utilized SH waste heat	1 415
	Excess heat from TES 1	512
	Excess heat from TES 2	316
Breakdown of heat energy of systems	Supply air heating	331
	Supply air preheating	292
	Pool water heating	687
	Water radiator heating	349
	Underfloor heating	112
	DHW preheating	540
	DHW heating	436
Breakdown of heat loads (+) and losses (-)	Sauna	530
	Lighting	177
	Ventilation fans	118
	Pool pumps	437
	Equipment	89
	HVAC aux.	104
	Occupants	300
	Radiation through windows	65
	DHW sewage losses	-715
	DH substation losses	-83
	Infiltration air	-90
	Envelope	-335
	Other heat loads(+) / losses(-)	-1 996

Notation: SH = swimming hall, HP = Heat Pump

Figure 59 presents the annual energies of the swimming hall presented in Table 35 as a connection scheme with. Figure 59 shows how different heat demands of the swimming hall are supplied mostly by high temperature waste heat. Pool water heating is supplied as the last heat demand with waste heat and thus only 41 % portion of pool water heating is supplied with waste heat. All other heat demands are mostly covered with waste heat.

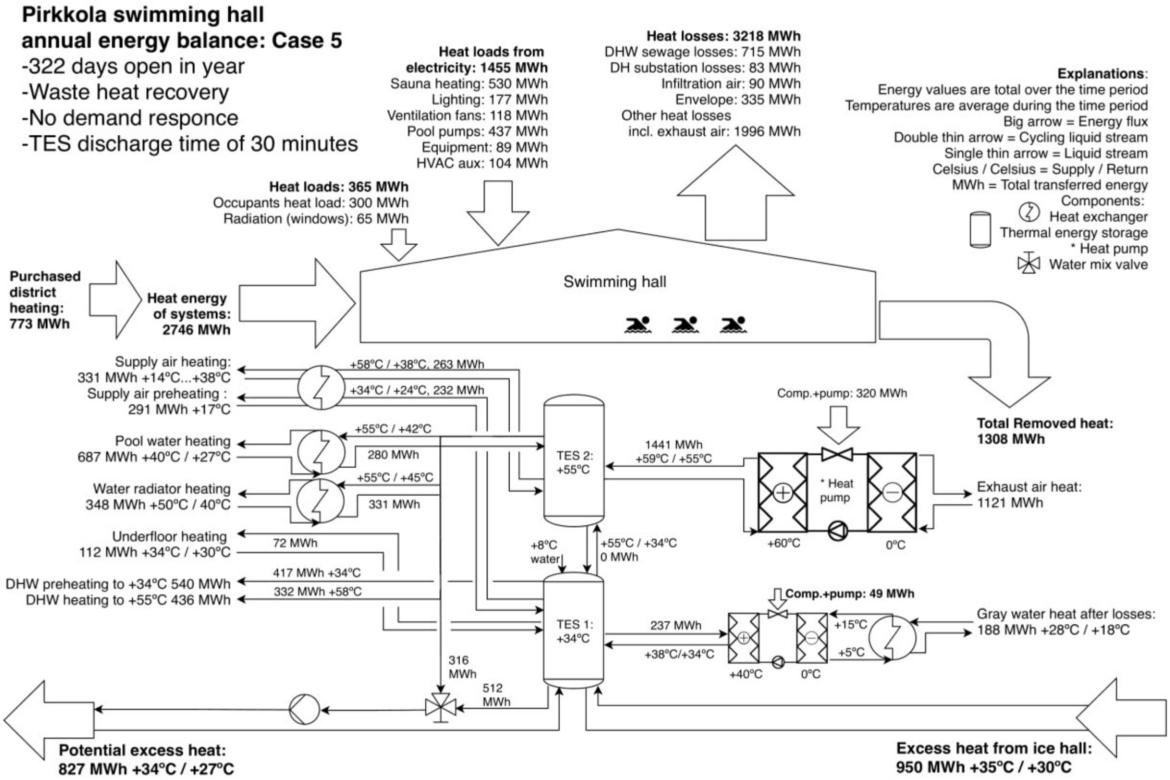

Figure 59. Annual energies of the swimming hall with utilization of waste heat and high temperature EAHP (Case 5) as a connection scheme of waste heat processing systems.

Figure 60 shows a stacked chart of total heat in the swimming hall (3600 MWh), divided to purchased district heat (red, 22 %), waste heat utilized in the swimming hall (orange, 55 %) and excess heat out from the swimming hall (purple, 23 %).

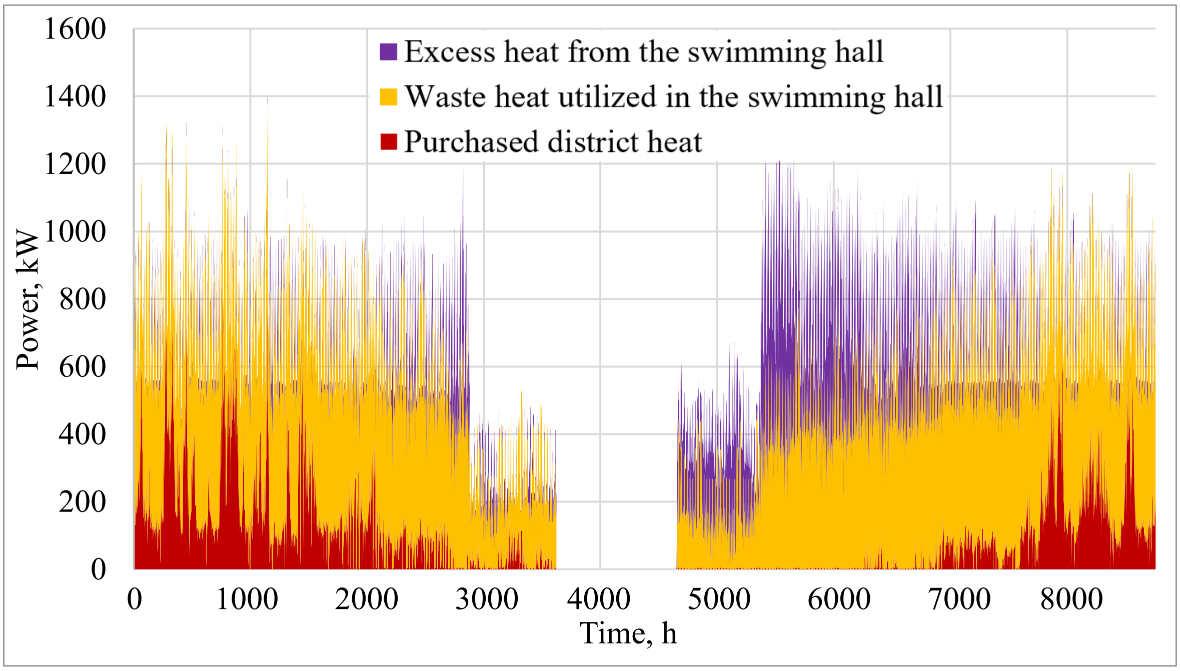

Figure 60. Total heat in the swimming hall divided to purchase district heat, waste heat utilized in the swimming hall and potential excess heat out from the swimming hall (Case 5).

Figure 60 shows visually the portion of the total heating power of the swimming hall (red + yellow) compared to total waste heat produced in the combined system (yellow + purple). Figure 60 shows the ratio of the utilized waste heat. Only a period after summer breaks has a significant amount of potential unused excess waste heat due to warmer season. Most of the potential excess heat to other buildings is low temperature excess heat from the ice hall.

4.4.3 Total energy balance

The energy balance 5 defined in Chapter 4.2 of total ice and swimming halls energy flows for cases 3, 4 and 5 are shown in Tables 36, 37 and 38. The total values are a sum from corresponding ice hall and swimming hall case values.

Table 36. Simulated combined annual energies for ice and swimming halls with utilization of waste heat but without EAHP (Case 3).

Combined energy flows [MWh/a]. Total floor area 14656 m²		
Energy balance 5: Total	Total purchased district heat	2 019
	Total purchased electricity	2 554
	Total heat loads	721
	Total heat losses	-4 558
	Total potential excess heat	-633

For the case 3, the total purchased district heat is reduced by 53 %, mostly due to self-sufficient the ice hall for heat. Total purchased electricity is increased only by 6 %, since only gray water heat pumps consume additional electricity compared to the reference case. Total heat loads stay the same, but heat losses decrease by 38 % due to recovered heat. Total potential excess heat (630 MWh/a) is not included in the heat losses, since it could be utilized elsewhere.

Table 37. Simulated combined annual energies for ice and swimming halls with utilization of waste heat with low temperature EAHP (Case 4).

Combined energy flows [MWh/a]. Total floor area 14656 m²		
Energy balance 5: Total	Total purchased district heat	1 531
	Total purchased electricity	3 205
	Total heat loads	721
	Total heat losses	-3 028
	Total potential excess heat	-2 267

For the case 4, the total purchased district heat is reduced by 65 %. Total purchased electricity is increased by 33 % mostly due to EAHP. Total heat loads stay the same, but heat losses decrease by 41 % due to recovered heat. Total excess heat is huge (2300 MWh/a), mostly due to EAHP.

Table 38. Simulated combined annual energies for ice and swimming halls with utilization of waste heat and high temperature EAHP (Case 5).

Combined energy flows [MWh/a]. Total floor area 14656 m²		
Energy balance 5: Total	Total purchased district heat	795
	Total purchased electricity	3 019
	Total heat loads	721
	Total heat losses	-3 477
	Total potential excess heat	-827

For the case 5, the total purchased district heat is reduced by huge amount of 82 %, while purchased electricity is increased only by 25 %. This indicates huge energy cost savings. Total heat loads stay the same, but heat losses decrease by 53 % due to recovered heat. Total amount of potential excess heat is moderate (830 MWh/a).

4.5 Annual energy cost savings and energies with smart control

This Chapter presents the results for smart control of energy system, which includes demand response and smart control of exhaust air heat pump (EAHP). The different systems controlled with demand response are ice refrigeration (Case 6_{IH}), swimming pool water (Case 7_{SH}), smart EAHP and saunas (Case 8). Cases 9 and 10 include all of the mentioned demand response systems. Case 10 is calculated as an exception without summer break. Cases 6_{IH} and 7_{SH} do not include waste heat recovery, which makes the analyzing of demand response more distinct. The methodology for these cases is presented in Chapter 3.5. Tables showing all annual energy fluxes for these cases are provided as an Appendix (Appendix H).

4.5.1 The ice hall

The case 6_{IH} does not include WHR to clarify the benefits of demand response of electricity for ice refrigeration. This system is implemented also for cases from 9 to 13.

The temperature set points for ice in demand response cases are -6 °C and -3 °C and in the reference case only -3 °C. In addition, the air temperature is lower in demand response cases. The temperature set point of indoor air is decreased the same amount as the ice temperature. The changes in annual energy consumptions are small compared to the reference case. Lower heating set point of indoor air causes lowered heat energy of systems, which are reduced by 3.7 % due to reduced heat demand of supply air. Purchased electricity remained about the same (-0.2 %), while electricity consumption of ice refrigeration is increased by 3 MWh/a and dehumidification is decreased by 5 MWh/a. Demand response of electricity for ice refrigeration decreases the total electricity costs of the ice hall by 1.9 %.

A surface temperature of ice is used to control the ice temperature. On average, the inside temperature of ice is 0.4°C colder than the surface. The temperatures of the two ice rinks in the ice hall are almost identical. The average temperature of the two rinks differs only by 0.07 °C. Figure 61 shows an example period (The two last weeks of December) for ice temperature of the ice rink in the bigger hall, which is controlled with demand response.

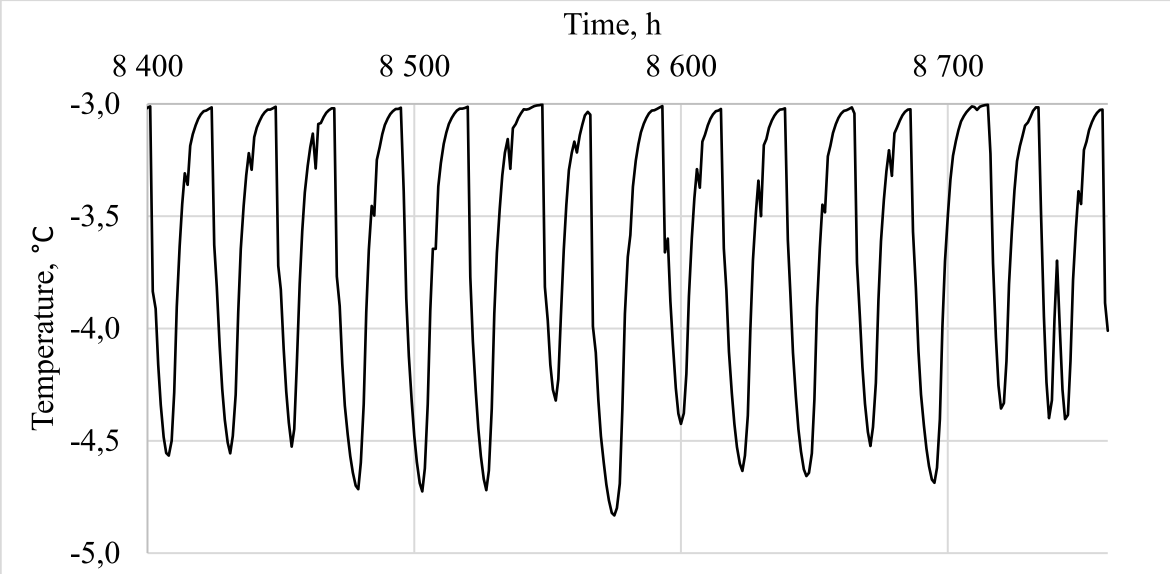

Figure 61. Ice temperature controlled with demand response for an example period (The two last weeks of December).

The ice temperature controlled with demand response follows daily cycles, since the electricity price is mostly classified as cheap during nights and as expensive during days (Figure 61). The cycles are not exactly identical, since the electricity price does differ between days. The average ice and air temperatures for the reference case are -3.0 °C and 6.0 °C, and for cases with demand response -3.6 °C and 5.4 °C respectively. Figure 62 shows the ice temperature for the first day (24 hours) of the two-week period, where the night time temperature of the ice is reduced to -4.5 °C and in the day time the temperature is normalized to -3.0 °C

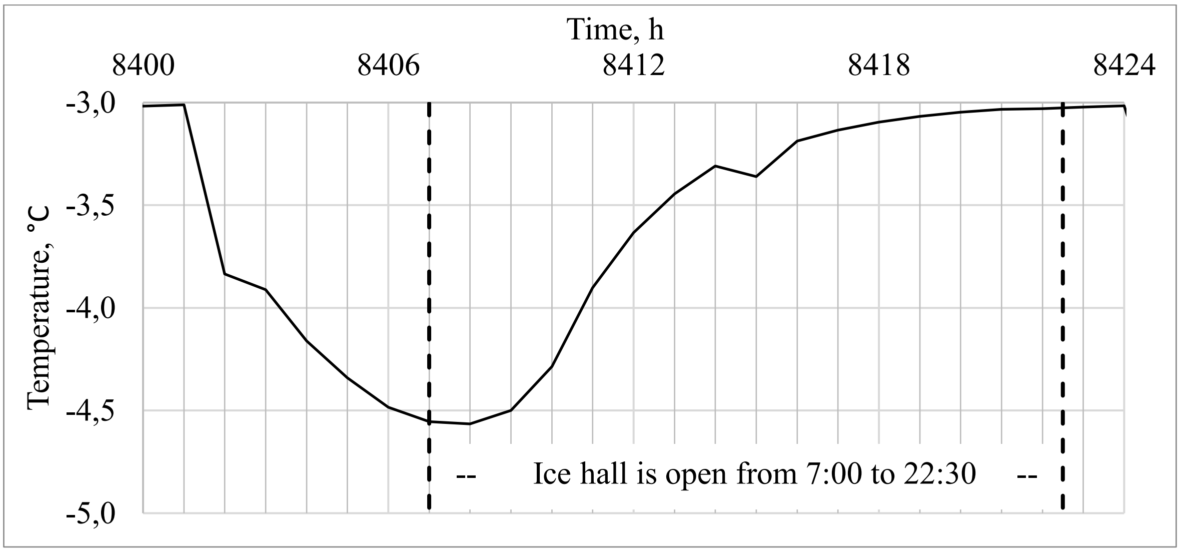

Figure 62. Ice temperature controlled with demand response for an example day.

