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Techno-economical analysis of adding a heat-exchanger for further heat recovery in block 7 at Mälarenergi

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ABSTRACT

This degree project investigates the techno-economic feasibility of installing an additional water to air heat-exchanger before the existing humidifier at block 7 at Mälarenergi, which is a wood waste fired CHP plant producing 150 MW of thermal power. The aim of this project is to evaluate if the low grade heat from the scrubber water can be both recovered and used to preheat combustion air, which will reduce the dependency on auxiliary steam. A comparative modeling approach using Ebsilon®Professional was used to compare both the current system at Mälarenergi and a modified design. The modified design incorporates an additional heat-exchanger with three different efficiencies of 70%, 80%, and 90%. Operational data was provided at one-minute resolution and was processed into ten-minute averages. The ten-minute averages was then simulated for both the replicated system and the modified design, and run three separate times with different humidity levels 30%, 50%, and 70% to reflect seasonal variations in the mixed indoor-outdoor combustion air. Economic performance was assessed using annual energy recovery and payback period was calculated using estimated investment cost and auxiliary steam price provide by Mälarenergi. The simulation results of this study show that the additional heat-exchanger can on average increase the outlet temperature by 1.4-1.8 ℃ depending on humidity and efficiency. While the annual recovered heat is between 359-423 MWh depending on the humidity and efficiency. The higher humidity yields slightly higher heat recovery though the main contributing factor is the heat-exchangers efficiency, around 32% extra heat recovered between 70% efficiency and 90% efficiency. The economic analysis yields long payback periods, between 56-75 depending on how favorable the costs/revenues and technical efficiencies are. This study concludes that the proposed heat-exchanger improves the thermal performance of the plant, the low grade heat and nature of the current energy market makes the investment of the new system unfeasible.

Keywords: CHP, Heat-exchanger, Heat recovery, Techno-economic assessment, Auxiliary Steam, Comparative analysis, Ebsilon®Professional

PREFACE

This degree project was written as a conclusion to our 5 year engineering journey at Mälardalen University in their Master of Science Program for Energy Systems. This journey allowed us to gather the knowledge to develop our own simulation and the required experience to conduct this research.

This work focuses on heat recovery and was carried out in collaboration with Mälarenergi. The degree project has provided valuable knowledge of heat-exchangers and heat recovery as well as experience in Ebsilon Professional and the general research process.

We express our sincere gratitude to our supervisor and examiner, Hao Chen and Anders Nordstrand for their guidance, constructive feedback, simulation assistance and continuous support throughout this degree project. We would also like to thank the engineers at Mälarenergi, Peter Svensson and Alexander Luukas for their insight and tour of Block 7. Additionally we thank Mattias Lindholm at Mälarenergi for their help in getting information to allow this work to get started. We are also grateful to Dennis Braun at Ebsilon®Professional for providing additional licenses the their software, allowing us to simulate at more convenient times.

Västerås in June 2026

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SUMMARY

The global transition toward more sustainable energy systems has increased the need to improve the efficiency of existing energy systems. One such system is combined heat and power (CHP) plants, though Mälarenergi already incorporates several heat recovery systems, there is still some improvements that can be made since they rely on auxiliary steam to preheat the combustion air. The purpose and aim of this degree project is to determine whether low grade heat from the scrubber can be used to reduce the auxiliary steam consumption for heating the combustion air, thus improving the overall efficiency of the plant. This is highlighted in the research questions below.

1. What is the potential temperature increase from adding different efficiency water-to-air heat-exchangers before a humidifier that uses low grade heat inside block 7, and how does different humidity environments effect operation?
2. What is the annual power saving as a consequence of using less auxiliary steam to preheat the combustion air?
3. What is the payback period of adding an additional heat-exchanger, considering investment cost, operational saving, and would it be economically feasible?

This degree project investigates how much additional recovered heat can be gained by adding an additional heat-exchanger, and in turn how much auxiliary steam can be saved annually by decreasing the energy needed for preheating the air before combustion. The investment costs are compared to the annual operational savings from the added heat-exchanger, and an estimated payback period is created. The method is done through a comparative modeling approach using Ebsilon®Professional, simulating both the current system configuration and the modified version with the additional heat-exchanger. The modified system used three different heat-exchanger configurations, 70%, 80%, and 90% efficiencies. One-minute resolution operational data was provided by Mälarenergi and was processed into ten-minute data averages and simulated three different times for each case using different humidity levels, 30%, 50%, and 70% to reflect seasonal variations in the mixed indoor-outdoor combustion air.

The results show the additional heat-exchangers increased the outlet air temperature by approximately 1.4-1.8 ℃across all humidity levels. This temperature increase equals an annual recovered energy ranging from 359-366 MWh at 70% efficiency, 415-423 MWh at 80%, and 474–482 MWh at 90%. Humidity also has a small but measurable influence in performance, with higher humidity yielding higher energy recovery due to the increased heat capacity of moist air. However the heat-exchanger efficiency has a strong impact: moving from 70% to 90% efficiency increases recovered energy by roughly 32%. The results also indicate that the small temperature increases provide limited benefit when working with low grade heat while the larger temperature differences show a disproportionately higher power saving.

The economical evaluation reveals that, despite the energy recovered the investment is not financially viable under the current market conditions. With the estimated installation cost of 7-10 million SEK and auxiliary steam price of 260 SEK/MWh from Mälarenergi, the payback period ranges from 56-75 years depending on humidity and efficiency. Theoretical calculations showing the payback period as if the system was installed in 2020 in addition to when the plant was built, this including historical electricity certificates and other revenues show that the payback period remains at 48-67 years. An additional sensitivity analysis show that under extremely favorable conditions, halving the investment cost to 5 million, and doubling the revenue to 500 SEK/MWh, the payback period

still remains at around 20 years.

The limiting factor is the low grade heat, and the amount that is available. Although there is improvements in thermal performance, the modest temperature increase is a direct consequence by the low and limited energy provided by the scrubber. Even though the efficiency of the heat-exchanger is a decisive factor, the highest efficiency scenario does not yield a financially attractive outcome, resulting in long payback periods despite favorable conditions.

In conclusion, the degree project demonstrates that the installation of an additional heat-exchanger before the humidifier in block 7 would provide measurable heat recovery and increase system efficiency, reducing the amount of auxiliary steam used for preheating the combustion air. But the energy recovered is insufficient to justify the investment cost, even under optimistic scenarios, and consequently fail to produce a reasonable payback period. The degree project concludes that the proposed modification to the existing system should not be implemented.

Sammanfattning

Den globala övergången till mer hållbara energisystem har ökat behovet av att förbättra effektiviteten i befintliga energisystem. Ett sådant system är kraftvärmeverk, även om Mälarenergi redan har flera värmeåtervinningssystem, finns det fortfarande vissa förbättringar som kan göras eftersom de förlitar sig på hjälpånga för att förvärma förbränningsluften. Syftet med detta examensprojekt är att avgöra om lågkvalitativ värme från skrubbaren kan användas för att minska den extra ånganvändningen för att värma förbränningsluften, vilket skulle förbättra anläggningens totala verkningsgrad. Detta framhävs i forskningsfrågorna nedan.

1. Vad är den potentiella temperaturökningen om man lägger till olika effektiva vatten-till-luft-värmeväxlare före en luftfuktare som använder lågkvalitativ värme inne i block 7 och hur påverkar olika fuktighetsmiljöer driften?
2. Vad är den årliga energibesparingen som en följd av att använda mindre hjälpånga för att förvärma förbränningsluften?
3. Hur lång är återbetalningstiden för att lägga till en extra värmeväxlare, med hänsyn till investeringskostnader, driftsbesparingar och skulle det vara ekonomiskt genomförbart?

Detta examensprojekt undersöker hur mycket extra återvunnen värme som kan utvinnas genom att lägga till en extra värmeväxlare, och i sin tur hur mycket hjälpånga som årligen kan sparas genom att minska energin som krävs för att förvärma luften före förbränning. Investeringskostnaden jämförs med de årliga driftsbesparingarna från den tillagda värmeväxlaren och en uppskattad återbetalningstid skapas. Metoden utförs genom en jämförande modelleringsmetod med Ebsilon®Professional, där både den nuvarande systemkonfigurationen och den modifierade versionen simuleras med den extra värmeväxlaren. Det modifierade systemet använde tre olika värmeväxlarkonfigurationer, 70%, 80% och 90% verkningsgrad. Operativa data med en minuts upplösning tillhandahölls av Mälarenergi och bearbetades till tiominutersdatagenomsnitt och simulerades tre olika tider för varje fall med olika fuktighetsnivåer, 30%, 50% och 70% för att spegla säsongsvariationer i den blandade inomhus-utomhusförbränningsluften.

Resultaten visar att de extra värmeväxlarna ökade utloppsluftens temperatur med cirka 1.4–1.8 ℃ över alla fuktighetsnivåer. Denna temperaturökning motsvarar en årlig återvunnen energi från 359–366 MWh vid 70% verkningsgrad, 415–423 MWh vid 80% och 474–482 MWh vid 90%. Fuktigheten har också en liten men mätbar inverkan på prestandan, där högre luftfuktighet ger högre energiåterställning tack vare den ökade värmekapaciteten hos fuktig luft. Dock har värmeväxlarens verkningsgrad en stark påverkan: att gå från 70% till 90% effektivitet ökar den återvunna energin med ungefär 32%. Resultaten visar också att de små temperaturökningarna ger begränsad nytta vid arbete med lågkvalitativ värme, medan de större temperaturskillnaderna visar en oproportionerligt högre energibesparing.

Den ekonomiska analysen visar att investeringen, trots den återvunna energin, inte är ekonomiskt hållbar under rådande marknadsförhållanden. Med den beräknade installationskostnaden på 7–10 miljoner SEK och en ångkostnad på 260 SEK/MWh från Mälarenergi, varierar återbetalningstiden mellan 56–75 år beroende på luftfuktighet och verkningsgrad. Teoretiska beräkningar som visar återbetalningstiden ifall systemet installerades 2020 när anläggningen byggdes, inklusive historiska elcertifikat och andra intäkter, visar att återbetalningstiden ligger kvar på 48–67 år. En ytterligare känslighetsanalys visar att under extremt gynnsamma förhållanden, med halvering av investeringskostnaden till 5 miljoner och en fördubbling av intäkterna till 500 SEK/MWh, ligger återbetalningstiden fortfarande på cirka 20 år.

Den begränsande faktorn är lågkvalitativ värme och mängden som finns tillgänglig. Även om det finns förbättringar i termisk prestanda, är den svaga temperaturökningen en direkt följd av den låga och begränsade energi som skrubbaren tillför. Även om värmeväxlarens effektivitet är en avgörande faktor, ger det högsta effektivitetsscenariot inte ett ekonomiskt attraktivt resultat, vilket resulterar i långa återbetalningstider trots gynnsamma förhållanden.

Avslutningsvis visar examensprojektet att installation av en extra värmeväxlare före luftfuktaren i block 7 skulle ge mätbar värmeåtervinning och öka systemets effektivitet, vilket minskar mängden hjälpånga som används för förvärmning av förbränningsluften. Men den återvunna energin är otillräcklig för att motivera investeringskostnaden, även under optimistiska scenarier och ger därför inte en rimlig återbetalningstid. Examensprojektet drar slutsatsen att den föreslagna modifieringen av det befintliga systemet inte bör genomföras.

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Nomenclature

ABBREVIATIONS

4E Energy, Exergy, Economic and Environmental

CCHP Combined Cooling, Heating and Power plant

CCS Carbon Capture and Storage

CCU Carbon Capture and Utilization

CHP Combined Heat and Power

CO_2 Carbon dioxide

COP28 28th United Nations Climate Change Conference

GHG Green House Gas

HXC Heat exchange capacity

HXFT heat-exchanger finned tube

HXLF heat-exchanger fin-less tube

NO_x Nitrogen Oxides

OHXFT Optimized heat-exchanger finned tube

ORC Organic Rankine Cycle

RH Relative Humidity

ROI Return on Investment

SRC Steam Rankine Cycle

TTA Thermal transmittance $\times$ area

VOCs Volatile Organic Compounds

DEFINITIONS

Colburn j-factor Dimensionless number that measures the effectiveness of heat and mass transfer

Kalina Cycle A thermodynamic cycle which uses a mixed working fluid for greater heat transfer

Regula - falsi Algorithm used to solve linear equations and smooth nonlinear equations for a single unknown value

VARIABLES

η Efficiency [%]

DP Pressure drop [Bar]

DQLR Heat loss through radiation (relative) [-]

DTLO Terminal temperature difference (lower) [°C]

DTUP Terminal temperature difference (upper) [°C]

F Mass flow relative to design [-]

H Enthalpy [kJ/kg]

KA Thermal transmittance × area [kW/K]

LMTD Logarithmic temperature difference [°C]

M Mass flow [kg/s]

P Pressure [Bar]

Q Energy flow [kW]

Q_e Exchanged heat [kW]

T Temperature [°C]

TOL Maximum permissible tolerance in energy balance for the internal iteration [kW]

1 Introduction

The current global transition towards sustainable energy systems and industrial practices is increasing the need for improving available systems. In the European Union the yearly energy efficiency progress is estimated to decrease to under 1%, which is well below the COP28 target of 4% (IEA, 2025). The increased reliance on energy recovery has become a necessary component in the global transition as fuel and CO_2 emission allowance prices has increased. One way to reach the current goal is improving the efficiency of existing combined heat and power (CHP) plants (Dzierwa et al., 2024).

Wood waste fired CHP plants are considered an environmentally friendly source of power due to releasing less pollutants than conventional fossil fuel plants. These plants usually use locally sourced waste from either residential sources, such as old wood furniture or wood scraps from wood processing plants in the local area. This waste wood is then used as a fuel source for a centralized CHP plant. By using the waste wood in this manner it is better utilized than conventional removal of wood waste, as CHP plants are able to capture some of the released pollutants (Marchenko et al., 2020).

In 2023 Sweden had around 10% of its total electricity output coming from CHP plants mainly powered by bio fuel, and in 2024 the electricity production from bio and waste fuel decreased to around 6.6% or 11415 GWh of electricity (Swedish-Institute, 2025; IEA, 2026). Along with electricity production CHP plants also produce heat, mainly used in district heating. According to SCB, 2025 CHP plants produced and delivered around 22738 GWh of district heating in the year 2024, highlighting the importance of CHP plants in the Swedish energy and heating system and the benefits of improving already existing plants.

1.1 Background

In this section we provide background information about Mälarenergi, Heat recovery system and Combined Heat and Power Plants.

1.1.1 Mälarenergi

Mälarenergi is a local energy producer in the county of Västmanland, Sweden. They produce both electricity and heat, delivering district heating across all of Västerås. As of 2015 they also provide installation of fiber optic cables, creating Sweden's biggest broadband network linking five municipalities, Västerås, Eskilstuna, Strängnäs, Hallstahammar and Arboga. Their main complex was built in the 1960's and is one of the largest in Sweden. Since that they have expanded their operation and continuously upgraded their hardware and equipment to ensure reliability and low environmental impact. As of today there is a total of four active blocks, a block is a boiler and turbine (Mälarenergi, 2026c). The four active blocks are the following:

- Block 3 is backup plant powered by an oil boiler.
- Block 4 is a turbine that is driven by a biofuel boiler.
- Block 6 is a bio and refuse derived fuel plant.
- Block 7 is a wood waste fired boiler.

