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Analysis on Methods and the Influence of Different System Data When Calculating Primary Energy Factors for Heat from District Heating Systems

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MASTEROPPGAVE

for

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Analyse av metoder og betydningen av ulike, relevante systemdata ved beregning av primærenergifaktorer for fjernvarme

Analysis on methods and the influence of different system data when calculating primary energy factors for heat from district heating systems

Bakgrunn og målsetting

Den nye, omarbeidete versjonen av "Directive 2002/91/EC on the energy performance of buildings"(EPBD) - Recast 2010/31/EU) sier at primærenergiforbruk, beregnet ved primærenergifaktorer, skal være obligatorisk indikator for angivelse av energiytelsen på energisertifikatene i hele EU-området inklusive EØS. For fjernvarmesystemer er følgende utsagn i Direktivet relevant; *"The energy performance of a building shall be expressed in a transparent manner and shall include an energy performance indicator and a numeric indicator of primary energy use, based on primary energy factors per energy carrier, which may be based on national or regional annual weighted averages or a specific value for on-site production."* For fjernvarme betyr dette at det er behov for å beregne spesifikke verdier for primærenergifaktoren for de respektive fjernvarmesystemer med den aktuelle systemkonstruksjon og de aktuelle typer brensel.

Hovedmålet med denne oppgaven er å gi en oversikt over relevante metoder og betydningen av ulike, relevante systemdata ved beregning av primærenergifaktorer relatert til fjernvarme. Hovedfokus rettes mot mindre systemer hvor CHP (Combined Heat and Pover) utgjør en sentral del av varmeproduksjonen til fjernvarmesystemet.

Oppgaven bearbeides ut fra følgende punkter

1. Klarlegg størrelsesorden for maksimal varmeeffekt og årlig varmeproduksjon for de mest aktuelle typer CHP-teknologier ved fjernvarmesystemer i Norge.
2. Klarlegg de typiske egenskapene ved de aktuelle CHP-teknologiene nevnt under pkt 1), og hvordan forholdet mellom varmeproduksjon og elektrisitetsproduksjon i prinsippet endres ved ulike, relevante systemdata.
3. Velg et relevant programverktøy for beregning av typiske endringer i systemytelsen ved predefinerte CHP-systemer og klarlegg konsekvensene ved valg av aktuelle, relevante data for fjernvarmesystemer i praksis.
4. Redegjør for aktuelle metoder for beregning av primærenergifaktorer for fjernvarmesystemer ved aktuelle typer brensel og med hovedvekt på systemer hvor CHP-systemer representerer en vesentlig del av varmeproduksjonen.

5. Bruk resultatene fra pkt 3) og 4) til å beregne typiske primærenergifaktorer for aktuelle fjernvarmesystemer med hovedvekt på systemer hvor CHP-systemer representerer en vesentlig del av varmeproduksjonen.

" - "

Senest 14 dager etter utlevering av oppgaven skal kandidaten levere/sende instituttet en detaljert fremdrift- og eventuelt forsøksplan for oppgaven til evaluering og eventuelt diskusjon med faglig ansvarlig/veiledere. Detaljer ved eventuell utførelse av dataprogrammer skal avtales nærmere i samråd med faglig ansvarlig.

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NTNU, Institutt for energi- og prosessteknikk, 1. august 2011

Olav Bolland
Instituttleder

Rolf Ulseth
Faglig ansvarlig/veileder

Medveileder(e)

Preface

This master's thesis was carried out from the beginning of August to the middle of December 2011 at the Department of Energy and Process Engineering at the Norwegian University of Science and Technology. It represents the final project and conclusion of my master degree.

It has been a special semester, with ups and downs, frustrations and breakthroughs.

There are many people I wish to thank for their advice, help and support.

I would like to thank my supervisor, Rolf Ulseth, for an interesting problem statement and good advice along the way.

A special thanks go to Monica Berner at Sintef, who has invested many hours in discussing project related challenges with me. In addition her support from reading drafts of the report and giving comments throughout the process has been crucial for the result.

Paul Andreas Ystad helped me overcome modelling challenges in a critical phase of the project, and always had time for discussing and giving advice.

Lars Olof Nord helped me interpreting some curious results.

Credit must also be attributed to all the people that provided information during the mapping phase of the project. Some of them are Roald Kirkengen from Hafslund Miljøenergi, Åmund Utne from Statkraft and Audun Amundsen from Mosseporten Miljøenergi, who all gave detailed information which helped me to better understand the real challenges of cogeneration connected to district heating networks.

Finally, this thesis would not have been possible to complete without the support of my boyfriend, who has been patient, helpful and understanding throughout the process.

Magnhild Kallhovd

Magnhild Kallhovd,

Trondheim 16.12.2011

ABSTRACT

A steady growing global demand for energy and rising greenhouse gas emissions has resulted in several initiatives from the European Union with the purpose of increasing energy efficiency. A part of this strategy is the introduction of energy performance certificates for buildings, containing a numerical primary energy indicator. Another instrument is to encourage an increased use of cogeneration. As a member of the European Economic Area agreement, these events also affect Norway.

The main aim of the project was to investigate how various relevant parameters influence the primary energy factor of district heating when a combined heat and power (CHP) plant is the heat producing unit. The study was to be based on Norwegian conditions.

To select relevant technologies, a mapping of existing and planned CHP facilities connected to district heating (DH) networks in Norway was carried out. The findings were that at present, there are nine steam cycle CHP plants connected to DH networks that are based on waste incineration, one steam cycle that is based on demolition wood and one reciprocating engine that is running on biogas. The installed electric capacity ranged from 0,3 MW to 22,8 MW and the annual district heating production from 1,5 GWh to 196 GWh. Based on this, it was decided to study steam cycle CHP plants further. Three different sizes were chosen: 2 MW_{el}, 10 MW_{el} and 25 MW_{el}.

In addition, the situation in Europe was looked into. Here, steam cycle and combined cycle were found to be the two most dominant CHP technologies. To have a different technology to compare with, a combined cycle with 22,7MW_{el} capacity was also included in the study.

By running plant simulations, the effects of part load operation, various district heating supply and return temperatures and different fuel types were quantified.

STEAM Pro was utilised to design the steam cycle models, while GT Pro was used to design the combined cycle models. STEAM Pro was also used to perform design simulations for different temperature levels in the DH network and to study the effect of different types of fuels. To be able to investigate the part load performance of the plants based on a given district heating demand, the models from STEAM Pro and GT Pro was imported into Thermoflex and modified.

Reducing the DH supply temperature from 120 to 80 °C and the return temperature from 80 to 35 °C in the 10 MW steam cycle plant increased the power efficiency by 25% and the power to heat ratio by 33%, but the total efficiency was only slightly increased. Variation of fuel, on the other hand, influenced the power efficiency and the total efficiency almost equally, and the power to heat ratio was hence left relatively unaltered.

The results from the simulations at the defined full load conditions showed that power efficiency was more than twice as high for the combined cycle than for the steam cycle plants, and the power to heat ratio was almost four times higher for the CC plant. The total efficiency was approximately 10 % lower for the combined cycle than for the steam cycles.

Performance also varied between the different sizes of steam cycles, and both boiler type and turbine size influenced power efficiencies and power to heat ratios. In contrast, the total efficiencies were close to equal.

Part load had a great influence on power efficiency and power to heat ratio for all technology types. Especially at very low load levels, the power efficiency was considerably reduced. The combined cycle experienced a total fall in power efficiency of 40%, while the reduction varied from 60% to only 29% for the steam cycle plants. The part load total efficiency was only slightly reduced for all plants.

Based on the part load simulations, annual efficiencies and power to heat ratios were calculated for different annual load distributions. The annual power to heat ratio and power efficiency was clearly influenced by changes in the annual load distribution pattern, while the effect was less notable for the annual total efficiencies.

To calculate the primary energy factors, the total efficiency and power to heat ratio results from the CHP plant simulations were implemented in an excel tool developed by [16]. Some other modifications were also performed.

The district heating primary energy factors (PEF_{DH}) for the defined base case varied from 0,85 for the Combined Cycle* alternative to 1,4 for the 2 MW steam cycle plant. The base case was defined to have medium energy density(8 MWh/m). This was later found to not represent the actual Norwegian conditions, where the average energy density is closer to 4 MWh/m. When this energy density was used, the PEF_{DH} for the 10 MW steam cycle plant increased 9,4%, from 1,38 to 1,51. This value is still considerably lower than the primary energy factor for the average electricity production in the Nordic countries, which is 2,16.

It was found that the combined heat and power plant parameters had a significant influence on the primary energy factors. The power to heat ratio was particularly important when the power bonus method was utilised. One main conclusion is therefore that it is important that the performance indicators that are used for the CHP plant are realistic, and takes into account technology type, part load performance and what load duration curve the plant is subject to.

In most of the cases studied, the fuel handling process and the use of additives contributed most to the primary energy losses related to the PEF_{DH}, while the sum of primary energy losses was dominated by the losses occurring in the CHP plant and the fuel handling. Nevertheless, what process and parameters that could potentially improve the PEF_{DH} most depended on technology and choice of allocation method. In all cases studied, pump work related to circulating the DH water and energy consumption related to ash transport, construction and dismantling of the CHP plant and DH pipes were negligible or close to negligible. Heat loss became a considerably more dominant primary energy loss contributor when a low energy density was assumed.

In the end, the calculation of primary energy factors involves many choices that influence the results. It is therefore important that the calculation method becomes more standardised. As it is today, some processes are optional, for instance the use of additives. In this study, the use of additives had a non-negligible influence on the results. Furthermore, the CHP simulation results underlined the importance of taking type of CHP technology and operational conditions into account when calculating primary energy factors for this kind of systems.

According to NS-EN 15316-4-5, the power bonus method is the allocation method that should be utilised when calculating primary energy factors for district heating. This makes the district heating primary energy factors extremely dependent on power to heat ratio and the choice of PEF for avoided electricity. If the amount of avoided electricity production in fact is smaller than the full amount of CHP production or if the PEF of the avoided electricity is lower than what is assumed, this might lead to a severe underestimation of the PEF_{DH}.

The ultimate goal with the use of primary energy is to encourage more efficient energy use. It is therefore important that the issues mentioned in the two paragraphs above are further studied and discussed as a part of exploring how a standard method should be designed to face this challenge.

SAMMENDRAG

Et stadig økende energibehov samt økende utslipp av klimagasser har ført til at EU de siste årene har lansert flere tiltak for å oppnå en mer effektiv energibruk. Et av virkemidlene er innføringen av en obligatorisk energimerkeordning for bygg. Ifølge denne skal alle bygg energimerkes, noe som innebærer at byggets forbruk av primærenergi skal tallfestes. Et annet initiativ er tiltak som skal føre til økt bruk av kogenerering. Som en del av EØS har disse hendelsene også betydning for Norge.

Hovedmålet med dette prosjektet var å undersøke hvordan forskjellige parametre påvirker primærenergifaktoren til fjernvarme når et kraftvarmeverk utgjør den varmeproduserende enheten. Studien skulle være basert på Norske forhold.

For å velge ut relevante teknologier ble en kartlegging av eksisterende og planlagte kogenereringsanlegg med fjernvarmetilknytning i Norge utført. Funnene viste at per dags dato finnes det ni dampturbinanlegg basert på avfallsforbrenning, ett dampturbinanlegg basert på returtre og en stempelmotor som går på deponigass. Installert elektrisk effekt varierte fra 0,3 MW til 22,8 MW, og årlig fjernvarmeproduksjon fra 1,5 GWh til 196 GWh. Basert på disse opplysningene ble det valgt å studere dampturbinanlegg nærmere. Tre forskjellige størrelser ble valgt: 2 MW_{el}, 10 MW_{el} og 25 MW_{el}.

I tillegg ble situasjonen i Europa undersøkt. Der er dampturbinanlegg og kombinerte kraftverk de mest dominerende teknologiene. For å kunne sammenligne dampturbinanleggene med en annen type teknologi, ble også et kombinert kraftverk inkludert i studien.

Ved å utføre simuleringer av anleggene ble virkningen av dellast, ulike tur- og returtemperaturer i fjernvarmenettet og forskjellige typer brensel tallfestet.

STEAM Pro ble brukt til å designe modellene av dampturbinanleggene, mens GT Pro ble brukt til å designe det kombinerte kraftverket. STEAM Pro ble også brukt til å utføre design simuleringer for forskjellige temperaturnivåer i fjernvarmenettet samt ulike brenseltyper. For å kunne utføre dellastsimuleringer hvor fjernvarmebehovet var kontrollvariabelen, måtte modellene fra STEAM Pro og GT Pro modifiseres i Thermoflex.

Ved å redusere turtemperaturen fra 120 til 80 °C og returtemperaturen fra 80 til 35 °C, oppnåde man en økning i elvirkningsgrad på 25% og kraft-varme forholdet på 33%. Totalvirkningsgraden ble bare marginalt forbedret. Endring av brensel påvirket på den andre siden elvirkningsgraden og totalvirkningsgraden noenlunde likt, og kraft-varmeforholdet forble dermed relativt uendret.

Resultatene fra simuleringene ved fullastforhold viste at elvirkningsgraden var over dobbelt så høy for det kombinerte kraftverket enn for dampturbinanleggene mens kraft-varmeforholdet var nesten fire ganger så høyt. Totalvirkningsgraden var circa 10% lavere for det kombinerte kraftverket enn for dampturbinanleggene.

Ytelsen varierte også mellom de forskjellige dampturbinanleggene, og det viste seg at både kjeltype og turbinstørrelse påvirket elvirkningsgraden og kraft-varmeforholdet. Totalvirkningsgraden viste seg derimot å være tilnærmelsesvis lik for alle de tre størrelsene.

Dellast-kjøring påvirket både elvirkningsgrad og kraft-varmeforhold kraftig for alle anleggene. Effekten var spesielt merkbar på det laveste dellast-nivået. Totalvirkningsgraden falt 40% for det kombinerte kraftverket mellom fullast og laveste dellastnivå, mens dampturbinanleggene opplevde fall på mellom 60%(minste anlegg) og 29%(største anlegg). Totalvirkningsgraden ble generelt bare minimalt påvirket.

Basert på dellastsimuleringene ble årsvirkningsgrader og årlige kraft-varmeforhold beregnet for ulike lastkurver. Det årlige kraft-varmeforholdet og elvirkningsgraden ble klart påvirket av endringer i lastkurven, mens innvirkningen var mindre merkbar for de årlige totalvirkningsgradene.

For å beregne primærenergifaktorene ble totalvirkningsgradene og kraft-varmeforholdene fra simuleringene av kogenereringsanleggene implementert i excel beregningsverktøyet som ble utviklet av [16]. Noen andre modifikasjoner ble også utført.

Primærenergifaktorene for fjernvarme (PEF_{DH}) i basisalternativet varierte fra 0,85 for det kombinerte kraftverket (CC alternativet) til 1,4 for dampturbinanlegget på 2 MW_{el}. Basisalternativet var definert til å ha medium energitetthet (8 MWh/m). Det ble senere i arbeidet oppdaget denne verdien sannsynligvis er alt for høy, ettersom gjennomsnittlig energitetthet i Norge i 1998 var på 4 MWh/m[1]. Da lav energitetthet (3 MWh) ble benyttet i beregningene i stedet for medium, økte PEF_{DH} for dampturbinanlegget på 10 MW_{el} med 9,4 %, fra 1,38 til 1,51. Denne verdien er imidlertid fortsatt betydelig lavere enn primærenergifaktoren for den nordiske elektrisitetsmiksen, som ligger på 2,16.

Ytelsesparametrene til kogenereringsanleggene hadde betydelig påvirkning på primærenergifaktorene. Kraft-varmeforholdet var spesielt avgjørende når power bonus metoden ble brukt. En hovedkonklusjon er derfor at det er viktig at ytelsesparametrene som blir brukt for å beskrive kogenereringsanlegget er realistiske, og tar i betraktning type teknologi, dellast-egenskaper og hvilken lastkurve anlegget følger.

I de aller fleste alternativene som ble undersøkt var brenselproduksjonskjeden og produksjon av forbrenningstilsetningsstoffer de to prosessene som bidro mest til primærenergitapene som var allokert til fjernvarmen. Ved lav energitetthet i basisalternativet sto varmetap i fjernvarmerørene for det nest største varmetapet. For de totale primærenergitapene var det enten tap i selve kogenereringsanlegget eller brenselproduksjonskjeden som påvirket mest. Det var imidlertid store forskjeller fra alternativ til alternativ vedrørende hvilken prosess og hvilke parametre som potensielt kunne bidratt mest til å forbedre primærenergieffektiviteten i de enkelte tilfellene. Dette var avhengig av teknologi og allokeringsmetode.

Imidlertid viste resultatene at primærenergitapene relatert til transport av aske, produksjon og legging av fjernvarmerør og pumpearbeid for å sirkulere fjernvarmevannet var neglisjerbare i alle beregningsalternativene. Konstruksjon og riving av kogenereringsanlegget var også neglisjerbart i de fleste tilfellene.

Beregning av primærenergifaktorer involverer svært mange valg som påvirker resultatene. Det er derfor viktig at beregningsmetodikken blir mer standardisert. I dag er noen prosesser valgfrie å inkludere, for eksempel produksjon av forbrenningstilsetningsstoffer. I denne studien hadde denne prosessen en betydelig påvirkning på PEF_{DH}. Resultatene viser også at det er viktig å ta i betraktning hvilken kogenereringsteknologi som brukes samt driftsforhold ved anlegget når man beregner primærenergifaktorer for denne typen systemer.

Ifølge NS-EN 15316-4-5 skal power bonus metoden brukes som allokeringsmetode ved beregning av PEF_{DH}. Dette gjør primærenergifaktorene svært sensitive for endringer i kraft-varmeforholdet og valg av PEF for elektrisitet. Dersom det viser seg at elektrisitetsproduksjonen andre steder faktisk ikke synker like mye som mengden elektrisitet produsert ved kogenereringsanlegget og/eller den antatte PEF verdien for elektrisitet er for høy, risikerer man en betydelig underestimering av PEF_{DH}.

Hovedmålet med bruk av primærenergifaktorer er økt total energieffektivitet. Forholdene nevnt i de to overstående avsnittene bør derfor undersøkes nærmere i videre studier slik at bruk av primærenergifaktorer bidrar til dette i størst mulig grad.

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NOMENCLATURE

Abbreviations:

CC	Combined cycle
CFB	Circulation fluidised bed
CHP	Combined heat and power
CTG	Combustion turbine generators
DH	District heating
DHC	District heating and cooling
DMFC	Direct methanol
FWH	Feed water heater
HRSG	Heat recovery steam generator
IC	Internal combustion
IEA	The International Energy Agency
IEA	The International Energy Agency
LHV	Lower heating value
MCFC	Molten carbonate
PAFC	Phosphoric acid
PEF	Primary energy factor
PHR	Power to heat ratio
PES	Primary energy savings
RIC	Reciprocating internal combustion
SO	Solid oxide
SOFC	Solid oxide fuel cell
ST	Steam turbine

Symbols:

T	[°C] / [K]	Temperature
E	[J] / [W]	Energy
$\mathbb{E}$	[J] / [W]	Exergy
W	[J] / [W]	Work
Q	[J] / [W]	Heat
U	[J] / [W]	Internal energy
V	[m/s] or [m^3]	Velocity or Volume
g	[m/s^2]	Gravity constant
Z	[m]	Elevation
h	[kJ/kg]	Enthalpy
s	[J/ K]	Entropy
σ	[J/ K]	Entropy production
η	[%]	Efficiency
p	[Pa]	Pressure
α	[-]	Power to heat ratio
f	[-]	Primary energy factor

Subscripts:

DH / dh	District heating
CV	Control Volume
B	Boundary
H	Hot heat reservoir
C	Cold heat reservoir
A	Ambient
Q	Heat
Elec	Electricity
El	Electricity
I	Inlet
E	Exit
Tot	Total
Chp	Combined heat and power
P	Primary energy
Del	Delivered (to user)

1 INTRODUCTION

A steady growing world energy demand and the emerging climate crisis constitute the backdrop for the topic of this project. As a response to the two above-mentioned challenges, the European Union has declared that there is a strong need for improving the energy efficiency of energy use, but also of the energy production systems[2]. Norway is connected to the continent both physically via power transmission cables, and politically through the European Economic Area agreement[1]. The energy system and energy policy on the continent thus influence Norway directly.

The task of making energy supply and consumption more efficient can be attacked from many angles. One is to choose a part process and study this in detail, trying to make improvements. This is of course central, as a chain of processes will never be more efficient than the least efficient one.

It is, however, important to not lose the overall picture. To assess the impact of a certain amount of energy consumption, it is vital to know how this energy was produced and supplied to the consumer, and what losses that occurred along the way. This analysis gives us the answer to the following questions: What is the efficiency of the whole energy supply chain, and where are the main potentials for improvements?

The background section gives further information on topics that are relevant for the formulation of the problem statement, which is presented in section 1.2. General assumptions and delimitations are outlined in section 1.3, while section 1.4 gives an overview of the structure of the report.

1.1 BACKGROUND

1.1.1 PRIMARY ENERGY FACTORS – A SHORT DEFINITION

Primary energy is defined as energy from renewable and non-renewable sources that have not been subject to any conversion or transformation process[3].

The term delivered energy, on the other hand, refers to the amount of energy that is actually delivered to the user. This is the number that occurs on people's energy bills, and is therefore the term that is easiest to relate to for most energy consumers.

Between the primary energy state and the delivered energy state lies the energy supply chain. Since energy losses occur along the whole chain from energy source to energy user, the amount of delivered energy does not necessarily reflect the amount of primary energy needed to supply the demand. This depends on the efficiency of the energy supply chain, which is in fact neglected when only the amount of delivered energy is considered.

The primary energy factor tells us how much primary energy that is needed to supply one unit of delivered energy. The main aim of using a primary energy factor is to look at the energy supply system in a more holistic way, and thus obtain a more efficient use of the world's energy resources.

[1] EØS avtalen

1.1.2 ENERGY PRODUCTION AND CONSUMPTION

According to the International Energy Agency (IEA), the world primary energy demand will increase by 33% from 2010 to 2035, with India and China accounting for more than 50% of the growth. The OECD countries are expected to have a far less pronounced growth, but still an increase.[4]

As of today, the European Union imports 50% of their energy requirement, and if the current trend persists this figure will increase to 70% in 2030[2]. Norway, on the other hand, has with its oil and gas production far more energy than can possibly be utilised within its borders.

When it comes to electricity, the situation is different. Since Norway's electricity production system is hydro power based, the production varies from year to year with the amount of precipitation. In years with low production, electricity needs to be imported to satisfy the domestic electricity demand. This is done by the mechanisms in the Nordic electricity market and is organised by Nordpool. Norway is connected to Russia, Sweden, Denmark, the Netherlands and Finland through several power transmission cables, and from 1993 to 2009, Norway was a net importer 7 out of 17 years. [5] The origin of the imported electricity in 2009 is shown in Figure 1.

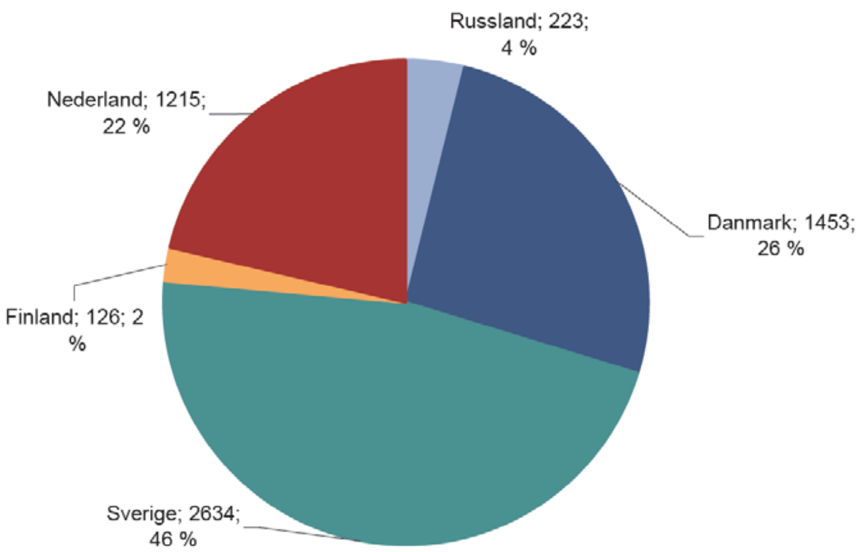


FIGURE 1: ELECTRICITY IMPORT TO NORWAY 2009. GWH AND PERCENTAGE. [5]

Another aspect that influences the Norwegian power system is problems with bottlenecks in the central transmission grid. This has resulted in price differences between the different price areas, and in some extreme situations the energy supply security has been reduced in some parts of the country.[6]

To reduce the electricity consumption and peak capacity demand, district heating is a part of Norway's energy strategy. One example of implementation of this strategy is Enova, which provides support for district heating projects. In 2011, 176 MNOK were given in support to projects that will result in 223 GWh of renewable heat production[7]. The development of district heating in Norway can be seen in Figure 2. The increase from 2009 to 2010 is considerable.

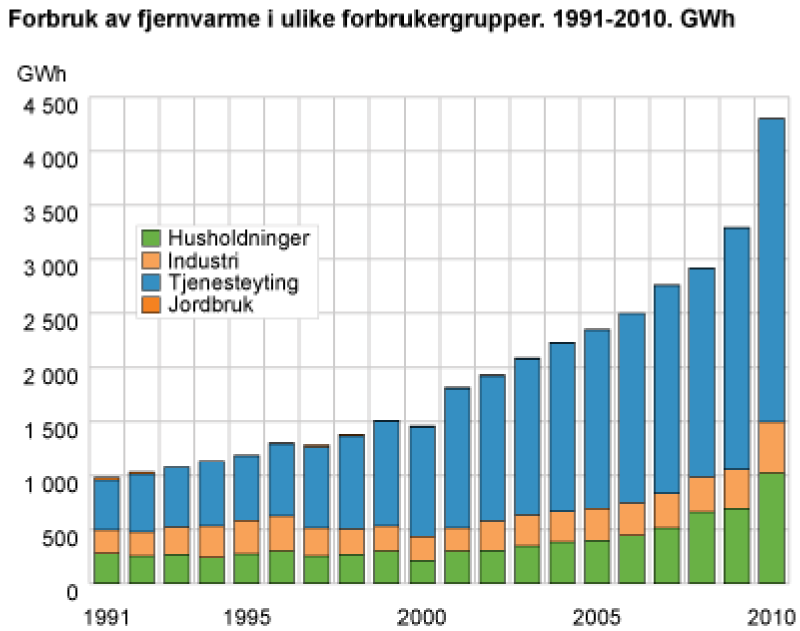


FIGURE 2: DEVELOPMENT OF DISTRICT HEATING IN DIFFERENT SECTORS. (DWELLINGS, INDUSTRY, SERVICE SECTOR AND AGRICULTURE)[8]

1.1.3 The Principle of Cogeneration

Cogeneration can be defined as the simultaneous production of several energy products from a single fuel source [9]. Normally the energy products in question are heat and electricity, but mechanical work and/or cold are also possible alternatives.

Figure 3 shows the principle in a simplified way: At the right, a heat demand of 160 units and a power demand of 100 units are supplied by two separate processes. The efficiency of the heat process is 80% and the electrical efficiency of the power station is 37,7%. This results in a total efficiency for the supplied heat and power of 55,9%.

On the left, the heat and power demand is supplied by a combined heat and power (CHP) plant. This gives a total efficiency of 80%.

This shows how combined heat and power plants might contribute to improving energy efficiency in situations where the alternative production is represented by conventional thermal power plants.

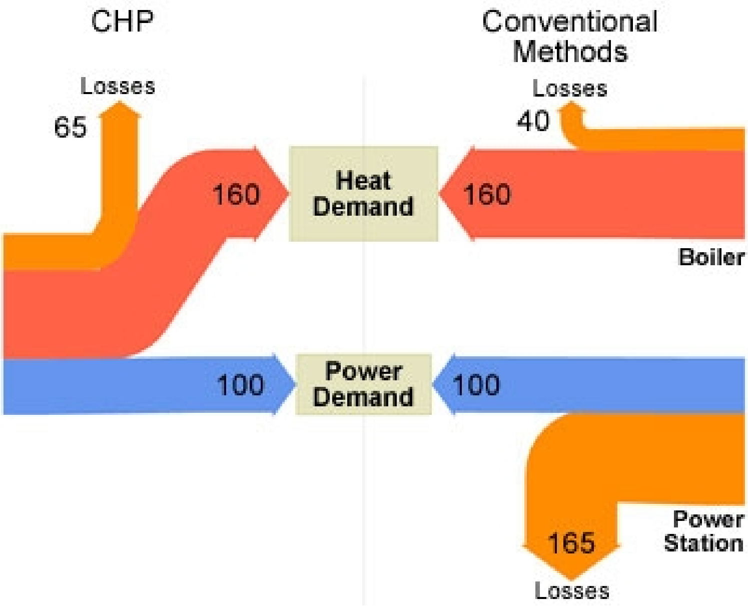


FIGURE 3: COGENERATION VS CONVENTIONAL PRODUCTION. REPRINTED FROM [10]

1.1.4 RELEVANT EU DIRECTIVES

The European Union has issued several directives that deal with energy use and energy production, of which the most relevant ones in this context are the Directive on the Energy Performance of Buildings, the Renewable Energy Directive and the Cogeneration Directive. They are all part of a strategy that involves reducing the Union's energy dependency and CO_2 emissions. This will both increase the energy security and allow the Union to comply with the Kyoto Protocol. [2, 3, 11]

To concretise the goals to be achieved within energy efficiency, greenhouse gas emissions and renewable energy, in 2007 the EU member states agreed on what is known as the 20-20-20 targets. These are:

- *"A reduction in EU greenhouse gas emissions of at least 20% below 1990 levels*
- *20% of EU energy consumption to come from renewable resources*
- *A 20% reduction in primary energy use compared with projected levels, to be achieved by improving energy efficiency."[12]*

THE DIRECTIVE ON THE ENERGY PERFORMANCE OF BUILDINGS

Buildings account for approximately 40 % of the energy consumption within the European Union[3]. The sector thus plays an important role if the Union is going to reach the 20-20-20 goals. In 2002, the European Union passed the first version of the directive on the Energy Performance of Buildings, and this was also implemented in Norway. In 2009 a recast was published, and this will take effect from the 12th of February 2012. [3] This description will therefore be based on the recast version.

The directive contains means to reduce the energy consumption in the building sector and increasing the use of renewable energy sources. The main actions are:

- All member states shall develop minimum requirements for the energy performance of buildings, which shall apply to all new buildings and to comprehensive rehabilitation projects. "The energy performance of a building" is in the directive defined as "the calculated or measured amount of energy that is needed to meet the energy demand associated with a typical use of the building. " The minimum requirements shall take the local climate into consideration, and be calculated according to standardised methods that are in accordance with ANNEX l in the Directive. In Norway, calculation methods that comply with the 2002 version of the directive are published in the standard NS 3031:2007.
- The member states shall assure that all new buildings are nearly zero-energy buildings within 31 December 2020. The authorities should lead by example, and is therefore obliged to build all new buildings as nearly zero-energy buildings from 31 December 2018. A "nearly zero-energy building" is defined as a building that has a very high energy performance and a nearly zero energy requirement that, to a very significant extent, is covered by energy from renewable sources.
- To make it easier for the consumers to orient themselves in the property market, the member states are obliged to implement an energy certificate. According to the recast directive, this should at least include one energy performance indicator and one numeric indicator of primary energy use. This last indicator should be based on primary energy factors per energy carrier. The primary energy factors may be based on national or regional weighted averages or a specific value for onsite production.

The energy certificate system that was implemented in Norway in July 2010 does not include a primary energy numeric indicator[13]. Thus, it does not comply with the recast of the directive.

THE RENEWABLE ENERGY DIRECTIVE

To achieve the renewable energy target it is necessary that all the member states contribute according to their premises. The renewable energy directive, that was passed the 23rd of April 2009, contains a matrix where all the countries have been assigned a renewable energy fraction goal. The fraction is calculated according to a scheme where the current renewable fraction and GDP are the most important parameters.[11]

Norway is also going to implement the renewable energy directive. If the renewable fraction were to be calculated according to the standard rules, this would have led to a renewable target of 73,6%, a considerable increase from 58,2% in 2005[14].

This was, however, regarded as too high by the Norwegian authorities. The 20th of July 2011 the Oil and Energy Department (OED) announced that Norway had managed to negotiate a lower target with the European Commisson. The renewable energy target for 2020 was set to 67,8%, 9,5 percent points higher than in 2005.

THE COGENERATION DIRECTIVE

The cogeneration directive contains steps for constructing a market where guarantee of origin certificates for electricity produced from high-efficiency cogeneration can be sold and purchased. The aim is to increase the demand for high-efficiency cogeneration power production, and by this reduce the primary energy consumption of the power sector.

The directive contains rules on how to calculate the amount of electricity that can be awarded a certificate:

- If the total efficiency is above a certain level, which is dependent on the technology, all the power produced are awarded certificates. For Combined Cycles and condensing steam extraction turbine plants, the requirement is a total efficiency of 80%, while the rest of the technologies need to have an efficiency of 75%
- If the total efficiency is below this level, the amount of electricity that can be awarded a guarantee of origin certificate is found by multiplying the yearly production of useful heat with the annual power to heat ratio. In this way, plants that are only run in cogeneration mode a part of the year do not get awarded certificates for all of their production.

The guarantee of origin certificate shall also contain the primary energy savings in percent, which in fact represents the fuel savings:

$$PES = \left(1 - \frac{1}{\frac{H_{\eta,CHP}}{H_{\eta,ref}} - \frac{E_{\eta,CHP}}{E_{\eta,ref}}} \right) \cdot 100\% \qquad (1)$$

Where:

PES	[%]	Primary Energy Savings
$H_{\eta, CHP}$	[%]	Heat production efficiency, CHP plant
$H_{\eta, ref}$	[%]	Heat production efficiency, reference facility
$E_{\eta, CHP}$	[%]	Electricity production efficiency, CHP plant
$E_{\eta, ref}$	[%]	Electricity production efficiency, reference facility

1.1.5 Renewable Energy Certificates

From the 1st of January 2012, a common Swedish-Norwegian market for renewable energy certificates will be opened. The certificates comprise renewable electricity production, and are meant to be energy neutral. This means that all renewable energy sources are included. For hydropower there are some restrictions, as only new facilities with less than 1 MW installed capacity has the right to get certificates. [15]

The certificates are distributed based on annual production, with one certificate per produced MWh. The goal is to realise a total of 26,4 TWh of renewable electricity production within 2020.

To create a demand for the certificates, all energy distributors that sell electricity to consumers need to buy certificates for a certain fraction of the total amount of electricity they sell. [15]

1.1.6 Recent Methodology Developments for Primary Energy Factor Calculations

In 2011, a report was published within Annex IX of the International Energy Agency - District Heating and Cooling[16]. The title was "The potential for Increased Primary Energy Efficiency and Reduced CO_2 Emissions by District Heating and Cooling", and one of the aims was to investigate the primary energy factors and CO_2 emissions for a district heating energy supply chain with a combined heat and power plant. The work was performed by SINTEF, SP Technical Research Institute of Sweden and KDHC – Korea District Heating Technology Research Institute.

As a part of this project, a calculation tool was developed in Excel to calculate primary energy factors for district heating under different circumstances. The main focus was not to study the combined heat and power plant itself, but all the processes surrounding it. The CHP plant was therefore modelled as black box with a constant efficiency and power to heat ratio. One of the main conclusions in the report was that the CHP plant was one of the most influential process regarding primary energy losses in the energy supply chain.

Consequently, a more detailed description of the CHP plant that takes into account technology type and part load operation would contribute to an increased accuracy of the primary energy factor calculations.

1.2 PROBLEM STATEMENT

Globally there is a strong need for more efficient use of energy resources, and within the EU there is a strong focus on efficient and renewable energy production. As participant in the Nordic electricity market and a member of the EEA agreement, Norway is also a part of this development.

District heating networks represent an alternative energy supply chain to the direct use of electricity which has traditionally been dominant in the Norwegian energy system[17]. This represents a way of reducing electricity consumption for heating purposes, and can thus lead to a decrease in both the total electricity demand and peak electricity load. This can in turn release electricity production for other purposes, and displace less efficient power production on the continent. District heating including combined heat and power production will have the same effects, but stronger. CHP production might in addition also help reduce bottle necks in the Norwegian transmission net.

This autumn Norway passed the renewable energy directive, and in January 2012 the renewable energy certificate market will be implemented. This might encourage more investment in renewable CHP plants connected to district heating networks.

Due to the changes in the recast of the Directive on Energy Performance of Buildings, it is a strong possibility that the Norwegian energy performance certificate system in the future will have to include the building's primary energy use. Thus, there is a need for calculating primary energy factors that are applicable for district heating under Norwegian conditions.

As described in paragraph 1.1.6, in district heating systems with cogeneration the CHP plant itself has a large influence on the primary energy factor.

Based on the preceding, the following questions emerge: Within what range are the primary energy factors for district heating with cogeneration in Norway? What parameters influence, and in what order of magnitude? How can the primary energy efficiency be increased?

1.3 ASSUMPTIONS AND DELIMITATIONS

The problem statement of the thesis is wide. To narrow down the scope of the study, adequate assumptions and delimitations have to be made.

The combined heat and power technologies studied will be chosen based on Norwegian conditions. A mapping of current and planned facilities will make the foundation of this choice.

As a consequence of this choice of emphasis, the calculation of the district heating load duration curve will not necessarily be based on a real case.

The project will hence be based on the following division of the original research questions:

- What is the current situation in Norway regarding CHP plants connected to district heating networks?
- What CHP technologies are available, and which are more relevant in a Norwegian context?

- How do these technologies perform under various part load conditions? What other parameters influence the performance, and in what order of magnitude?
- How does the CHP plant performance influence the district heating primary energy factor compared to other parameters?
- What other processes are the most important ones to improve to increase the primary energy efficiency?
- What parameters influence the primary energy factor, and in what order of magnitude?

In addition to the main assumptions and delimitations listed above, more simplification will need to be done along the way. This is especially true for the modelling and operation strategy for the cogeneration plant. These will be described where it is suitable in the report.

1.4 Report Structure

Chapter 2: Combined Heat and Power Plants, contains the thermodynamic theory on CHP, descriptions of different technologies and the mapping study on the current situation in Norway regarding combined heat and power plants connected to district heating networks.

In Chapter 3: Primary Energy Factors, the theory behind primary energy factors are explained more in detail. The basics of life cycle assessment are presented, and the reasoning behind various allocation methods is explained.

The methodology that constitutes the basis for all simulations and calculations are explained in detail in Chapter 4: Methodology. Section 4.2 and 4.3 treat the modelling of the CHP plants, while section 4.4 gives an overview of the calculation tool that is utilised to calculate the primary energy factors.

Chapter 5: Case Descriptions, describes the cases that have been studied. Section 5.1 describes the simulations that were run to investigate the performance of the CHP plants, while section 5.2 presents the cases that were studied using the primary energy factor calculation tool.

The results of the CHP plant simulations are presented in Chapter 6: Results and Analysis – CHP Plant Performance.

Chapter 7: Results and Analysis – Primary Energy Factors, contain the results from the cases presented in section 5.2.

The results and methods are discussed in Chapter 8: Discussion.

Conclusions are drawn and further work suggested in Chapter 9: Conclusions.

2 COMBINED HEAT AND POWER PLANTS

This chapter begins with an explanation of the basic thermodynamic principles that are relevant when analysing cogeneration. In section 0, the main different technologies are explained, and in section 2.3 special features on combined heat and power (CHP) plants in district heating networks are clarified. Section 2.4 gives an overview on the situation for CHP plants in Europe, while section 2.5 provides an up to date status on existing and planned CHP plants connected to district heating networks in Norway.

2.1 THERMODYNAMIC CONCEPTS

The analysis of a CHP plant is based on thermodynamic relations, of which the most important ones are presented in this section. Note that all equations are based on the sign convention where heat is positive when entering a system, while work is positive when leaving a system. Equations in this section are reproduced from [18] unless otherwise is stated.

2.1.1 THE FIRST LAW OF THERMODYNAMICS

The first law of thermodynamics states that the total amount of energy is conserved in all energy conversions and transfers. In a closed system, energy can be transferred by the means of heat or work. If one assumes that kinetic and potential energy is neglected, this results in a change in the amount of internal energy.

$$dU = \delta Q - \delta W \tag{2}$$

Where:

U	Internal energy	[J]
Q	Heat	[J]
W	Work	[J]

For an open system, energy will also be transferred to and from the control volume that encloses the system by mass flows entering and exiting across the system boundary. At rate form, this can be expressed as:

$$\frac{dE_{CV}}{dt} = \dot{Q}_{CV} - \dot{W}_{CV} + \sum_i \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i\right) - \sum_e \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e\right) \tag{3}$$

Where:

$\frac{dE_{CV}}{dt}$	Energy change within the control volume (CV)	[W]
$\dot{Q}_{CV}$	Heat transferred across the system boundary	[W]
$\dot{W}_{CV}$	Work related to shafts, displacement of boundary and el. Effects	[W]
$\dot{m}$	Mass entering or exiting CV	[kg]
h	Enthalpy	[J/kg]
V	Velocity	[m/s]
g	Gravity constant	[m/s^2]
z	Elevation	[m]

2.1.2 The Second Law of Thermodynamics

The first law of thermodynamics does account for the energy balance, but there are several questions that cannot be answered by applying this law.

The second law of thermodynamics gives us the possibility to predict the direction of a process, establish conditions for equilibrium, and determining the best theoretical performance of cycles and engines. Also, it is possible to do an analysis on every element of a cycle, and determine where losses occur that prevents us from reaching the theoretical maximum efficiency.

There are many alternative statements of the second law. One of them is the Clausius statement:

"It is impossible for any system to operate in such a way that the sole result would be an energy transfer by heat from a cooler to a hotter body"[18]

Another is the Kelvin-Planck statement:

"It is impossible for any system to operate in a thermodynamic cycle and deliver a net amount of energy by work to its surroundings while receiving energy by heat transfer from a single thermal reservoir."[18]

To do quantitative calculations based on the second law, the property entropy is used. Entropy is not a physical property that can be measured with instruments, but a theoretical quantity that says something about the disorder within the system. The amount of entropy in the universe is ever increasing, and in all real processes, there will be entropy production.