The ice temperature has a delay from 1 to 2 hours, before it starts fully increasing (Figure 62). This delay is caused either by heat loads of ice hall, which take hours to have a full effect after the hall opening, or by ice refrigerator system, which takes time to adjust to the new temperature set point. The refrigerator works in decreased power during the time, when the ice temperature is increasing. Thus, the model for demand response of ice refrigeration works as intended. The low thermal capacity of ice is the reason, why only a small portion of the daily expensive electricity can be saved.

With demand response of ice refrigeration the average price of purchased electricity and the total electricity costs of the ice hall are decreased by 1.9 %. Purchased district heat energy is reduced by 3.7 % due to lower average indoor temperature and district heating costs are reduced by 3.9 %. The total energy cost saving in ice hall is 2.9 %. Figure 63 shows electricity prices as a function cumulative distribution of purchased electricity energy for the reference case and a case with demand response of ice refrigeration.

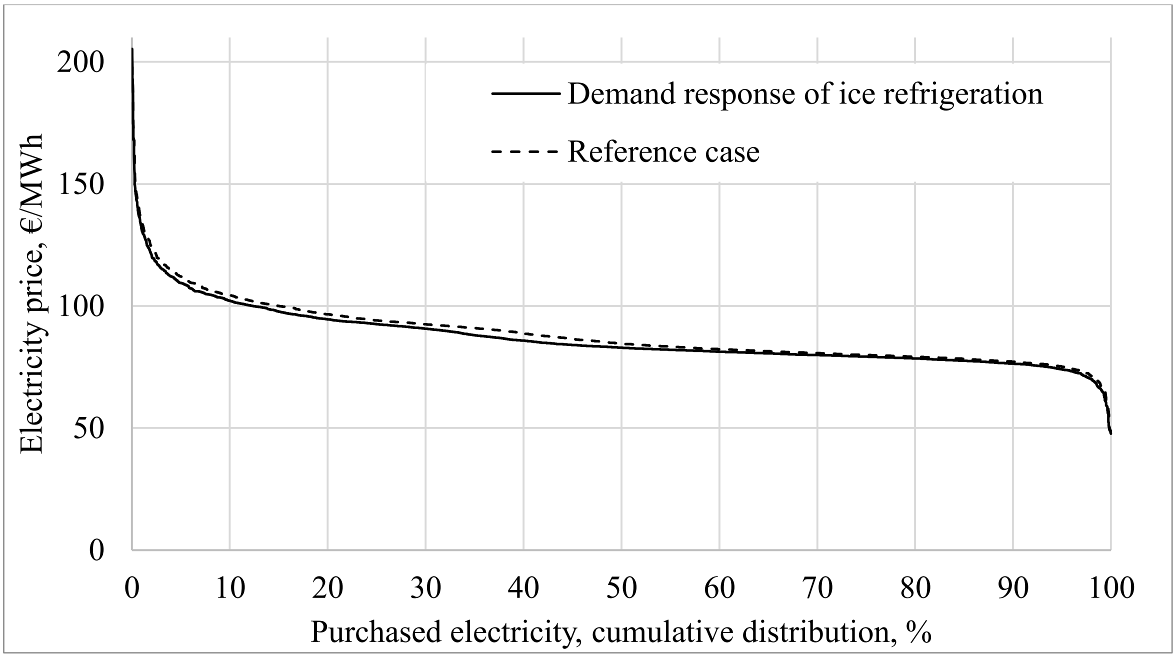

Figure 63. Electricity prices as a function of cumulative distribution of purchased electricity energy for reference (Case 1_{IH}) and with demand response of ice refrigeration (Case 6_{IH}).

Figure 63 shows the difference between electricity prices of purchased electricity energies. The difference can be seen for 50 % of the expensive end of the cumulative distribution. For example at electricity price of 90 €/MWh, the cumulative purchased electricity is for the reference case 39.6 % and for demand response case only 37.7 %. This difference of 1.9 % corresponds well to the reduced electricity costs of 1.9 % for ice hall.

The case 10 for the ice hall includes demand response as case 6_{IH} does, except it does not include summer breaks. The length of full year simulation for ice hall is 40 % longer, than with summer break. Most of the heat loads and system annual energies are increased by the corresponding 40 %. However, the benefits of demand response of ice refrigeration increase even more, since the ice refrigeration electricity is increased by +46 %.

4.5.2 The swimming hall

Case 7, demand response of district heat for pool water heating

This case does not include WHR to clarify the benefits of only demand response of district heat for swimming pool and pool space air. The average temperature of big pool water increases from the normal set point of 26.5 °C to 27.3 °C and the children pool water increased from 26.5 °C to 26.8 °C. The temperature set point of pool space air is kept 2 °C higher than pool water temperature. These higher heating set points increased heat energy of systems and purchased district heat by 1.7 %. In this case, the district heating costs of swimming hall decrease by 1.1 %. Pool water heating is increased by 5.5 %. Other than these changes, the annual energies of swimming hall have stayed the same as in the reference case 1_{SH}.

Figure 64 shows water temperature through the year for the big pool and Figure 65 for the children pool. The big pool has 4 times more surface area and thus has bigger heat demand.

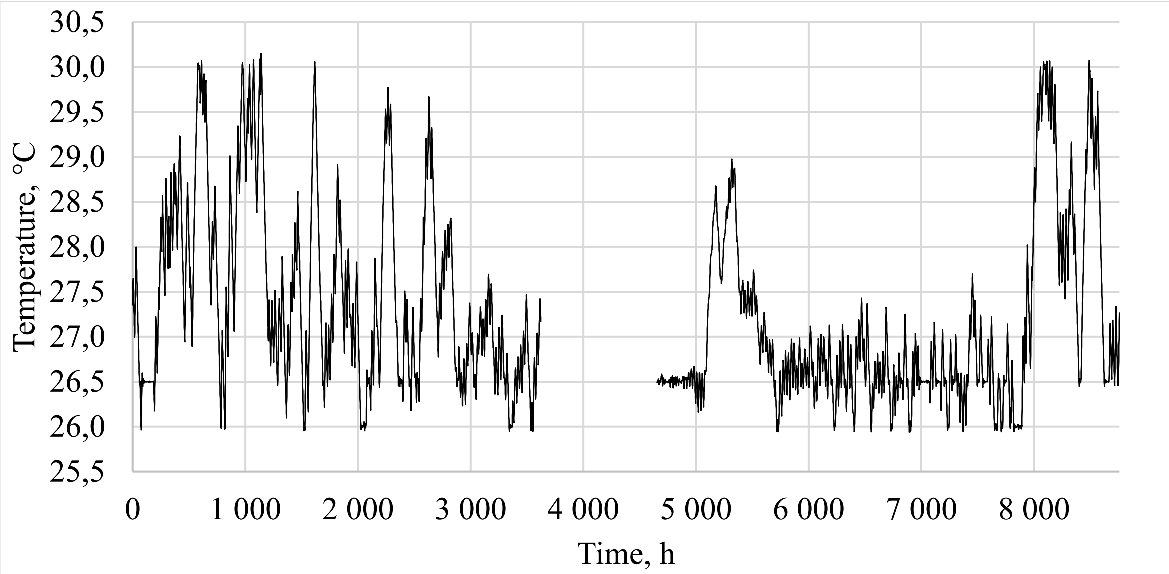

Figure 64. Water temperature of the big pool with demand response through the year with summer break.

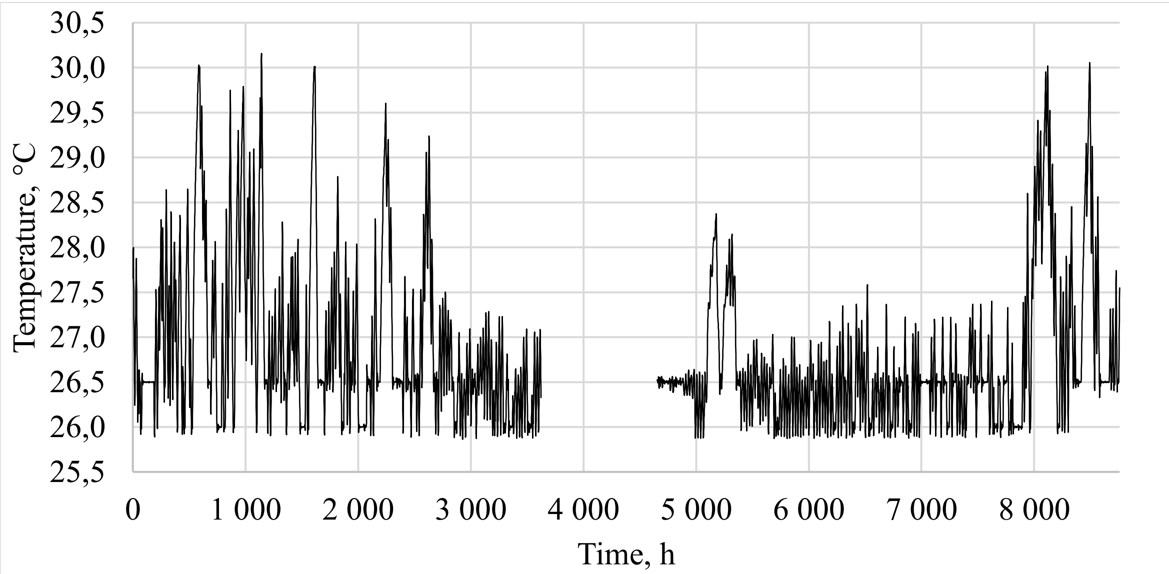

Figure 65. Water temperature of the children pool with demand response through the year with summer break.

The big pool is average of 2 times more deep than the children pool and thus has about two times higher thermal capacity. The bigger thermal capacity of water increases the potential of demand response. With the used control strategy, the longest period, when the water temperature is over the normal set point of 26.5 °C is 25 days (hours 200 to 800, Figure 64). The average temperatures are higher during the first half of the year (27.5 and 27.0 °C) than the second half of the year (27.1 and 26.7 °C). This is caused by district heating trends of decreasing price during the first half of the year and increasing price trends during the second half of the year. This effect could be compensated with a more advanced control algorithm, which takes into account the price trends.

The average purchased district heat price is decreased by 2.8 % in the swimming hall with demand response of pool water heating. Figure 66 shows district heat price as a function of cumulative distribution of purchased district heat energy for the reference case and a case with demand response of pool water heating. The total district heat costs of swimming hall are reduced by 1.1 %, which is less than the change in average price, since the total purchased district heat has increased. The total energy cost saving in swimming hall is 0.4 %.

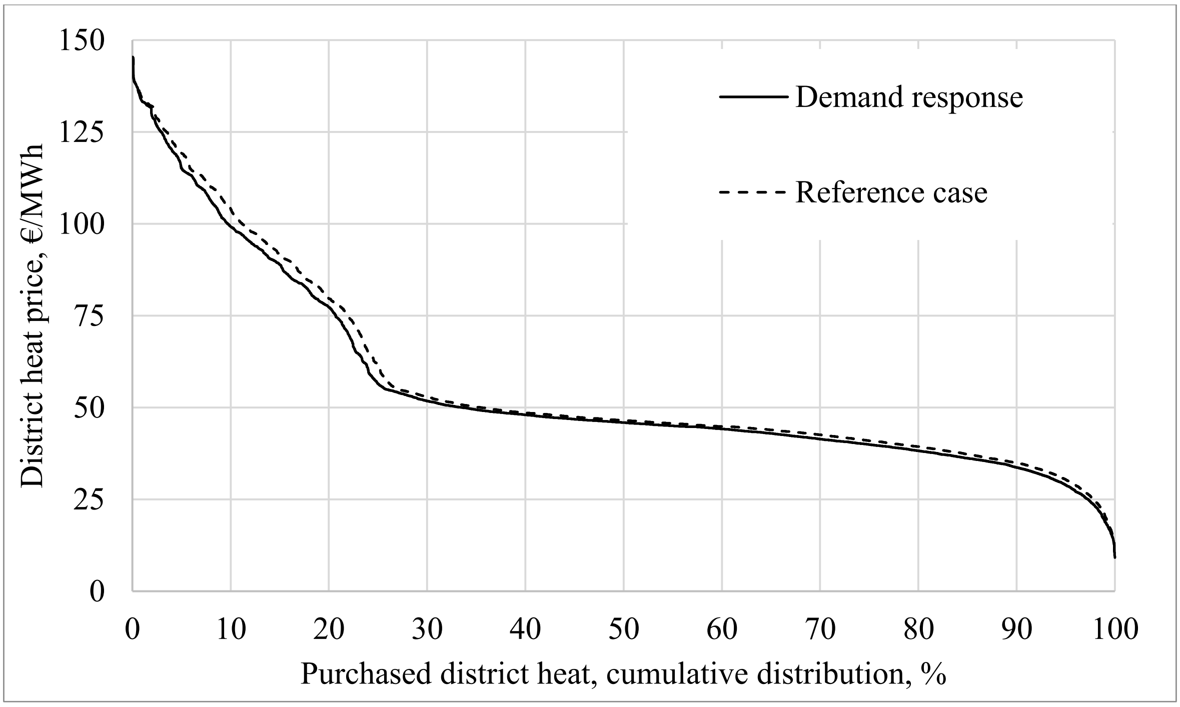

Figure 66. District heat price as a function of cumulative distribution of purchased district heat energy for the reference case 1_{SH} and with demand response of district heat for pool water heating (Case 6_{SH}).

Figure 66 shows the difference between prices of purchased district heat. The difference can be seen for about 25 % of the expensive end of the cumulative distribution. For example at district heat price of 75 €/MWh, the cumulative purchased district heat is for the reference case 21.9 % and for demand response case only 20.7 %. This difference of 1.2 % corresponds well to the reduced district heat costs of 1.1 % for the swimming hall.

Case 8, demand response of electricity in swimming hall and smart EAHP

Demand response of electricity in swimming hall and smart EAHP reduce the total energy cost of the swimming hall by 23 %. The district heating costs of the swimming hall decreased by 64 %, while the electricity costs of swimming hall increased only by 23 %. The comparison is done to reference (Case 1_{SH}). The annual energies in this case are the same as in case 5, except for EAHP, pool water heating and saunas. Swimming pool water heating, sauna heating and EAHP are controlled with demand response of electricity in this case.

The average price of purchased electricity in the swimming hall decreases by 2.0 % compared to the reference case and 1.2 % compared to the case 5 without demand response. Figure 67 shows electricity price as a function of cumulative distribution of purchased electricity for the reference case and this case with demand response of electricity in the swimming hall.

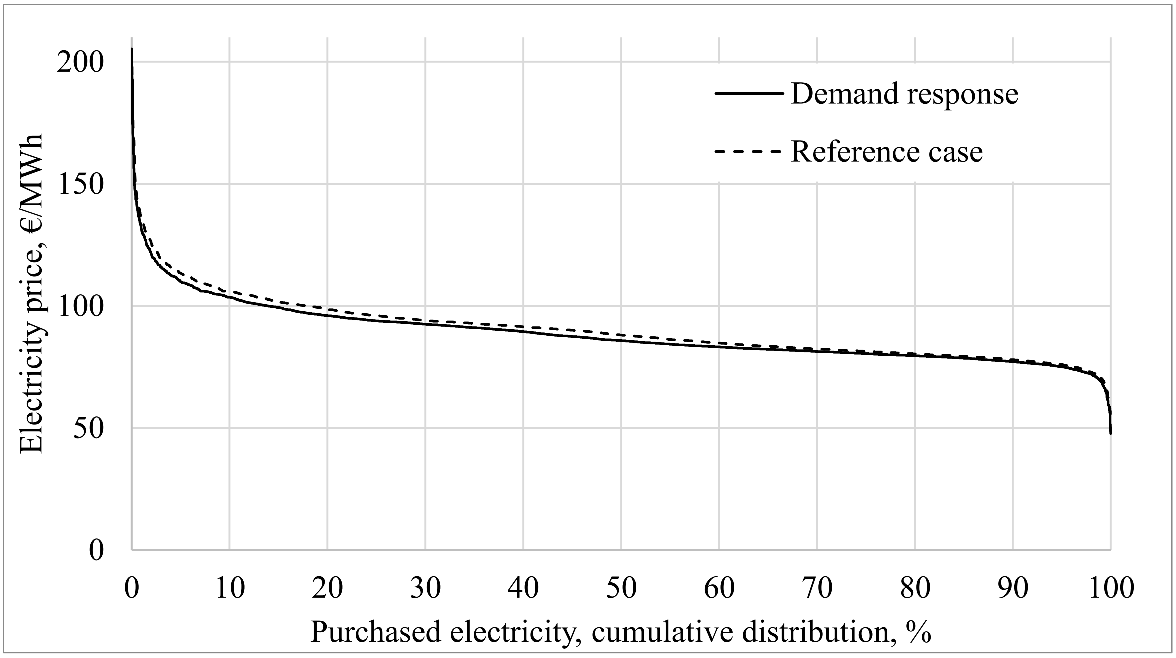

Figure 67. Electricity price as a function of cumulative distribution of purchased electricity energy for reference (Case 1_{SH}) and with demand response of electricity for swimming hall (Case 8_{SH}).

