Waste heat utilization for central heating and ventilation systems at Chassix Norway

By

Kjartan Topstad Konradsen

May 31, 2018

Supervisor at University of Agder:
Souman Rudra
Supervisor at Chassix Norway:
Harry Danielsen

Department of Engineering

Faculty of Engineering and Science

Abstract

Chassix from south of Norway attempt to reduce their energy demand by sending waste heat to a Central Heating System, (CHS), named Case 1. The CHS heat the ventilation systems, space heat and water heating. The waste heat used is from the Sand Recycling Process, (SRP), and the Aluminum Cooling Process, (ACP). The ACP and a $600kW$ cooling tower are installed in May/June. Since the CHS is close to the ventilation systems using Battery Heat Exchanger, BHE, a possibility for utilizing the ventilation systems as a cooling tower, named Case 2. Case 2 is an attempt to reduce investment cost by replacing investment cost for installing the cooling tower. The potential heat recovery for CHS needed to be estimated. The estimation was based on dimension data at the building and from an Enova statistic. Estimated capacity was then compared with actual capacity from CHS. The potential heat recovery was estimated to be 1 012-1 $142 MWh/year$, but actual data indicate that it might be lower. Case 1 is possible with reduced production at ambient temperatures above $17.5°C$. With temperature above $22°C$, the process needs to operate at 93%, due to possible cooling. Installing Case 1 and Case 2 are estimated as the most economically efficient that only need 19% support from Enova to have a payback time in three years. Case 1 needs 34% support from Enova. Since the ventilation systems were 20 years old and are soon to rebuild, Chassix decided not to install Case 2.

Preface

By studying renewable energy sources for 4.5 years and reading about the future increase of grid infrastructure problems, I can conclude that waste heat utilization is essential for reducing peaks in the winter. Chassix is at the beginning of reducing their energy demand and wanted to recruit a master student to find waste heat solutions. This task felt rewarding due to the positive impact it would have on cost and infrastructure. The utilization of waste heat in this report is transferred to heat the buildings and shower water. By finding economically beneficial solutions for waste heat, this could motivate other companies to do the same.

This paper has used a significant time to understand and find solutions. Chassix were in a period with a lot to do since they changed the owner from Benteler to Chassix and the ACP were built. Chassix were in the start phase of energy management and how all the systems were gathered was not always clear.

Chassix has been helpful with the data they have provided, and several employees have been involved namely Harry Danielsen, Ola Brunvoll, Leif Rinde and Sverre Gardol. To write the report Souman Rudra, Kristine Bjordal and Malini Sarah Espeland have helped with the writing and structure of the report.

Individual mandatory declaration

1.	I hereby declare that my thesis is my own work and that I have not used any other sources or have received any other help than mentioned in the thesis.	☒
2.	I further declare that this thesis: - has not been used for another exam at another department/university/university college in Norway or abroad; - does not refer to the work of others without it being stated; - does not refer to own previous work without it being stated; - have all the references given in the literature list; - is not a copy, duplicate or copy of another's work or manuscript.	☒
3.	I am aware that violation of the above is regarded as cheating and may result in cancellation of exams and exclusion from universities and colleges in Norway, see Universitets- og høgskoleloven §§4-7 og 4-8 og Forskrift om eksamen §§ 31.	☒
4.	I am aware that all submitted theses may be checked for plagiarism.	☒
5.	I am aware that the University of Agder will deal with all cases where there is suspicion of cheating according to the university's guidelines for dealing with cases of cheating.	☒
6.	I have incorporated the rules and guidelines in the use of sources and references on the library's web pages.	☒

Publishing Agreement

Authorization for electronic publishing of the thesis.

Author(s) have copyrights of the thesis. This means, among other things, the exclusive right to make the work available to the general public (Åndsverkloven. §2).

All theses that fulfill the criteria will be registered and published in Brage Aura and on UiA's web pages with author's approval.

Theses that are not public or are confidential will not be published.

I hereby give the University of Agder a free right to

make the task available for electronic publishing: ☒JA ☐NEI

Is the thesis confidential? ☐JA ☒NEI

(confidential agreement must be completed)

- If yes:

Can the thesis be published when the confidentiality period is over? ☐JA ☐NEI

Is the task except for public disclosure? ☒JA ☐NEI

(contains confidential information. see Offl. §13/Fvl. §13)

Contents

List of Tables

List of Figures

Symbols

β	Temperature efficiency
η	Percentage of potential heat recovery
A	Area
c_p	Specific heat
E_{NOK}	Energy cost
E_r	Reduced energy cost
E_{inv}	Investment cost
k	Thermal conductivity
$LMTD$	Log Mean Temperature Difference
L	Length
m	Mass
m_a	Mass of air
m_{Alu}	Mass of aluminum
$m_{Alu,tot}$	Mass of aluminum per year
$\dot{m}$	Mass flow
P_{CHS}	Electrical energy from CHS
$\dot{P}_{CHS}$	Electrical power from CHS
Q	Amount of heat
Q_{red}	Reduced heat demand
Q_{pot}	Potential heat recovery
Q_{WH}	Amount of heat to water heat
Q_{ACP}	Waste heat from ACP
Q_{stat}	Enova statistic heat demand
Q_{SRP}	Waste heat from SRP
Q_{HB}	Amount of energy in heat battery
$\dot{Q}$	Heat transfer
$\dot{Q}_C$	Condution
$\dot{Q}_{Alu}$	Heat transfer aluminium
$\dot{Q}_{HB}$	Capacity of heat battery
$\dot{Q}_{WH}$	Capacity to water heat
$\dot{Q}_{HB}$	Capacity to water heating
T_a	Ambient temperature
T_{in}	Temperature after ventilation
T_{out}	Temperature out from the location
$T_{Y,sf}$	Temperature of a medium s, in direction f, from heat exchanger Y
T_{HB}	Temperature out form HB
$T_{HE1,ao}$	Air temperature out from HB1
$T_{HE2,ai}$	Air temperature in from HB2
$T_{HE2,ao}$	Air temperature out from HB2
$T_{HE2,gi}$	Glycol temperature in from HB2
$T_{HE3,gi}$	Glycol temperature out from HB3
$T_{HE3,go}$	Glycol temperature out from HB3

$T_{HE3,wi}$	Water temperature out from HB3
$T_{HE3,wo}$	Water temperature out from HB3
T_x	Unknown temperature
U	Over all heat transfer
X_e	Employee

1 Introduction

Global warming and the increasingly negative impacts to the climate forces the world to take action. These actions focus on sharing energy from renewable resources and increasing energy efficiency. Emissions of greenhouse gases from energy production in an industry are one of the main reasons for the increased CO2 levels in the atmosphere in the world today[1]. Utilization of waste heat is reducing the growing heat demand and the industry more energy efficiency and Co2 friendly[2]. Reduction of energy consumption in an industry has the advantage of promoting eco-efficiency and minimize operation costs[3],[4].

Two methods of utilizing waste heat are: indirect and direct. The indirect method converts waste heat to mechanical and electrical energy such as organic Rankine-, Bryton cycle, power cycle, turbines, termoelectric[5],[6]. The direct method utilizes waste heat to usable heat such as heat pump, heat exchanger, heat pipe, boiler and district heating[6], [7]. The temperature of waste heat is significant for possibilities for utilization and is categorized in low, medium or high grade waste heat.

In Norway, the main energy production is electrical and increasing[8]. The grid infrastructure is dimensioned after peak capacity. The electrification of the transport sector forces the energy company to upgrade the grid[9]. NVE (Norwegian Water Resources and Energy Directorate) are introducing power tariff which gives an economic benefit off decreasing the maximum capacity[10]. Norway has a cold climate and the highest peaks are in the winter/January due to temperature[8]. Waste heat utilization for heating the buildings locally, or district heating, is important for grid infrastructure to reduce the electrical peak. In a study from Enova potential waste heat from industry are, 25-60$°C$ and 7 $TWh/year$, 3.1 $TWh/year$ at 60-140$°C$, 9.2$TWh/year$ at <140$°C$[11]. Temperatures in a building are 20-22$°C$ in air and 65-75$°C$ in water [12],[13],[14]. Which indicate that waste heat for heating buildings has potential.

1.1 Background

This report discuss the possibilities for utilizing waste heat locally for space heating the buildings and water heating at Chassix Norway. Chassix is located in the south of Norway and produces aluminum parts to high-end vehicles. Since the aluminum process is energy demanded, Chassix has cooling towers with a capacity of 2.4MW and considering using this as waste heat. The company is in the start phase by integrating energy management, and it is desired to utilize the waste heat be more energy efficient. Several methods for utilizing waste heat were used as explained in section 1.2, before utilizing waste heat in a direct method in ventilation, space heating and water heating. Today this heat demand is heated by a CHS and heat demand at the CHS is unknown and has to be estimated. Utilizing waste heat in CHS is named Case 1.

The waste heat process selected is the ACP. Chassix is installing ACP, in May/June, which requires a new cooling tower at 600kW. Since the distance from CHS to the ventilation system is close. The ventilation systems with BHE is evaluated if it could replace the cooling tower. If the BHE has the cooling capacity at 600kW and replace the investment cost of installing the new cooling tower and if it is cheaper. Another waste heat source is the cooling tower from the SRP. This is also considered due to of grid heating and could be replaced if ACP has no waste heat. The solution to utilize the BHE as a cooling tower is named Case 2.

1.2 Preparatory work and approach

This theses started by visiting Chassix. There were no direct problems expected to be solved. By attending meetings different process were addressed and problems to investigate.

A cooling channel, cools down aluminum after the ACP with air. With high ambient temperature, the cooling capacity is reduced. A geothermal cooling was looked into but ended after looking at cost and time of problems. If some other process would be beneficial with using geothermal, it could be used combined with this to reduce cost.

It was attempted to make SRP more energy efficient by recovering exhaust air to preheat the sand. If the sand is heated to boiling point, unhealthy gases are released. There were attempts to find out by calling the firm Ashlad chemical S.A, without any definite answer. Then opportunities to find the boiling point at the University in Agder, but ended after considering my own competence and the time-consuming workload.

Chassix consumes $20 GWh/year$, Enova concluded that $250 GWh/year$ consumption and high temperature is necessary to have economically beneficial of indirect waste heat utilization[11].

District heating is utilized by Alcoa located 400 meters away, and Chassix wishes to utilize the waste heat in the building before considering district heating input or output.

After looking in the archive, the BHE were addressed, and was a known heat recovery method with low temperature efficiency. Then it was found that Chassix uses a CHS to heat ventilation systems, space heat, and water heat. The heat demand at CHS is uncertain and had to be estimated. Utilizing waste heat in the CHS was seen as a possibility with a match from ACP which is built in May 2018. With learning about the new ACP, and that it needed a cooling tower, there was seen an opportunity to use the BHE as a cooling tower, named Case 2. Then the consultant company Norsk Energi became involved, since the paper would not be finished before June 2018. After a while, it was mentioned by Chassix that the ventilation systems is 20 years old and should soon be rebuilt since the typical lifetime is 20-25 years. Since this solution had been calculated, it presided in case on another point would be an opportunity.

Grey water from showering, was seen as a possibility reducing water heat and peeks. However, this heat is distributed from the CHS, it should be done after waste heat utilization in CHS.

1.3 Research questions

Chassix has possibilities for utilizing waste heat from Case 1 and Case 2. Case 2 is mainly looked into to reduce investment cost. Before installing, an economic evaluation needs to be estimated and these questions needed to be answered:

- How much energy could be utilized by waste heat?
- Can the BHE replace the cooling tower?
- What is the investment cost with and without replacing the cooling tower?
- Are the two cases within the range of three years payback, set from Chassix?

1.4 Limitation

The processes use analog sensors, and there are many temperatures and mass flows which varies with ambient temperature. The uncertainties result in assumptions and average estimations. Chassix is at the beginning of energy management, and have available data or report to give an overview and proper insight of how efficient the system operated and heat is distributed. During the progress of this study, new information came and work had to be redone.

In the systems, the liquid temperatures have a reduced impact on heat demand, in the regulation, it has a higher impact due to temperature difference or capacity of the heat exchanger. The exhaust air temperature has a more significant impact on the heat demand since it regulates the bottom line capacity for the whole year.

Collecting data with logging or digitizing the sensors results in more accurate estimation of values are possible. The digital logging of CHS would be the most accurate.

The Aspen Plus program was needed to be learned. At the beginning of the report, I was not able to use it competently before late March. This delay was the reason the excel simulation was used in the beginning and time to learn the investment cost method in this program.

2 Existing systems at Chassix

To understand the existing systems, each system is explained individually in these sections. It is the CHS which heat the ventilation systems, space heat and water heat as shown in Figure 1. There are two waste heat systems ACP and SRP. ACP has a capacity of $600kW$ and SRP has $300kW$.

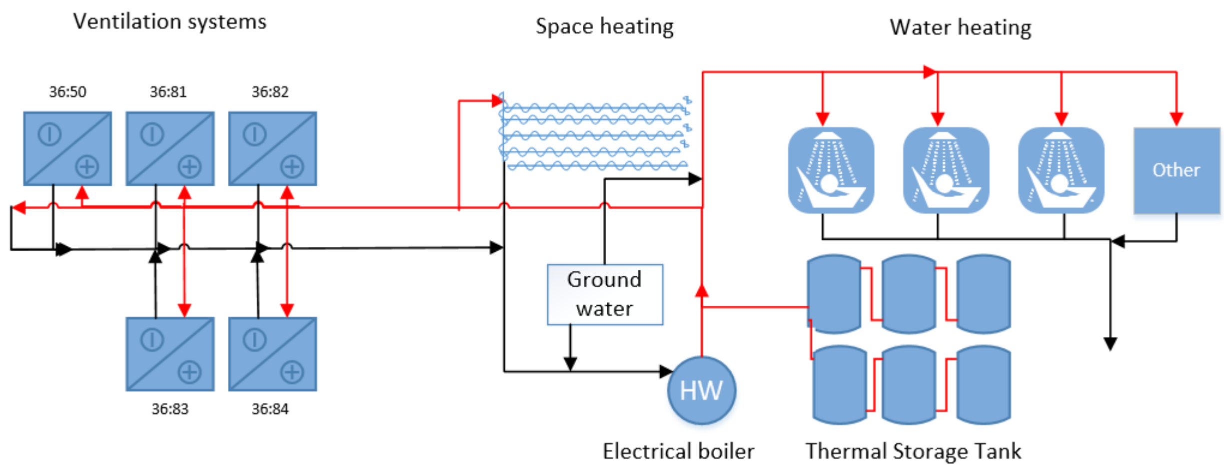


Figure 1: Drawing of how ventilation, space heat and water heat are connected with CHS

The building is separated into two parts; Production hall and administration hall. The ACP and SRP are in the production hall, and the CHS is in the administration hall. The distance from CHS to ACP and SRP are estimated to be between 120 and 150 meters, and the ventilation is estimated to be 32 meters away from the CHS.

2.1 Central Heating System

CHS is the heat source of water to ventilation, space heating and water heating in the building. The capacity is uncertain but with the principle shown with an energy flowchart below.

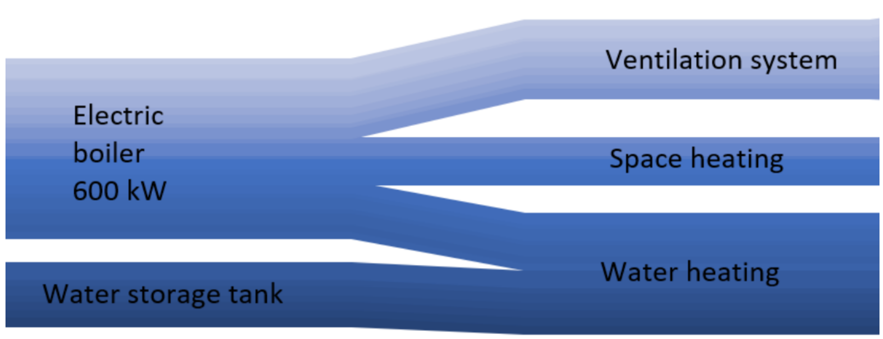


Figure 2: Energy flow ventilation, space heating and water heating are heated

An electrical boiler heats the water with a capacity of $600kW$ to 65°C. If the electrical boiler is insufficient, the water is mixed with water from six Thermal Storage Tanks, (TST), at 300 liters per tank at 70-80°C. The storage tank has a circulating system.

2.2 Ventilation systems

The ventilation systems are 20 years old and utilize heat recovery with BHE and RHE. How a BHE and heat RHE are explained in sections 3.4.1-3.4.2. Total mass flow is $18kg/s$ and specification for each ventilation system is shown in Table 2.

Table 2: Ventilation specifications

Ventilation system [-]	Air mass flow [kg/s]	Glycol mass flow [kg/s]	Heat battery [kW]	β [%]
36:50	2	-	40	70.3
36:81	2.67	1.09	56	52
36:82	2.33	1.37	70	51
36:83 & 36:84	5.00	2.05	105	50

In this report, the efficiency of the BHE is considered to be constant at 51%. From the installation report shown in appendix A.25, the ventilation system operate from Monday 06:00-Friday 20:00, which equals to 5720 *h/year*.

2.3 Space heating

The administration hall is 7 $200m^3$ to offices and $300m^3$ to others shown in appendix A.24. The heating is considered to be on at all time of year. The heating is by radiators and floor heating in the shower room. There are many windows and many angles which increase the number of cooling bridges.