The entropy balance for a closed system going from state 1 to state 2 can be stated the following way:

$$S_2 - S_1 = \int_1^2 \left(\frac{\delta Q}{T} \right)_b + \sigma \tag{4}$$

Where:

S	Entropy	[J/K]
T	Temperature	[K]
$\boldsymbol{\sigma}$	Entropy production	[J/K]

The integral $\int_1^2 \left(\frac{\delta Q}{T}\right)_b$ represents the entropy change due to heat transfer across the system boarder. σ is the symbol of entropy production. If σ is equal to zero, there are no irreversibilities present within the system. This means that the system is fully reversible, which is only possible in theory.

If σ is greater than 0, irreversibilities are present within the system. By calculating the entropy-production of a process, one can assess the potential for improvement compared to the theoretical maximum performance.

From equation(5), it can be seen that an adiabatic, fully reversible process is equal to an isentropic process.

2.1.3 THE CARNOT POWER CYCLE

The Carnot power cycle is a thermodynamic cycle consisting of four internally reversible processes, and is thus a fully reversible cycle.

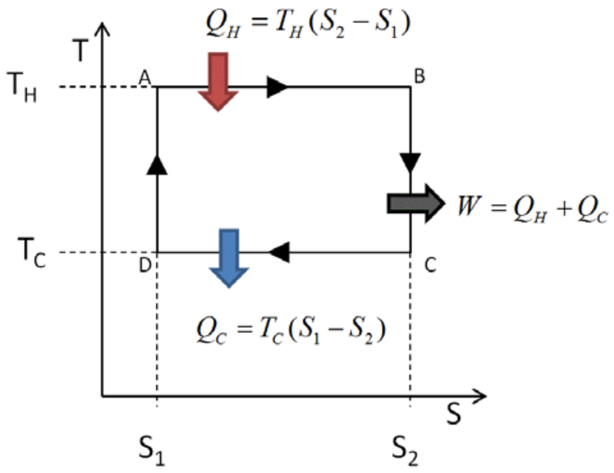


FIGURE 4: CARNOT POWER CYCLE

Figure 4 shows the temperature - entropy (T-s) diagram for the cycle. From state A to state B, heat is transferred isothermally into the system. The entropy increases due to the entropy that accompanies this heat transfer. Process B to C is adiabatic and reversible, and therefore also isentropic. Process C to D consists of an isothermal heat removal, and process D to A is an isentropic temperature increase. The work that can theoretically be produced is the sum of the heat that transferred to and from the system.

The Carnot Power cycle is used to calculate the highest theoretically possible efficiency for any power cycle operating between two heat reservoirs at different temperatures. This efficiency is called the Carnot efficiency, and can be expressed by the temperatures of the hot and cold reservoir:

$$\eta_{Carnot} = 1 - \frac{T_C}{T_H} \tag{6}$$

Where:

T_C	Temperature of cold reservoir	[K]
T_H	Temperature of hot reservoir	[K]

2.1.4 EXERGY

As the Carnot efficiency implies, it is not possible to convert all the energy in a thermal reservoir into work, even with a fully internally reversible power cycle. To describe the theoretical maximum quantity of work that can be obtained from a system as it reaches equilibrium with its surroundings, the term exergy is used. The remaining energy contained in the system is called anergy. The total amount of energy in a system is thus the sum of exergy and anergy.

To describe power cycles, exergy is a useful term, as it helps us to determine how much influence energy losses in a power cycle will have on power output.

Exergy is closely related to the concepts presented in paragraph 0, but unlike the Carnot efficiency, exergy is always calculated relative to ambient conditions. As the Carnot efficiency describes the maximum theoretical efficiency for power production from two heat reservoirs at different temperatures, it follows that the exergy content of a certain amount of heat energy can be described as:

$$\mathbb{E}_Q = Q\left(1 - \frac{T_a}{T_H}\right) \tag{7}$$

Where T_a is the ambient temperature in Kelvin.

I can be shown that the exergy balance for a closed system going from state 1 to state 2 is:

$$\mathbb{E}_2 - \mathbb{E}_1 = \int_1^2 \left(1 - \frac{T_a}{T_b}\right)\delta Q - \left[W - p_a\left(V_2 - V_1\right)\right] - T_a\sigma \tag{8}$$

$\int_1^2 \left(1 - \frac{T_a}{T_b}\right)\delta Q$ is the exergy transferred across the system boarder accompanying heat transfer at T_b, which is the temperature at the system boarder.

$\left[W - p_a\left(V_2 - V_1\right)\right]$ is the total amount of work that will leave the system. V is in this case volume[m^3]. W is work produced within the system, while $p_a\left(V_2 - V_1\right)$ is work used for expanding the system boarder.

$T_a\sigma$ is the *exergy destruction.* In a fully reversible system, the energy destruction will be zero, while it will be non-zero in a real system. Clearly, entropy production and exergy destruction are closely related.

2.1.5 THERMODYNAMIC PERFORMANCE INDICATORS

The first law efficiency of a thermodynamic cycle can be expressed as:

$$\eta_I = \frac{W_{out}}{Q_{in}} \tag{9}$$

Where W_{out} represents the net work output from the plant and Q_{in} is the total heat input measured by the lower heating value (LHV) if combustion of a fuel is involved. If electricity is the only work output from the plant, this efficiency is often called the power efficiency or the electric efficiency, and referred to as η_{el}.

In the case of CHP production, a part of the waste heat is also a useful product. In this case, the total efficiency can be expressed as:

$$\eta_{tot} = \frac{W_{out} + Q_{out,useful}}{Q_{in}} \tag{10}$$

For a CHP plant, η_{tot} is often referred to as η_{CHP}.

Sometimes it is also of interest to explore the total exergy efficiency of a cycle, or to assess the exergy performance of the different processes of the cycle. The exergy efficiency is simply defined as:

$$\eta_{II} = \frac{\mathbb{E}_{out}}{\mathbb{E}_{in}} \tag{11}$$

This is also called the second law efficiency. To calculate the total exergy efficiency of a thermal power cycle, (11) equals:

$$\eta_{II} = \frac{W_{out}}{\mathbb{E}_{Q}} \tag{12}$$

Another important parameter to assess a CHP plant is the power to heat ratio (PHR) :

$$\alpha = \frac{W_{out}}{Q_{out}} \tag{13}$$

In the rest of the report, α and PHR will both be utilised to refer to the power to heat ratio, α mostly in equations and PHR in the text.

Efficiencies are normally given for a system operating at steady state at its design point. However, in many cases the load will vary throughout the year, and the performance of the system vary with the load. Then it is more suitable to use annual efficiencies and power to heat ratios to describe the performance of the plant.

2.2 Combined Heat and Power Plant Technologies

Combined heat and power production is a very wide term that covers a vast range of applications and technologies. When sorted by application, normally four different categories are used[19]:

- Industrial applications
- Energy supply to large buildings or a small cluster of buildings
- District heating and cooling
- Micro CHP for households

The categories are, however, not mutually exclusive. It is perfectly possible for a plant to be included in more than one category, and the most common combination is a district heating CHP plant that also delivers heat and electricity to industrial applications, or vice versa.

There are two main principles for combined heat and power production[9]:

1. *Topping-cycle*. Power is produced utilizing high temperature energy, and rejected heat is used for thermal purposes.
2. *Bottoming-cycle*. Energy input is first used to supply a high temperature thermal load, and rejected heat emerging from the thermal application is then used for power production. This is for instance the case when waste heat from industrial appliances is used for power production.

When a CHP plant is supplying heat to a DH network, this is of course an example of a topping-cycle plant.

Within the district heating and cooling application category, many possible CHP technologies exist. In this report, they are sorted and presented by prime mover.

2.2.1 Combustion Turbine Generators with Heat Recovery

This category contains all sorts of gas turbines. The key components are a compressor, a combustor and a turbine. The working fluid is normally air, which is taken from the environment around the facility [20]. The ambient conditions of the air, like temperature and humidity, are therefore of interest when the performance of the gas turbine is calculated. For instance, the electrical efficiency will increase when the ambient temperature decreases[21].

A wide variety of fuels are applicable. Natural gas is the most common, but biomass gases, manufactured gases, fuel oils and liquefied petroleum gases are also an option. Liquid fuels will, however, need pre treatment before use.[22]

To produce the heat, heat recovery steam generators (HRSG) are used. The HRSG is a heat exchanger where the hot flue gas exiting the gas turbine (typically 450-650 °C) is used to heat a fluid, normally oil or water. Sometimes the HRSG also has supplementary firing, which often is called duct firing. This makes it possible to add more heat if the heat content of the flue gas supplied by the gas turbine is not enough. Large temperature differences between the hot and the cold stream in the HRSG will give large exergy losses, and the heat recovery is therefore often done in two or more stages.

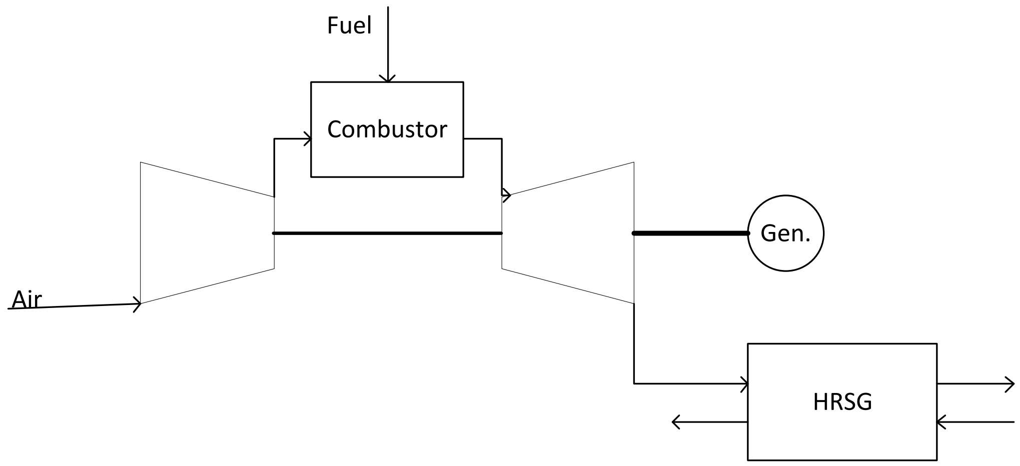


FIGURE 5: GAS TURBINE WITH HEAT RECOVERY

The amount of heat extracted does not influence the amount of power produced, only the amount of heat released to the environment. If the heat demand is low, it is possible to reduce the power output by running the gas turbine at part load.

Combustion turbine generators (CTGs) exist in a wide range of sizes, from about 50 kW to above 300 MW. In cogeneration facilities, the size of the turbine is normally in the range from about 2 MW to 20 MW. [9]

2.2.2 Steam Turbines

Steam turbine systems are based on the Rankine cycle. The working fluid is normally water, but other fluids might also be used. This can for example be the case if one wants to exploit waste heat for power production, because then the working fluid needs to have a lower boiling temperature than water. If an organic fluid is used, this is called an Organic Rankine Cycle (ORC).

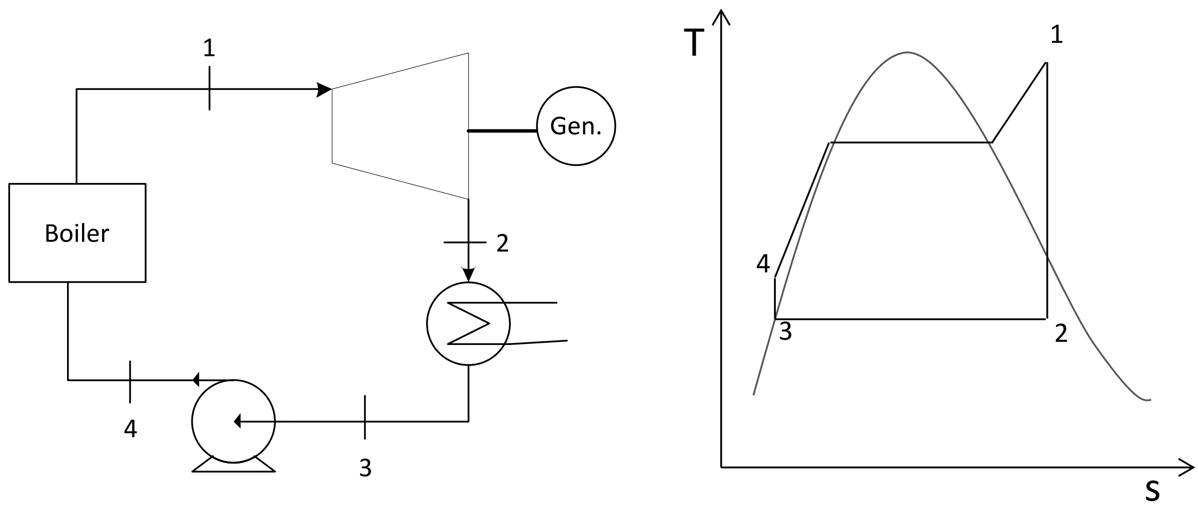


FIGURE 6: IDEAL RANKINE CYCLE

The flowsheet and T-s diagram of an ideal Rankine cycle is shown in Figure 6. The cycle consists of four fully reversible processes. In process 3-4, condensed liquid is pumped isentropically to a higher pressure. From 4 to 1, heat is added to the system at a constant pressure and the working fluid goes from a subcooled to a superheated state. From state 1 to state 2, the steam is expanded isentropically, normally all the way to the two phase area. Then the steam-liquid mixture is condensed, either by a water cooling loop or in air cooled cooling towers.

Work is created when the steam expands through the turbine.
Isentropic efficiency for a turbine is defined as:

$$\eta_{is} = \frac{(h_1 - h_{2,real})}{(h_1 - h_{2s})} \tag{14}$$

Where $h_{2,real}$ is the actual end state of the expansion on h_{2s} is the isentropic end state. As the Rankine cycle in Figure 6 is an ideal one, the expansion is isentropic. Thus state 2 of course equals state 2s which results in an isentropic efficiency of 100%.

For a pump (or compressor in the case of gas turbines) the isentropic efficiency becomes:

$$\eta_{is} = \frac{(h_3 - h_{4s})}{(h_3 - h_{4,real})} \tag{15}$$

The work of a steam cycle is expressed by:

$$W = \eta_{is} \cdot (h_1 - h_2) \tag{16}$$

There are many ways of increasing the power efficiency of a steam cycle. Extra superheating of the steam, feedwater heaters, and multi-stage expansion with reheating are some examples.[22]

Steam cycles can in theory operate on all sorts of fuels, as the combustion is happening outside the closed cycle of the working fluid. Coal is the most common option, while solid biomass and waste also represent relevant alternatives. Large solar power plants and nuclear power plants are also based on steam cycles.

Steam turbines can be found in any size, but are normally used from 250 kW and upwards. It is, however, not common to install steam turbines in small facilities (<1MW)[9].

Steam Cycles as CHP plants

There are two main categories for steam turbines used for cogeneration:

- Back pressure turbines
- Extraction condensing turbines

In a back pressure turbine, the steam is not expanded to the lowest possible level, but to the pressure required by the process.

Condensing turbines expand the steam as much as possible, and well below atmospheric pressure. Combined with an extraction system, part of the steam can be extracted at an intermediate (or many) pressure level(s). This way, it is possible to supply steam at high pressures and temperatures, for instance for process applications. On the other hand, condensing turbines require large dimensions due to major volume flows as the pressure approaches the minimum level.[23]

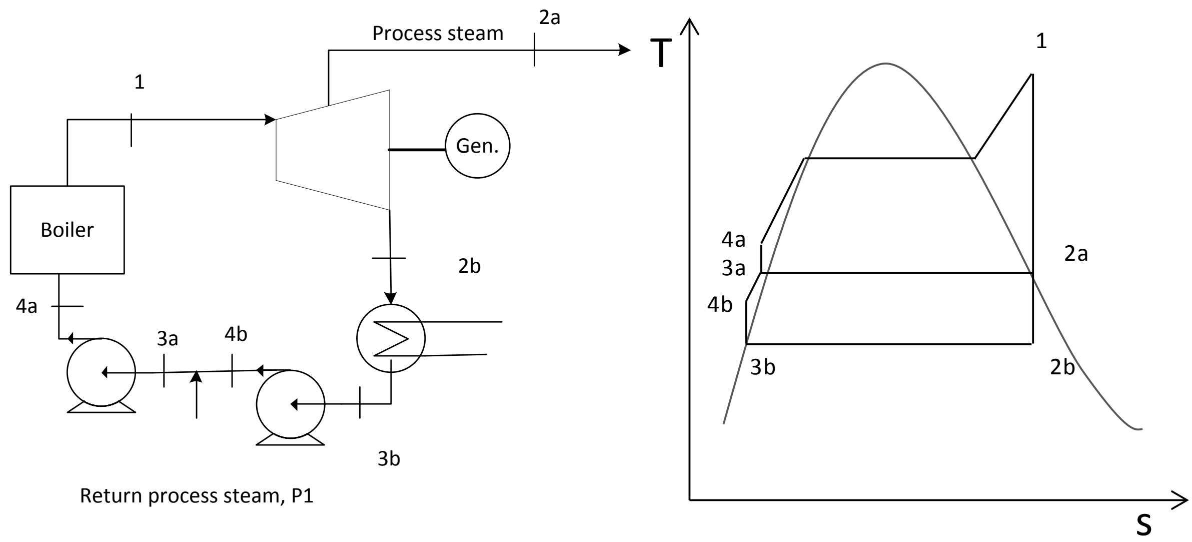


FIGURE 7: SIMPLIFIED STEAM CYCLE WITH EXTRACTION

In an extraction system, increased heat production will reduce the power production. The ratio between lost power production and increased heat production is expressed by the C_v factor[19]:

$$C_v = \frac{|\delta P|}{|\delta Q|} \tag{17}$$

Figure 7 illustrates the flow sheet of a steam cycle with extraction for a process and the corresponding T-s diagram. A fraction of the steam, y [%], is extracted at state 2a instead of being expanded to state 2b. As the expansion is isentropic, the corresponding power loss is:

$$\delta P = y \cdot (h_{2a} - h_{2b}) \tag{18}$$

The corresponding heat output depends on the enthalpy of the return process stream, P1:

$$\delta Q = y \cdot (h_{2a} - h_{P1}) \tag{19}$$

The C_v factor is then:

$$C_v = \frac{(h_{2a} - h_{2b})}{(h_{2a} - h_{P1})} \tag{20}$$

The higher the pressure is of the steam extracted, the more the power output will be reduced per unit of heat that is extracted. The heat/power ratio is thus not fixed, and the power production will therefore reach its maximum when heat production is zero.

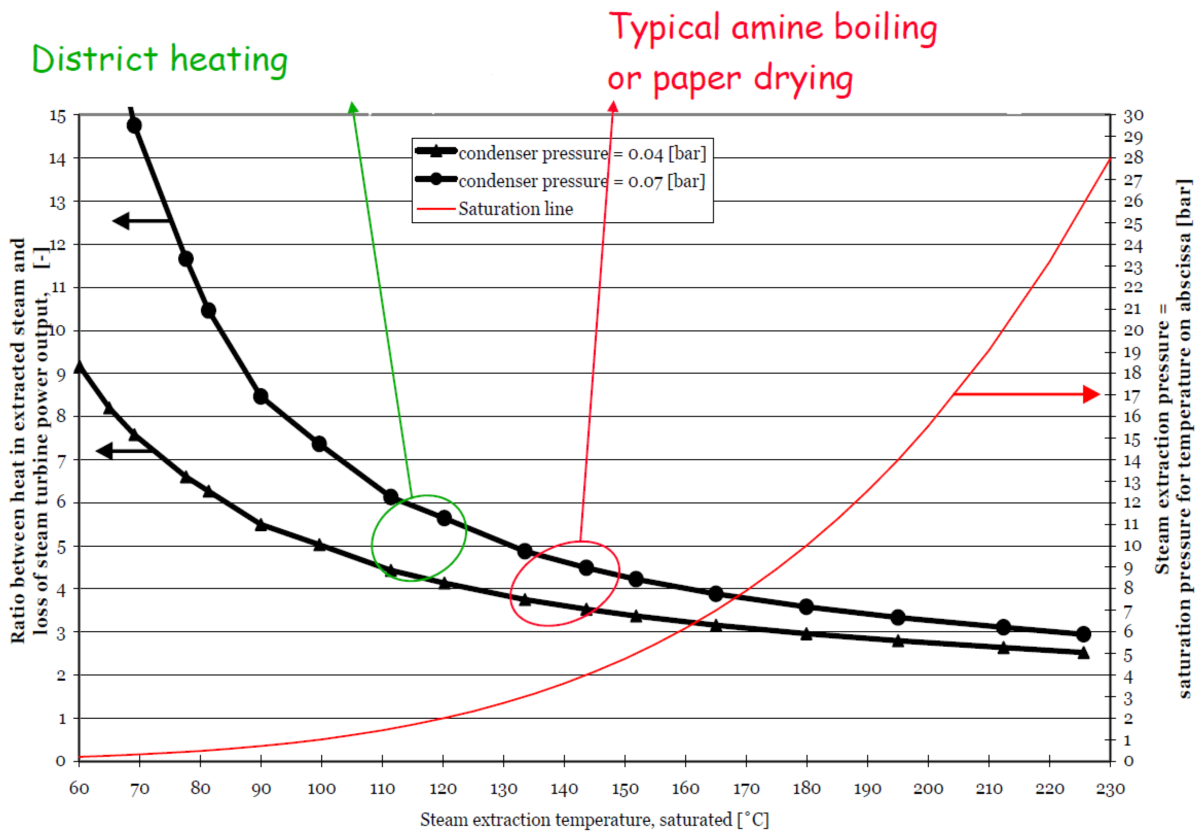


FIGURE 8: THE INVERSE OF CV PLOTTED AGAINST STEAM EXTRACTION TEMPERATURE. REPRINTED FROM [20]

In Figure 8, the ratio between heat in extracted steam and loss of steam turbine power output is plotted for two different condenser pressure. This ratio corresponds to the inverse of the C_v factor and it can be viewed as the CHP version of the coefficient of performance (COP) for heat pumps.

The figure shows clearly that by extracting steam at low temperatures, the increased heat output is far higher than the power loss. When the temperature increases, the heat output is lower per unit of lost power.

The condenser pressure plays an important role, and a low condenser pressure gives a higher ratio of heat output to power loss. This is of course because when the steam can expand to a lower pressure, more work is produced after the extraction point compared to a higher condenser pressure. At the same time, a higher condenser pressure might make also the condenser heat usable for heating purposes.

In a back pressure system, the pressure ratio of the turbine is quite fixed, and the power production is decided by the steam flow. If the heat production is reduced, this means that the power production is also reduced. This results in a direct coupling between heat and power production.[23]

The temperatures of the heat demand will, however, influence the power output also in this case. If the temperature requirement is high, the condenser pressure needs to be raised accordingly. This will of course lead to that the steam expands less, and the amount of work produced is reduced.

2.2.3 COMBINED CYCLE

In a combined cycle, the hot flue gas of the gas turbine is used to generate steam in a steam cycle. The principle flow diagram is shown in Figure 9.

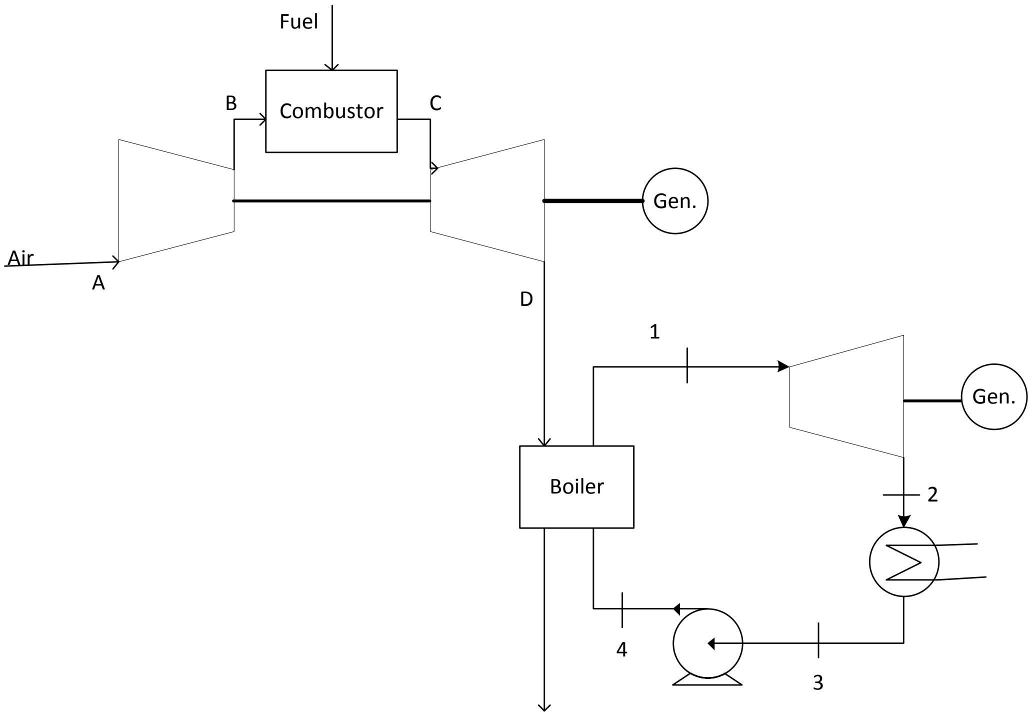


FIGURE 9: FLOW SHEET OF A COMBINED CYCLE

The corresponding T-s diagram is visualised in Figure 10.

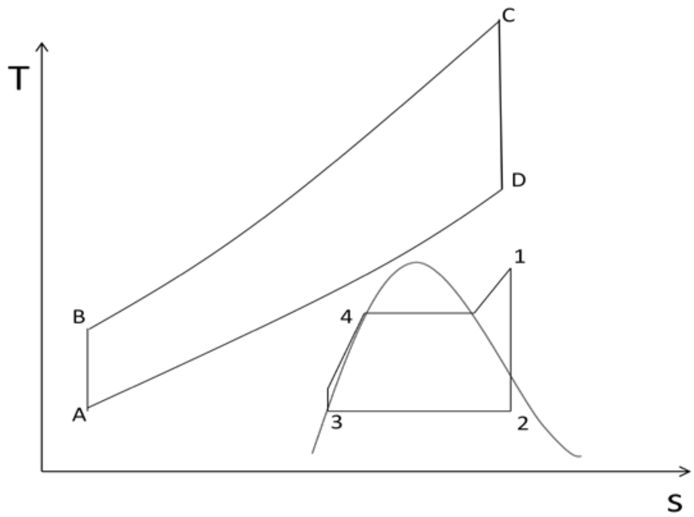


FIGURE 10: ILLUSTRATIVE T-S DIAGRAM OF A COMBINED CYCLE. THE GAS TURBINE IS REPRESENTED BY A BRAYTON CYCLE.

The combined cycle offers higher power efficiency than what can be obtained with a gas turbine of steam cycle separately. In state-of-the-art facilities, power efficiency range between 57% and 60%[24].

As this kind of plants require large capital investments, the technology is normally not utilised for plants with a power output of less than 20MW[19].

2.2.4 INTERNAL COMBUSTION RECIPROCATING ENGINES

In a reciprocating internal combustion (RIC) engine, the combustion takes place inside a cylinder, and the induced volume change moves a piston which in turn generates rotation on a shaft. This mechanical work can then be utilized to produce electricity in a generator.

Both liquid and gaseous fuels can be utilized, but the engine has to be designed for the fuel in question. This leads to a general separation between gas engines and diesel engines, where one of the main difference is that gas engines have an ignition source while diesel engines uses compression to initiate ignition.[18]

Heat is mainly recovered from the cooling jacket and the flue gases, but heat from intercoolers and the lube oil is also recovered in some cases. The temperatures available vary depending on the type of engine. The cooling water exit temperature ranges from 90 to 120°C, while the temperature of the flue gases ranges from 250 to 650 °C depending on type of fuel and engine model. Engines running on diesel generally have lower heat recovery temperatures than gas engines.[19]

In Figure 11, a flow sheet of a natural gas engine with heat recovery is presented. Here, exhaust gas, jacket water and lube oil heat are all exploited.

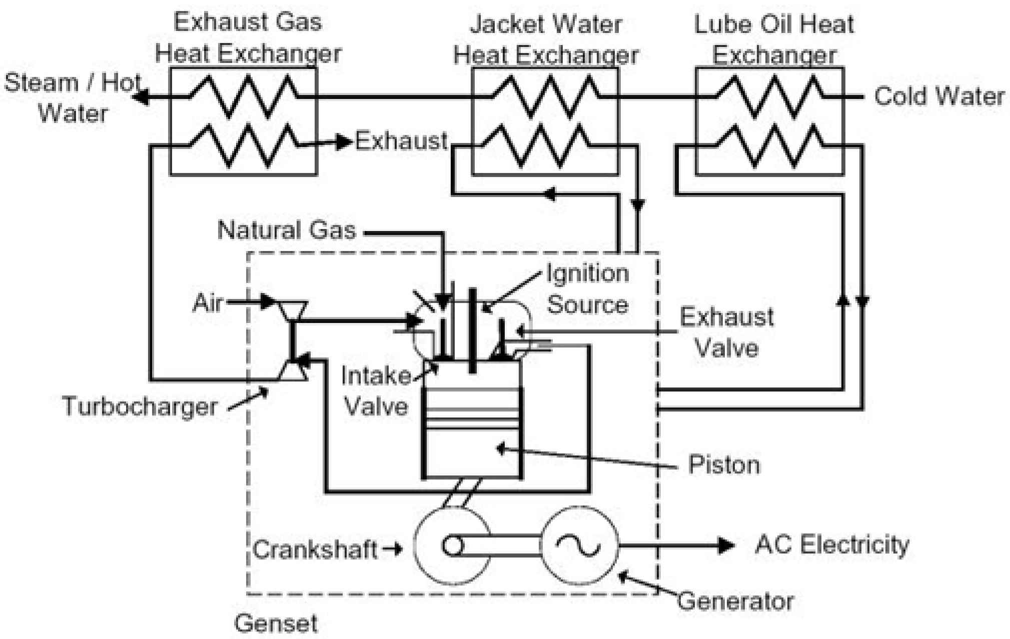


FIGURE 11: GAS ENGINE WITH HEAT RECOVERY. REPRINTED FROM [25]

The power efficiency of RIC engines is typically in the range of 40-43%, and the total efficiency at design conditions can get as high as 92%.[19]

RIC engines are normally used as prime mover in smaller CHP plants with less than 2 MW installed capacity (electricity), but engines exist in the range from approximately 30 kW to 15 MW[19].

2.2.5 Fuel cells

A fuel cell is an electrochemical device that converts fuel into electricity via a chemical reaction. The most common fuel is hydrogen, and the products are electricity, heat and water. Natural gas, biogas and diesel are also usable fuel options. There are many different types of fuel cells that cover most types of application within central and stand-alone CHP generation and mobile use in vehicles, but not all of them are commercially mature. Still, interest is picking up, both in micro-scale domestic CHP applications, and in larger distributed energy facilities[26].

The main technologies at the time are:

- Solid oxide fuel cell (SOFC)
- Molten carbonate (MCFC)
- Phosphoric acid (PAFC)
- Solid oxide (SOFC)
- Direct methanol (DMFC)

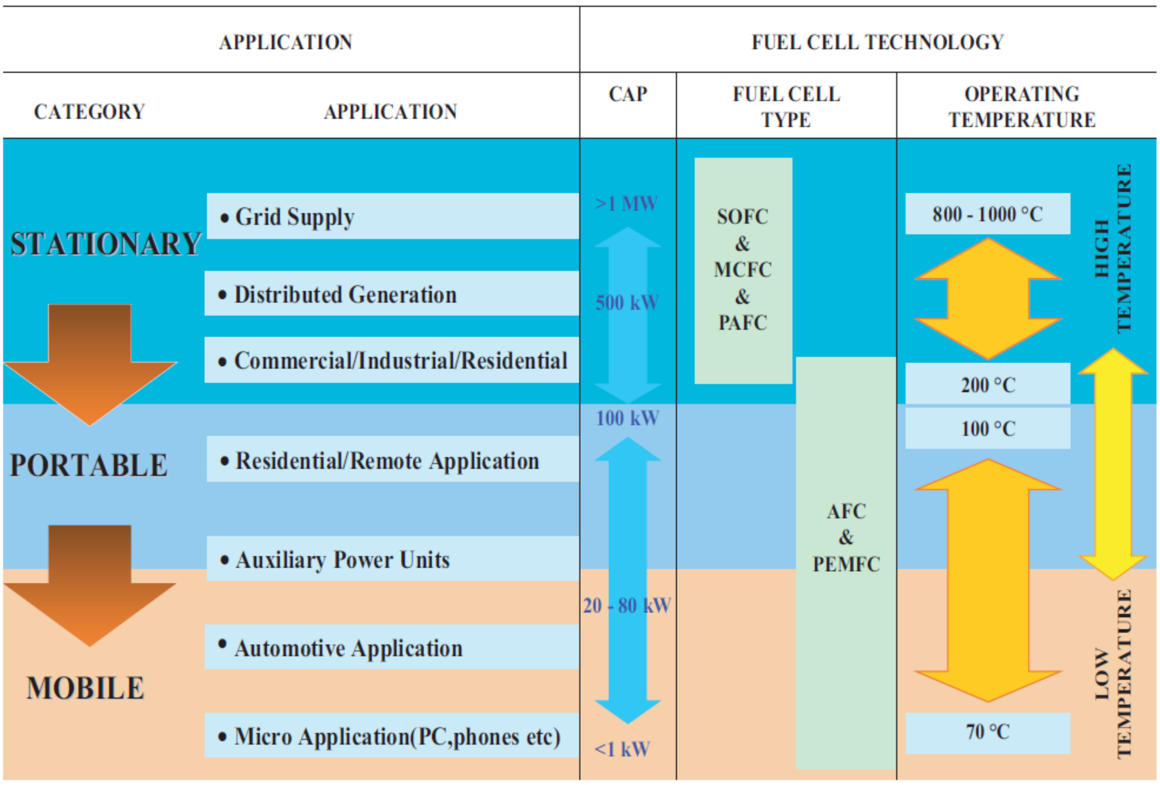


FIGURE 12: APPLICATION OF FUEL CELLS. REPRINTED FROM [26]

In Figure 12 the different fuel cells and their operating temperatures and applications are summarised. The power efficiencies ranges from 35 to 55 %[26], while the total efficiency in CHP applications can reach 85 %[9].

2.2.6 STIRLING ENGINES

Stirling engines are based on external heat supply, and can therefore run on any sort of heat supply. In addition to normal combustion, stirling engines based on solar energy do also exist.

Theoretically, a stirling engine can reach Carnot efficiency, but for the time being the power efficiencies normally range from 15 to 30%[27], but have been reported up to 40%[28].

Even though the stirling cycle has been known for a long time, there has yet to be introduced a commercially competitive engine. Research and product development are, however, ongoing. Small size, quiet operation and full fuel flexibility are, however, all attributes that make it probable that it can play a role in the future.[28]

2.2.7 CHOICE OF PRIME MOVER IN CHP SYSTEMS

In Table 1 capacity ranges, efficiencies and heat supply flexibility for different technologies are listed. The capacity ranges for most of the technologies are very large, and to decide what prime mover that is the most adequate, many factors need to be considered. The most important ones are:

- What is the main product, heat or electricity?
- Power to heat ratio
- Heat and/or power demand profiles
- Efficiency at part loads
- Fuel
- Total efficiency
- Possibility for heat accumulation
- Possibility of purchasing and selling electricity from/to the grid

To decide which one is the most appropriate prime mover in a specific situation, all these aspects need to be considered. In addition comes the economic aspect.

TABLE 1 : DIFFERENT CHP SOLUTIONS SUMMARISED. INFORMATION ON POWER EFFICIENCY, TOTAL EFFICIENCY AND FLEXIBILITY IN HEAT SUPPLY IS REPRINTED FROM [5], EXCEPT THE INFORMATION ON FUEL CELLS WHICH IS FROM [26] AND STIRLING ENGINES THAT ARE FROM [27] AND [28].

	Typical Power Capacity	Power efficiency [%]	Total efficiency [%]	Flexibility in heat supply
Diesel engines	30 kW – 15 MW	35 – 45	80 – 92	Low
Natural gas engines	30 kW – 15 MW	30 – 42	80 – 92	Low
Combustion turbine generators	1 MW – 100 MW	20 – 37	70 – 92	Large
Steam turbine systems	250 kW -	15 – 42	20 – 92	Large
Combined cycles	20 MW -	40 – 60	47 – 92	Large
Fuel cells	1 kW – 1 MW	35 – 55	Up to 85	n.a.
Stirling Engine	1 kW – 1,5 MW	15 – 40	65 - 85	n.a.

2.3 CHP Plants in District Heating Systems

When CHP-plants are integrated in a district heating grid, the main aim of the facility is normally to satisfy a given district heating demand. The electricity production can in this situation be regarded as a bonus, as the main product is the supplied heat. However, as mentioned in the first section of this chapter, some plants deliver energy to both industrial applications and district heating networks. In these cases, it is likely that the CHP plant is not alone responsible for the heat supply to the DH network, and other parameters than district heating demand might influence the operational strategy.

Waste incineration plants represent a special case of CHP plants in DH networks. Their task is to handle the incoming waste. If the heat demand is not matching the flow of incoming waste, the heat produced might be cooled away and not made use of. The use of absorption chillers makes it possible to employ the excess heat, but this is not a common installation in Norway.

In all the cases mentioned above, the electricity production from can be regarded as a secondary product, or a bonus. This is of importance when the primary energy factor is calculated, as will be further explained in the methodology chapter.

2.3.1 Design

There are many design options for including a CHP-plant into a district heating network. In some cases, the design will include process heat delivery in addition to district heating, in other cases heat storage might be a possibility. The design of the plant will be dependent on the following parameters[9]:

- Magnitude, duration and coincidence of electrical and thermal loads
- Choice of prime mover
- Waste heat recovery systems employed
- Facility location and distance from load centers
- Need for backup
- Staff capability and training

In the end, costs will be the most important decision variable regarding the design of the system.

2.3.2 Sizing

Normally, a CHP plant is not dimensioned to cover peak load, but all or a part of the base load. If the plant is oversized, this will lead to dumping of waste heat and low overall efficiencies. In addition, an oversized plant will have to run a lot on part load, which will lower the yearly efficiency.

An undersized system will, however, lead to a situation where the heat load is not served.

In reality, an optimisation of plant size is a complex task where capital costs and operation and maintenance costs for the given heat demand will decide the optimal size.

2.3.3 Operation Strategies

The operation strategy depends on the choice of prime mover and the design of the system. To optimise the operation is a complicated task, especially if there are alternative supply possibilities for the heat and/or electricity.

The parameters that need to be controlled in a district heating cogeneration plant, are the thermal output and the water temperature. The choice of technology, design and the sizing of the system will, however, influence the control strategy.

In a thermodynamic-economic optimisation, the costs of fuel, electricity and operation and maintenance, as well as start and stop costs will have to be taken into consideration. Technical aspects, like design efficiency, part-load behaviour and the possibility for storing thermal heat production either in a cool or heat storage will of course influence these costs.

The optimisation problems regarding sizing and operation will not be elaborated on in this thesis, as this is not the main focus of the problem statement.

2.4 Cogeneration Plants in Europe

The use of cogeneration differs a lot across Europe. Figure 13 is generated based on numbers from Eurostat, and shows the percentage of the total electricity production that origins from combined heat and power. Denmark, Finland and the Netherlands are amongst the countries with the highest fraction, while Norway and France are in the lowest category.

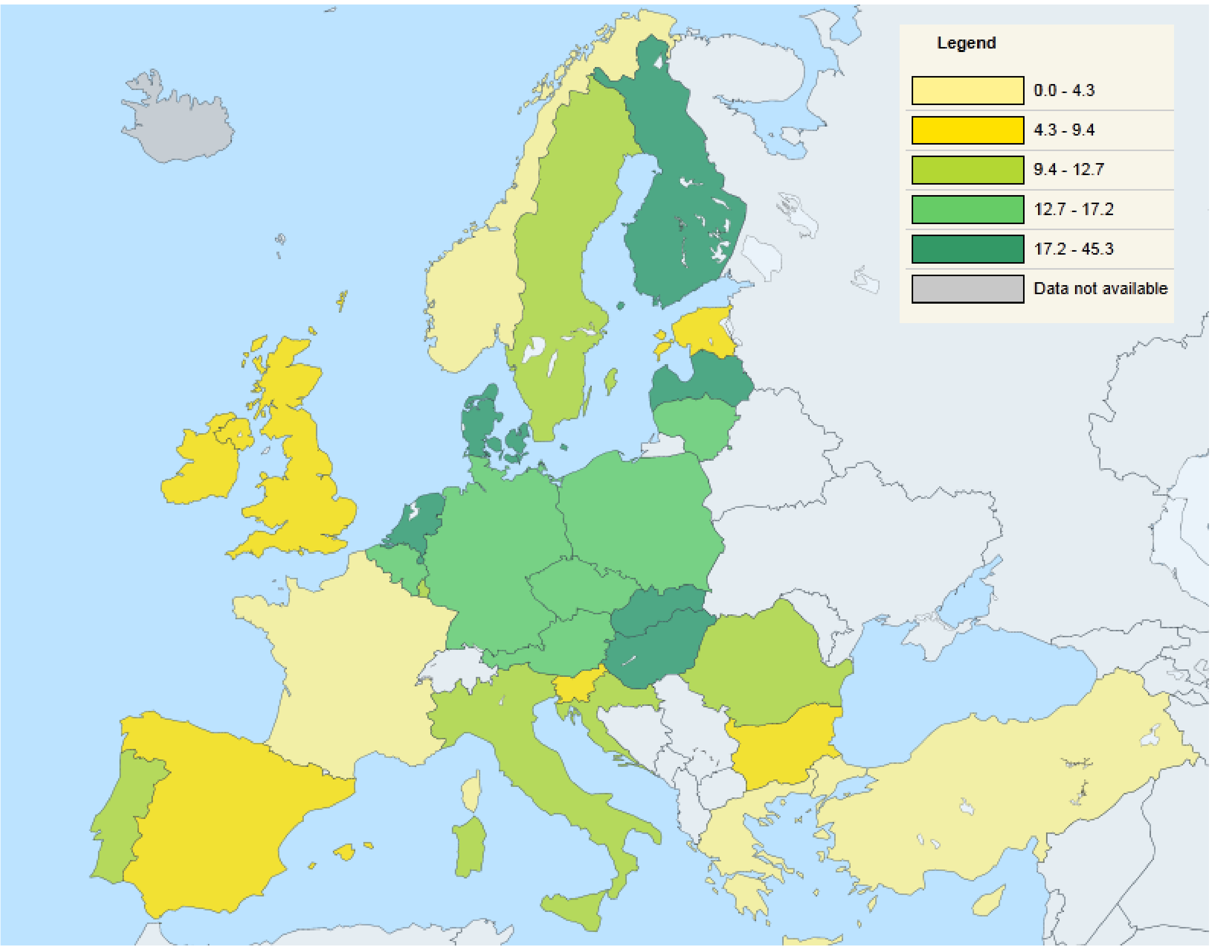


FIGURE 13: AMOUNT OF ANNUAL ELECTRICITY PRODUCTION SUPPLIED BY CHP PLANTS IN 2010, [%] [29]

Figure 14 show the relative utilisation of different technologies based on the ratio of CHP electricity to the total electricity production. Figure 15 visualises what technologies that are most utilised based on the fraction of total heat output.