In 1992, Mälarenergi also became the first company in Sweden to produce and distribute district cooling for commercial use (Mälarenergi, 2026b).

1.1.2 Mälarenergi's Environmental Goals

At Mälarenergi they have their own goal of releasing 0 CO_2 emissions when delivering their products to the customer, such as district heating and electrical power, which they want to have completed by the year 2030. However, this goal has recently been expanded to 0 CO_2 emissions for the entirety of their operation. This goal is called "Resan mot noll" or "The journey to zero", this is done to keep in line with the Paris agreement which states that the global temperature rise should not increase by more than 1.5° by 2030. To help achieve this goal, they invested in using an old cavern space that was used for oil storage when they used that in their boilers. This cavern is approximately 300k cubic meters and is used to store district heating water, this allows them to store around 13 GWh of thermal energy. The thermal storage is calculated by Mälarenergi to be able to reduce their CO_2 emissions by 1600 tons every year. This also connects to how they want to optimize and reduce their current operation, heat recovery is a small part of this. If they cannot reduce their CO_2 emissions to 0 so to compensate for it they will utilize CCS/CCU (Carbon Capture and Storage/Utilization) (Mälarenergi, 2024; United-Nations, 2015).

1.1.3 CHP plant

CHP plant or combined heat and power plant is defined as a power plant that produces two or more forms of energy simultaneously from one main fuel source (Rzetelska and Combrinck, 2022). Integrating both heat and electricity production distinguishes CHP plants from other conventional energy systems that would produce either electricity or heat in separate power plants (Pariasamy et al., 2022). Rather than rejecting the waste heat that is created when producing electricity, the CHP avoids some of those losses by capturing it, and using it for productive thermal applications. This in turn greatly improves fuel utilization and overall plant efficiency as less energy is wasted to the environment (Costea and Feidt, 2022).

Breeze, 2018 describes that there are two types of CHP plant cycles, it is either a topping cycle or a bottoming cycle. In a topping cycle, the fuel is first combusted and the energy released from combustion is used to generate electricity. The excess heat is then captured and repurposed using heat recovery equipment such as, heat-exchangers, or economizers to heat steam, water or air. The topping cycle is great and appropriate when you have a need for both electricity and thermal energy. The bottoming cycle is opposite to the topping cycle. In a bottoming cycle you first use your fuel in a high temperature thermal process, this can either be in a industrial furnace or kilns. The waste heat from the thermal process is then used to produce electricity, most commonly by utilizing a steam turbine or an ORC. Bottoming cycles are far less common than topping cycles but can be a great addition and are more advantages in industries using high temperature processes, like glass manufacturing, cement production or metal processing (Breeze, 2018; Pariasamy et al., 2022).

1.1.4 Heat-exchangers & Economizers

Heat-exchangers are one of the most fundamental components in a CHP system. These components allow the system to recover additional heat, particularly waste heat from various processes. Capturing waste heat improves overall system efficiency by reducing energy losses and thereby lowering environmental impacts. There are many different kinds of heat-exchangers and they can use different types of working fluids, such as air-to-air, water-to-water (Srimuang and Amatachaya, 2012). In this thesis a water-to-air heat-exchanger is planned to be used.

One of the primary mechanics for heat recovery is to use economizers to preheat feedwater entering the boiler (Dzierwa et al., 2024). By doing this you can use the energy from the unused flue

gas instead of the standard alternative of using the steam from the turbine bleed to preheat the feedwater. This implies that when the flue gas is used to heat the water less steam is extracted from the turbine and in turn can be used to produce more electricity, using less fuel thus increasing the efficiency of the entire power plant (Xu et al., 2013). Another application of economizers is to preheat the intake air going into the boiler. By preheating the air the enthalpy rise, adding more energy to the system (Chen and Mauro, 2024). With more energy in the combustion air the amount of energy needed for combustion is reduced. As there is more energy in the system the chemical energy from combustion is directed to increasing the flame temperature instead of heating the fuel to combustion temperature, this in turn leads to a fuel reduction (Ibrahim and Farrukh, 2019).

1.1.5 Scrubbers & humidifiers

CHP plants designed for delivering their heat and power to a city, also release a great amount of greenhouse gas (GHG) emissions. To combat some of the emissions such as NO_x, SO_x and VOCs, scrubbers are often installed. The scrubber operate by spraying water into the flue gas, as it evaporates the moisture content increases to 100% and as a consequence the water in the flue gas condensates and the flue gas temperature decrease. The scrubber does not only work like a contact heat-exchanger but as previously mentioned it also acts as a cleaning process as it helps contain and reduce emissions of harmful substances, collecting heavy metals and dust particles in the water condensate (Dzierwa et al., 2024; Saari et al., 2023).

Scrubbers are particularly important in large CHP plants that utilize biomass as fuel, primarily to reduce NO_x emissions generated during combustion. These pollutants can cause significant environmental damage, such as acid rain and are therefore strictly regulated by environmental standards. In order for CHP plants to comply with legal emission limits, the installation of scrubbers are often necessary (Johansson et al., 2021).

To further improve the effectiveness of heat recovery systems, Zhang et al., 2020 proposed that adding a humidifier to the combustion air could increase the heat recovery efficiency by 12.2 %, coupled with a waste heat utilization increase of 6.9%. By increasing the humidity in the flue gas, this allows for more water to condense in different heat recovery systems thus leading to more latent heat recovery. Zhang et al., 2020 continues to talk about that additionally from only increasing the efficiency, the installation of a combustion air humidifier could also reduce NO_x emissions of by around 82% compared to the standard emissions for an in-use boilers in Beijing.

1.1.6 Block 7 at Mälarenergi

Boiler 7 at Mälarenergi is their newest boiler which began its operation in 2020, this CHP plant has around 150 MW of thermal power. This allows the boiler to supply thermal energy to around 36% of the nearby municipalities, these are Västerås, Surahammar, Hallstahammar, Skultuna och Kolbäck. Combined with block 6 the plants cover 86% of the total thermal demand in these municipalities. The primary goal of this boiler was to reduce their dependence of fossil fuels as such boiler 7 uses wood waste as its primary fuel. This fuel does require some preprocessing, namely crushing to reduce large chunks of this wood to smaller sizes. The former wood products are also filled with several metallic components such as nails and other reinforcements. With this boiler operational they allowed Mälarenergi to shut down their oldest boilers, 1 & 2 (Mälarenergi, 2026a).

1.1.7 Block 7 heat recovery systems

The first heat recovery method that is utilized at block 7 is the use of indoor air, which has been heated by all the process of the plant. This air can be mixed with outdoor air too, depending on how much is available indoors. The first system this slightly preheated air encounters is a humidifier, if the outside air has not reach 100% humidity yet. The humidifier uses some of the exhaust heat of the scrubber to inject hot water against the gathered air. After this the humid air is yet again heated with a heat-exchanger. This heat-exchanger uses the exhaust heat of the economizer, and if more heating is required it uses axillary steam bleed from the turbines. Then some of the air splits with it going directly to the boiler. The other part of the split goes to the economizer to get heated further. The air is then combusted in the boiler, simultaneously heating circulating water that evaporate to steam going to the turbines. After the turbines the steam goes through condensers connected to the district heating system taking advantage of the high temperature steam while cooling it to go back to the economizer and finally connecting back to the boiler to be heated again. The flue gasses progress though the boiler, going to the economizer to preheat the circulating water and air going into the boiler. The flue gas is then transported to the baghouse filter where it is filtered to remove dust and other debris. After the baghouse filter the flue gasses are cooled in a scrubber, heating up water sprayed into the gasses. The heated water is then transported through a heat-exchanger transferring some of its heat to the district heating system and later either recirculated back into the scrubber or split going to the humidifier to then return back to the scrubber after humidifying the combustion intake air.

1.2 Purpose/Aim

The purpose/aim with this degree project is to do a techno-economical assessment, adding an additional heat-exchanger before the humidifier in a existing power plant, Mälarenergi block 7. Simulating different load scenarios and observing when and how the heat-exchanger impacts the system, increasing the outlet air temperature and reducing the amount of auxiliary steam used to preheat the combustion intake air.

1.3 Research questions

1. What is the potential temperature increase from adding different efficiency water-to-air heat-exchangers before a humidifier that uses low grade heat inside block 7, and how does different humidity environments effect operation?
2. What is the annual power saving as a consequence of using less auxiliary steam to preheat the combustion air?
3. What is the payback period of adding an additional heat-exchanger, considering investment cost, operational saving, and would it be economically feasible?

1.4 Delimitation

The first delimitation of this project is the model boundary which includes the humidifiers air/water intake and the air/water exhaust. No other components are modeled. Due to the Ebsilon®Professional software not having a humidifier that reflects the humidifier in block 7 at Mälarenergi in its premade components, the humidifier is created with other components which would entail that the heat transfer simulation happens in two steps rather than one. The first would simulate the humidification process, injecting the water into the air and the second step would simulate the heat transfers of the fluids. Compared to reality where these two would happen simultaneously.

2 Method

In this section we describe how we have acquired our results in Ebsilon and MATLAB.

2.1 Literature Review

For the literature review Scopus & MDPI was used as the primary scholarly search engines. Our key topic was `CHP` & `Heat Recovery` this was combined with a boolean search strategy to find the articles used in this report. Along with our primary keywords the secondary keywords that were used is: `heat-exchanger`, `economic assessment`, `efficiency`, `exergy`, `system performance`, `waste heat` and `environmental impacts`.

2.2 Data and Data collection

The data received from Mälarenergi is sensor data with a resolution of one minute from the years 2021-2025. The sensors are pressure, temperature and flow sensors. Other types of data like pipe dimensions were also provided in CAD designs of the entirety of block 7.

2.3 Seven stage modeling approach

We have used a seven stage modeling approach:

1. Define the problem.
2. Identify controlling factors.
3. Evaluate the data.
4. Develop equations.
5. Implement the solution procedure.
6. Verify the model solution.
7. Validate model.

2.4 Define the problem

The method we used is a comparative simulation of block 7 at Mälarenergi, comparing the system with an added heat-exchanger using Ebsilon®Professional and without it. The simulation was carried out in two parts, first the humidifier was created and made to operate like a real life humidifier under normal operation conditions. The second part consists of adding the additional heat-exchanger and connecting it to the replicated system.

The system is treated as a semi-lumped process, as the heat-exchanger and humidifier is calculated as an average and do not have specific sections. The model uses static, steady-state conditions over the course of a year with a ten minute interval. This is because the focus is on long-term operational behavior rather than transient dynamics. The key inputs include, air and water mass flow, temperature and pressure at inlet points. The key outputs include, recovered heat, efficiency, and calculated economic values, such as annual heat recovery and payback period. These are the most important factors for comparing the baseline configuration with the modified system, and allowing a structured comparison of both technical and financial outcomes.

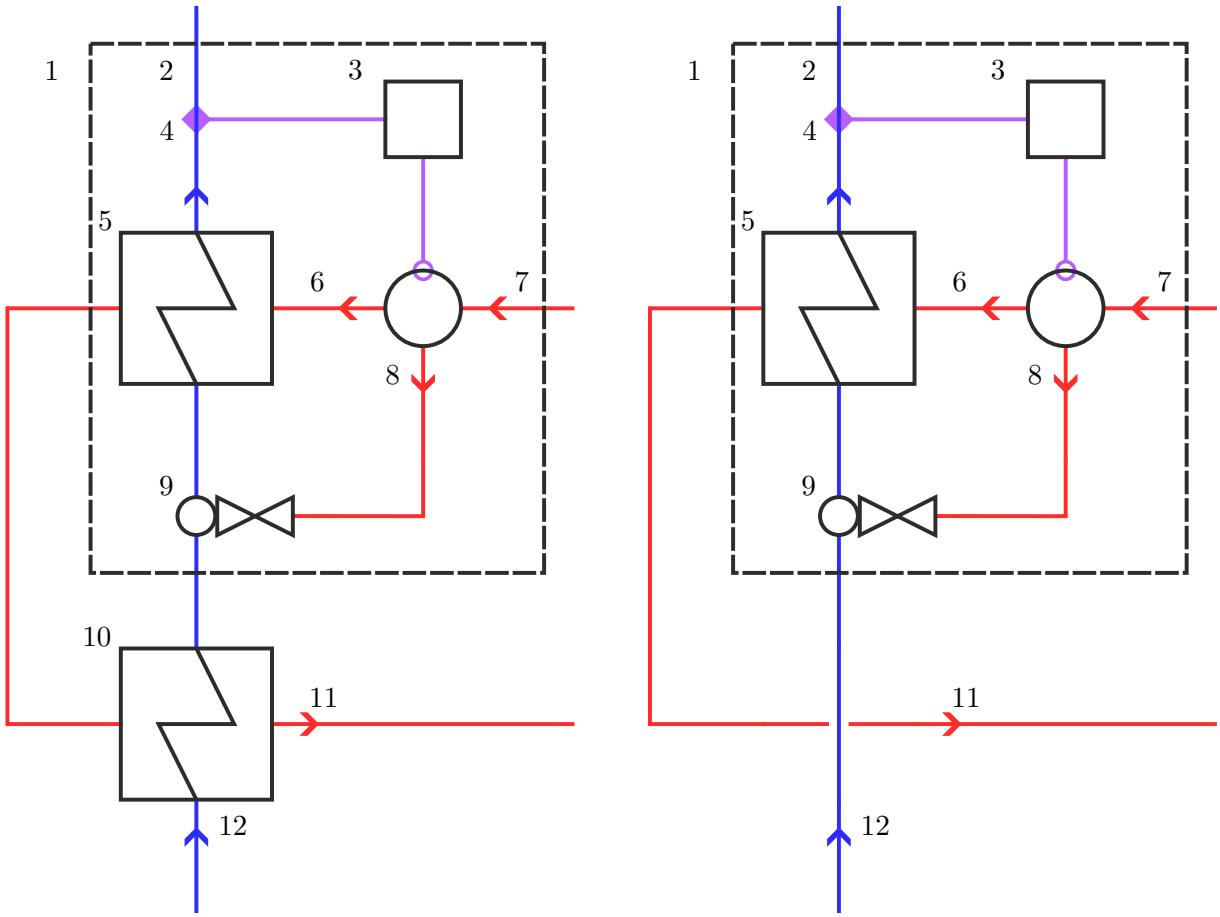

Figure 1: Left: New system. Right: Replicated System.

1. Boundary of the humidifier.
2. Heated air, towards boiler.
3. Controller unit.
4. Data collection point for controller unit.
5. Heat-exchanger 1.
6. Hot water splitter outlet.
7. Hot water input.
8. Hot water branch towards injector.
9. Injector.
10. Heat-exchanger 2.
11. Hot water outlet.
12. Cold air inlet.

2.4.1 Hot water side

Hot water enters the system from the scrubber through channel (7). At the splitter, the flow divides into two branches. The primary branch (6) supplies the heating element inside the humidifier, while the secondary (8) branch feeds the water injector (9) responsible for humidifying the combustion air.

After passing through the humidifier section, the water temperature has decreased due to heat and mass transfer to the air stream. The cooled water then continues to the additional heat-exchanger (10), where it preheats the incoming combustion air (12) drawn from a controlled mixture of indoor and outdoor air. Following this heat-exchange process, the water returns (11) to the scrubber, thereby completing the circulation loop.