2.4 Water heating

Water heating is chosen due to the amount of showering, and in this report the rest is neglected, due to lack of data. Chassix estimated that 100 employees are showering per day over three shifts. The water heating is estimated to be only from showering in this report. An employee reported the time to get comfortable hot water was high due to long pipelines.

2.5 Sand Recycling Process

The SRP is a process in which recycled sand, mixed with glue. In the process, the sand mixture is burned with propane to $700°C$ as shown in Figure 3. The figure is from the catalog given by Chassix. The sand is then cooled down with a furnace preheat module which recovers heat back to the PXG Fluid Bed. The sand is then cooled down with a cooler classifier module with water and cooling tower. The cooling tower has a capacity of $300kW$ at common operation inlet/outlet temperature $32/24°C$. In year 2015, 3 $313MWh/year$ of propane with an operation time of 6 $000h/year$.

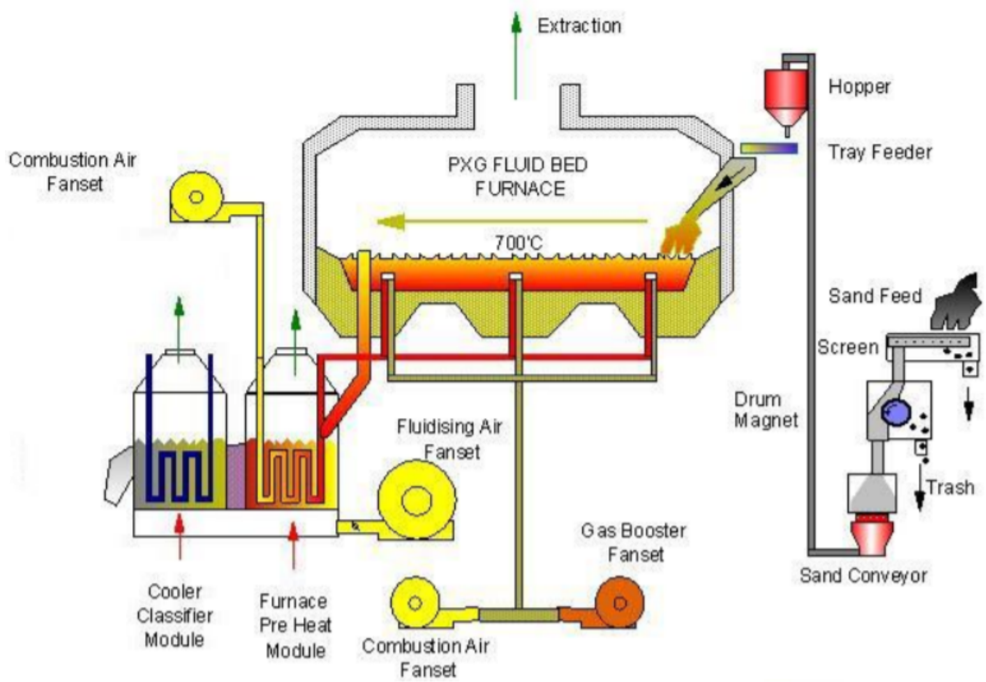


Figure 3: Sketch of Sand Recycling Process

2.6 Aluminum Cooling Process

The ACP shown in Figure 4, is cooling down a rack of $700kg$ aluminum, from $540°C$ to $150°C$ with a $15m^3$ water tank at $65±5°C$. The cooling period is five min, with a new rack every 15 min. The average heat transfer at full capacity is estimated to be $600kWh/h$. The producer mentions that the capacity might be less due to latent heat and water refilling. With $600kW$ capacity the water tank temperature increases with approximately $0.5°C/min$ on average and $1.5°C/min$ at the time the rack is in the water.

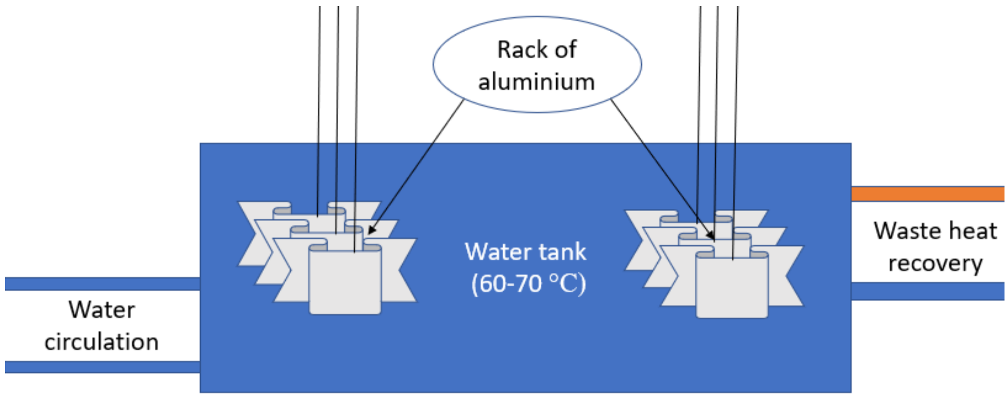


Figure 4: Aluminum Cooling Process

The ACP is planned to be installed in May/June 2018, and they need to install a cooling tower to maintain $65±5°C$. It is due to the installation that they have the possibility to utilize the BHE instead of installing a cooling tower. This report was not finished in time to know if it was economically responsible, so Chassix hired a consulting firm Norsk Energi, to speed up the calculations.

3 Theory

The theory chapter include how heat losses in buildings occur, what is impact heat transfer, and technology for heat exchangers.

3.1 Types of heat exchanger

This section will explain the advantages and disadvantages of some heat exchangers that could be relevant for this project. The two major types of heat exchangers are shell and tube-, and plate heat exchangers. The shell and tube is designed with tubes or finned tubes. The main difference between these heat exchangers is that in a shell and tube heat exchanger the fluid runs over and through, while in plates heat exchanger the fluid is flowing through a baffle which separates the fluid [15]. Heat exchanger design is evolving with trying to make them smaller and more efficient. In the recent years, heat exchangers with phase changing material to support local storage have been researched [16].

3.1.1 Finned Tube Heat Exchanger

Finned tube heat exchanger, (FTHE), is a type of shell and tube heat exchanger with fins as shown in Figure 5. The fin uses the difference in the area at the two sides to increase the heat transfer. FTHE is often used with air and liquid in ventilation ducts.

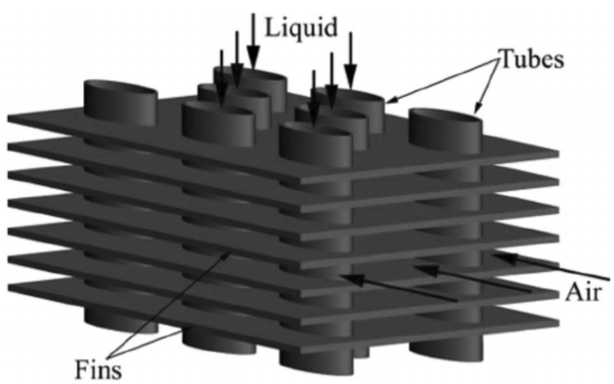


Figure 5: Finned tube heat exchanger[17]

3.1.2 Plate heat exchanger

Plate and frame heat exchanger for general refinery service can be referred to as gasket plate heat exchangers. The plate heat exchanger consists of a frame, which consists of a head, follower, column, carrying bar, guiding bar, and some clamping bolts. Advantages and disadvantages of plate heat exchanger compared with shell and tube are listed under[18].

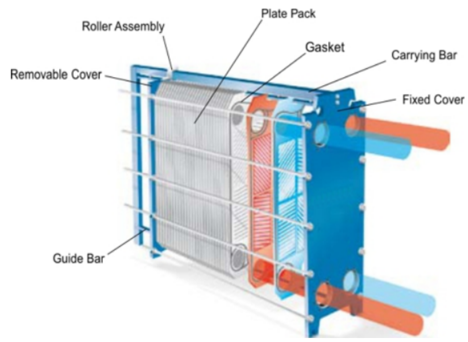


Figure 6: Gasket plate heat exchanger[19]

Plate heat exchanger advantages

- High overall heat transfer coefficient, compared with shell and tube heat exchanger or spiral heat exchanger.
- The combination high U-value and compact configuration the size could be as much as five times the size of a shell and tube.
- Easy maintenance and cleaning since it could be taken apart.
- The area could be increased or decreased by addition or removal to expand or reduce heat transfer.

Plate heat exchanger disadvantages

- Higher potential for leakage
- Higher pressure drop
- Dose not work well with high fluid temperature, and the gasket imposes temperature limitations.

3.2 Heat in the building

These sections explain heat losses in a building structure.

3.2.1 Conduction loss

Conduction losses in a building are heat losses through main elements of the building fabric (through roof, walls, windows and floor) calculated with equation

$$\dot{Q}_C = \frac{A \cdot k \cdot (T_{in} - T_a)}{L} \tag{3.1}$$

$$\tag{3.2}$$

where $\dot{Q}_C$ is conduction loss, A is area, k is thermal conductivity, T_{in} is the temperature into the building, T_a is ambient temperature and L is the thickness of the substance. 20-50 % extra through thermal bridge heat losses (corners, junctions, and structural elements penetrating the insulation layer) [20].

3.2.2 Ventilation

The ventilation circulates the air to maintain healthy indoor environment. A balanced ventilation system utilizes heat recovery from the exhaust air and a heat battery to preheat the ventilation air. The efficiency of heat recovery ,N_{TE}, is dependent of types of method. Different types are explained in section 3.4. The heat demand for a heat battery Q_{HB} is calculated with equation

$$\dot{Q}_{HB} = \left(1 - \frac{\beta}{100}\right) \cdot \dot{m} \cdot c_p \cdot (T_{in} - T_a) \tag{3.3}$$

where β is heat recovery efficiency [21].

Eq. 3.3 is independent on the relation of inlet and exhaust air temperature. The heat recovery efficiency is calculated with

$$\beta = \frac{T_{HE1,ao} - T_a}{T_{out} - T_a} \tag{3.4}$$

where $T_{HE1,ao}$ is inlet temperature after heat recovery and T_{out} is hot exhaust air. With using Eq. 3.4 and Eq. 3.9 the heat capacity to HB is

$$\dot{Q}_{HB} = \dot{m} \cdot c_p \cdot (T_{in} - \beta \cdot (T_{out} - T_a) - T_a) \tag{3.5}$$

The capacity of Heat Battery, (HB), dependent on temperature efficacy and exhaust air temperature is shown in Figure 7, where T_{in} is set to 22 $°C$.

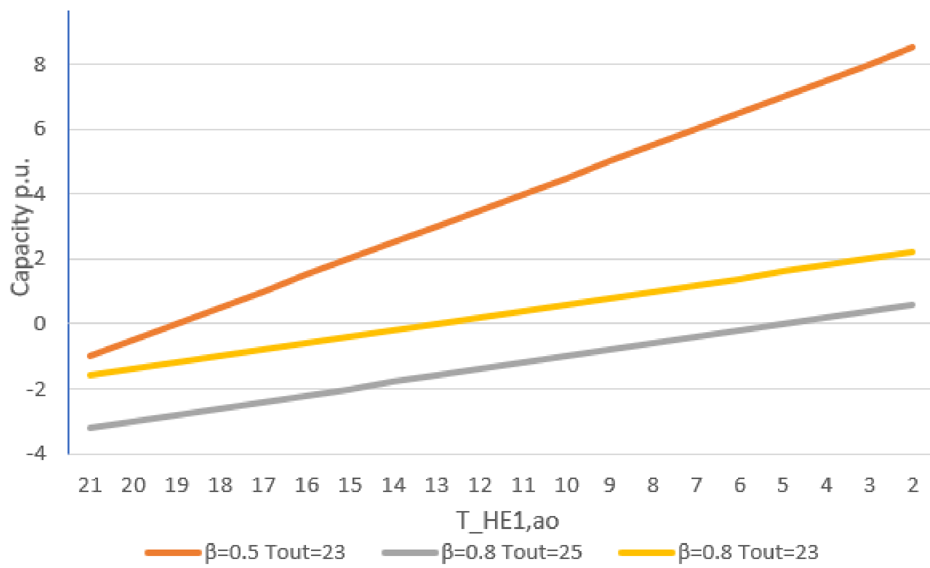


Figure 7: Capacity of HB dependent on β, T_{in} and T_{out}

When the graph is 0, heat battery is not used. If it is negative the heat recovery method needs to operate less efficiently since the inlet temperature would be higher then required.

3.3 Heat transfer

These sections explain the section of how the heat transfer varies, dependent on temperature, mass flow, thermal resistance, mediums and typical values.

3.3.1 Log Mean Temperature Difference

Log mean temperature difference, (LMTD), describes how the temperature difference behaves logarithmically within the heat exchanger instead of linearly[22], and is shown in Figure 8

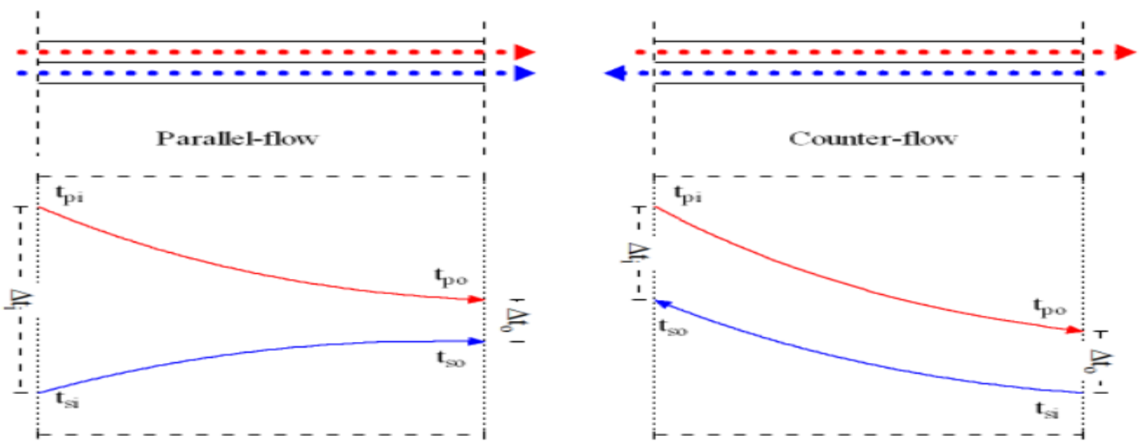


Figure 8: LMTD with counter- and parallel flow [23]

The difference between calculating counter- and parallel flow is shown with Eq. 3.6

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{ln(\frac{\Delta T_1}{\Delta T_2})} \tag{3.6}$$

$$\Delta T_1 = T_{hot,in} - T_{cold,out} \tag{3.7}$$

$$\Delta T_2 = T_{hot,out} - T_{cold,in} \tag{3.8}$$

The heat transfer in each medium with constant pressure are

$$\dot{Q} = \dot{m} \cdot c_p \cdot \Delta T \tag{3.9}$$

3.3.2 Overall heat transfer

Overall heat transfer, U, describes the rate of heat exchanged inside a heat exchanger. The calculations of U depends on heat transfer, area and LMTD with EQ. 3.10.

$$U = \frac{\dot{Q}}{LMTD \cdot A} \tag{3.10}$$

The U-value varying by several factors as fluid, geometry, viscosity, temperature, thermal resistance, fouling, mass flow and flow path [24]. In Table 3, media with typical U-value i present.

Table 3: Typical U-value for different medium [25]

Fluid 1	Type of HE	Fluid 2	U-value $[W/(m^2 \cdot K)]$
Air (atm. pressure)	Fin and tube	air (atm.)	5-35
Water	Fin and tube	Air or gas (atm.)	15-70
Water	Fin and tube	Water	150-1 200
Water	Plate	Water	1 000-4 000
Water vapor	Plate	condensation of water vapor	30 000-140 000

3.3.3 U-values efficiency dependent on optimal mass flow

The optimum mass flow is defined as Eq. 3.12 and is based on Eq. 3.11 but the ΔT is the same on both sides.

$$\dot{m}_{opt} \cdot c_{p,opt} \cdot \Delta T = \dot{m}_1 \cdot c_{p,1} \cdot \Delta T \tag{3.11}$$

$$\dot{m}_{opt} = \frac{\dot{m}_1 \cdot c_{p,1}}{c_{p,opt}} \tag{3.12}$$

The effectiveness of mass flow and temperature is visualized in Figure 9 with variation of heat transfer coefficient,K, and temperature and mass flow in Figure 10.