When the technologies are presented based on heat output, steam backpressure turbine systems are by far the most utilised technology with a fraction of 47%. Combined cycle comes second with 20% and steam condensing turbine comes third with 14%. Gas turbines with heat recovery and internal combustion engines account for respectively 10 and 7 %, while 2% is supplied by other technologies.

When produced electricity is the basis for the graph, combined cycle is the most utilised technology, with 31% of the electricity contribution. Steam backpressure turbine follows closely with 30% of the production, while steam condensing turbine, gas turbine with heat recovery and internal combustion engine accounts for 12 and 13 % each. Other technology supply 2% of the total amount of CHP electricity.

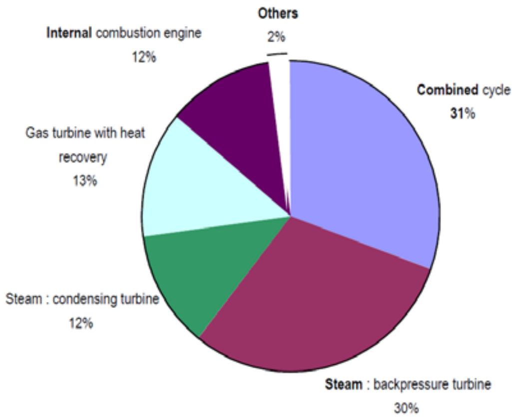


FIGURE 14: CHP ELECTRICITY PRODUCTION WITHIN THE EU BY TECHNOLOGY[30]

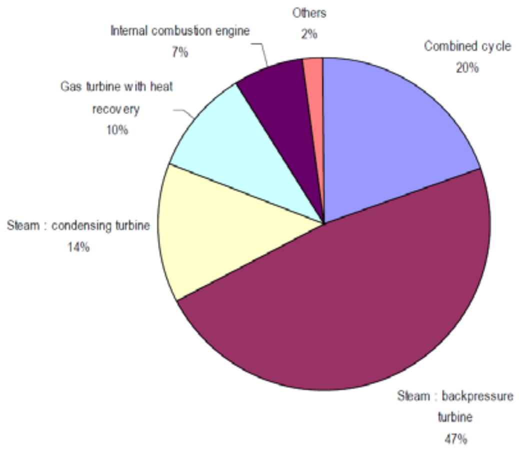


FIGURE 15: CHP HEAT PRODUCTION WITHIN THE EU BY TECHNOLOGY[30]

Figure 16 display the annual CHP efficiencies in various countries in the EU. The variation is notable, from above 85% in Luxenburg to approximately 55% in Greece. Sweden and Finland are second and third best, with efficiencies just above and below 80 %.

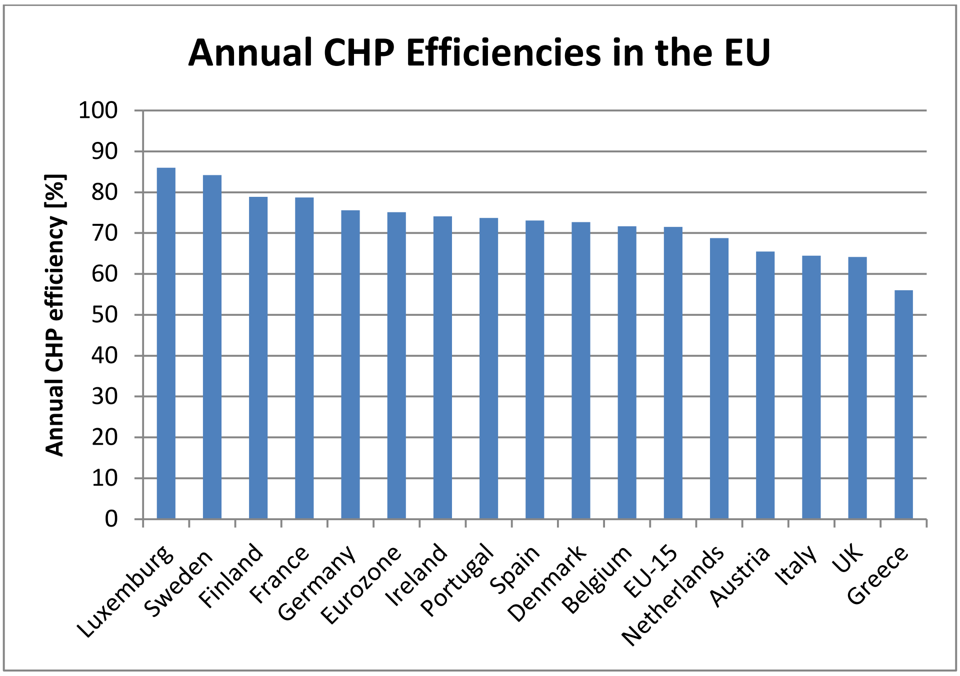


FIGURE 16: ANNUAL EFFICIENCY FOR CHP PLANTS IN THE EU FOR THE YEAR 2000, SORTED BY COUNTRY. BASED ON DATA FROM [30]

Figure 17 displays the annual CHP efficiencies in the EU per Technology. Also here there are differences, with the condensing turbine at 60%, while the rest of the technologies obtain between 75% and 78%. The large variation in efficiencies between the countries implies that there are large differences in the quality of the facilities. Therefore, state of the art facilities of the respective technologies will obtain higher efficiencies than displayed in the graph.

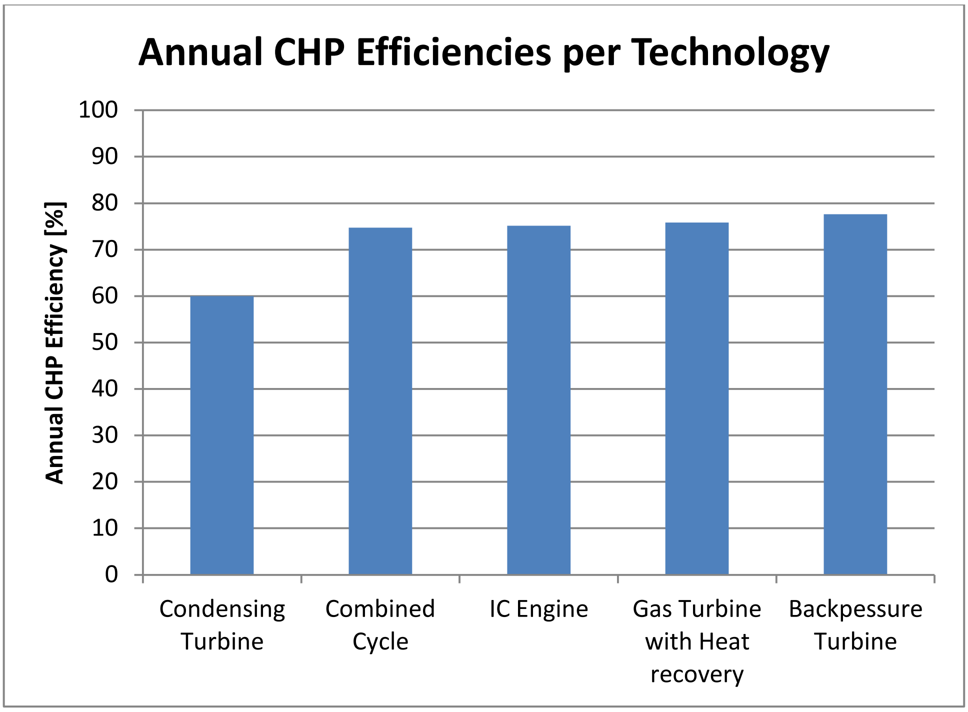


FIGURE 17: ANNUAL EFFICIENCY FOR CHP PLANTS IN THE EU FOR THE YEAR 2000, SORTED BY TECHNOLOGY. BASED ON DATA FROM [30]

2.5 CHP Plants Delivering Heat to DH Networks in Norway

In this section, two existing and one planned CHP-plant that deliver heat to district heating networks in Norway are described. In paragraph 2.5.4, a comprehensive overview of all Norwegian CHP plants connected to DH networks are presented. The information on the Norwegian CHP plants was acquired from many different sources. A standard letter with questions was sent to many of the facilities. This letter is found last in the appendix.

2.5.1 Fredrikstad Bio-El

In Fredrikstad, Hafslund has a CHP plant which is fired on homogenous, sorted waste, of which 80% origin from the industry and 20% is municipality waste.

The facility consists of a 25 MW boiler, one back pressure turbine and one condensing turbine. The turbines are operated on the same shaft like a two-stage turbine. The back pressure unit is a Dresser Rand B5S-5, while the condensing unit is a Dresser Rand B7S-3. The maximum output from the generator is 5,68 MW.

The supply temperature of the district heating water varies between 120 and 90 °C according to outdoor temperature, but is usually not above 115°C. The return temperature varies between 60 and 80 °C.

In addition to electricity, the plant delivers heat to a district heating network and process steam. The operational strategy is thus influenced by district heating demand, process steam demand and electricity prices. The facility is dimensioned so that at maximum load, the turbine will be stopped and all the energy produced is transferred to the district heating network.

The facility is quite new, and the production started in 2008. The production in 2010 was 53 GWh of district heating, 49,8 GWh of steam and 14,2 GWh of electricity.

2.5.2 Mosseporten Miljøenergi

Mosseporten was put into operation in 2010, and is a CHP plant which consists of a gas engine that operates on landfill gas.

The engine is of the brand Jenbacher, and maximum electricity power is 1063 kW. This gives a full load net electric efficiency of 40,43%. Maximum heat production capacity is 1122 kW, and thus the full load total efficiency is 83,11% and the power to heat ratio 0,94. The water in the heat recovery system is heated to 92 °C, and thereafter heat exchanged with the district heating water, which obtains a supply temperature of 88°C. As a result of this temperature constraint, heat from the engine is utilised for district heating only approximately 2500 h per year.

In Table 2, production figures for 2010 and estimates for 2011 and 2012 can be seen.

TABLE 2: PRODUCTION FIGURES FOR MOSSEPORTEN MILJØENERGI

	Production 2010 [MWh]	Estimated figures 2011 [MWh]	Estimated figures 2012 [MWh]
Electricity	252	734	8500
District heating	3183	6500	7000
Cold	9034	9100	9100
Steam	822	1300	1500

2.5.3 RANHEIM HEATING CENTRAL

Trondheim has a large district heating network, and Trondheim Energi has explored the possibility of building a CHP facility at Ranheim to be able to meet an increasing heat demand. At Ranheim, there is also a paper factory, and one of the alternatives considered included delivery of process steam to this factory. The location at Ranheim was, however, rejected by Trondheim municipality in 2011, and the process steam scenario will therefore not be included in this description of the plant. At the moment, Trondheim Energi is searching for a new location and it is uncertain when, where and if the facility will be constructed. Nevertheless, the facility will be addressed as Ranheim Heating Central in this thesis[1, 31].

The plant is expected to cover a part of the base load in the district heating network in Trondeim. It is designed according to a duration curve that is adjusted to the expected total heat demand in 2016, which is calculated to be 730GWh (The heat production in 2009 was 549 GWh). The CHP-plant is expected to deliver 202,6 GWh heat and 80 GWh electricity, which gives a yearly power to heat ratio of 0,39. Installed boiler capacity is 76 MW, maximum heat capacity delivered to the district heating network is 50 MW, and maximum power production from the generator is 21,7 MW. This gives a power to heat ratio of 43% at design conditions, assuming a district heating supply temperature of 75 °C. It is assumed an operating time of 4800h, of which 2600h are at full load.

Figure 18 shows the simplified duration curve that constitutes the basis for the dimensioning of Ranheim heating central.

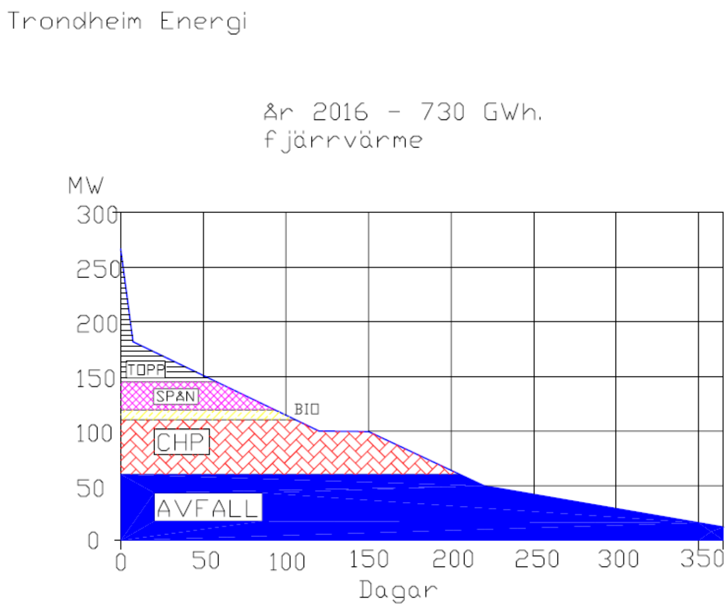


FIGURE 18: LOAD DURATION CURVE FOR RANHEIM HEATING CENTRAL[1]

2.5.4 OVERVIEW OF CHP PLANTS CONNECTED TO DH NETWORKS IN NORWAY

BIOENERGY PLANTS

In addition to the examples mentioned in the sections above, more cogeneration facilities exist. Two of them are outside the category of waste incineration plants, and these are listed in Table 3.

Solør Bioenergi in Kirkenær is running on impregnated wood waste that is grounded into wood chips. As of today, this is the only operational CHP-DH plant based on solid bioenergy in Norway.

Mosseporten Miljøenergi is, on the other hand, the only cogeneration facility connected to a DH grid that is fuelled on bio gas.

TABLE 3: EXISTING COGENERATION PLANTS CONNECTED TO DH NETWORKS RUNNING ON BIOENERGY[32, 33]

Plant	Prime mover	Fuel	Power capacity [MW]	Thermal capacity [MW]	District heating [GWh/year]	Steam production, [GWh/year]	Electricity production, [GWh/year]	PHR, design /per year
Solør Fjernvarme, Kirkenær	Steam turbine	Wood waste [2]	1,8	Boiler: 10	36	0	17	-/ 0,47
Mosseporten[3] Miljøenergi, Moss	Gas engine	Landfill gas	1,063	1,122	1,5	7	8,5	0,95/ -[4]

[2] Impregnated demolition wood that is grounded to wood chips.
[3] Estimated production numbers for 2012
[4] At Mosseporten cold energy is also produced. The PHR is therefore not calculated on an annual basis.

WASTE INCINERATION PLANTS

There are nine waste incineration plants in Norway that produce power and deliver heat to district heating networks.

TABLE 4: WASTE INCINERATION COGENERATION PLANTS. PRODUCTION IN 2010. [34-37]

Plant	Power capacity [MW]	Total energy recovery [%]	District heating [GWh/year]	Steam production, [GWh/year]	Electricity production, [GWh/year]	PHR for 2010
BIR, Bergen	20	63	173	36	65	0,31
Forus Energigjenvinning	2,8	69,4	59,6	0	8,5	0,14
Bio-el, Fredrikstad	5,68	70,1	53	49,8	14,2	0,14
FREVAR KF, Fredrikstad	1	90,5	6,8	190	0,6	0,31
EGE Klemetsrud, Oslo	22,8[5]	67	196	0	70	0,36
Eidsiva Trehørningen[6]	6,8	-	100	50	50	0,25
Returkraft[7]**, Kristiansand**	14,3	67,3	80	0	25	0,31
Senja avfallssekskap, Sørreisa	0,3	42,6	5,8	0	1	0,17
Tafjord Kraftvarme, Ålesund	5	77,5	99,1	0	14,3	0,14

In Table 4 they are all listed. One of numbers provided is the total energy recovery percentage. This number is calculated according to guidelines from Klif, and represents the ratio between the total amount of sold energy and the total amount of steam produced in the plant. Thus, the denominator in the ratio is *not* the total energy input from the waste, as it would be according to the definition of η_{tot} in equation(21).

Some of the plants have a low energy recovery percentage (<70%). This is due to the fact that they are operated based on the rate of incoming waste, and not heat or electricity demand. During low heat demand seasons, this results in that heat needs to be cooled off, which of course reduces the total energy recovery percentage.

The yearly power to heat ratio can be seen in the last column. It can be seen that this number ranges from 0,12 to 0,36. It should be remarked that the plant of Returkraft is not fully operational yet, and that the annual production is estimated to rise to 250 GWh district heating and 95 GWh electricity when the district heating network is expanded[35].

[5] 12,3MW was recently installed, in 2010 the installed capacity was 10,5 MW
[6] Opened in 2011, production figures are based on projections.
[7] Only limited operation during 2010.

PLANNED PROJECTS

As mentioned earlier, the Ranheim project is paused until a possible location is agreed upon with the municipality of Trondheim. The Eidsiva project, however, has scheduled construction start up in 2012, and aims to start producing around the end of 2013/beginning of 2014[38]. The key figures for the two projects are presented in Table 5.

TABLE 5: PLANNED CHP-DH PROJECTS[1, 31, 38]

Plant	Prime mover	Fuel	Power capacity [MW]	Thermal capacity [MW]	District heating [GWh/year]	Steam production, [GWh/year]	Electricity production, [GWh/year]	PHR, design/per year
Ranheim	Steam turbine	Wood waste/ wood chips	21,7	50	202,6	0	80	0,43/ 0,39
Eidsiva Bioenergi	Steam turbine	Wood waste[8]	4	49[9]	67	90	30	0,08/ 0,19

[8] Returtre og hogstavfall/GROT
[9] District heating plus industrial steam production

3 PRIMARY ENERGY FACTORS

As mentioned in the introduction, the primary energy factor tells us how much primary energy that is needed to supply one unit of delivered energy. To calculate the primary energy factor is, however, not straight forward. This chapter provides the framework for the calculations.

3.1 SYSTEM BOUNDARIES

According to EN 15603, the primary energy factor shall include:

"at least:

- Energy to extract the primary energy carrier
- Energy to transport the energy carrier from the production site to the utilisation site
- Energy used for processing, storage, generation, transmission, distribution and any other operations necessary for delivery to the building in which the delivered energy is used"

In addition, the primary energy factor *may* also include energy to build the transformation units, energy to build the transportation system and energy to clean up or dispose the wastes. In Figure 19, the main processes in the energy supply chain are shown along with energy input and useful energy output.

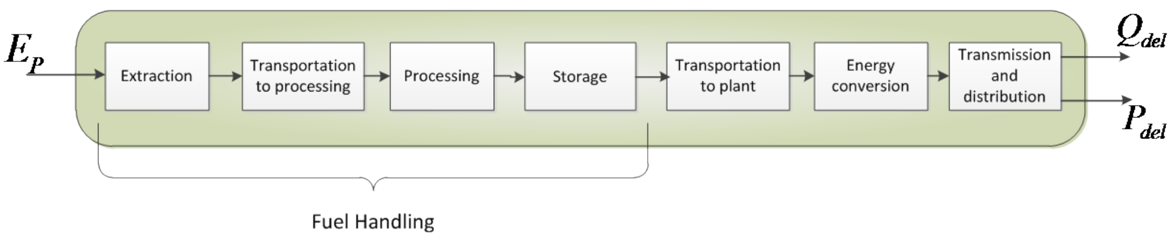


FIGURE 19: ENERGY SUPPLY CHAIN

E_P is the total amount of primary energy supplied across the system boarder, while Q_{del} and P_{del} are respectively the heat and the electricity delivered to the users.

There are two conventions for defining primary energy factors[39]:

- *Total primary energy factor*
  This conversion factor represents all primary energy input needed to supply one unit of delivered energy, both fossil and renewable. The total primary energy factor will always exceed unity.

- *Non-renewable primary energy factor*
  To get a more nuanced picture, it is possible to distinguish between renewable and non-renewable energy in the calculations. The non-renewable PEF excludes the renewable energy component of the primary energy needed, which may lead to a conversion factor that is less than unity if renewable energy resources are utilised.

3.2 THE PRIMARY ENERGY BALANCE FOR A COGENERATION SYSTEM

In a district heating system with cogeneration, the energy balance can be visualised as in Figure 20.

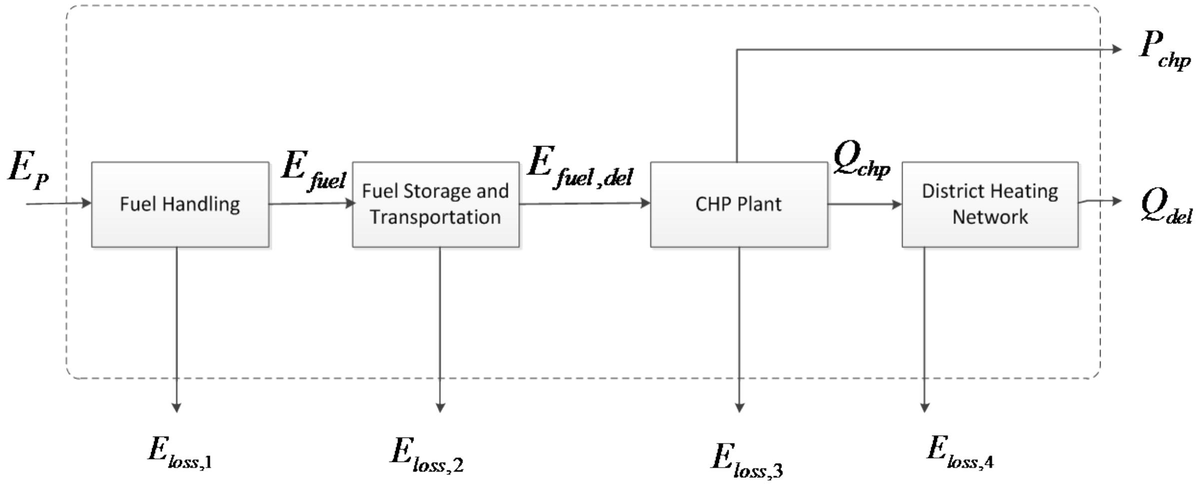


FIGURE 20: ENERGY BALANCE, DISTRICT HEATING WITH COGENERATION

When η_{chp} is the total efficiency of the plant and η_{dhn} is the efficiency of the district heating network, we get the following relations:

$$Q_{chp} = Q_{del} + E_{loss,4} = \frac{Q_{del}}{\eta_{dhn}} \tag{22}$$

$$E_{fuel,del} = P_{chp} + Q_{chp} + E_{loss,3} = \frac{(P_{chp} + Q_{chp})}{\eta_{chp}} \tag{23}$$

$$E_{fuel} = E_{fuel,del} + E_{loss,2} \tag{24}$$

$$E_P = E_{fuel} + E_{loss,1} = f_{P,fuel} E_{fuel} \tag{25}$$

$f_{P,fuel}$ is the primary energy factor for all the processes that are included in the "Fuel Handling" box, i.e. extraction, transportation to processing, storage, preparing and processing.

The total amount of primary energy needed is expressed by the help of the primary energy factors for the delivered heat and the produced electricity, $f_{P,dh}$ and $f_{P,el}$

$$E_P = f_{P,dh} Q_{del} + f_{P,el} P_{chp} \tag{26}$$

The system in Figure 19 is a simplified version of an energy supply chain, and represents the minimum of what should be included in the primary energy factor, assuming that "Fuel Processing" includes energy for extraction, processing and transport of the fuel from extraction location to processing facility.

The real system is, however, far more complex. In Figure 21, a more detailed version is presented. Here, relevant sub-processes are added, such as the construction of the CHP plant and the fuel processing facility, the production of district heating pipes and combustion additives, and the production and use of gasoline for transportation.

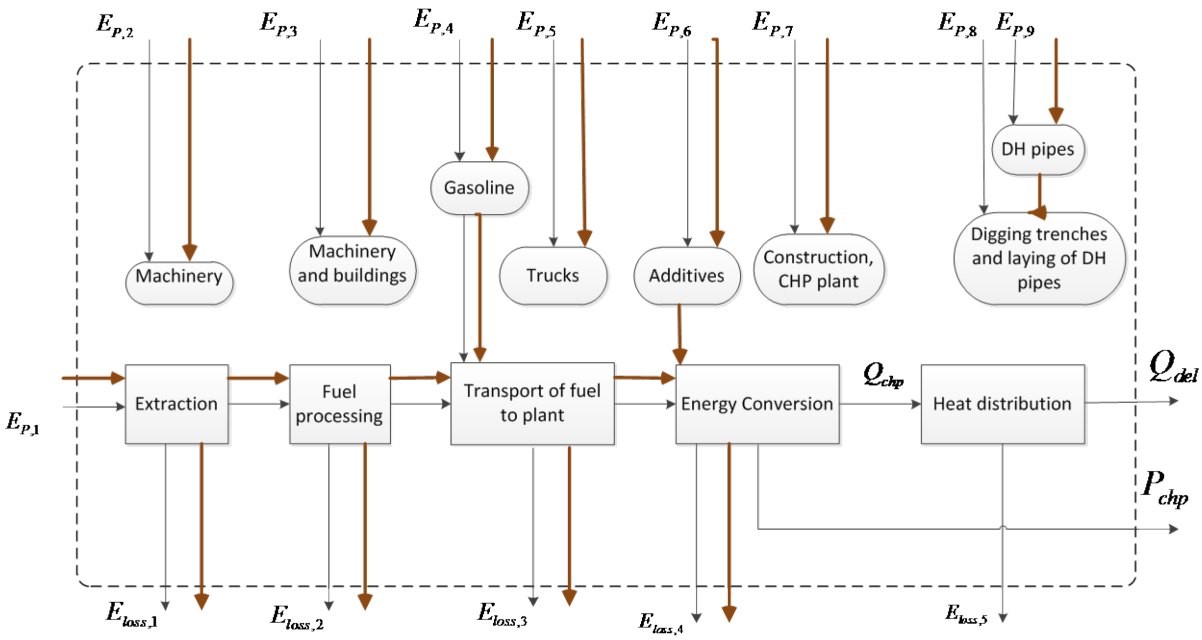


FIGURE 21: ENERGY SUPPLY CHAIN WITH SUB PROCESSES AND MATERIAL FLOWS

The thin, black arrows represent primary energy flows, while the thicker brown arrows represent material flows. It can be noted that energy is added across the system boarder to many of the processes within the system boarder, not only the extraction process. In addition, the material supplied requires energy to produce, which should also be included in the total energy requirement of the system.

The material input and output flows represent raw materials, products, waste and emissions. Upstream of all these sub processes there are even more sub processes with material and energy input and output, in the form of waste, emissions, products and energy loss. “Fuel Processing” is an example of a process that can be very complex, especially in the case of biofuels[40].

To cope with this complexity, a method is needed to assess the energy consumption related to the energy and material flows that cross the system boarder. The method used is Life Cycle Analysis, which is briefly presented in the following section.

An important part of Life Cycle Analysis is to clearly define the system boarders. As mentioned previously, EN 15603 states that some processes are mandatory to include in the primary energy factor, while some are optional. This gives room for adjustment to national conditions, but also for confusion.

The standard EN 15316-4-5:2007 contains a calculation method for obtaining primary energy factors of district heating systems. Here, the system boarders are defined so that only the compulsory elements defined in EN 15603 are included. The definitions of total and renewable factors are the same as in EN 15603, e.g. the total primary energy factor should always exceed unity.

In the calculation examples provided in the Annex, however, the total primary energy factor of district heating is less than unity. This contradicts the definition in both standards if the definition in the standard means that the total primary energy factor per energy carrier always should exceed unity. If the definition means that the total primary energy factor for the whole system should exceed unity, the PEF for the district heating might be less than one. This should, however, be stated more clearly.

In this project, the calculated primary energy factors will be presented as they are even if they turn out to be negative or less than unity. This is to make the calculations more transparent.

3.3 Life Cycle Analysis

Life cycle assessment is a standardised method (ISO 14040 series) that was developed as a tool to quantify all the environmental impacts of a product from its cradle to the grave. This means that all the phases in the product life cycle should be included: Production, use phase and product disposal. The extraction and processing of all material and energy needed in the processes involved should be included.

The method is now not only used for product assessment, but also for comparing different energy chains, services and processes.

In short, there are four steps involved when an LCA is performed, which are visualised in Figure 22.

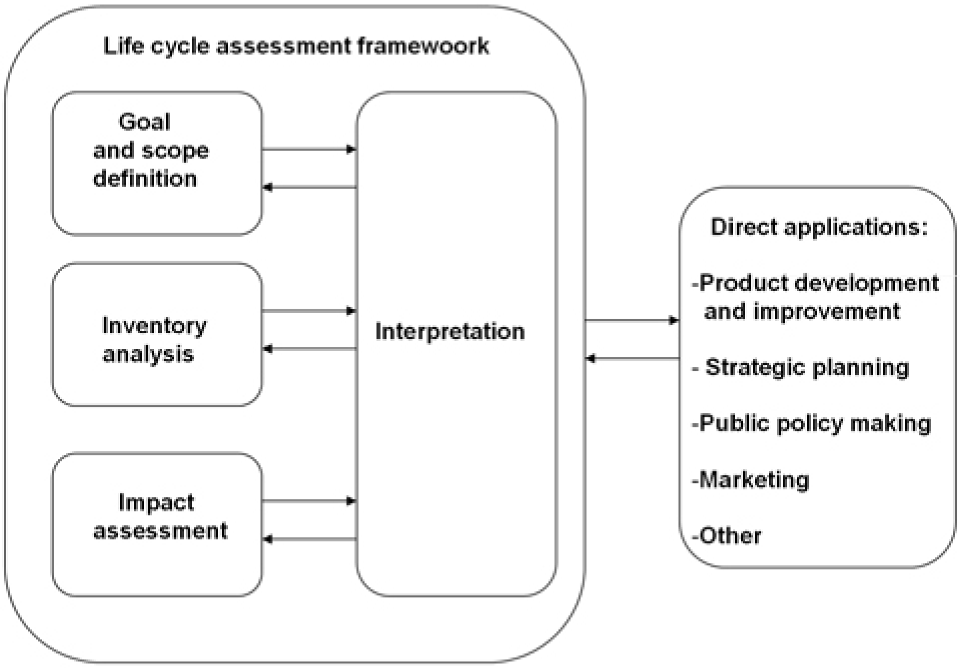


FIGURE 22: THE LIFE CYCLE ASSESSMENT FRAMEWORK

1. Goal and scope definition

   The goal and scope of the analysis should be clearly stated. This involves choosing a functional unit (FU) that is appropriate and setting the system boundaries in a way that includes all the relevant processes. Every process will, however, have input that is an output from another process, which again is supplied by other processes. The path could in theory be infinite. Normally, a cut-off rule is applied. If a flow is accounting for less than a certain percentage of the total environmental impact, it can be neglected.

2. Inventory analysis

In this step, all the inputs and outputs for all the processes involved should be mapped and quantified. If a process has more than one output, the environmental impact has to be divided between the different products. This is done by allocating emissions between the products, using one of several methods which will be further described in section 3.4.

The inventory analysis results in a matrix where all material- and energy flows are listed. These flows are then connected to the emissions related to them, and this makes the basis of the impact assessment. The data sources should be presented in a transparent way, and the credibility should be indicated. This step requires a lot of work. If a cut off rule is applied, it should be clarified what flows are neglected.

3. Impact assessment

The impact assessment involves the classification and categorisation of the different emissions resulting from the processes within the system boundaries. The classification process is done by grouping all emissions according to what impact categories they contribute to. There are two main groups of impact categories: midpoint and endpoint.

The midpoint categories represent different environmental issues, like climate change and eutrophication. The emissions that contribute to one specific category are multiplied with a scientifically decided factor, and the emissions are then added to one value representing the environmental impact of one functional unit within this environmental field. The methods used in midpoint categories for aggregating emissions and presenting results are scientifically accepted.

For instance, the scientific factor for the midpoint category “climate change” is based on the concept of global warming potential. The global warming potential represents the amount of positive radiative forcing of a gas during a certain time frame compared to CO_2. IPCC has calculated GWPs for a great number of gases, which can be found in Working Group 1’s contribution to the Fourth Assessment Report of 2007 (4AR). Within the framework of the Kyoto-protocol, only six gases should be accounted for in the national inventories: CO_2, CH_4, N_2O, HFCs, PFCs and SF_6. In this way, emissions of different gases can be aggregated into a single indicator that indicates the global warming potential.

The other main group of impact categories are the end point categories. These categories represent the damage done by the emissions at the end of their path. Examples of categories are human health and ecosystems. The aggregation of impact from different sources into these endpoint categories are more disputed. The factors used are less scientific, and the result is less transparent. On the other hand, the use of end point categories might make the results easier to understand for those the study is intended for. Hence, there exists a trade off between accuracy and communication when presenting LCA results.

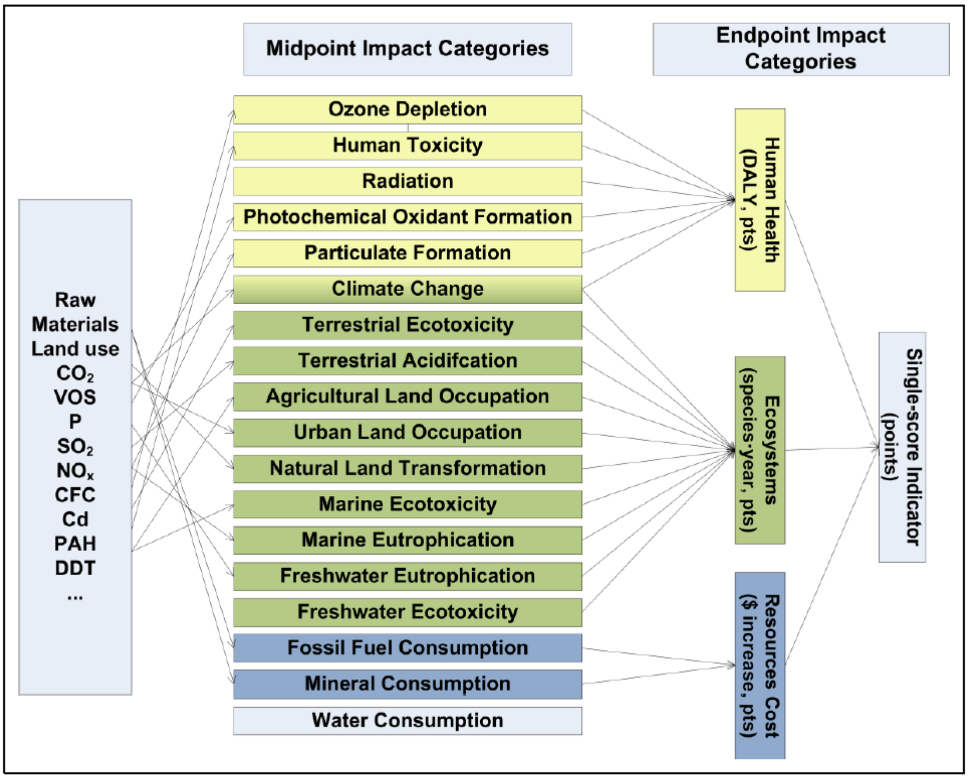


FIGURE 23:MIDPOINT AND ENDPOINT CATEGORIES. REPRINT FROM [26]

There are many different impact assessment methods. Some of them include only midpoint, some only endpoint, while others include a single score indicator of the total environmental impacts of one functional unit.

The impact assessment can also include normalisation and weighting. The normalisation is done by dividing the impact results by a reference value. This could be the impact of another product, or of a region. Weighting is done when all the impacts are aggregated into one value.

4. Interpretation

   Along the way, all choices, values and results need to be interpreted and analysed. Adjustment might be needed as new knowledge is acquired during the working process.

3.4 Multiproduct Processes and Allocation Methods

When a process has more than one product, the primary energy consumption and environmental impacts need to be divided between the different products. This can be done in several ways. The ISO standard 14040 recommends that allocation is avoided by dividing a multiproduct process into sub processes, or by system expansion. If this is not possible, it is recommended that the environmental burdens should be divided between the products using an allocation method that is based on physical properties of the products[41].

The term "allocation" is, however, normally used for all of the methods mentioned above: Dividing the process into sub-processes, system expansion and what the ISO standard refers to as physical and economic allocation. "Allocation method" will in this report refer to all the options available for dividing environmental impact between the outputs of a multiproduct process.

To illustrate the different allocation methods, a general example is used. In Figure 24, a multiproduct process with one input flow and two output flows is shown. Output 1 represents one functional unit (FU) of the main product, but from this figure it is not possible to tell how much of the environmental burden that should be attributed to each of the outputs.

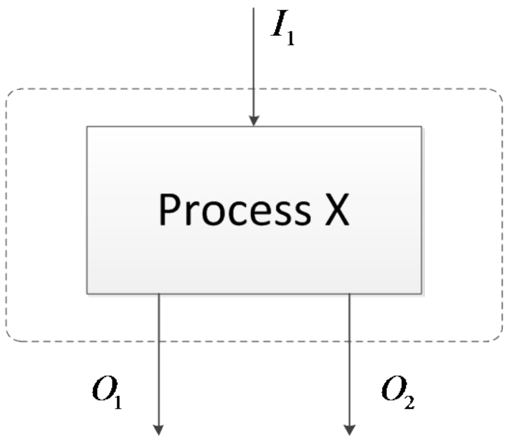


FIGURE 24: MULTIPRODUCT PROCESS

3.4.1 DIVIDING OF MAIN PROCESS INTO SUB PROCESSES

Figure 25 shows a situation where it is possible to divide the main process into two sub processes, and distinguish how much of input 1 that is needed to supply one FU of output 1. This is the best and most accurate solution, but it is seldom possible, as a multiproduct process normally is an integrated process that is not easily divided into several sub processes.

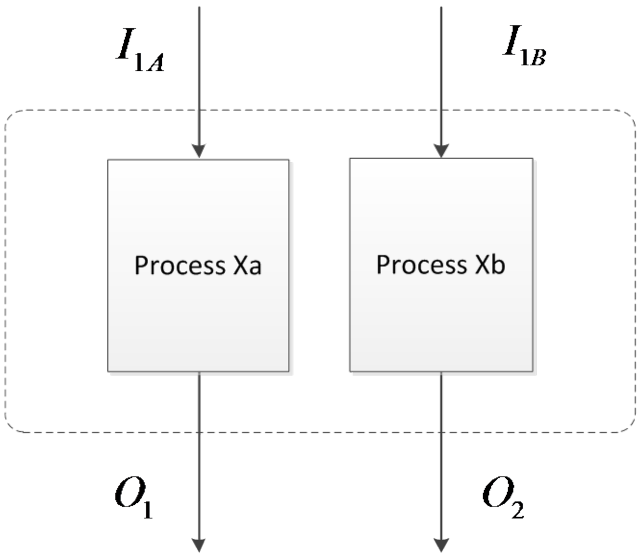


FIGURE 25: DIVIDING INTO SUB PROCESSES

If there exist separate processes that produce the same outputs elsewhere, it is possible to use data from these processes to divide the environmental impact from process X. Let us say that output 1 is heat and output 2 is electricity. These products are possible to produce in separate processes with different efficiencies. By defining two reference processes for the production of electricity and heat, it is possible to calculate the respective environmental impact for the same amounts of heat and electricity that is produced in process X. The ratio between the two separate processes can then be used to divide the environmental impact from process X between the heat and electricity produced.

3.4.2 System Expansion

In Figure 26, the system boarders are expanded.

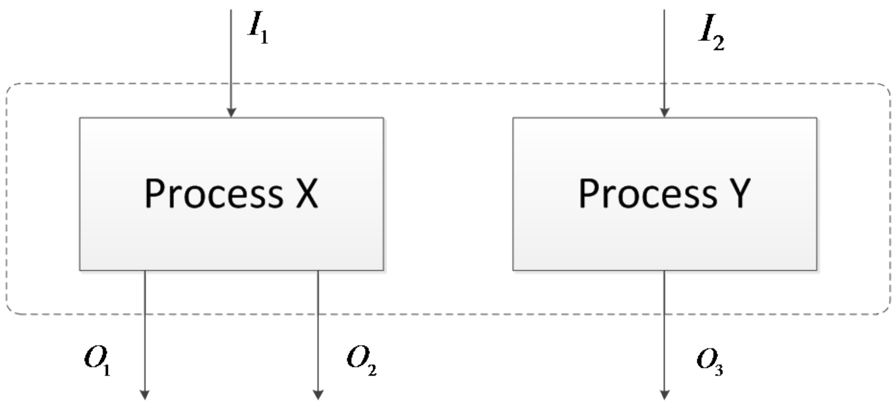


FIGURE 26: SYSTEM EXPANSION

In the initial state, Process Y is not within the system boarders, and produces a certain amount of output 3. Output 3 represents the exact same product as output 2. The concept of system expansion is based on the assumption that the need for the product that output 3 and output 2 represent is constant. When Process X is introduced, the production of output 3 in process Y is reduced with the same amount as output 2, and in this way emissions are avoided compared to the initial situation.

The avoided emissions are subtracted from the total environmental burden of Process X, and the remaining environmental burden is attributed to output 1.

3.4.3 Physical and Economic allocation

If system expansion is not possible or desirable, the ISO standard recommends that allocation is done according to physical properties of the outputs, such as mass or energy content. If this is not possible, economic value of the products can be used as basis for the allocation.

4 METHODOLOGY

The aim of the CHP plant analysis is to assess how relevant types and sizes of plants behave regarding power to heat ratio and power and total efficiency when various district heating input parameters are varied. To be able to do this, it is necessary to use adequate modelling software. In section 4.1, the technologies that have been chosen for further study are presented. In section 4.2, the modelling software and the main components of the models are introduced.

To assess how off-design operation influences the plant's annual performance, the plants are operated after a varying district heating load. In section 4.5, the calculation of district heating load is discussed.

In section 4.3 the calculation of the primary energy factors is explained in detail.

4.1 THE SELECTED TECHNOLOGIES

The chosen technologies should be relevant in the context of district heating in Norway. The most common type of combined heat and power plant in Norway today is without competition back pressure steam cycle based on waste incineration. The sizes range from 0,3 to 22,8 MW. This technology should therefore be included in the study in different sizes.