2.4.2 Combustion air side

Combustion air is supplied from a mixture of indoor and outdoor air (12). The air first passes through the added heat-exchanger (10), where it is preheated by the water exiting the humidifier. The preheated air then enters the humidifier and reaches the water injector (9).

The injector (9) introduces water into the air stream until the target humidity level is achieved, as regulated by a dedicated control system (3). After humidification, the air continues through the humidifier to the heating element (5), where it is heated. Finally, the conditioned air exits the humidifier (2) and flows toward the boiler.

2.5 Identify controlling factors

The system is governed by thermodynamic and hydraulic factors that determine how both the replicated and modified configurations perform. One of the most important factors is the heat-transfer mechanisms, which directly influence the effectiveness of the additional heat-exchanger. Temperature gradients and overall heat-transfer coefficients determine how much thermal energy can be recovered. Mass-flow rates, pressure levels, and pressure drops across components influence the distribution of energy across the system. Flow related variables such as frictional losses, and pressure drops are also taken into consideration to ensure that the modified system remains within realistic operational values.

2.6 Evaluate the Data

The empirical foundation of the study consists of operational measurements from Block 7, including temperature and flow rates. All the data is cleaned, filtered, and evaluated to remove outliers and ensure that only representative operating conditions are used. The baseline Ebsilon®Professional model is iteratively adjusted until the simulated design outputs align with performance within acceptable real life operation conditions. Once validated the additional heat-exchanger was implemented. Simulation results from both configurations are extracted and compared, focusing on heat recovery potential. The economic evaluation is based on calculated heat gains, estimated investment costs, and operational savings.

2.7 Develop Equations

The analytical framework relies on thermodynamic and economic equations describing heat-transfer processes, energy balances, and cost relationships. The energy balance is estimated using different equations that in turn control how the different components in the model behave. Economic assessment uses payback period to evaluate financial viability.

2.7.1 Injector

This is component (comp 38) from EBSILON®Professional, 2026.

1. Fluid inlet.
2. Fluid outlet.
3. Injection Line.

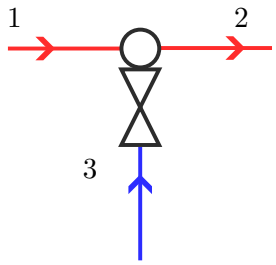

Figure 2: Injector (Comp 38)

$$M_2 = M_1 + M_3, \quad P_2 = P_1 \tag{1}$$

if line 1, 2 = flue gas or air and line 3 = water, then $H_{diff} = -2500 kJ/kg$

$$H_2 \times M_2 = H_1 \times M_1 + (H_3 + H_{diff}) \times M_3 \tag{2}$$

2.7.2 Splitter

This is component (comp 4) from EBSILON®Professional, 2026.

1. Fluid inlet.
2. Fluid outlet.
3. Branch.

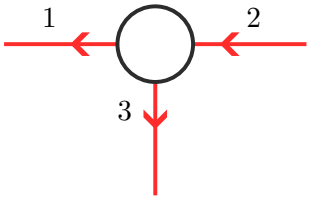

Figure 3: Splitter (Comp 4)

$$P_2 = P_1, \quad P_3 = P_1 \tag{3}$$

$$T_2 = T_1, \quad T_3 = T_1 \tag{4}$$

$$H_2 = H_1, \quad H_3 = H_1 \tag{5}$$

$$M_2 = M_1 - M_3 \tag{6}$$

2.7.3 Heat-exchanger

This is component (comp 25) from EBSILON®Professional, 2026.

1. Primary inlet, cold side.
2. Primary outlet, cold side.
3. Secondary inlet, hot side.
4. Secondary outlet, hot side.

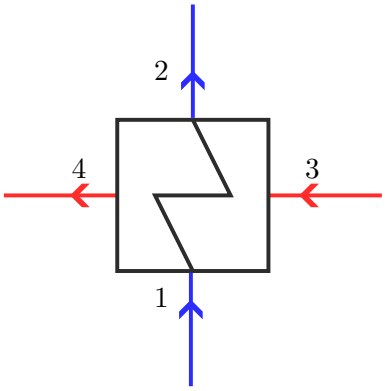

Figure 4: Heat-exchanger (Comp 25)

Estimation for iteration 1:

$$F_1 = (M_1/M_{1,N})^2 \tag{7}$$

$$F_3 = (M_3/M_{3,N})^2 \tag{8}$$

if GLOBAL = design, then $F_1 = 1.0$, if GLOBAL = design, then $F_3 = 1.0$

$$P_2 = P_1 - DP_{12,N} * F_1, \quad M_2 = M_1 \tag{9}$$

$$FK_1 = f(M_1/M_{1,N}), \quad FK_2 = f(M_3/M_{3,N}) \tag{10}$$

$$KA = KA_N * FK_1 * FK_2 \tag{11}$$

$$P_4 = P_3 - DP_{34,N} * F_3 \tag{12}$$

Maximum/Minimum values for the iteration

$$H_{2,max} = f(P_2, T_3), \quad H_{4,min} = f(P_4, T_1) \tag{13}$$

$$Q_{12,max} = M_1 * (H_{2,max} - H_1), \quad Q_{34,max} = Q_3 - M_4 * H_{4,min} \tag{14}$$

$$Q_{max} = min(Q_{12,max}, Q_{34,max}) \tag{15}$$

$$Q_{12} = 0.5 \times Q_{max} \tag{16}$$

Each iteration:

$$H_4 = (Q_3 - Q_{12}/(1 - DQLR))/M_4 \tag{17}$$

$$T_4 = f(P_4, H_4), T_2 = f(P_2, H_2) \tag{18}$$

$$H_2 = H_1 + Q_{12}/M_2 \tag{19}$$

$$DTLO = T_4 - T_1, \quad DTUP = T_3 - T_2 \tag{20}$$

$$LMTD = (DTUP - DTLO)/(ln(DTUP) - ln(DTLO)) \tag{21}$$

$$Q_e = KA \times LMTD \tag{22}$$

$$DQQ_1 = DQQ, \quad DQQ = Q_{12} - Q_e \tag{23}$$

Regula - falsi method (Gradient $= (Q_{12} - (Q_{12})_1)/(DQQ - DQQ_1)$ at iteration step 1: Gradient of the last global step)

$$Q_{12,X} = Q_{12} - DQQ \times Gradient, \quad Q_{12,1} = Q_{12}, \quad Q_{12} = Q_{12,X} \tag{24}$$

$$DQ = |DQQ/((Q_{12} + Q_e) \times 0.5)| \tag{25}$$

if $DQ < TOL$ then end iteration step X, else continue the iteration $X + 1$

$$Q_N = Q_3 - M_4 \times H_4 \tag{26}$$

$$KA \times LMTD = M_2 \times H_2 - M_1 \times H_1 \tag{27}$$

$$KA \times LMTD = M_3 \times H_3 - M_4 \times H_4 - Q_N \times DQLR \tag{28}$$

All formulas and definitions surrounding each component has been taken from Ebsilon®Professional Online Documentation, from their respective component page.

2.7.4 Payback period

For the economic equations we have used the following. With data from Mälarenergi we have received an estimated CAPEX and the price of the auxiliary steam in SEK/MWh, that the new heat-exchanger is meant to reduce. This is will be the indicator for the new system so observe its profitability. As it will show how long it will take the new system generate an annual return equal to the CAPEX.

$$\frac{CAPEX}{\text{MWh/y} \times SEK/MWh} \tag{29}$$

2.8 Implement the solution procedure

2.8.1 Ebsilon

After completing the simulations for both configurations, the baseline replica model and the modified model with an additional heat-exchanger, the two scenarios were compared and analyzed. The respective energy outputs were evaluated, and the annual differences were plotted for further interpretation. To examine the influence of ambient conditions, each scenario was simulated using three different inlet relative humidity levels: 30%, 50%, and 70%. For each humidity case, the heat-exchanger efficiency was also varied across three settings: $\eta = 70\%$, 80%, and 90%.

2.8.2 Payback Period

The first step to this was to create a range of values for the CAPEX and revenue streams. This range is 5-10 million SEK for the CAPEX and 100-500 SEK/MWh for the revenue. This allows us to represent uncertainty in the calculations. To visualize this in a intuitive way a mesh grid from MATLAB has been utilized as that allows all possible combinations to be viewed for the payback period. This is then plotted in a 3 dimensional surface plot for intuitive understanding. For this only the 50% RH and 80% η data was used as that is the middle ground.

2.9 Verify the model solution

To verify the model performance, the results from the different simulation scenarios were examined and analyzed. After assessing whether the outcomes were reasonable, the model was either adjusted or rerun to confirm the results. In addition, the input values provided by Mälarenergi were evaluated to ensure that they were reasonable and representative of normal operating conditions. Based on this assessment, the results were interpreted and, where necessary, adjusted to reflect how the system would operate with the additional heat-exchanger configuration installed. To verify the economic aspect of this report a surface plot was made with a theoretical span of the CAPEX and revenue.

2.10 Validate model

When validating the system we assume Ebsilon®Professional and its Excel add-in perform thermodynamic and off-design calculations accurately, within a reasonable margin of error, with the provided real operating conditions. This approach is necessary because Mälarenergi did not provide validation data; in particular, there are no recorded measurements of the outlet parameters that this study focuses on. Without validation data from Mälarenergi, particularly outlet parameter measurements, a direct comparison between simulated and measured values is impossible. The model's validity thus hinges on the software's established reliability and the physical plausibility of the simulated results.

3 Theoretical framework/Literature study

In this section we gather the knowledge needed to complete the method and discussion section.

3.1 Industrial heat-exchangers

The thermal conductivity and the heat transfer is one of the most important parts of a heat-exchanger, as it is what fundamentally determines the effectiveness and the economic viability of the heat recovery system. Today's heat-exchanger design advances focuses on enhancing these traits, looking at transfer coefficients through modifications in geometry and flow manipulation. (J. Wang et al., 2025; Wardach-Swiecicka et al., 2025) One of the most common air-to-water heat-exchanger is the finned tube heat-exchanger. This is because it compensates the low heat transfer coefficient found in air or gas, by utilizing and taking advantage of an extended surface area. Some of the resent research has explored new fin geometries that in turn enhance heat transfer though both vortex generation and flow mixing (Sheikholeslami and Ganji, 2016; El Maakoul et al., 2017; J. Wang et al., 2025; Wardach-Swiecicka et al., 2025).

3.1.1 Tube bundle heat-exchanger

J. Wang et al., 2025 conducts a study on optimizing the heat transfer in a tube bundle heat-exchanger for waste heat recovery. The study focuses on investigating different tube bundled heat-exchanger fin designs, looking at different fin incline angles ($\theta = 30°, 35°, 40°, 45°, 50°$) and fin diameter (D= 62, 68, 74 mm) under varying fluid velocity and temperature profiles. Looking at the different scenarios the result show that the most optimal configuration that result in the best heat transfer coefficient and performance where at ($\theta = 45°$,D = 74 mm). This configuration resulted in a 22.76% to 31.22% heat transfer coefficient increase compared to traditional smooth fin-less tube bundles.

Another study, Maghsoudali et al., 2022 focus and investigate three different bundled shell and tube heat-exchangers including, a fin-less tube model, a finned tube model and an optimized finned model. The focus of the study was to assess and discuss how different design changes affect and change the heat transfer, pressure drop and performance of the different heat-exchangers. Fins in heat exchanges are traditionally used to enhance the surface area of the heat-exchanger, in some cases this enhancement can lead to altercations in the fluid flow behavior, preventing air flow over the pipes and subsequently leading to a reduction in efficiency despite the use of the extra heat transfer area. When discussing the result Maghsoudali et al., 2022 talks about different heat transfer values under different loads ($m_s = 0.08\ kg/s$: HXLF = $70\ W/m^2K$, HXFT = $90\ W/m^2K$, OHXFT = $280\ W/m^2K$). Under special conditions and fluid mass flow rates, the normal generic finned pipe bundle perform worse than the smooth model, this is a consequence of the fluid flow in the heat-exchanger. Under special conditions the fluid in the normal finned heat-exchanger is compromised, resulting in only one seventh of the fluid passing over the heat-exchanger pipes, greatly reducing the heat transfer rate. This is avoided in the optimized heat-exchanger, by changing the geometry of the fins and adding shell-side protrusions the flow is manipulated into passing over the tubes themselves and greatly increasing the heat transfer. The study emphasize the importance of guided flow in finned heat-exchangers, and how geometric modification of the flow is important and can recover or enhance the performance of the fins effectiveness.

3.1.2 Cross-Flow heat-exchanger

Like previously mentioned, gas-liquid heat-exchangers are often limited by the gas-side heat transfer which often is significantly lower than the liquid side. Choi et al., 2010 mentions that to overcome this limitation the gas side is often fitted with extra fins, enlarging the total heat transfer area leading to better heat transfer capabilities. Choi et al., 2010 studies the heat transfer behavior of discrete fins in gas-liquid heat-exchangers, adjusting fin pitch, number of tube rows, fin alignment and vertical fin space. The aim of the study is to understand how the different parameters affect the Colburn j-factor, which is a way of evaluate the effectiveness of heat and mass transfer during a convection processes. One of the most notable discoveries from the study was that increasing fin pitch consistently increased the Colburn j-factor throughout all the tested Reynolds numbers on the system. Choi et al., 2010 elucidates that with larger fin pitch the boundary layer between the fluid and the plates gets interrupted, this means that there is a greater delay before a boundary layer is formed leading to greater heat transfer. Another find was that using discrete plates instead of one continuous large one also increased heat transfer. This is because the gaps between the plates crates a break in the boundary layer as a new leading edge is introduced. The use of discrete fins outperform long continuous fins by 6-11.6 % within the fin-pitch range of 7.5-15 mm.

F. Wang et al., 2026 focus on creating and studying a two-stage Taguchi-ANOVA + RSM optimization framework, that will identify what the most important factors are when maximizing thermal hydraulic performance on an aluminum slit fin and tube heat-exchanger. The results from F. Wang et al., 2026 show that the two-stage optimization work and that thermal hydraulic can be improved significantly by optimizing the geometry of the heat-exchanger. The six different geometric parameters that are studied are as followed, slit height, slit width, slit length, fin thickness, transverse tube pitch, and longitudinal tube pitch. One of the key take away from the results is that the best overall performing values occur when the geometry is built to maximize the heat transfer rate coupled with a modestly good flow resistance. F. Wang et al., 2026 explains that minimizing friction alone does not maximize the performance as with there testing the worst friction control is case still yields a 10.2% worse performance than the best case. F. Wang et al., 2026 emphasize that it is only when coupled with heat transfer optimization, the metric that dominates performance, that friction factor makes a difference. Another key result from the study is that the most importance geometry is not the slit height, slit width, or slit length but the longitudinal tube pitch and the transverse tube pitch. This is because the spacing between the slits is far more important than the fin shapes themselves, as the spacing contributes the most to the boundary layer redevelopment and the flow mixing intensity.

3.2 Heat recovery

The purpose of this section is to examine existing research on the technical aspects and the performance improvements of heat recovery systems designed for CHP plants and the economic implications of integrating the heat recovery systems. This is done to provide an comprehensive overview of the advancements in technology, operational efficiencies, and sustainability of the heat recovery systems. This is important to address the need for optimizing CHP systems to meet growing demands while minimizing the environmental impacts and allowing economic viability.