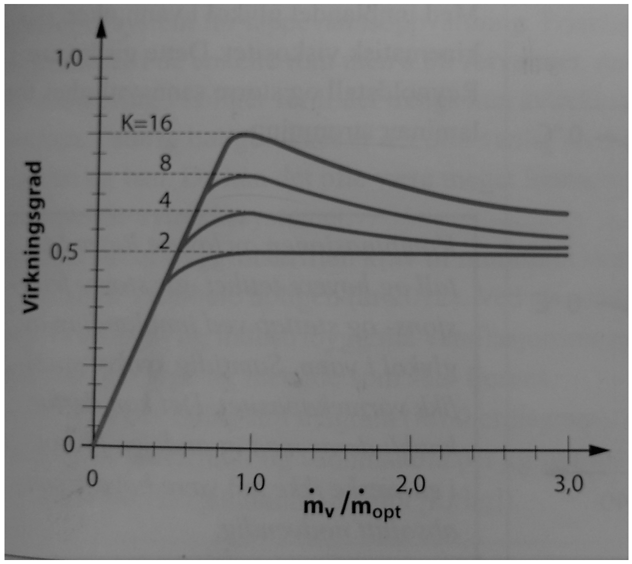


Figure 9: Effectiveness of optimum mass flow[26]

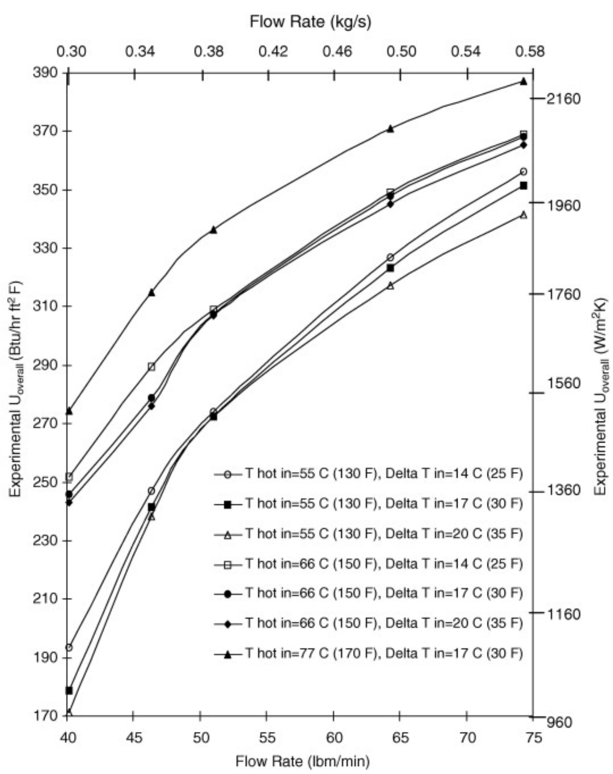


Figure 10: U-value dependent on temperature and mass flow[27]

3.3.4 Relation of area and heat transfer

The Eq. 3.13 shows that if the U-value from one substance is low, the area could compensate. This is typically for fin and tube heat exchanger. One example is a pipe where the inside area is smaller than outside area.

$$U_1 \cdot A_1 = U_2 \cdot A_2 \tag{3.13}$$

3.3.5 Velocity impact on heat transfer

The velocity has an impact on heat transfer, as examples are listed in Table 4.

Table 4: Velocity impact of U-value[28]

Velocity	Typical U-value $[W/(m^2 \cdot K)]$	Variation $[W/(m^2 \cdot K)]$
Natural convection air (<1m/s)	10	20...20
Forced convection air (>5m/s)	50	20...200
Natural convection water (<0.1m/s)	200	10...1 000
Forced convection water (>0.5m/s)	5 000	50...20 000

3.3.6 Laminar flow

Laminar flow works as a thermal resistance. The thicker the laminar sublayer is, the higher the thermal resistance is. With low turbulence or low Reynolds number, there would be a sublayer of laminar flow into the surface. By increasing the velocity, the Reynolds number would increase the laminar flow within the surface and would decrease as the turbulence increase. Turbulence has higher convection then laminar which explains this behavior[22].

3.3.7 Fouling

Fouling on a surface would also behave as thermal resistance since it increases the thickness of the surface. The resistance of fouling and laminar is shown in Figure 11

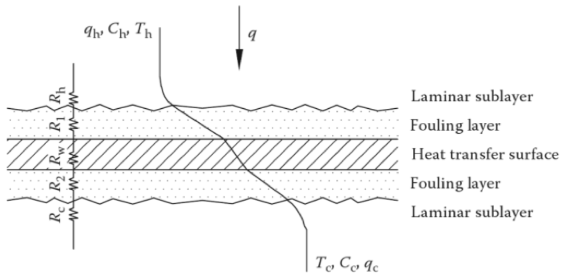


Figure 11: Thermal Resistance of Laminar Fouling

Even if fouling is a disadvantage as thermal resistance, it could also increase heat transfer. Fouling sometimes occurs in shapes that make the laminar sublayer begin to be turbulent. This phenomenon is also used when designing the heat exchanger as shown in Figure 12. The fouling layer occurs faster with laminar flow than with turbulent flow[15].

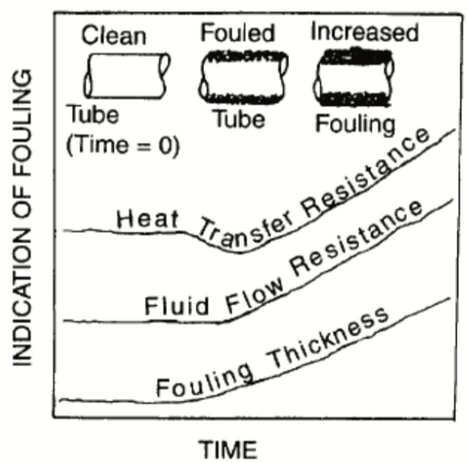


Figure 12: Fouling in tubes[29]

3.4 Heat recovery in balanced ventilation

This section will explain two heat recovery methods in balanced ventilation: Rotating heat exchanger and battery heat exchanger.

3.4.1 Rotating Heat Exchanger

Rotating heat exchanger, (RHE), is a wheel that rotates in the outlet and inlet air. The temperature efficiency is 70-80%[26]. One of the disadvantages of RHE is that some of the exhaust air is sent back to the inlet air. The recovered air is a problem in some places where there is a restriction for this, like hospitals or placeses where unhealthy chemicals could be released from exhaust air duct to inlet air.

3.4.2 Battery Heat Exchanger

Battery heat exchanger, (BHE), uses two finned tube heat exchangers; one in the exhaust air duct and one in the inlet air duct. In between a glycol circuit transports the heat as shown in Figure13. This method reuse no exhaust air from exhaust air and has a temperature efficiency of 50-55%[26].

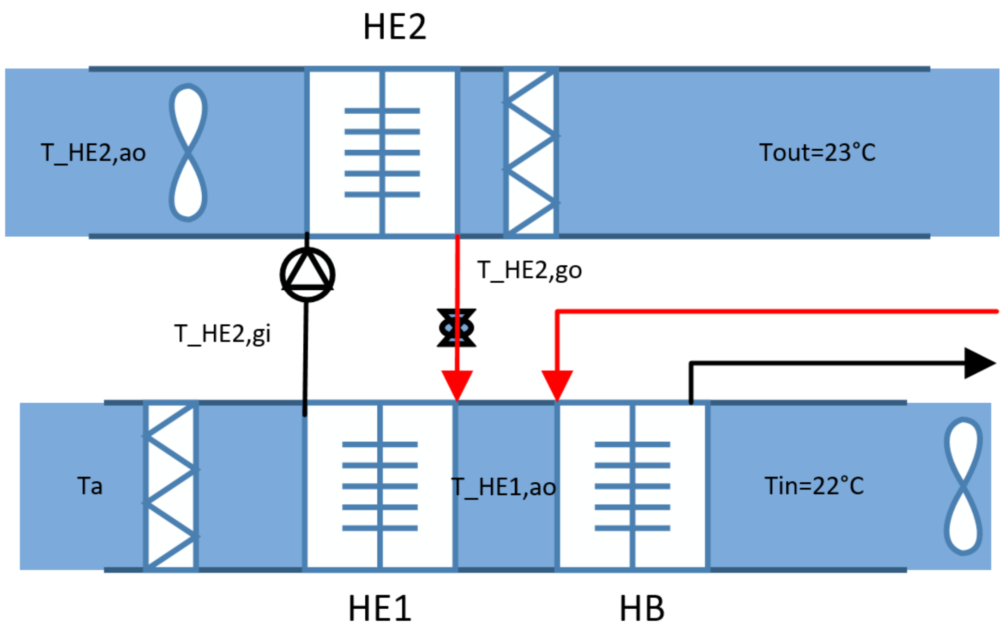


Figure 13: Battery heat exchanger

4 Method

With the knowledge of the existing systems, methods for estimating heat demand at CHS and heat recovery of waste heat is estimated. When the heat recovery is estimated, the two cases are explained with calculations to determine missing data needed before mass flows and heat exchangers are sized. After the system is designed the system is simulated and an estimation of the efficiency of the system is estimated. From this the economic evaluations are explained.

4.1 Heat demand in the systems

4.1.1 Aluminum Cooling Process

The capacity at full production is $600kW$. Based on the monthly aluminum production, an average capacity, $\dot{Q}_{ACP,m}$, in the month is calculated with

$$\dot{Q}_{ACP,m} = \frac{Q_{Alu,max} \cdot m_{month}}{m_{rack} \cdot h} \tag{4.1}$$

where $\dot{Q}_{Alu,max}$ is the capacity at full load, m_{rack} is mass of aluminum per rack, h is operation time and m_{month} is the monthly aluminum production, shown in appendix 11.

If Case 2 has lower capacity then $600kW$, the reduced production capacity, η_{ACP}, is decreased with

$$\eta_{ACP} = \frac{\dot{Q}_{HE3}}{\dot{Q}_{ACP}} = \frac{h_{ACP,n}}{h_{ACP}} = \frac{m_{rack}}{m_{rack,max}} \tag{4.2}$$

4.1.2 Ventilation heat demand

The heat demand in the ventilation systems is dependent on time with different ambient temperature h_{T_a}. Data of ambient temperatures was collected in the period 0.1.01.2015-31.12.2017 from eklima.no at Lindesnes ($20km$ from Chassix). By using this data and Eq. 3.5 the heat demand is calculated with,

$$Q_{HB} = \sum_{T_a=min}^{max} \frac{\dot{m} \cdot c_p \cdot 5720}{8760} (T_{in} - \beta \cdot (T_{out} - T_a) - T_a) \cdot h_{T_a} \tag{4.3}$$

Where min is lowest temperature measured, max is when $T_{HE1,ao}$=T_{in}

Data was collected by creating an account at eklima.no. Press "Observations" in the main menu. In the type of report choose "observation- hour and minutes list." In the setting list, insert time interval "all" and then temperature. Press next two times, and the data is in the List of reports on the front page. Copy the temperature in excel and use equation 3.5 for each temperature and summarized for each year. To count the number of hour per temperature countif function is used.

4.1.3 Space heating

The administration hall uses the ventilation systems to preheat the air and space heating to increase the temperature locally based on employee preferences. For calculating space heat demand, the capacity is estimated. The space heat capacity depends on heat loss explained in section 3.2 and heat gain from energy sources as lamps, computers, humans, cooking,$\dot{Q}_{dic}$ explained with the equation

$$\dot{Q}_{SH} = \dot{Q}_C + \dot{Q}_R + \dot{Q}_L - \dot{Q}_{EL} - \dot{Q}_{div} - Q_{v\dot{e}nt} \tag{4.4}$$

Due to lack of data and time to complete the equation, an alternative equation was made based on how CHS is dimension and after -18 °C[30]. The space heating from CHS with ventilation and water heating. The space heat is then calculated with,

$$Q_{SH} = \frac{(\dot{P}_{CHS,max} - \dot{Q}_{HB,max}) \cdot 6000}{8760} \cdot \sum_{T_a=-18}^{T_{out}} \frac{(T_{out} - T_a) \cdot h_{T_a}}{T_{out} + 18} \tag{4.5}$$

The accuracy of this method is based on capacity data from CHS that was collected. The data is compared with space heat and ventilation since the amount of space heat impact the capacity in BHE as shown in Figure 7. Data from water heat was not available, so this is neglected and need to be considered if unexpected high capacity.

$$\dot{Q}_{SH} + \dot{Q}_{HB} = \dot{P}_{CHS} \tag{4.6}$$

The capacity was measured, and weather conditions noted from [31]. The data was collected 9 times at 7 days by employee Sverre Gardl.

4.1.4 Water heating

Water heating, the dominant heat demand is assumed to be from showering and the rest is neglected. It is assumed to be 100 employees showering every day over three shifts, and is calculated with

$$Q_{WH} = \dot{m} \cdot h \cdot c_p \cdot \Delta T \cdot X_e \cdot \frac{50 \cdot 5}{60 \cdot 60} \tag{4.7}$$

The assumption of the calculations are listed below:

- Employees shower per day: 100 employees/day.
- Peak number of employees shower at the same time: 25,X_e.
- Time used per shower: 7.5min, h.
- Mass flow: 12l/min, $\dot{m}$[32].
- Hot temperature: 38°C[32].
- Cold temperature: 6°C[33].

4.1.5 Heat demand based on Enova statistic

The method based on Enova statistics from buildings in Enova report[34], which was the method also used by Norsk Energy. The statistic calculates energy based on total heat demand divided by the area. The heat demand for an office on the south coast in Norway was based on 57 office buildings and consume $197kWh/m^2$, Q_{stat}. The area connected to CHS is 7 $500m^2$, A, which give a yearly heat demand calculated with

$$Q_{CHS} = Q_{stat} \cdot A \tag{4.8}$$

and potential waste heat utilization

$$Q_{CHS} = \frac{Q_{stat} \cdot A \cdot 6000}{8760} \tag{4.9}$$

4.2 Case 1: Waste heat to Central Heating System

The CHS is heated by an electric boiler and TST. To reduce the heat demand from the electrical boiler, the waste heat is implemented before the electrical boiler with a plate heat exchanger as shown in Figure 14

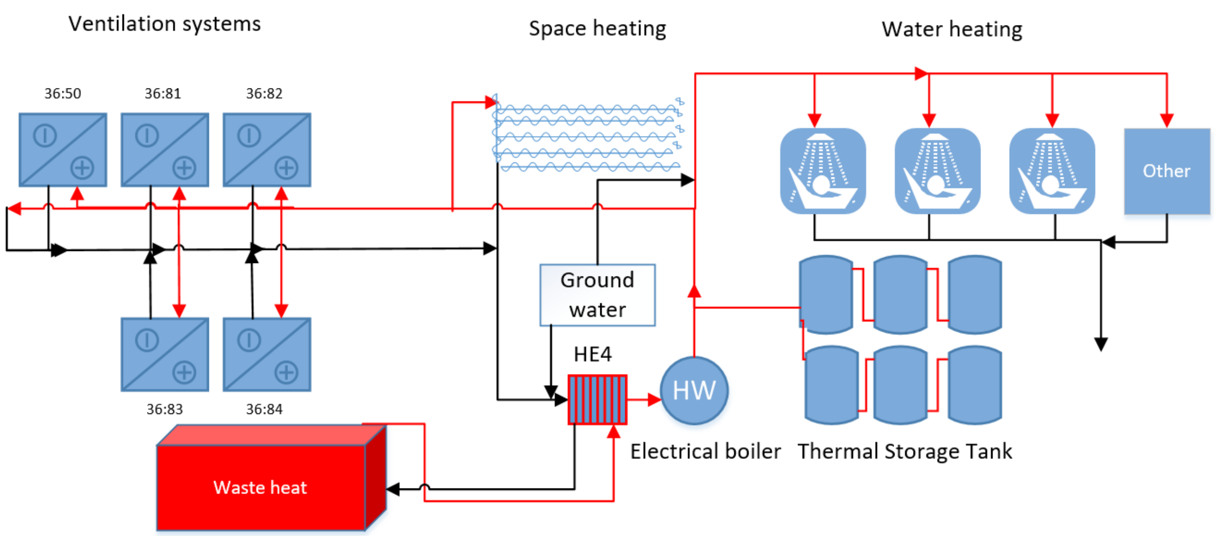


Figure 14: Waste heat sent from ACP to CHS

It was considered using waste heat in the TST, but the temperature at TST is 70-80 °C and uses circulation pumps which limit the possibilities. The TST is in use when a high amount of water is used, and with implementing waste heat before the boiler, the TST would not be in use without changing the regulation.

An alternate solution instead of using HE4 would be using valves to prevent temperature drop in the HE4. This alternative is shown in appendix C.2. The use of valves is possible since both are liquid water.

4.2.1 Estimation of mass flow

The mass flow in the CHS is related to the capacity of the heat battery, space heat, and use of water. The ΔT in heat battery is higher since the temperature is lower in the ventilation. The ΔT in space heat is set to be $30°C$ at all time in this report.

$$\dot{m}_{CHS} = \frac{\dot{Q}_{HB}}{c_p \cdot \Delta T} + \frac{\dot{Q}_{SH}}{c_p \cdot 30} + \dot{m}_{WH} \cdot X_e \tag{4.10}$$

4.2.2 Inlet temperature

The inlet temperature into the CHS is estimated with Eq. 4.11. The temperature of the heat battery and space heating would increase with lower ambient temperature due to increase in mass flow and capacity.

$$T_{cold} = \frac{\dot{m}_{HB} \cdot c_p \cdot T_c + c_p \cdot T_c + \dot{m}_{SH} \cdot c_p \cdot T_c}{\dot{m}_{HB_{tot}} \cdot c_p + \frac{\dot{Q}_{SH}}{c_p \cdot \Delta T} \cdot c_p + \dot{m}_{WH} \cdot c_p} \tag{4.11}$$

4.3 Case 2: Battery Heat Exchanger as a cooling tower

This case utilizes the waste heat in the four ventilation systems with BHE and the utilization of waste heat from the SRP or ACP. The primary purpose of this case is to consume $600kW$ in the BHE with use of HE1 and HE2. The fluid in SRP and ACP is water, and the ventilation system has 30% ethylene glycol which required a heat exchanger in between, HE3, in each ventilation system as shown in Figure 15.