When it comes to fuel, the main aim of a waste incineration plant is to reduce the volume of the waste. The operation is based on this, and not the district heating load. As the choice of technology also is based on the possibility of studying the influence of parameters like load variations, different flow and return DH temperatures and other variables that might be of interest, waste is excluded as the main fuel. Based on the introduction of renewable energy certificates and the renewable energy directive, it is fair to assume that biofuels in various forms will be relevant as fuel. Solid wood waste is therefore chosen as main fuel for the steam cycles when part load performance is the main focus.

It would also be interesting to be able to compare the results from plants relevant in a Norwegian context with a different kind of technology. In section 2.4, the status of CHP plants in Europe where summarised, and it was clear that in the European Union, Combined Cycles are widely used.

On this foundation, the following technologies and sizes have been chosen for further studies:

- Back pressure steam cycle, 2 MWel
- Back pressure steam cycle, 10 MWel
- Back pressure steam cycle, 25 MWel
- Combined cycle, 22,7 MWel

4.2 MODELLING OF THE CHP PLANTS

4.2.1 CHOICE OF MODELLING SOFTWARE

There are mainly two types of modelling software that are applicable for the task of thermodynamic modelling of CHP plants: Fully flexible programs and application specific programs.

In a fully flexible program, the user chooses all components and links them together using streams. When the system consists of many components, this way of modelling is a demanding task, as the user needs to be truly familiar with how the system is put together, know the approximate size of the equipment and reasonable initial values and settings for the components in question. This approach is useful if the aim is to study a simplified process or if the model is based on a specific plant layout that is available to the modeller. It can also be a good option if the goal of the modelling is to evaluate different sizing of one component or to assess how the introduction of an additional component influence the system. Hysys, Aspen and PRO II are all examples of fully flexible simulation tools.

Application specific software is made so that the user can model complex processes without the detailed knowledge of the exact attributes of all the included components or the connections between them. Reasonable assumptions and initial values are set by the program based on experienced values, and if the user knows that these values are different in his case, they can be changed. User input is automatically cross checked and validated, so that inconsistencies and software crashes are less likely than in a fully flexible program.

The use of application specific software makes it possible to model larger and more realistic systems in much shorter time and with less likelihood of errors than with fully flexible programs. It is less likely that mistakes and unrealistic assumptions are made, as reasonable default values are provided. At NTNU, GT Pro and STEAM Pro by Thermoflow are the most used application specific modelling programs.

The disadvantage is of course that application specific software contains a limited choice of system layouts. If the user wants to model a plant layout that is not included in the program, this is simply not possible.

In this project, the best from both worlds are combined in the use of the software package from Thermoflow. The plants are designed using GT Pro and STEAM Pro. In this process a variety of input data is provided to the program and based on this information a realistic plant layout with properly sized and initialized components are created.

The plant model is then imported into Thermoflex, which is a fully flexible program that is compatible with models created in GT Pro and STEAM Pro. This makes it possible to adjust the model without building it from scratch. The Thermoflow program package is mainly designed with power production in mind. There are some pre-made district heating options, but these are not made for simulations of CHP plants that have heat as their main product. By adapting the model from STEAM Pro or GT Pro in Thermoflex, it is possible to control more variables, and thus run the cases that are interesting in a district heating point of view.

4.2.2 CALCULATION MODES

There are two main calculation modes available in the Thermoflow programs: Design and off-design. In design mode, thermodynamic criteria dictate the size of the components. In off-design mode, it is possible to observe the thermodynamic performance of the fixed system under varying conditions.

The design mode is in this context useful to study different technologies, fuels and design temperature levels.

Off design mode is used to study how the different systems perform when the district heating load, and supply and return temperatures vary throughout the year.

As the main aim of the Thermoflow software package is to study power production, it required a lot of “manual” work to obtain the results on off design performance for varying district heating load.

4.2.3 MAIN COMPONENTS AND INPUTS

The models are constituted of different components that are connected by streams. There are many input choices, and the most relevant related to the components used are described here.

BOILER

STEAM Pro offers an advanced boiler model that is based on the choice between the following three alternatives:

Conventional boiler

> If this alternative is chosen, the boiler is designed after the pattern of a pulverised coal boiler. The solid fuel is grinded into small particles, and combustion takes place in radiant furnace enclosure at a high temperature.

Circulating fluidised bed boiler

> This alternative initialises a boiler design employing a circulating fluidized bed. This combustion technology involves a bed substance that consists of small particles of an incombustible matter, for instance limestone. These particles mix with the fuel and contribute to mix fuel and air. This leads to a more complete combustion process, and a more even temperature in the combustion chamber. The flue gas is then lead into a cyclone where the bed particles are separated from the rest of the flue gas. The bed particles are then reintroduced into the fluidised bed.

Grate boiler

> Grate boilers are a part of the boiler technologies with a fixed combustion bed. This is useful for combustion of heterogeneous fuel, and fuel that is no easily grinded to small particles. It exist many types of grate boilers, with fixed, moving, horizontal or inclined grate. In STEAM Pro, the grate boiler is modelled as a horizontal moving grate boiler. The combustion takes place on and above the grate, with primary air supplied from below through the grate, and secondary air supplied from above.

When boiler type was chosen for the different steam cycles, this was done based on the boiler size that is needed to supply the turbine. When solid biofuel is considered, grate fired boilers are normally used for sizes up to approximately 40 MW_{th}, while CFB boilers are used for boilers from 15 MW_{th} and upwards[42].

Grate boilers were therefore chosen for the 2 MW_{el} and 10 MW_{el} steam cycle plant, while a circulating fluidised bed (CFB) boiler was chosen for the 25 MW_{el} plant.

STEAM TURBINE

There are four different main choices regarding steam turbines in STEAM Pro

- Non condensing back pressure
- Non reheat condensing
- Single reheat condensing
- Double reheat condensing

The back pressure and non reheat condensing turbine can be seen in Figure 27. The only difference for the user will be that the exhaust pressure of the condensing turbine is set by the program, while for a back pressure turbine this can be chosen by the user.

From a modelling point of view, there are differences between the condensing and the back pressure steam turbine. A steam turbine will choke if the volumetric flow rate gets too high and the steam reaches sonic velocity. In the Thermoflow programs, this is of course always taken into account in design mode, but in off design mode, this effect is only checked for if one uses the condensing turbine model[43]. However, in this project the cycles are only run below the design point, and the use of the back pressure turbines at off design is therefore not a problem.

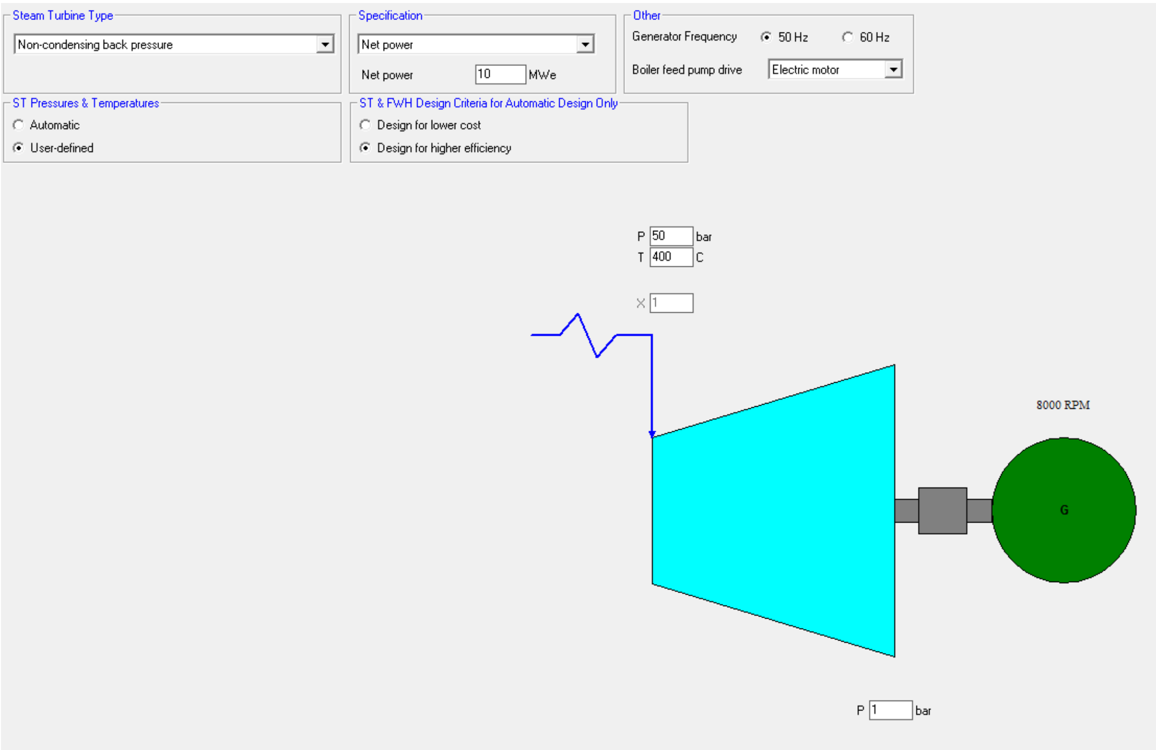


FIGURE 27: BACK PRESSURE TURBINE, MAIN INPUT SCREEN FROM STEAM PRO

Figure 28 shows a single reheat condensing turbine. Here, the steam is reheated between two of the turbine stages to obtain a higher efficiency. Single and double reheat arrangements are mainly used in larger power plants, and in STEAM Pro it is recommended if the power output is above 80 MW.

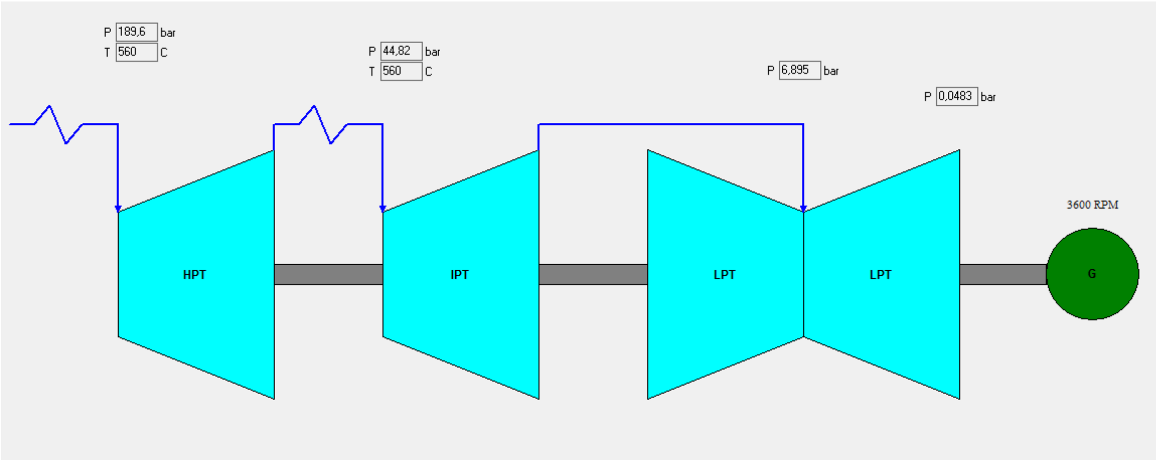


FIGURE 28: SINGLE REHEAT CONDENSING TURBINE, SCREEN SHOT FROM STEAM PRO

In this thesis, the steam cycles have been modelled using back pressure turbines.

Several choices need to be made during the design phase. Below some of the alternatives are listed, with the ones that were actually used written in bold:

- Main specification
  - Gross power output
  - **Net power output**
  - High pressure turbine steam flow
  - Low pressure turbine outlet exhaust flow
- Steam turbine (ST) pressures and temperatures
  - **Automatic**
  - User defined: This option makes it possible for the user to define input pressure and back pressure, in addition to the temperatures in the feed water heaters train.
- Steam turbine and feed water heater (FWH) design criteria
  - Design for lower cost
  - **Design for higher efficiency**
- Generator frequency
  - **50 Hz**
  - 60 Hz

In addition there are many more advanced input options. It is possible to adjust pressure control strategy within the steam turbine, the steam seal system, steam turbine leaks, design assumptions regarding losses and stage efficiencies, as well as assumptions regarding pumps and boiler sizing. These assumptions were left in their default setting, as it would have been outside the scope of this thesis to review all of them in detail.

Figure 29 shows the Thermoflex symbol of steam turbine with four groups with steam extraction at two different pressure levels. One group contains a varying amount of stages, which consist of a stationary blade and a rotating blade.

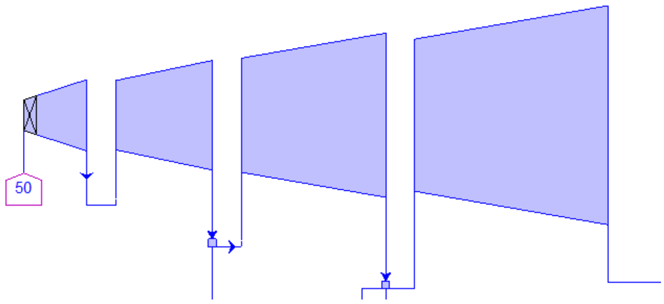


FIGURE 29: THERMOFLEX SYMBOL FOR A STEAM TURBINE WITH CONTROLLED INLET PRESSURE, 4 STAGES AND TWO BLEED PRESSURES

District Heater

The district heater component is a shell and tube heat exchanger with three different alternatives for parameter control in design mode:

- *"Exit T limited by source P, district heater sets steam flow & network sets water flow".*
  If this option is chosen, the exit temperature is decided by the incoming steam pressure, which is calculated exogenously from the district heater. The district heating water flow is also set by the network, which means that this value is calculated somewhere else in the system. With the district heating flow and inlet and exit temperatures known, the district heater imposes a steam rate from the network that comply with the district heating load.
- *"Exit T sets source P, district heater sets steam flow & network sets water flow".*
  This option lets the district heater decide the pressure of the steam, so that the desired exit temperature can be reached. This is an useful option if the district heater is connected to a steam bleed. The district heating water flow is imposed by the network, and the heat load is served by dictating the steam mass flow of the extracted steam.

  This option was chosen on the district heaters that was coupled to the steam bleed.

- *"Exit T sets source P, network sets steam flow and district heater sets water flow".*
  Like above, source pressure is dictated by the district heater, but in this case the steam flow is set by the network. When the district heater is coupled to the exhaust of a back pressure turbine, this option is the one that is used. The district heater is in this case also the condenser of the steam cycle, so the district heating water flow is therefore adjusted so that the condensing heat of the steam is carried away.

  This option was chosen on the district heaters that were coupled directly to the exhaust from the steam turbine.

In off design mode, the control options are a bit different. For the district heaters that were directly coupled to the exhaust of the steam turbine, the option chosen was "Network sets steam flow and district heater sets water flow and source pressure". Choosing this alternative, it is possible to set the district heater water flow as an input value, and the outlet temperature is then calculated by the program.

The district heaters that were coupled to the extraction steam, were run on the control option “Valve modulated to control water temperature”. In this mode, a valve is introduced at the steam inlet of the heat exchanger, allowing control over the outlet temperature of the district heating water.

The Thermoflex symbol of the district heaters can be seen in Figure 30.

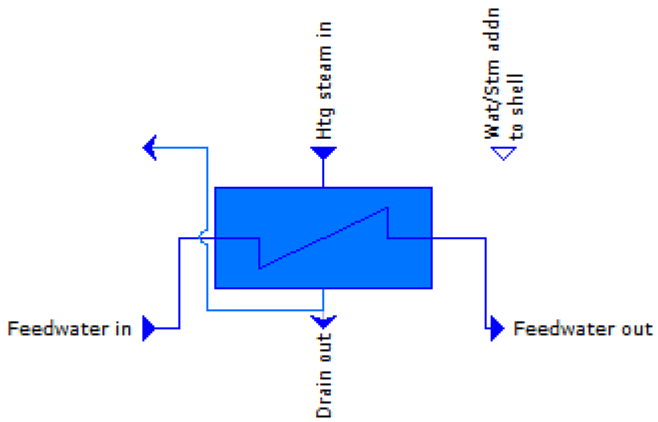


FIGURE 30: DISTRICT HEATER, THERMOFLEX SYMBOL

GAS TURBINE

In GT Pro, it is possible to choose from wide range of gas turbines that are based on real models. The choice of gas turbine will in reality rely on many aspects: Desired electrical output, efficiency versus price evaluations, hours per year at full load operation, part load operation performance, compatibility with existing equipment, availability etc.

In this project, a high efficiency engine from General Electric was chosen, the LM1800e. This choice was made after sorting the engines available after size and efficiency. In the desired size range, the LM1800e was the engine with the highest efficiency. The product description at GE’s web paged stated that it is an “Ideal choice for power generation, specifically cogeneration and CHP applications”, and that it has “Best in class efficiency from 16,5 – 18 MW” [44]. More details on the gas turbine is found in appendix A.

In Figure 31, the gas turbine symbol of Thermoflex can be seen.

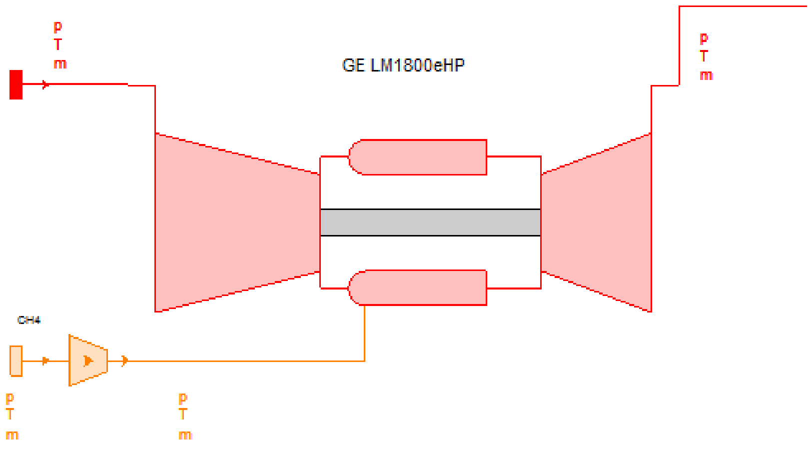


FIGURE 31: GAS TURBINE WITH FUEL SUPPLY AND FUEL COMPRESSOR, THERMOFLEX SYMBOL

Heat Recovery Steam Generator

The heat recovery steam generator consists of different heat exchangers that transfer the heat from the gas turbine flue gas to the steam cycle working fluid. The design of the HRSG was done by GT Pro and Thermoflex, and no changes were made to the initial assumptions.

Figure 32 contains the HRSG symbol used in Thermoflex. The blue streams are water and steam streams, while the red stream represents the flue gas from the gas turbine.

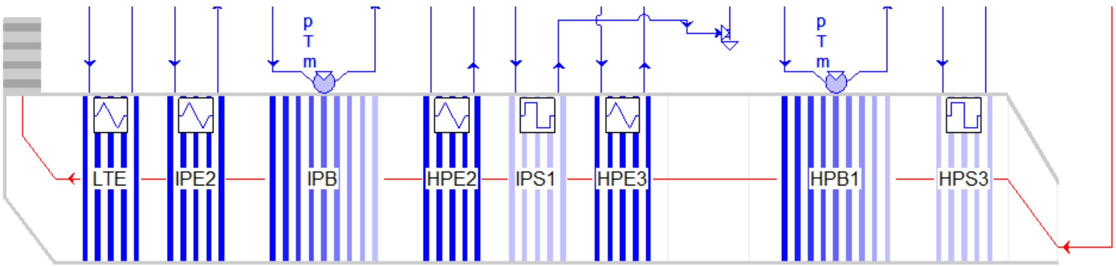


FIGURE 32: HEAT RECOVERY STEAM GENERATOR, THERMOFLEX SYMBOL

Pumps and Valves

As for pumps, the default multistage centrifugal pump with an isentropic efficiency of 75% was not altered.

Figure 33 shows the Thermoflex symbols for pumps and valves.

Deaerator

Deaerators are used to remove dissolved gases from the working fluid of a steam cycle. This is done by heating the working fluid to saturation temperature, after which the emancipated gases are vented away. The deaerator symbol can be seen in Figure 34.

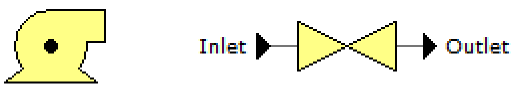


FIGURE 33: PUMP AND VALVE, THERMOFLOW SYMBOL

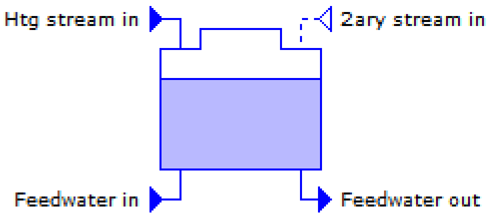


FIGURE 34: DEAERATOR, THERMOFLEX SYMBOL

Feed Water Heater

The feed water heater is a shell and tube heat exchanger that is used to preheat the condensed working fluid of the steam cycle before it is fed into the boiler. The STEAM Pro software uses a feed water heater with a small condensate pump, which can be seen in Figure 35.

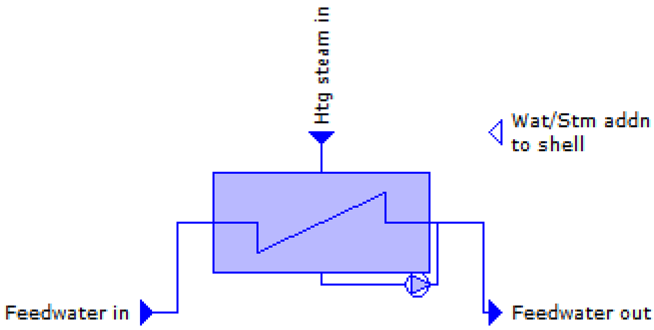


FIGURE 35: FEEDWATER HEATER WITH PUMP, THERMOFLEX SYMBOL

Splitters and Mixers

Splitters and mixers are used to respectively separate one stream into two or more, or to mix two or more streams into one stream. Mixing of streams at different temperature levels causes exergy losses in the system. The symbols for splitters and mixers can be seen in Figure 36.

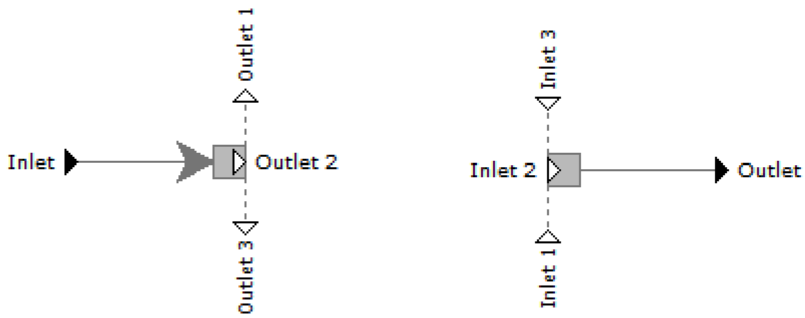


FIGURE 36: SPLITTER AND MIXER, THERMOFLEX SYMBOL

MAKEUP AND BLOWDOWN

In the Rankine cycle, the working fluid cycle is idealised as a closed loop. In reality, some of the water will replaced with makeup water during operation. This is modelled using the makeup and blowdown component. In addition, closed loops represent a challenge in the calculation of models, and another important task for this component is simply to break the loops. The makeup/blowdown component is shown in Figure 37.

FIGURE 38: WATER SOURCE AND WATER SINK, THERMOFLEX SYMBOL

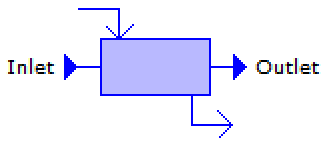


FIGURE 37: MAKEUP AND BLOWDOWN, THERMOFLEX SYMBOL

WATER SOURCE AND SINK

The district heating water network was modelled by using a water source and a water sink. The symbols are shown in Figure 38.

4.3 Plant Descriptions and Flow Sheets

4.3.1 The steam cycle based CHP plants, STEAM Pro and Thermoflex

The initial design of these plants were done in STEAM Pro with the following input in addition to the ones explained in paragraph 4.2.3.

- Ambient temperature 6 °C, which is close to the average temperature in Trondheim and Oslo[45]
- Back pressure 1 bar

Before the design-simulation, "Net power" was chosen as specification parameter in the main steam turbine input screen, and the desired amount of electric power was chosen: 2, 10 or 25 MW_{el}.

It should be noted that pump work to drive the district heating water is regarded as outside the system boarder of the CHP plant, and is therefore not included in any of the cases.

Models Used for Part Load Simulations

To be able to assess the part load performance of the CHP plants, the STEAM Pro model was imported into Thermoflex. There, some adjustments were made. District heaters were added and coupled to the extraction port in the steam turbine, and makeup/blowdown components and pumps were added if needed. After these modifications, a new round of design simulations was made with the desired return and supply temperatures for the district heating water. The design simulation defined the maximum district heating load, and the part loads were calculated based on this. Based on the part loads and desired temperatures, the necessary district heating mass flow was calculated. Then, the Thermoflex model was put into off-design mode, and by using the multiple simulations function and the control loop function, the fuel input was adjusted so that the district heating load was satisfied.

Even though each of these models had to be created separately, the flow sheets are similar. The Thermoflex flow sheet of the 10 MW plant steam cycle is shown as an example in Figure 39. The boiler flow sheet for both the grate boiler and the circulating fluidised bed boiler is found in appendix A.

Models Used for Simulations Studying Fuel Choice and Design Temperatures

To study the effect of different fuel and design temperatures in the district heating system, STEAM Pro was used alone, as off design simulations were not needed. The same input values were used and the same assumptions were made as in section 4.3.1, with one exception. Now the built-in district heating opportunity was used.

It was, however, not possible to use the multiple run function in an efficient way, as the bleed pressure had to be adjusted manually every time the temperatures of the district heating was changed.

The flow sheet for this model is shown in Figure 40.

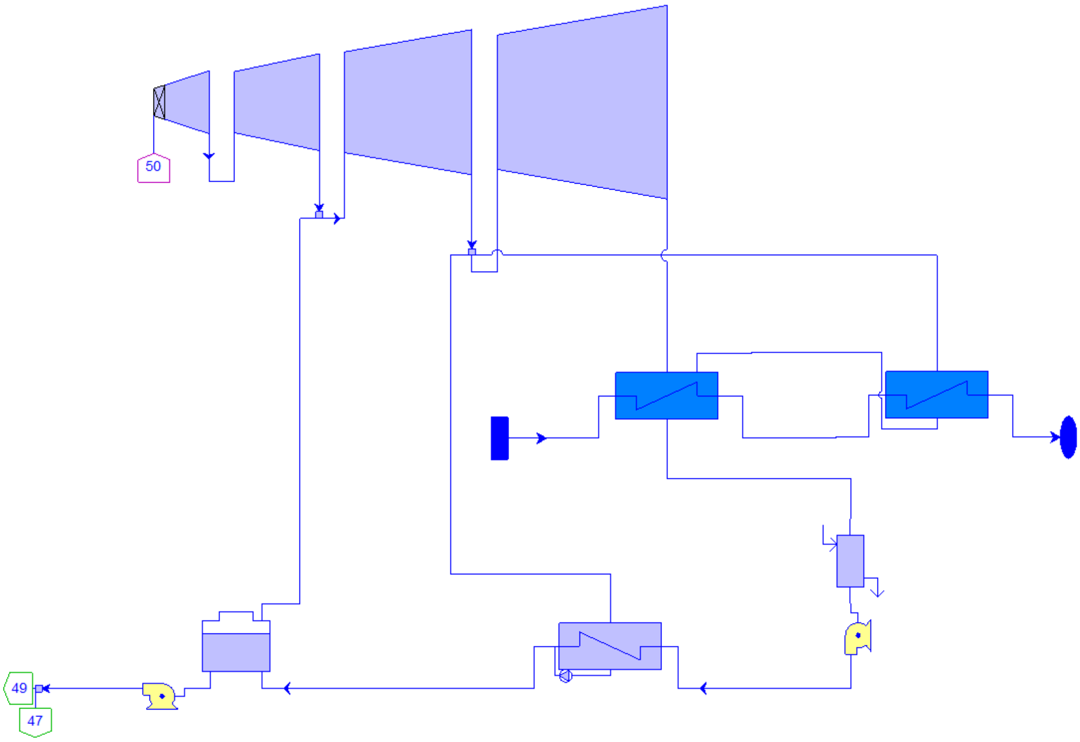


FIGURE 39: STEAM CYCLE CHP PLANT, FLOW SHEET FROM THERMOFLEX

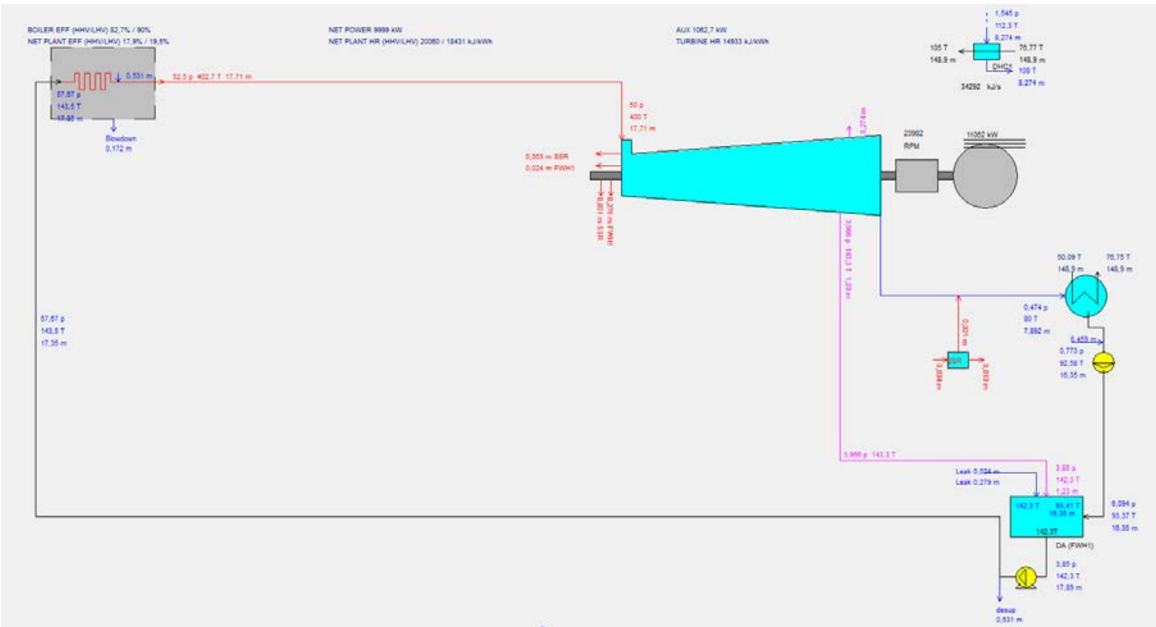


FIGURE 40: STEAM CYCLE CHP PLANT, FLOW SHEET FROM STEAM PRO

4.3.2 THE COMBINED CYCLE PLANT

The main design of the combined cycle plant was initialised using GT Pro. The ambient temperature was chosen to be 6 °C, and a district heating system similar to the one coupled to the steam CHP plants was chosen. The gas turbine used was the GE 1800e, as described in paragraph 4.2.3. The fuel utilised was CH_4.

Combined cycle with district heating, GT Pro

In Figure 41, the flow sheet from GT Pro can be seen. This model was used for simulating different design temperatures for the district heating network. Note that the flow sheet is a bit confusing in the sense that the condenser that is coupled directly to the exhaust of the steam turbine actually is the same as the bottom heat exchanger marked "COND" in the district heating system on the left in the figure.

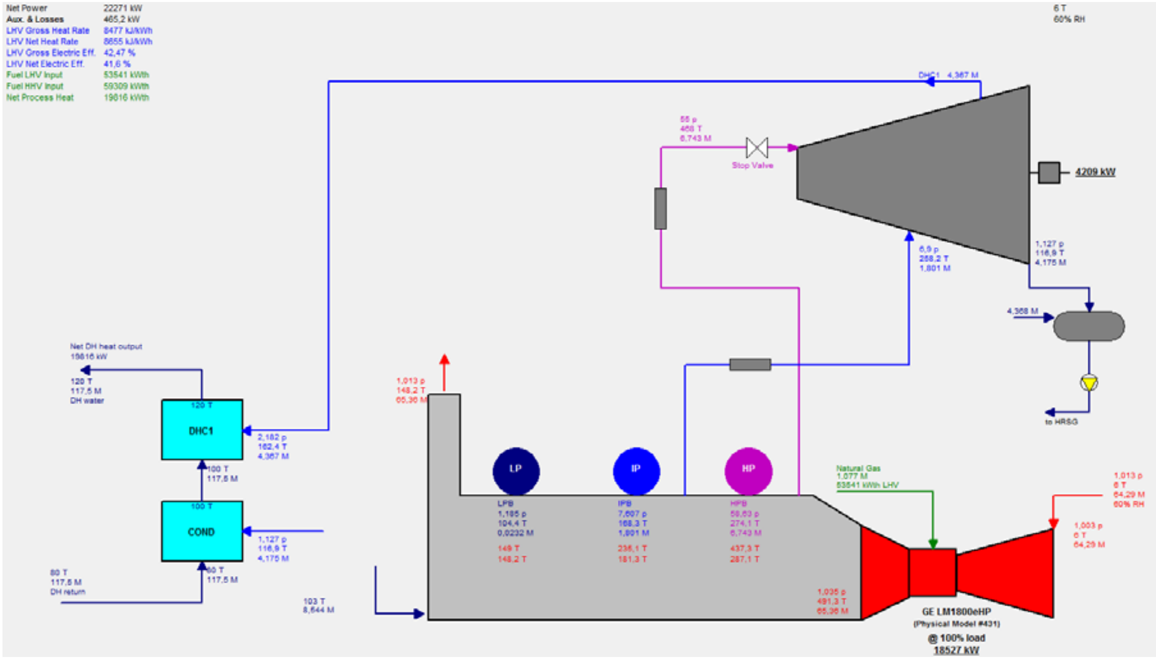


FIGURE 41: COMBINED CYCLE CHP PLANT WITH DISTRICT HEATING, FLOW SHEET FROM GT PRO

COMBINED CYCLE WITH DISTRICT HEATING, THERMOFLEX

To study the off design performance of the CHP plant, it was necessary to import the GT Pro model into Thermoflex and do the simulations there. To be able to control the district heating temperatures and flow, the condenser was replaced by a district heater, and a pump was added. The flow sheet is shown in Figure 42.

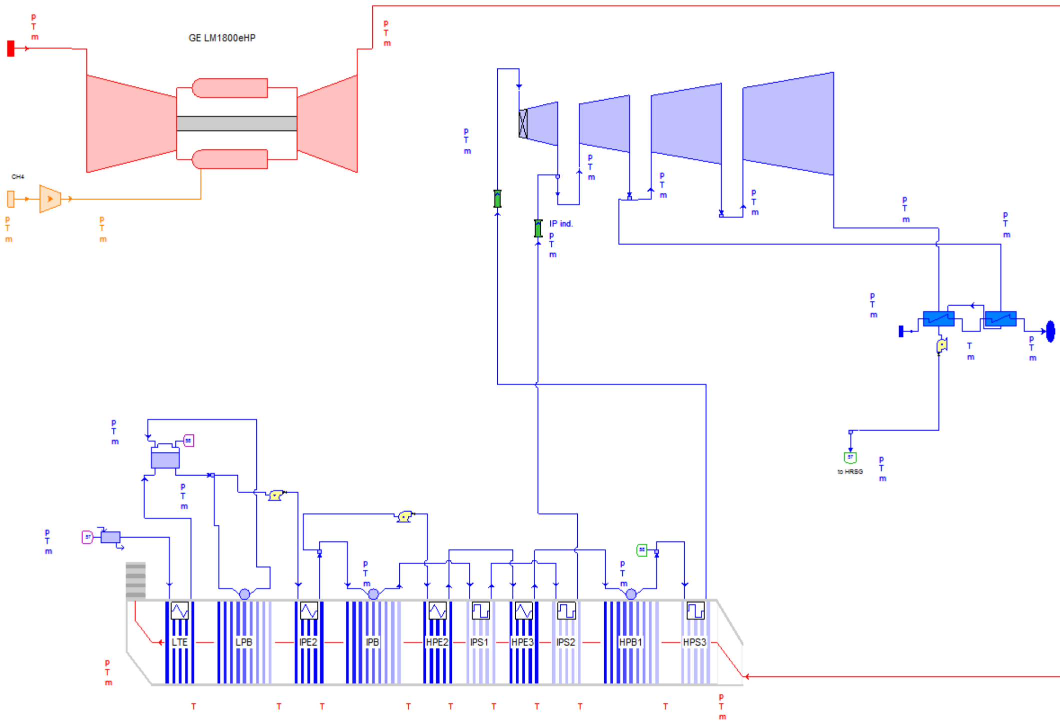


FIGURE 42: COMBINED CYCLE, THERMOFLEX MODEL

4.4 THE PRIMARY ENERGY FACTOR CALCULATION TOOL

To calculate the primary energy factors, a calculation tool developed by [16] was used. This is an excel based implement where all the main components in the district heating energy chain have been taken into consideration. The original layout is shown in Figure 43, and the following section gives a short description of the model, regarding system boundaries, functional unit, energy balance and the different processes included.

Some modifications were done to the original model, and these are summarised in paragraph 4.4.8.

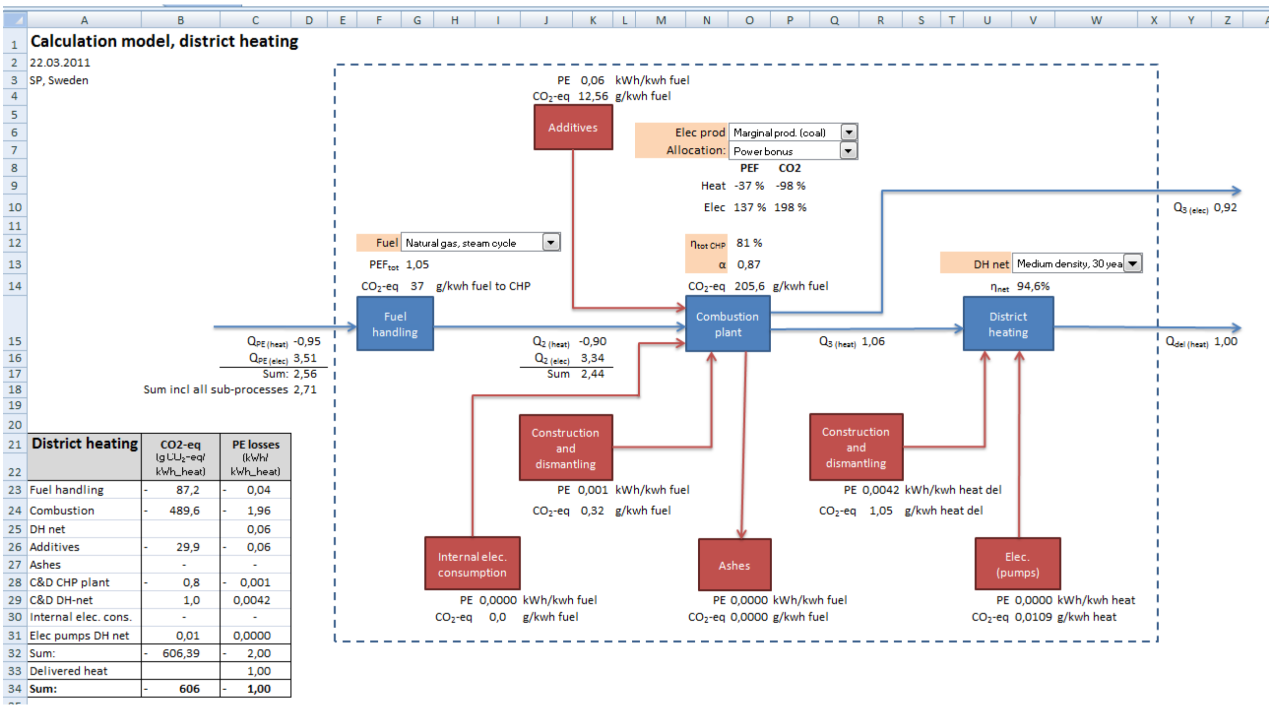


FIGURE 43: PRIMARY ENERGY FACTOR CALCULATION TOOL DEVELOPED BY [16], SCREENSHOT OF ORIGINAL MODEL

4.4.1 SYSTEM BOUNDARIES AND FUNCTIONAL UNIT

The functional unit of the model is 1 kWh of delivered energy by district heat. The system boundaries are set so that the whole energy supply chain plus some additional processes are covered. The blue flows and processes represent the main processes and the red flows and processes represent sub processes.

4.4.2 THE FUEL HANDLING PROCESS

This process includes extraction of raw materials, processing and transportation of the fuel to the CHP plant. The numbers are calculated using Simapro.

4.4.3 THE COMBUSTION PLANT PROCESS

The combined heat and power plant itself is simplified as a black box process with a total efficiency of 81% and a power to heat ratio of 0,87. In this project, the results from the CHP plant simulations done in the Thermoflow programs were implemented in a drop down menu.

Four sub processes are related to the combustion plant:

- **Additives**
  To obtain the desired combustion characteristics, various chemical additives are used during incineration. The necessary amount of six different additives is based on data from Swedish

and Korean facilities. Simapro was used to quantify primary energy need and greenhouse gas emissions related to the production of these additives.

- **Construction and dismantling**
  This sub process comprises estimated primary energy consumption and GHG emissions for construction and dismantling of the CHP plant. The numbers are again calculated by the use of Simapro, and the material inventory is based on the material requirements for a 100 MW plant with and expected lifetime of 30 years.

- **Ashes**
  This sub process includes transport to the deposit site for the ashes.

- **Internal electricity consumption**
  In this process, the user can include internal electricity consumption in the plant. This is, however, already taken into account in the CHP plant results calculated in Thermoflow. In all the results, net electric output is used, which is defined as gross electricity production minus auxiliary loads. The value of internal electricity consumption was therefore maintained at the default value, which is zero.

4.4.4 THE DISTRICT HEATING PROCESS

In the district heating process, heat loss from the pipes, pump electricity consumption and construction and dismantling of the pipes are the main elements. Pipe construction and dismantling and pump electricity consumption are in the model separated from the main process as two sub processes.

- Construction and dismantling
  In addition to the material and energy required for the construction and dismantling of the pipes, this process also includes an estimate of energy and material consumption related to excavation of trenches and laying of pipes. The final values for primary energy consumption and GHG emissions are given for different levels of energy density per meter pipeline and for different assumptions regarding the lifetime of the pipes. The composition of pipe dimensions is based on an existing DH network.