Figure 67 shows the difference between prices of purchased electricity. The difference can be seen for about 60 % of the expensive end of the cumulative distribution. For example at electricity price of 90 €/MWh, the cumulative purchased electricity is for the reference case 45 % and for demand response case only 39 %.

The swimming pool heating uses also district heating during times, when waste heat is not available. The maximum design power of the big pool is limited in this case to 120 W/m^2 (per water surface area) from the default of 200 W/m^2. Without doing this reduction in this case, the temperature of the big pool water would not normalize to the normal set point of +26.5 °C from the higher temperature set point of +30 °C. This is caused by higher temperature set point during each night, which is caused by cheap electricity during the nights. In addition, the significant thermal capacity of the big swimming pool prevents the pool from cooling down. Limiting the maximum design power prevented the temperature of the big pool water from increasing over +27 °C. A maximum temperature between +28 °C and +30 °C would have been optimal but was not achieved.

Temperatures of big pool and children pool water are both controlled with demand response. The following figures present the temperature of the big pool. In this case, the average temperature of the big pool water decreased from the normal set point of 26.5 to 26.3 °C. Pool water heating stays about the same as in case 5 (-0.5 %), even though the average temperature is lower. This is probably caused by time mismatch between heat received from EAHP and heat demand of pool water heating. Figure 68 shows the water temperature through the year for the big pool.

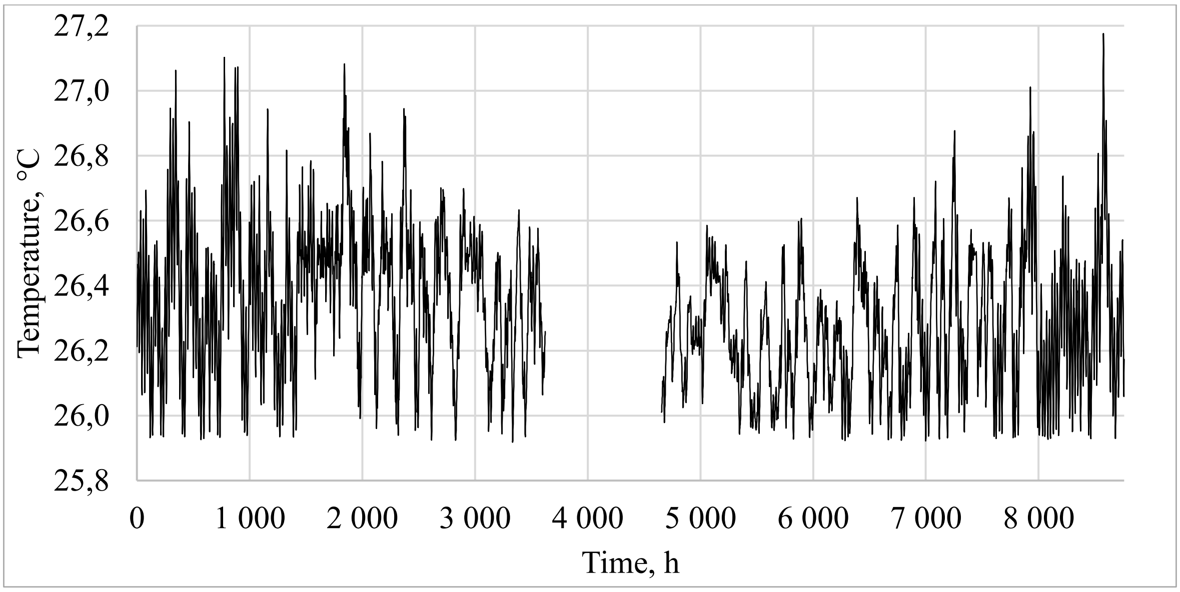

Figure 68. Water temperature of the big pool through the year with summer break. Demand response of electricity for EAHP is utilized in swimming pool water heating.

The water temperature stays close to the normal set point of 26.5 °C through the year, which is not intended. This relatively constant temperature limits the benefits of demand response. Thus, the performance of the algorithm of pool water heating controlled with demand response of EAHP electricity is not optimal. Still, the total energy cost savings is higher than any previous cases.

Saunas are heated during opening times. With demand response of electricity, saunas have two temperature set points, 75 and 90 °C, compared to one temperature set point of 85 °C in reference (Case1$_{IH}$). Average sauna temperature stays close to reference with 1 °C decrease and heating energy of saunas decreased by 2 % compared to the reference case. Figure 69 shows sauna temperature controlled with demand response of electricity during an example period (The last week of December).

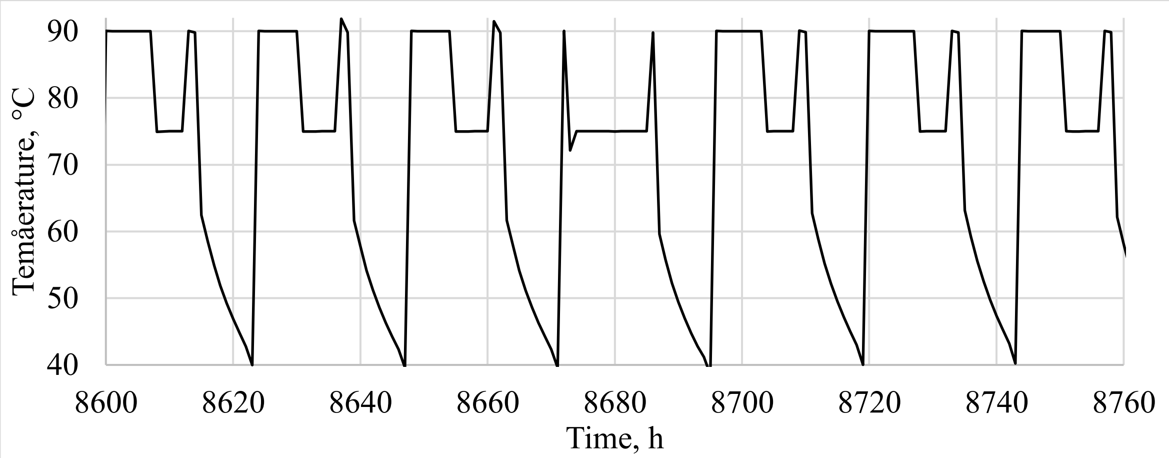

Figure 69. Sauna temperature controlled with demand response of electricity during an example period (The last week of December).

Figure 69 shows, how the sauna temperature decreases rapidly, when the temperature set point is lowered. Thus, saunas are not suitable for loading strategy of demand response. The saunas in the model did have lower thermal capacity than the rest of the building. While increasing the concrete thickness would increase thermal capacity of saunas, it would not decrease the heat losses. Thus, saunas are suitable mainly for peak clipping strategy of demand response. The sauna temperature set point is decreased for most of the days from 17 to 20 according to the highest electricity prices of the day.

In the case 5, the EAHP with constant evaporation temperature produces a lot of excess heat. In this case 8 and the following cases, the smart EAHP adjusts the temperature set point of exhaust air to match the heat demand of the upcoming hour. The algorithm is presented in Chapter 3.5.4. The condensation temperature of EAHP is kept at the same constant value of +60 °C as in the case 5.

The smart EAHP decreases the electricity consumption by 29 MWh/a compared to the case 5, which corresponds to 1.6 % decrease in electricity consumption of the swimming hall. The energy cost savings are assumable even more, since the smart EAHP is turned off during 12 % of hours of most expensive electricity price. The smart EAHP produces 1310 MWh/a heat, of which 87 % (1130 MWh/a) is utilized. The EAHP with a constant evaporation temperature of +10 °C (Case 5) produces 1440 MWh/a heat, of which 78 % (1130 MWh/a) is utilized. Thus, the smart EAHP is about 10 % more efficient in terms of producing utilized waste heat.

A breakdown of total heat in the swimming hall is very close to a stacked chart presented for case 5 in Chapter 4.4.2 (Figure X). The small changes being that the purchased district heat is decreased from 22 to 20 %, waste heat utilized in the swimming hall is increased from 55 to 57 % and potential excess heat from the swimming hall stays at 23 %.

Cases 9 and 10 , demand response of combined electricity and district heat in swimming hall and smart EAHP

These cases include demand response of electricity for smart EAHP and saunas, and as a new system to previous cases; demand response of combined electricity and district heat for heating of swimming pool water and pool space air. The yearly opening times are different for cases 9 and 10, since case 10 does not include a summer break.

The only difference between case 9 and the previous case 8 is the addition of demand response of district heat for heating of swimming pool water and indoor air. Figure 70 shows water temperature through the year for the big pool. The average water temperatures during the year are for these cases 26.9 °C (Case 9) and 26.8 °C (Case 10), which are 0.4 °C lower than with demand response of electricity (Case 8), but 0.5 °C higher than with demand response of district heat (Case 7). The average temperature of pool water for these cases is with the set targeting range.

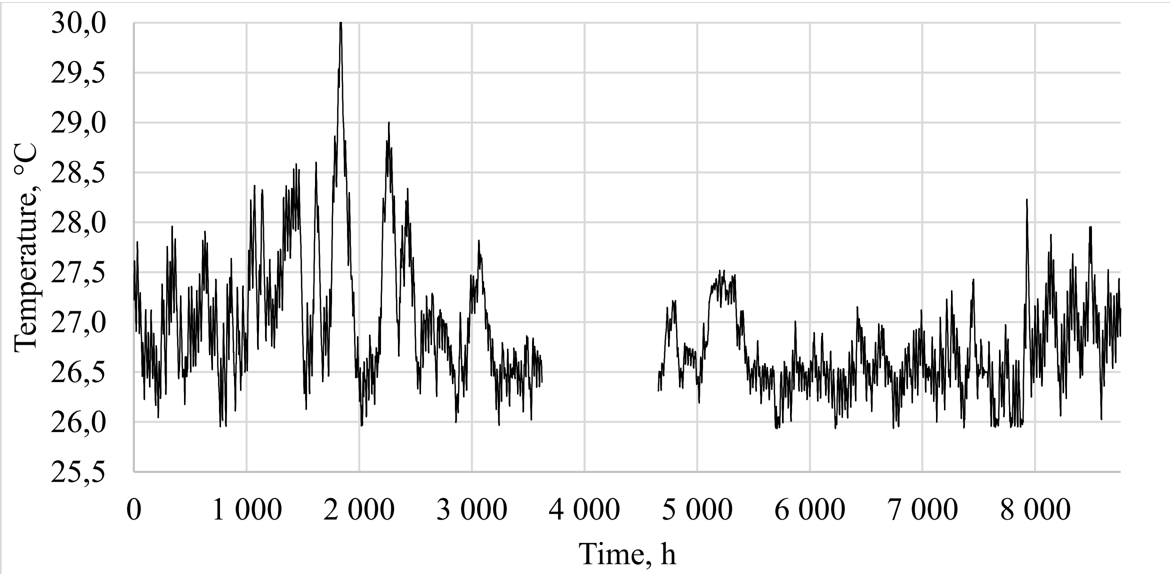

Figure 70. Water temperature through the year for the big pool with demand response of the combined electricity and district heat for heating of the water.

Demand response of electricity (Case 8) caused long periods of high water temperature during winter. With this combined electricity and district heat algorithm, the long periods of high water temperature were not caused. Figure 70 shows, that the only period of high water temperature (hours 1800 to 2000) was reversed to normal temperature set point of 26.5 °C as soon as it reached the higher temperature set point of 30 °C. Thus, the algorithm keeps the water temperature at intended ranges better than in case 8. However, compared to case 8, the utilized waste heat to pool water heating is decreased by 8 % (30 MWh/a). This decrease in utilized heat is assumable caused by the time mismatch of demand response of EAHP and the actual heating power of pool water heating.

The average price of purchased electricity in swimming hall in this case 9 is decreased compared to reference (Case 1_{SH}) by the same amount of 2.0 % as in case 8. The average price of purchased district heat is increased by 12 % compared to the reference case. However, this increase does not imply the faultiness of demand response, since the annual purchased district heat in swimming hall is reduced by 68 % due to utilization of waste heat, which changes the profile of purchased district heat prices.

Case 10 for swimming hall is identical to the presented case 9, except it does not include summer breaks. The length of full year simulation for the swimming hall is 13 % longer, than with summer break. Most of the heat loads and system annual energies are increased by the corresponding 13 %. Exception increases in annual energies are pool water heating (+10 %), supply air heating (+3 %) and water radiator and underfloor heating (+5 %). Purchased district heat is increased only by 2 % due to increased excess heat from the ice hall (+68 %) and lower space heating demand, which are caused by the warm summer season.

4.5.3 Total energy balance

The energy balance 5 (defined in Chapter 4.2) of total ice and swimming halls annual energy flows for cases 6 to 10 are shown in Table 39. The energy balance for case 6_{IH} is calculated by combining the ice hall energy flows with reference (Case 1_{SH}) swimming hall energy flows. The energy balance for case 7_{SH} is calculated by combining the swimming hall energy flows with reference (Case 1_{IH}) ice hall energy flows.

Explanations for the cases:

- Case 6_{IH} + Case 1_{SH}: demand response of electricity for ice refrigerator.
- Case 7_{SH} + Case 1_{IH}: demand response of district heat for pool water heating.
- Case 8: demand response of electricity in swimming hall and smart EAHP.
- Case 9: demand response of electricity for ice refrigerator, demand response of combined electricity and district heat in swimming hall and smart EAHP.
- Case 10: same as case 9, but full year without summer breaks.

Table 39. Simulated combined annual energies for ice and swimming halls for cases 6 to 10.

Combined energy flows [MWh/a] Total floor area 14656 m²		Case 6_{IH} + Case 1_{SH}	Case 7_{SH} + Case 1_{IH}	Case 8	Case 9	Case 10
Energy balance 5: Total	Total purchased district heat	4 271	4 378	795	921	949
	Total purchased electricity	2 410	2 411	2 948	2 947	3 720
	Total heat loads	718	718	722	717	1 147
	Total heat losses	-7 303	-7 409	-3 558	-3 547	-3 881
	Total potential excess heat	0	0	-696	-847	-1 514

Table 39 shows the differences between the cases where cases 6_{IH} and 7_{SH} do not include waste heat recovery and thus have energy flows close to the reference case (Case 1_{IH} + Case 1_{SH}) with differences less than 1 %. Cases 8 and 9 have the least purchased district heat and electricity of the presented cases. Case 10 annual energy values are bigger than in cases 8 and 9, since it does not include summer break.

4.6 Purchased district heat with bigger thermal energy storages

This Chapter presents the results for cases 8_{2h}, 9_{2h} and 10_{2h}. These cases are identical to cases 8, 9 and 10, which are presented in the Chapter 4.5, except the thermal energy storage (TES) capacities are increased. The increase in TES capacities allows more waste heat to be utilized, which decreases the amount of purchased district heat. The TES capacities are expressed as discharge times during average heat demands. The TES discharge times are increased from 30 minutes to 2 hours. The methodology for TES is presented in Chapter 3.4.1. Tables showing all annual energy fluxes for these cases are provided in Appendix H.