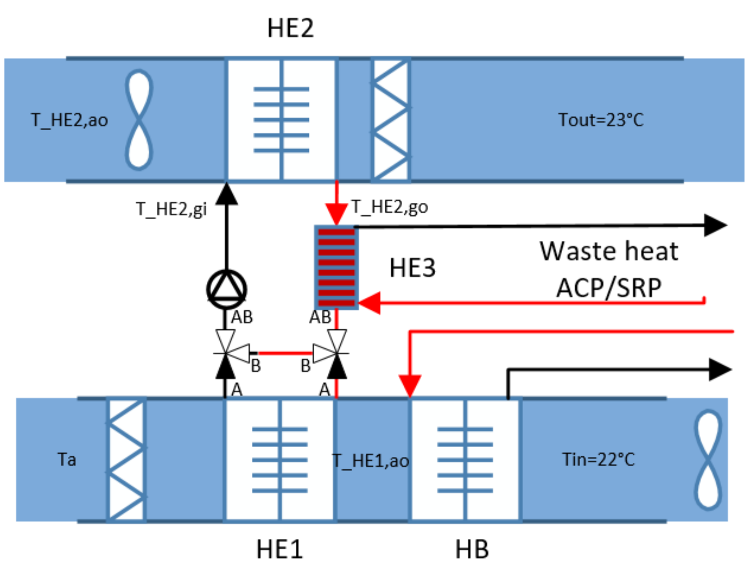


Figure 15: HE3 placed in the glycol circuit in each ventilation system 36:81-84

The HE3 is placed on the hot side since the waste heat fluid is water and the glycol temperature at the hot side could be below freezing point. Placing it on the cold side would also increase the heat recovery from the exhaust air. The splitter regulates the mass flow in the glycol circuit to maintain the inlet temperature to 22°C.

4.4 Dimension of component

To simulate the system and set limitation this section is made.

4.4.1 Capacity of heat exchanger

This report includes several heat exchangers and their capacity is dependent on a correlation between area and heat transfer, UA-value. The area is a constant, and the heat transfer varies with mass flow, turbulence, temperature, thickness, materials, medium, velocity and fouling, explained in section 3.3. In this report, constant UA-value is considered as a proper method, even if this is not accurate when changing the mass flow or temperature.

To minimize pipe size, and not oversize the heat exchangers a minimum temperature approach is set. A graph of how UA-value capacity depends on LMTD and Q is demonstrated with using Eq. 3.10 and varying temperatures at optimum mass flow(T1=T2),

$$UA = \frac{m \cdot c_p \cdot (\Delta T + 0.2 \cdot y)}{LMTD - 0.2 \cdot y}, \qquad y = 0, 1, 2, 3...25 \tag{4.12}$$

also, without optimum mass flow (T1≠T2)

$$UA = \frac{\dot{Q}}{\frac{T1-(T2-0.2\cdot y)}{LN\left(\frac{T1}{T2-0.2\cdot y}\right)}}, \qquad y = 0, 1, 2, 3...25 \tag{4.13}$$

Fig. 16 shows the graph of the Eq. 4.12-4.13

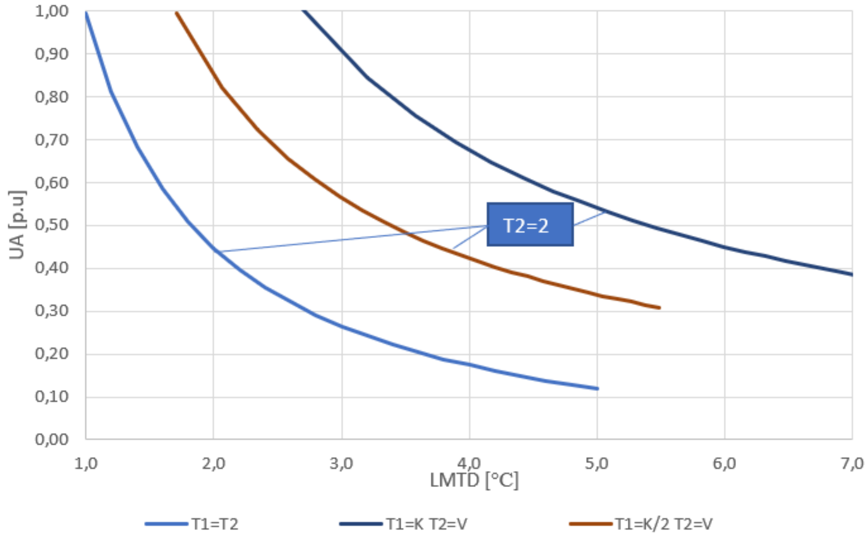


Figure 16: How UA-value varies dependent on minimum temperature approach

This report concludes that minimum temperature approaches at dimensioned conditions when T2=2. The operation time at max capacity is also low due to the number of hours at max capacity.

4.4.2 Mass flow

The main purpose of mass flow in this system is to transport heat, and its capacity is based on specific heat and change in temperature. To reduce pipelines cost, implemented mass flow, m_{new}, is minimized with utilizing highest possible ΔT_{max}. Due to results from section 4.4.1, $\Delta T_{max} - 2$ is used, as well as the follow equation

$$\dot{m}_{new} = m \cdot \frac{c_p \cdot \Delta T_{dim}}{c_{p,new} \cdot (\Delta T_{max} - 2)}, \qquad \Delta T_{dim} \leqslant \Delta T_{max} - 2 \tag{4.14}$$

4.5 Simulations

Excel and Aspen Plus were used to simulate the energy flow. The UA-value used at HE3, is the same in both simulations methods.

4.5.1 Excel simulation

To predict if Case 2 were possible before Aspen Plus was available, the results were simulated in Excel using the equations below

$$\dot{Q}_{HE3} = \dot{Q}_{HE1} + \dot{Q}_{HE2} \tag{4.15}$$

$$\frac{\dot{m} \cdot c_p \cdot \Delta T}{UA_{HE3} \cdot LMTD} = \frac{\dot{m} \cdot c_p \cdot \Delta T}{UA_{HE1} \cdot LMTD} + \frac{\dot{m} \cdot c_p \cdot \Delta T}{UA_{HE2} \cdot LMTD} \tag{4.16}$$

4.5.2 Simulation in Aspen Plus

Aspen Plus is a simulation program widely used with chemicals. It has best in the class database of pure component and phase equilibrium data for conventional chemicals[35]. Aspen Plus was used to calculate and support calculations from Excel simulations. The method used is NonRandom Two-Liquid ,(NRTL), model and is utilized widely in phase equilibrium calculations[36]. Sizes of the heat exchangers are calculated in section 4.3 and all data is shown in appendix C.1.1- C.1.2.

4.5.3 Case 1: Waste heat to Central Heating System

The systems are connected as in the drawing and the input as shown in appendix C.1.1

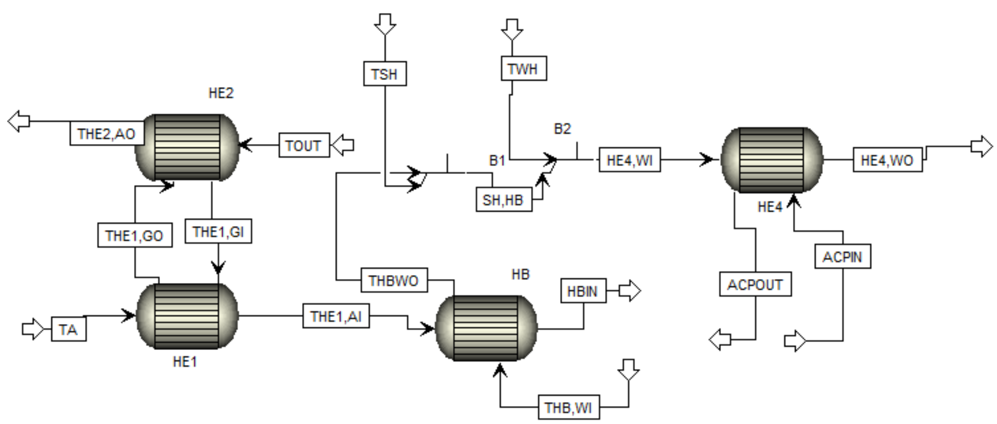


Figure 17: Aspen Plus simulation drawing of case 1

In heat battery and space heat it is used $65°C$, since the output from CHS is $65°C$.

4.5.4 Case 2: Battery Heat Exchanger as a cooling tower

The systems is connected as in the drawing and the input as shown in appendix C.1.2

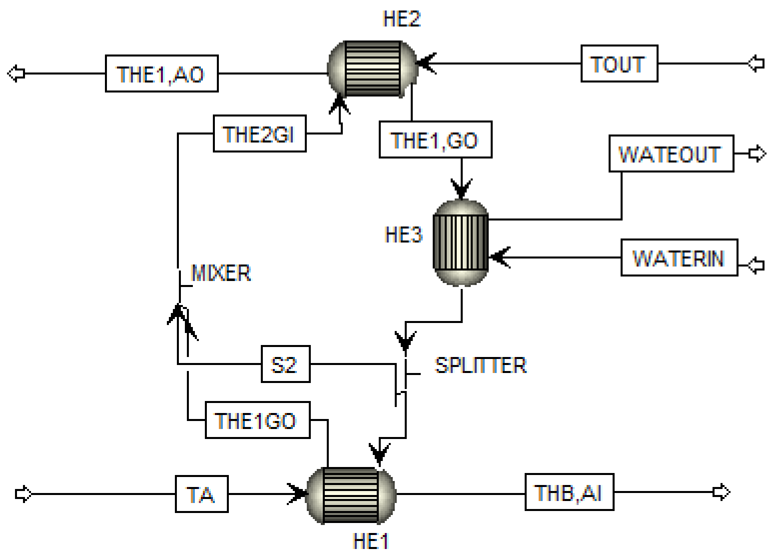


Figure 18: Aspen Plus simulation drawing of case 2

when simulating waste heat in the exhaust air hot and cold side needs to be changed.

4.6 Waste heat utilization in Case 1 and Case 2

The efficiency, η, of heat demand reduced, Q_{red}, dependent on potentional heat recovery, Q_{pot}, is calculated with equation

$$\eta = \frac{Q_{red}}{Q_{pot}} \tag{4.17}$$

Q_{red} is estimated by simulating the ambient temperature all heat is recovered with waste heat, T_{dim}, and at a lower temperature ,T_y. Using the calculated capacity of CHS needed $\dot{Q}_i$, and considerable linearity, the efficiency of utilize waste heat is estimated with equation

$$Q_{red} = \sum_{T_a=T_{min}}^{T_{dim}} \frac{(T_a - T_{dim})}{T_y - T_{dim}} \cdot \dot{Q}_i \tag{4.18}$$

The water heating is estimated based on how much is utilized if 25 employees shower at the same time with ambient temperature at 9°C.

$$N_{WH} = \frac{\dot{Q}_{CHS} - \dot{Q}_{HB} - \dot{Q}_{SH}}{\dot{Q}_{WH}} \tag{4.19}$$

4.7 Economical evaluation

This section explains how the economic evaluations of the two cases are calculated. First the energy savings, then the investment cost. Then how much Enova support need to have three years payback time.

4.7.1 Energy savings

The energy savings are estimated based on waste heat replaced with heat demand. The heat demand and the waste heat is estimated at 8760 $h/year$ and available waste heat is considered to be 6 000$h/year$ and multiply by the reduced heat demand from section 4.6 the energy saving is estimated with equation

$$E_r = (Q_{HB} \cdot \eta_{HB} + Q_{SH} \cdot \eta_{SH} + \Delta Q_{WH} \cdot \eta_{WH}) \cdot E_{NOK} \tag{4.20}$$

where energy price include all, E_{NOK}. Note that water heating is not multiplied with estimated operation hours, this is due to showering is primarily when production is on.

4.7.2 Investment cost

To find investment cost, Aspen Plus was initially used, but with no good results. A Norwegian company named Norsk Energi estimated the cost of installing pipes from ACP to CHS to be 2 420 000NOK shown in appendix B.26. A factor dependent on that calculation then estimates this. Since the size of the pipes needs to be three times higher, four more heat exchangers and 26% longer the price is set to be 1.5 more than the solution of only CHS.

The company Tratec estimated the investment cost to be 1 666 500NOK shown in appendix B.27. If the BHE replaced the cooling tower, the investment cost needs to be lower than this to be economically efficient without considering maintenance.

4.7.3 Payback time

With investing in energy saving, or low emissions solutions, some of the investment could be supported by Enova. Enova is a central instrument in the development of the low-emission society and the future energy systems like this[37]. The payback time is due to Enova support, dependent on the amount of Enova support. The support need from Enova to be in the three years payback is calculated with equation

$$\eta_{Enova} = \frac{3 \cdot E_r}{E_{inv}} \tag{4.21}$$

where η_{Enova} is the percentage support need from Enova and E_{inv} is the investment cost.

5 Results

The results are presented and the detailed information are placed in method and existing processes. First data of heat demand and the process, before an overview of estimated heat demand and temperatures. Then the two cases is presented with results from simulations and an economic calculations.

5.1 Waste heat and heat demand

These sections estimate the capacity of the ACP after production of aluminum. Estimating the heat demand and capacity to the CHS individually by compering of real data to Enova statistics. In the end the temperatures and heat are shown to give and overview.

5.1.1 Aluminum Cooling Process

The waste heat capacity of ACP is varying on how often a new rack is cooled down and kg aluminum per rack. The average monthly capacity due to production is calculated based on data from section 4.1.1 and shown in Figure 19

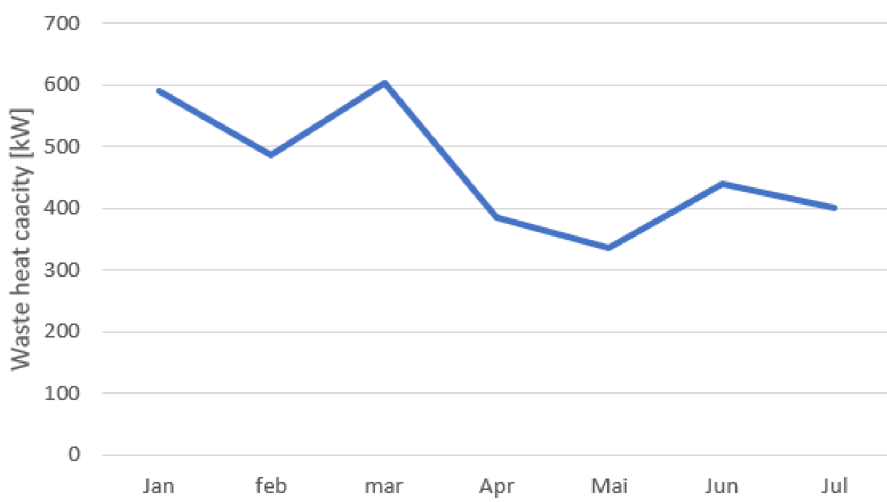


Figure 19: Average monthly capacity from ACP

This shows the average capacity collected is between 334-600kW. The water tank should have a temperature at 60-70°C. If estabilising a homogeneous temperature difference of 7°C is possible. The water tank has energy storage of 105kWh calculated with Eq. 3.9.

5.1.2 Ventilation

The heat demand for the ventilation system is dependent on the ambient temperature. The ambient temperature variations and the profile of the temperature are collected from eklima.no as explained in section 4.1.2.

The year 2017 is 1.6% higher than 2015 and the hours difference in ambient temperature are shown in Figure 20. The average heat demand and hours of ambient temperature is shown in Figure 22 which gives 878$MWh/year$ and estimated based on the ventilation system operating 8760$h/year$

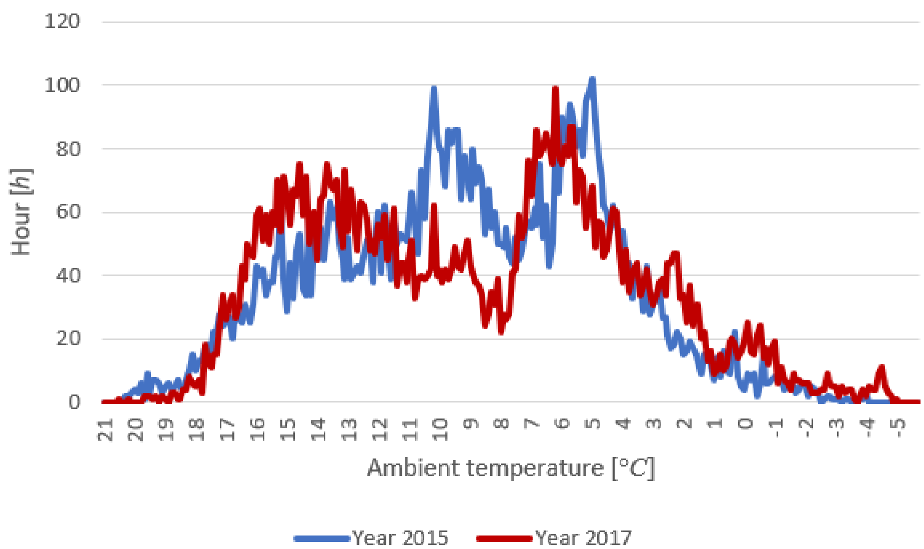


Figure 20: Number of hours per year of given ambient temperature

The graph shows that the temperature varies most between 7-12°C and the highest peaks are between 5-6°C and at 10 °C in year 2015.