- Pump electricity
  In a district heating network, one or several pumps are needed to circulate the water to the farthest customer and back. The work required is proportional to the head loss, which is a function of mass flow, viscosity, pipeline diameter, roughness and water density. Viscosity and density are temperature dependent, and the supply and return temperatures will thus influence the pump work. The supply and return temperatures also define the temperature difference, which again defines the DH water mass flow. The pump work is calculated for a given heat load curve and temperature profile per MWh heat delivered by the district heating network for three different levels of energy density. The pump has an assumed efficiency of 85%.

The heat loss from the pipes is calculated based on the same composition of pipe dimensions and supply and return temperatures as in "Construction and Dismantling" and "Pump electricity". The user has the possibility to choose energy density and expected life time of the pipes. Based on this, the heat loss, the pump electricity consumption and the construction and dismantling numbers change accordingly. Further details regarding the calculations and values in paragraph 4.4.2, 4.4.3 and 4.4.4 are found in [16].

4.4.5 Energy Balance Equations

The equations are based on the flow sheet in Figure 43.

Q_{del} is the functional unit, and will therefore always be equal to one.

$Q_{3(heat)}$ is the amount of heat that is it necessary that the CHP plant produces to be able to deliver 1 kWh of heat to the user, and is calculated based on the efficiency of the district heating network:

$$Q_{3(heat)} = \frac{Q_{del(heat)}}{\eta_{dh}} \tag{27}$$

The corresponding amount of electricity produced is found by using the power to heat ratio:

$$Q_{3(elec)} = \alpha Q_{3(heat)} \tag{28}$$

The necessary amount of energy input to the CHP plant is then found when dividing the production by the total efficiency:

$$Q_2 = \frac{Q_{3(elec)} + Q_{3(heat)}}{\eta_{chp}} \tag{29}$$

The total amount of primary energy input necessary to supply one FU is in the excel sheet marked as "Sum incl. all sub processes", and is here referred to as $Q_{1(tot)}$. The number after "Sum" denotes the primary energy requirement when sub processes are excluded, and are here referred to as Q_1.

$$Q_1 = f_{P,fuel} Q_2 \tag{30}$$

$$Q_{1(tot)} = Q_1 + E_{P,sub} \tag{31}$$

Where $E_{P,sub}$ is the primary energy requirement of the sub processes per FU.

To be able to calculate the primary energy factor of the delivered district heating, the total primary energy input needs to be allocated between the heat and the electricity produced.

4.4.6 Allocation Methods

The calculation tool has five different allocation methods to choose from.

Energy Allocation

In this case, the primary energy consumption is allocated between heat and electricity according to how much heat and electricity that are produced in the CHP plant. This allocation method is an example of physical allocation, as explained in paragraph 3.4.3.

$$A_{heat} = \frac{Q_{3(heat)}}{Q_{3(heat)} + Q_{3(elec)}} \tag{32}$$

$$A_{el} = 1 - A_{heat} \tag{33}$$

The corresponding primary energy factors then become:

$$f_{P,dh} = \frac{A_{heat} Q_{1,tot}}{Q_{del(heat)}} \tag{34}$$

$$f_{P,el} = \frac{A_{el} Q_{1,tot}}{Q_{3(elec)}} \tag{35}$$

Exergy Allocation

This allocation method takes the quality of the energy produced into account, as the allocation factors are based on the exergy content of the heat and electricity produced. This method is also an example of a physical allocation method.

Electricity consists of 100% exergy. The exergy accompanying heat transferred at a constant temperature T_H, can be calculated using equation (5):

$$\mathbb{E}_Q = Q\left(1 - \frac{T_a}{T_H}\right)$$

When the heat is transferred at a sliding temperature, however, this equation is not valid. It can be shown that the temperature T_H then can be replaced by the logarithmic mean temperature of the temperatures at which the heat is transferred. In the case with district heating, these temperatures are represented by the supply and return temperatures.

$$T_m = \frac{(T_S - T_R)}{\ln\left(\frac{T_S}{T_R}\right)} \tag{36}$$

Then, the exergy content of the produced district heating can be found by replacing T_H with T_m:

$$\mathbb{E}_Q = Q_{3(heat)}\left(1 - \frac{T_a}{T_m}\right) \tag{37}$$

T_a is chosen to be 6 °C, and T_S and T_R are calculated based on the yearly load distribution and the corresponding supply and return temperatures.

The allocation factors can then be calculated:

$$A_{heat} = \frac{\mathbb{E}_Q}{Q_{3(elec)} + \mathbb{E}_Q} \tag{38}$$

$$A_{el} = 1 - A_{heat} \tag{39}$$

The primary energy factors are then calculated according to equation (23) and (24).

Alternative Production Method

The alternative production method is an allocation method where the environmental burdens are distributed according to how it would be if the heat and electricity were to be produced separately. The CHP process is thus divided into two sub processes with different efficiencies, and is an example of the allocation method described in paragraph 3.4.1.

$$A_{heat} = \frac{\frac{Q_{3(heat)}}{\eta_{alt.heat}}}{\frac{Q_{3(heat)}}{\eta_{alt.heat}} + \frac{Q_{3(elec)}}{\eta_{alt.elec}}} \tag{40}$$

Where $\eta_{alt.heat}$ is the efficiency of the alternative heat production and $\eta_{alt.elec}$ is the efficiency of the alternative electricity production.

The allocation factor for electricity and the final primary energy factors are calculated according to equations (22) to (24).

Power Bonus Method

This method uses the principle of system expansion and avoided impact, as explained in paragraph 3.4.2. The produced heat is regarded as the main product, while the electricity is a bonus that would have been produced elsewhere if the heat was not produced. This is the method that should be utilised when calculating primary energy factors for district heating according to EN 15316-4-5:2007.

Since the electricity produced is regarded as a bonus, the primary energy that would have been used to produce this electricity elsewhere can be subtracted from the primary energy input to the CHP process. This requires knowledge of the primary energy factor for the electricity production that is assumed to be avoided.

The primary energy factors are in this case calculated the following way:

$$f_{P,el} = f_{P,alt.el.production} \tag{41}$$

$$f_{P,dh} = \frac{Q_{1(tot)} - Q_{3(elec)} f_{P,el}}{Q_{del(heat)}} \tag{42}$$

The allocation factor then becomes:

$$A_{heat} = \frac{f_{P,dh} Q_{del(heat)}}{Q_{1(tot)}} \tag{43}$$

The allocation factor for electricity is then calculated according to equation (22)

200% Method

The 200% method is not based on a physical relation, the dividing of a main process into sub processes or the assumption of avoided impact. The allocation factor is calculated according to the following equation:

$$A_{heat} = \frac{Q_{3(heat)}}{2 \cdot Q_2} \tag{44}$$

The allocation factor for electricity and the corresponding primary energy factors are then calculated according to equations (22) to (24).

4.4.7 Electricity Production Mix

When the Power Bonus method is utilised, it is necessary to have a primary energy factor that is representing the electricity production that is avoided due to the electricity production in the CHP plant.

In the model, two different alternatives were available: The UCPTE mix and marginal production, which is assumed to come from coal power plants. The primary energy factors for were taken from EN 15603[39].

As this thesis is focusing on combined heat and power in Norway, the author included a third electricity mix: NORDEL. This mix represents the average production within the Nordic electricity market. The primary energy factor for the NORDEL mix was derived from the results presented in [46]. The study did not present a corresponding CO_2-eq emission per kWh, so the NORDEL option is only valid for primary energy calculations.

TABLE 6: ELECTRICITY MIXES AND PRIMARY ENERGY FACTORS

Electricity Production	Primary Energy Factor	Source
Electricity from coal power plant	4,05	[39]
Electricity Mix UCPTE	3,31	[39]
Electricity Mix NORDEL	2,16	[46]

4.4.8 Modifications

Some modifications were done by the author to the existing calculation tool, and these are summarised in the list below. In addition, a screen shot of the modified input screen is added in appendix A.

1. The results from the simulations of the different CHP technologies were implemented so that the efficiency and power to heat ratio now depend on the technology chosen.
2. NORDEL was introduced as an electricity mix.
3. Three different alternatives regarding yearly load distribution were implemented. (These are described later in the report, in paragraph 5.1.3) In this way it is possible to study how part load operation influences the total primary energy factor, as the yearly total efficiency and power to heat ratio changes when the load distribution pattern changes. It is, however, important to emphasise that changes in heat loss from pipes etc were *not* changed accordingly.
4. The exergy allocation method was developed so that the effect of sliding temperature during heat transfer was taken into account. The temperatures are automatically updated according to the yearly load distribution profile that is chosen.

4.5 District Heating Load and Temperatures

The district heating load varies from day to day, from hour to hour. To obtain a detailed load duration curve for a specific area containing a set of buildings, it is possible to use relations that connect the mean outdoor temperature with a heat load characteristic for the building categories in question. Temperature-dependent heat models that can predict the heat consumption hour by hour for typical building categories and week days have been developed for Norwegian conditions by [47].

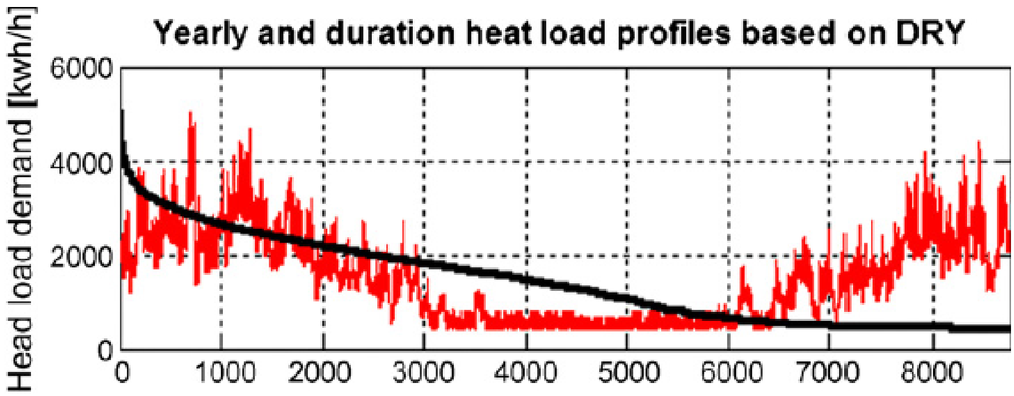


FIGURE 44: EXAMPLE OF YEARLY AND DURATION HEAT LOAD PROFILES, REPRINTED FROM [47]

This amount of detail is, however, not always necessary. To simplify, it is possible to separate the load into high, mid and low season. In this way load variations are taken into account, without having to run the models on infinite different part load levels, which is practically impossible with the software used.

To be able to integrate the results from the CHP plant simulations with the primary energy factor calculation tool in a coherent way, each case was run with the same district heating load curve and temperatures that was used in the development of the PEF calculation tool done by [16]. In this way, the annual efficiency of the plants was calculated on the same basis as the heat loss from the district heating pipes and the pressure drop in the same pipes, which also contributes to the primary energy factor.

The estimation done in [16] is based on aggregated measurements of the load from two different district heating producers in the Nordic countries. The base case regarding load and temperatures is thus defined according to the information given in Table 7.

The same information is visualised in Figure 45.

TABLE 7: DISTRICT HEATING LOAD AND SUPPLY AND RETURN TEMPERATURES

Season	% of the year	% of maximum DH load	% of max load for CHP plant	Supply Temperature [°C]	Return Temperature [°C]
Winter	25	67	100	105	50
Spring/Autumn	50	38	56	80	40
Summer	25	18	26,5	70	35

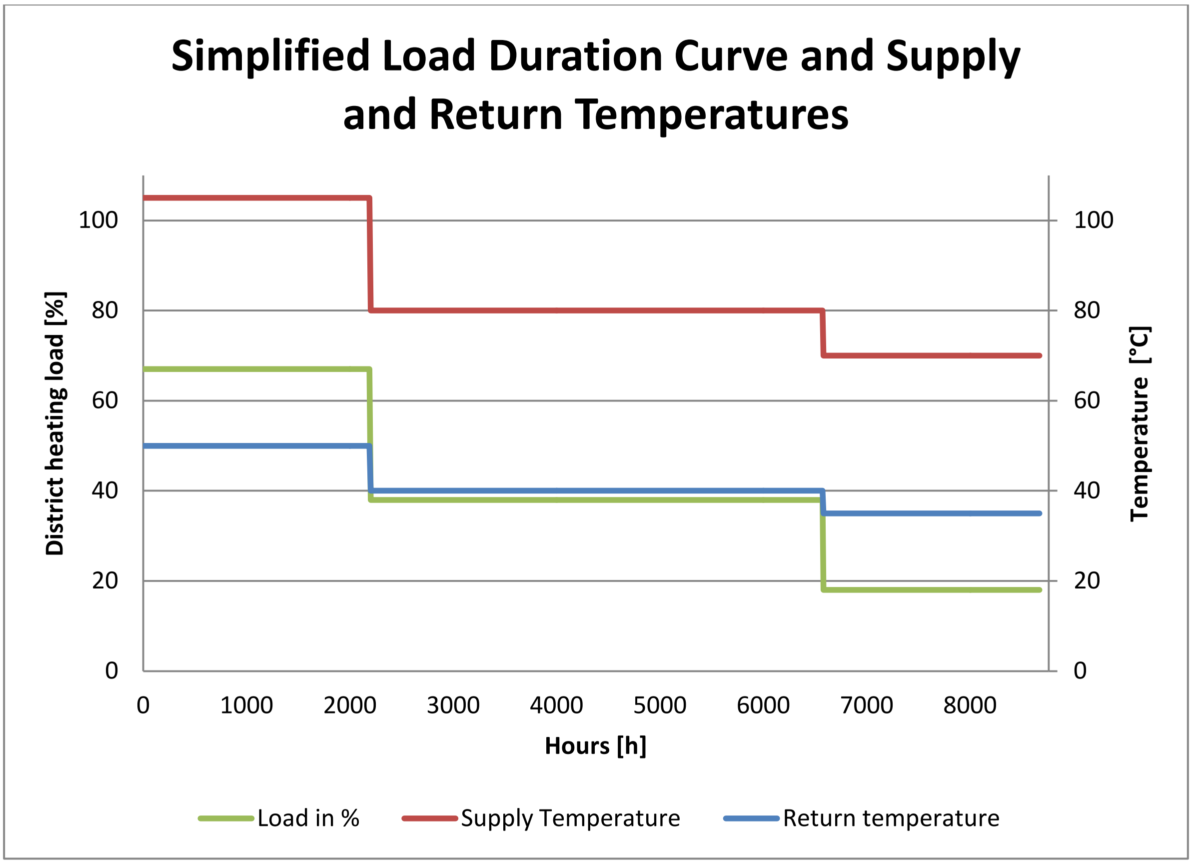


FIGURE 45: LOAD DURATION CURVE AND SUPPLY AND RETURN TEMPERATURES

5 CASE DESCRIPTIONS

5.1 CHP PLANTS

5.1.1 PART LOAD SIMULATIONS

All the chosen technologies were run at three different district heating load levels with corresponding temperature levels according to the information in Table 7: District heating load and supply and return temperatures. The load levels were 100%, 56,5% and 26%.

Natural gas was used as fuel in the combined cycle.

For the rest of the plants, wood waste with a LHV of 15665 kJ/kg at 25 °C and a moisture weight percentage of 10,2% was used as fuel. Further details on the fuel are found in the appendix B.

5.1.2 FUEL TYPE

Grate boilers can be adapted to many types of fuel. Due to different fuel characteristics, this will influence the performance of the plant.

To study the effect of different fuel types, design simulations were run in STEAM Pro, doing the same assumptions as for the $10MW_{el}$ plant described in section 4.3.1 in page 51. The only exception is that the supply temperature was set to 120°C and the return temperature to 80°C.

The main characteristics for the fuels chosen can be seen in Table 8. Further details are found in appendix B.

TABLE 8: FUEL PROPERTIES, FROM STEAM PRO

	Wood Waste	Pellets	Municipality Waste	Demolition Wood
LHV [kJ/kg]	15665	16784	10133	15464
Moisture [w%]	10,2	8,7	25,2	9,01
Ash [w%]	6,2	0,5	21,0	11,94

5.1.3 Alternative Yearly Load Distributions

To study the effect of different distributions of part load time, three alternatives were defined. These can be seen in Table 9. "Year 1" is equal to the load distribution in Figure 45 in page 62, while "Year 2" and "Year 3" represents alternatives where the CHP plants spend less time at the lowest part load. In Year 3, the CHP plant is only operated at full and 56% load, where the 56 % part-load occurs 4 months during the year. This case might represent a future scenario where buildings have a lower heat loss, and therefore a smaller heat load ratio between summer and winter. In this case, the distribution of load with respect to time will also change. During the summer and early autumn, the need for domestic hot water represents the only heat load, while space heating is necessary during the other 8 months. It is assumed that the CHP will be dimensioned to cover the average load during the months were space heating is necessary, and only hot water load during the rest of the year. Therefore, it will not be necessary to run the plant on the lowest part load.

TABLE 9: ANNUAL LOAD DISTRIBUTION CASES

DH load in %	*Year 1*	*Year 2*	*Year 3*
100	0,25	0,4	0,67
56	0,5	0,5	0,33
26,5	0,25	0,1	0

5.1.4 Alternative Design DH Temperatures

To investigate the effect of different design temperatures, ten different cases have been studied. In Figure 46, the temperatures used in 5 of the simulations are plotted, and it should be noted how the return temperature is held constant.

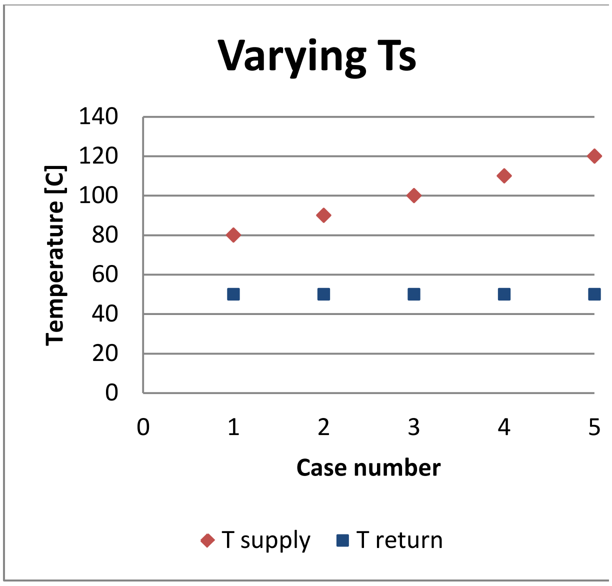


FIGURE 46: VARYING DESIGN TEMPERATURES, RETURN TEMPERATURE HELD CONSTANT

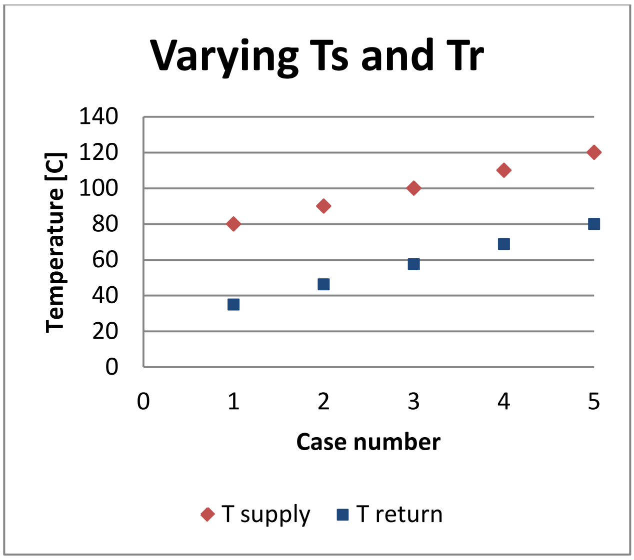


FIGURE 47: VARYING DESIGN TEMPERATURES, BOTH SUPPLY AND RETURN TEMPERATURES

In Figure 47, both supply and return temperatures are varied. The return temperature is based on information on low temperature and high temperature systems found in [48].Case 1 in this figure represents full load in a low temperature network, while case 5 represents full load in a high temperature network. The temperatures in between are found by linear interpolation. The exact temperatures are found in appendix B.

5.2 Case Description - Primary Energy Factors

The results on total efficiency and power to heat ratio from the CHP part load simulations were implemented in the excel calculation tool. Then a number of different cases were studied.

5.2.1 Base Case with different CHP plants

In the base case, the following parameters were equal for all the technologies:

TABLE 10: BASE CASE PRIMARY ENERGY FACTOR CALCULATIONS

Electricity production:	NORDEL mix
Load distribution:	Year 1
Allocation method	Power Bonus
Energy density, district heating	Medium
Life time, pipes district heating	60 years

For the steam cycles, wood chips were chosen as fuel (in the excel tool, in the STEAM Pro simulations "Wood Waste" was used). This fuel had a PEF of 1,19. Natural gas was chosen for the combined cycle and template alternatives. The PEF of this fuel was 1,05.

5.2.2 Allocation Methods

The choice of allocation method will influence the value of the PEF_{DH}. To explore the effect of different allocation methods, all the methods implemented in the excel tool was utilised and compared, except the 200% method.

TABLE 11: ALTERNATIVE PRODUCTION EFFICIENCIES

	Alternative Heat Production	Alternative Electricity Production
Steam Cycle, all sizes	90%	38%
Combined Cycle	90%	58%
Template	90%	58%

The alternative production method is an allocation method that is based on dividing the multiproduct process into separate processes. This means that it is assumed that the same fuel could be used in another process to produce the same output. Consequently, there are different alternative efficiencies for different fuel, and the assumed efficiencies in this case are displayed in Table 11.

5.2.3 CHOICE OF ELECTRICITY PRODUCTION

When the Power Bonus method is used, the primary energy factor of electricity production should represent the electricity production that is being avoided by producing electricity in the CHP plant. The production that is being avoided will always be the marginal production.

But what is the marginal electricity production in Norway?

According to energy planning theory, the marginal production is the cheapest production capacity that is available in the system[49]. In Europe, this is normally assumed to be coal power. As Norway is directly linked to the continent, some argue that coal power should be regarded as marginal power production also here.

In fact, what is actually the marginal capacity varies with load level. [50] argue that coal power only represents the marginal capacity during low load hours, while during peak load hours gas power takes the role as marginal producer.

For this reason, the moment for power production in the CHP facility will influence what power production that is being avoided.

Yet another aspect of this discussion is that it is difficult to know what will be the marginal production in a longer term[51]. Hopefully future marginal production will become more efficient than the current one.

When CO_2 emissions get introduced, the situation gets even more difficult because of EU's CO_2 quota system. It can be argued that when it comes to greenhouse gas emissions, the power bonus method is not applicable, as avoided missions CO_2 emissions will result in unused quotas that will be sold to another polluter. In theory, the CO_2 emissions would remain constant.

Nevertheless it is of interest to study what difference it makes to vary the choice of avoided electricity production. In this case, marginal coal production is compared to the NORDEL average production.

5.2.4 FUEL CHAIN PEF

The fuel chain is complex, and there are no standard rules on how the system boarders should be set. Some studies include materials for construction of the processing facilities, some do not. Bio fuels are especially difficult to assess[40]. It is therefore relevant to study the effect of different primary energy factors for the fuel chain. In Table 12, different primary energy values are presented, and these are used to study the impact of this parameter. Municipal waste has a value that is below unity. This is because the "Polluter Pays" principle has been used. According to this way of reasoning, the environmental burden of waste is allocated to the ones that bought the products in the first place, and thus were the reason that the product was produced. The reason why the value is not 0 but 0,04 is that the transport and pre treatment of the waste before incineration is regarded as fuel handling.

TABLE 12: PRIMARY ENERGY FACTOR, DIFFERENT FUEL CHAINS

Fuel	PEF, Fuel Chain	Source
Wood Chips	1,19	Value from[16], where it is calculated based on the Ecoinvent database.
Pellets	1,11	[52] – Miljöfaktaboken 2011
Wood Waste	1,03	[52] – Miljöfaktaboken 2011
Municipal Waste	0,04	[52] – Miljöfaktaboken 2011

5.2.5 Different Load Distribution

The total efficiency and power to heat ratio of the CHP plants vary according to load percentage. As a result of this, the annual efficiency will vary with respect to how much time the plant will have to be run on part load.

To study the effect of different annual load distributions, annual efficiencies and power to heat ratios were calculated for the different CHP technologies from the results of the simulations described in section 5.1.1: Part Load Simulations, and the yearly distributions presented in table Table 9: Annual load distribution cases.

5.2.6 Impact of Energy Density and Life Time of Pipes

The energy density in the area where the district heating is supplied influences the relative heat loss from the pipes and the pump work. In [16], values for heat loss and pump work has been calculated for different energy densities, assuming the temperatures and load distribution presented in Figure 45. The values, assuming a pipe lifetime of 30 or 60 years, are reprinted in Table 13.

TABLE 13: ENERGY LOSSES RELATED TO THE CONSTRUCTION OF AND DISTRIBUTION IN THE DISTRICT HEATING NETWORK. VALUES FROM [16].

	PEF_{PIPES}, 30 yr [kWh/kWh_{del}]	PEF_{PIPES}, 60 yr [kWh/kWh_{del}]	Energy Loss [%]	Pump Electricity [kWh/kWh_{del}]
Low (3 MWh/m)	0,0116	0,0058	13,3 %	2,23E-05
Medium (8 MWh/m)	0,0042	0,0021	5,4 %	8,13E-06
High (15 MWh/m)	0,0008	0,0004	0,9 %	1,48E-06

To study the effect of different energy densities, calculations were performed with low, medium and high energy density with a pipe life time of 60 years. To analyse the impact of pipe life time, a case with medium energy density and a pipe life time of 30 years was also calculated.

6 RESULTS AND ANALYSIS - CHP PLANT PERFORMANCE

In this chapter the results from the cases described in section 5.1 are presented. In section 6.1, the results from the part load simulations are presented in a tabular format. The values are then compared and analysed in section 6.2. The impact of different fuel types, yearly load profiles and supply and return temperatures are then presented and analysed in sections 6.3

6.1 IMPACT OF PART LOAD OPERATION

All the plants were simulated according to the load profile and supply and return temperatures presented in Table 7 and Figure 45. This was done to analyse the general differences in performance between the plants, in addition to studying the relationship between design performance and annual performance.

6.1.1 2 MW_{EL} STEAM TURBINE CHP PLANT, GRATE BOILER

In Table 14 , the results from the part load calculations for the smallest steam cycle plant are summarised.

TABLE 14: RESULTS PART LOAD CALCULATIONS, 2MWEL, GRATE BOILER

Parameter	Full load		Med. Load		Low load	
District heating load	8094	kW	4552	kW	2185	kW
DH load, %	100	%	56,2	%	26,99	%
T supply	105	°C	80	°C	70	°C
T return	50	°C	40	°C	35	°C
Fuel input (LHV)	11521	kW	6335	kW	2794	kW
η_{el}	17,36	%	15,27	%	7,02	%
η_{CHP}	87,61	%	87,12	%	85,21	%
p exhaust steam	0,4833	bar	0,189	bar	0,167	bar
α	0,244	-	0,212	-	0,09	-

6.1.2 10 MW_{EL} PLANT STEAM TURBINE CHP PLANT, GRATE BOILER

Table 15 shows the results for the 10 MW_{el} steam cycle plant.

TABLE 15: RESULTS PART LOAD CALCULATIONS, 10 MWEL, GRATE BOILER

Parameter	Full load		Med. Load		Low load	
District heating load	32768	kW	18349	kW	8672	kW
DH load, %	100	%	56	%	26,5	%
T supply	105	°C	80	°C	70	°C
T return	50	°C	40	°C	35	°C
Fuel input (LHV)	49536	kW	27086	kW	11741	kW
η_{el}	20,32	%	18,11	%	10,69	%
η_{CHP}	86,44	%	85,86	%	84,56	%
p exhaust steam	0,474	bar	0,523	bar	0,313	bar
α	0,307	-	0,267	-	0,145	-

6.1.3 25 MW_{EL} STEAM TURBINE CHP PLANT, CFB BOILER

Table 16 shows the results for the 25 MW_{el} CHP plant.

TABLE 16: RESULTS PART LOAD CALCULATIONS, 25 MWEL, CFB BOILER

Parameter	Full load		Med. Load		Low load	
District heating load	77807	kW	43568	kW	20623	kW
DH load, %	100	%	56	%	26,5	%
T supply	105	°C	80	°C	70	°C
T return	50	°C	40	°C	35	°C
Fuel input (LHV)	117620	kW	64036	kW	28813	kW
η_{el}	21,26	%	19,39	%	15,11	%
η_{CHP}	87,4	%	87,4	%	86,7	%
p exhaust steam	0,484	Bar	0,524	bar	0,318	bar
α	0,321	-	0,285	-	0,211	-

6.1.4 COMBINED CYCLE, 22,7 MW_{EL}

The lower limit for part load operation of the gas turbine was 15,1% of full load. This resulted in a district heating load of 6843 kW, which is 35,8% of maximum DH load.

The values in the Low Load column are found by assuming that the surplus heat is cooled off. This alternative will in the rest of the report be referred to as Combined Cycle or CC.

It is also possible to assume that the extra heat is made use of, and that no heat needs to be cooled away. This gives the results that are listed in the column "Minimum Load". These results, where the heat production in reality is higher than the DH demand, is marked with a star and referred to as Combined Cycle* or CC*.

TABLE 17: RESULTS PART LOAD OPERATION, COMBINED CYCLE CHP PLANT

Parameter	Full Load		Med. Load		Minimum Load		Low Load	
District heating load	19129	kW	10717	kW	6843	kW	4974	kW
DH load, %	100	%	56	%	35,8	%	26	%
T supply	105	°C	80	°C	70	°C	70	°C
T return	50	°C	40	°C	35	°C	35	°C
Fuel input (LHV)	52428	kW	25203	kW	13997	kW	13997	kW
η_{el}	43,36	%	35,42	%	27,02	%	27,02	%
η_{CHP}	79,84	%	77,94	%	75,91	%	62,55	%
p exhaust steam	0,483	bar	0,3627	bar	0,331	bar	0,331	bar
α	1,188	-	0,833	-	0,553	-	0,760	-

6.2 COMPARISON AND ANALYSIS, PART LOAD SIMULATIONS

6.2.1 POWER EFFICIENCY

Figure 48 shows the power efficiencies of the different CHP plants plotted against the district heating load. The combined cycle power efficiency is, as expected, in general significantly higher than the power efficiencies of the pure steam cycle plants. As the district heating load decreases, however, the CC efficiency plunges almost 40%, from a design value of 43,36% to a low load value of 27,02%.

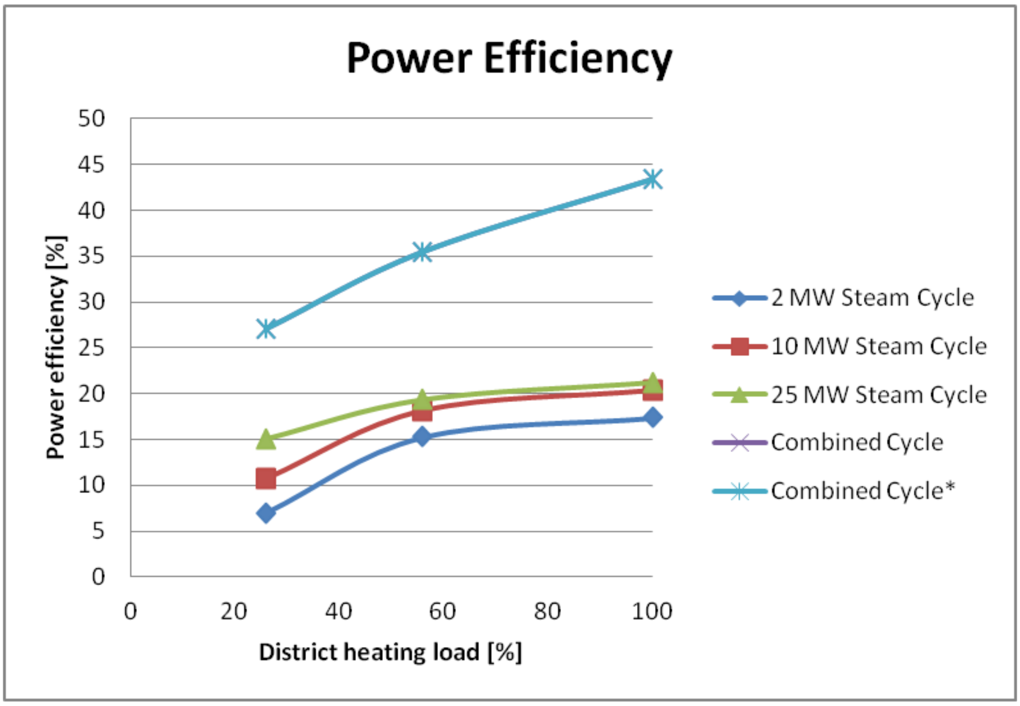


FIGURE 48: COMPARISON OF POWER EFFICIENCY

The steam cycles also experience a substantial drop in power efficiency from full load to part load. The behaviour diverges between the two plants with grate boiler and the largest plant with a CFB boiler. For the two grate boiler plants, the power efficiency drop rapidly when the DH load is reduced to 26% of full load. The 2 MW_{el} plant experiences a 60% drop in efficiency, while the 10MW_{el} plant suffers a 47% drop. For both plants, approximately 80% of the efficiency drop occurs between medium and low load.

The plant with the CFB boiler handles the part load operation better than the other plants, and the drop in efficiency from full to minimum load is only 29%. Approximately 30% of the efficiency drop occurs between full and medium load, and 70% between medium and low load.

It is important to remember that also the district heating temperatures change during part load. Lower supply temperatures means that the steam in principle can expand further. Since the CHP plants are run in off-design mode at part load, it is limited how much this can be exploited, but surely the power efficiencies would have been even lower if the supply and return temperatures were held constant during part load operation.

6.2.2 TOTAL EFFICIENCY

The part load total efficiencies are shown in Figure 49. They show a remarkable stable behaviour, except for the Combined Cycle alternative which has a very different progression compared to the others. This is due to the minimum load restriction of the gas turbine combined with the assumption that the surplus heat is cooled off. For the Combined Cycle* alternative, it is assumed that the surplus heat is made use of in another application, and as a consequence the CHP efficiency has a similar trend as the other alternatives.

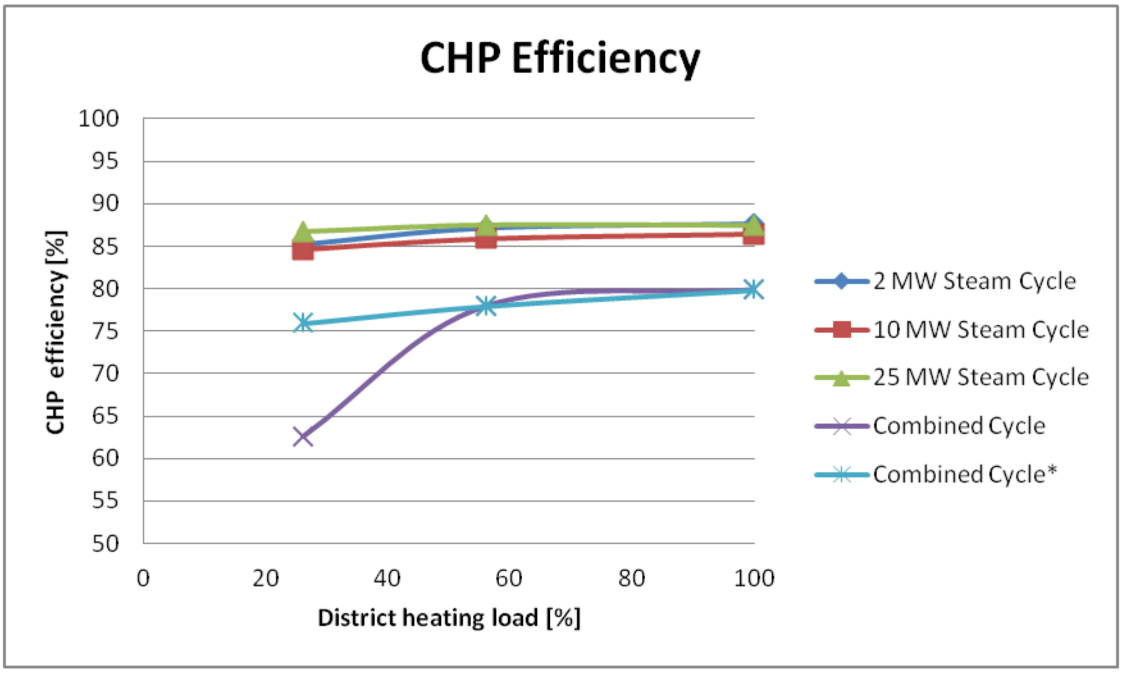


FIGURE 49: CHP PART LOAD EFFICIENCY

The values for the CHP efficiency varies from 79,84% at full load to 75,91% at minimum load for the Combined Cycle alternative. This represents a drop in efficiency of 4,9%, which is far less than the 40% drop of the power efficiency. The efficiency for the Combined Cycle* alternative drops to 62,55% due to the waste of heat energy, which represents a drop of 21,6%.

The steam cycle plants are in general one cut above the combined cycle plant when it comes to the total efficiency. As for the power efficiency, the CFB steam cycle plant seems to handle the part load operation better than the grate boiler plants. The total efficiency decreases only 0,7 percent points from full to minimum load, while for the 10 MW and 2 MW plant, the decrease is respectively 1,8 and 2,4 percent points. This represents an efficiency drop of respectively 0,8%, 2,1% and 2,7%. It is interesting to observe that the smallest grate boiler power plant achieve a slightly higher total efficiency than the other steam cycle plants at full load.

6.2.3 Power to Heat Ratio

The power to heat ratio of the different steam cycles can be seen in Figure 50. It is a significant difference between the two largest and the smallest CHP plant.

The 2 MW plant has a maximum PHR of 0,25 at maximum load. At medium load this drops to 0,18, before plunging to 0,09, which corresponds to a total drop of 64%. The 10 MW plant has a similar development, but the starting point is substantially higher with a PHR of 0,31 a full load. It decreases in total 52%, ending at 0,14. 75% of the decrease happens when the DH load decreases from 56% to 26%.

The 25MW plant with CFB boiler obtains a PHR of 0,32 at full load, which is 30% higher than the 2 MW plant, but only 5% higher compared to the 10 MW plant. The decrease is, however less rapid than for the 10 MW plant, and the total reduction is only 34%, resulting in a PHR at 0,21 at minimum load.

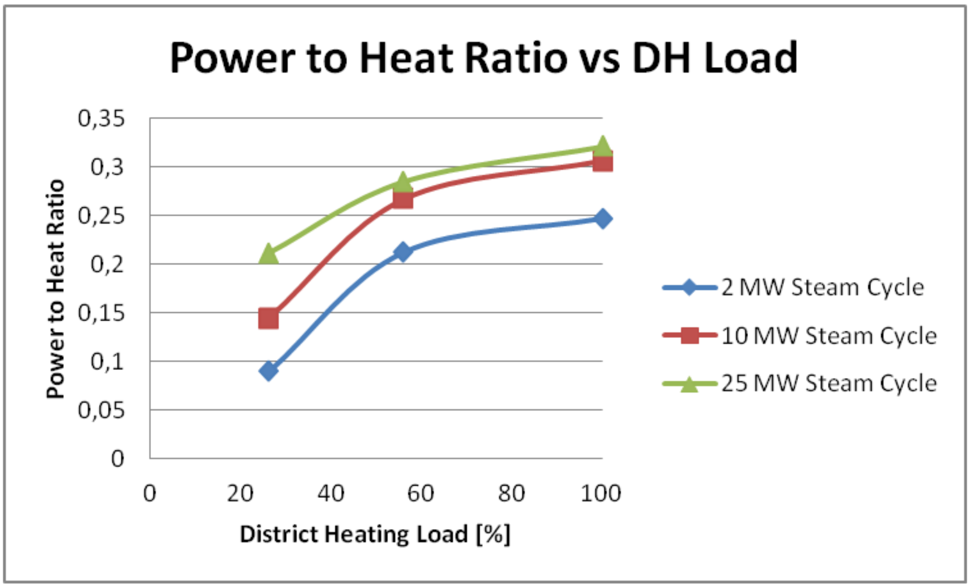


FIGURE 50: POWER TO HEAT RATIO, STEAM CYCLES

It seems like increased size helps to reach a higher PHR, but that the boiler type is dominant when it comes to deciding the behaviour at particularly low part loads.

Figure 51 shows the PHR of the combined cycle. As expected, the power to heat ratio is higher at minimum load in the scenario were heat is cooled off. At full load the PHR is 1,19, while it decreases to 0,83 at medium load. This represents a 30% decrease. The minimum load PHR is 0,76 for the Combined Cycle and 0,55 for the Combined Cycle*. The total decrease is thus respectively 36% and 54%.