The specific objectives of the literature study are the following:

1. Evaluate current knowledge on technical advancements and performance improvements of CHP with heat recovery systems.
2. Benchmarking existing methodologies for assessing the environmental impacts of heat recovery

systems.

3. Identifying the economic implications of integration of heat recovery systems in CHP plants.
4. Comparing the effectiveness of different heat recovery systems when integrated with a CHP plant.
5. Deconstruct the role of the system components and operational parameters influencing the overall energy efficiency and sustainability.

3.2.1 Technical Efficiency

In the article written by Ling and Chen, 2024 the researchers evaluated a waste-to-energy plant in which they modeled a new system that could better utilize the waste heat from the flue gas. This resulted in an energy efficiency increase from 66% to 73%. The system that they modeled was a Rankine cycle which directly recovered high temperature waste heat from the flue gas. After this they used an organic Rankine cycle to capture the waste heat from the steam exiting the RC turbine. Finally they used a heat pump to extract further heat form the flue gas, reducing its exhaust temperature and increasing the useful heat output. These methods are also called cascading waste heat recovery due to their nature of feeding in to the next step in the process, this arrangement allows the system to better utilize the waste heat at different temperatures. Other studies like Ghema et al., 2024 evaluated waste heat recovery systems in poly-generation plants from different studies. This article discusses current limitations, technological advancements and existing challenges, this allows them to establish research gaps in the field of heat recovery. The key metrics which the article defines allows it to evaluate the different systems from a structured framework. A systematic literature review was also conducted by the authors on the existing research on heat recovery systems, this clarifies the current knowledge and highlights the current performance related challenges. The method of evaluation that has been used in the article is a 4E framework, this assess the energy, exergy, economic and environmental aspects. This approach allows the authors to provide a multidimensional understanding of the practical potential and the system performance. The article then consolidates their findings and proposes directions for future research. From their results we can observe that the amount of energy efficiency increases ranges from 20.8%-96.9% from heat recovery systems. In the study by Zhu et al., 2024 they evaluated a Combined Cooling, Heating and Power plant (CCHP). In this plant, a biomass combustion process supplies thermal energy to an intermediate pressurized hot water loop. This heated water transfers energy to an ORC evaporator, vaporizing the organic working fluid. The vapor is used in a turbine to generate electricity. They also use an additional heat recovery step which is to supply heat to the combustion air via an air heater. Using this method they achieved a primary energy saving ratio of around 13%.

The study written by Wu and Xu, 2024 proposes and thermodynamically models a landfill biogas driven multi generation system. This produces electricity, cool water, hot water, hydrogen and fresh water. The biogas from the landfill is the primary fuel source. The system is structured around cascading heat recovery and thermal integration. The heat from the biogas boiler is transferred to the desalination unit where high pressure seawater is evaporated which generates steam to a Kalina cycle. The residual brine water supplies heat to an ORC. There is also a Ammonia Rankine Cycle ARC which generates additional electricity with help from waste heat from the Kalina cycle. Together all these parts produce an energy efficiency of around 93.77%. The system proposed by Liu et al., 2024 is a biomass-natural gas hybrid system cogeneration system with cascading waste heat recovery for imrproved efficiency. The high temperature exhaust gas is sequentially sent through a

steam Rankine cycle (SRC) for power generation. The steam from the SRC is then conveyed to an absorption chiller to produce cooling. The remaining low temperature heat is then recovered by an ORC for additional power generation. The final achieved thermal efficiency is 75.69%.

From these sources it can be seen that integration with heat recovery system often leads to significant energy efficiency improvements. Ranging from a couple percentage points to massive increases in 90% range, depending on what the system is and what fuel type is used.

3.2.2 Environmental Impact

The article written by Y. Wang et al., 2023 conducts an environmental and performance assessment of biomass driven energy configurations focusing on CO_2 emissions and system efficiency. The metric that is used is the CO_2 per unit of electricity generated. This is evaluated through three different systems, a standalone fuel cell, fuel cell with an engine and finally a fuel cell with an engine with heat recovery. The system with heat recovery achieves 93.6% less CO_2 emission compared to the standalone engine system and 33.1% less than the fuel cell hybrid system.

In the study made by Nikafshan Rad et al., 2024 they develop and evaluate a cogeneration plant designed for multi quality waste heat recovery from different industrial processes. An ORC cycle is used for power generation, a vapor ejection refrigeration cycle is used for cooling production. Though they also propose to use three heat-exchangers in their model compared to their basic cycle which only employed one heat-exchanger. This allows the process to recover additional waste heat to improve their heat recovery. This waste heat is channeled to a ejection based refrigeration process, which doesn't use electricity which makes the ejectors improve the general performance of the plant. In their reference system which uses a compressor for cooling production, it can be seen that a large part of equivalent CO_2 is related to the compressor. By using the ejector supplied with waste heat they can reduce CO_2 equivalents by around 11640 tons over its lifetime.

The report written by Nami et al., 2020 which suggest improvements to a previous concept of a CHP & ORC plantation. These modifications include a flue gas condensation unit to recover additional exhaust heat. This additional heat is then used to increase the heat recovered in the ORC cycle, while also lowering the flue gas temperature for use in the district heating section. The evaluated working fluids in the ORC are also chosen for their environmentally friendliness, such as R123/R124. This plant uses municipal solid waste as fuel for their incinerator which is equipped with a steam generator which makes it a conventional Rankine based CHP plant. In this case they use the 4E methodology mentioned earlier and for the environmental part they first compare a standard waste incineration plant to a landfill. For this they used 840 kg of CO_2-equivalents/ton of processed for the land fill and around half that for a average waste incinerator at 415 kg of CO_2-equivalents/ton. To calculate their environmental impact they have considered the released CO_2 and NO_x emissions from the incineration process. This made their system go from a 1.264 sustainability index to a 1.28. Which would means that the system is able to both use the energy more efficiently while also keeping environmental impacts low.

3.2.3 Economic performance

For the study made by Yang et al., 2024 they propose a model of a CHP system in which they evaluate the performance of the energy, exergy and techno-economics. For this an optimization framework has been used which uses polynomial regression and a genetic algorithm, with thermal/exergy efficiency and return on investment as objective functions. Their suggested design uses a shared evaporator for the ORC and a vapor compression heat pump for the waste heat recovery.

The ORC turbine is places after this evaporator and is connects to their first heat intermediate heat-exchanger, while their intermediate heat-exchanger 2 is functioning as a condenser for the ORC system, this allows them to replace the vapor compression heat pump's expansion valve with a pump. This increases the productivity and overall energy efficiency while reducing equipment costs. For the economic analysis they have chosen ROI specifically because it encompasses both the costs and the profits. They achieve a ROI of 0.3 which would mean that their system is profitable, under the optimal conditions for this ROI the exergy destruction of the heat-exchangers is increased. This would entail that the system prioritizes profitability over environmental well being.

In the article written by Karimi et al., 2024 they do an evaluation for the efficiency and cost of a modular cogeneration plant that has a battery storage system, combined with and ORC to recover additional waste heat from the processes. This system is based on an internal combustion engine which is a CHP for this study. The ORC is added to the exhaust line to recover the waste heat. Excess electricity is stored in the battery, while cooling and heating is supplied on demand. For the economic simulations they have focused on two configurations, maximum energy storage and maximum energy savings. Though a result from this study is that the addition of the heat recovery ORC increases the payback period and the net present value of the system to a small degree. Though the findings from the article also show that the addition of the ORC helped to reduce the environmental cost of the entire system, this combines well with the minor economic consequences of the implementation of the ORC.

3.2.4 Analysis and Synthesis

The literature on the combined heat and power plants with integrated heat recovery systems demonstrates the highlighted technical advancements improve system performance and efficiency. These innovations include Rankine, Organic Rankine, Kalina, absorptions cycles and multi-level heat recovery systems – enable energy efficiencies exceeding 70% and exergy efficiencies of 50%. Hybrid systems that combine these cycles alongside the integration of renewable fuels such as biomass and biogas consistently show enhanced thermodynamic performance and operational flexibility. Retrofitting approaches further bolster these gains by incorporating gas turbines and heat recovery boilers, this yields improved power output and fuel savings. Parametric and multi-objective optimization techniques are critical for designing operational parameters like turbine inlet temperature, compressor pressure ratio, and refrigerant selection to maximize efficiency and sustainability.

The environmental assessment of the reviewed works highlight significant reductions in GHG emissions and other pollutants achieved through heat recovery integration. Several systems approach zero or even negative CO_2 emissions by using renewable fuels and CO_2 capture technologies. The environmental benefits extend to lowering NO_x and SO_2 emissions and mitigating waste heat discharge impacts. Environmental evaluations mostly focus on operational emissions with less attention to life cycle assessments and indirect environmental costs related to materials and refrigerants. Studies do highlight the critical influence of the working fluid and refrigerant choices based on environmental footprints.

The economic analyses consistently confirm the financial viability of integrating heat recovery systems in CHP plants, with payback periods commonly ranging from 1-7 years. Multi-objective optimization frameworks balance cost, efficiency and environmental goals, improving return on investments and reducing operational expenses. Sensitivity analysis reveal that fluctuations in fuel prices, electricity cost and components substantially affect economic outcomes. While hybrid configurations with energy storage and advanced cycles may entail highers initial capital cost, their long term benefits in cost savings and emissions reduction are beneficiary.

Together the literature indicates that the effectiveness of heat recovery technologies is strongly influenced by system configuration complexity, operational parameters and fuel type. Cascade and multi-level heat recovery systems outperform single stage setups, especially when combined with environmentally friendly refrigerants and optimize thermal matching. However the diversity of methodologies and system design limits the generalization of findings, signaling the need of standardized evaluation protocols. Furthermore real world validation, long term operational data and integrated techno-economic-environmental frameworks remain underdeveloped areas. Addressing these gaps is critical for advancing the deployment of high efficiency, sustainable CHP plants with heat recovery systems that optimize energy use, minimize environmental impact and ensure economic feasibility.

3.2.5 Summary Table of the Reports for heat recovery

Article	Technical	Environmental	Economic	Configuration	Influence
Ling and Chen, 2024	Energy efficiency up to 73% with cascade system.	Significant CO_2 reduction via exergy analysis.	NPV of 1.59 million USD, payback 6.5 years.	Cascade of Rankine, ORC, heat pump cycles.	Pump and turbine isentropic efficiency critical.
Ghema et al., 2024	Energy efficiency range 20.8–96.9%, exergy 24.1–63.6%.	Average 2.45 kt CO2 reduction per MW annually.	Payback 1.4–6.7 years, competitive LCOE.	WHR-based polygeneration systems.	heat-exchanger and system design affect efficiency.
Zhu et al., 2024	Primary energy saving ratio up to 13%.	Negative CO_2 emissions due to CO_2 capture.	Optimization of thermo-economics with a payback analysis.	Biomass CCHP with an ORC and CO_2 capture.	Refrigeration coefficient and condensation temperature are decisive.
Wu and Xu, 2024	Energy & Exergy efficiency 93.77% & 55.17%.	0.3 CO_2 ton/MWh.	17.65 $/GJ.	Multi-heat recovery with Kalina, ORC and ejector.	Refrigerant influences performance.
Liu et al., 2024	Thermal & Exergy efficiency 75.69% & 41.76%.	Combustion chamber is main component of exergy destruction.	Sum unit cost of the product is 13.37 $/GJ	Gas turbine with steam Rankine, ORC and absorption chiller.	Turbine inlet pressure and condensation temperature are decisive.
Y. Wang et al., 2023	Energy&Exergy efficiency 69.4% & 52.1%.	97.7% lower emissions than petroleum plants.	Specific fuel cost 1.96 USD/GJ.	Biomass gasification hybrid.	Waste heat improved overall efficiency.
Nikafshan Rad et al., 2024	Exergy performance increased by 10.3%i	CO_2 reduction of 11640 tons	Annual cost reduced by 21.6%	Multi level heat recovery with ORC and ejector	Refrigerant choice influences performance
Nami et al., 2020	Sustainability index increased from 1.264 to 1.28.	Most exergy destruction in the incinerator.	30.88$/GJ ORC turbine	Waste-driven hybrid CHP with flue gas condensation.	Flue gas condensation improves system sustainability.
Yang et al., 2024	Thermal & Exergy efficiency 17% & 97%.	Negligible global warming impact.	ROI 0.30.	Mass-integrated with multi-objective optimization.	heat-exchanger has the highest exergy destruction.

Table 1: Table of compiled sources, part 1

Article	Technical	Environmental	Economic	Configuration	Influence
Karimi et al., 2024	Primary Energy Saving improved by 22%	CO_2 reduced by 25%	Increase in payback period.	Modular cogeneration with ORC and battery storage.	Economic parameters affect payback.
Dzierwa et al., 2024	15MW gained in the recovery boiler.	Reduction of 26 Mkg CO_2/year.	Payback period of about 4 years.	Flue gas heat recovery with condensation exchanger.	Flue gas temperature and condensate impact performance.
Hai et al., 2024	Exergy efficiency increased by 10.1%	CO_2 emissions reduced by 11000 tons.	Annual cost reduced by 21.16%	Multi level heat recovery with ORC and ejector.	Refrigerant selection critical for environmental impact.
Meftahpour et al., 2024	Efficiency increase from 53.60% to 54.31%.	Insignificant environmental cost.	8.8 % reduction for unit cost.	Kalina cycle integrated with gas turbine and HRSG.	Combustion chamber and HRSG dominate the exergy losses.
Nutakki et al., 2024	Energy & Exergy efficiency 54.26% & 29.14.	CO_2 emissions 0.65 kg/kWh.	Unit exergy cost 10.28 $ /GJ	Biomass gas turbine with ORC, absorption chiller and desalination.	Thermal matching is critical for performance.
Hu et al., 2024	Exergy efficiency 63.09%	0 CO_2 emissions achieved	Levelized cost of storage 0.1$/kWH.	System with ORC, CO_2 capture and LNG cold energy.	Exergy destruction mainly in cold energy subsystem.

Table 2: Table of compiled sources, part 2

4 Current study

In this section we explain the empirical foundation and the process used for collection and processing of the operational data used. We also talk about the design features of our model, some assumptions we made, the simulation process, and the overall structure of the economic evaluation.

4.1 Raw Data

Examples of the minute-resolution operational data retrieved from Mälarenergi's recorded data bank.