With the use of Eq. 4.3, the heat demand for the whole year are presented in Table 5.

Table 5: Ventilation heat demand at 8760$h/year$

Year	2015	2016	2017
Heat demand [$MWh/year$]	868	880	882

5.1.3 Space heat demand

The space heat demand was calculated based on the dimensioned capacity of CHS, with subtracting HB and considering that water heat is dimensioned for TST. The data is then compared with actual measurements from CHS shown in Figure 21. The results of collecting data explained in section 4.1.3 are listed appendix 12 and shown with graph below.

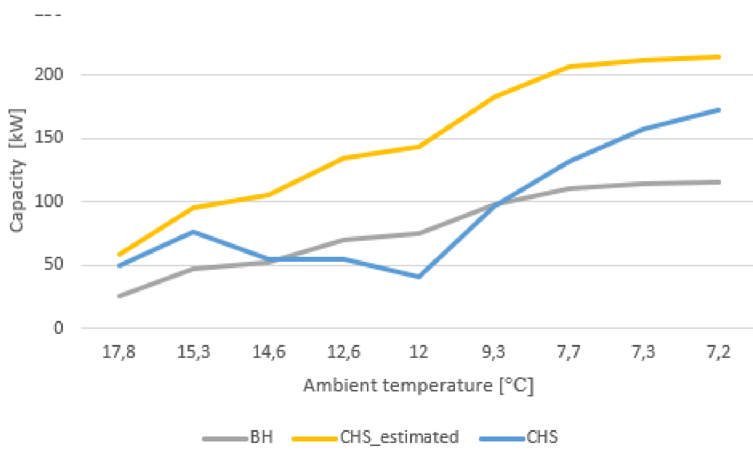


Figure 21: Difference in assumed CHS and real CHS

As shown in the graph the estimated is higher than the reality. The calculated capacity from heat battery is lower than measured at data from 12-14.7$°C$. Every data above 8$°C$ had sunny days.

5.1.4 Summarize waste heat and heat demand

The heat demand in ventilation and space heating was based on the ambient temperature and calculated with Eq. 4.3 and 4.5, shown in Figure 6. Water heating is impulsive and occur in between shifts and calculated with heat demand and capacity for the shower by Eg. 4.7. Table 6 show the results for average yearly heat demand in the systems.

Heat demand to the Central heating is shown in table below

Table 6: Capacity and heat demand distribution in CHS

	Heat battery	Space heat	Water heat	Sum	Enova
Max capacity [kW]	348	252	651	1 251	-
Heat consumption [MWh/year]	573	717	78	1 368	1 478
Heat recovery [MWh/year]	573	491	78	1 142	1 012

Water heating needs the highest capacity and has the lowest heat demand. The capacity appears in short time intervals and depends on how many employees are showering at the same time. The ventilation system has highest heat demand and utilizing waste heat in the ventilation would have the highest energy potential to reduce. The heat demand on Enova report is higher in heat demand and lower at the heat recovery potential. This is as shown in Table 6 due to a generalization of operation time.

An overview is represented by an energy flow chart in Figure 22. The energy flow chart is based on temperature since ventilation and space heating are correlated to ambient temperature.

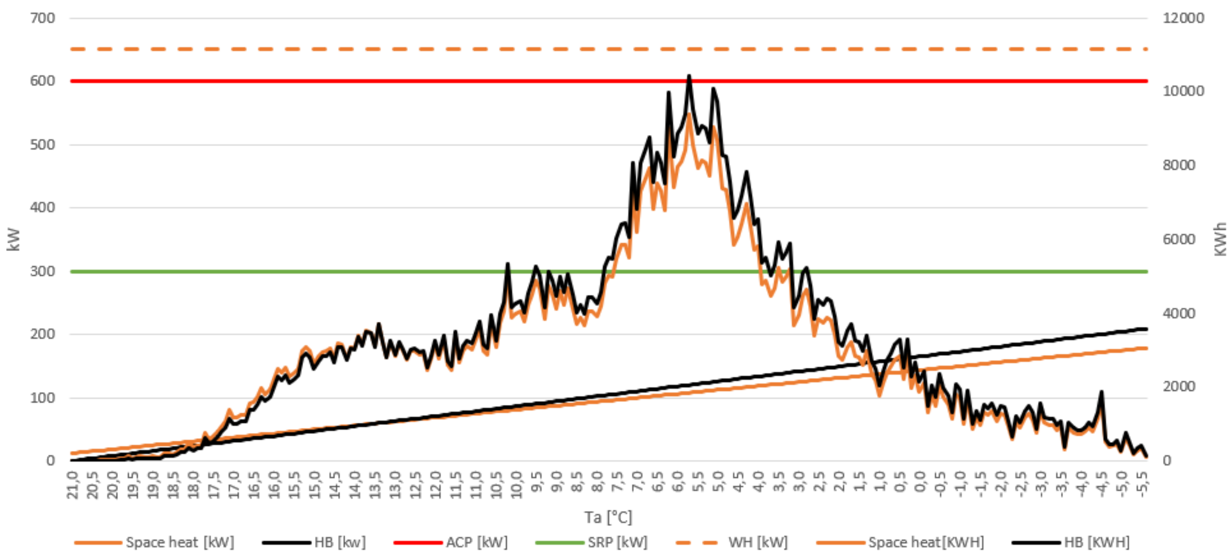


Figure 22: Heat capacity and heat demand in the systems

ACP can reduce the energy demand from ventilation and space heat at the temperature collected. The maximum capacity of water heating needs assistance from the TST. The highest heat demand occurs at 5-6°C, and the heat demand decline in between 0-5$°C$. At temperatures above 0$°C$, 90% of the heat demand is covered, and the CHS is dimensioned at this temperature to not oversize [38].

For utilizing waste heat, the temperatures difference has an impact on potential. Higher temperature difference increases the mass flow and pipe cost. Table 7 shows the temperature difference and the capacity needed at 0°C.

Table 7: Overview of temperatures and capacity

	Waste heat		Heat demand			
	Aluminium Cooling Procsess	Sand Recycling Process	Heat Battery	Battery Heat Exchanger	Space heating	Water heating
Temperature Hot/cold [$°C$]	70/1	32/1	60/17	23/-5	60/23	65/6
Capacity at 0$°C$ [kW]	334-600	300	171	165	144	651

ACP has the highest temperature which would give the lowest pipe cost compared with SRP. Water heating and BHE has the lowest temperatures, but BHE would have a higher temperature with implementing the waste heat due to higher $T_{HE,go}$. The cold side temperature from heat battery and space heating would increase to 30$°C$ as the ambient temperature decreases to -20$°C$. At the waste heat, the temperature has minimum 1$°C$ due to minimum temperature approach and freezing point.

5.2 Case 1: Waste heat to Central Heating System

This section dimensions and calculates the reduced heat demand with utilized waste heat from ACP to CHS. The waste heat is connected to the electrical boiler shown in Figure 14.

The CHS heat the building (ventilation, space heating, and water heating). The mass flow dimensions after space heating and ventilation heating. When water heat increases, the CHS inlet temperature would decrease and the capacity of HE4 increase.

The method for dimension and calculations are in section 4.2 and shown in Table 8.

Table 8: Dimensions and heat transfer

	$CHS_{HB,SH}$ T_a=0°C	$CHS_{HB,SH}$ T_a=-8°C	$CHS_{HB,SH,WH}$ T_a=9°C	Eq. no
$\dot{m}_{HE4,wi}$ [kg/s]	2.2	2.7	3.9	4.10
$T_{HE4,wi}$ [°C]	27	27.8	12.4	4.11
UA [kW/K]	72735	72735	72735	3.10
$\dot{Q}_{El}$ [kW]	0	78	374	3.10
η [%]	99.6	99.6	44	4.17,4.19

The capacity of waste heat utilization is limited by minimum temperature approach due to mass flow at -8°C. The $T_{H4,wi}$ varies and limit the capacity of waste heat utilization.The η at column three is the estimation of water heat recovery.

5.3 Case 2:Battery Heat Exchanger as a cooling tower

This solution transports waste heat from the ACP direct to the BHE after CHS, with implementing a heat exchanger, HE3. The SRP is also being included in case of future opportunities when ACP is not operating, or if ACP should be beneficial elsewhere, or increase the capacity on slow production. The HE3 is dimensioned to maximum capacity for ACP and how it would behave with minimum capacity or with changing to SRP.

Figure 15 shows a detailed version and Figure 23 shows the principle of the solution. Waste heat is sent to the glycol circuit and transported to the HE1 and HE2.

Table 9 shows the results from excel simulation, explained in section 4.5.1 and Aspen Plus, shown in appendix C.1.2.

ACP and SRP are sufficient waste heat sources for dimension reduce the heat demand from ventilation above 99%. At -8 degree the SRP heat 300kW, if heated with HB it would be 227kW, this is caused by the increasing temperature in $T_{H2.gi}$ which decreases the heat recovery in HE2.

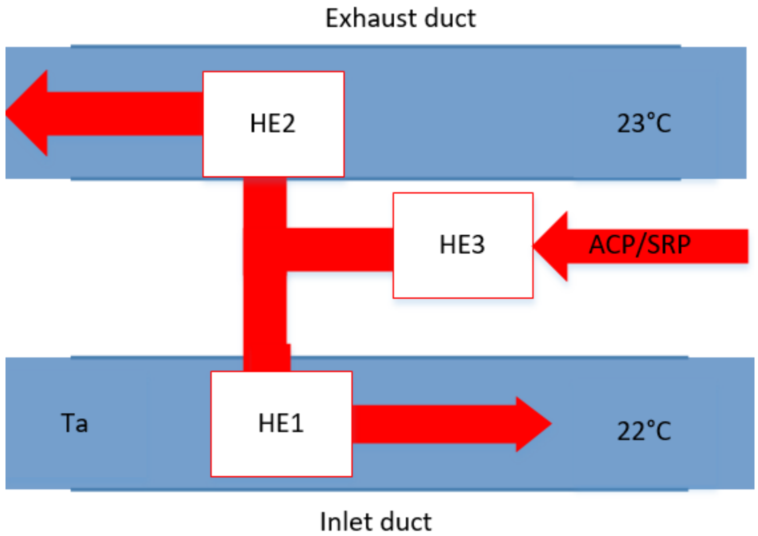


Figure 23: Waste heat sent to the inlet and exhaust duct at BHE

Table 9: ACP and SRP implemented in BHE dimension for heat demand and waste heat cooling

Case 2	ACP	SRP	Ref.
Mass flow waste heat [kg/s]	6	6	4.14
$UA_{3.33}$ $[W/K]$	58 6811	58 681	3.10
$\dot{Q}_{HE3}$ at T_a=22°C [°C]	563/558	107/106	4.5
T_a=600/300 kW [°C]	17.5	-8	4.5
η [%]	100	99.6	4.19

ACP has the capacity to being used as a cooling tower when the ambient temperature is below 17.5°C. If the temperature is above 22°C the process needs to be reduced by 7% either with longer time between the racks or reduced mass as calculated with Eq. 4.2.

The time above 17.5°C is estimated to be 3% based on the data collected from eklima.no. If SRP would be used, it would utilize all heat from the cooling tower at -8°C. From temperature data, this has not occured in the period 2015-2017.

The temperature in the water tank increases with 1°C/min when cooling with 600kw at the time when aluminum cooling is in the tank. This indicates that the water tank should have 65°C before a new rack. Norsk Energi chose not use a the HE4 and rather use valves and replaced the water to reduce cost and neglect the temperature drop in HE4. This solution is shown in appendix C.2

5.4 Economic evaluation

The economic perspective in this report is considering investment cost and savings. Maintenance and operation cost is not evaluated.

The energy savings are based on the amount of waste heat replaced by the electrical boiler in CHS. The energy savings are calculated with Eq. 4.20 and the estimation of investments cost are explained in section 4.7.2. The data is then represented in Table 10 where CT is the cost of installing the Cooling tower. The energy cost at Chassix was confidential, so a price of $0.5 NOK/kWh$ were chosen.

Table 10: Savings, cost and necessary Enova support for solutions

Case	CHS(1)	CHS+BHE (1+2)	CHS+BHE-CT (1+2-CT)
Economic savings [$NOK/year$]	530 000	530 000	530 000
Investment [$NOK/year$]	2 420 000	3 630 000	1 964 000
Enova support [%]	34	56	19

Chassix mentions that economical support from Enova is up to 30%. To manage three years savings the ventilation should be used as a cooling tower. If a cooling tower is not replaced, few advantages to using it without knowing maintenance cost and Case 1 is innstalled.

6 Discussion

The basis of this report is heat demands. The dominating heat demand are from the ventilation systems and space heating, which has the main impact on the natural force, ambient temperature. The ambient temperature is unreliable, which gives uncertainties in heat demand. The heat demand in the three years estimated, had 1.6% difference. However, the next years could be different.

6.1 Heat demand and waste heat

Ambient temperature impacts the heat capacity of the ventilation system and space heating. Degradation on the BHE is not considered, and the ventilation system is 20 years old and close to the coast where air contains salt. Degradation in BHE would increase the heat demand, and since humidity and latent heat is not considered BHE would be increased. Space heating depends on people in the building decreasing or increasing the exhaust air temperature and humidity, which impacts the BHE heat recovery shown in Figure 7. A reduction in BHE capacity increases the heat demand for ventilation. The exhaust air temperature has a significant impact on heat demand and data from this should be collected. This report uses $23°C$ as an average.

The comparison of estimated capacity related to reality was lower, and at some measurements lower than in heat battery. Reasons for this could be: solar irradiance, the TST was used, something was turned off/on, the ambient temperature/outlet is lower/higher. The outlet temperature in the exhaust air is only estimated and is uncertain. Higher outlet temperature reduces the heat demand in heat battery and increases the heat demand in space heaters. However, since all the measurements were lower, the space heating may be lower than estimated. The data diverged most from the data were taken at temperatures which have a lower impact on the total heat demand. There should be more measurements at temperatures below $8°C$.

The Enova method is estimated based on average energy consumption. The type of buildings were offices, and in the Enova report, the ventilation system only operates at $8h/day$, while Chassix uses the offices $24h/day$. The office area is also high compared with some of them in the statistics. This impact gives ambiguous results, but the direct calculation indicates that it should be lower.

The water heating is only based on estimated employees showering. This is estimated numbers and the reality could be different. Employees showering change by season, for example if employees are cycling or begin cycling to work. Water heating is not only based on the shower. It could be include other water uses such as washing hands, dishes, washing. Other use of hot water would increase the heat demand. The method estimated heat recovery could be questioned since it is conservative and employees shower at the same time is uncertain.

For utilizing waste heat, the process needs to operate, and the operation time depends on the production demand. The SRP operates at its most energy efficient conditions and the operation time is based on previous year. The ACP depends on continuity of aluminum cooling demand. Figure 19 shows that the production of aluminum is higher in the winter, which is beneficial for utilizing waste heat to heat since the ambient temperature is lower and power prices are higher from the grid company. In the month with lower aluminum production, it would be beneficial for utilizing the waste heat that production is reduced and continuous. Reduced production may not be optimal for the processes, with utilizing work capacity, or it could be less energy efficient.

6.2 Simulations

The difference in simulation and excel sheet is close to the same. The difference seems to be most in Aspen Plus which has different specific capacity especially in the air. The humidity in the air has not been considered and would have an impact on increasing the specific heat. As shown with the different UA-value in HE1 and HE2 when sensible heat occurs, and water has higher specific heat capacity. Aspen plus uses several iterations which make it more accurate.

6.3 Case 1

The outlet temperature from ACP is uncertain, and this report has estimated it to be constant $70°C$. The temperature before each rack should be $65°C$ which decreases the capacity of HE4, due to temperature difference from ACP and CHS. The temperature could also be higher than $70°C$ dependent on the circulation in the water tank. If the temperature is lower than $70°C$, in some scenarios the heat recovery decreases. Using the alternative method from Norsk Energi with valves instead of HE4, shown in appendix C.2 minimize the reduced heat recovery.

Utilizing waste heat in CHS reduces the capacity to the grid. However, at work shifts the peak from showering at cold ambient temperatures could reach $600kW$. The TST is used after the electric boiler, and the waste heat does not cover all the shower peaks. The TST is less used when waste heat is utilized and compensated with the electric boiler. From the simulation at -8$°C$, the potential heat recovery from heat battery and space heating is close to full capacity. At lower temperatures the heat recovery would not increase, due to waste heat mass flow and the temperature from heat battery and space heat would increase. However, if water to water heating is used the heat recovery increases due to decreased temperatures. In periods of temperatures below -8$°C$ the electrical boiler is used. The weather data collected zero time of temperature below -5$°C$. This could happen and would decrease the percentage of heat recovery.

The cold temperature from space heat and heat battery is uncertain. It has been used constant $35°C$ at space heat and UA method from heat battery. These temperatures are essential due to the design and potential heat transfer if using valves for mass flow is more important.

If Case 1 and Case 2 are installed, it would be cheaper to increase the dimensions, since Case 2 has higher mass flow than Case 1. However, Case 2 would reduce capacity demand from the ventilation system.

6.4 Case 2

Utilizing waste heat direct in the ventilation system has the advantage to deliver heat at low temperatures and can consume high demand of waste heat even if heat demand is neglected. The high consumption made it possible to utilize it as a cooling tower. The system can consume $600kW$ at an ambient temperature below 17.5°C. Above the production needs to be reduced, at $22°C$, the reduction is 7%. The time of temperatures above $17.5°C$ is estimated to be 3% of the year.