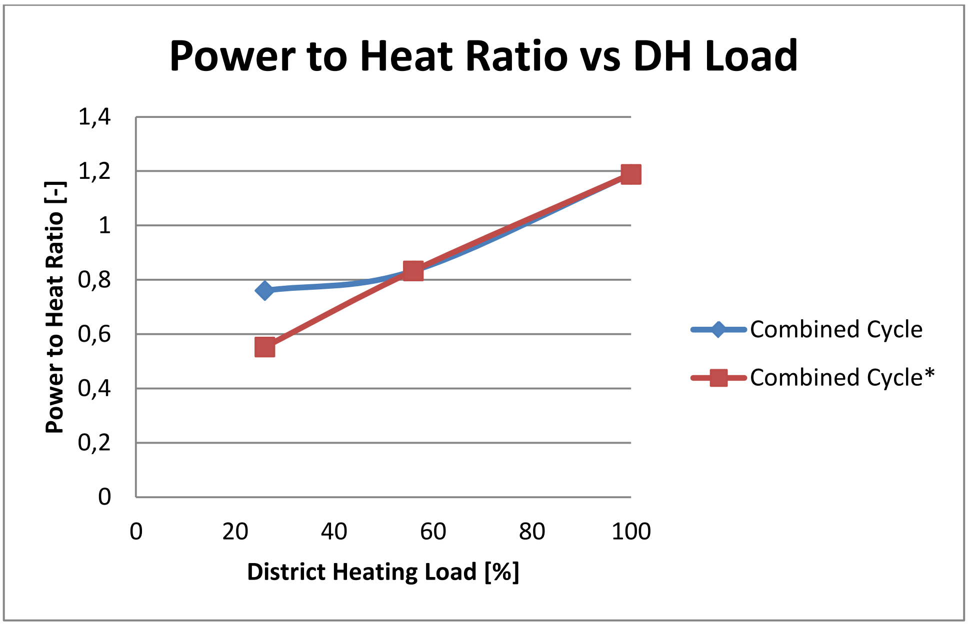
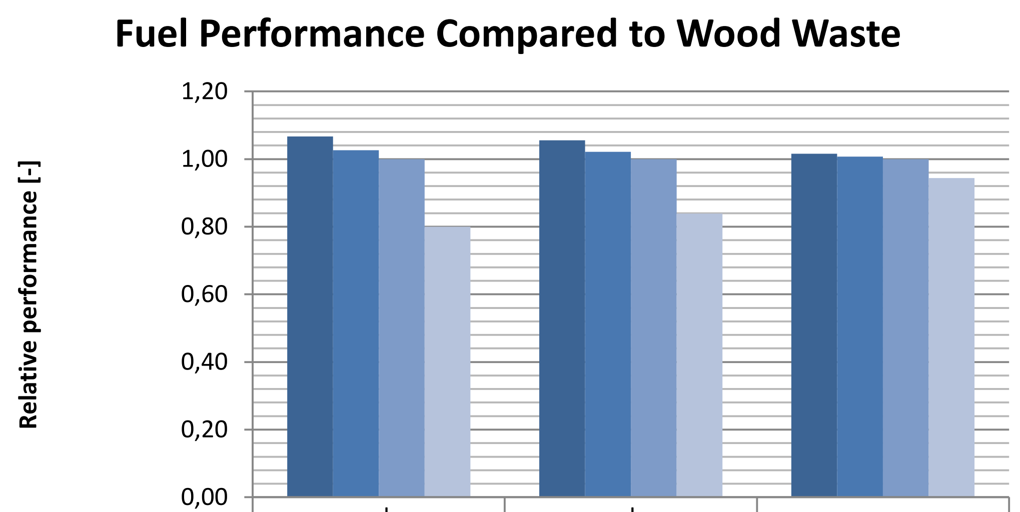


FIGURE 51: POWER TO HEAT RATIO, COMBINED CYCLE

6.3 Impact of Different Fuel

The power plant performance for the different fuel types are displayed in Figure 52, and it shows that the differences are significant. Pellets gives the highest power efficiency at 18,85%, demolition wood and waste wood follows closely with 18,14% and 17,67%. Municipal waste stands out with a low value of 14,13%, which is not surprising. A high moisture content and a low LHV makes the waste plant by far the least effective.



	ηel	ηchp	α
Pellets	1,07	1,06	1,02
Demolition Wood	1,03	1,02	1,01
Wood Waste	1,00	1,00	1,00
Waste	0,80	0,84	0,94

FIGURE 52: IMPACT OF DIFFERENT FUEL ON POWER EFFICIENCY, GRATE BOILER 10 MWEL

For the total efficiency, the pattern is similar. Also here the municipal waste stands out with a distinctly lower value than the rest, only 71%. Pellets excel in the other range of the scale with a total efficiency of 89%, while the wood waste and demolition wood rates at 84% and 86%.

The power to heat ratio differ less than the two preceding efficiencies. Pellets, demolition wood and wood waste have very similar values of respectively 0,269, 0,267 and 0,265. Municipal waste has a PHR of 0,250.

6.4 Impact of Yearly Load Distribution

The impact of different load distributions on the annual power efficiencies is shown in Figure 53. The "Design" column represents the annual efficiency if the plant were run at 100% load all year. For the combined cycle, the annual efficiency ranges from the design value at 43% all the way down to 35%. This represents a decrease of 18,6%.

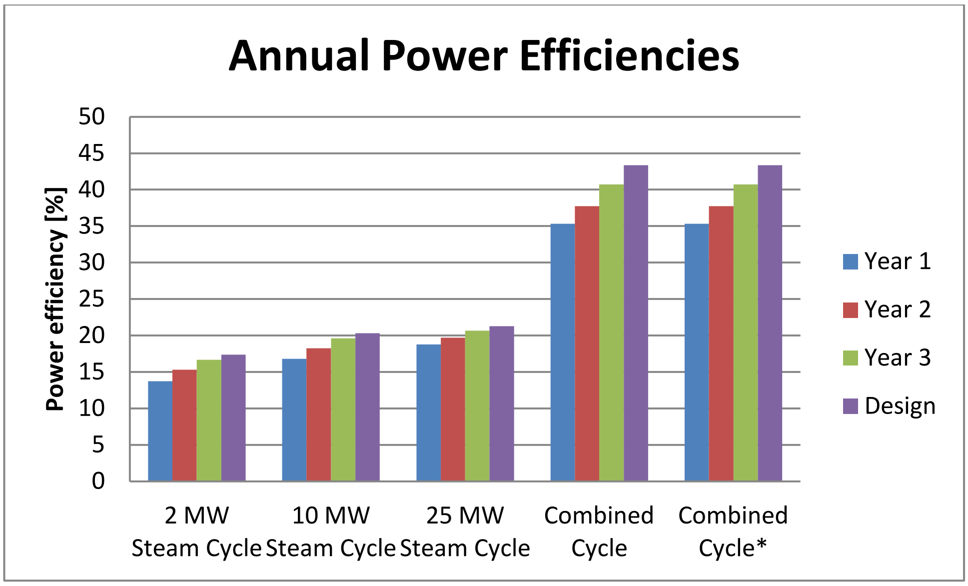


FIGURE 53: ANNUAL POWER EFFICIENCIES, DIFFERENT LOAD DISTRIBUTIONS

The 2 MW and 10 MW steam cycle have similar reductions of 20,9% and 17,3%. The steam cycle with the fluidised bed, however, only get an annual power efficiency reduction of 11,6% due to its relative smaller difference between maximum and minimum part load power efficiency.

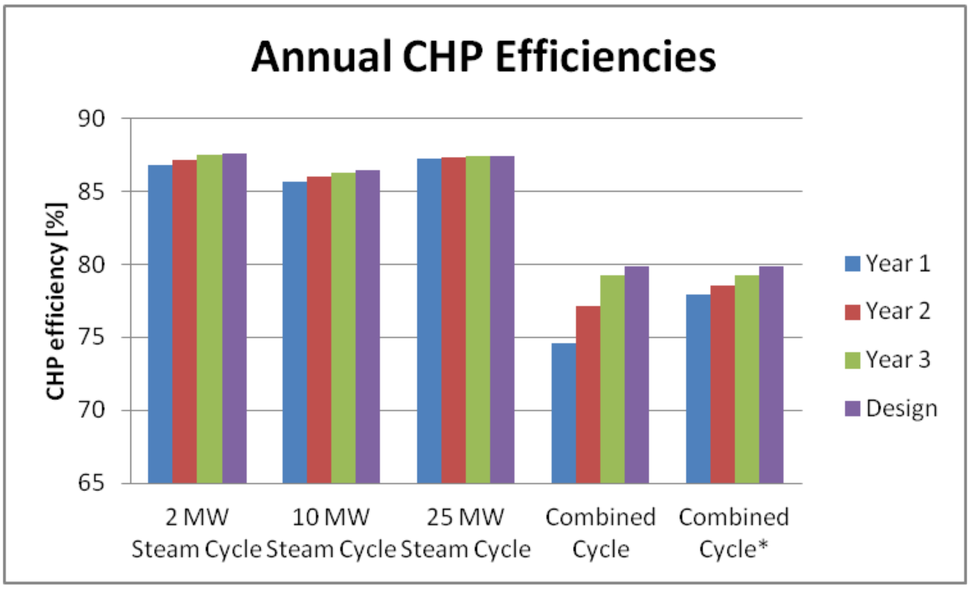


FIGURE 54: ANNUAL CHP EFFICIENCIES, DIFFERENT LOAD DISTRIBUTIONS

In Figure 54 the annual CHP efficiencies are visualised. As expected, the differences are rather minute, as the CHP efficiencies were quite stable at all DH loads. For the Combined Cycle, however, the differences are dramatic in the cases Year 1 and Year 2. In these cases the lowest part load is included, and as the total efficiency was very low at this level the annual efficiencies are severely reduced. It the Combined Cycle* alternative this effect is less profound due to the assumption that the excess heat produced is used for something useful, which again leads to a decent CHP efficiency even at minimum load. Still, the difference in for the Combined Cycle* alternative are larger than for the steam cycle alternatives, with a difference from design to Year 1 of 6,6%, while the corresponding number for all the steam cycles are less than 1%.

The annual power to heat ratios for the steam cycles are given in Figure 55. As the power efficiency was significantly influenced during part load operation, the PHRs are considerably different for each load distribution case. The 25 MW steam cycle is slightly less influenced due to better performance at part load, and has a reduction from Design to Year 1 of 14%. The 2 MW and 10 MW steam cycles have differences of 23% and 20%.

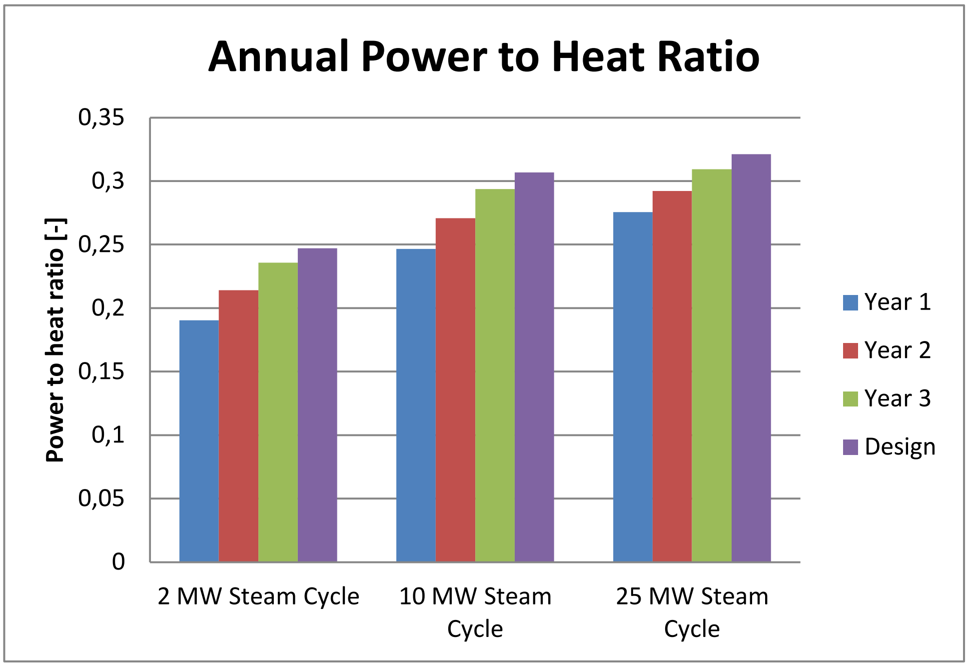


FIGURE 55: ANNUAL POWER TO HEAT RATIO STEAM CYCLES, DIFFERENT LOAD DISTRIBUTIONS

The annual PHR for the combined cycle plant is shown on the graph in Figure 56. The Design and Year 3 cases are equal for both the Combined Cycle and the Combined Cycle* alternative, as the lowest load level is not included. For Year 1 and Year 2, however, there are some differences. The reduction from Design to Year 1 is 24% for the Combined Cycle alternative and 28% for the Combined Cycle* alternative.

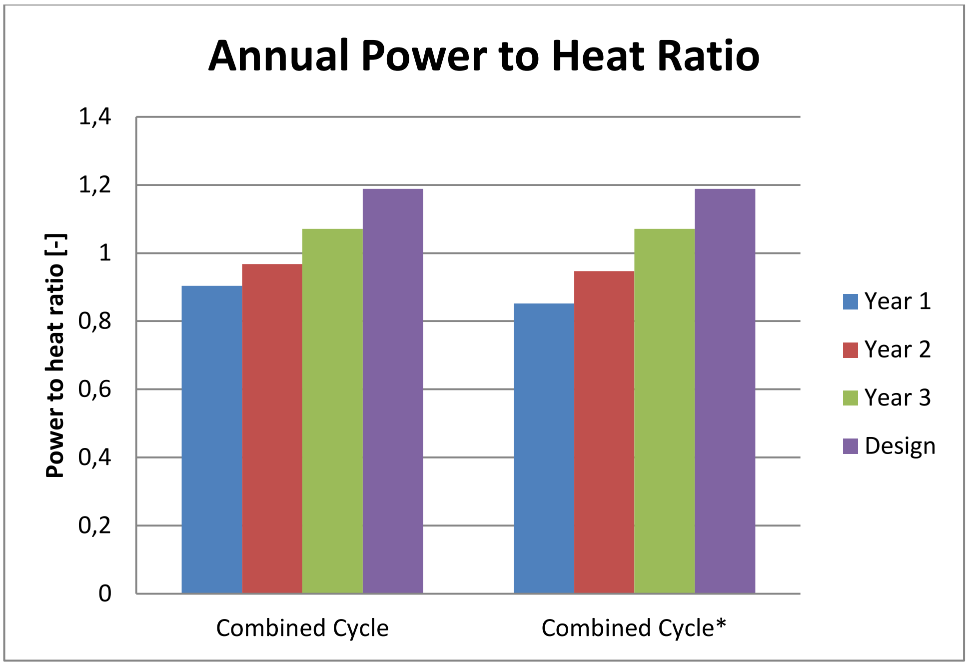


FIGURE 56: ANNUAL POWER TO HEAT RATIO, COMBINED CYCLE, DIFFERENT LOAD DISTRIBUTIONS

6.5 IMPACT OF TEMPERATURE LEVELS IN THE DISTRICT HEATING NETWORK

6.5.1 POWER EFFICIENCY

Figure 57 shows the change in power efficiency for the 10 MW steam cycle plant when the DH supply and return temperatures are varied. The blue curve shows the results when the return temperature is held constant at 50°C as described in Figure 46 while the supply temperature is altered. The red curve shows the results when the return temperature is varied according to Figure 47.

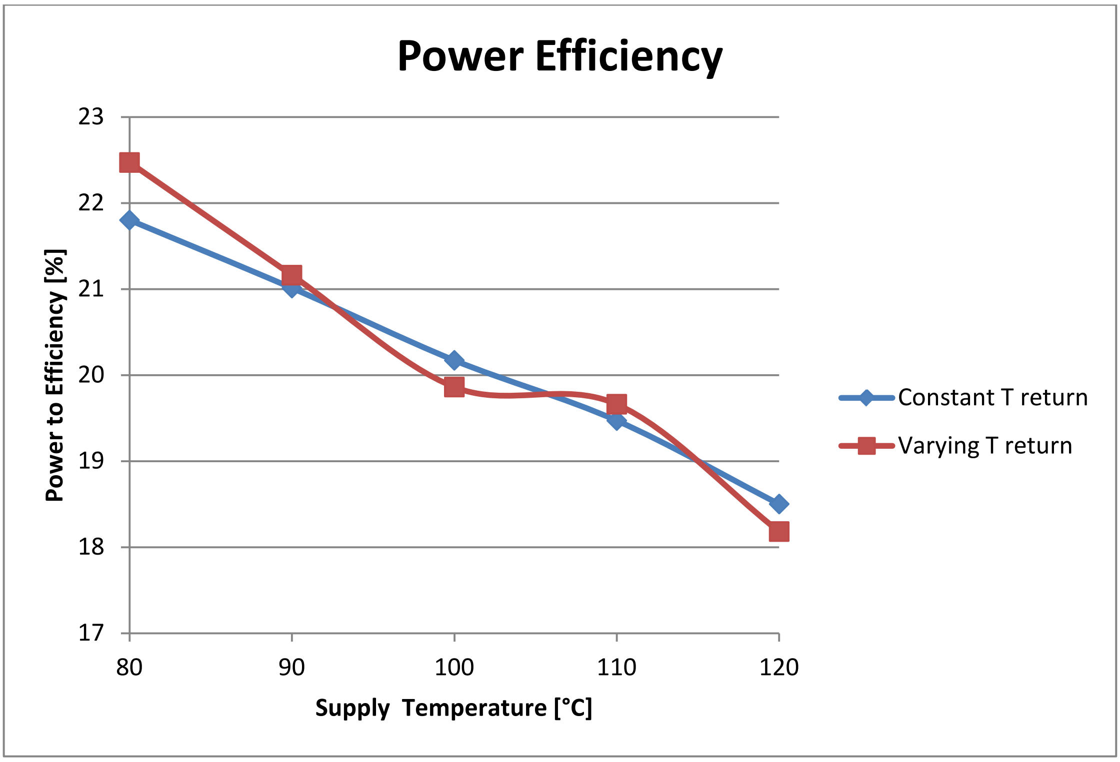


FIGURE 57: POWER EFFICIENCY, DIFFERENT SUPPLY AND RETURN TEMPERATURES

The constant return temperature alternative shows an almost linear behaviour, and it shows clearly that a higher supply temperature reduces the power efficiency. This is as expected, because a higher supply temperature makes it necessary to extract steam at a higher pressure. Higher pressure means that the steam now contains more exergy. As the bleed steam expands less than with a lower supply temperature, less work is produced per kg of input steam and the power efficiency decreases. ¨

The red curve starts off with a higher efficiency than the constant T return alternative. This is because a lower return temperature makes it possible to reduce the condenser pressure, and thus expand the steam further and produce more work.

Between a Ts of 90 °C and 100 °C, the return temperature in the "Varying T return" case intersects the constant T return case, and the efficiency thus falls below the constant Tr case.

Between 100 °C and 110 °C, the pattern is broken. Suddenly the power efficiency is almost constant. Between 110 °C and 120 °C, the decline is similar to the trend from 80°C to 100°C again.

Why does this happen?

The answer lies in the way STEAM Pro designs the power plants. Since these simulations are design simulations, the components are not equally sized for all runs. To understand the behaviour between 100°C and 110°C, it is necessary to have a close look at the details of the simulation results.

After studying all parameters closely, it turned out that the enthalpy of the bleed steam changed very little from 100 °C to 110 °C compared to the other intervals. Further studies of the steam turbines showed a radical change in design parameters in the same interval.

A steam turbine is constructed of several stages, and the stages are in STEAM Pro sorted into groups. The efficiency of each group is described by three efficiencies: Dry step efficiency, group blading efficiency and the group overall efficiency. The group overall efficiency includes all losses: From valves, moisture and exit loss at the outlet. The efficiency is defined as the real enthalpy difference across the group divided by the isentropic enthalpy difference.

The steam turbine in this case consists of four groups, and in Figure 58 the development of the four group efficiencies are shown compared to the extraction steam enthalpy. The steam is extracted after group 2.

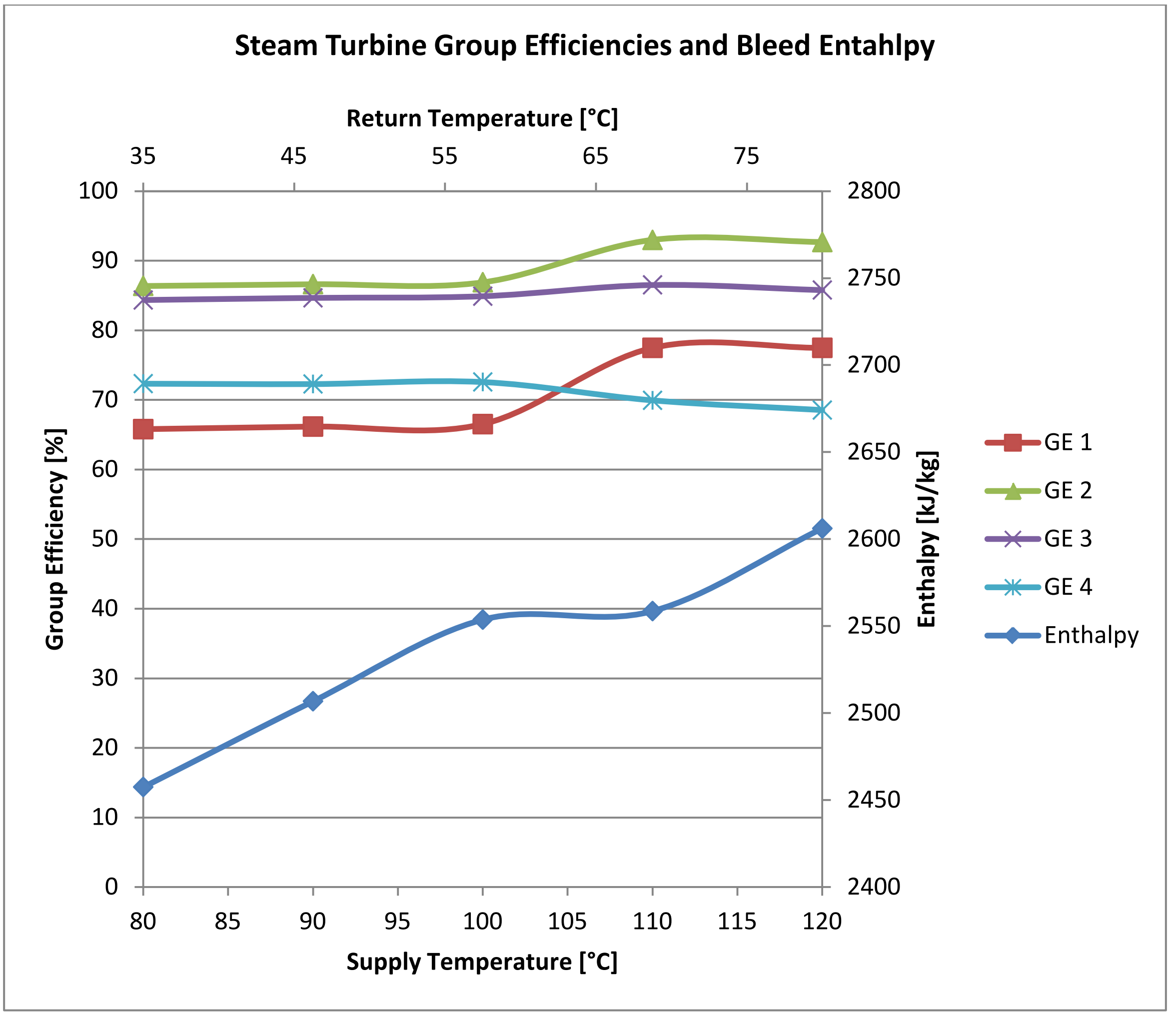


FIGURE 58: STEAM TURBINE GROUP EFFCICIENCIES AND BLEED STEAM ENTALPHY

It can clearly be seen that the design of the steam turbine changes when the return temperature goes from 57,5 to 68,75 °C. Group efficiency 1 and 2 increase considerably, group efficiency 3 increase marginally while group efficiency 4 decrease approximately 3 percent points. The change seems to be discrete, as the new values stay almost completely constant afterwards.

The increased efficiency of the steam turbine counteracts the negative influence of increased bleed pressure, and as a consequence the power efficiency remains almost constant from Ts equal 100 to Ts equal 110 °C. When Ts is increased to 120°C, and thus the bleed pressure further increased, the power efficiency continues to fall because the group efficiencies now are close to constant. The decrease is a little bit steeper than before the change of the ST design, and this can be explained by the slight efficiency decrease in Group 4 from Ts equal 110°C to Ts 120°C, while the group efficiencies was completely constant from case 1 to 3.

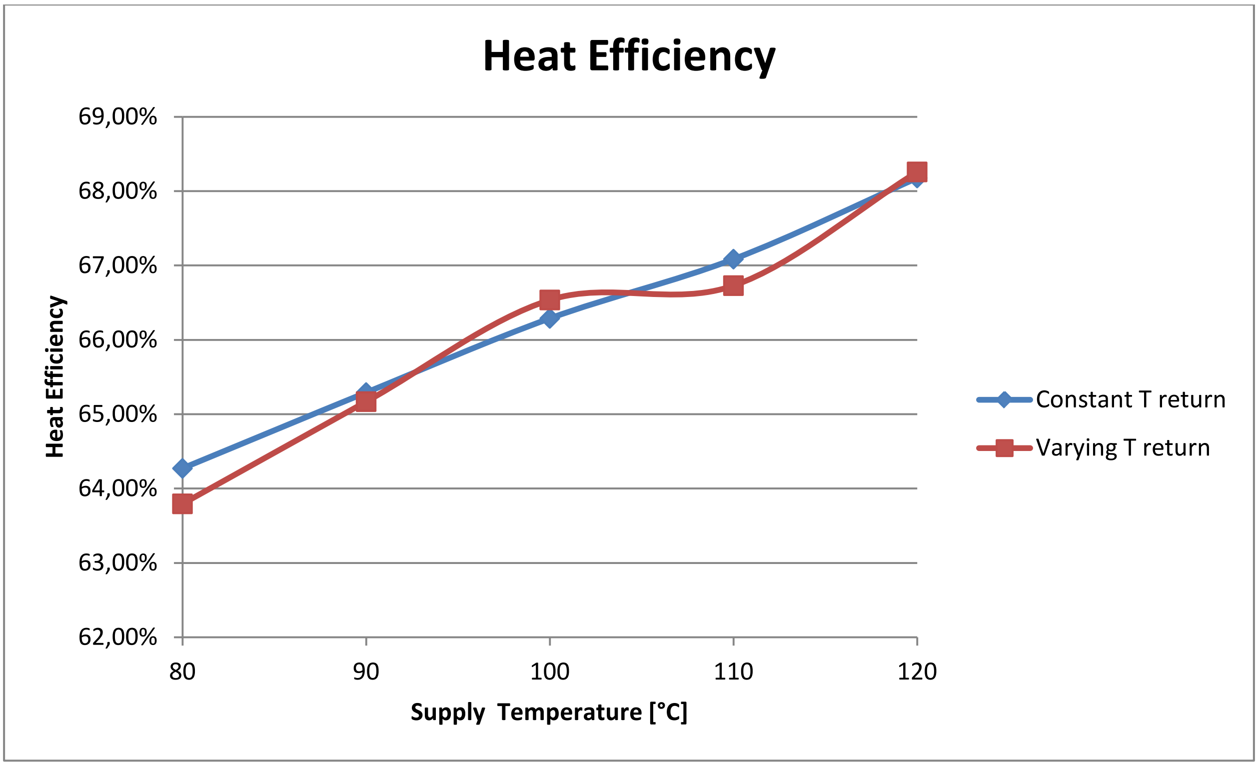


FIGURE 59: HEAT EFFICIENCY, DIFFERENT SUPPLY AND RETURN TEMPERATURES

Figure 60 shows the heat efficiency for the different cases. The form is mirrored compared to the power efficiency graph because of the discrete change in the steam turbine.

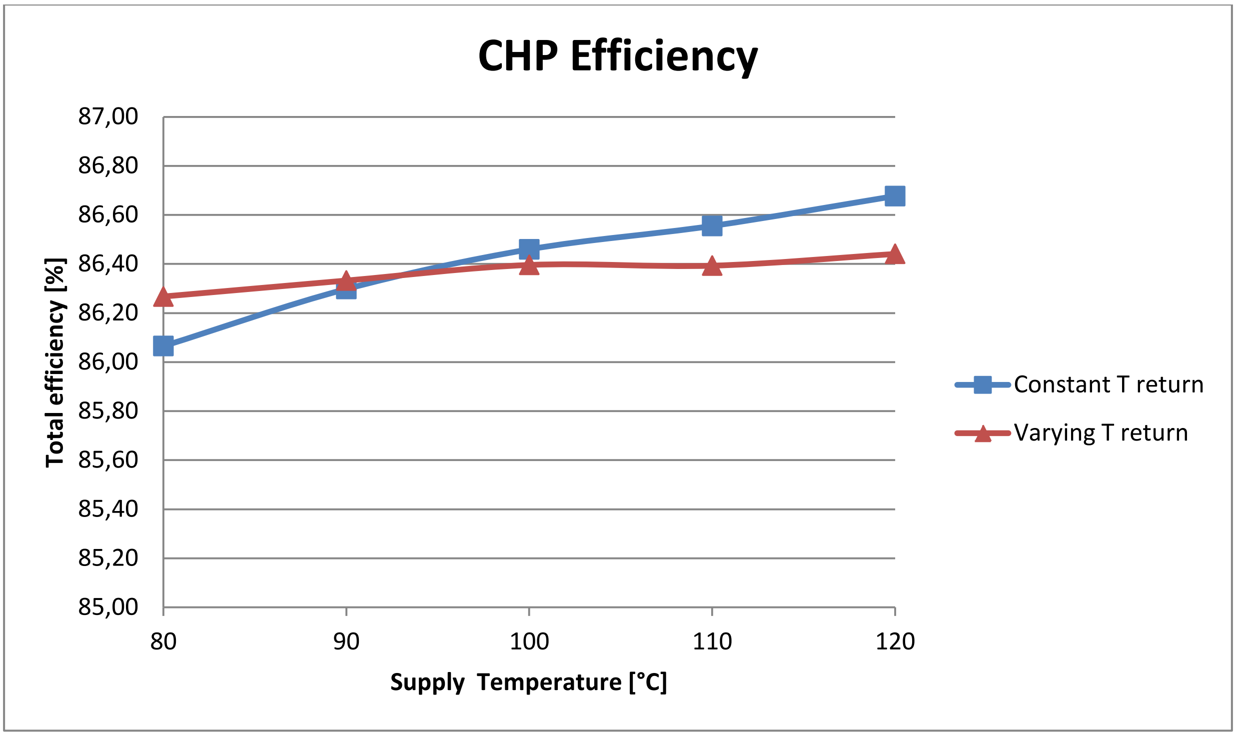


FIGURE 60: CHP EFFICIENCY, DIFFERENT SUPPLY AND RETURN TEMPERATURES

The CHP efficiency is equal to the sum of the heat efficiency and the power efficiency. Figure 60 shows that the increase in heat efficiency is somewhat larger than the decrease in power efficiency, and both the Tr constant and Tr varying case show a slight increase in CHP efficiency. The change is, however, minuscule when T return is varied and less than 1 percent point when T return is held constant.

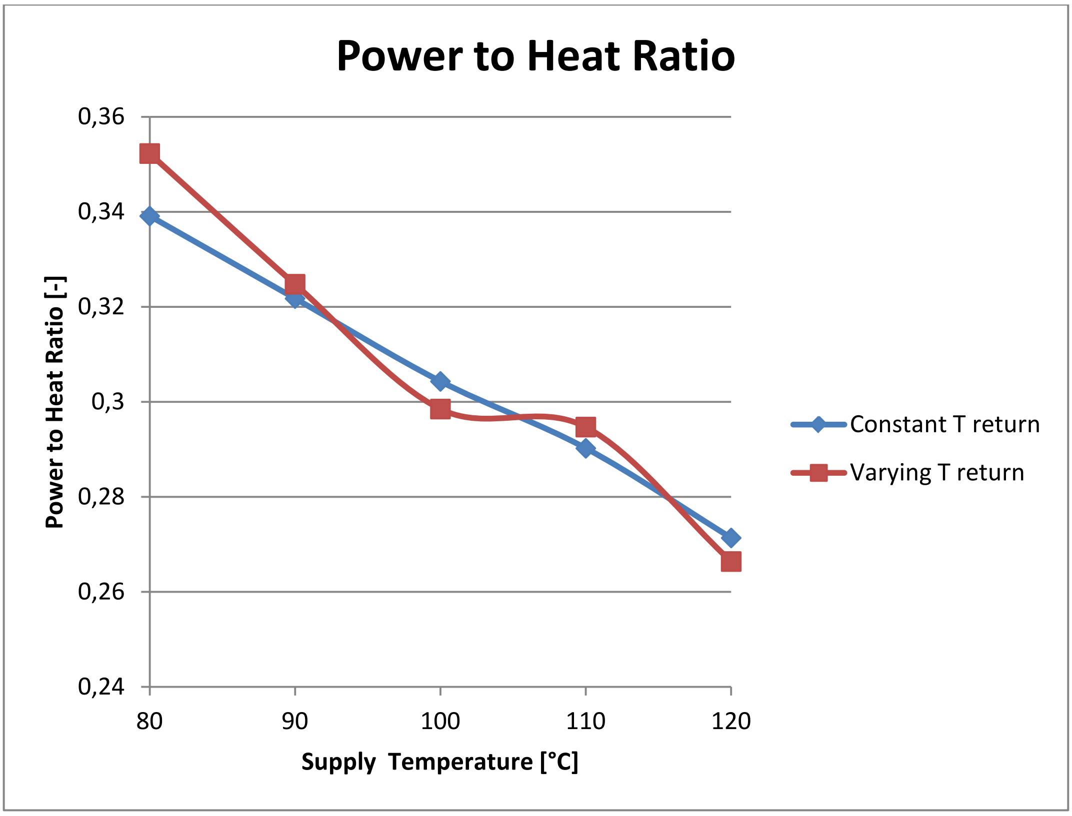
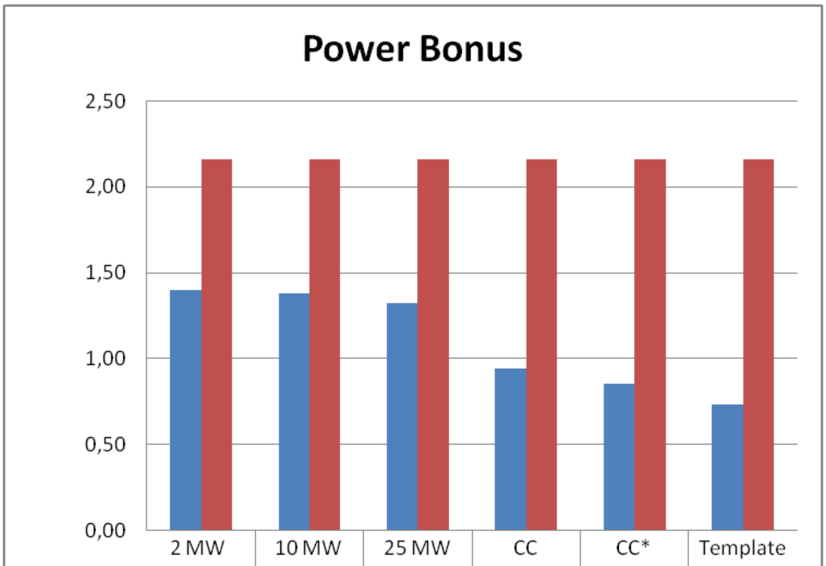


FIGURE 61: POWER TO HEAT RATIO, DIFFERENT SUPPLY AND RETURN TEMPERATURES

The power to heat ratio, as shown in Figure 61, shows a similar behaviour as the power efficiency. The only difference is that the relative change is larger due to the increased heat efficiency.

7 RESULTS AND ANALYSIS- PRIMARY ENERGY FACTORS

7.1.1 BASE CASE – ALL TECHNOLOGIES



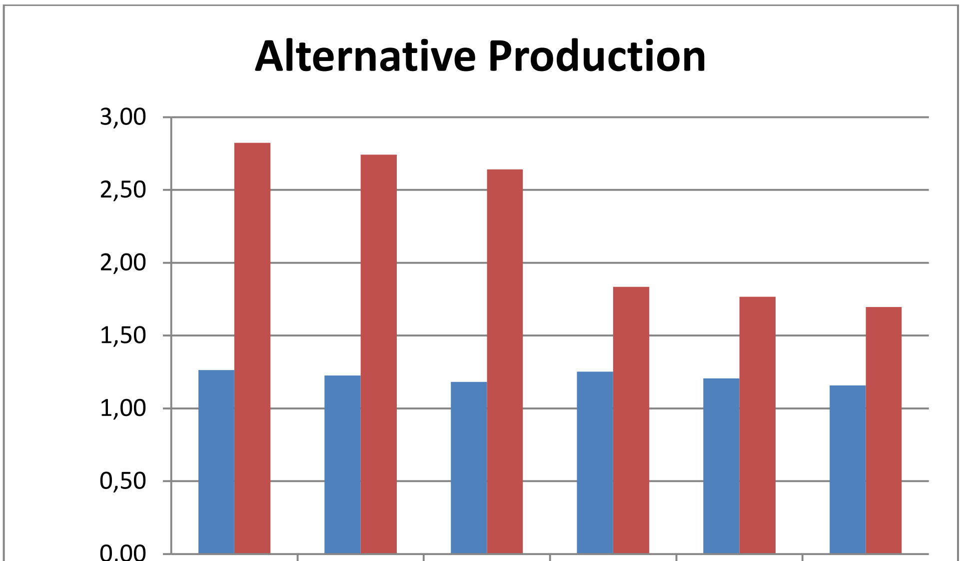
	2 MW	10 MW	25 MW	CC	CC*	Template
PEF, DH	1,40	1,38	1,32	0,94	0,85	0,73
PEF, El	2,16	2,16	2,16	2,16	2,16	2,16

FIGURE 62: PRIMARY ENERGY FACTORS, BASE CASE CONDITIONS WITH POWER BONUS ALLOCATION

The results from the base case scenario described in 5.2.1 can be seen in Figure 62. Because the Power Bonus method is used, the primary energy factor of the produced electricity is equal to the assumed avoided production, which in this case is represented by the average production within the Nordic electricity market.

For the PEF_{DH}, there is a great difference between the steam cycle plants and the rest. The steam cycle plants obtain values between 1,32 and 1,4, while all the other alternatives get values of less than unity. This shows that the power to heat ratio has a greater influence on the PEF_{DH} than the total efficiency of the plant.

7.1.2 Impact of Allocation Method



	2 MW	10 MW	25 MW	CC	CC*	Template
PEF, DH	1,26	1,23	1,18	1,25	1,21	1,16
PEF, El	2,82	2,74	2,64	1,83	1,77	1,70

FIGURE 63: PRIMARY ENERGY FACTORS, BASE CASE CONDITIONS WITH ALTERNATIVE PRODUCTION ALLOCATION

When the alternative production method is used to allocate the primary energy consumption, the result is a very different one. As can be seen in Figure 63, the PEF_{DH} values vary less than 10% between maximum and minimum value. The PEF_{El} values, on the other hand, change a lot between the steam cycle based CHP plants, and the natural gas based. This is due to the different alternative efficiency that is used for the steam cycles and the others, as stated in Table 11.

The relation between the district heating PEF and the electricity PEF is determined by the following expression that can be derived from the equations in section 4.4.5 and 4.4.6, assuming that the impact from pumps and C&D of pipes are negligible:

$$\frac{PEF_{DH}}{PEF_{El}} = \frac{\eta_{alt.elec}}{\eta_{alt.heat}\eta_{dh}} \tag{45}$$

The chosen alternative production efficiencies do therefore have a great influence on the results.

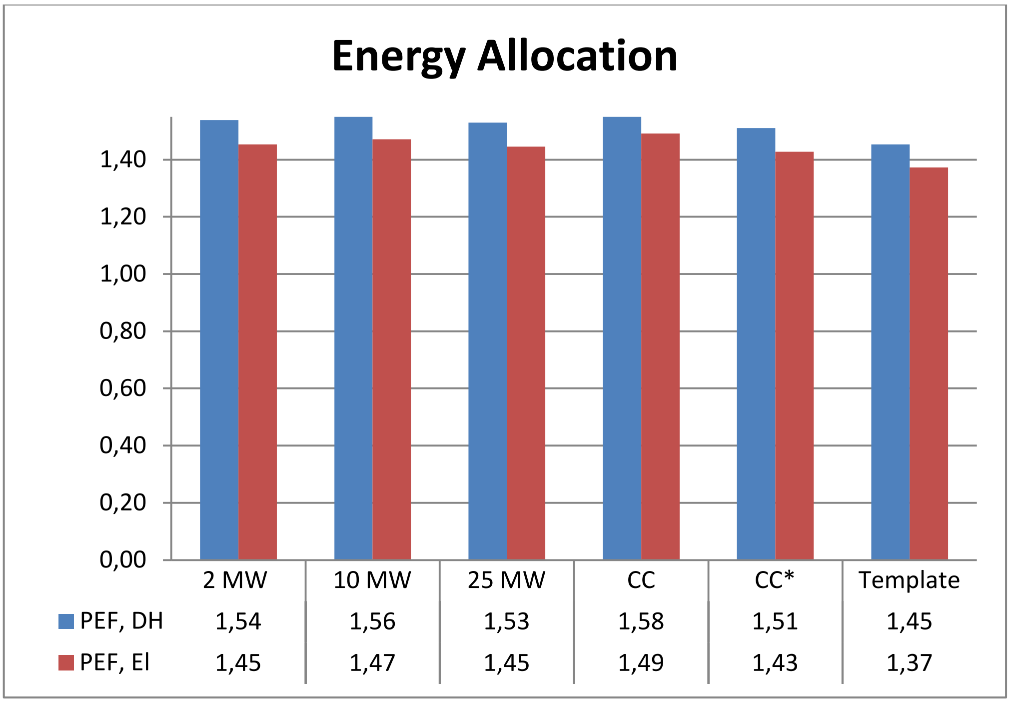


FIGURE 64: PRIMARY ENERGY FACTORS, BASE CASE CONDITIONS WITH ENERGY ALLOCATION

Figure 64 shows how the PEFs vary, and when compared to the values of total annual efficiency for year 1 in Figure 54, the changes in the PEFs correspond to the differences in total efficiency. The PEF values for the CC* and Template alternative is, however, lower than for the steam cycles, which have higher efficiencies. The reason for this is that the CC, CC* and Template alternative uses natural gas as fuel, and this option has a much lower primary energy factor for the fuel chain.

Based on this we can conclude that when energy allocation is used, the total efficiency of the cycle becomes the most important parameter, followed by the PEF value of the fuel. The power to heat ratio is of no importance.

It can be noted that the ratio between the PEFs are quite similar. In fact, the ratio is determined by the efficiency of the district heating network. By manipulation of the equations in section 4.4.5 and 4.4.6, assuming that the impact from pumps and C&D of pipes are negligible, it can be shown that:

$$\frac{PEF_{DH}}{PEF_{El}} = \frac{1}{\eta_{dh}} \tag{46}$$

If the efficiency of the district heating network approaches 1, the primary energy factors for heat and electricity will be equal.