The ice hall is almost self-sufficient in terms of heat (99 %) even with smaller TES tanks. By increasing the TES tanks for cases 8, 9 and 10, the amounts of annual purchased district heat decrease from 22 to 14 MWh/a (Case 8_{2h}), from 50 to 19 MWh/a (Case 9_{2h}) and from 57 to 19 MWh/a (Case 10_{2h}). The swimming hall has a bigger potential for increasing TES capacities, since swimming hall still purchased about 30 % of its heat demands as district heat for cases 8, 9 and 10. The changes for total utilized waste heats for the cases are presented in Table 40.

Table 40. Changes in total utilized waste heats for cases 8, 9 and 10 with bigger TES tanks in ice and swimming halls.

Case number Unit	8 [MWh/a]	8_{2h} [ΔMWh/a]	9 [MWh/a]	9_{2h} [ΔMWh/a]	10 [MWh/a]	10_{2h} [ΔMWh/a]
Total utilized waste heat in ice hall	1 603	+8	1 506	+32	1 979	+38
Total utilized waste heat in swimming hall	1 965	+45	1 890	+73	2 110	+92
Total purchased district heat in ice and swimming hall	795	-53	921	-105	949	-130

TES tanks with 2 hour discharge time decrease the amount of the total purchased district heat a further 7 %, 11 % and 14 % for cases 8, 9 and 10 respectively (Table 40). Case 9 benefits more from bigger TES tanks than case 8, since more waste heat is unutilized due to different algorithm. Case 10 without summer breaks benefits even more than case 9 due to longer operation period. The amount of purchased district heat stay relatively low, since the smart EAHP is not able to produce enough high temperature waste heat even with the minimum evaporation temperature of 0 °C (maximum power) during the coldest periods of the year. The coldest periods of the year are in average longer than the discharge time of 2 hours used for the bigger TES tanks. The investment for bigger TES tanks in swimming hall is more cost-effective than in ice hall.

4.7 Annual energy comparison

Table 41 shows annual purchased district heat, electricity and total energies for the swimming hall, the ice hall and the total energy consumption for all the 13 cases. The numbers are rounded to nearest tens. The reference cases 1 and 2 are used as a comparison and are thus not included in Table 41. In addition, the non-worthwhile case with low condensation temperature EAHP (Case 4) is left out. Cases 10 and 10_{2h} are compared to the reference without summer breaks (Case 2) and the rest of the cases are compared to the reference with summer breaks (Case 1). Negative values mean a decrease and positive values mean an increase compared to the reference case.

Table 41. Absolute and relative changes in annual energies compared to the reference cases.

Case [ΔMWh/a]		3	5	6	7	8	9	10	8_{2h}	9_{2h}	10_{2h}
SH	DH	-720	-1 940	0	+50	-1 940	-1 850	-2 060	-1 990	-1 920	-2 150
	EL	+60	+520	0	0	+450	+450	+490	+460	+460	+530
	Tot	-660	-1 430	0	+50	-1 500	-1 390	-1 560	-1 520	-1 460	-1 620
IH	DH	-1 590	-1 590	-60	0	-1 590	-1 560	-2 050	-1 600	-1 590	-2 090
	EL	+90	+90	0	0	+90	+90	+170	+90	+90	+170
	Tot	-1 510	-1 500	-60	0	-1 500	-1 480	-1 880	-1 510	-1 510	-1 920
SH + IH	DH	-2 310	-3 540	-60	+50	-3 540	-3 410	-4 110	-3 590	-3 510	-4 240
	EL	+140	+610	0	0	+540	+540	+660	+550	+540	+700
	Tot	-2 170	-2 930	-60	+50	-3 000	-2 870	-3 450	-3 030	-2 970	-3 540
Case [Δ%]		3	5	6	7	8	9	10	8_{2h}	9_{2h}	10_{2h}
SH	DH	-26 %	-72 %	0 %	+1.7 %	-72 %	-68 %	-70 %	-73 %	-71 %	-73 %
	EL	+4 %	+37 %	0 %	-0.0 %	+32 %	+32 %	+31 %	+33 %	+33 %	+33 %
	Tot	-16 %	-35 %	0 %	+1.1 %	-36 %	-34 %	-34 %	-37 %	-35 %	-36 %
IH	DH	-99 %	-99 %	-3.7 %	0 %	-99 %	-97 %	-97 %	-99 %	-99 %	-99 %
	EL	+9 %	+9 %	-0.2 %	0 %	+9 %	+9 %	+12 %	+9 %	+9 %	+12 %
	Tot	-57 %	-57 %	-2.4 %	0 %	-57 %	-56 %	-53 %	-58 %	-58 %	-54 %
SH + IH	DH	-53 %	-82 %	-1.4 %	+1.1 %	-82 %	-79 %	-81 %	-83 %	-81 %	-84 %
	EL	+6 %	+25 %	-0.1 %	-0.0 %	+22 %	+22 %	+22 %	+23 %	+23 %	+23 %
	Tot	-32 %	-43 %	-0.9 %	+0.7 %	-44 %	-43 %	-42 %	-45 %	-44 %	-44 %
SH = swimming hall, IH = ice hall, DH = district heat, EL = electricity, Tot = total DH + EL											

The purchased district heat in swimming hall decreases significantly more in cases, which include EAHP with high condensation temperature (case 5 and cases 8 onwards). The downside of the EAHP with high condensation temperature is the increased electricity consumption. Most of the waste heat in ice hall does not require additional electricity consumption, since the ice refrigeration and dehumidification are mandatory. This shows in bigger total consumed energy reduction with utilization of all waste heat in ice hall (-53 to -58 %) than in swimming hall (-34 to -37 %). (Table 41)

Cases 6 and 7 include demand response without utilization of waste heat. The annual energies in these cases stay close to reference, since the demand response does not aim on changing the energy mounts. The total decrease in annual purchased district heat due to bigger TES tanks for cases 8_{2h}, 9_{2h} and 10_{2h} are -1.2 %, -2.4 % and -2.6 % respectively, but the values are overwhelmed by the decreases due to utilization of waste heat. (Table 41)

4.8 Cost investment analysis

This Chapter presents the savings and cost investment analysis. Table 42 shows annual purchased district heat, electricity and total energy costs for swimming hall, ice hall and total energy costs for all the 13 cases. The numbers are in thousands of euros and are calculated with hourly energy prices, which include all fees and value-added taxes (VAT). The reference case total energy costs are with summer breaks 463 000 € and without summer breaks 549 000 €.

Table 42. Annual purchased district heat, electricity and total energy costs for swimming hall, ice hall and total energy costs for all the 13 cases. The absolute values are in thousands of euros.

Case [k€]		1	2	3	4	5	6	7	8	9	10	8_{2h}	9_{2h}	10_{2h}
SH	DH	155	165	120	88	58	155	153	54	56	58	52	52	53
	EL	127	143	132	188	172	127	127	164	164	184	164	164	184
	Tot	282	308	252	276	231	282	280	218	220	242	216	216	237
IH	DH	92	112	4	2	2	88	92	2	4	4	1	2	2
	EL	89	128	95	97	97	87	89	97	95	140	97	95	140
	Tot	181	240	99	100	100	176	181	100	99	145	99	97	142
SH + IH	DH	247	278	123	90	60	243	245	56	60	62	53	54	55
	EL	216	271	227	286	270	214	216	261	259	324	261	259	324
	Tot	463	549	351	376	330	457	461	317	319	386	314	313	379
SH = swimming hall, IH = ice hall, DH = district heat, EL = electricity, Tot = total DH + EL														

Table 43 shows the absolute and relative changes in annual energy costs. The reference cases 1 and 2 are used as a comparison. The reference cases and case 4 are not included in Table 43. Cases 10 and 10_{2h} are compared to the reference without summer breaks (Case 2) and the rest of the cases are compared to the reference with summer breaks (Case 1). Negative values mean a decrease and positive values mean an increase compared to the reference case.

Table 43. Absolute and relative changes in annual energy costs compared to the reference cases. The absolute values are in thousands of euros.

Case [Δk€]		3	5	6	7	8	9	10	8_{2h}	9_{2h}	10_{2h}
SH	DH	-35	-97	0	-2	-101	-99	-107	-103	-103	-112
	EL	+5	+45	0	-0	+37	+37	+41	+37	+37	+41
	Tot	-30	-51	0	-2	-64	-62	-67	-66	-65	-71
IH	DH	-88	-90	-4	0	-90	-88	-108	-90	-90	-111
	EL	+6	+8	-2	0	+8	+6	+12	+8	+6	+12
	Tot	**-82**	**-81**	**-5**	**0**	**-81**	**-82**	**-96**	**-82**	**-84**	**-98**
SH + IH	DH	-123	-186	-4	-2	-190	-187	-215	-193	-193	-222
	EL	+11	+54	-2	-0	+45	+43	+53	+45	+43	+53
	Tot	**-112**	**-133**	**-5**	**-2**	**-145**	**-144**	**-162**	**-148**	**-149**	**-169**
Case [Δ%]		3	5	6	7	8	9	10	8_{2h}	9_{2h}	10_{2h}
SH	DH	-23 %	-62 %	0 %	-1.1 %	-65 %	-64 %	-65 %	-66 %	-66 %	-68 %
	EL	+4 %	+36 %	0 %	-0.0 %	+29 %	+29 %	+28 %	+29 %	+29 %	+28 %
	Tot	-11 %	-18 %	0 %	-0.6 %	-23 %	-22 %	-22 %	-23 %	-23 %	-23 %
IH	DH	-96 %	-98 %	-3.9 %	0 %	-98 %	-96 %	-96 %	-98 %	-98 %	-98 %
	EL	+7 %	+9 %	-1.9 %	0 %	+9 %	+7 %	+10 %	+9 %	+7 %	+10 %
	Tot	-45 %	-45 %	-2.9 %	0 %	-45 %	-45 %	-40 %	-45 %	-46 %	-41 %
SH + IH	DH	-50 %	-75 %	-1.4 %	-0.7 %	-77 %	-76 %	-78 %	-78 %	-78 %	-80 %
	EL	+5 %	+25 %	-0.8 %	-0.0 %	+21 %	+20 %	+20 %	+21 %	+20 %	+20 %
	Tot	-24 %	-29 %	-1.1 %	-0.4 %	-31 %	-31 %	-30 %	-32 %	-32 %	-31 %
SH = swimming hall, IH = ice hall, DH = district heat, EL = electricity, Tot = total DH + EL											

The most important values in Table 43 are the change in absolute energy costs for ice hall and for the total energy costs. These values include summer breaks. The ice hall alone is able to save annually between 81 000 and 84 000 € (-45 %). Almost all of the energy savings for ice hall are brought by utilization of waste heat (81 000 €, 97 %). The rest of the energy savings are from demand response of ice refrigeration and bigger thermal energy storages (TES). The demand response of ice refrigeration without utilization of waste heat (Case 6) brings cost saving of 5 000 €, but when combined with utilization of waste heat, the cost saving are reduced to under 1 000 €. This is caused by breaks in waste heat recovery.

Table 43 shows the cumulative energy cost savings for the ice and swimming halls, which are annually between 112 000 (-24 %) (Case 3) and 149 000 € (-32 %) (Case 9_{2h}). The breakdown of energy cost savings for smart control measures and utilization of waste heat (Case 9_{2h}) are shows visually in Figure 71. The biggest portion, after utilizing waste heat in ice hall, is the utilization of waste heat from ice hall to swimming hall, which brings annual savings of 30 000 €. Using an EAHP with high condensation temperature of constant +60 °C brings annual savings of 21 000 € and adding the smart control an additional 6 000 €. The demand response of district heat in swimming hall brings annual savings of 5 000 € and using bigger TES tanks increases the savings by an additional 3 000 €. Without summer breaks, the annual savings would increase from 149 000 € (Case 9_{2h}) to 169 000 € (Case 10_{2h}).

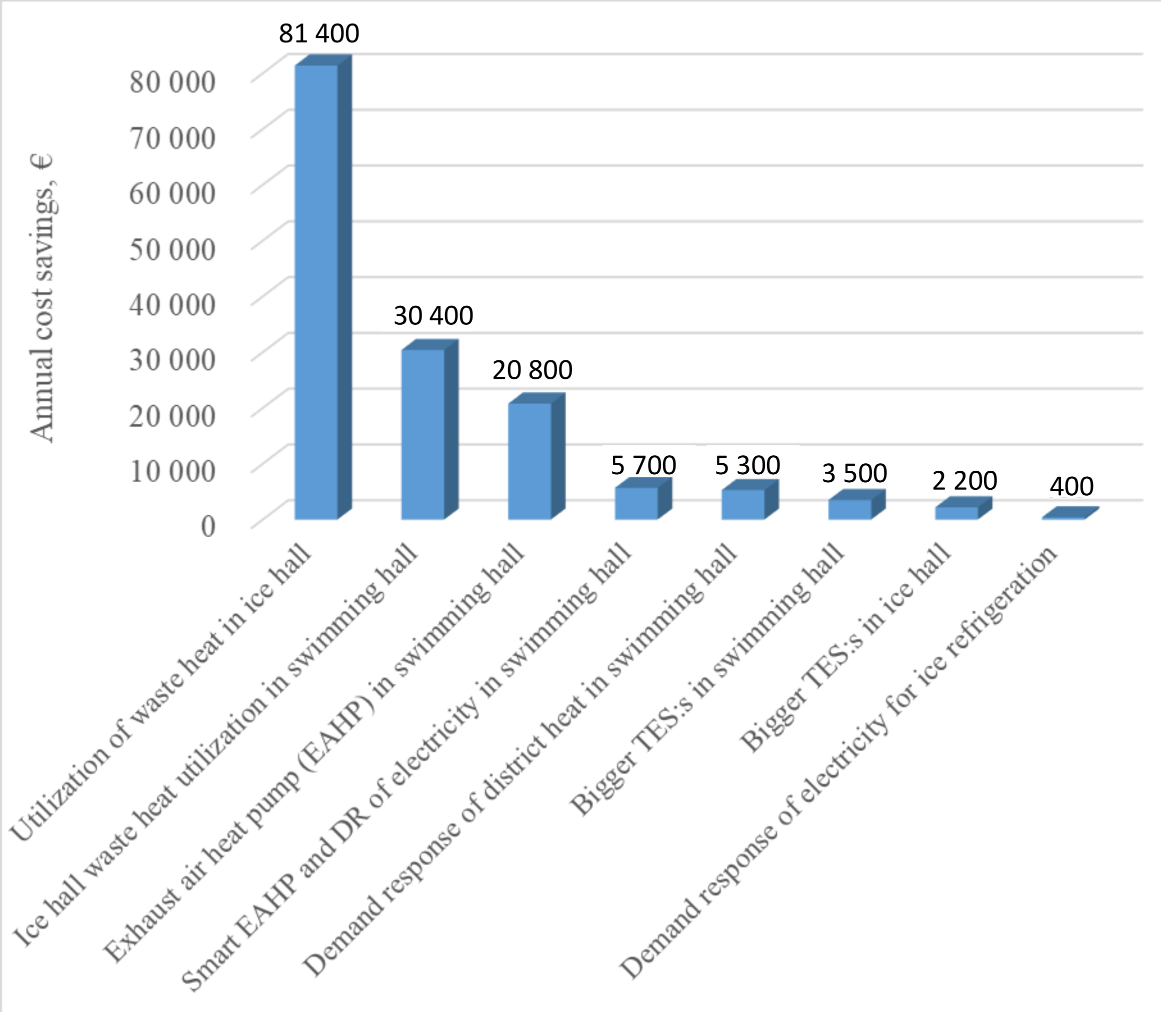

Figure 71. Breakdown of energy cost savings for smart control measures and utilization of waste heat in ice and swimming hall (Case 9_{2h}).

Table 44 presents the total energy cost savings for the three repayment periods, which are the same as the maximum cost of profitable investments. The methodology of the calculation is presented in Chapter 3.6. Swimming hall is studied separately only for demand response of district heat, which brings annual savings of 5 000 € (Case 7). Other cases assume utilization of waste heat from ice hall to swimming hall.

Table 44. The maximum cost of profitable investment in thousands of euros for three repayment periods.