The reduction could be achieved by increasing time per rack or decrease mass of aluminum parts. Decreasing the mass of aluminum parts would keep the temperature within temperature ratio. Increasing time would increase the water temperature, which increases the cooling capacity, and may not increase the time. How the regulation of time at the rack is uncertain. The $600kW$, is also uncertain due to the fact that the calculation has not included latent heat and water refilling.

The cooling capacity is calculated based on $70°C$. What temperature is out from the ACP may be lower or higher due to the time of the process. If $70°C$ is maximum the capacity decreases and the solution needs to re-evaluated. At high ambient temperature the humidity increases and specific capacity increases, this has not been evaluated in this report. Another constraint is that the ventilation is turned off in the weekend and needs to be turned on if the ACP could be the function. Also the exhaust air is chosen to be $23°C$ at high ambient temperature. This could be higher and reduce the cooling capacity.

If there should be found other places to utilize waste heat from the production hall to the administration hall, a connection is possible. The connection of SRP to the system is an advantage if the ACP has periods it is not in use, due to waste heat utilization. As Figure 19 shows, the ACP would not operate at full capacity in some month. In these periods SRP could be used if the ACP is operating at full capacity.

The ventilation duct has condensed water due to humidity in the exhaust air if the ambient temperature is low. With using waste heat, water would vaporize increased heat capacity and prevent correction. The impact of high-temperature in the duct and heat exchanger has not been evaluated. Since the mass flow in HE1 would be low due to the increase in temperature the fouling factor would increase. It is essential that the system is operating most efficient in the summer period, so cleaning of heat exchangers should be done before the summer.

The ventilation systems are soon to be changed, and this report was ended too late and the cooling tower had to start building. Therefore this solution misses high investment savings by replacing it.

6.5 Economic

Norsk Energi economic evaluation is used without knowing exact dimensions of the system, and it also includes internal work which may not exceed any cost. The price could be higher or lower, and the multiplications factor to installing BHE is based on a guess. The economical method for down payment is how a company is calculating it, as is taxes cut, the internal rent which also makes it more economical. The price for energy is also different, but real numbers are also different.

7 Conclusion

Utilizing waste heat in CHS within a payback time of three years were based on energy saving and the investment cost. Waste heat recovery was estimated to 1 012-1 142$MWh/year$. With investment cost, the alternative to use waste heat, to preheat the CHS and replace the cooling tower with use of BHE was the most cost-efficient. However, the ACP productions needed to be reduced by 7% when ambient temperatures were above 17.5$°C$. Case 1 was only cost efficient if it could replace the cooling tower investment, with reduced Enova support from 56-19%. Due to time the cooling tower had to be built before this report was ready, and since the ventilation system was 20 years old, it was soon to be rebuilt. The solution of using waste heat only to the CHS needed 34% Enova support. The economic Enova support is individual, but Chassix mentions up to 30% was possible. This results is than 4% more then three years payback time.

7.1 Future work

Future work would be to collect more data from the existing system especially in the CHS and ACP. Since the ACP soon is installed, knowledge of operating time, temperature output and capacity would give an indication of how cost efficient the solution would be. The economic evaluation has to be recalculated with their numbers of prices and handling of investment cost.

To maintain low energy peeks from the CHS and reduce the water heat demand. A method of using grey water in shower to preheat the water at cold side. Change how the TST is used or installing a new TST withing the waste heat circuit to have higher capacity to heat higher mass flow.

Since the pipes are installed from the production hall to the administrative hall, other heat demanded system also could be connected to this utilization.

References

[1] Sintef. *Energisparetiltak i industrien - SINTEF*. https://www.sintef.no/siste-nytt/energisparetiltak-i-industrien/. (Accessed on 04/12/2018). Jan. 2014.

[2] Clemens Forman et al. "Estimating the global waste heat potential". In: *Renewable and Sustainable Energy Reviews* 57 (2016), pp. 1568–1579. ISSN: 1364-0321. DOI: https://doi.org/10.1016/j.rser.2015.12.192. URL: http://www.sciencedirect.com/science/article/pii/S1364032115015750.

[3] Janez Petek, Peter Glavi, and Anja Kostevek. "Comprehensive approach to increase energy efficiency based on versatile industrial practices". In: *Journal of Cleaner Production* 112 (2016), pp. 2813–2821.

[4] Sharifah Rafidah Wan Alwi et al. *Sustainability engineering for the future*. 2014.

[5] Adriano Desideri et al. "Experimental comparison of organic fluids for low temperature ORC (organic Rankine cycle) systems for waste heat recovery applications". In: *Energy* 97 (2016), pp. 460–469.

[6] Feng Huang et al. "Heat recovery potentials and technologies in industrial zones". In: *Journal of the Energy Institute* (2016).

[7] Hao Fang et al. "Industrial waste heat utilization for low temperature district heating". In: *Energy policy* 62 (2013), pp. 236–246.

[8] NVE. *Energi og effekt - NVE*. https://www.nve.no/energibruk-og-effektivisering/energibruk-i-norge/energi-og-effekt/. (Accessed on 04/25/2018). June 2016.

[9] Hegnar.no. *Har stromnettet kapasitet til elektriske biler, busser og ferger? - hegnar.no*. http://www.hegnar.no/Nyheter/Energi/2017/10/Har-stroemnettet-kapasitet-til-elektriske-biler-busser-og-ferger. (Accessed on 04/25/2018). Oct. 2017.

[10] Energi Norge. *Effekttariffer pa horing*. https://www.energinorge.no/fagomrader/stromnett/nyheter/2017/effekttariffer-pa-horing/. (Accessed on 04/25/2018). Nov. 2017.

[11] Enova. *Untitled*. https://webcache.googleusercontent.com/search?q=cache:PT1Su0ltlREJ:https://www.enova.no/download/%3FobjectPath%3Dupload_images/.pdf+&cd=10&hl=en&ct=clnk&gl=no. (Accessed on 04/12/2018). Feb. 2009.

[12] Line Gjerstad. *Denne temperaturen bor du ha inne na - Aftenposten*. https://www.aftenposten.no/bolig/Denne-temperaturen-bor-du-ha-inne-na-9102b.html. (Accessed on 04/25/2018). Feb. 2016.

[13] Enok og Innklima (AEI) Akerhus. *Innetemperatur*. https://www.enok.no/enokguiden/02_1.html. (Accessed on 04/25/2018).

[14] *Hvor varmt skal varmtvannet vaere? - DinSide*. https://www.dinside.no/bolig/hvor-varmt-skal-varmtvannet-vaere/61310529. (Accessed on 04/25/2018). Oct. 2012.

[15] KLM. *Microsoft Word - ENGINEERING_DESIGN_GUIDELINES_heat_exchanger_Rev04all.docm*. http://www.klmtechgroup.com/PDF/EGD2/ENGINEERING_DESIGN_GUIDELINES_heat_exchangers_sizing_and_selection_rev_web.pdf. (Accessed on 11/18/2017).

[16] Goran Hed and Rickard Bellander. "Mathematical modelling of PCM air heat exchanger". In: *Energy and Buildings* 38.2 (2006), pp. 82–89.

[17] *Sketch of a finned oval tube heat exchanger element.* https://www.researchgate.net/figure/233288303_fig1_Figure-1-Sketch-of-a-finned-oval-tube-heat-exchanger-element. (Accessed on 11/15/2017). May 2016.

[18] Thermaxxjackets. *Plate and Frame Heat Exchangers Explained.* https://www.thermaxxjackets.com/plate-and-frame-heat-exchangers-explained/. (Accessed on 05/29/2018).

[19] JBJ. *Gasketed plate heat exchangers.* https://www.jbj.co.uk/plate-heat-exchangers.html. (Accessed on 11/15/2017).

[20] schoeck. *Heat Loss in Buildings - Schock Ltd.* https://www.schoeck.co.uk/en-gb/heat-loss-in-buildings. (Accessed on 04/23/2018).

[21] The Engineering Toolbox. *Heat Loss from Buildings.* https://www.engineeringtoolbox.com/heat-loss-buildings-d_113.html. (Accessed on 04/23/2018).

[22] R. H. Perry and D. Green. "Perry's Chemical Engineers' Handbook". In: 7th. Singapore: McGraw-Hill International Editions, 1984, 10:11–25.

[23] *Arithmetic and Logarithmic Mean Temperature Differences in Heat Exchangers.* https://www.engineeringtoolbox.com/arithmetic-logarithmic-mean-temperature-d_436.html. (Accessed on 11/13/2017).

[24] F. P. Incropera et al. "Principles of Heat and Mass transfer". In: 7th. Singapore: John Wiley & Sons Inc, 2013. Chap. 11.

[25] Engineers Edge. *Overall Heat Transfer Coeffisient Chart.* URL: http://www.engineersedge.com/thermodynamics/overall%5C_heat%5C_transfer-table.htm.

[26] D. Zijdemans. "Vannbaserte oppvarings- og kjolesystemer". In: Oslo: Skarland Press AS, 2012, pp. 299–311.

[27] FSK Warnakulasuriya and WM Worek. "Heat transfer and pressure drop properties of high viscous solutions in plate heat exchangers". In: *International Journal of Heat and Mass Transfer* 51.1 (2008), pp. 52–67.

[28] I. Martinez. "Heat and Mass convection. Boundary Layer flow". In: (2015), p. 20. URL: http://webserver.dmt.upm.es/~isidoro/bk3/c12/Heat%5C%20convection.%5C%20Boundary%5C%20layer%5C%20flow.pdf.

[29] Mostafa M Awad. "Fouling of heat transfer surfaces". In: *Heat Transfer-Theoretical Analysis, Experimental Investigations and Industrial Systems.* InTech, 2011.

[30] VariantVVS. *VariantVVs.indd.* http://byggebolig.no/imageoriginals/.pdf. (Accessed on 05/23/2018).

[31] Yr. *Yr Datosok Farsund (Vest-Agder).* https://www.yr.no/sted/Norge/Vest-Agder/Farsund/Farsund/almanakk.html?dato\unhbox\voidb@x\bgroup\let\unhbox\voidb@x\setbox\@tempboxa\hbox{2\global\mathchardef\accent@spacefactor\spacefactor}\accent222\egroup\spacefactor\accent@spacefactor018-04-23. (Accessed on 05/24/2018). Apr. 2018.

[32] Din side. *Dusjing: Hvor mye koster det a ta en dusj? - DinSide.* https://www.dinside.no/bolig/hvor-mye-koster-det-a-ta-en-dusj/60988994. (Accessed on 04/19/2018). 2016.

[33] NVE. *Grunnvann i Norge - NVE*. https://www.nve.no/hydrologi/grunn-og-markvann/grunnvann-i-norge/. (Accessed on 04/19/2018). July 2017.

[34] Enova. *Bygningsnettverkets energistatistikk 2004 - Enova*. https://www.enova.no/download/?objectPath=up Tabell 3.2. (Accessed on 05/19/2018). 2004.

[35] Aspen Plus. *Aspen Plus - Process Simulation for Chemicals*. https://www.aspentech.com/products/engineering/aspen-plus. (Accessed on 05/26/2018).

[36] Solomon Gebreyohannes, Brian J. Neely, and Khaled A.M. Gasem. "One-parameter modified nonrandom two-liquid (NRTL) activity coefficient model". In: *Fluid Phase Equilibria* 379 (2014), pp. 196–205. ISSN: 0378-3812. DOI: https://doi.org/10.1016/j.fluid.2014.07.027. URL: http://www.sciencedirect.com/science/article/pii/S0378381214004129.

[37] Enova. *Om organisasjonen Enova*. https://www.enova.no/om-enova/om-organisasjonen/. (Accessed on 05/28/2018).

[38] Sintef. *Moler bruk av varmtvann - SINTEF*. https://www.sintef.no/siste-nytt/maler-bruk-av-varmtvann/. (Accessed on 05/22/2018). Oct. 2017.

Appendices

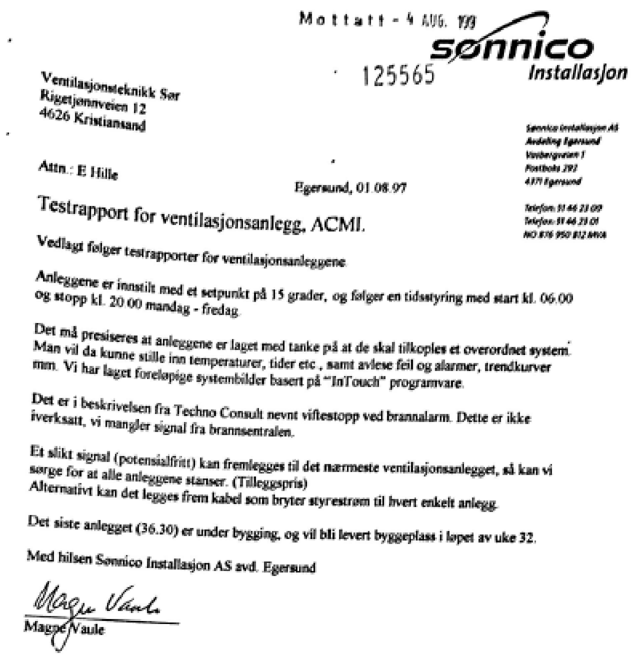
A Existing

A.1 Space heating

Beregninger av arealer målt til utvendig konstruksjon				
Beskrivelse	Lengde m	Bredde m	Areal m²	Gr.fl. M²
Hall	145	133	19285	19285
Vannbehandling	30	8	240	240
Kompressor	18	8	144	144
Trafo	18	7.5	135	135
Trafo	18	7.5	135	135
CKO 2-etg.	25	18	450	
Lager 2-etg.	25	25	625	
Kontorer 2-etg.	18	9	162	
Treningsrom	25	12	300	
3-etg.	55	25	1375	
			0	
Kandtlager	51.5	18	927	927
Kantine	52	18	936	936
Gang	18	3	54	54
			0	
Kontorer	30	12	360	360
Kontorer	68	18	1224	1224
Vent. På tak	22	5	110	
			0	
Kjemikalielager	14	9	126	126
Brannstasjon	6	5.5	33	33
Stigerør/sandblåseverksted	16	6.5	104	104
		Sum	**26725**	**23703**

Figure A.24: The area of building based on outside construction

A.2 Ventilation system



Mottatt - 4 AUG. 1997

sønnico Installasjon

125565

Ventilasjonsteknikk Sør
Rigetjønnveien 12
4626 Kristiansand

Sønnico Installasjon AS
Avdeling Egersund
4371 Egersund

Telefon: 51 46 23 00
Telefax: 51 46 23 01

Attn.: E Hille

Egersund, 01.08.97

Testrapport for ventilasjonsanlegg, ACMI.

Vedlagt følger testrapporter for ventilasjonsanleggene

Anleggene er innstilt med et setpunkt på 15 grader, og følger en tidsstyring med start kl. 06.00 og stopp kl. 20.00 mandag - fredag.

Det må presiseres at anleggene er laget med tanke på at de skal tilkoples et overordnet system. Man vil da kunne stille inn temperaturer, tider etc., samt avlese feil og alarmer, trendkurver mm. Vi har laget foreløpige systembilder basert på "InTouch" programvare.

Det er i beskrivelsen fra Techno Consult nevnt viftestopp ved brannalarm. Dette er ikke iverksatt, vi mangler signal fra brannsentralen.

Et slikt signal (potensialfritt) kan fremlegges til det nærmeste ventilasjonsanlegget, så kan vi sørge for at alle anleggene stanser. (Tilleggspris)
Alternativt kan det legges frem kabel som bryter styrestrøm til hvert enkelt anlegg.

Det siste anlegget (36.30) er under bygging, og vil bli levert byggeplass i løpet av uke 32.