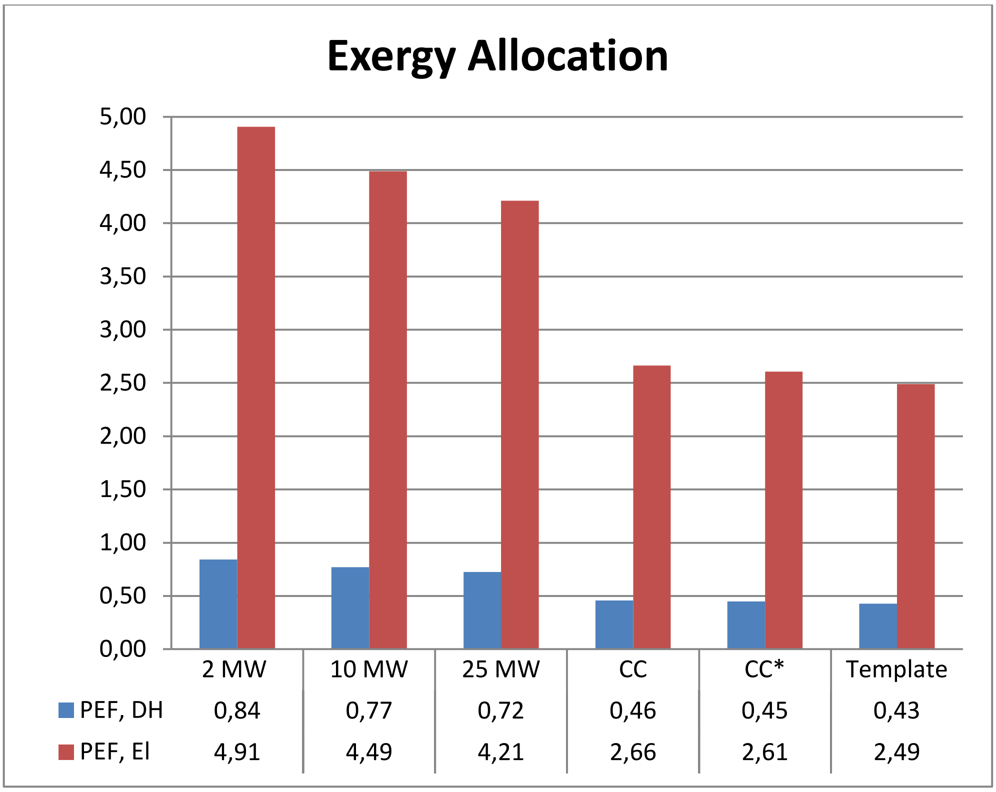


FIGURE 65: PRIMARY ENERGY FACTORS, BASE CASE CONDITIONS WITH EXERGY ALLOCATION

The PEFs that result from the use of exergy allocation can be seen in Figure 65. As electricity is pure exergy while the temperature levels in the district heating network results in a low energy quality for the district heating, this allocation method gives in a very high PEF value for the electricity produced.

The ratio between the district heating PEF and electricity PEF is, however, equal for all the technologies, and is expressed by equation (47)when assuming that the impact from pumps and C&D of pipes are negligible:

$$\frac{PEF_{DH}}{PEF_{El}} = \frac{\eta_{dh}}{\left(1-\frac{T_0}{T_m}\right)} \tag{47}$$

The expression in the denominator on the right hand side of the equation expresses the energy quality of the district heating.

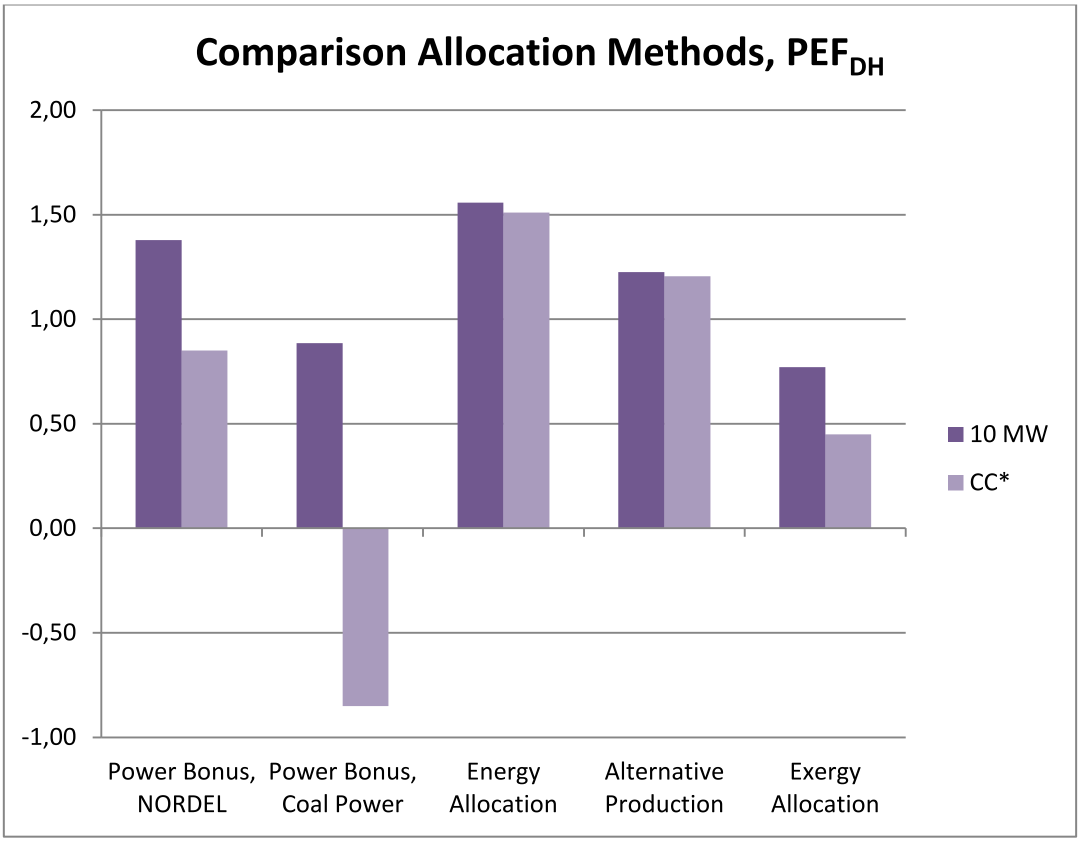
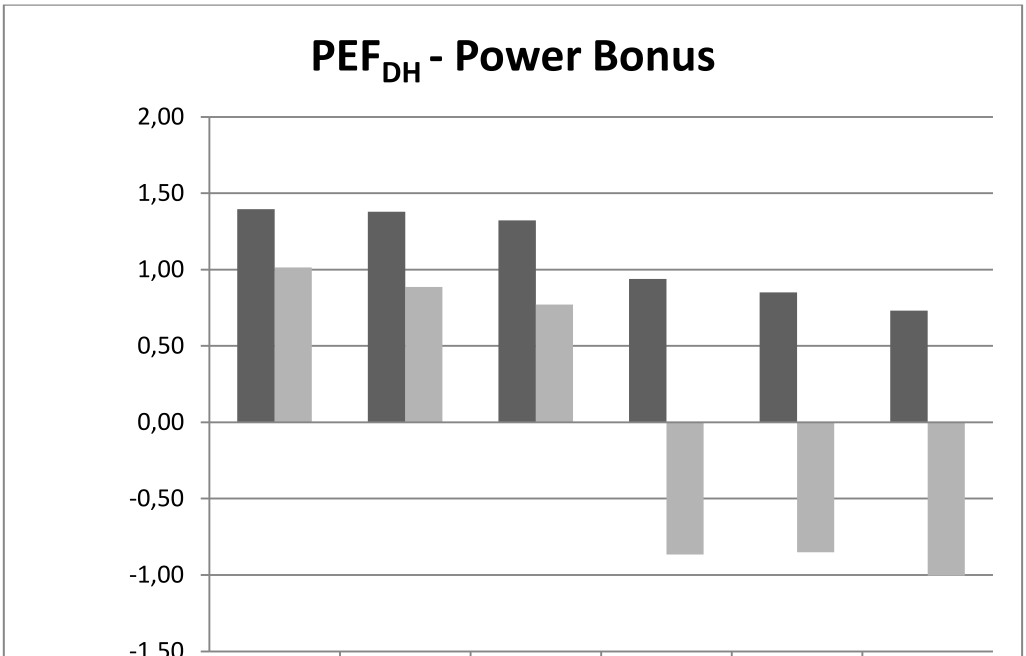


FIGURE 66: COMPARISON ALLOCATION METHODS

Figure 66 gives an overview over the district heating primary energy factors for the 10 MW steam cycle and the CC* alternative when using different allocation methods. The difference is notable for both technologies, but even more substantial for the CC* alternative. For the 10 MW steam cycle, the PEF varies from 1,56 with energy allocation to 0,77 with exergy allocation. The CC* values vary from 1,51 with energy allocation to -0,85 when the power bonus method is used with coal power as the avoided production.

7.1.3 Impact of Alternative Electricity Production

The electricity production chosen is of high importance when the power bonus method is used. Figure 67 visualises the differences between an average production scenario and a marginal production scenario.



	2 MW	10 MW	25 MW	CC	CC*	Template
NORDEL	1,40	1,38	1,32	0,94	0,85	0,73
Coal Power	1,02	0,89	0,77	-0,87	-0,85	-1,00

FIGURE 67: COMPARISON OF RESULTS, POWER BONUS METHOD WITH DIFFERENT ELECTRICITY MIX

In the average scenario, the avoided production is assumed to be represented by the average production within the Nordic electricity market, NORDEL. In the marginal scenario, it is assumed that the CHP plant power production replaces the marginal production in the Nordic electricity market, which is assumed to be coal power.

The impact is more dominant for technologies with a high power to heat ratio.

7.1.4 Dominance Analysis

In this section, the results presented in section 5.2.2 are repeated. The difference is that this time the graphs are broken down into processes, and only the losses are included, not the delivered energy. This results in graphs were one can study and analyse what processes that contribute more to the primary energy losses along the energy supply chain.

The graphs marked DH in Figure 68 represent the primary energy losses allocated to DH per kWh of district heating delivered to the user. If 1 kWh is added, one obtains the PEF_{DH}. The graphs marked EL represent primary energy losses allocated to the electricity per 1 kWh produced electricity from the plant. If one adds 1 kWh to this number, the result will be the PEF_{EL}. The SUM columns represent the sum of primary energy losses for 1 kWh delivered district heat and 1 kWh produced electricity. Note that the process that is named “Combustion” represents all the losses related to the CHP process, not only the losses during combustion.

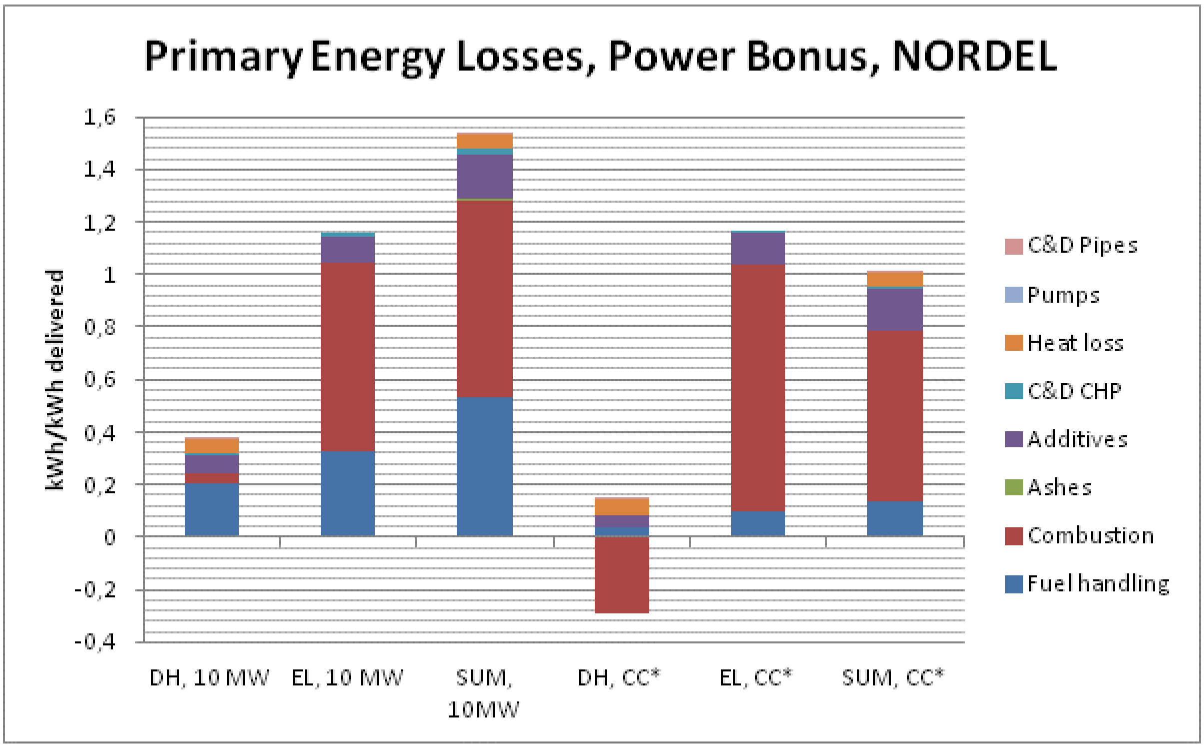


FIGURE 68: PRIMARY ENERGY LOSSES, POWER BONUS METHOD, NORDEL EL-MIX

The use of the power bonus method results in the distribution of primary energy losses that can be seen in Figure 68. As expected, the sum of primary energy losses for electricity is 1,16 in both alternatives. This is because the PEF_{EL} is set equal to the PEF for the NORDEL mix, which is 2,16.

For the 10 MW plant, the main part of the energy losses for the district heating PEF is the fuel handling. The second largest contribution comes from the use of additives in the combustion, while the third largest comes from heat loss in the district heating pipes. Construction and dismantling of the CHP plant add a minuscule contribution, while the impact of transport of ashes and construction and laying of pipes are negligible.

The sum of the primary energy losses shows that the combustion is the largest contributor for both technologies. For the 10 MW plant, the fuel handling and additives come second and third. For the

CC* alternative, the order is opposite. For both technologies, the loss in the district heating pipes comes in fourth place, and the construction and dismantling of the CHP plant in fifth.

Due to the use of power bonus method and the vast difference in PHR, the sum of primary energy losses is lower for the CC* alternative than for the 10 MW steam cycle plant.

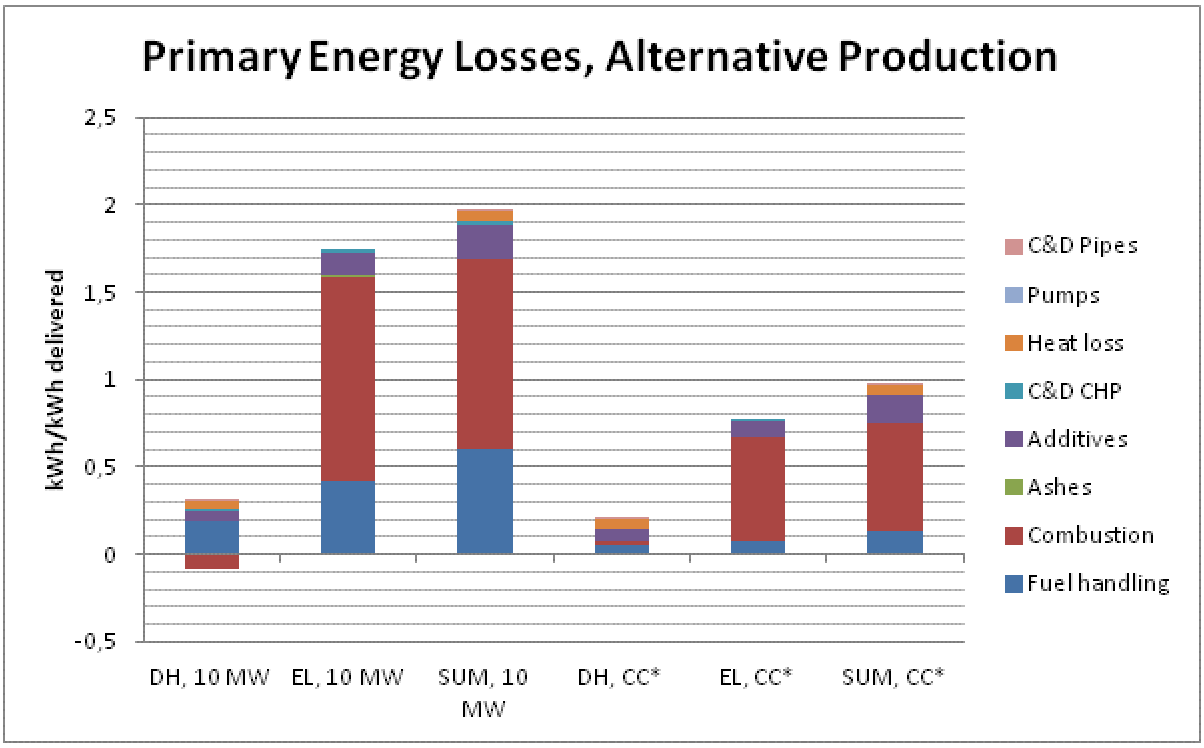


FIGURE 69: PRIMARY ENERGY LOSSES, ALTERNATIVE PRODUCTION METHOD

For the 10 MW plant, the use of alternative production allocation results in a primary energy input to the CHP plant related to DH production that is smaller than the amount of produced district heating. Therefore, the primary energy loss connected to the combustion process becomes negative for the district heating. Fuel handling is again the largest loss contributor, while the use of additives and heat loss from the DH network constitutes most of the remaining losses. Again, construction and dismantling of pipes and pump work is not even visible in the graph, while construction and dismantling of the CHP plant has a contribution of 3,5% of the total primary energy losses.

In sum, combustion, fuel handling and additives are the largest loss contributors.

For the CC* plant, the alternative production efficiency for electricity is higher, and therefore there is no negative PE loss related to the combustion process for the district heating this time. The contribution from fuel handling is lower due to a more efficient fuel handling process with a lower PEF. Construction and dismantling of the CHP plant still has a negligible contribution.

The sum shows that in this case the additives has the second largest contribution to PE losses, and that the combustion process represents a far larger fraction of the total losses.

Also, the sum of primary energy losses are far lower for the CC* alternative than for the 10 MW alternative. This is because the alternative power production efficiency is set far higher for the combined cycle than for the steam cycle.

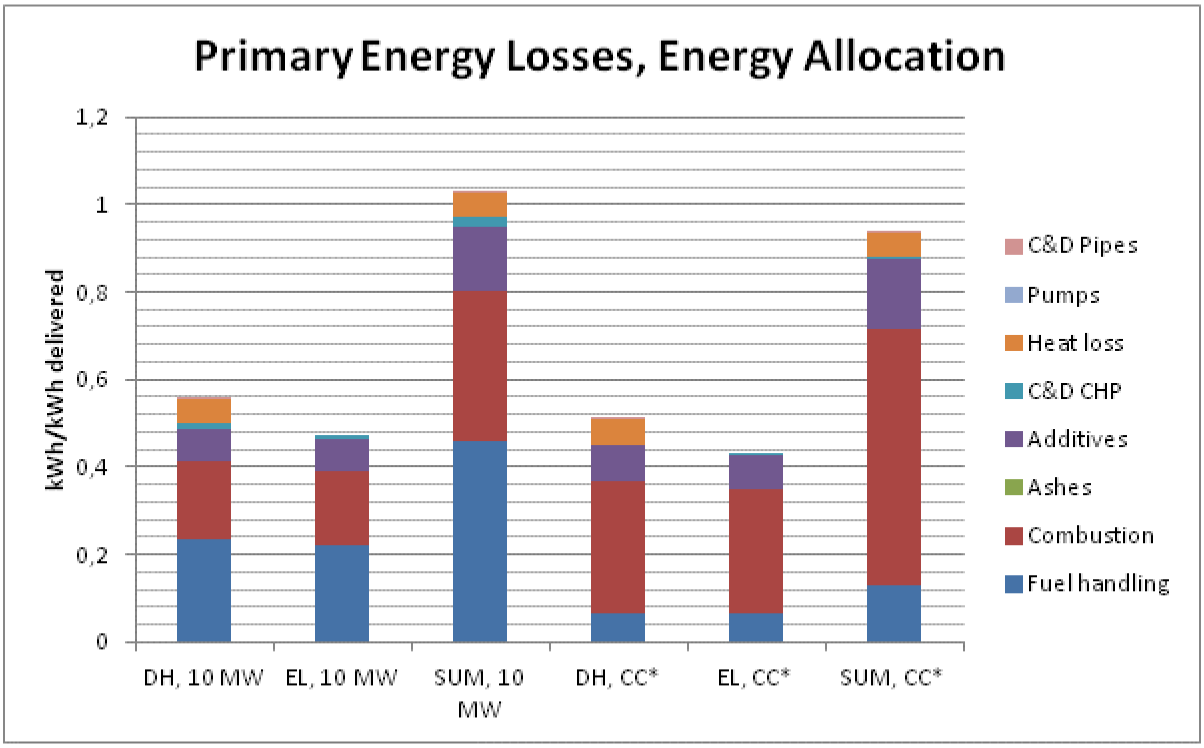


FIGURE 70: PRIMARY ENERGY LOSSES, ENERGY ALLOCATION

When energy allocation is used, the primary energy losses are distributed relatively equally between the district heating and the electricity, as can be seen in Figure 70. This is because the energy losses from the common processes are distributed so that the district heating accounts for $1/\eta_{dh}$. On top of that, the district heating specific heat loss is added.

The sum columns show that also here the CC* alternative get a lower primary energy loss in total. This is due to the lower PEF of natural gas compared to Wood chips. If the fuel had the same PEF, the primary energy losses would be far higher for the CC* alternative than for the steam cycle, due to a lower total efficiency. As a consequence of the low efficiency, combustion is by far the largest contributor to primary energy losses in the CC* plant.

7.1.5 Impact of Fuel

In section 6.3, results were presented on how the 10 MW steam cycle plant performed when the fuels listed in “Table 8: Fuel properties, from STEAM Pro” was used.

These results were implemented in the excel calculation tool, so that the annual total efficiency and PHR of the plant was altered.

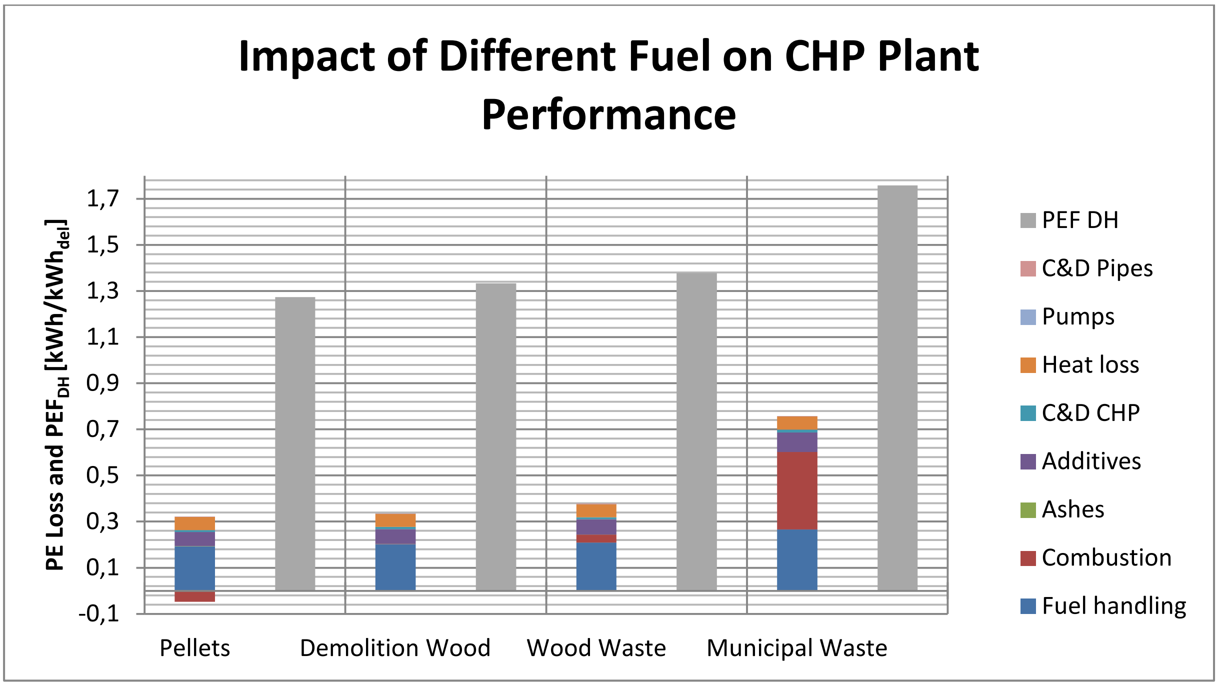


FIGURE 71: IMPACT OF FUEL TYPE ON CHP PERFORMANCE AND PEF DH, POWER BONUS METHOD

The pellets alternative gives a PEF_{DH} of 1,27, Demolition Wood 1,33, Wood Waste 1,38 and Municipal Waste 1,76.

In the results displayed in Figure 71, the PEF of the fuel is held constant at 1,19, which is the PEF for Wood Chips. In other words, different fuels are used in the CHP simulations in STEAM Pro, while the same fuel, Wood Chips, is used in the excel tool.

It is of course not fair to compare the different CHP plant performances without taking into account the varying PEF values for the corresponding fuel chain.

Figure 72 compares the previous results with the results obtained when the fuel input in the excel calculation tool is chosen so that it corresponds to the fuel used in the STEAM Pro simulations. The primary energy factors for the different fuels are listed in Table 12.

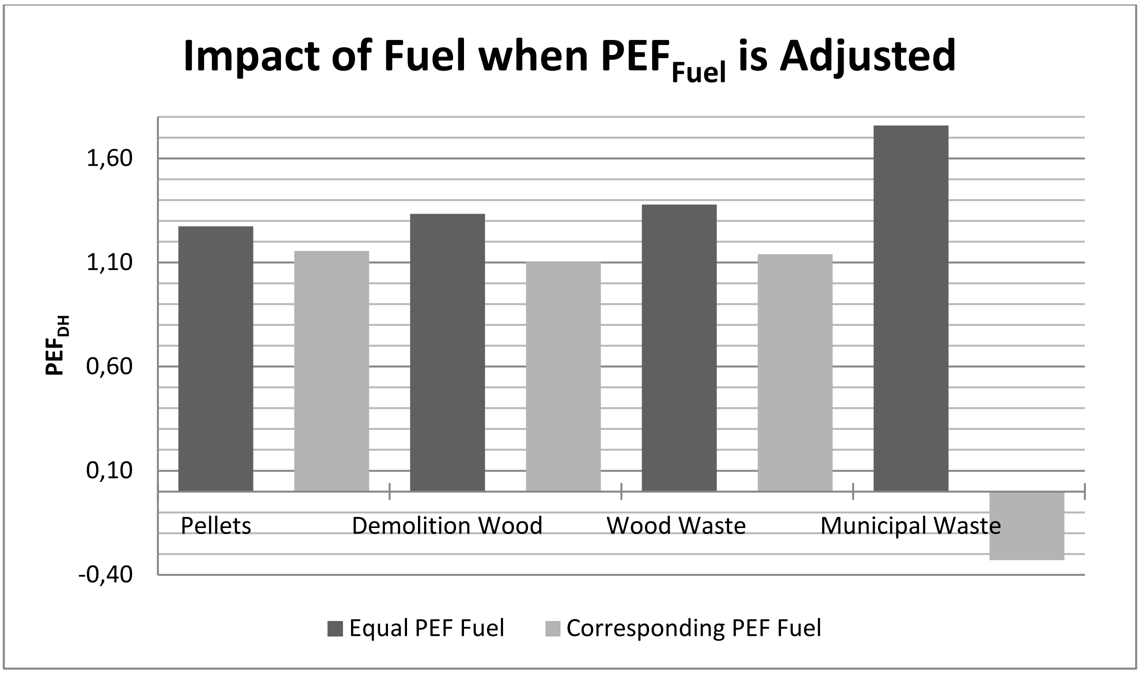


FIGURE 72: IMPACT OF FUEL, CORRESPONDING FUEL CHAIN PEF

For pellets, demolition wood and wood waste, the differences in CHP efficiency is now counteracted by the primary energy factor of the corresponding fuel. This results in quite similar PEF_{DH} values for the three fuels.

When it comes to municipal waste, the difference between the two cases is considerable, mainly due to the use of the "polluter pays" principle for municipal waste as fuel.

7.1.6 Impact of Fuel PEF

In Figure 73: Impact of different PEF fuel values, 10 MW plant, Power Bonus and Energy allocation, the CHP efficiency is held constant at the base case value, while the fuel is changed in the excel tool, and by this also the primary energy factor of the fuel.

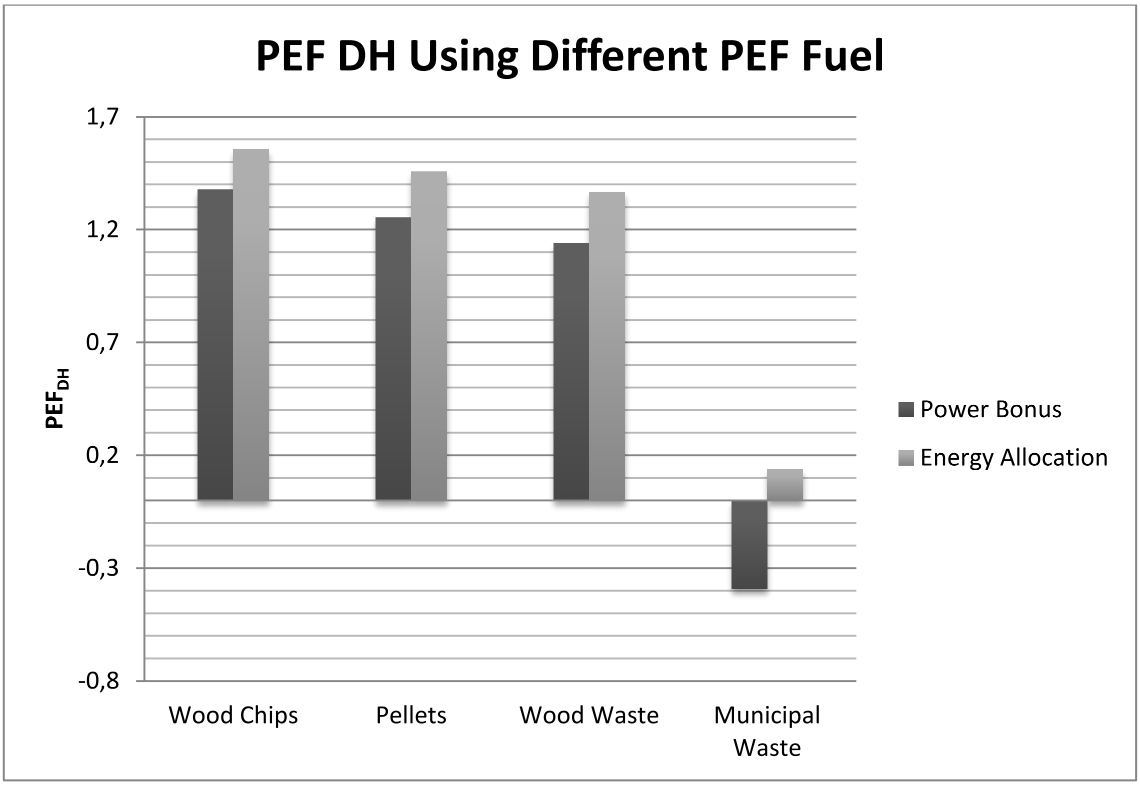


FIGURE 73: IMPACT OF DIFFERENT PEF FUEL VALUES, 10 MW PLANT, POWER BONUS AND ENERGY ALLOCATION

The PEF_{DH} is influenced considerably by changing the primary energy factor of the fuel. By replacing wood chips with wood waste, the PEF_{DH} decreases 17% when power bonus allocation is used, and 12% with energy allocation.

Municipal waste stands out from the rest, with a negative value when power bonus allocation is used, and a very low value of 0,14 when energy allocation is used.

7.1.7 DIFFERENT LOAD DISTRIBUTION

In Figure 74, the impact of the different annual load distribution patterns presented in Table 9 is shown.

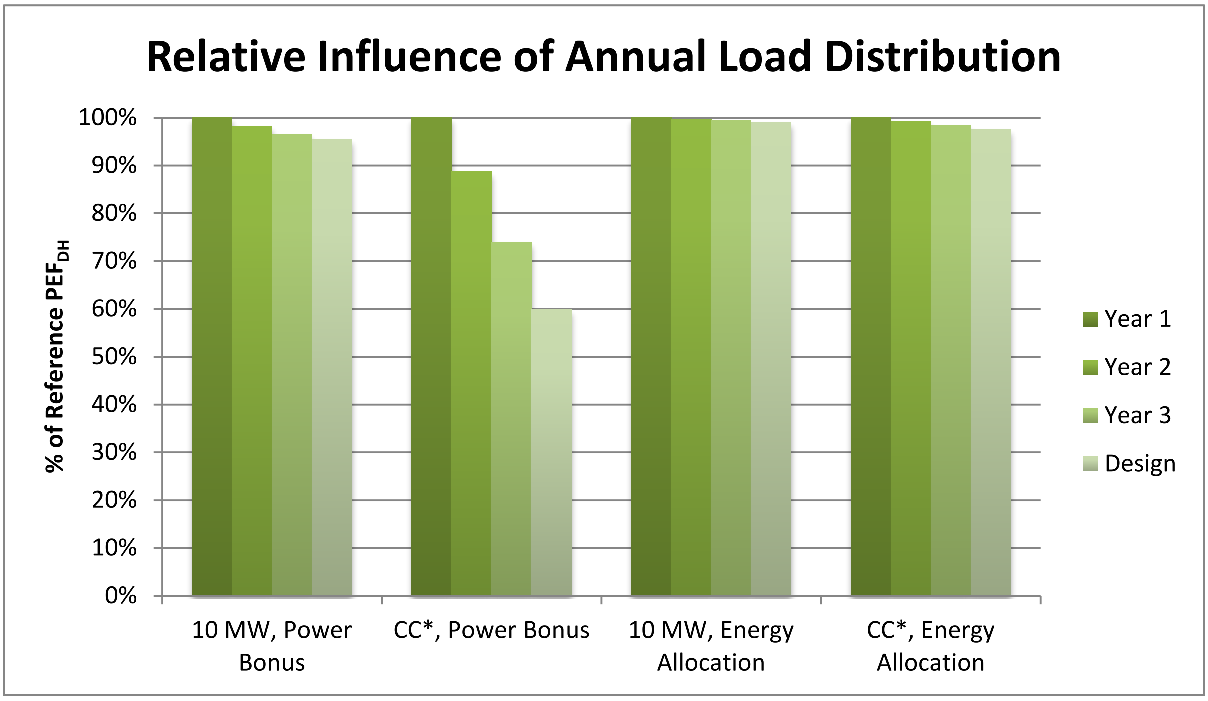


FIGURE 74: IMPACT OF DIFFERENT ANNUAL LOAD DISTRIBUTION PATTERNS

The first column, Year 1, represents the reference annual load distribution that has been used in the base case. Year 2 and 3 represents alternative load distributions where the plants are run less on part load. The last column, "Design", shows the PEF_{DH} relative to Year 1 if the plant were to be run on 100% load 8760h per year.

For the 10 MW plant with power bonus allocation, the annual load distribution pattern has a small influence. The PEF_{DH} is only 4 %-points lower for the design case compared to the Year 1 case.

In the CC* alternative, however, the PEF_{DH} is greatly influenced by the annual load pattern. The difference between Year 1 and Year 3 is 26 percent points, and between Year 1 and Design entire 40 percent points. This is mostly due to the considerable change in power to heat ratio, that was presented in Figure 56, but the change in total efficiency presented in Figure 54 also plays a role.

When energy allocation is used, the changes between the different annual load distribution scenarios are minuscule. This is true for both the 10 MW plant and the CC* plant.

7.1.8 Energy Density

Figure 75 shows the PEF_{DH} for different energy densities.

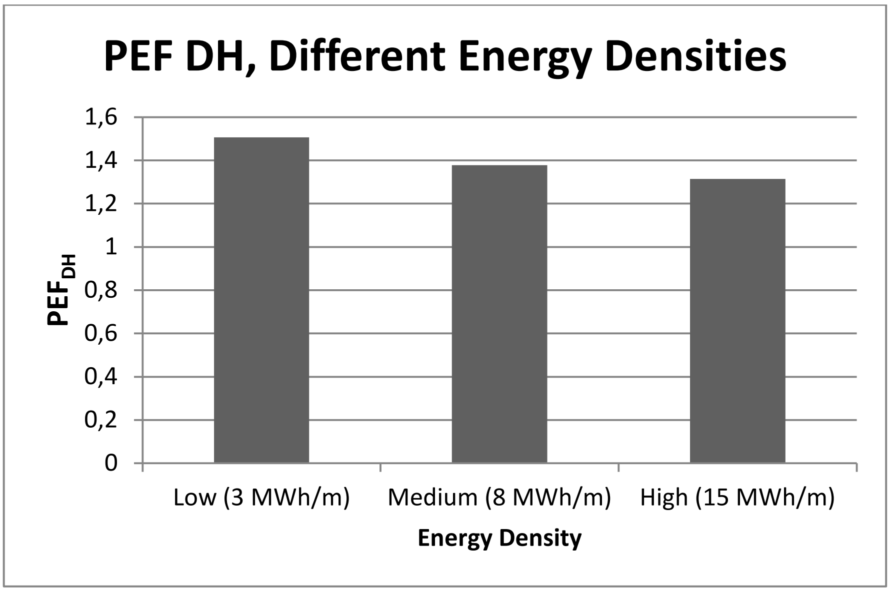


FIGURE 75: IMPACT OF DIFFERENT ENERGY DENSITIES REGARDING THE DISTRICT HEATING NETWORK

The medium value is the same as in the base case, 1,378. When the energy density is low, the PEF_{DH} increases 9,4% to a value of 1,507. With a high energy density, the PEF_{DH} decreases 4,7% to 1,314.

The distribution of energy losses allocated to district heating change considerably from alternative to alternative. The distribution of energy losses for the different energy densities are visualised in Figure 76, Figure 77 and Figure 78.

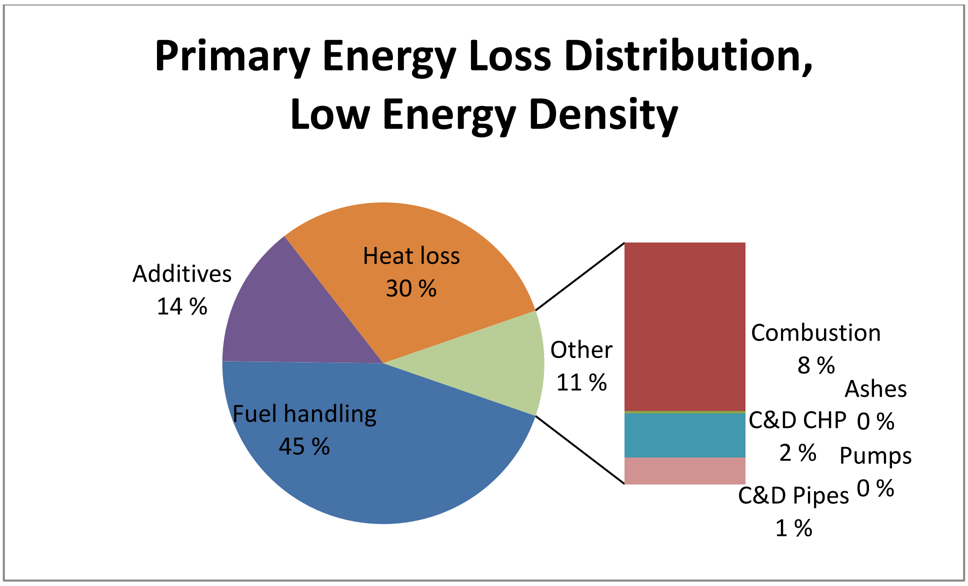


FIGURE 76: ENERGY LOSS DISTRIBUTION, LOW ENERGY DENSITY

In the low energy density case, the heat loss accounts for 30 % of the total losses. Fuel handling and additives account for the second and third largest losses, while the impact from ashes and pumps are negligible. Construction and dismantling of the CHP plant and the district heating pipes account for respectively 1 % and 2 % of the total.

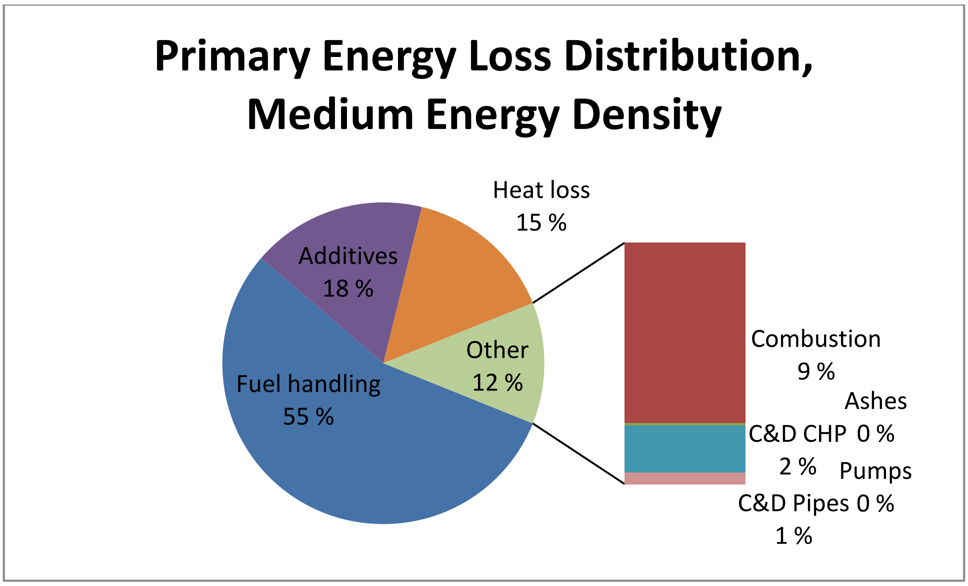


FIGURE 77: ENERGY LOSS DISTRIBUTION, MEDIUM ENERGY DENSITY

When the energy density is increased from 3 to 8 MWh/m, the heat loss is severely reduced. The relative impact from the heat loss is now 15%, which is a 50% decrease relative to the low density scenario. The impact from fuel handling and additives now account for a larger share of the total energy loss than in the first scenario.