Case [Δk€]		3	5	6	7	8	9	10	8_{2h}	9_{2h}	10_{2h}
IH	7 a	510	510	30	0	510	510	590	510	520	610
	10 a	700	690	40	0	690	700	810	700	720	840
	15 a	970	970	60	0	970	970	1 140	980	1 000	1 170
SH + IH	7 a	700	820	30	10	900	890	1 010	920	930	1 050
	10 a	950	1 130	40	10	1 240	1 220	1 380	1 260	1 270	1 440
	15 a	1 330	1 580	60	20	1 730	1 710	1 930	1 760	1 780	2 010
SH = swimming hall, IH = ice hall											

The investment is profitable, if the cost of the investment is lower than the achieved total energy cost savings. The maximum cost of investment is for the ice hall between 500 000 € and 1 000 000 € depending mainly on repayment period. For the combined ice and swimming halls the maximum cost of investment is between 700 000 € and 2 000 000 €, depending on repayment period and used measures. These values include utilization of waste heat and summer breaks. The total energy cost savings for demand response only are for ice refrigeration between 30 000 and 60 000 € and for demand response of district heat in swimming hall between 10 000 and 20 000€. (Table 44)

5 Discussion

New ice halls are built in Finland an average of 5 per year and 100 are to be renovated in the near future according to Laitinen et al. (2014:5). The amount and age profile of Finnish swimming halls is similar to ice halls and thus is the amount of upcoming swimming hall renovations. Average of 2 new swimming halls per year have been built in the last 10 years (Jyväskylä University, 2018b). This high amount of building and renovating of these type of buildings in Finland create many opportunities to implement utilization of waste heat and smart energy systems for ice and swimming halls.

Recommendations

Utilization of heat from ice refrigeration in an ice hall reduces almost all of the need for purchased district heat for a training ice hall with low indoor temperature, while the increase in electricity consumption is marginal. The investment for the utilization of heat from ice refrigeration seems to be profitable especially in new ice halls. Utilization of heat from ice refrigeration should be set as a regulation for new ice halls, since it alone about halves the total energy consumption of an ice hall with a relatively small investment cost.

An ice hall has excess waste heat even after utilization in the ice hall. A portion of this excess heat can be utilized in a nearby swimming hall for underfloor heating or preheating of DHW or supply air. However, swimming halls have a lot of potential heat in exhaust air and sewage water, which can also be recovered and used to reduce heat demands of the swimming hall, without the need for extra piping from an ice hall to a swimming hall. However, if the temperature of the waste heat is increased with a heat pump close to +60 °C, the waste heat can be utilized to all of swimming hall heat demands. The full heat demand of a swimming hall is so big, that both waste heat sources from the ice and swimming halls can be utilized. This utilization of all waste heat sources in Pirkkola swimming hall reduces the total energy consumption of the combined energy system up to 43 %. Based on this thesis, I would recommend the utilization of all waste heat sources by combining the energy systems of upcoming closely located ice and swimming halls.

From the combined energy system of the Pirkkola ice and swimming halls, about 800 MWh/a (20 %) of all waste heat is left as a potential excess waste heat, due to alternating heat demands and waste heats. This amount of heat could heat 27 000 m^2 of modern building area with average annual heat demand of 30 $kWh/m^2/a$. However, a long-term thermal energy storage would be needed, since most of the excess waste heat is clustered to period during August and September.

The demand response systems in ice and swimming halls have a small investment cost, since the adjusting is done based on temperature set points with specific algorithms. Thus, demand response is possibly profitable even with the relatively low energy cost savings, which are less than tenth of the energy cost savings achieved with utilization of waste heat.

Reliability of the results

The reliability of the results is affected by inaccuracy of the used parameters, the assumptions made, the simplifications in the simulation model. However, the results were compared to similar buildings or studies of similar buildings. In addition, in the post-processing of utilization of waste heat, the energy balances were verified to hold true with a maximum error of about 5 %. Thus, at least the magnitude of the results can be held reliable.

Future research topics

The topic of utilization of waste heat in combined ice and swimming halls was studied in this thesis thoroughly for a new training ice hall with two ice rinks and an old medium sized swimming hall. However, the results are not to be generalized for different size of halls, for different climate or for a different age of ice or swimming halls. New calculations are needed for each case with different baselines.

The topic of demand response in ice and swimming halls is not previously studied so this research has a significant novelty value. Since there are no earlier studies on the topic, the results of this study should be compared to future studies. The algorithms of this thesis were not optimized, and thus the potential energy cost saving could be bigger than what is presented in this thesis. The future research could be focused on finding more optimal demand response methods and algorithms for ice and swimming halls. Especially the significant thermal capacity of swimming pool water yields a significant potential for demand response, which should be researched more.

Accuracy of power control of energy systems of ice refrigeration and pool water heating should be improved in the future simulation studies to achieve more saving potential for demand response control.

In addition, demand response of district heat in not yet implemented in Finland and for it to occur, more research and co-operation between different companies in fields of energy production and consumption are needed.

6 Conclusions

The objectives of this thesis were to analyze the potential reduction of energy consumption and energy cost savings with waste heat recovery and smart control of energy systems in the combined energy system of the ice and swimming halls. This thesis studied a case in Pirkkola (Helsinki), which includes an old existing swimming hall and a new training ice hall. Measures for smart control of energy systems in this thesis consisted of demand response of electricity and district heat and a demand based control of exhaust air heat pump. In addition, the combined effect of waste heat recovery, short-time storing of waste heat and demand response were evaluated. This study was carried out by dynamic simulations and post-processing of the simulation results. The models of the ice and swimming halls, the ventilation and the component models that were not available were built in this study.

Utilization of the waste heat from ice refrigeration in the ice hall is the most profitable and the most suggested investment found in this thesis. Including condensing and gray water heat recoveries in the ice hall, the total annual savings are 81 000 €, which is 45 % of the ice hall energy costs. In the Pirkkola ice hall, up to 99 % of purchased district heat can be replaced by waste heat mainly from ice refrigeration, while the total electricity demand in the ice hall increases only by 9 %. Thus, the ice hall can be self-sufficient in terms of heat.

The swimming hall was not studied separately, since the swimming hall does not have any source of waste heat, which could be utilized in the swimming hall without the need of additional electricity consumption of a heat pump. One focus of utilization of waste heat in this thesis was transferring excess heat from the ice hall to the swimming hall. By transferring the excess heat and utilizing the waste heats of the swimming hall, the total purchased district heat in the swimming hall is reduced by 72 % and the electricity consumption is increased by 37 % resulting in a total reduction of consumed energy of 35 % in the swimming hall. Utilization of the excess heat from the ice hall to the swimming hall is the second most profitable investment bringing annual savings of 30 000 €.

In the combined energy system of the ice and swimming halls, up to 81 % (3540 MWh/a) of the waste heat can be utilized. Most of the utilized waste heat in the swimming hall is from an exhaust air heat pump (EAHP) with high condensation temperature and the smaller portions are from the excess heat from the ice hall and from the heat recovery of gray water in the swimming hall. The EAHP has the biggest portion, because it is the only waste heat able to supply most of the high temperature heat demands in the swimming hall. The changes in the annual energy consumptions of the combined energy system are 82 % reduced purchased district heat, 25 % increased electricity and 42 % reduced total consumed energy. The annual energy cost savings the EAHP brings is 21 000 €.

Demand response of electricity for ice refrigeration decreases the total electricity costs of the ice hall by 1.9 %. The low thermal capacity of the ice limited the demand response potential. In addition, the amount of waste heat utilized in the ice hall was decreased due to demand response.

Demand response of district heat for swimming pool water and pool space air decreases the average price of purchased district heat in the swimming hall by 2.8 % and the total district heat costs in the swimming hall by 1.1 %. Demand response of electricity for heating of pool space water and indoor air with heat from EAHP did not decrease the total energy costs.

Demand response of electricity for saunas in the swimming hall did not decrease the total energy costs and resulted in rapid temperature decreases due to big heat losses when the temperature set point was lowered. Thus, saunas are not suitable for loading strategy of demand response, but are suitable for peak clipping strategy of demand response.

The increase in TES capacities allows more waste heat to be utilized, which decreases the amount of purchased district heat. In utilization of waste heat, TES tanks with discharge time of 2-hour compared to 30 minutes decrease the annual purchased district heat in the ice and swimming halls up to 2.6 %.

With all the measures for the ice and swimming halls the annual energy cost savings are 149 000 € (-32 %). The total energy cost savings during the lifecycle of the systems equal the maximum cost of profitable investment, which are for the ice hall between 500 000 and 1 000 000 € and for the combined ice and swimming halls between 700 000 and 2 000 000 €.

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Appendices

Appendix A. Calculation for value mentioned in introduction

Energy consumption of ice and swimming halls in Finland of total energy consumption of Finnish building sector is

$$p = \frac{N_{IH} * Q_{avg,IH,a} + N_{SH} * Q_{avg,SH}}{Q_{BS}} \tag{A.1}$$

where

N_{IH}	amount of ice halls in Finland (Hemmilä & Laitinen, 2018:8)
$Q_{avg,IH,a}$	average annual energy consumption of an ice hall in Finland (Jyväskylä University, 2018a) [MWh/a]
N_{SH}	amount of swimming halls in Finland (Hemmilä & Laitinen, 2018:8)
$Q_{avg,SH}$	average annual energy consumption of a swimming hall in Finland (Jyväskylä University, 2018b) [MWh/a]
Q_{BS}	total annual energy consumption of Finland building sector in 2016 (SVT, 2017) [MWh/a].

Appendix B. Two example views of IDA ICE advanced level interface

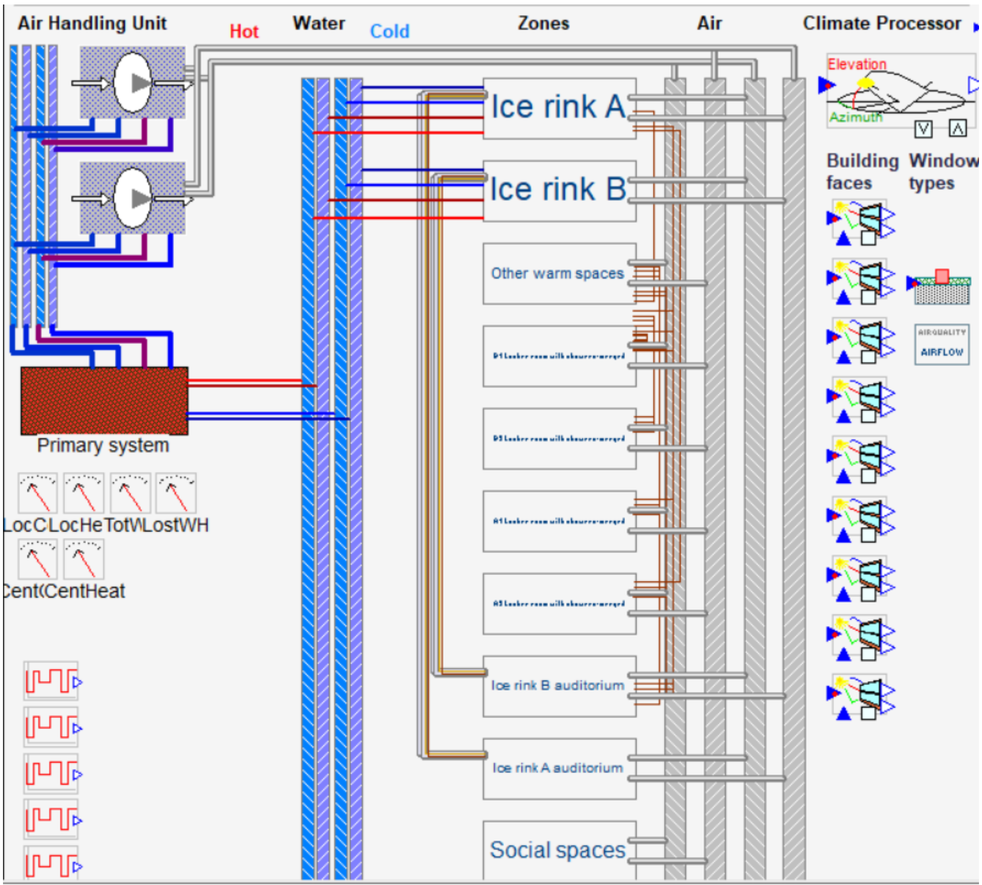


Figure B.1. View of ice hall schematic.

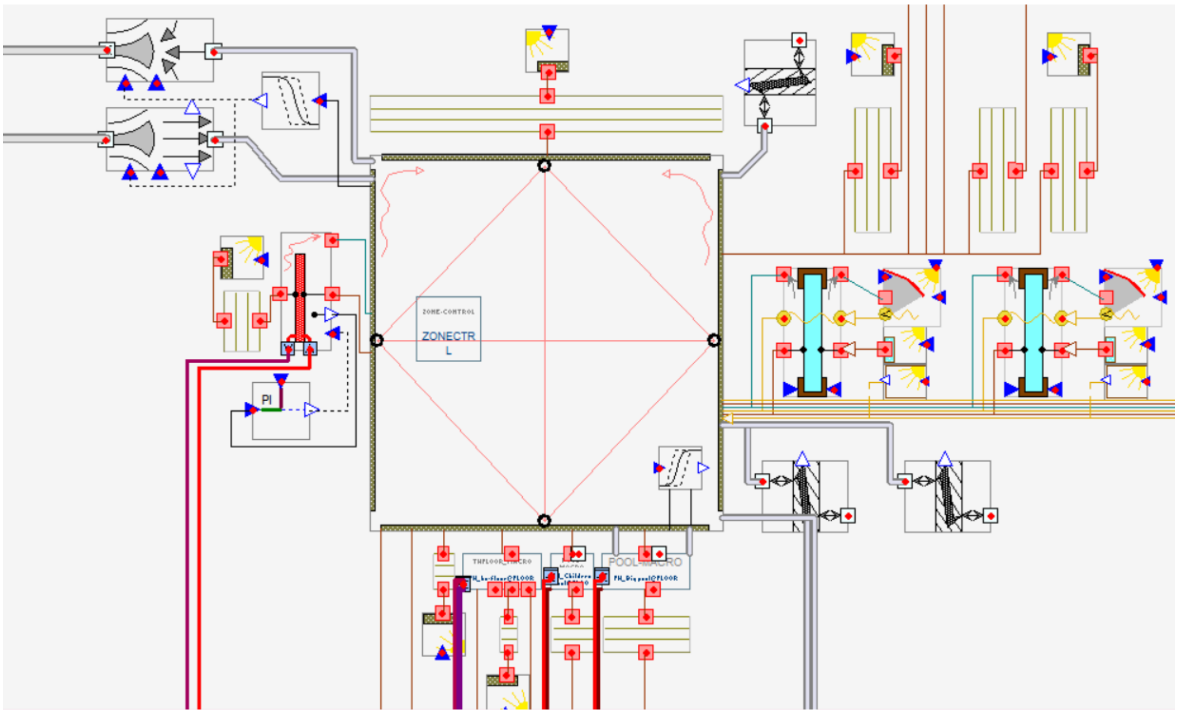


Figure B.2. View of main pool space in swimming hall.

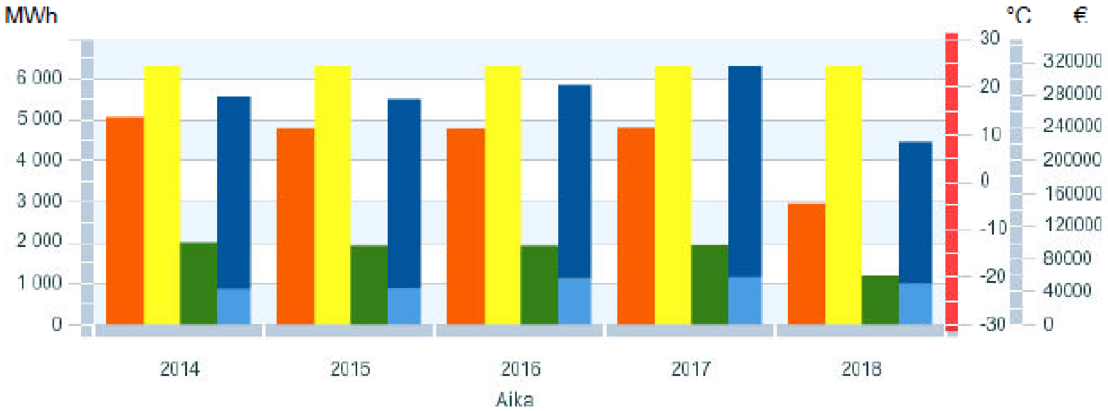
Appendix C. Energy reports for Pirkkola and calculation of the energy consumption division between Pirkkola ice and swimming halls

HELEN

Kaukolämmön seuranta

Käyttöpaikan numero: 4681
Pirkkolan Liikuntapuisto, 00630 Helsinki

Raportti ajanjaksolle: Käyttö 2014 - 2018
Raportti luotu: 4.10.2018 klo 08:39:05



Käyttö
Summa: 22 385,40 MWh
Keskiarvo: 4 477,08 MWh
Minimi: 2 946,28 MWh (2018)
Maksimi: 5 059,27 MWh (2014)

Ennuste
Summa: 31 650,00 MWh
Keskiarvo: 6 330,00 MWh
Minimi: 6 330,00 MWh (2014)
Maksimi: 6 330,00 MWh (2014)

Primäärienergia

Vesivirtamaksu

Energiamaksu

Figure C.1. Report for annual consumption of district heat in Pirkkola ice and swimming halls.