Med hilsen Sønnico Installasjon AS avd. Egersund

Magne Vaule

Magne Vaule

Figure A.25: Ventilation's systems operation time

B Method

B.1 Aluminum production

Month	January	February	Mars	April	May	June	July	August
Kg of aluminum	1321	1089	1349	861	750	987	898	680

Table 11: Mass of aluminum per month

B.2 Ambient temperature data

Ta	år 2015	år 2016	år2017	år avg	Ta	år 2015	år 2016	år2017	år avg	Ta	år 2015	år 2016	år2017	år avg
21,0	0	1	0	0,3	17,9	13	21	7	13,7	14,8	33	73	67	57,7
20,9	0	1	0	0,3	17,8	13	24	3	13,3	14,7	48	59	65	57,3
20,8	0	1	0	0,3	17,7	17	35	18	23,3	14,6	53	47	75	58,3
20,7	0	1	0	0,3	17,6	18	24	12	18,0	14,5	36	61	59	52,0
20,6	0	2	0	0,7	17,5	15	31	11	19,0	14,4	34	73	71	59,3
20,5	0	1	1	0,7	17,4	22	33	15	23,3	14,3	57	67	50	58,0
20,4	0	4	0	1,3	17,3	22	43	15	26,7	14,2	34	60	58	50,7
20,3	2	3	0	1,7	17,2	28	38	24	30,0	14,1	54	53	60	55,7
20,2	2	3	1	2,0	17,1	24	54	34	37,3	14,0	48	70	45	54,3
20,1	3	2	0	1,7	17,0	25	43	26	31,3	13,9	55	60	64	59,7
20,0	4	2	0	2,0	16,9	27	36	31	31,3	13,8	45	54	65	54,7
19,9	3	4	0	2,3	16,8	20	44	34	32,7	13,7	53	55	75	61,0
19,8	6	7	0	4,3	16,7	30	39	27	32,0	13,6	63	45	69	59,0
19,7	1	6	2	3,0	16,6	26	62	30	39,3	13,5	58	32	67	52,3
19,6	9	8	2	6,3	16,5	25	49	43	39,0	13,4	61	55	70	62,0
19,5	2	6	2	3,3	16,4	31	57	39	42,3	13,3	50	53	56	53,0
19,4	7	6	1	4,7	16,3	29	62	50	47,0	13,2	45	43	49	45,7
19,3	7	8	2	5,7	16,2	25	56	47	42,7	13,1	39	44	73	52,0
19,2	6	6	0	4,0	16,1	31	58	46	45,0	13,0	52	31	54	45,7
19,1	3	7	2	4,0	16,0	43	50	59	50,7	12,9	39	46	67	50,7
19,0	5	11	1	5,7	15,9	40	69	61	56,7	12,8	41	40	59	46,7
18,9	6	8	0	4,7	15,8	42	67	51	53,3	12,7	43	36	48	42,3
18,8	4	6	3	4,3	15,7	34	73	59	55,3	12,6	41	33	63	45,7
18,7	5	14	3	7,3	15,6	38	60	50	49,3	12,5	46	28	62	45,3
18,6	7	15	1	7,7	15,5	38	55	60	51,0	12,4	51	22	58	43,7
18,5	5	14	1	6,7	15,4	46	55	56	52,3	12,3	51	31	48	43,3
18,4	4	17	4	8,3	15,3	46	71	70	62,3	12,2	38	20	51	36,3
18,3	7	23	4	11,3	15,2	62	74	54	63,3	12,1	50	25	47	40,7
18,2	10	14	8	10,7	15,1	39	71	71	60,3	12,0	60	21	56	45,7
18,1	15	26	6	15,7	15,0	29	63	66	52,7	11,9	41	27	52	40,0
18,0	10	21	5	12,0	14,9	44	69	56	56,3	11,8	62	20	59	47,0

Ta	år 2015	år 2016	år2017	år avg	Ta	år 2015	år 2016	år2017	år avg	Ta	år 2015	år 2016	år2017	år avg
11,7	50	16	45	37,0	8,6	70	34	34	46,0	5,5	81	74	63	72,7
11,6	39	14	51	34,7	8,5	53	46	24	41,0	5,4	86	63	73	74,0
11,5	57	22	61	46,7	8,4	67	33	28	42,7	5,3	78	70	71	73,0
11,4	50	23	37	36,7	8,3	55	30	35	40,0	5,2	95	58	55	69,3
11,3	53	26	44	41,0	8,2	60	41	31	44,0	5,1	98	82	62	80,7
11,2	52	31	44	42,3	8,1	50	42	39	43,7	5,0	102	62	68	77,3
11,1	51	33	38	40,7	8,0	50	53	22	41,7	4,9	89	58	49	65,3
11,0	56	31	45	44,0	7,9	49	56	28	44,3	4,8	77	60	57	64,7
10,9	66	25	51	47,3	7,8	55	71	26	50,7	4,7	70	51	56	59,0
10,8	57	28	33	39,3	7,7	46	84	28	52,7	4,6	62	45	46	51,0
10,7	47	27	39	37,7	7,6	44	71	41	52,0	4,5	60	49	48	52,3
10,6	73	32	40	48,3	7,5	52	76	42	56,7	4,4	54	56	55	55,0
10,5	58	21	39	39,3	7,4	45	76	59	60,0	4,3	62	56	61	59,7
10,4	77	26	40	47,7	7,3	48	79	52	59,7	4,2	56	47	60	54,3
10,3	87	24	42	51,0	7,2	54	57	56	55,7	4,1	52	45	48	48,3
10,2	99	27	62	62,7	7,1	57	88	76	73,7	4,0	54	55	38	49,0
10,1	85	20	40	48,3	7,0	55	66	65	62,0	3,9	44	28	48	40,0
10,0	81	24	42	49,0	6,9	61	81	76	72,7	3,8	44	43	35	40,7
9,9	79	32	38	49,7	6,8	56	85	86	75,7	3,7	33	38	40	37,0
9,8	68	27	42	45,7	6,7	75	81	78	78,0	3,6	40	34	42	38,7
9,7	86	28	39	51,0	6,6	52	68	80	66,7	3,5	38	47	44	43,0
9,6	82	38	42	54,0	6,5	62	72	85	73,0	3,4	38	47	34	39,7
9,5	86	39	49	58,0	6,4	43	87	81	70,3	3,3	29	56	37	40,7
9,4	86	36	43	55,0	6,3	50	70	75	65,0	3,2	43	41	42	42,0
9,3	64	29	42	45,0	6,2	76	82	99	85,7	3,1	28	26	35	29,7
9,2	78	40	47	55,0	6,1	66	67	78	70,3	3,0	31	33	31	31,7
9,1	70	35	51	52,0	6,0	90	60	75	75,0	2,9	33	40	34	35,7
9,0	64	35	43	47,3	5,9	83	64	81	76,0	2,8	37	35	38	36,7
8,9	80	36	41	52,3	5,8	89	68	78	78,3	2,7	27	33	39	33,0
8,8	69	36	38	47,7	5,7	94	79	87	86,7	2,6	27	19	34	26,7
8,7	74	46	37	52,3	5,6	90	59	87	78,7	2,5	21	25	44	30,0

Ta	år 2015	år 2016	år2017	år avg	Ta	år 2015	år 2016	år2017	år avg	Ta	år 2015	år 2016	år2017	år avg
2,4	17	26	44	29,0	-0,7	6	8	17	10,3	-3,8	2	9	3	4,7
2,3	18	25	47	30,0	-0,8	6	5	12	7,7	-3,9	1	8	4	4,3
2,2	22	19	47	29,3	-0,9	7	12	17	12,0	-4,0	3	8	2	4,3
2,1	20	26	33	26,3	-1,0	8	7	19	11,3	-4,1	0	9	5	4,7
2,0	15	16	34	21,7	-1,1	7	7	6	6,7	-4,2	0	12	4	5,3
1,9	16	21	25	20,7	-1,2	8	14	11	11,0	-4,3	0	10	4	4,7
1,8	19	14	37	23,3	-1,3	4	6	7	5,7	-4,4	0	11	9	6,7
1,7	17	32	24	24,3	-1,4	6	11	6	7,7	-4,5	0	17	11	9,3
1,6	15	18	31	21,3	-1,5	5	10	4	6,3	-4,6	0	4	5	3,0
1,5	13	19	31	21,0	-1,6	5	12	9	8,7	-4,7	0	4	3	2,3
1,4	9	29	20	19,3	-1,7	3	14	7	8,0	-4,8	0	5	2	2,3
1,3	15	29	22	22,0	-1,8	4	15	7	8,7	-4,9	0	8	0	2,7
1,2	14	25	13	17,3	-1,9	6	9	6	7,0	-5,0	0	3	1	1,3
1,1	12	20	16	16,0	-2,0	5	14	6	8,3	-5,1	0	11	0	3,7
1,0	7	23	9	13,0	-2,1	2	16	6	8,0	-5,2	0	7	0	2,3
0,9	14	22	10	15,3	-2,2	5	10	3	6,0	-5,3	0	3	0	1,0
0,8	8	29	15	17,3	-2,3	4	4	3	3,7	-5,4	0	5	0	1,7
0,7	16	29	10	18,3	-2,4	3	14	3	6,7	-5,5	0	6	0	2,0
0,6	10	37	12	19,7	-2,5	0	13	4	5,7	-5,6	0	2	0	0,7
0,5	9	33	19	20,3	-2,6	1	17	4	7,3	-5,7	0	0	0	0,0
0,4	13	14	20	15,7	-2,7	2	13	9	8,0					
0,3	22	21	18	20,3	-2,8	2	13	6	7,0					
0,2	8	20	14	14,0	-2,9	1	8	5	4,7					
0,1	5	28	16	16,3	-3,0	1	19	5	8,3					
0,0	4	16	19	13,0	-3,1	1	16	2	6,3					
-0,1	9	10	25	14,7	-3,2	0	13	5	6,0					
-0,2	7	4	16	9,0	-3,3	1	14	3	6,0					
-0,3	9	13	15	12,3	-3,4	1	10	4	5,0					
-0,4	2	8	21	10,3	-3,5	0	13	4	5,7					
-0,5	6	12	24	14,0	-3,6	0	5	1	2,0					
-0,6	15	6	14	11,7	-3,7	1	15	0	5,3					

B.3 CHS data

Date [dd.mm.yyyy-xx:xx]	Weather [-]	CHS [kW]	T_a [degree]	SH [kW]	Data no. [-]
25.04.2018-09:03	Sun	173	7.2	65	1
04.05.2018-11:03	Fog	157	7.3	50	2
24.04.2018-14:45	Cloudy	132	7.7	28	3
11.05.2018-09:00	Sun	97	9.3	6	4
07.05.2018-08:00	Sun	95	12	-29	5
07.05.2018-12:00	Sun	50	12.7	-11	6
08.05.2018-08:00	Sun	55	14.6	5	7
09.05.2018-08:00	Sun	76	15.3	32	8
08.05.2018-12:45	Sun	50	17.9	26	9

Table 12: Show the data and calculation of space heat

Tabell 1 – Spillvarme fra vannbad til varmeanlegg

Spillvarme til varmeanlegg		
Investeringskalkyle		
Ventilasjonsanlegg	kr	-
Riving/ombygging av aggregat	kr	-
Kjølemaskin	kr	-
Rørarbeid	kr	800 000
Pumper	kr	80 000
Elektro/automatikk	kr	200 000
Styring/regulering	kr	200 000
Instrumentering og reguleringsventiler	kr	70 000
Stengeventiler	kr	40 000
Prosjektering og prosjektledelse (25 %)	kr	750 000
Uspesifisert/usikkerhet (20 %)	kr	280 000
Total investering	**kr**	**2 420 000**
Energibesparelse		
Strømbesparelse, varme	kWh	1 000 000

Figure B.26: Norsk Energi estimation of installation cost to Alt 1

TRATEC

TRATEC TEKNIKKEN AS

Benteler Automotive Farsund AS
Lundevågen
4550 Farsund

BUDSJETTPRIS RØR OG ELEKTRO REV 1

Deres ref.: Sverre Gardøl **Vår ref.:** Odd Arne Reiersen Farsund, 18.9.2017

Sak/Prosjekt: **Varmegjennvindingsanlegg**

Vi takker for Deres forespørsel og har med dette gleden av å gi budsjettpris på rør og elektro for nevnte prosjekt.

Vi ønsker å levere et produkt med kvalitet til en konkurransedyktig pris

Vi har beregnet følgende oppsett:

Alternativ 1, kun dump av varme fra herdebad

Rør

1 stk	sirkulasjonspumpe Grundfos TPE 3 80-180 mellom veksler og akkumulator
1 stk	sirkulasjonspumpe Grundfos Magna 3 mellom avtrekksvifte og akkumulator
1 stk	sirkulasjonspumpe Grundfos TPE 3 80-180 mellom akkumulator og glykolveklser
1 stk	sirkulasjonspumpe Grundfos TPE 3 80-180 mellom glykolveklser og herdekar
3 stk	5m3 akkumulatortanker
12 m	DN 200 rør fra kjølekar til veksler
12 m	DN 200 rør fra veksler til akkumulator
24 m	DN 200 rør fra akkumulator til glykolveksler på messanin
24 m	DN 200 rør fra glykolveksler på messanin til tørrkjøler over tak
1 stk	Plateveklser 600 kw
1 stk	Tørrkjøler 600 kw TTC
1 stk	Glykol påfyllingsarrangement

Delsum 1 146 566 kr eks mva

Elektro/Automatikk

5 stk	Oppkopling av sirkulasjonspumper
3 stk	Oppkopling av energimålere fra pumper
1 stk	Tilførsel automatikktavle
1 stk	Automatikktavle
1 stk	Oppkopling av tørrkjøler
3 stk	Oppkoplig av temptransmittere
1 stk	Oppkopling av filterspyling
8 stk	Temperaturmåler PT 1000
1 stk	Programmering og dokumentasjon

Delsum 136 750 kr eks mva

Sum alternativ 1 1 666 750 kr eks mva

Figure B.27: Tratec estimated investment cost for installing cooling tower

B.4 Economic

C Results

C.1 Simulation

C.1.1 Aspen Plus: Case 1

	CHS Ta=0				CHS Ta=-8				CHS water heat = 24 employee, Ta= 9			
Name	HB	HE1	HE2	HE4	HB	HE1	HE2	HE4	HB	HE1	HE2	HE4
Hot side property method	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL
Hot side use true species approach for electrolytes	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES
Hot side free-water phase properties method	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA
Hot side water solubility method	3	3	3	3	3	3	3	3	3	3	3	3
Cold side property method	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL
Cold side use true species approach for electrolytes	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES
Cold side free-water phase properties method	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA
Cold side water solubility method	3	3	3	3	3	3	3	3	3	3	3	3
Constant UA [J/sec-K]	10070	33592	33592	72735	10070	33592	33592	72735	10070	33592	33592	72735
Minimum temperature approach [C]	0,0001	1	1	0,00001	0,0001	1	1	0,00001	0,0001	1	1	0,00001
Hot side outlet pressure [bar]	0	0	0	0	0	0	0	0	0	0	0	0
Cold side outlet pressure [bar]	0	0	0	0	0	0	0	0	0	0	0	0
Inlet hot stream temperature [C]	65,0	15,5	23,0	70,0	65,0	12,9	23,0	70,0	65,0	18,4	23,0	70,0
Inlet hot stream pressure [bar]	3,0	3,0	1,0	3,0	3,0	3,0	1,0	3,0	3,0	3,0	1,0	3,0
Inlet hot stream vapor fraction	0,0	0,0	1,0	0,0	0,0	0,0	1,0	0,0	0,0	0,0	1,0	0,0
Outlet hot stream temperature [C]	15,8	7,4	11,4	31,9	18,5	2,2	7,5	29,1	16,5	13,5	15,9	12,9
Outlet hot stream pressure [bar]	3,0	3,0	1,0	3,0	3,0	3,0	1,0	3,0	3,0	3,0	1,0	3,0
Outlet hot stream vapor fraction	0,0	0,0	1,0	0,0	0,0	0,0	1,0	0,0	0,0	0,0	1,0	0,0
Inlet cold stream temperature [C]	11,5	0,0	7,4	27,0	7,7	-8,0	2,2	27,8	15,9	9,0	13,5	12,4
Inlet cold stream pressure [bar]	1,0	1,0	3,0	3,0	1,0	1,0	3,0	3,0	1,0	1,0	3,0	3,0
Inlet cold stream vapor fraction	1,0	1,0	0,0	0,0	1,0	1,0	0,0	0,0	1,0	1,0	0,0	0,0
Outlet cold stream temperature [C]	22,0	11,5	15,5	66,0	22,3	7,7	12,9	58,0	22,0	15,9	18,4	42,0
Outlet cold stream pressure [bar]	1,0	1,0	3,0	3,0	1,0	1,0	3,0	3,0	1,0	1,0	3,0	3,0
Outlet cold stream vapor fraction	1,0	1,0	0,0	0,0	1,0	1,0	0,0	0,0	1,0	1,0	0,0	0,0
Heat duty [cal/sec]	40377,8	43895,7	44464,6	77021,1	55832,7	60123,5	59480,6	82707,2	23313,5	26458,3	27185,1	114477,6
Calculated heat duty [cal/sec]	40377,8	43895,7	44464,6	77021,1	55832,7	60123,5	59480,6	82707,2	23313,5	26458,3	27185,1	114477,6
Required exchanger area [sqm]	11,8	39,5	39,5	85,6	11,8	39,5	39,5	85,6	11,9	39,5	39,5	85,6
Actual exchanger area [sqm]	11,8	39,5	39,5	85,6	11,8	39,5	39,5	85,6	11,8	39,5	39,5	85,6
Average U (Dirty) [cal/sec-sqcm-K]	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
UA [cal/sec-K]	2405,2	8023,1	8021,3	17372,5	2404,4	8023,4	8021,7	17372,4	2406,0	8021,5	8023,4	17372,5
LMTD (Corrected) [C]	16,8	5,5	5,5	4,4	23,2	7,5	7,4	4,8	9,7	3,3	3,4	6,6
LMTD correction factor	1	1	1	1	1	1	1	1	1	1	1	1

Mixer						
Name	B1	B2	B1	B2	B1	B2
Property method	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL
Use true species approach for electrolytes	YES	YES	YES	YES	YES	YES
Free-water phase properties method	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA
Water solubility method	3	3	3	3	3	3
Specified pressure [bar]	0	0	0	0	0	0
Outlet temperature [C]	26,9932804	26,9932804	27,7961115	27,7961115	27,428461	12,4326204
Calculated outlet pressure [bar]	3	3	3	3	3	3
First liquid /Total liquid	1	1	1	1	1	1