Timestamp	**Air inlet temperature** ℃	**Air $\dot{Q}$** Nm^3/s	**Water inlet temperature** ℃	**Water $\dot{Q}$** m^3/s
2025-01-01 00:00	10.2525	47.6462	32.0462	449.4366
2025-01-01 00:01	10.2525	47.7763	32.0462	450.5306
2025-01-01 00:02	10.2525	47.4778	32.0462	448.5756
2025-01-01 00:03	10.2525	47.4987	32.0462	449.3228
2025-01-01 00:04	9.8538	47.4221	32.0462	450.9851
2025-01-01 00:05	9.7261	47.4221	32.0462	450.7623
2025-01-01 00:06	9.7261	47.2704	32.0462	446.7478
2025-01-01 00:07	9.7261	47.6034	32.2721	450.4442
2025-01-01 00:08	9.7261	48.4097	32.4147	448.7571
2025-01-01 00:09	9.7261	47.9576	32.4147	452.2243

Table 3: Measured values for air temperature, combustion air flow, water temperature, and water flow (January 1)

Timestamp	**Air temperature** ℃	**Combustion air $\dot{Q}$** Nm^3/s	**Water temperature** ℃	**Water $\dot{Q}$** m^3/s
2025-08-17 00:00	29.8451	24.6846	51.3848	449.0780
2025-08-17 00:01	29.8451	24.6773	51.3848	450.3321
2025-08-17 00:02	29.8451	24.5531	51.3848	450.4068
2025-08-17 00:03	29.8451	24.9235	51.3848	448.9572
2025-08-17 00:04	29.8451	24.9642	51.3848	448.6612
2025-08-17 00:05	29.8451	24.7751	51.3848	448.5090
2025-08-17 00:06	29.8451	24.7078	51.3848	450.2908
2025-08-17 00:07	29.8451	24.7799	51.3848	447.9677
2025-08-17 00:08	29.8451	24.7854	51.3848	451.0130
2025-08-17 00:09	29.8451	24.7854	51.3848	450.5672

Table 4: Measured values for air temperature, combustion air flow, water temperature, and water flow (August).

Timestamp	**Air temperature** ℃	**Combustion air** $\dot{Q}$ Nm^3/s	**Water temperature** ℃	**Water** $\dot{Q}$ m^3/s
2025-12-01 00:00	11.9371	38.0669	43.2296	449.9518
2025-12-01 00:01	11.9371	38.1491	43.2296	451.2736
2025-12-01 00:02	11.9371	38.0004	43.2296	447.6768
2025-12-01 00:03	12.1276	38.3112	43.2296	449.4352
2025-12-01 00:04	12.2530	38.2984	43.2296	448.9975
2025-12-01 00:05	12.2530	37.9968	43.2296	449.9176
2025-12-01 00:06	12.2530	37.8405	43.2296	450.3452
2025-12-01 00:07	12.2530	38.3101	43.2296	451.5175
2025-12-01 00:08	12.2530	38.2516	43.2296	451.4284
2025-12-01 00:09	12.2530	37.8740	43.2296	449.3908

Table 5: Measured values for air temperature, combustion air flow, water temperature, and water flow (December 1).

4.2 Data processing

To process the data and make it more usable the minute data throughout the year 2025 was divided into 10 minute averages. This was done in excel using a block-averaging formula. The following expression was used:

=AVERAGE(INDEX($AD$11:$AD$525600;(ROW()-11)*10+1):INDEX($AD$11:$AD$525600;

(ROW()-11)*10+10))

and it creates a shifting reference window that extracts consecutive ten-row segments from the data range AD11:AD525600. The function ROW() is used to generate a counter as the formula is filled downward. By subtracting 11 from the row number, the counter is aligned such that it begins at zero in the first output row. This counter is then multiplied by ten to determine the offset required to locate the start of each ten-row block within the dataset.

The two INDEX functions use this offset to identify the first and last entries of the current block, thereby defining a range of ten observations. As the formula is copied downward, the start and end positions shift by exactly ten rows for each successive output row. The AVERAGE function then computes the mean value of the observations contained within this defined interval. Through this, each row of the resulting column represents the average of a distinct ten-row segment of the original time-series data.

In conclusion the Excel formula computes the average of non-overlapping blocks of 10 consecutive samples. For the $k-$th output row (staring at $k = 0$), the block-average is defined as:

$$\overline{x}_k = \frac{1}{10} \sum_{j=10k+1}^{10k+10} x_j \tag{30}$$

10 minute average of the minute-resolution operational data retrieved from Mälarenergi's recorded data bank.

Timestamp	**Air temperature** ℃	**Combustion air** $\dot{Q}$ Nm^3/s	**Water temperature** ℃	**Water** $\dot{m}$ kg/s
2025-01-01 00:00	9,9494	47,6484	37,0923	124,9385
2025-01-01 00:10	9,9239	47,5444	37,4132	125,0321
2025-01-01 00:20	10,4042	46,6577	37,7074	124,8620
2025-01-01 00:30	9,9337	46,2722	38,0081	125,0835
2025-01-01 00:40	9,7261	46,5684	38,2995	124,9883
2025-01-01 00:50	10,2459	46,7566	38,2477	125,0220
2025-01-01 01:10	9,7519	46,8794	38,3204	124,9170
2025-01-01 01:20	9,6259	47,2935	38,3613	124,9922
2025-01-01 01:30	10,8492	46,8190	38,1763	125,0588
2025-01-01 01:40	10,4672	46,7329	37,9706	124,9996

Table 6: Average measured values for air temperature, combustion air flow, water temperature, and water flow (January 1).

During the evaluation of the results, some data points were identified as unrealistic and were therefore removed. The issue originated from periods of low water-side mass flow ($0 < \dot{m} < 7$). Under these conditions, the additional heat-exchanger heated the combustion air before it entered the humidifier. Because warmer air can hold a higher moisture content, the injector supplied more water than intended, which in turn caused a larger temperature drop than in the reference system, as more water evaporated. With the low water-side mass flow, the heating element in the humidifier was unable to compensate for the increased cooling caused by the water injection, which resulted in a reduction of the overall air temperature.

To address this issue, a decision was made to bypass the humidifier whenever the water-side mass flow fell below 1 kg/s. This ensured more realistic system behavior and more reliable results.

4.3 Design features

The study evaluates four system designs: the replicated system, which represents Mälarenergi's current configuration, and three modified designs that incorporate an additional heat-exchanger with total efficiencies of 70%, 80%, and 90%, respectively. A single model structure is used for both the replicated and the heat-exchanger configurations, with identical boundary conditions and input parameters; the only differences in the results arise from the inclusion of the additional heat-exchanger and the variation in its design efficiency.

The humidifier implemented in the model represents a simplified approximation of a real humidification process. The main deviation from an actual system concerns the water injection mechanism. Since Ebsilon®Professional does not provide a humidifier component capable of producing a positive temperature increase on the air side, a custom configuration was developed. In contrast to a real humidifier, where water injection and convective heat transfer occur simultaneously, the model separates these processes into two distinct stages.

First, the humidity level is measured by a controller located at the outlet of the humidifier section. Based on this measurement, the controller adjusts the splitter fraction to regulate the amount of water injected into the air stream, ensuring that the relative humidity reaches 95% at the humidifier outlet. As a result of this configuration, the relative humidity upstream of the heat-exchanger section

becomes significantly higher, approaching nearly 100%. When the air is subsequently heated, its capacity to hold moisture increases, causing the relative humidity to decrease accordingly.

For the replica system, the water outlet temperature, water mass flow, and air inlet conditions were provided as fixed inputs. This in turn gave us the resulting water inlet temperature. Since the outlet temperature change after adding the additional heat-exchanger the inlet temperature from the replica system was instead used as a input in the additional heat-exchanger system.

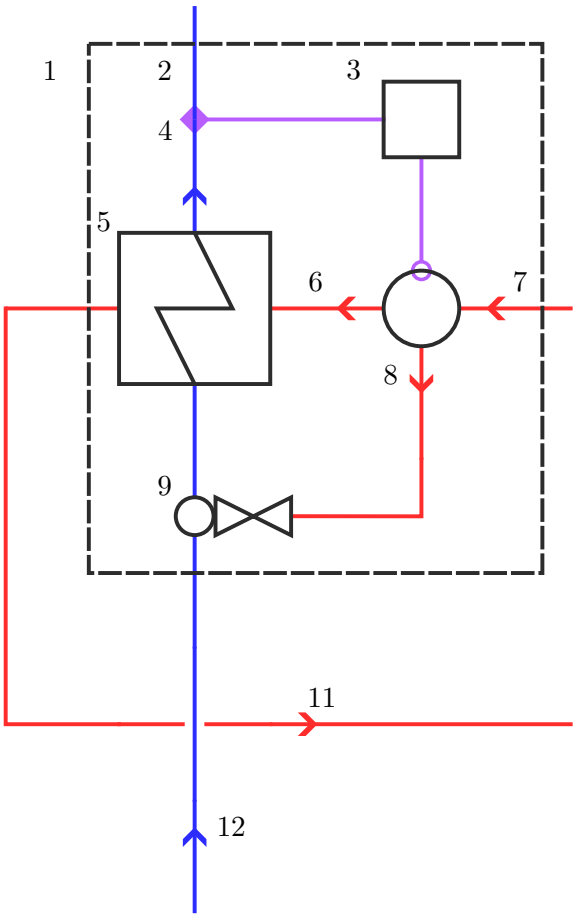

Figure 5: Replicated system.

The design values for the different scenarios:

Design	**Replica**	**70%**	**80%**	**90%**
Water Inlet (7)				
T ℃	38.876	38.876	38.876	38.876
P bar	1.000	1.000	1.000	1.000
m kg/s	125.398	125.460	125.469	125.479
Air Inlet (12)				
T ℃	22.514	22.514	22.514	22.514
P bar	1.000	1.000	1.000	1.000
m kg/s	40.451	40.451	40.451	40.451
RH %	30	30	30	30

Table 7: System design values:

Design	**70%**	**80%**	**90%**
TTA (nominal) KA_N [kW/K]	50.797	68.348	98.539
HXC (nominal) Q_N [kW]	373.221	428.169	483.531

Table 8: Heat-exchanger design values:

The use of characteristics curves in the humidifier and the added heat-exchanger is used to more accurately represent real life operations. Since the heat-exchanger is built and designed for a specific water and air mass flow, having a efficiency characteristics curve would in turn give more accurate simulations. When using part load on the system, since it is designed for a 100% load using less load would in turn lower the efficiency of the heat-exchanger. With the characteristics curve this reduction or increase in efficiency can be simulated when using off-design input parameters. The characteristic curves of the humidifier differ between the water side (4.3) and the air side (4.3). This design choice arises from the need to construct a custom humidifier model, as the real system's design features were not available. The use of separate characteristic curves, one for the water side and one for the air side follows the default configuration of the component employed in the model (comp 25), as described in EBSILON®Professional, 2026.

The additional heat-exchanger (HEX) is modeled using the same characteristic curves (4.3) for both the air and water sides. All heat-exchangers with efficiencies of 90%, 80%, and 70% are assigned identical characteristic behavior. This modeling choice enables a clearer isolation of the effects of efficiency variation on temperature profiles and, consequently, on the overall performance of the system. In the following graph, kA_2 referrer to the thermal transmittance in the water side, and kA_1 referrer to the thermal transmittance in the air side. The kA_N referrer to the design kA at the specific medium illustrated by the graphs.

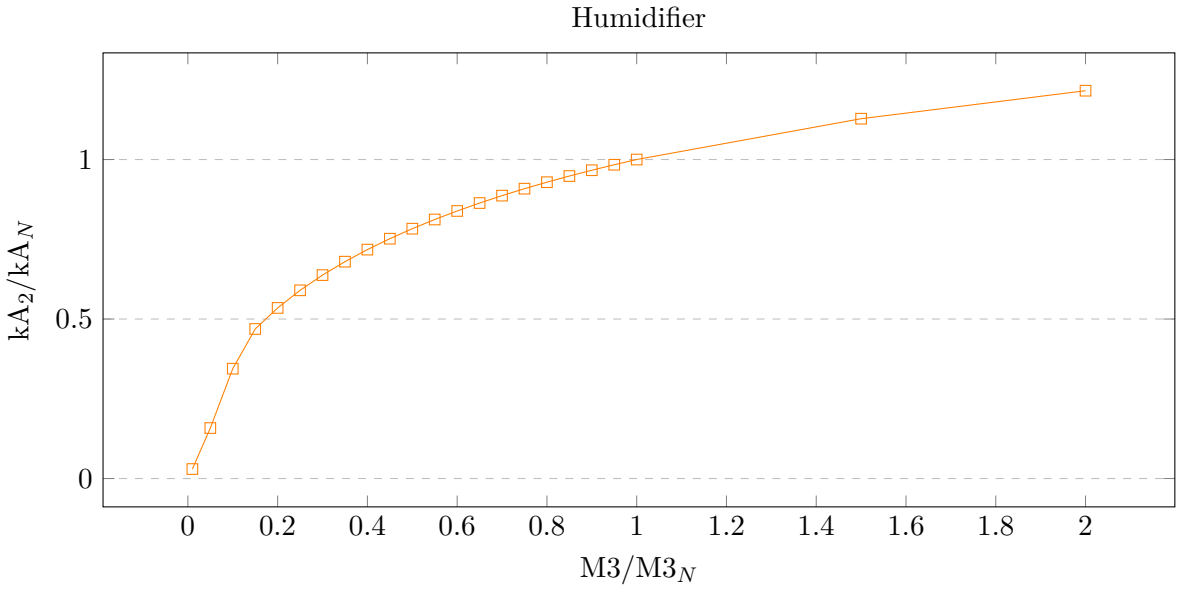

Figure 6: Humidifier water side characteristics

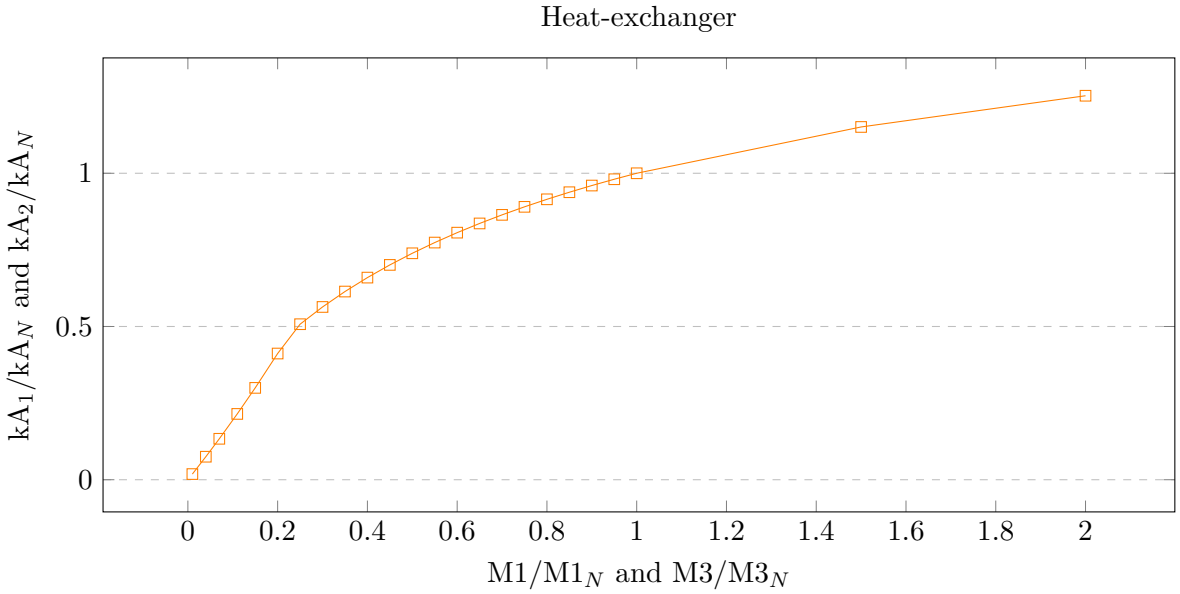

Figure 7: Humidifier air side characteristics, Heat-exchanger water and air side characteristics

4.4 Simulations

All calculations were performed in Excel using the Ebsilon®Professional add-in to execute and simulate off-design calculations for the four different system configurations.