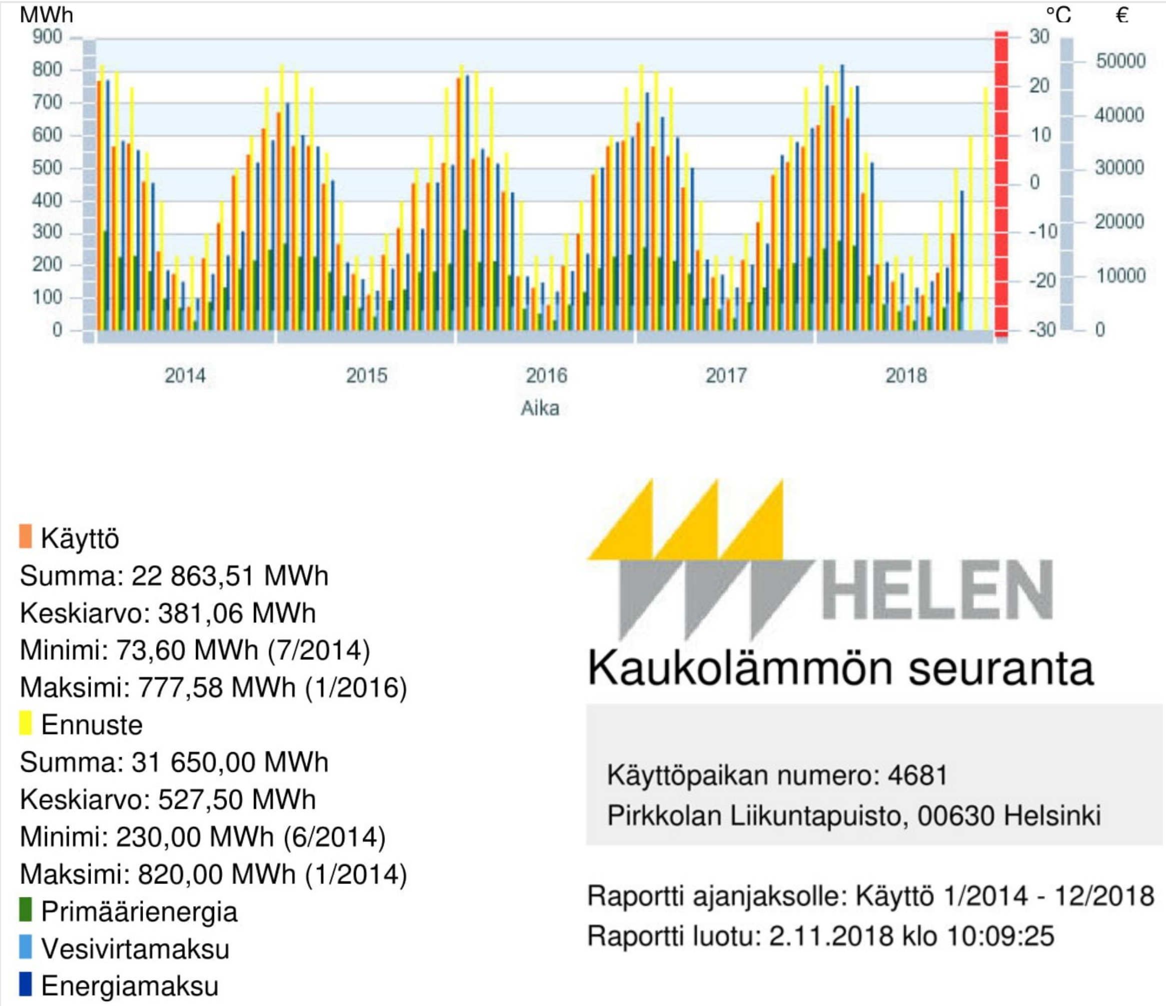


Figure C.2. Report for monthly consumption of district heat in Pirkkola ice and swimming halls.

The latest monthly district heat consumptions are from months September and October 2018 (Figure C.2). The ice hall has been closed April 2018 and thus, the consumption of these months is assumed to come fully from the swimming hall. To get the approximation for district heating in the swimming hall only of the total consumption $p_{DH,SH}$, the consumption of these months (9 and 10) is compared to consumption of the same months in the year 2017 as follows

$$p_{DH,SH} = \frac{Q_{DH,2018,9} + Q_{DH,2018,10}}{Q_{DH,2017,9} + Q_{DH,2017,10}} = \frac{(175 + 300)MWh}{(350 + 490)MWh} = 56.6\%. \qquad \text{(C.1)}$$

Figure C.1 shows the annual district heat consumption to be about 4850 MWh/a. By multiplying this consumption by the percentage calculated in Equation C.1, the consumptions acquired are for the swimming hall 2700 MWh/a and for the ice hall 2100 MWh/a.

Figure C.3. Report for annual electricity consumption in Pirkkola ice and swimming halls.

The total monthly electricity consumption are provided as an excel Table (Korva, 2018). The approximation for electricity in the swimming hall only is done in the same way as with district heat. The total electricity consumption of the swimming hall $p_{EL,SH}$ is

$$p_{EL,SH} = \frac{Q_{EL,2018,9} + Q_{EL,2018,10}}{Q_{EL,2017,9} + Q_{EL,2017,10}} = \frac{(161 + 175)MWh}{(322 + 334)MWh} = 51.0\%. \qquad \text{(C.2)}$$

Figure C.3 shows the annual district heat consumption to be about 3200 MWh/a. By multiplying this consumption by the percentage calculated (Equation C.2), the consumptions acquired are for the swimming hall 1600 MWh/a and for the ice hall 1600 MWh/a.

Appendix D. Schematic of DHW system in Pirkkola

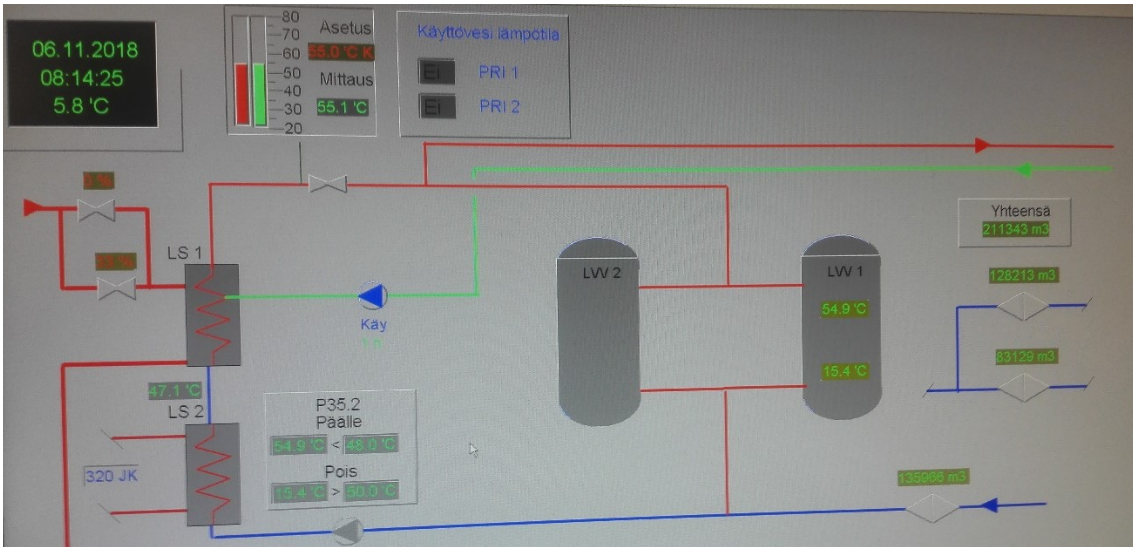


Figure D.1. View of control panel for DHW system in the Pirkkola ice and swimming halls.

Appendix E. Schematic of air handling unit in the main Pirkkola swimming hall space

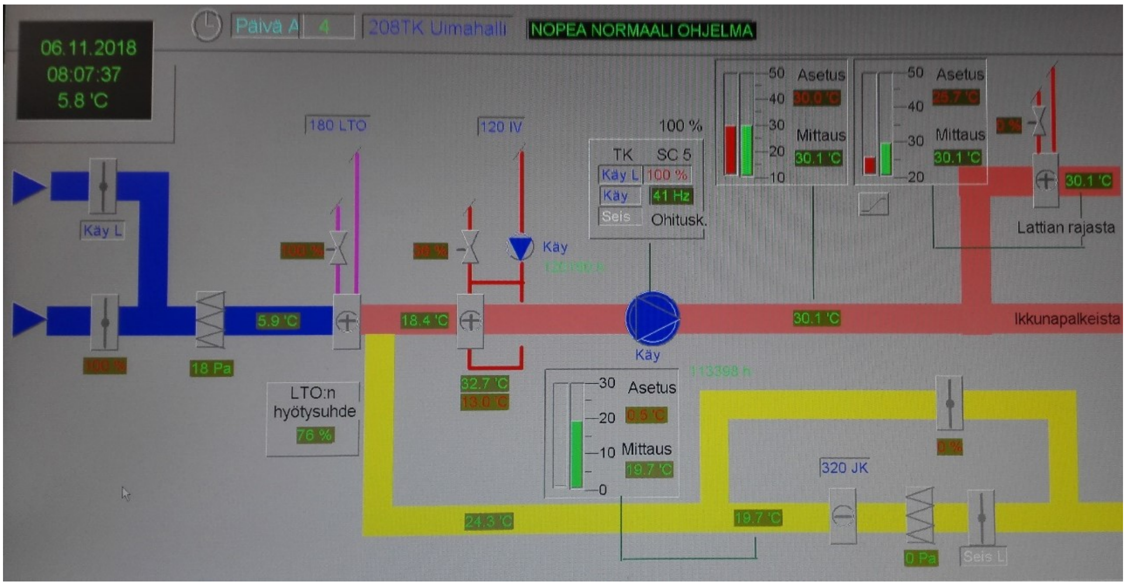


Figure E.1. Schematic of air handling unit in the main Pirkkola swimming hall space (1/2).

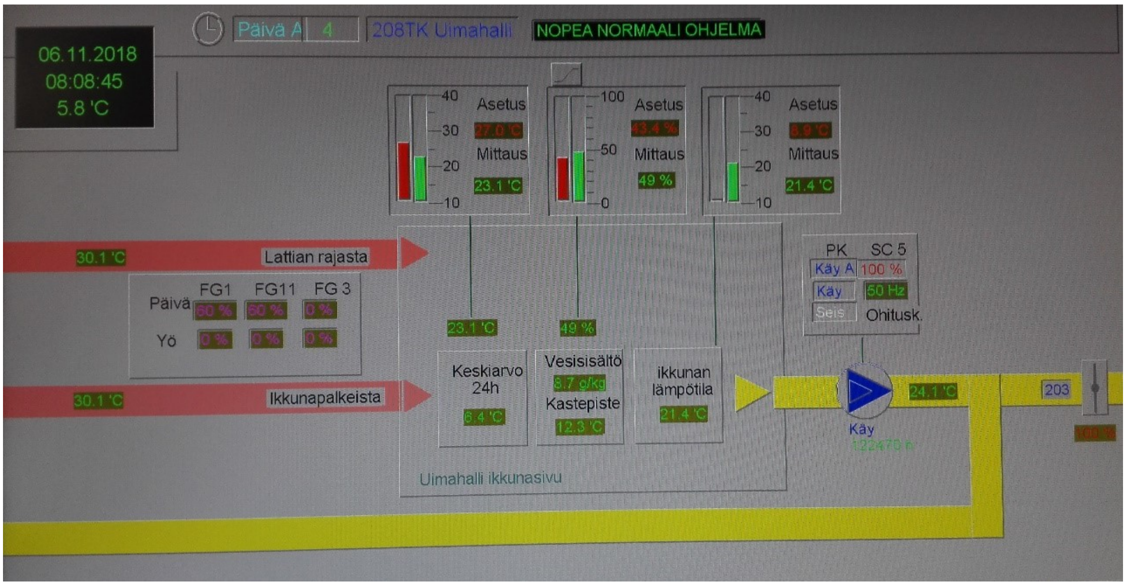


Figure E.2. Schematic of air handling unit in the main Pirkkola swimming hall space (2/2).

Appendix F. Efficiency of gray water heat exchangers

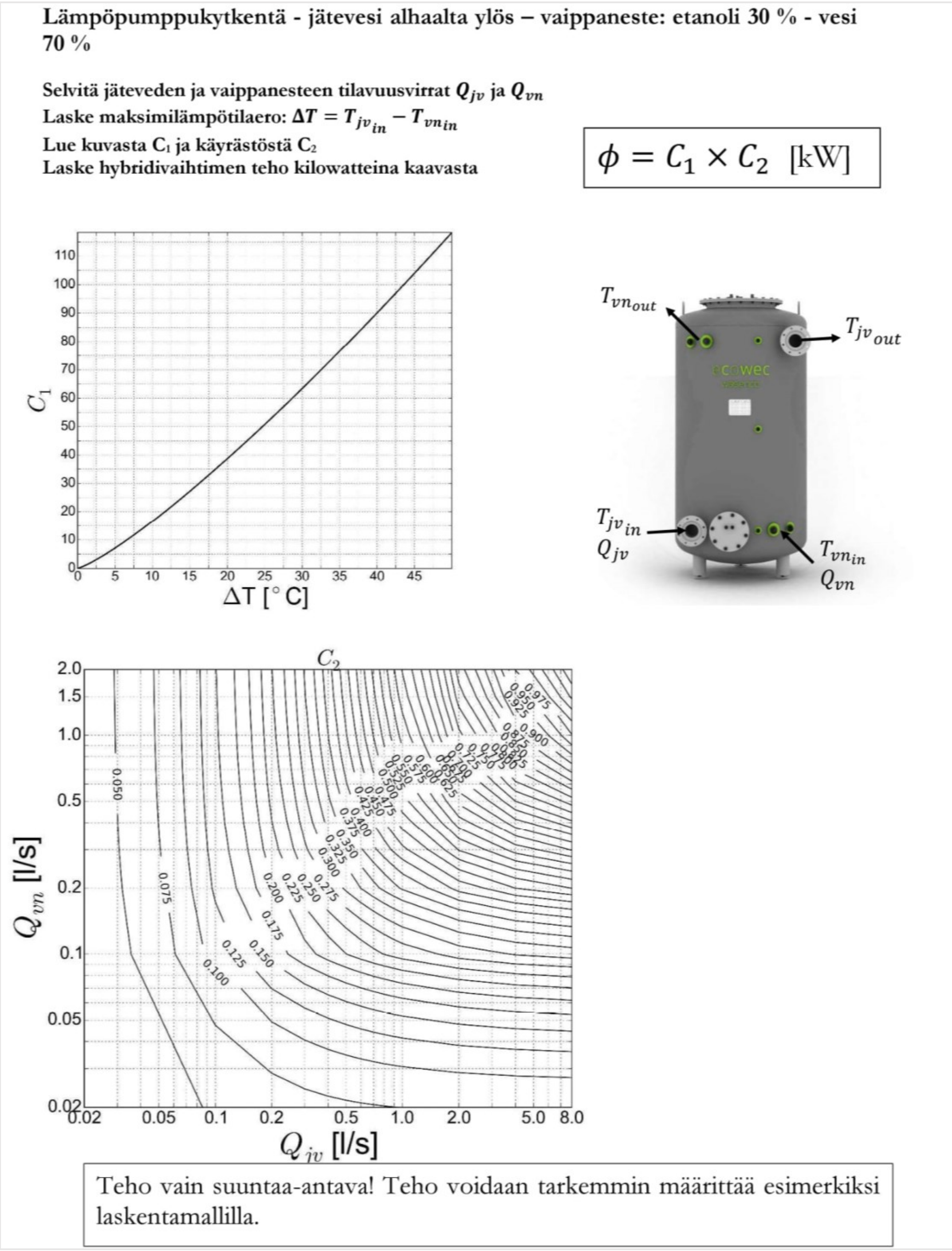


Figure F.1. Power dimensioning guide for gray water heat exchanger, which is combined to a heat pump with coolant (Ecowec, 2018).