Figure C.28: Simulation of CHS

C.1.2 Aspen Plus: Case 2

	Ta=0			Ta=-8			Ta=22			THE3,wi=65		
Name	HE1	HE2	HE3	HE1	HE2	HE3	HE1	HE2	HE3	HE1	HE2	HE3
Hot side property method	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL
Hot side use true species approach for electrolytes	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES
Hot side free-water phase properties method	STEAM-T/	STEAM-TA	STEAM-TA	STEAM-T/	STEAM-T/	STEAM-TA	STEAM-T/	STEAM-T/	STEAM-TA	STEAM-T/	STEAM-T/	STEAM-TA
Hot side water solubility method	3	3	3	3	3	3	3	3	3	3	3	3
Cold side property method	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL	NRTL
Cold side use true species approach for electrolyte	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES	YES
Cold side free-water phase properties method	STEAM-T/	STEAM-TA	STEAM-TA	STEAM-T/	STEAM-T/	STEAM-TA	STEAM-T/	STEAM-T/	STEAM-TA	STEAM-T/	STEAM-T/	STEAM-TA
Cold side water solubility method	3	3	3	3	3	3	3	3	3	3	3	3
Constant UA [J/sec-K]	7259	7519	4148	7259	7519	4148	7292	7292	4148	7259	7519	4148
Minimum temperature approach [C]	1	1	2	1	1	2	1	1	1	1	1	0,1
Inlet hot stream temperature [C]	29,9	23,0	70,0	28,3	23,0	70,0	39,9	39,9	70,0	28,9	23,0	65,0
Inlet hot stream pressure [bar]	3,0	1,0	3,0	3,0	1,0	3,0	3,0	3,0	3,0	3,0	1,0	3,0
Inlet hot stream vapor fraction	0,0	1,0	0,0	0,0	1,0	0,0	0,0	0,0	0,0	0,0	1,0	0,0
Outlet hot stream temperature [C]	14,5	16,5	21,1	9,5	12,7	18,8	1,0	31,8	33,4	15,0	16,9	21,1
Outlet hot stream pressure [bar]	3,0	1,0	3,0	3,0	1,0	3,0	3,0	3,0	3,0	3,0	1,0	3,0
Outlet hot stream vapor fraction	0,0	1,0	0,0	0,0	1,0	0,0	0,0	0,0	0,0	0,0	1,0	0,0
Inlet cold stream temperature [C]	0,0	14,5	18,9	-8,0	9,5	16,6	0,0	24,0	31,8	2,0	15,0	19,2
Inlet cold stream pressure [bar]	1,0	3,0	3,0	1,0	3,0	3,0	1,0	1,0	3,0	1,0	3,0	3,0
Inlet cold stream vapor fraction	1,0	0,0	0,0	1,0	0,0	0,0	1,0	1,0	0,0	1,0	0,0	0,0
Outlet cold stream temperature [C]	22,7	18,9	29,9	19,6	16,6	28,0	0,0	36,1	39,9	22,4	19,2	29,0
Outlet cold stream pressure [bar]	1,0	3,0	3,0	1,0	3,0	3,0	1,0	1,0	3,0	1,0	3,0	3,0
Outlet cold stream vapor fraction	1,0	0,0	0,0	1,0	0,0	0,0	1,0	1,0	0,0	1,0	0,0	0,0
Heat duty [cal/sec]	18088,9	5202,0	12795,5	21937,8	8223,1	13366,6	0,5	9644,8	40,4	16276,5	4869,2	11449,2
Calculated heat duty [cal/sec]	18088,9	5202,0	12795,5	21937,8	8223,1	13366,6	0,5	9644,8	40,4	16276,5	4869,2	11449,2
Required exchanger area [sqm]	8,5	8,8	4,9	8,5	8,8	4,9	0,0	8,6	4,9	8,5	8,8	4,9
Actual exchanger area [sqm]	8,5	8,8	4,9	8,5	8,8	4,9	8,6	8,6	4,9	8,5	8,8	4,9
Average U (Dirty) [cal/sec-sqcm-K]	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
Average U (Clean)												
UA [cal/sec-K]	1733,2	1794,0	989,5	1733,9	1795,2	988,8	0,0	1741,4	988,7	1733,1	1795,5	990,7
LMTD (Corrected) [C]	10,4	2,9	12,9	12,7	4,6	13,5	10,6	5,5	9,8	9,4	2,7	11,6
LMTD correction factor	1	1	1	1	1	1	1	1	1	1	1	1
Number of shells in series	1	1	1	1	1	1	1	1	1	1	1	1
Number of shells in parallel												

Mixer		Ta=-8	Ta=22	THE3,wi=65
Name	B12	B12	B12	B12
Property method	NRTL	NRTL	NRTL	NRTL
Use true species approach for electrolytes	YES	YES	YES	YES
Free-water phase properties method	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA
Water solubility method	3	3	3	3
Temperature estimate [C]	16	16	14	16
Outlet temperature [C]	14,478171	9,5234006	39,895392	15,025944
Calculated outlet pressure [bar]	3	3	3	3
First liquid /Total liquid	1	1	1	1

FSplit				
Name	B1	B1	S1	B1
Property method	NRTL	NRTL	NRTL	NRTL
Use true species approach for electrolytes	YES	YES	YES	YES
Free-water phase properties method	STEAM-TA	STEAM-TA	STEAM-TA	STEAM-TA
Water solubility method	3	3	3	3
First outlet stream	1	1	0,00001	1
First specified split fraction	1	1	0,00001	1
First calculated split fraction	1	1	0,00001	1

Figure C.29: Simulation of ACP_{min} solution

	ACP Ta=-20			ACP ta=22			ACP=600kw		
Name	HE1	HE2	HE3	HE1	HE2	HE3	HE1	HE2	HE3
Hot side property method	**NRTL**	**NRTL**	**NRTL**	**NRTL**	**NRTL**	**NRTL**	**NRTL**	**NRTL**	**NRTL**
Hot side use true species approach for electrolytes	**YES**	**YES**	**YES**	**YES**	**YES**	**YES**	**YES**	**YES**	**YES**
Hot side free-water phase properties method	**STEAM-T**	**STEAM-**	**STEAM-TA**	**STEAM-TA**	**STEAM-**	**STEAM-TA**	**STEAM-1**	**STEAM-**	**STEAM-TA**
Hot side water solubility method	**3**	**3**	**3**	**3**	**3**	**3**	**3**	**3**	**3**
Cold side property method	**NRTL**	**NRTL**	**NRTL**	**NRTL**	**NRTL**	**NRTL**	**NRTL**	**NRTL**	**NRTL**
Cold side use true species approach for electrolyte	**YES**	**YES**	**YES**	**YES**	**YES**	**YES**	**YES**	**YES**	**YES**
Cold side free-water phase properties method	**STEAM-T**	**STEAM-**	**STEAM-TA**	**STEAM-TA**	**STEAM-**	**STEAM-TA**	**STEAM-1**	**STEAM-**	**STEAM-TA**
Cold side water solubility method	**3**	**3**	**3**	**3**	**3**	**3**	**3**	**3**	**3**
Constant UA [J/sec-K]	**7292**	**7292**	**58681**	**7292**	**7292**	**58681**	**7292**	**7292**	**58681**
Minimum temperature approach [C]	**1**	**0,0001**	**0,1**	**1**	**0,0001**	**0,1**	**1**	**0,0001**	**0,1**
Inlet hot stream temperature [C]	67,2	23,0	70,0	68,5	68,5	70,0	68,4	66,1	70,0
Inlet hot stream pressure [bar]	3,0	3,0	3,0	3,0	3,0	3,0	3,0	3,0	3,0
Inlet hot stream vapor fraction	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
Outlet hot stream temperature [C]	23,0	23,0	28,0	20,0	45,7	48,3	20,0	44,4	47,1
Outlet hot stream pressure [bar]	3,0	3,0	3,0	3,0	3,0	3,0	3,0	3,0	3,0
Outlet hot stream vapor fraction	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
Inlet cold stream temperature [C]	-20,0	23,0	23,0	19,0	23,0	45,7	19,0	23,0	44,4
Inlet cold stream pressure [bar]	1,0	1,0	3,0	1,0	1,0	3,0	1,0	1,0	3,0
Inlet cold stream vapor fraction	1,0	1,0	0,0	1,0	1,0	0,0	1,0	1,0	0,0
Outlet cold stream temperature [C]	46,6	23,0	67,2	19,0	57,8	68,5	22,6	56,0	68,4
Outlet cold stream pressure [bar]	1,0	1,0	3,0	1,0	1,0	3,0	1,0	1,0	3,0
Outlet cold stream vapor fraction	1,0	1,0	0,0	1,0	1,0	0,0	1,0	1,0	0,0
Heat duty [cal/sec]	53037,7	0,0	53015,2	0,3	27741,6	27768,7	2901,2	26265,7	29165,4
Calculated heat duty [cal/sec]	53037,7	0,0	53015,2	0,3	27741,6	27768,7	2901,2	26265,7	29165,4
Required exchanger area [sqm]	8,6	0,0	69,0	0,0	8,6	69,0	1,2	8,6	69,0
Actual exchanger area [sqm]	8,6		69,0	8,6	8,6	69,0	8,6	8,6	69,0
Average U (Dirty) [cal/sec-sqcm-K]	0,0		0,0	0,0	0,0	0,0	0,0	0,0	0,0
Average U (Clean)									
UA [cal/sec-K]	1741,5		14015,4	0,0	1740,8	14015,7	247,7	1741,3	14015,6
LMTD (Corrected) [C]	30,5		3,8	12,4	15,9	2,0	11,7	15,1	2,1
LMTD correction factor	1,0		1,0	1,0	1,0	1,0	1,0	1,0	1,0
Number of shells in series	1	1	1	1	1	1	1	1	1

	ACP ta=-20	ACPta=22	ACP =600kw
Name	B12	B12	B12
Property method	**NRTL**	**NRTL**	**NRTL**
Use true species approach for electrolytes	**YES**	**YES**	**YES**
Free-water phase properties method	**STEAM-TA**	**STEAM-TA**	**STEAM-TA**
Water solubility method	**3**	**3**	**3**
Specified pressure [bar]	**0**	**0**	**0**
Temperature estimate [C]	**14**	**14**	**14**
Outlet temperature [C]	23,0	68,5	66,1
Calculated outlet pressure [bar]	3	3	3
First liquid /Total liquid	1	1	1

FSplit			
Name	S1	S1	S1
Property method	**NRTL**	**NRTL**	**NRTL**
Use true species approach for electrolytes	**YES**	**YES**	**YES**
Free-water phase properties method	**STEAM-TA**	**STEAM-TA**	**STEAM-TA**
Water solubility method	**3**	**3**	**3**
First outlet stream	**1**	**5E-06**	**0,05**
First specified split fraction	**1**	**5E-06**	**0,05**
First calculated split fraction	1	5E-06	0,05
Second calculated split fraction	0	1	0,95

Figure C.30: Simulation of ACP_{max} solution

	SRP at - 8 degree				SRP at max cooling		
Name	HE1	HE2	HE3		HE1	HE2	HE3
Hot side property method	NRTL	NRTL	NRTL		NRTL	NRTL	NRTL
Hot side use true species approach for electrolytes	YES	YES	YES		YES	YES	YES
Hot side free-water phase properties method	STEAM-TA	STEAM-TA	STEAM-TA		STEAM-TA	STEAM-TA	STEAM-TA
Hot side water solubility method	3	3	3		3	3	3
Cold side property method	NRTL	NRTL	NRTL		NRTL	NRTL	NRTL
Cold side use true species approach for electrolytes	YES	YES	YES		YES	YES	YES
Cold side free-water phase properties method	STEAM-TA	STEAM-TA	STEAM-TA		STEAM-TA	STEAM-TA	STEAM-TA
Cold side water solubility method	3	3	3		3	3	3
Constant UA [J/sec-K]	7259	7519	58681		7292	7292	58681
Minimum temperature approach [C]	1	1	0,1		1	0,0001	0,1
Hot side outlet pressure [bar]	0	0	0		0	0	0
Inlet hot stream temperature [C]	31,1411235	23	32		31,7187	31,7187	32
Inlet hot stream pressure [bar]	3	1	3		3	3	3
Inlet hot stream vapor fraction	0	1	0		0	0	0
Outlet hot stream temperature [C]	10,9075148	13,7320035	18,7875368		0,99999	27,2316	27,7235
Outlet hot stream pressure [bar]	3	1	3		3	3	3
Outlet hot stream vapor fraction	0	1	0		0	0	0
Inlet cold stream temperature [C]	-8	10,9075158	17,2638413		0	23	27,2316
Inlet cold stream pressure [bar]	1	3	3		1	1	3
Inlet cold stream vapor fraction	1	0	0		1	1	0
Outlet cold stream temperature [C]	21,6919435	17,2638413	31,1406058		-6E-06	29,6325	31,7186
Outlet cold stream pressure [bar]	1	3	3		1	1	3
Outlet cold stream vapor fraction	1	0	0		1	1	0
Heat duty [cal/sec]	23639,2484	7380,75017	16257,8877		3,6E-05	5283,29	22,1199
Calculated heat duty [cal/sec]	23639,2484	7380,75017	16257,8877		3,6E-05	5283,29	22,1199
Required exchanger area [sqm]	8,53907893	8,84585191	69,029879		2E-08	8,57884	69,0352
Actual exchanger area [sqm]	8,54	8,84588235	69,0364706		8,57882	8,57882	69,0365
Average U (Dirty) [cal/sec-sqcm-K]	0,0203019	0,0203019	0,0203019		0,0203	0,0203	0,0203
Average U (Clean)							
UA [cal/sec-K]	1733,59537	1795,87612	14014,3779		4E-06	1741,67	14015,5
LMTD (Corrected) [C]	13,6359665	4,10983258	1,1600863		8,88617	3,03347	0,37696
LMTD correction factor	1	1	1		1	1	1

Figure C.31: Simulation of *SRP* solution

C.1.3 Excel simulation

Exhaus air cold	23	Glycol Hot	67,88	T3 ExhaustAir/Glycol	-10,14					
Exhaus air hot	54,78	Glycol Cold	44,08	T4 ExhaustAir/Glycol	-21,08					
Air Cold	17,50	Water Hot	70,00	T1 Air/Glycol	47,42	T1 Glycol/Water	2,12			
Air Hot	22,00	Water Cold	46,18	T2 Air/Glycol	45,88	T2 Glycol/Water	2,10			
	Qprikk	mprikk	Cp	Tinn	Tut	dT				
Inlett air	15 434,55	3,33	1030	17,50	22,00	4,50				
Exhaust air	109 002,22	3,33	1030	23,00	54,78	31,78	UA_HE3 ACP_max			
							2,67	3,33	5,00	mdot air
Glycol	124 436,77	1,37	3830	67,88	44,08	23,80	46998	58681	88110	UA
		0,41				2,95	1,00	1,25	1,88	mdot water
Water	124 436,77	1,25	4180	70,00	46,18	23,82		563		Q
	UA	Q	U	LMTD	K=1					
Glycol/Water	58861	124 436,77		2,11	1					
Area	58861	7 587,00								
InletAir/Glycol	331	15 434,55		46,65	1,00	kW				
Area H1	7292					600,6841506				
ExhaustAir/Glycol	-7290	109002,22		-14,95						
Area H2	7292					15,997				

Figure C.32: Cooling capacity= 600 kW

C.2 Case 1: Alternative Solution

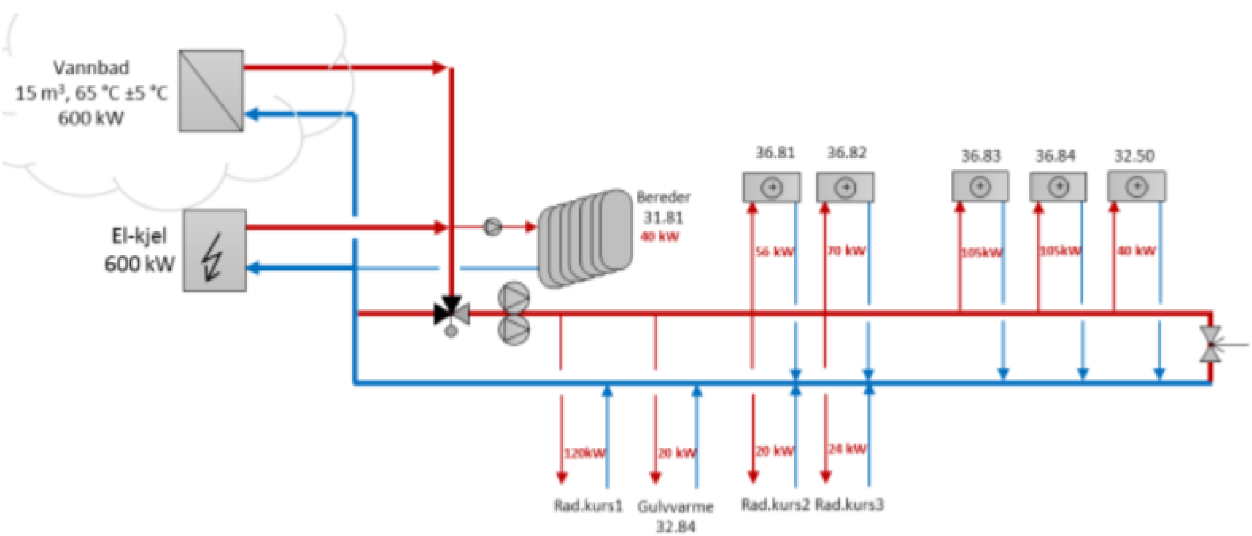


Figure C.33: Norsk Energi solution instead of HE4