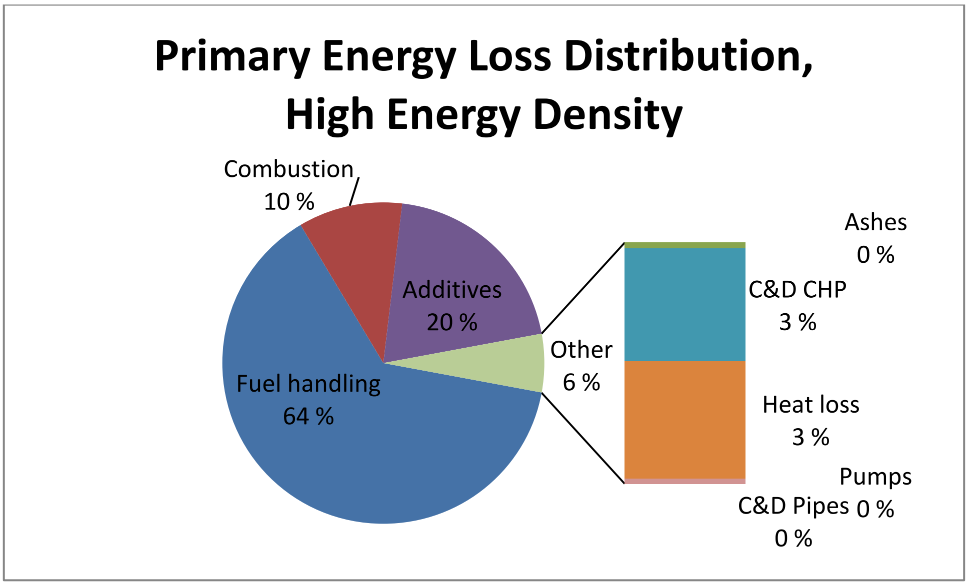


FIGURE 78: ENERGY LOSS DISTRIBUTION, HIGH ENERGY DENSITY

The high energy density scenario results in an even more notable change in the heat loss fraction. In this scenario the heat loss only accounts for 3% of the total energy losses. The additives and fuel handling account for a still larger part of the losses than in the previous scenarios, while the impact from construction and dismantling of pipes now account for approximately 0 % of the total energy loss. This is because the impact from the building of pipes now is distributed on more units of delivered energy throughout its life time.

The energy density thus has a considerable impact on the PEF_{DH}.

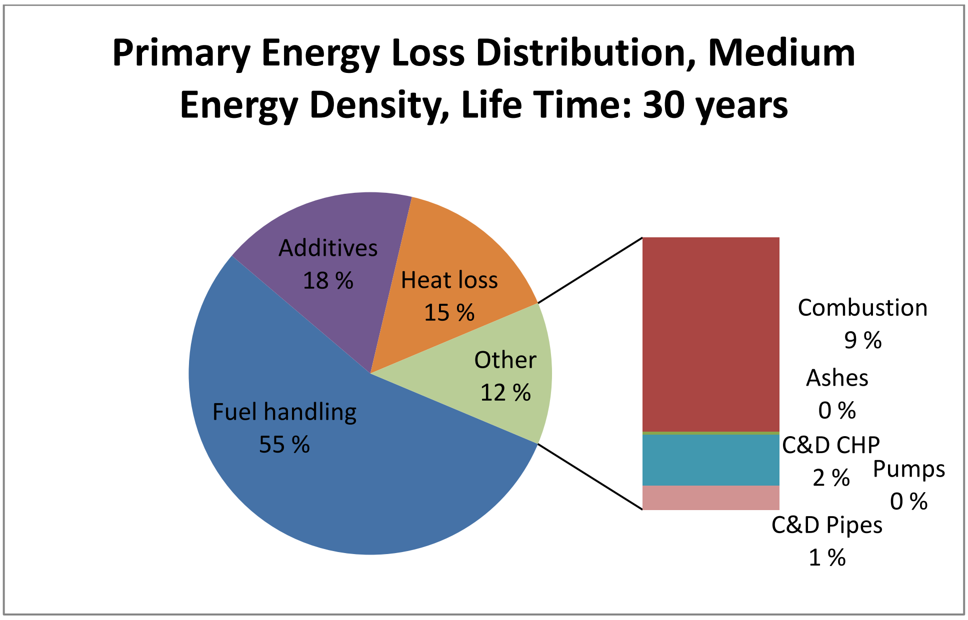


FIGURE 79: ENERGY LOSS DISTRIBUTION, MEDIUM ENERGY DENSITY AND PIPE LIFE TIME 30 YEARS

Figure 79 shows the energy loss distribution when the assumed life time of pipes is reduced from 60 to 30 years. The C&D Pipe loss do increase from 0,002126 kWh/kWh_{del} to 0,0042 kWh/kWh_{del}, which is a 97% increase. This does, however, not influence the overall picture. There is no visible change compared to Figure 77, where a life time of 60 years was assumed.

8 DISCUSSION

8.1 CHP SIMULATIONS

8.1.1 OPERATION

The simulations of the CHP plants were performed at full load and two part load levels according to a simplified DH load duration curve. This is not necessarily a realistic way of operating a CHP plant. During the first part of the project period, I contacted many of the Norwegian CHP facilities to ask about design, technical details and operation of their CHP facility. The conclusion from this survey was that it is difficult to say something very general about how CHP plants connected to district heating networks are designed and operated, as they all have different premises. Some, as for example Fredrikstad Bio-el, deliver steam to industries in addition to district heating. When the heat demand is high, the steam turbine is turned completely off to be able to provide enough heat. Mosseporten delivers both district heating, industry steam and heat to provide cooling. BIR has to cool off heat during the summer due to the constant inflow of waste. In the preliminary project for Ranheim heating central, the CHP facility is planned to operate for only approximately 4800h per year, and it is not possible to see how large load variations the plant is expected to have.

Real CHP plants are operated according to many factors that have not been taken into account in this work. If the plant delivers heat to more than one purpose, and if it is connected to the electricity grid, the energy prices for fuel, electricity and alternative heat production will for instance influence the operation.

Maybe even more important is the role of the CHP plant in the district heating network: Is it a base load plant, as all the waste incineration plants? Or is it a plant that is supposed to run from autumn through winter until spring, like the Ranheim plant is scheduled for? In other words, it is very difficult to say something general on how large DH load variations a CHP plant can expect to have.

The district heating load duration curve used was considerably simplified compared to a real load duration curve. The information gathered in the first phase of this project showed that none of the Norwegian CHP DH plants followed the district heating load hour by hour. The choice of DH load duration curve is therefore a simplification that can be justified.

8.1.2 DIMENSIONING AND MODELLING

Optimal dimensioning and design of a CHP plant is a complex task. It has not been within the scope of this project to investigate the dimensioning process in detail, but to run different technologies at equal load patterns to be able to compare their behaviour. Therefore STEAM Pro and GT Pro was used to do the initial plant design and dimensioning. In these programs, reasonable assumptions regarding initial values and efficiencies of the components are taken by experts with vast experience within this field. Furthermore, real components with realistic characteristics and standard sizes are utilised. This gives the models a higher degree of realism than what could have been achieved if the models had to be created from scratch.

The drawback with this method is that the equations and reasoning behind the design is more or less hidden for the user. This was clearly demonstrated in section 6.5, where different design temperatures were studied. In this case it was difficult to find the source of the sudden change in behaviour of the power efficiency. Even though an explanation could be found be investigating the vast amount of output, it was still not evident exactly *why* this change had occurred exactly where it did.

The more realistic the design and simulation is, the more effects are included. This can in some cases make it difficult to distinguish the effect of what you are trying to investigate, in this case the supply and return temperatures, from other secondary effects. If further design studies on different temperatures are to be made using STEAM Pro, it should be investigated even more in detail what parameters that should be held constant. In the case of supply and return temperatures, maybe the results would be more representative if the auto efficiency function for the steam turbine was turned off and replaced by adequate and equal efficiencies for all return and supply temperatures. It could, however, be that this solution would give less realistic results. Further investigations are necessary to reach a conclusion on what would be the better option.

8.1.3 Economic Aspects

Economy is maybe the most important aspect in both the operational and dimensioning phase. Higher efficiency means higher investment costs but also lower operational costs. The cost-optimal size of a plant can be found through an economic-thermodynamic optimisation process, and the cost-optimal operation by intricate optimisation algorithms that include the prices in the energy markets. In this work, the simulation program was instructed to design for higher efficiency, neglecting the economic implications.

8.2 Primary Energy Factors

8.2.1 Definition of System Boarders

The definition of primary energy factors is ambiguous. As mentioned in chapter 3, the definition in EN 15603 states that some elements are compulsory while others are optional. As there exists no clear guidelines on what to include, very different values might be obtained. One example is the sub process “Additives”. This process is not mandatory to include according to the standard, but as it represents a rather large share of the primary energy losses (Ref. section 7.1.4) it would give quite different results if it was not included. The ambiguity regarding system boarders makes crucial that all primary energy factor calculations are done in a transparent way.

In the NS-EN 15316-4-5 standard, rules for calculating the primary energy factor for district heating systems with combined heat and power plants are explained in more detail. Here, only the mandatory elements are included in the calculations. A consequence of this is less room for uncertainties regarding the results, as less data need to be collected.

In general, the more processes one includes in an analysis of a system, the more uncertain the results become. When the goal is to get a holistic picture of the system, however, it is important to study the relative influence of sub processes before they are excluded from the studies. If sub processes are excluded before we know the approximate size of their impact, the primary energy factor will not give a correct picture of the real primary energy consumption.

8.2.2 Definition of Total Primary Energy Factor

The definition in EN 15603 and NS-EN 15316-4-5 states that the total primary energy factor always shall exceed unity. The use of the power bonus method on a CHP district heating network does, however, in many cases result in a district heating primary energy factor of less than one, in some cases even negative. This is also the case in the examples that are included in the Annex of NS-EN 15316-4-5.

In this work, the results have been presented as they are, even if the PEF has been less than one or negative. This is because the main goal has been to study the impact from different parameters, and these would be hidden if all PEFs that were less than unity or negative were set to one. Still, the definition and the examples provided in NS-EN 15316-4-5 give two different signals on how PEFs that are lower than unity should be handled.

8.2.3 Data Quality

The data quality is of high importance when systems are studied using life cycle analysis. In this project, most of the data collecting was done by [16], which is judged to be a reliable source. Still, there are many parameters that influence, and the author has not been able to search through relevant literature to check the values against other studies.

Some small additions have been done regarding electricity and fuel primary energy factors. This information has been found in published material from acknowledged institutions.

There are, however, always uncertainties connected to the input data. Especially the values regarding the primary energy factor of the fuel is sometimes difficult to compare because there are no general rules on what processes that should be included within the system boarders.

The results show that the additives influence the results notably. In the calculations, the same amount of additives is included for all fuels. This is probably not correct, as natural gas has very different combustion characteristics than for instance wood, waste of pulverised coal.

8.2.4 Coherency Between CHP Simulations and PEF Calculation Tool

When the annual load distribution is altered, this again changes the amount of time that the corresponding temperatures occur in the district heating pipes. This will again influence the annual heat loss from the pipes.

In the calculation tool, the heat loss is only calculated for one annual heat load distribution. When the other annual heat load distributions were implemented the corresponding heat loss was not changed, which would not be the case in reality. To quantify this effect, new heat loss calculations would have to be done. It is probable that the effect would be noticeable in the low energy density case because the heat loss there contributes significantly to the primary energy losses. In the high energy density case, on the other hand, the effect would probably be negligible as the heat loss only account for 3 % of the DH losses.

8.2.5 Choice of Energy Density

In the base case, a medium energy density was chosen. This was simply because this was the middle value, and thus thought to represent an average case. When this value was compared to experienced values, however, it became clear that 8 MWh/m is the double of the average energy density in DH grids in Norway in 1998, which was 4 MWh/m [53]. As the energy efficiency requirements for buildings have become stricter since then, it is not very likely that the energy density for DH networks has increased the last 13 years. To get an impression of the impact a lower energy density would have, a comparison was performed. (Figure 75: Impact of different energy densities regarding the district heating network). This showed that the PEF_{DH} increased 9,4% compared to the base case. Hence more calculations should have been done using the low energy density, as this would probably represent Norwegian conditions better.

9 CONCLUSIONS

9.1 CHP PLANT PERFORMANCE

In Norway at present, there are 9 waste incineration steam cycle CHP plants connected to DH networks, one steam cycle that is based on demolition wood and one reciprocating engine that is running on biogas. Based on the current situation in Norway and Europe, back pressure steam cycle and combined cycle were chosen as technologies to study more in detail.

The simulations showed that the plant performance vary greatly between the technologies. At design conditions, the combined cycle had an over two times higher power efficiency and a 3,7 times higher power to heat ratio than the 10 MW steam cycle power plant. The total efficiency, however, was 10 % lower for the combined cycle than for the 10 MW steam cycle plant.

Within the steam cycle power plants, the size and boiler type influenced the performance. The tendency for the steam cycles with grate boilers was that increasing the size led to higher power efficiency, but a slightly lower total efficiency. The steam cycle with a circulating fludised bed boiler had slightly higher power efficiency than the 10 MW plant at full load, and an almost equal total efficiency. The effect of improved boiler efficiency therefore seems to have evened out the negative effect of size increase regarding the total efficiency

At part load, both the combined cycle and the steam cycles experienced severe drops in power efficiency. The effect was most pronounced for the combined cycle and the 2 MW steam cycle plant. Within the steam cycles, the plant with a CFB boiler had a less significant drop in efficiency than the grate boiler steam cycles. The impact of part load operation is not linear, and the reduction in performance was more drastic from 56% load to 26,5% load than from 100% load to 56%.

The total efficiency was not influenced as severely by part load operation as the power efficiency. For the steam cycles the total efficiency varied by less than 2 percent points, while the combined cycle had a drop of four percent points. In the scenario where it was assumed that surplus heat was cooled off at the lowest part load, the CC alternative, the total efficiency was reduced all the way down to 62,55%. This result shows that oversizing the plant affects the total efficiency severely.

Three different annual load distributions were studied using the results from the part load calculations. When the annual power efficiency and power to heat ratio was studied, the effect of annual load distribution was notable for all alternatives. For the annual total efficiencies, the effect of different annual load distributions was notable for the combined cycle alternatives, negligible for the CFB boiler steam cycle and almost negligible for the two other steam cycle plants.

Different dimensioning supply and return temperatures in the district heating network influenced the power efficiency and power to heat ratio visibly. When only the supply temperature was varied, the power efficiency was altered 3 % points, while a variation of both supply and return temperature resulted in a total change of 4,5 % points. The total efficiency showed a slight decrease with increasing temperatures in the district heating network, but the effect was small. The value of the PHR was 33% higher at the lowest temperatures than at the highest when both T_s and T_r was varied. These results are only indicative due to the discrete change in steam turbine efficiency described in section 6.5.

For the 10 MW plant, different fuels were tested to see how the performance was influenced. Pellets and Demolition wood gave a quite significant improvement of total efficiency, while the use of municipal waste plunged the total efficiency.

Due to the simplifications done regarding operation and dimensioning of the plants, it is important to underline that the annual performance results presented are not generally valid for all plants of corresponding technology and size. The full load values can, however, be considered to represent realistic values, as the simplifications do not impact on these results, and the simulation tool used provides realistic design and internal losses.

As all technologies were subject to the same assumptions, the annual values are still well suited to compare the different technologies and sizes against each other.

9.2 PRIMARY ENERGY FACTORS

The base case scenario gave PEF_{DH} values from 0,85 to 0,94 for the combined cycle based alternatives, and 1,32 to 1,4 for the steam cycle based alternatives. In this case, the power to heat ratio was the most influential CHP plant parameter due to the use of the power bonus method.

When different allocation methods were used, the 10 MW steam cycle plant got PEF_{DH} values ranging from 0,77 with exergy allocation to 1,56 with energy allocation. The CC* alternative experienced a change from 0,45 with exergy allocation to 1,51 with energy allocation. This shows how highly influential the choice of allocation method is. Even though the largest value was more than three times higher than the smallest one, it is interesting to observe that all the values were considerably below 2,16, which is the PEF value of the Nordic electricity mix.

According to EN-NS 15316-4-5, the power bonus method should be used when calculating primary energy factors for district heating. While the energy allocation method and the alternative production allocation method gave fairly similar results the steam cycles and the combined cycles, the power bonus method gave quite diverging results for the different technologies. This is because the power to heat ratio becomes very important, as all avoided impact from produced electricity can be subtracted from the total impact. This leads to that the decision on what electricity production that is assumed to be avoided, becomes decisive for the result.

When a marginal scenario was implemented, with the assumption that the marginal production in the Nordic electricity market is coal power, the results changed dramatically compared to the base case scenario. The combined cycle alternatives got negative PEF_{DH} values, and the steam cycle alternatives got values ranging from 0,77 to 1,02. The choice of electricity mix is therefore a very important parameter when the power bonus method is used.

When the distribution of primary energy losses was studied, it turned out that it varies according to allocation method and technology where the largest fraction of losses occurs. It was, however, possible to conclude that pump work, ash transport and construction and dismantling of district heating pipes all give negligible contributions for the cases studied. This leaves combustion, fuel handling and additives as the main contributors to energy loss, while heat loss from the pipes in general accounted for a smaller part of the total loss, but a considerable part of the district heating loss. The results should, for the reasons explained in the discussion, be reviewed and compared to results where a lower energy density is used.

The heat loss increased dramatically when the energy density was decreased from 8 MWh/m to 3 MWh/m; from 15% to 30% of the primary energy losses related to the district heating. When the energy density was assumed to be 15 MWh/m, the heat loss only represented 3% of the energy losses related to the district heating. Within the scope of this project, it was not possible to adjust the heat loss and pump work to the new annual load distributions. Therefore these results are only indicative. Still, the results indicate that the energy density have a notable influence on the $PEF_{DH.}$

The primary energy factor of the fuel has a great influence on the PEF_{DH}. By replacing wood chips with wood waste, this alone reduced the PEF_{DH} with 17%. The fuel handling process is thus plays a very important role when it comes to increasing primary energy efficiency of the DH energy supply chain.

The 10 MW plant was simulated in STEAM Pro with different types of fuel, and the effect on the total efficiency was notable. This effect was, however, almost completely counterbalanced when the corresponding PEF fuel value was used to calculate the PEF_{DH}. The lesson is that if the aim is to improve the PEF_{DH}, a very efficient CHP plant is of little use if this plant is based on fuel that is notably more energy consuming to produce than the fuel needed for a less efficient plant.

Three different annual load distribution patterns were used to observe the importance of part load operation of the CHP plant. It turned at that the impact depended a lot on what allocation method that was used. When energy allocation was used, the difference in primary energy factor was nearly negligible. When the power bonus method was used, the change in power to heat ratio at part loads resulted in a large variation for the CC* alternative. Again, this shows that use of the power bonus method makes the PEF_{DH} extremely sensitive to variations in the power to heat ratio. The 10 MW plant, with a far smaller change in power to heat ratio during part load, only got a minor change in PEF_{DH} when the annual load patterns were altered.

If the use of primary energy factors becomes mandatory in Norway as a consequence of the directive on the energy performance of buildings, it is necessary that an unambiguous framework with consistent system boundaries is developed. If some processes continue to be optional, like additives, it will be difficult to compare values for different plants. A relevant conclusion from this project is that construction and laying of DH pipes, pump work, transport of ashes and construction and dismantling of the CHP plant are negligible. This should of course be verified by more studies, but it seems like these sub processes can be left out of the PEF_{DH} calculations without large consequences.

Other aspects that would need to be defined are what primary energy factor that should be used for electricity and the different fuels in question. Furthermore, the results from this project clearly show that it is important that values on total efficiency and power to heat ratio are set according to the CHP technology and operational situation in question.

9.3 Further Work

9.3.1 CHP Plant Performance

The results from the simulations indicate that the size of the CHP plant influences the plant performance. More sizes should be studied to quantify this effect better. It also turned out that boiler type had a quite significant effect on part load performance, and this effect should also be studied further.

In this study, only steam cycles and combined cycle were studied. As presented in the CHP theory chapter, there are many more interesting technologies. As an example, natural gas engines are somewhat relevant in a Norwegian context. In the future, fuel cells might be an option.

The supply and return temperatures were only studied at design conditions. In future work, also the corresponding part load conditions should be studied to get a more complete picture. Then it would be possible to use the results in the calculation of primary energy factors. As the impact of DH temperatures was significant when regarding the power efficiency and PHF, lower temperatures would surely influence the PEF_{DH} in a positive direction. Furthermore, heat losses would also decrease, which in this study was a notable part of the district heating related PE losses.

More design temperature simulations should nevertheless be done to see how the results would be with a constant steam turbine group efficiency.

Economy and operational strategy are two important aspects that were ignored in this project. This is a vast field, and surely the inclusion of economic aspects might give different results. One effect could be that the power efficiency of the CHP plant decreases because it is very expensive to build state-of – the –art efficient plants. Another possibility could be that the power efficiency increased, because economic criteria would result in less part load operation.

In this project, simulations were carried out to obtain annual values for total efficiency, power efficiency and power to heat ratio for different technologies. As discussed, this is a difficult task because dimensioning, design, delivery of other energy products and other differences related to site specific conditions vary from plant to plant. As a consequence, plants with the same technology and similar size can have widely different performance during the year.

For this reason it might be a better idea to gather data from relevant plants that are already running. As described in the CHP-chapter, in Norway it does actually not exist a single cogeneration plant that is not a waste incineration plant, which only delivers district heating, and no other energy products in addition, as process steam or cold. It would therefore be necessary to utilise Swedish and Danish data.

9.3.2 PRIMARY ENERGY FACTOR CALCULATIONS

The impact of additives should be further studied to confirm that the influence actually is as important as it seems.

Through all the cases, the heat loss from the district heating pipes was calculated according to the temperatures and load pattern of the base case. To get more accurate results, it should be implemented a possibility in the excel tool to adjust the heat loss to the actual load pattern and district heating temperatures.

As mentioned in the discussion, more calculations should be performed utilising the low energy density, as this setting is closer to the Norwegian reality.

9.3.3 PRIMARY ENERGY FACTOR CALCULATION METHODOLOGY

The primary energy factor should be a tool for reducing the overall primary energy consumption. It is important that we avoid a situation where methodological choices hide the real potential for improvement.

The principle for the power bonus method is easy: Electricity produced by the CHP displaces production somewhere else, and due to market mechanisms it will be the marginal electricity production that is avoided. Another market mechanism is, however, that higher supply shifts the supply curve so that prices are reduced and the consumption increases. It is therefore not evident that all the electricity production from the CHP plant actually is avoided somewhere else. It is also not completely straight forward to decide what the PEF of the marginal production is. The consequence of this might be that instead of a holistic PEF_{DH} value that represents the real primary energy burden of producing district heating, we end up with a value that is not related to physical realities.

By analysing the power flow in Europe, it could maybe be possible to make a more accurate model on the amount of electricity production avoided on the continent as a consequence of CHP electricity production in Norway. It would also be necessary to calculate the average PEF of this electricity production. These developments would bring the PEF methodology a large step forward.

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APPENDIX A

BACKGROUND INFO FOR CHAPTER 4: METHODOLOGY

GAS TURBINE DETAILS

TABLE 18: GAS TURBINE PROPERTIES

Manufacturer & Model	GE LM1800e High Power
Shafts	2
RPM	3000
Pressure ratio	15,8
Turbine inlet temperature[i] (TIT)	1121 °C
Turbine exit temperature (TET)	494 °C
Air flow	68 [kg/s]
Generator power	17725 kW_{el}
LHV efficiency	34,6 %

The design efficiency for a gas turbine is given for full load at ISO conditions: 15 °C, 100 kPa and 60% relative humidity for the ambient air.

[1] The TIT definition used is the stagnation temperature at the first rotor inlet. This TIT definition is often called "Firing inlet temperature".

Grate Boiler Flow sheet

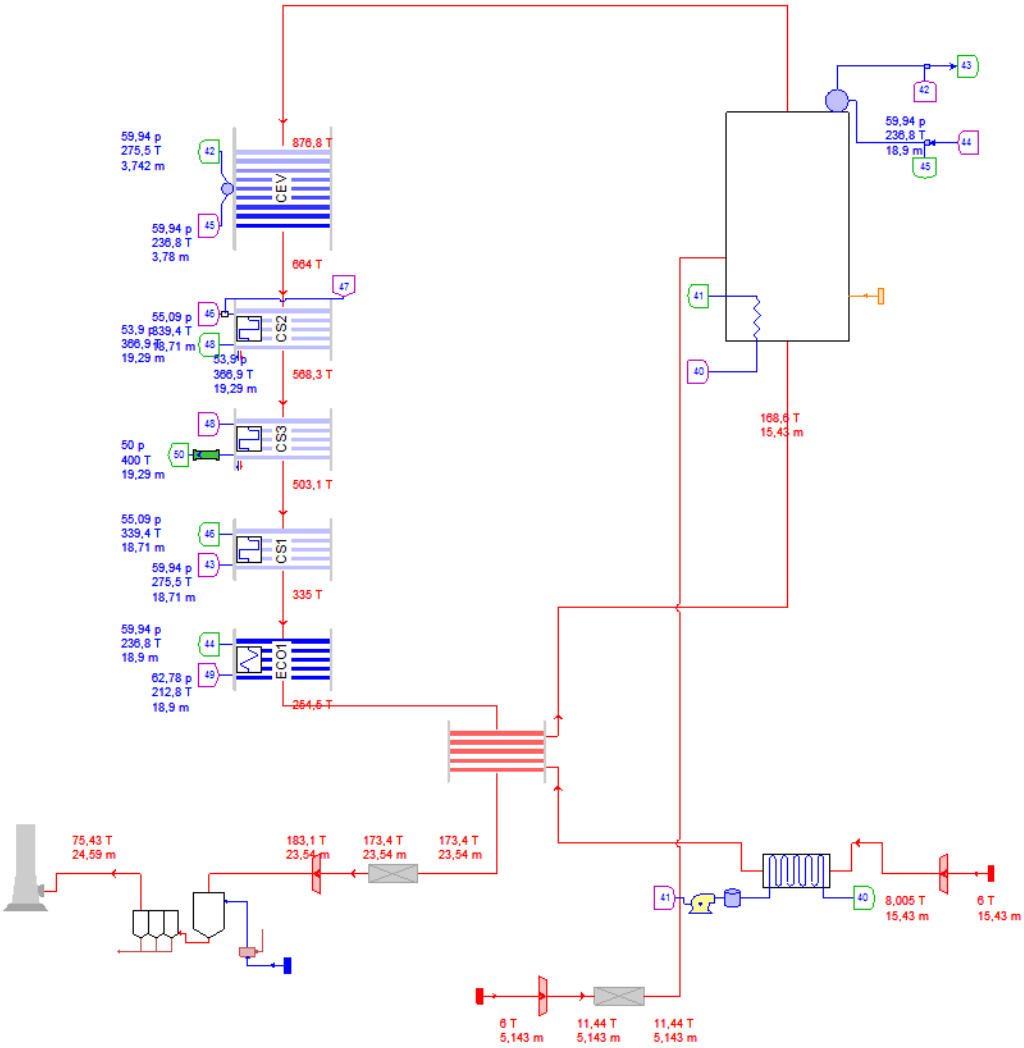


FIGURE 80: GRATE BOILER, THERMOFLEX FLOW SHEET

CIRCULATING FLUIDISED BED BOILER FLOW SHEET

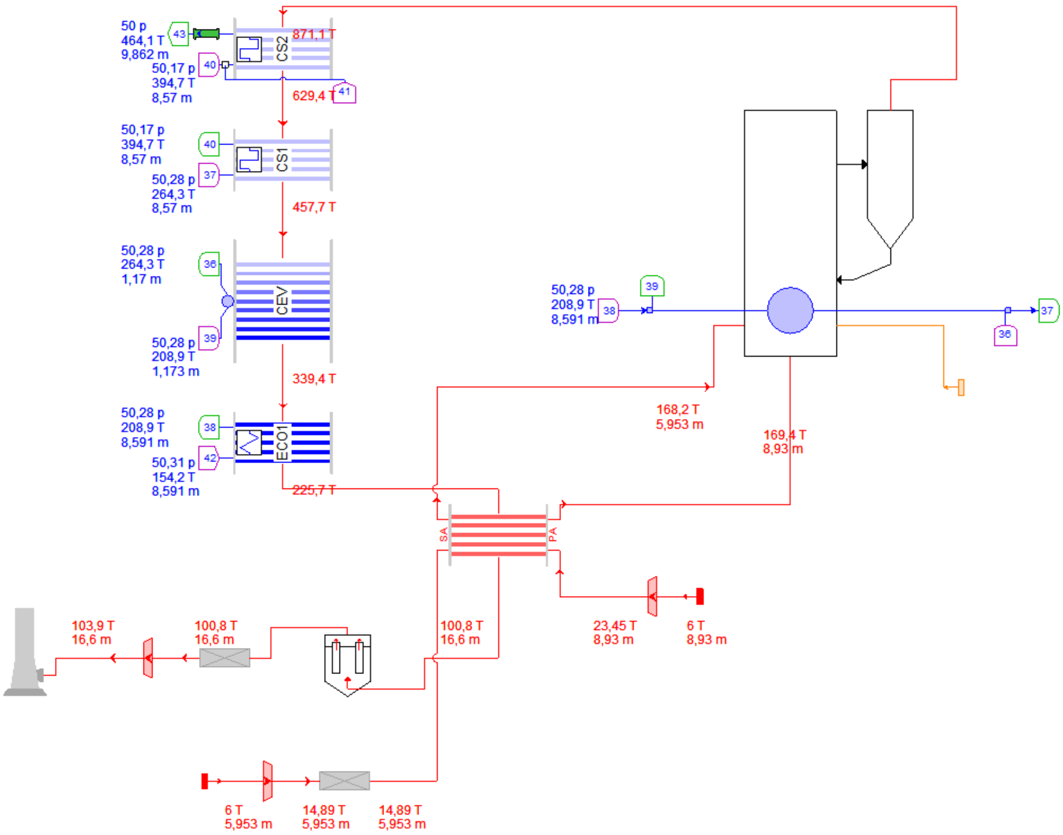
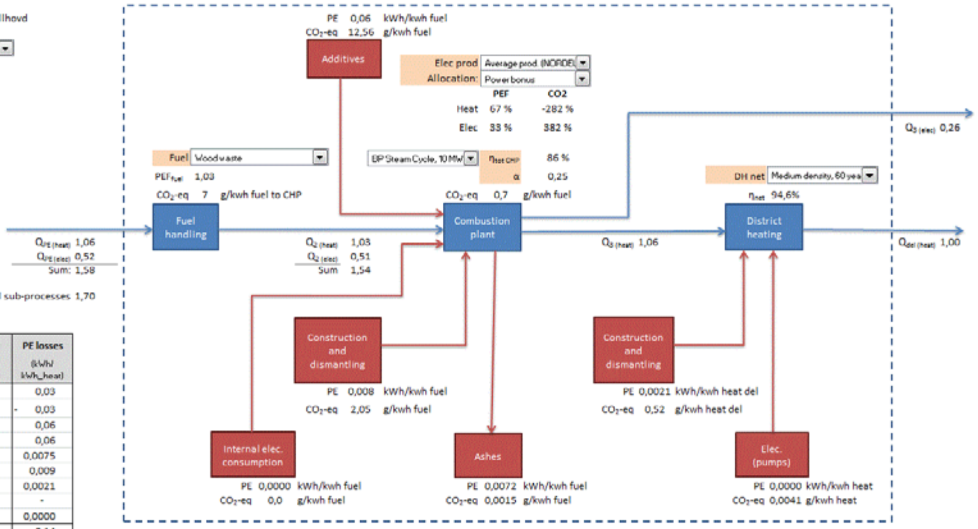


FIGURE 81: CIRCULATING FLUIDISED BED BOILER, THERMOFLEX FLOWSHEET

Screen shot of modified Excel calculation tool

22.11.2011
SP, Sweden. Modified by Magnhild Kallhovd

Yearly load distribution | Year 1



District heating	CO2-eq ($g\ CO_2$-eq/ kWh_heat)	PE losses (kWh/ kWh_heat)
Fuel handling	- 29,7	0,03
Combustion	- 3,1	- 0,03
DH net		0,06
Additives	- 54,5	0,06
Ashes	- 0,00639	0,0075
C&D CHP plant	- 8,9	0,009
C&D DH-net	0,5	0,0021
Internal elec. cons.	-	-
Elec pumps DH net	0,00	0,0000
Sum:	- 95,69	0,14
Delivered heat		1,00
Sum:	- 96	**1,14**

APPENDIX B

BACKGROUND INFO CHAPTER 5

DETAILED FUEL INFO FROM STEAM PRO

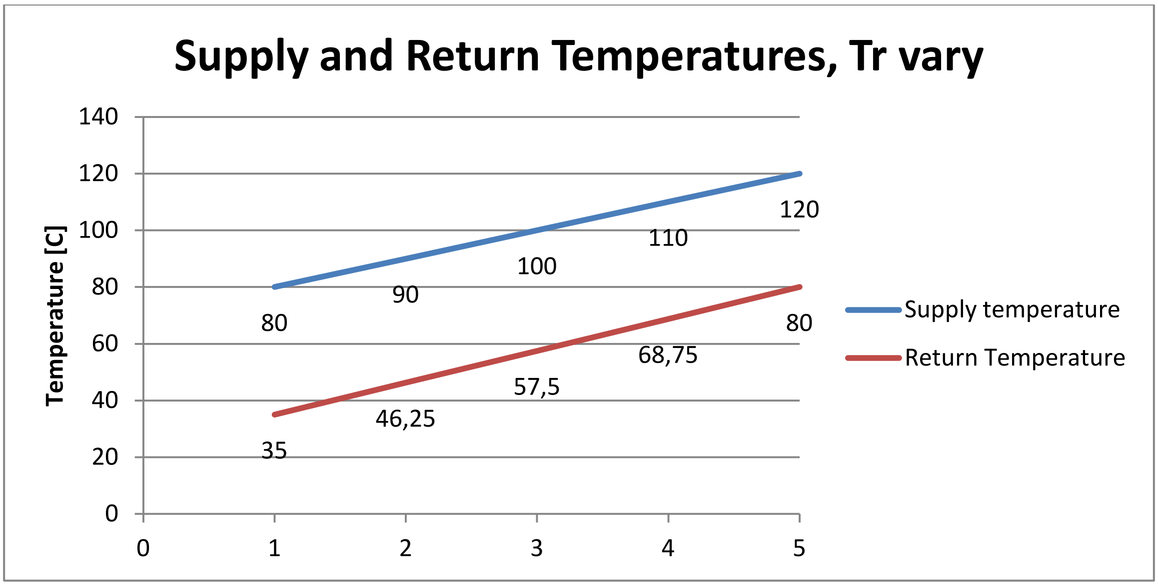
Fuel Name: Demolition Wood		
Type: Biomass--Wood		
Fuel supply temperature	25	C
Total LHV + Sensible heat @ 25C	15464	kJ/kg
Total fuel enthalpy reference to 0C	16815	kJ/kg
Heating Values (at 25C)		
LHV (moisture and ash included)	15464	kJ/kg
HHV (moisture and ash included)	16753	kJ/kg
Ultimate Analysis (weight %)		
Moisture	9,01	%
Ash	11,94	%
Carbon	42,1	%
Hydrogen	4,9	%
Nitrogen	0,52	%
Chlorine	0,05	%
Sulfur	0,11	%
Oxygen	31,37	%
Total	100	%
Proximate Analysis (weight %)		
Moisture	9,01	%
Ash	11,94	%
Volatile Matter	67,84	%
Fixed Carbon	11,21	%
Total	100	%
Other Properties		
Specific Heat @ 25C, dry	1,602	kJ/kg-C
Specific Heat @ 300C, dry	2,42	kJ/kg-C
Ash Analysis (weight %)		
SiO2	45,91	%
Al2O3	15,55	%
Fe2O3	12,02	%
CaO	13,51	%
MgO	2,55	%
Na2O	1,13	%
K2O	2,14	%
TiO2	2,09	%
P2O5	0,94	%
SO3	2,45	%
Other	1,71	%
Total	100	%
Ash Characteristics		
Fouling	High	
Ash Total Alkali Content	233,1	ng/J
Estimated Ash Deposition Temperature	1032,4	C

Solid Fuel		
Fuel Name: MSW-3		
Type: Waste--MSW		
Fuel supply temperature	25	C
Total LHV + Sensible heat @ 25C	10133	kJ/kg
Total fuel enthalpy reference to 0C	11675	kJ/kg
Heating Values (at 25C)		
LHV (moisture and ash included)	10133	kJ/kg
HHV (moisture and ash included)	11600	kJ/kg
Ultimate Analysis (weight %)		
Moisture	25,2	%
Ash	21	%
Carbon	28,1	%
Hydrogen	3,9	%
Nitrogen	0,4	%
Chlorine	0,5	%
Sulfur	0,3	%
Oxygen	20,6	%
Total	100	%
Proximate Analysis (weight %)		
Moisture	25,2	%
Ash	21	%
Volatile Matter	25	%
Fixed Carbon	28,8	%
Total	100	%
Other Properties		
Specific Heat @ 25C, dry	1,182	kJ/kg-C
Specific Heat @ 300C, dry	1,881	kJ/kg-C
Ash Analysis (weight %)		
SiO2	45	%
Al2O3	26	%
Fe2O3	8	%
CaO	11	%
MgO	2	%
Na2O	5	%
K2O	1	%
TiO2	0	%
P2O5	0	%
SO3	2	%
Other	0	%
Total	100	%
Ash Characteristics		
Fouling	Severe	

Fuel Name: Wood pellets		
Type: Biomass--Wood		
Fuel supply temperature	25	C
Total LHV + Sensible heat @ 25C	16784	kJ/kg
Total fuel enthalpy reference to 0C	18265	kJ/kg
Heating Values (at 25C)		
LHV (moisture and ash included)	16784	kJ/kg
HHV (moisture and ash included)	18197	kJ/kg
Ultimate Analysis (weight %)		
Moisture	8,7	%
Ash	0,5	%
Carbon	45,8	%
Hydrogen	5,5	%
Nitrogen	0,08	%
Chlorine	0,01	%
Sulfur	0,01	%
Oxygen	39,4	%
Total	100	%
Proximate Analysis (weight %)		
Moisture	8,7	%
Ash	0,5	%
Volatile Matter	74,4	%
Fixed Carbon	16,4	%
Total	100	%
Other Properties		
Specific Heat @ 25C, dry	1,686	kJ/kg-C
Specific Heat @ 300C, dry	2,57	kJ/kg-C
Ash Analysis (weight %)		
SiO_2	24,1	%
Al_2O_3	4,22	%
Fe_2O_3	3,46	%
CaO	37,95	%
MgO	8,28	%
Na_2O	1,66	%
K_2O	15,81	%
TiO_2	0	%
P_2O_5	4,52	%
SO_3	0	%
Other	0	%
Total	100	%
Ash Characteristics		
Fouling	Severe	
Ash Total Alkali Content	48	ng/J
Estimated Ash Deposition Temperature	1159,2	C

Fuel Name: Wood Waste		
Type: Biomass--Wood		
Fuel supply temperature	25	C
Total LHV + Sensible heat @ 25C	15665	kJ/kg
Total fuel enthalpy reference to 0C	17116	kJ/kg
Heating Values (at 25C)		
LHV (moisture and ash included)	15665	kJ/kg
HHV (moisture and ash included)	17049	kJ/kg
Ultimate Analysis (weight %)		
Moisture	10,2	%
Ash	6,2	%
Carbon	44,02	%
Hydrogen	5,2	%
Nitrogen	0,69	%
Chlorine	0,11	%
Sulfur	0,07	%
Oxygen	33,51	%
Total	100	%
Proximate Analysis (weight %)		
Moisture	10,2	%
Ash	6,2	%
Volatile Matter	67,3	%
Fixed Carbon	16,3	%
Total	100	%
Other Properties		
Specific Heat @ 25C, dry	1,613	kJ/kg-C
Specific Heat @ 300C, dry	2,461	kJ/kg-C
Ash Analysis (weight %)		
SiO2	57,5	%
Al2O3	10,3	%
Fe2O3	5,1	%
CaO	10,4	%
MgO	2,9	%
Na2O	2,3	%
K2O	3,7	%
TiO2	0	%
P2O5	0,1	%
SO3	0	%
Other	7,7	%
Total	100	%
Ash Characteristics		
Fouling	Severe	
Ash Total Alkali Content	218,2	ng/J
Estimated Ash Deposition Temperature	1037,4	C

SUPPLY AND RETURN TEMPERATURES WHEN T RETURN VARY



APPENDIX C

EXAMPLE OF INFORMATION REQUEST LETTER

Statkraft Varme AS
Sluppenv. 6,
7005 Trondheim

Trondheim 10.08.2011

Magnhild Kallhovd
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7018 Trondheim
magnhik@stud.ntnu.no
tlf: 980 23 676

Forespørsel om informasjonsbidrag i forbindelse med masteroppgave

Høsten 2011 skal jeg gjennomføre min avsluttende masteroppgave ved Energi og miljø, NTNU. Hoveddelen av oppgaven skal omhandle bruk av kraftvarmeverk i fjernvarmeanlegg. Som en del av denne skal jeg gjøre full- og dellastsimuleringer av forskjellige typer og forskjellige størrelser CHP-anlegg som er aktuelle i fjernvarmenett i Norge.

CHP-anlegg finnes som kjent i svært mange varianter. For å kunne gjøre en begrunnet utvelgelse av hva slags type anlegg jeg ønsker å fokusere på i oppgaven, driver jeg nå med en kartlegging av hva som allerede finnes, samt hva som planlegges av kraftvarmeverk rundt om i landet. Derfor lurer jeg på om dere har noe informasjon om eksisterende og/eller planlagte anlegg som jeg kunne fått tilgang til?

Det er ikke noe mål at anleggene jeg simulerer skal være helt like konkrete, eksisterende, norske anlegg, men de skal representere relevante alternativer. All informasjon dere kan bidra med vil derfor være til stor hjelp med å legge et godt grunnlag for realistiske valg når jeg avgjør hvilke anlegg jeg skal inkludere i simuleringsdelen av oppgaven.

Det jeg først og fremst lurer på er:

- Hva slags type kraftproduksjonsenhet (prime mover) benyttes i anlegget? (Stempelmotor, gassturbin, dampturbin, annet?)
- Hvor stort er anlegget?(Installert kapasitet, el/termisk)
- Hvilken type brensel benyttes i anlegget?
- Hva er total- og elvirkningsgrad og Power/Heat-ratio ved design drift?

Annen informasjon som også hadde vært nyttig er:

- Hvilken type motor/turbin brukes i anlegget? (type, produsent, modell)
- Hvilke tur- og returtemperaturer leverer man til fjernvarme-nettet? (store forskjeller vinter/sommer?)
- Hva er nedre grense for dellast?
- Hva er total- og elvirkningsgrad og power/heat-ratio ved off-design drift? Hva er (forventet) snittverdi på disse gjennom et år?
- Flytdiagram av anlegget

Det hadde vært til stor hjelp om dere kunne bidra med informasjon, både små og store bidrag mottas med takk. Ta gjerne kontakt dersom dere skulle ha noen spørsmål.

Med vennlig hilsen,

Magnhild Kallhovd