Efficiency of gray water heat exchanger in the ice hall

The power dimensioning is done according to guide shown in Figure F.1. The ice hall is dimensioned with one heat exchanger. The average gray water flow in the ice hall is

$$q_{DHW} = \frac{V_{DHW,d}}{t_{sh} \cdot n_{g,IR}} = \frac{12 \frac{m^3}{d} \cdot 1000 \frac{l}{m^3}}{2000\ s \cdot 9.5 \frac{1}{d}} = 0.63\ \frac{l}{s} \qquad \text{(F.1)}$$

where

q_{DHW} average domestic hot water (DHW) flow during shower usage [l/s]
$V_{DHW,d}$ average DHW usage per day [m^3/d]
t_{sh} average time, during which the showers are taken after a game [s]
$n_{g,d,IR}$ average number of games per day per ice rink [1/d].

The coolant flow in the heat exchanger is assumed the same as the gray water flow. The minimum temperature of the coolant is assumed 0 °C and the input temperature of gray water is assumed 10 °C lower than the average used water temperature of 38 °C. Thus, the temperature difference is 28 °C. The average gray water flow and the temperature difference give according to Figure F.1 the average power for the heat exchanger

$$\phi_{he} = C_1 \cdot C_2 = 60\ kW \cdot 0.44 = 26.4\ kW. \qquad \text{(F.2)}$$

The power used for heating of the DHW for a time step, during which the showers are taken, is received from IDA ICE simulations. The heat exchanger efficiency is defined as the relation of heat recovered from the gray water to the heating of DHW as follows

$$\eta_{he} = \frac{\phi_{he}}{P_{DHW}} = \frac{26.4\ kW}{77.5\ kW} = 34\ \%. \qquad \text{(F.3)}$$

Efficiency of gray water heat exchangers in the swimming hall

The swimming hall is dimensioned with two heat exchanger due to high gray water flow. The heat gray water drained to sewers consists of only shower water and washing water, which is 67 % of total swimming hall DHW usage according to Hemmilä and Laitinen (2018:13). The average gray water flow in the sewers in the swimming hall is

$$q_{DHW,he} = \frac{V_{DHW,d} \cdot p_{sh,ww}}{t_{open,d} \cdot N_{he}} = \frac{51 \frac{m^3}{d} \cdot 1000 \frac{l}{m^3} \cdot 0.67}{15.5\ h \cdot 3600 \frac{s}{h} \cdot 2} = 0.31\ \frac{l}{s} \qquad \text{(F.4)}$$

where

$q_{DHW,he}$ average DHW flow in one heat exchanger during opening times [l/s]
$V_{DHW,d}$ average DHW usage per day [m^3/d]
$p_{sh,ww}$ portion of shower water and washing water of total DHW
$t_{open,d}$ daily opening time [s]
N_{he} number of heat exchangers.

The coolant flow in the heat exchanger is assumed the same as the gray water flow. The minimum temperature of the coolant is assumed 0 °C and the input temperature of gray water is assumed 10 °C lower than the average used water temperature of 37 °C. Thus, the temperature difference is 27 °C. The average gray water flow and the temperature difference give according to Figure F.1 the average power of the heat exchangers during opening times are

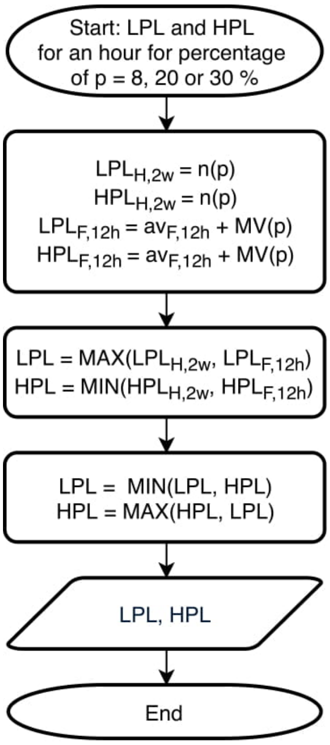
$$\phi_{he} = C_1 \cdot C_2 \cdot N_{he} = 57\ kW \cdot 0.30 \cdot 2 = 34.2\ kW. \tag{F.5}$$

The average heating of DHW in the swimming hall during opening times is received from the IDA ICE simulations. The heat exchanger efficiency in this thesis is defined as the relation of heat recovered from the gray water to the heating of DHW as follows

$$\eta_{he} = \frac{\phi_{he}}{P_{DHW},avg} = \frac{34.2\ kW}{189.6\ kW} = 18\ \%. \tag{F.6}$$

Appendix G. Calculation process for dynamic energy price limits

Higher price limit (HPL) and lower price limit (LPL) are defined for each hour based on two limiting values, which are the last 2 weeks of history data and the following 12-hour data. The prices are known 12 hours upfront. The calculation process of the algorithm for price limits is shown in Figure G.1.



Notation:

p	percentage, which each limiting value excludes
$av_{F,12h}$	average price of following 12-hour energy prices
MV(p)	marginal value, which excludes the corresponding percentage (p), when added to the average of following 12-hour energy prices ($av_{F,12h}$)
n(p)	n:th value from an order of magnitude of the last 2 weeks of history prices, which excludes the corresponding percentage of energy prices (p)
$LPL_{H,2w}$	limiting value for lower price limit from last 2 weeks of history data
$LPL_{F,12h}$	limiting value for lower price limit from following 12-hour data
LPL	final lower price limit
$HPL_{H,2w}$	limiting value for higher price limit from last 2 weeks of history data
$HPL_{F,12h}$	limiting value for higher price limit from following 12-hour data
HPL	final higher price limit

Figure G.1. The calculation process of algorithm for higher price limit (HPL) and lower price limit (LPL) for each hour.

This paragraph explains Figure G.1. The limiting value from the last 2 weeks of history data is chosen as the value from an order of magnitude of the history prices, which excludes the corresponding percentage of energy prices. The limiting value from following 12-hour prices is calculated by adding a chosen constant marginal value (positive or negative) to the average value of the following 12-hour prices, which excludes the corresponding percentage. The marginal value can be approximated for example with history data. The price limits HPL and LPL are a combination of these two limiting values; for LPL the bigger value is chosen and for HPL the smaller value is chosen. Finally, LPL and HPL are prevented from exceeding each other.

Appendix H. Tables showing all annual energy fluxes for cases from 6 onwards

Table H.1. Annual energies for the ice and swimming halls with demand response of electricity for ice refrigeration (Case 6).

Energy flows [MWh/a]. Ice hall floor area 6674 m²		
Energy balance 1: Building	Heat energy of systems	1 554
	Heat loads from electricity	244
	Other heat loads	353
	Heat losses	-2 150
	Total removed heat for use	0
Energy balance 2: Technical systems	Ice refrigeration heat	1 568
	Ice refrigeration HP electricity	581
	Gray water heat	0
	Gray water HP electricity	0
	Dehumidification heat	207
	Dehumidification electricity	68
	Supply air heating	0
	DHW heating	0
	Ice resurfacing water	0
	Water radiator heating	0
	Ground frost heating	0
	Excess heat to swimming hall	0
Total values	Utilized waste heat	0
	Purchased district heat	1 554
	Purchased electricity	1 004
Breakdown of heat energy of systems	Supply air heating	1 240
	DHW heating	191
	Ice resurfacing water freezing	42
	Water radiator heating	24
	Ground frost heating	58
Breakdown of heat loads (+) and losses (-)	Ice resurfacing water freezing	155
	Lighting	174
	Occupants	90
	Ventilation fans	69
	DHW sewage losses	-191
	DH substation losses	-47
	Infiltration air	-31
	Envelope	-46
	Other heat loads(+) / losses(-)	-1 801

Combined energy flows [MWh/a]		
Energy balance 5: Total	Total purchased district heat	4 271
	Total purchased electricity	2 410
	Total heat loads	718
	Total heat losses	-7 303
	Total potential excess heat	0

Notation: SH = Swimming Hall, HP = Heat Pump

Energy flows [MWh/a]. SH floor area 7982m²		
Energy balance 3: Building	Heat energy of systems	2 717
	Heat loads from electricity	1 445
	Other heat loads	365
	Heat losses	-4 487
	Total removed heat for use	0
Energy balance 4: Technical systems	Heat from ice hall	0
	Exhaust air HP heat	0
	Exhaust air electricity	0
	Gray water heat	0
	Gray water HP electricity	0
	Supply air preheating	0
	Supply air heating	0
	Water radiator heating	0
	Underfloor heating	0
	DHW heating	0
	DHW preheating	0
	Pool water heating	0
	Potential excess heat	0
Total values	Utilized waste heat	0
	Purchased district heat	2 717
	Purchased electricity	1 404
Breakdown of waste heat	Utilized ice hall waste heat	0
	Utilized SH waste heat	0
	Excess heat from TES 1	0
	Excess heat from TES 2	0
Breakdown of heat energy of systems	Supply air heating	322
	Supply air preheating	292
	Pool water heating	669
	Water radiator heating	348
	Underfloor heating	110
	DHW preheating	540
	DHW heating	436
Breakdown of heat loads (+) and losses (-)	Sauna	527
	Lighting	177
	Ventilation fans	119
	Pool pumps	437
	Equipment	89
	HVAC aux.	96
	Occupants	300
	Radiation through windows	65
	DHW sewage losses	-976
	DH substation losses	-82
	Infiltration air	-92
	Envelope	-335
	Other heat loads(+) / losses(-)	-3 263

Table H.2. Annual energies for the ice and swimming halls with demand response of district heat for the swimming pool water and pool space air heating (Case 7).

Energy flows [MWh/a]. Ice hall floor area 6674 m²		
Energy balance 1: Building	Heat energy of systems	1 554
	Heat loads from electricity	244
	Other heat loads	355
	Heat losses	-2 212
	Total removed heat for use	0
Energy balance 2: Technical systems	Ice refrigeration heat	1 593
	Ice refrigeration HP electricity	578
	Gray water heat	0
	Gray water HP electricity	0
	Dehumidification heat	221
	Dehumidification electricity	73
	Supply air heating	0
	DHW heating	0
	Ice resurfacing water	0
	Water radiator heating	0
	Ground frost heating	0
	Excess heat to swimming hall	0
Total values	Utilized waste heat	0
	Purchased district heat	1 614
	Purchased electricity	1 007
Breakdown of heat energy of systems	Supply air heating	1 240
	DHW heating	191
	Ice resurfacing water freezing	42
	Water radiator heating	24
	Ground frost heating	58
Breakdown of heat loads (+) and losses (-)	Ice resurfacing water freezing	155
	Lighting	174
	Occupants	90
	Ventilation fans	69
	DHW sewage losses	-118
	DH substation losses	-48
	Infiltration air	-37
	Envelope	-56
	Other heat loads(+) / losses(-)	-1 842

Combined energy flows [MWh/a]		
Energy balance 5: Total	Total purchased district heat	4 378
	Total purchased electricity	2 411
	Total heat loads	718
	Total heat losses	-7 409
	Total potential excess heat	0

Notation: SH = Swimming Hall, HP = Heat Pump

Energy flows [MWh/a]. SH floor area 7982m²		
Energy balance 3: Building	Heat energy of systems	2 765
	Heat loads from electricity	1 445
	Other heat loads	362
	Heat losses	-4 531
	Total removed heat for use	0
Energy balance 4: Technical systems	Heat from ice hall	0
	Exhaust air HP heat	0
	Exhaust air electricity	0
	Gray water heat	0
	Gray water HP electricity	0
	Supply air preheating	0
	Supply air heating	0
	Water radiator heating	0
	Underfloor heating	0
	DHW heating	0
	DHW preheating	0
	Pool water heating	0
	Potential excess heat	0
Total values	Utilized waste heat	0
	Purchased district heat	2 765
	Purchased electricity	1 404
Breakdown of waste heat	Utilized ice hall waste heat	0
	Utilized SH waste heat	0
	Excess heat from TES 1	0
	Excess heat from TES 2	0
Breakdown of heat energy of systems	Supply air heating	338
	Supply air preheating	291
	Pool water heating	706
	Water radiator heating	353
	Underfloor heating	100
	DHW preheating	540
	DHW heating	436
Breakdown of heat loads (+) and losses (-)	Sauna	526
	Lighting	177
	Ventilation fans	119
	Pool pumps	437
	Equipment	89
	HVAC aux.	96
	Occupants	300
	Radiation through windows	62
	DHW sewage losses	-715
	DH substation losses	-84
	Infiltration air	-94
	Envelope	-336
	Other heat loads(+) / losses(-)	-3 302

Table H.3. Annual energies for the ice and swimming halls with demand response of electricity for the swimming hall, utilization of waste heat and smart EAHP (Case 8).

Energy flows [MWh/a]. Ice hall floor area 6674 m²		
Energy balance 1: Building	Heat energy of systems	1 614
	Heat loads from electricity	244
	Other heat loads	355
	Heat losses	-259
	Total removed heat for use	-1 953
Energy balance 2: Technical systems	Ice refrigeration heat	1 593
	Ice refrigeration HP electricity	578
	Gray water heat	64
	Gray water HP electricity	16
	Dehumidification heat	221
	Dehumidification electricity	73
	Supply air heating	-1 266
	DHW heating	-182
	Ice resurfacing water	-42
	Water radiator heating	-23
	Ground frost heating	-54
	Excess heat to swimming hall	-950
Total values	Utilized waste heat	1 603
	Purchased district heat	22
	Purchased electricity	1 098
Breakdown of heat energy of systems	Supply air heating	1 304
	DHW heating	191
	Ice resurfacing water freezing	42
	Water radiator heating	23
	Ground frost heating	54
Breakdown of heat loads (+) and losses (-)	Ice resurfacing water freezing	155
	Lighting	174
	Occupants	90
	Ventilation fans	69
	DHW sewage losses	-118
	DH substation losses	-48
	Infiltration air	-37
	Envelope	-56
	Other heat loads(+) / losses(-)	110

Combined energy flows [MWh/a]		
Energy balance 5: Total	Total purchased district heat	795
	Total purchased electricity	2 948
	Total heat loads	722
	Total heat losses	-3 558
	Total potential excess heat	-696

Notation: SH = Swimming Hall, HP = Heat Pump

Energy flows [MWh/a]. SH floor area 7982m²		
Energy balance 3: Building	Heat energy of systems	2 738
	Heat loads from electricity	1 444
	Other heat loads	367
	Heat losses	-3 299
	Total removed heat for use	-1 205
Energy balance 4: Technical systems	Heat from ice hall	950
	Exhaust air HP heat	1 018
	Exhaust air electricity	291
	Gray water heat	188
	Gray water HP electricity	49
	Supply air preheating	-233
	Supply air heating	-270
	Water radiator heating	-317
	Underfloor heating	-62
	DHW heating	-300
	DHW preheating	-417
	Pool water heating	-334
	Potential excess heat	-696
Total values	Utilized waste heat	1 965
	Purchased district heat	773
	Purchased electricity	1 849
Breakdown of waste heat	Utilized ice hall waste heat	550
	Utilized SH waste heat	1 415
	Excess heat from TES 1	520
	Excess heat from TES 2	175
Breakdown of heat energy of systems	Supply air heating	327
	Supply air preheating	293
	Pool water heating	684
	Water radiator heating	360
	Underfloor heating	98
	DHW preheating	540
	DHW heating	436
Breakdown of heat loads (+) and losses (-)	Sauna	518
	Lighting	177
	Ventilation fans	118
	Pool pumps	437
	Equipment	89
	HVAC aux.	104
	Occupants	300
	Radiation through windows	67
	DHW sewage losses	-715
	DH substation losses	-83
	Infiltration air	-92
	Envelope	-332
	Other heat loads(+) / losses(-)	-2 077

Table H.4. Annual energies for the ice and swimming halls with demand response of the ice and swimming halls, utilization of waste heat and smart EAHP (Case 9).