The calculations were carried out in two sequential steps. In the first step, all replica-system simulations were performed using ten-minute averaged input data for each humidity level (30%, 50%, and 70%). Once the replica simulations were completed, the resulting outlet temperatures were used to determine the corresponding inlet temperatures required for the second step. In the second step,

the system including the additional heat-exchanger was simulated. All simulations in this stage used identical boundary conditions, namely the same inlet temperature, pressure, and volumetric flow rate. The only parameter that differed between the scenarios was the total heat-exchanger efficiency, while the underlying heat-exchanger characteristic curves remained unchanged.

The results from the various efficiency scenarios were then recorded and compared to one another to enable further analysis and discussion regarding the potential power savings and the associated payback period after the potential installation of the heat-exchanger.

4.5 Assumptions

Since we did not get any design features or performance metrics on the humidifier we designed our own in Ebsilon®Professional. All pressure losses in the air side was assumed to be zero to make calculations easier. During modeling of the system it was observed that changing the water pressure had minimal impact on the system performance and was therefore negligible, thus the humidifier inlet water pressure was set to the air pressure.

During the design and simulation process, the water density is assumed to be constant when calculating the water mass flow from the given volume-flow, $\rho = 1000\, kg/m^3$ and the mass flow rate is defined immediately downstream of the humidifier. This configuration ensures that the simulations remain feasible even when the humidifier is bypassed (i.e., when the water-side mass flow is set to 0 kg/s). Ebsilon®Professional requires a non-zero mass flow through the water injector at all times; however, this does not influence the analysis, as the study focuses on differences in energy output, and the imposed minimum flow does not introduce any measurable deviation.

With this setup, the heated air stream is capable of absorbing a larger amount of water vapor when the heat-exchanger is active. Consequently, the total water mass flow upstream of the humidifier becomes higher than in the reference model. This effect is most pronounced in the 90% efficiency case, where the total mass flow increases by approximately 0.041 kg/s. This discrepancy is taken into account when interpreting the results and formulating the conclusions.

4.6 Economy

From Mälarenergi the estimated cost of installation is 7-10 MSEK for the additional heat-exchanger, this is our CAPEX. From this heat-exchanger we have estimated to save around 359-482 MWH energy per year. This can reduce their use of auxiliary steam by the same amount every year. This auxiliary steam cost them around 260 SEK/MWH. Mälarenergi also gets energy certificates from block 7, these can vary in price, ranging from 10 SEK/MWh to 0.1 SEK/MWh. In addition to this Mälarenergi also gets a flat rate of 10 SEK/MWh for its produced power, due to it being considered environmentally friendly. This results in the following:

$$\frac{7 \times 10^6 \text{ SEK}}{359.1922 \text{ MWH/y} \times 260 \text{ SEK/MWH}} = 74.95 \text{ years} \tag{31}$$

$$\frac{7 \times 10^6 \text{ SEK}}{420.1263 \text{ MWH/y} \times 260 \text{ SEK/MWH}} = 64.08 \text{ years} \tag{32}$$

$$\frac{7 \times 10^6 \text{ SEK}}{482.4944 \text{ MWH/y} \times 260 \text{ SEK/MWH}} = 55.79 \text{ years} \tag{33}$$

With data from Energimyndigheten, 2024 we can make an estimation of revenue gained if the proposed system were to be installed since block 7's inception.

	2020	2021	2022	2023	2024	2025	Unit
Mean Price	11.9	1.7	0.8	0.4	0.4	0.5	[SEK/MWh]
Revenue 1	10575.9	5635.4	5205.6	5000.8	5008.8	5061.0	[SEK]
Revenue 2	7877.1	4197.3	3877.2	3724.6	3730.6	3769.5	[SEK]
Payback 1	51.5	52.4	51.6	50.7	49.7	48.6	Years
Payback 2	69.2	70.7	70.0	69.1	68.1	67.0	Years
Payback Utopia	19.9					15.3	Years

Table 9: Dynamic payback period with energy certificates included.

$$\frac{\text{CAPEX} - \sum_{i=1}^{n}(\text{INCOME}_{i-1})}{\text{INCOME}_i} \tag{34}$$

This is the equation used to calculate the payback period, where i is the year. The CAPEX used for this table is 7 MSEK, with the exception of the utopia column, which uses the best scenario from figure 14. These cost are 5 MSEK for the CAPEX and an auxiliary steam price of 500 SEK/MWh. The revenue 1 is calculated using the high end of the simulated power gain while revenue 2 is using the low end of the results, this is also the case for payback 1/2. What the payback period shows in this table is actual payback period if said calculation had been done at that point in time. This is also why the payback period goes up and down in the beginning, when the revenue was high due to high electricity certificate prices.

5 Results

5.1 Simulations

In this section we have plotted the results in graphs showcasing both the temperature increase and power increase from the simulated additional heat-exchanger.

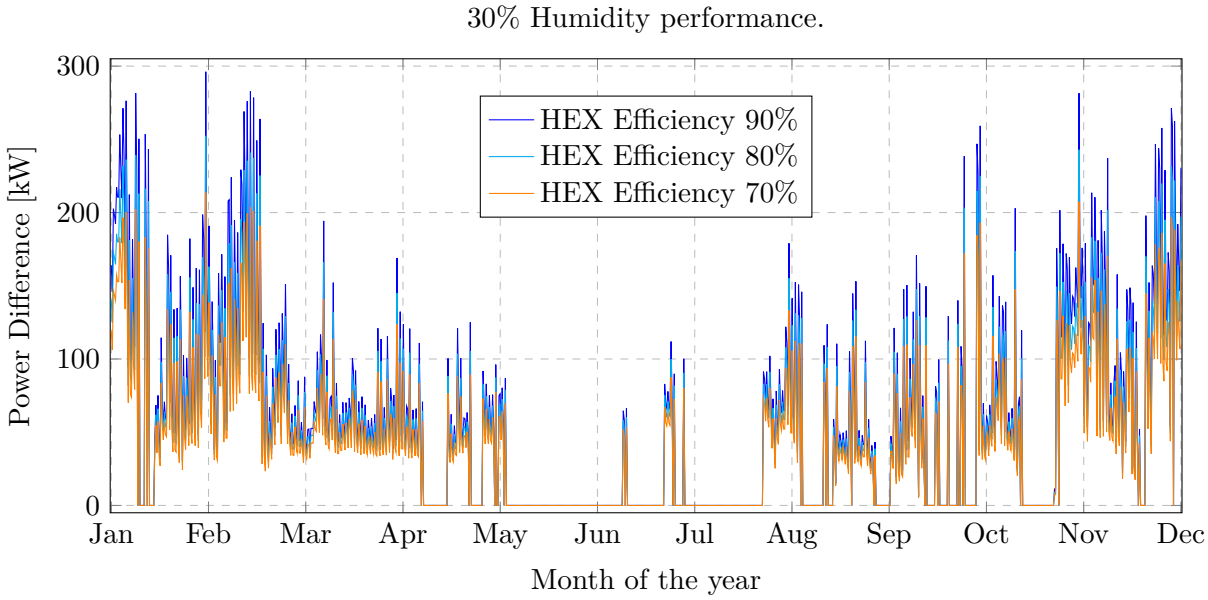

Figure 8: Power saving throughout the year, 30% humidity relative humidity.

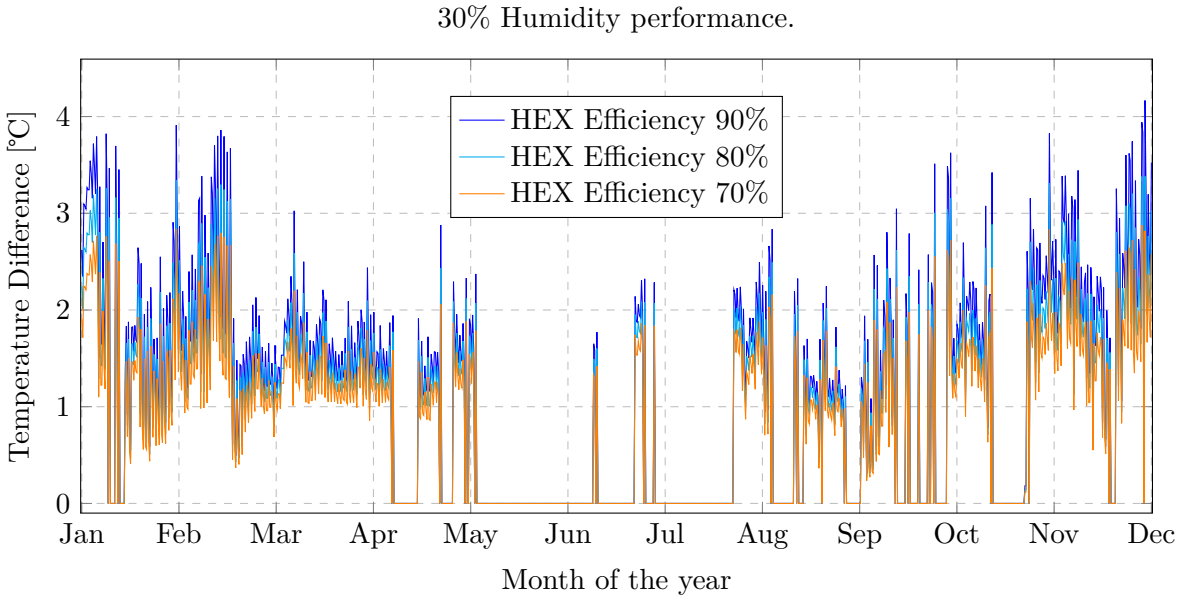

Figure 9: Temperature difference throughout the year, 30% humidity relative humidity.

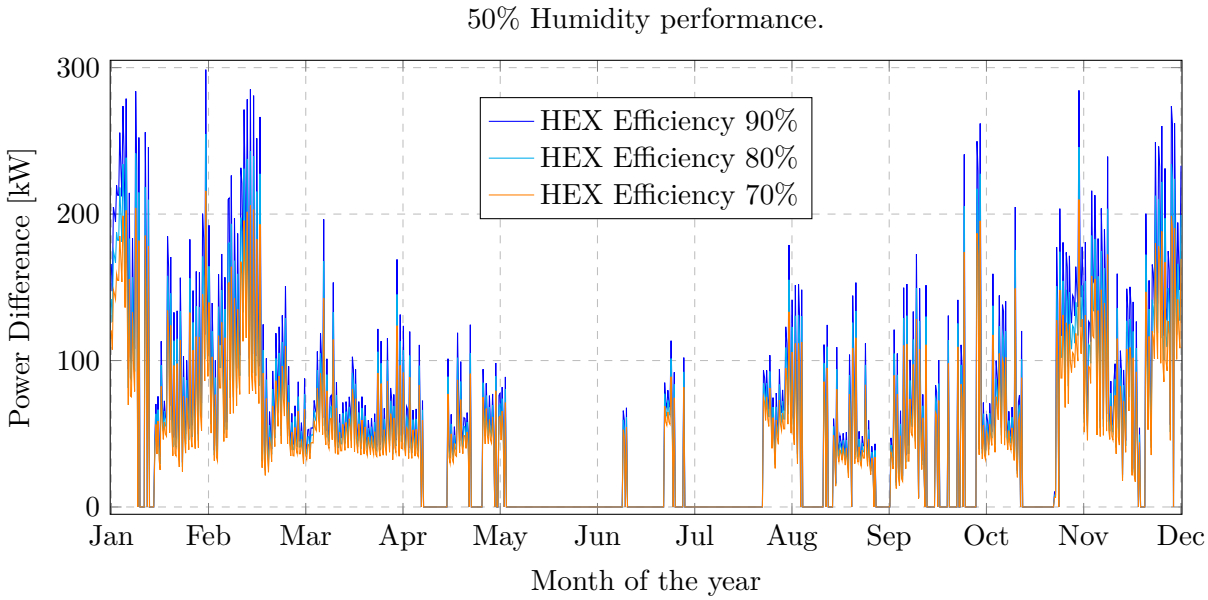

Figure 10: Power saving throughout the year, 50% humidity relative humidity.

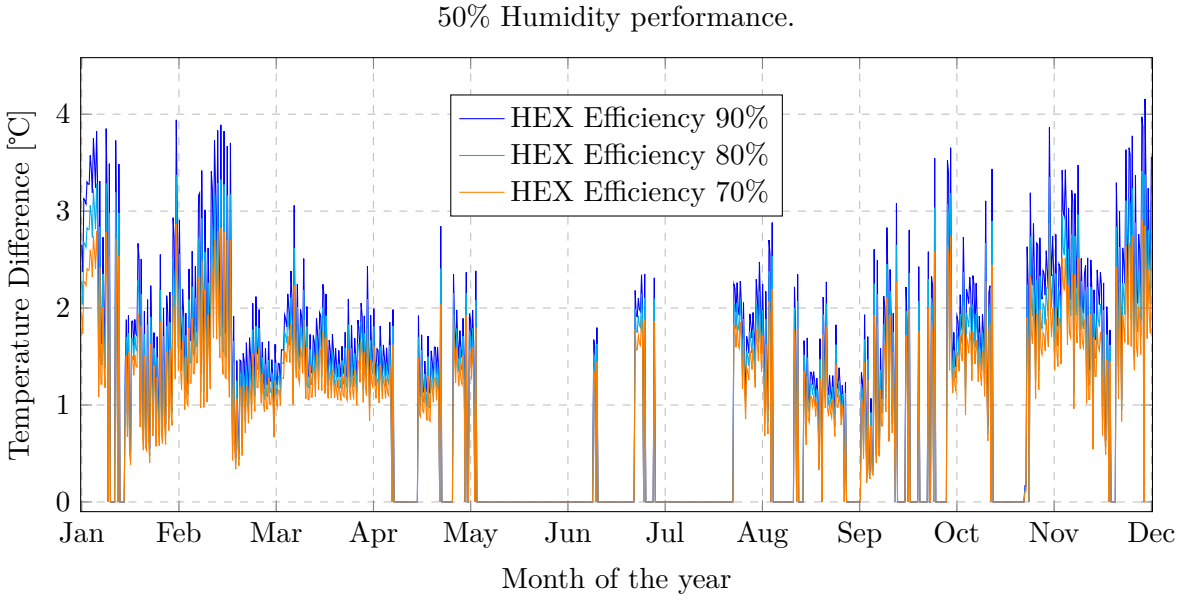

Figure 11: Temperature difference throughout the year, 50% humidity relative humidity.