Energy flows [MWh/a]. Ice hall floor area 6674 m²		
Energy balance 1: Building	Heat energy of systems	1 554
	Heat loads from electricity	244
	Other heat loads	353
	Heat losses	-242
	Total removed heat for use	-1 908
Energy balance 2: Technical systems	Ice refrigeration heat	1 568
	Ice refrigeration HP electricity	581
	Gray water heat	64
	Gray water HP electricity	16
	Dehumidification heat	207
	Dehumidification electricity	68
	Supply air heating	-1 205
	DHW heating	-182
	Ice resurfacing water	-42
	Water radiator heating	-24
	Ground frost heating	-58
	Excess heat to swimming hall	-1 006
Total values	Utilized waste heat	1 506
	Purchased district heat	50
	Purchased electricity	1 093
Breakdown of heat energy of systems	Supply air heating	1 240
	DHW heating	191
	Ice resurfacing water freezing	42
	Water radiator heating	24
	Ground frost heating	58
Breakdown of heat loads (+) and losses (-)	Ice resurfacing water freezing	155
	Lighting	174
	Occupants	90
	Ventilation fans	69
	DHW sewage losses	-118
	DH substation losses	-47
	Infiltration air	-31
	Envelope	-46
	Other heat loads(+) / losses(-)	108

Combined energy flows [MWh/a]		
Energy balance 5: Total	Total purchased district heat	921
	Total purchased electricity	2 947
	Total heat loads	717
	Total heat losses	-3 547
	Total potential excess heat	-847

Notation: SH = Swimming Hall, HP = Heat Pump

Energy flows [MWh/a]. SH floor area 7982m²		
Energy balance 3: Building	Heat energy of systems	2 761
	Heat loads from electricity	1 444
	Other heat loads	364
	Heat losses	-3 306
	Total removed heat for use	-1 220
Energy balance 4: Technical systems	Heat from ice hall	1 006
	Exhaust air HP heat	1 032
	Exhaust air electricity	295
	Gray water heat	188
	Gray water HP electricity	49
	Supply air preheating	-220
	Supply air heating	-280
	Water radiator heating	-314
	Underfloor heating	-65
	DHW heating	-301
	DHW preheating	-362
	Pool water heating	-308
	Potential excess heat	-847
Total values	Utilized waste heat	1 890
	Purchased district heat	871
	Purchased electricity	1 854
Breakdown of waste heat	Utilized ice hall waste heat	459
	Utilized SH waste heat	1 431
	Excess heat from TES 1	640
	Excess heat from TES 2	206
Breakdown of heat energy of systems	Supply air heating	336
	Supply air preheating	292
	Pool water heating	698
	Water radiator heating	357
	Underfloor heating	101
	DHW preheating	540
	DHW heating	436
Breakdown of heat loads (+) and losses (-)	Sauna	518
	Lighting	177
	Ventilation fans	118
	Pool pumps	437
	Equipment	89
	HVAC aux.	104
	Occupants	300
	Radiation through windows	64
	DHW sewage losses	-715
	DH substation losses	-84
	Infiltration air	-93
	Envelope	-334
	Other heat loads(+) / losses(-)	-2 080

Table H.5. Annual energies without summer breaks for the ice and swimming halls with demand response of the ice and swimming halls, utilization of waste heat and smart EAHP (Case 10).

Energy flows [MWh/a]. Ice hall floor area 6674 m²		
Energy balance 1: Building	Heat energy of systems	2 032
	Heat loads from electricity	341
	Other heat loads	690
	Heat losses	-236
	Total removed heat for use	-2 820
Energy balance 2: Technical systems	Ice refrigeration heat	2 224
	Ice refrigeration HP electricity	849
	Gray water heat	90
	Gray water HP electricity	22
	Dehumidification heat	379
	Dehumidification electricity	125
	Supply air heating	-1 547
	DHW heating	-255
	Ice resurfacing water	-58
	Water radiator heating	-26
	Ground frost heating	-81
	Excess heat to swimming hall	-1 692
Total values	Utilized waste heat	1 979
	Purchased district heat	57
	Purchased electricity	1 636
Breakdown of heat energy of systems	Supply air heating	1 600
	DHW heating	267
	Ice resurfacing water freezing	58
	Water radiator heating	26
	Ground frost heating	81
Breakdown of heat loads (+) and losses (-)	Ice resurfacing water freezing	217
	Lighting	244
	Occupants	126
	Ventilation fans	97
	DHW sewage losses	-165
	DH substation losses	-61
	Infiltration air	-10
	Envelope	43
	Other heat loads(+) / losses(-)	304

Combined energy flows [MWh/a]		
Energy balance 5: Total	Total purchased district heat	949
	Total purchased electricity	3 720
	Total heat loads	1 147
	Total heat losses	-3 881
	Total potential excess heat	-1 514

Notation: SH = Swimming Hall, HP = Heat Pump

Energy flows [MWh/a]. SH floor area 7982m²		
Energy balance 3: Building	Heat energy of systems	3 003
	Heat loads from electricity	1 635
	Other heat loads	457
	Heat losses	-3 645
	Total removed heat for use	-1 399
Energy balance 4: Technical systems	Heat from ice hall	1 692
	Exhaust air HP heat	1 187
	Exhaust air electricity	339
	Gray water heat	213
	Gray water HP electricity	55
	Supply air preheating	-207
	Supply air heating	-297
	Water radiator heating	-328
	Underfloor heating	-74
	DHW heating	-354
	DHW preheating	-478
	Pool water heating	-363
	Potential excess heat	-1 514
Total values	Utilized waste heat	2 110
	Purchased district heat	893
	Purchased electricity	2 085
Breakdown of waste heat	Utilized ice hall waste heat	629
	Utilized SH waste heat	1 481
	Excess heat from TES 1	1 214
	Excess heat from TES 2	300
Breakdown of heat energy of systems	Supply air heating	355
	Supply air preheating	293
	Pool water heating	768
	Water radiator heating	375
	Underfloor heating	106
	DHW preheating	612
	DHW heating	494
Breakdown of heat loads (+) and losses (-)	Sauna	585
	Lighting	201
	Ventilation fans	135
	Pool pumps	495
	Equipment	101
	HVAC aux.	118
	Occupants	340
	Radiation through windows	117
	DHW sewage losses	-810
	DH substation losses	-91
	Infiltration air	-98
	Envelope	-360
	Other heat loads(+) / losses(-)	-2 287

Table H.6. Annual energies for the ice and swimming halls with bigger TES discharge time, demand response of electricity for the swimming hall, utilization of waste heat and smart EAHP (Case 8_{2h}).

Energy flows [MWh/a]. Ice hall floor area 6674 m²		
Energy balance 1: Building	Heat energy of systems	1 614
	Heat loads from electricity	244
	Other heat loads	355
	Heat losses	-259
	Total removed heat for use	-1 953
Energy balance 2: Technical systems	Ice refrigeration heat	1 593
	Ice refrigeration HP electricity	578
	Gray water heat	64
	Gray water HP electricity	16
	Dehumidification heat	221
	Dehumidification electricity	73
	Supply air heating	-1 266
	DHW heating	-182
	Ice resurfacing water	-42
	Water radiator heating	-23
	Ground frost heating	-54
	Excess heat to swimming hall	-947
Total values	Utilized waste heat	1 607
	Purchased district heat	14
	Purchased electricity	1 098
Breakdown of heat energy of systems	Supply air heating	1 304
	DHW heating	191
	Ice resurfacing water freezing	42
	Water radiator heating	23
	Ground frost heating	54
Breakdown of heat loads (+) and losses (-)	Ice resurfacing water freezing	155
	Lighting	174
	Occupants	90
	Ventilation fans	69
	DHW sewage losses	-118
	DH substation losses	-48
	Infiltration air	-37
	Envelope	-56
	Other heat loads(+) / losses(-)	110

Combined energy flows [MWh/a]		
Energy balance 5: Total	Total purchased district heat	743
	Total purchased electricity	2 948
	Total heat loads	722
	Total heat losses	-3 558
	Total potential excess heat	-648

Notation: SH = Swimming Hall, HP = Heat Pump

Energy flows [MWh/a]. SH floor area 7982m²		
Energy balance 3: Building	Heat energy of systems	2 738
	Heat loads from electricity	1 444
	Other heat loads	367
	Heat losses	-3 299
	Total removed heat for use	-1 205
Energy balance 4: Technical systems	Heat from ice hall	947
	Exhaust air HP heat	1 018
	Exhaust air electricity	291
	Gray water heat	188
	Gray water HP electricity	49
	Supply air preheating	-231
	Supply air heating	-271
	Water radiator heating	-319
	Underfloor heating	-66
	DHW heating	-311
	DHW preheating	-421
	Pool water heating	-363
	Potential excess heat	-648
Total values	Utilized waste heat	2 009
	Purchased district heat	729
	Purchased electricity	1 867
Breakdown of waste heat	Utilized ice hall waste heat	553
	Utilized SH waste heat	1 411
	Excess heat from TES 1	511
	Excess heat from TES 2	137
Breakdown of heat energy of systems	Supply air heating	327
	Supply air preheating	293
	Pool water heating	684
	Water radiator heating	360
	Underfloor heating	98
	DHW preheating	540
	DHW heating	436
Breakdown of heat loads (+) and losses (-)	Sauna	518
	Lighting	177
	Ventilation fans	118
	Pool pumps	437
	Equipment	89
	HVAC aux.	104
	Occupants	300
	Radiation through windows	67
	DHW sewage losses	-715
	DH substation losses	-83
	Infiltration air	-92
	Envelope	-332
	Other heat loads(+) / losses(-)	-2 077

Table H.7. Annual energies for the ice and swimming halls with bigger TES discharge time, demand response of the ice and swimming halls, utilization of waste heat and smart EAHP (Case 9_{2h}).

Energy flows [MWh/a]. Ice hall floor area 6674 m²		
Energy balance 1: Building	Heat energy of systems	1 554
	Heat loads from electricity	244
	Other heat loads	353
	Heat losses	-242
	Total removed heat for use	-1 908
Energy balance 2: Technical systems	Ice refrigeration heat	1 568
	Ice refrigeration HP electricity	581
	Gray water heat	64
	Gray water HP electricity	16
	Dehumidification heat	207
	Dehumidification electricity	68
	Supply air heating	-1 205
	DHW heating	-182
	Ice resurfacing water	-42
	Water radiator heating	-24
	Ground frost heating	-58
	Excess heat to swimming hall	-957
Total values	Utilized waste heat	1 536
	Purchased district heat	19
	Purchased electricity	1 093
Breakdown of heat energy of systems	Supply air heating	1 240
	DHW heating	191
	Ice resurfacing water freezing	42
	Water radiator heating	24
	Ground frost heating	58
Breakdown of heat loads (+) and losses (-)	Ice resurfacing water freezing	155
	Lighting	174
	Occupants	90
	Ventilation fans	69
	DHW sewage losses	-118
	DH substation losses	-47
	Infiltration air	-31
	Envelope	-46
	Other heat loads(+) / losses(-)	108

Combined energy flows [MWh/a]		
Energy balance 5: Total	Total purchased district heat	817
	Total purchased electricity	2 947
	Total heat loads	717
	Total heat losses	-3 547
	Total potential excess heat	-743

Notation: SH = Swimming Hall, HP = Heat Pump

Energy flows [MWh/a]. SH floor area 7982m²		
Energy balance 3: Building	Heat energy of systems	2 761
	Heat loads from electricity	1 444
	Other heat loads	364
	Heat losses	-3 306
	Total removed heat for use	-1 220
Energy balance 4: Technical systems	Heat from ice hall	957
	Exhaust air HP heat	1 032
	Exhaust air electricity	295
	Gray water heat	188
	Gray water HP electricity	49
	Supply air preheating	-223
	Supply air heating	-283
	Water radiator heating	-317
	Underfloor heating	-69
	DHW heating	-313
	DHW preheating	-380
	Pool water heating	-351
	Potential excess heat	-743
Total values	Utilized waste heat	1 963
	Purchased district heat	798
	Purchased electricity	1 862
Breakdown of waste heat	Utilized ice hall waste heat	461
	Utilized SH waste heat	1 481
	Excess heat from TES 1	585
	Excess heat from TES 2	159
Breakdown of heat energy of systems	Supply air heating	336
	Supply air preheating	292
	Pool water heating	698
	Water radiator heating	357
	Underfloor heating	101
	DHW preheating	540
	DHW heating	436
Breakdown of heat loads (+) and losses (-)	Sauna	518
	Lighting	177
	Ventilation fans	118
	Pool pumps	437
	Equipment	89
	HVAC aux.	104
	Occupants	300
	Radiation through windows	64
	DHW sewage losses	-715
	DH substation losses	-84
	Infiltration air	-93
	Envelope	-334
	Other heat loads(+) / losses(-)	-2 080

Table H.8. Annual energies without summer breaks for the ice and swimming halls with bigger TES discharge time, demand response of the ice and swimming halls, utilization of waste heat and smart EAHP (Case 10_{2h}).

Energy flows [MWh/a]. Ice hall floor area 6674 m²		
Energy balance 1: Building	Heat energy of systems	2 032
	Heat loads from electricity	341
	Other heat loads	690
	Heat losses	-236
	Total removed heat for use	-2 820
Energy balance 2: Technical systems	Ice refrigeration heat	2 224
	Ice refrigeration HP electricity	849
	Gray water heat	90
	Gray water HP electricity	22
	Dehumidification heat	379
	Dehumidification electricity	125
	Supply air heating	-1 547
	DHW heating	-255
	Ice resurfacing water	-58
	Water radiator heating	-26
	Ground frost heating	-81
	Excess heat to swimming hall	-1 656
Total values	Utilized waste heat	2 015
	Purchased district heat	19
	Purchased electricity	1 636
Breakdown of heat energy of systems	Supply air heating	1 600
	DHW heating	267
	Ice resurfacing water freezing	58
	Water radiator heating	26
	Ground frost heating	81
Breakdown of heat loads (+) and losses (-)	Ice resurfacing water freezing	217
	Lighting	244
	Occupants	126
	Ventilation fans	97
	DHW sewage losses	-165
	DH substation losses	-61
	Infiltration air	-10
	Envelope	43
	Other heat loads(+) / losses(-)	304

Combined energy flows [MWh/a]		
Energy balance 5: Total	Total purchased district heat	819
	Total purchased electricity	3 720
	Total heat loads	1 147
	Total heat losses	-3 881
	Total potential excess heat	-1 386

Notation: SH = Swimming Hall, HP = Heat Pump

Energy flows [MWh/a]. SH floor area 7982m²		
Energy balance 3: Building	Heat energy of systems	3 003
	Heat loads from electricity	1 635
	Other heat loads	457
	Heat losses	-3 645
	Total removed heat for use	-1 399
Energy balance 4: Technical systems	Heat from ice hall	1 656
	Exhaust air HP heat	1 187
	Exhaust air electricity	339
	Gray water heat	213
	Gray water HP electricity	55
	Supply air preheating	-213
	Supply air heating	-299
	Water radiator heating	-331
	Underfloor heating	-80
	DHW heating	-367
	DHW preheating	-497
	Pool water heating	-412
	Potential excess heat	-1 386
Total values	Utilized waste heat	2 202
	Purchased district heat	801
	Purchased electricity	2 118
Breakdown of waste heat	Utilized ice hall waste heat	655
	Utilized SH waste heat	1 541
	Excess heat from TES 1	1 148
	Excess heat from TES 2	238
Breakdown of heat energy of systems	Supply air heating	355
	Supply air preheating	293
	Pool water heating	768
	Water radiator heating	375
	Underfloor heating	106
	DHW preheating	612
	DHW heating	494
Breakdown of heat loads (+) and losses (-)	Sauna	585
	Lighting	201
	Ventilation fans	135
	Pool pumps	495
	Equipment	101
	HVAC aux.	118
	Occupants	340
	Radiation through windows	117
	DHW sewage losses	-810
	DH substation losses	-91
	Infiltration air	-98
	Envelope	-360
	Other heat loads(+) / losses(-)	-2 287