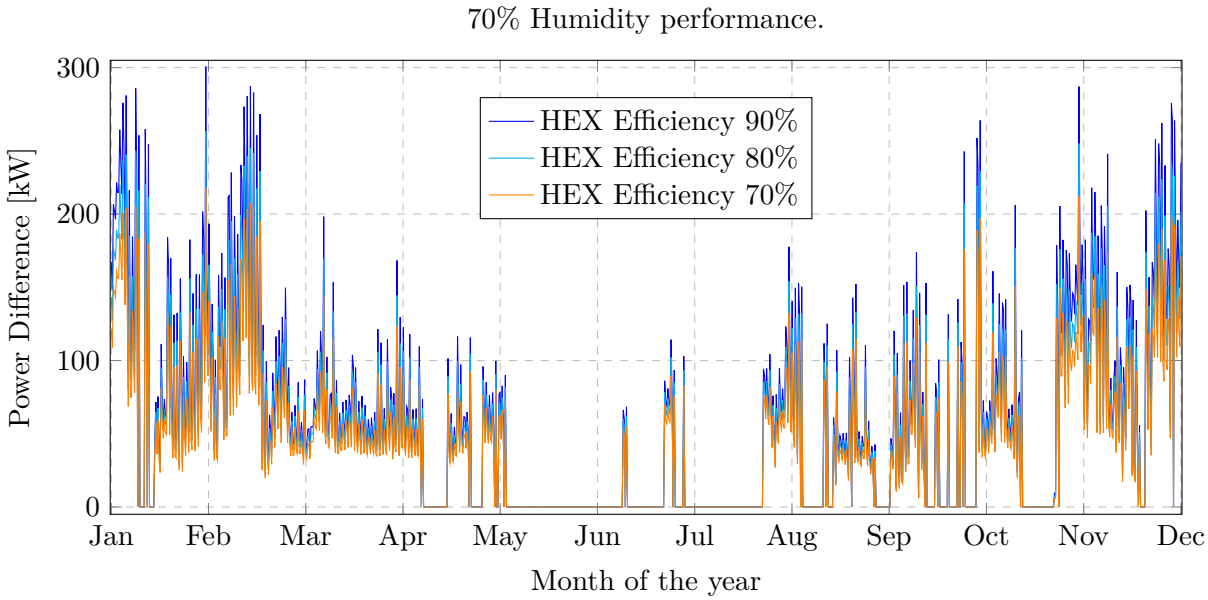

Figure 12: Power saving throughout the year, 70% humidity relative humidity.

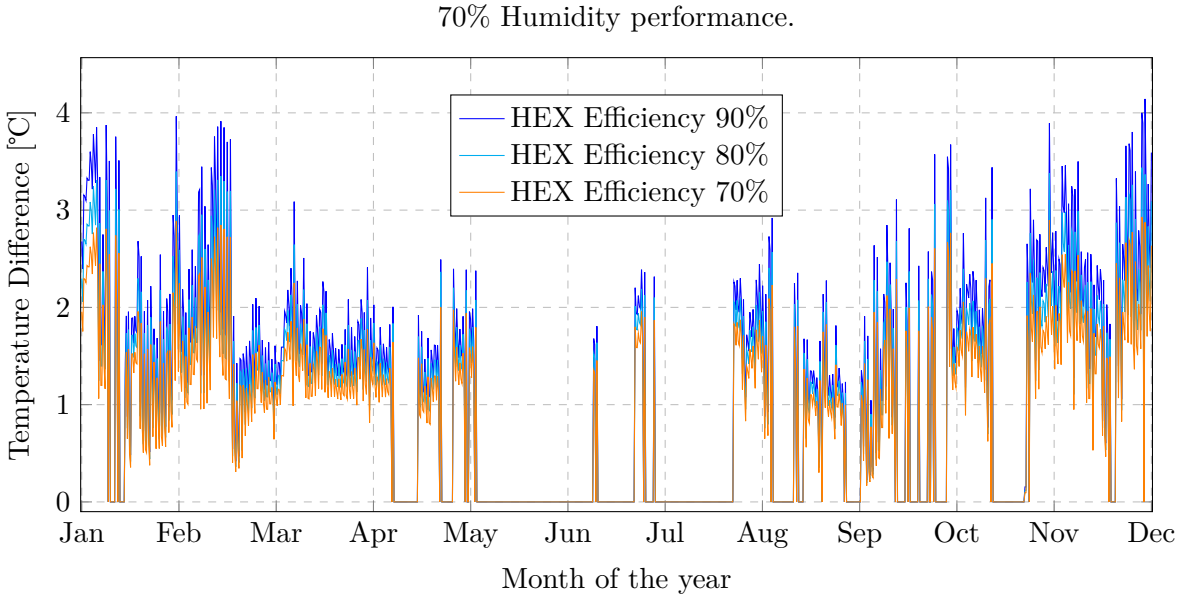

Figure 13: Temperature difference throughout the year, 70% humidity relative humidity.

RH \ η	70%	80 %	90%
30%	359.1922	415.0789	473.9739
50%	363.824	420.1263	479.3952
70%	366.5789	423.074	482.4944

Table 10: Yearly energy recovered [MWh]

RH \ η	70%	80 %	90%
30%	1.4036	1.6105	1.8223
50%	1.4243	1.6328	1.8460
70%	1.4357	1.6447	1.8582

Table 11: Average air outlet temperature increase [°C]

$$\frac{479.3952 \quad [\text{MWh}]}{363.824 \quad [\text{MWh}]} = 1.3177 \tag{35}$$

5.2 Calculations

Average power increase at different temperature increases. (calculations done at 90% efficiency, 70% humidity)

$$k = \text{temperature } ^\circ\text{C}, \quad j = \text{power } [kW]$$
$$\frac{\sum_{i=1}^{n} j(i)}{n} = [kW] \tag{36}$$

$$I = \{i \in \{1, ..., 31387\} \mid 0.9 < k(i) < 1.1\}$$
$$n = length(I) = 1297, \quad \frac{\sum_{i=1}^{1297} j(i)}{1297} = 51.7301\,[kW] \tag{37}$$
$$I = \{i \in \{1, ..., 31387\} \mid 1.9 < k(i) < 2.1\}$$
$$n = length(I) = 3358, \quad \frac{\sum_{i=1}^{3358} j(i)}{3358} = 86.1021\,[kW] \tag{38}$$
$$I = \{i \in \{1, ..., 31387\} \mid 2.9 < k(i) < 3.1\}$$
$$n = length(I) = 548, \quad \frac{\sum_{i=1}^{548} j(i)}{548} = 182.9316\,[kW] \tag{39}$$

°C \ η	70%	80 %	90%	Average
1	42.3938	46.8387	51.7301	46.9875
2	109.7712	104.1950	86.1021	100.0228
3	198.7599	200.4163	182.9316	194,0359

Table 12: Average power increase taken at 70% humidity at different temperature increases [kW].

5.3 Payback Period

Below here is the calculated payback period with varying costs and revenue.

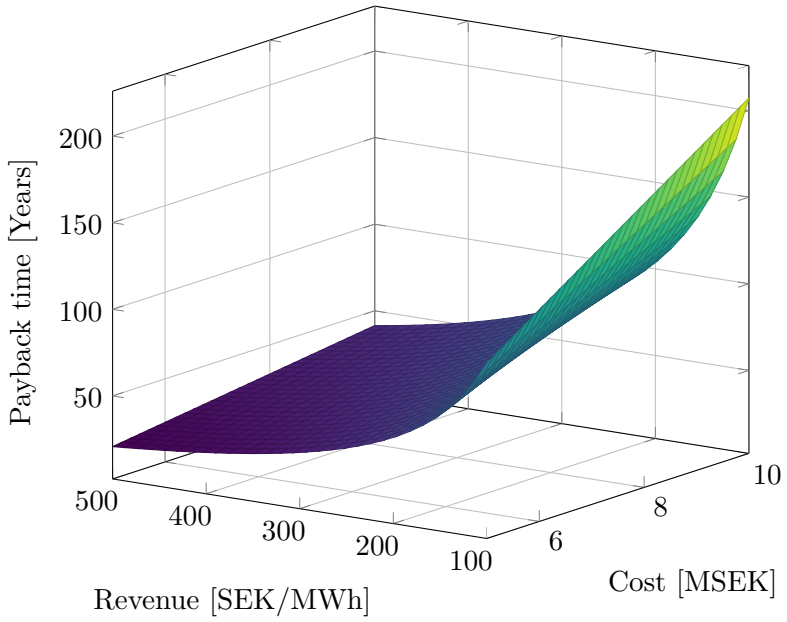

Figure 14: Payback period, as a function of initial investment cost and revenue.

RH \ η	70%	80 %	90%
30%	74.9545	64.8625	56.8029
50%	74.0003	64.0833	56.1605
70%	73.4442	63.6368	55.7998

Table 13: Payback period over the different RH and η, at 260 [SEK/MWh].

6 Discussion

In this section, the results of the study is discussed in relation to our research objectives and previous literature. The influence of humidity, heat-exchanger efficiency and temperature on the system performance is discussed and analyses. Along with a discussion about economic feasibility and different methodological approaches.

6.1 Result

The resulting power savings and their respective temperature gain show that there is a positive outcome, and we observe that there is a noticeable energy increase by using our installed heat-exchanger. In practical terms, the system recovers more energy than in the reference case, and this effect is visible across the simulated operating conditions. But with the current goals and purpose of the study other factors must also be taken into consideration, like cost and payback period. By looking at and using our study we see that the feasibility of installing a similar heat-exchanger in block 7 would be impractical. This study has shown that utilizing low grade energy sources, or sources with low energy content is both hard and result in low energy recovery.

When looking at our results we observe that inlet humidity, even though it is small, makes a difference, especially when you compare low intake air humidity levels (30%) and high humidity levels (70%). In all three efficiency cases the extra saved energy is around 10 MWh, with a small decline in growth between the three humidity levels. This is also the case for the temperature increase, though less noticeable, where it increases by around 0.03℃ between the 30% humidity and 70% humidity. Air with higher RH before the humidification process can absorb heat at higher rate due to having already higher water content, this is due to water having higher heat transfer coefficient than air and the higher the RH the higher the water content in the fluid.

Comparing the different results we can see unlike the difference between the humidity cases, using a higher efficiency heat-exchanger has a great impact on the system performance. Between the 70% efficiency heat-exchanger and the 90% the increase in saved energy is around one third, a 32% recovered energy increase as seen in equation 35. A higher efficiency heat-exchanger allows higher transfer rate between the two fluids, this would entail more heat from the water side being transferred to the air side. With higher inlet and outlet temperature of the humidifier the less auxiliary steam is needed to heat it up before the boiler.

The temperature increase from the results which can be seen in the three graphs (9, 11, 13) and in a condensed form in table 11. Through this table it can be seen that the average temperature increase from η regardless of the efficiency is around 0.21℃. To analyze the gain at each temperature step (T = 1, 2, 3) we have taken the average gain in power in the span of T $\pm$ 0.1℃to get sufficient data points. The increase at each of the steps are different, at T = 1 the gain in power is around 47 kw, while at T = 2 its around 100 kw and finally at T = 3 its 194 kw. This pattern is interesting and show that working with low grade energy, and low temperature differences can be difficult as these small temperature increases give less energy benefit than if you would have a large temperature increase.

Like with the saved power, the payback period also shows a the declined growth between the humidity cases, with a rough approximation of a 0.9-0.7 years between 30% humidity and 50% across the different efficiencies and 0.55-0.35 years between 50% and 70% humidity. Also like the saved power, the difference in the payback period between efficiencies is almost constant, with a almost linear decrease of 10-8 years between the different efficiencies.

In figure 14 a sensitivity analysis has been conducted, where the cost and revenue have been varied. The cost ranges from 5-10 million SEK while the revenue has a range of 500-100 SEK/MWh. This was done on the data set of 50% RH and 80% efficiency. Even though both the revenue and investment cost has been changed to benefit a smaller payback period it doesn't change it to a reasonable degree. Even though the investment has been halved and the revenue has been doubled, the payback period still sits around 20-25 years.

From the simulation results it can be observed that the heat-exchanger efficiency is the most influential parameter affecting the system performance. From those results it can be observed that transitioning from a 70% efficiency to a 90% efficiency will yield around 32% more recovered energy to the system. This is a rather large improvement which demonstrates the importance of using high efficiency heat-exchangers for waste heat recovery systems in CHP plants.

The heat-exchanger efficiency determines the fraction of theoretical thermal energy that can be transferred between the hot and cold fluid with a heat-exchanger. Thereby a high efficiency heat-exchanger can extract more thermal energy from the water side and delivering it to the air side. The enhanced heat transfer achieved with higher efficiency equipment directly reduces auxiliary steam consumption in the CHP system. By elevating the air before it goes to the boiler less auxiliary steam is needed for normal operation.

By analyzing the annual heat recovery and the sensitivity analysis we see that despite using a great efficiency heat-exchanger and a low investment cost, the limiting factor to why the installation is unfeasible is because of the low energy availability in the hot side medium coming form the scrubber. The amount of energy provided from the scrubber is too low, making the heat-exchanger despite its great efficiency unprofitable. This entails that the available heat source has already been cooled to the point where no meaningful energy remains to be recovered.

6.2 Method

The main reason for selecting EBSILON®Professional as our simulation tool is because of its wide component library, and because it is designed to be used for power plant and energy system simulations. With the steady-state modeling that EBSILON®Professional uses their simulation model align well with our research objectives, focusing on energy balance, economic performance and of design simulations. With EBSILON®Professional and the excel add-in it enabled us to do multiple simulations from an array of data points, calculating the annual power savings by simulating the entire year.

The use of different humidity levels stems from two reason, the first is that Sweden has a varying climate, and such the humidity level changes drastically with the seasons. The second reason is that no information or data was provided by Mälarenergi as to what the humidity the air had going into the humidifier, thus calculations on different humidities seemed fitting. As to why 30%, 50% and 70% were chosen was because Mälarenergi specified that the used air for combustion is a mixture of outdoor and indoor air. This means that even tough, for example the outdoor air would be -7 °C during winter and have a humidity of 90%, the in indoor air that is heated would have a lower relative humidity. Mixing indoor and outdoor air would then imply a reduction in relative humidity (nomadseason, 2025). In conclusion, testing three evenly spaced humidity levels was a resolvable solution, looking at the impact of fluctuating humidity between seasons and the impact that would have on the simulations.

As for different simulation tools that could have been used is Matlab or any other programming language. The purpose of them would be to modeling the heat-exchangers size and geometry,

making sure that the heat-exchangers would fit across all the efficiencies, within the limited space restrictions that were given. One program in particular that could have been used is Simulink, which is a simulation tool in MATLAB that lets you program and design heat-exchangers, letting you change both the geometry of the outside but also the inside of the heat-exchanger. With the Simulink simulation tool you can change the air and water side dynamics, the tube diameter, pressure losses, heat transfer, fin surface area and much more creating a more accurate model. Since the resulting payback period is large across all efficiencies the need for more accurate heat-exchanger modeling were not required.

7 Conclusions

The purpose going into this thesis was to do a techno-economical assessment, adding an additional heat-exchanger to block 7 at Mälarenergi. Our goal was to observe and simulate how the heat-exchanger would act and how it would impact the system. This was achieved by using the simulation tool Ebsilon®Professional, modeling the system and simulating year-round operational data provided by Mälarenergi. The study compared the current configuration with heat-exchanger with three different efficiency levels and analyzed the results.

The answer to the research question are as following:

1. The potential temperature increase from installing the new heat-exchanger would be between 1.4-1.8 ℃ averaged over a year. The humidity have a very small linear increase for the temperature, while the efficiency have a more modest increase at around 0.4 ℃.

2. The yearly heat recovery and power saving across all the different scenarios were as following: recovered energy ranged from 359–366 MWh at 70% efficiency, 415–423 MWh at 80% efficiency, and 474–482 MWh at 90% efficiency.

3. The payback period for this system is between 75 years and 56 years depending on how favorable the humidity and efficiency are. If this heat-exchanger were to be installed when block 7 was constructed and was to be used with electricity certificates along with the flat rate of revenue received due to being a green energy source. This heat-exchanger would have had a 48-67 year payback period in 2025. In the absolute best scenario, where the capital cost would be halved and the revenue doubled the system would still have a payback period of around 20 years and when considering the electricity certificates they would have an payback period of 15 years in 2025.

Unlike the efficiency the humidity is undetermined, meaning that it is an unpredictable variable changing with both the climate and the current weather. The simulations done across three different humidity level show that despite its influence being low, it is still a modest measurable heat and temperature difference. Thus influencing the heat-exchanger efficiency decision when deciding what heat-exchanger you might want to use. The efficiency of the heat-exchangers has a much larger impact, where the difference between the efficiencies is around a 32% recovered energy increase. The lowest energy recovered is 359 MWh and the highest is 474 MWh at the same humidity level. Despite this with these potential energy savings it is still not enough in the current energy market to consider installing the heat-exchanger. This is because low-grade heat sources inherently limit performance as the amount of energy able to be recovered is insufficient.

This aligns with the payback period being unreasonable long despite doubling revenue and halving the cost, this in turn makes the project impossible to have financial gain in block 7's lifetime, unless electricity certificates price sky rocket or new financial incentives arrive.

8 Suggestions for further work

Although the single heat-exchanger system was found to be economically unfeasible, several more advanced heat recovery technologies may offer greater potential for thermal heat recovery. These technologies include Organic Rankine Cycle, Casacding Rankine cycles, Kalina Cycles. In particular, an ORC system could be investigated as a means of converting low grade heat into additional electrical power, bypassing the auxiliary steam and going immediately to power generation. This approach has the potential to reduce overall payback period which would improve the financial viability of the investment.

Along with a financial evaluation, a model could be made to observe the integration of the orc system with existing heat recovery infrastructure of block 7. This would entail how the ORC system could utilize the heat from the scrubber water or other low temperature sources to produce power.

9 Declaration of Large Language Models (LLMs) and AI-assisted Technologies in the Writing Process

The authors of this degree report acknowledges that we have utilized Microsoft Copilot GPT-5.1 for text enhancement and research assistance. Though all core research, analysis and theoretical contributions and academic conclusions are purely made by the authors of this project. We as the authors take complete responsibility for the validity, accuracy and academic integrity in this thesis.

References

Breeze, P. (2018). Chapter 3 - combined heat and power principles and technologies. In P. Breeze (Ed.), *Combined heat and power* (pp. 21–32). Academic Press. https://doi.org/10.1016/B978-0-12-812908-1.00003-1

Chen, L.-Q., & Mauro, J. C. (2024). A modern statment of the fist law of thermodynamics. *MRS Bulletin*, *49*, 1175–1185. https://doi.org/10.1557/s43577-024-00802-w

Choi, J. M., Kim, Y., Lee, M., & Kim, Y. (2010). Air side heat transfer coefficients of discrete plate finned-tube heat exchangers with large fin pitch. *Applied Thermal Engineering*, *30*(2), 174–180. https://doi.org/10.1016/j.applthermaleng.2009.08.001

Costea, M., & Feidt, M. (2022). A review regarding combined heat and power production and extensions: Thermodynamic modelling and environmental impact. *Energies*, *15*(23). https://doi.org/10.3390/en15238782

Dzierwa, P., Trojan, M., Peret, P., Taler, J., Taler, D., Kaczmarski, K., Wrobel, W., & Bator, J. (2024). Technological and economical analysis of the heat recovery system from flue gas in a thermal waste treatment plant. *Energy*, *307*, 132708. https://doi.org/10.1016/j.energy.2024.132708

EBSILON®Professional. (2026). *Component objects - general and categories*. Retrieved April 8, 2026, from https://help.ebsilon.com/EN/ComponentsGeneralCategories.html

El Maakoul, A., El Metoui, M., Ben Abdellah, A., Saadeddine, S., & Meziane, M. (2017). Numerical investigation of thermohydraulic performance of air to water double-pipe heat exchanger with helical fins. *Applied Thermal Engineering*, *127*, 127–139. https://doi.org/10.1016/j.applthermaleng.2017.08.024

Energimyndigheten. (2024). *Genomsnittliga månadspriser på elcertifikat, sek/mwh, 2003-*. Retrieved May 11, 2026, from https://pxexternal.energimyndigheten.se/pxweb/sv/Energimyndighetens_statistikdatabas/Energimyndighetens_statistikdatabas___Energiindikatorer___22._Elcertifikatsystemet/EN_IND22B.px/

Ghema, M., El Fadar, A., & Achkari, O. (2024). Capacity of waste heat recovery-based polygeneration to achieve sustainable development goals. *Science of The Total Environment*, *927*, 171983. https://doi.org/10.1016/j.scitotenv.2024.171983

Hai, T., Omar, I., El-Sharkawy, M. R., Kassim, M., Rajab, H., Said, E. A., Hussein, A. H. A., Hassan Alhaidry, W. A. A., Idan, A. H., & Alizadeh, M. (2024). Multiaspect analysis and optimization of a power and cooling cogeneration plant integrated with a multilevel waste heat recovery system. *International Journal of Low-Carbon Technologies*, *19*, 1801–1813. https://doi.org/10.1093/ijlct/ctae134

Hu, T., Li, Y., Wang, J., & Fan, Y. (2024). Thermodynamic and economic analysis of a high-efficiency system integrating orc power generation, co2 capture, and co2 storage driven by complementary lng cold energy and flue gas waste heat. *Academic Journal of Science and Technology*, *13*(1), 185–188. https://doi.org/10.54097/9gekrg40

Ibrahim, U., & Farrukh, S. (2019). Optimization of fuel in saturated steam boiler through preheating of controlled air-fuel mixture. *IEEE INTERNATIONAL CONFERENCE ON COMPUTING, MATHEMATICS AND ENGINEERING TECHNOLOGIES*. https://doi.org/10.1109/ICOMET.2019.8673398

IEA. (2025). Energy efficiency 2025. *IEA*. https://www.iea.org/reports/energy-efficiency-2025

IEA. (2026). Retrieved May 13, 2026, from https://www.iea.org/countries/sweden/energy-mix

Johansson, J., Normann, F., & Andersson, K. (2021). Techno-economic evaluation of co-removal of nox and sox species from flue gases via enhanced oxidation of no by clo2—case studies of

implementation at a pulp and paper mill, waste-to-heat plant and a cruise ship. *Energies*, *14*(24). https://doi.org/10.3390/en14248512

Karimi, A., Gimelli, A., Iossa, R., & Muccillo, M. (2024). Techno-economic simulation and sensitivity analysis of modular cogeneration with organic rankine cycle and battery energy storage system for enhanced energy performance. *Energy*, *295*, 131021. https://doi.org/10.1016/j.energy.2024.131021

Ling, J., & Chen, X. (2024). Energy and economic analysis of a new combination cascade waste heat recovery system of a waste-to-energy plant. *Energies*, *17*(20). https://doi.org/10.3390/en17205196

Liu, J., Ren, J., Zhang, Y., Huang, W., Xu, C., & Liu, L. (2024). Exergoeconomic evaluation of a cogeneration system driven by a natural gas and biomass co-firing gas turbine combined with a steam rankine cycle, organic rankine cycle, and absorption chiller. *Processes*, *12*(1). https://doi.org/10.3390/pr12010082

Maghsoudali, Y., Rastegarkoutenaei, A., Sahami, M., & Bandpy, M. G. (2022). Investigation of the effect of using the finned tubes on the performance of shell and tube heat exchanger by 3d modeling. *Journal of Energy Storage*, *56*, 106031. https://doi.org/10.1016/j.est.2022.106031

Mälarenergi. (2024). *Hållbar utveckling sedan 1861*. Retrieved March 3, 2026, from https://www.malarenergi.se/om-malarenergi/miljo-och-hallbar-utveckling/

Mälarenergi. (2026a). *Block 7 - framtidens energibehov*. Retrieved March 3, 2026, from https://www.malarenergi.se/om-malarenergi/framtidens-samhalle/vara-anlaggningar/kraftvarmeverket-vasteras/block-7/

Mälarenergi. (2026b). *Fjärrkyla - behagligt inomhusklimat*. Retrieved March 12, 2026, from https://www.malarenergi.se/foretag/varme-kyla-foretag/fjarrkyla/

Mälarenergi. (2026c). *Kraftvärmeverket västerås*. Retrieved March 3, 2026, from https://www.malarenergi.se/om-malarenergi/framtidens-samhalle/vara-anlaggningar/kraftvarmeverket-vasteras/

Marchenko, O., Solomin, S., Kozlov, A., Shamanskiy, V., & Donskoy, I. (2020). Economic efficiency assessment of using wood waste in cogeneration plants with multi-stage gasification. *Applied Sciences*, *10*(21). https://doi.org/10.3390/app10217600

Meftahpour, H., Saray, R. K., Aghaei, A. T., & Bahlouli, K. (2024). Comprehensive analysis of energy, exergy, economic, and environmental aspects in implementing the kalina cycle for waste heat recovery from a gas turbine cycle coupled with a steam generator. *Energy*, *290*, 130094. https://doi.org/10.1016/j.energy.2023.130094

Nami, H., Anvari-Moghaddam, A., Arabkoohsar, A., & Razmi, A. R. (2020). 4e analyses of a hybrid waste-driven chp–orc plant with flue gas condensation. *Sustainability*, *12*(22). https://doi.org/10.3390/su12229449

Nikafshan Rad, H., Ghasemi, A., & Marefati, M. (2024). Cost and environmental analysis and optimization of a new and green three-level waste heat recovery-based cogeneration cycle: A comparative study. *Heliyon*, *10*(7), e29087. https://doi.org/10.1016/j.heliyon.2024.e29087

nomadseason. (2025). *Climate of sweden*. Retrieved May 11, 2026, from https://nomadseason.com/climate/sweden/index.html

Nutakki, T. U. K., Agrawal, M. K., Tahir Chauhdary, S., Ahmad, S. F., Ayadi, M., Hedi, E., Muhammad, T., & Xiao, F. (2024). Thermo-economic-environmental analysis of a sustainable heat integration design for biomass-fueled power plant using integration of cchp and sweater desalination application. *Desalination*, *577*, 117404. https://doi.org/10.1016/j.desal.2024.117404

Pariasamy, S. G., Venkiteswaran, V. K., Kumar, J., & Awad, M. M. (2022). Industrial chp with steam systems: A review of recent case studies, trends and relevance to malaysian industry. *Energies*, *15*(20). https://doi.org/10.3390/en15207491

Rzetelska, D., & Combrinck, M. (2022). Fuel poverty and low carbon emissions: A comparative study of the feasibility of the hybrid renewable energy systems incorporating combined heat and power technology. *Frontiers in Energy*. https://doi.org/10.1007/s11708-021-0748-x

Saari, J., Kaikko, J., Sermyagina, E., Hamaguchi, M., Cardoso, M., Vakkilainen, E., & Haider, M. (2023). Recovery boiler back-end heat recovery. *Heat Recovery*, *22(3)*. https://doi.org/10.32964/TJ22.3.174

SCB. (2025). *District heating (gwh)*. Retrieved May 13, 2026, from https://www.scb.se/en/finding-statistics/statistics-by-subject-area/energy/energy-supply-and-use/annual-energy-statistics-electricity-gas-and-district-heating/pong/tables-and-graphs/district-heating-gwh/

Sheikholeslami, M., & Ganji, D. (2016). Heat transfer enhancement in an air to water heat exchanger with discontinuous helical turbulators; experimental and numerical studies. *Energy*, *116*, 341–352. https://doi.org/10.1016/j.energy.2016.09.120

Srimuang, W., & Amatachaya, P. (2012). A review of the applications of heat pipe heat exchangers for heat recovery. *Renewable and Sustainable Energy Reviews*, *16*(6), 4303–4315. https://doi.org/10.1016/j.rser.2012.03.030

Swedish-Institute. (2025). *Swedes use a lot of energy, yet emissions are low. the key? renewable energy*. Retrieved May 13, 2026, from https://sweden.se/climate/sustainability/energy-use-in-sweden

United-Nations. (2015). *7. d paris agreement*. Retrieved March 3, 2026, from https://treaties.un.org/pages/ViewDetails.aspx?src=TREATY&mtdsg_no=XXVII-7-d&chapter=27&clang=_en

Wang, F., Li, J., Zhang, G., Liu, H., Zhang, X., Liu, X., Sun, B., Xu, J., & Guo, W. (2026). Performance optimization of aluminum slit fin-and-tube heat exchangers based on orthogonal experiments and response surface modeling. *International Journal of Thermal Sciences*, *223*, 110659. https://doi.org/10.1016/j.ijthermalsci.2026.110659

Wang, J., Liu, H., Wu, L., Yu, L., Liu, P., & Liu, Z. (2025). Heat transfer enhancement of tube bundle with symmetrically inclined annular fins for waste heat recovery. *Energies*, *18*(18). https://doi.org/10.3390/en18184964

Wang, Y., Xu, H., Li, Y., Lin, N., & Xu, P. (2023). Technical design of an innovative biomass / gasification-driven power plant with heat recovery hybrid system: Co2 emission comparison between the designed plant and fossil fuel-powered plants. *Chemosphere*, *340*, 139818. https://doi.org/10.1016/j.chemosphere.2023.139818

Wardach-Swiecicka, I., Polesek-Karczewska, S., & Dabrowski, A. (2025). Predicting the performance of a helically coiled heat exchanger for heat recovery from a waste biomass incineration system. *Sustainability*, *17*(2). https://doi.org/10.3390/su17020759

Wu, Y., & Xu, Y. (2024). A comprehensive evaluation of an environmentally friendly multi-heat recovery system using biogas energy source from thermodynamic, economic, and environmental perspectives. *Journal of Environmental Chemical Engineering*, *12*(3), 112783. https://doi.org/10.1016/j.jece.2024.112783

Xu, G., Huang, S., Yang, Y., Wu, Y., Zhang, K., & Xu, C. (2013). Techno-economic analysis and optimization of the heat recovery of utility boiler flue gas. *Applied Energy*, *112*, 907–917. https://doi.org/10.1016/j.apenergy.2013.04.048

Yang, J., Li, C., Teng, J., Zhang, Y., Wang, Y., Hou, Y., Xia, L., Sun, X., Wang, L., & Xiang, S. (2024). Multi-objective optimization and life cycle assessment of mass-integrated combined

heat and power system. *Science of The Total Environment*, *951*, 175811. https://doi.org/10.1016/j.scitotenv.2024.175811

Zhang, Q., Niu, Y., Yang, X., Sun, D., Xiao, X., Shen, Q., & Wang, G. (2020). Experimental study of flue gas condensing heat recovery synergized with low nox emission system. *Applied Energy*, *269*, 115091. https://doi.org/10.1016/j.apenergy.2020.115091

Zhu, Y., Zhang, C., Yan, M., Liu, Z., Li, W., Li, H., & Wang, Y. (2024). Biomass-fired combined heat, cool and power system incorporating organic rankine cycle and single-effect lithium bromide absorption refrigeration integrated with co2 capture: Thermo-economic analysis. *Energy*, *304*, 132096. https://doi.org/10.1016/j.energy.2024.132096