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Science and Technology

# Performance Evaluation of Combined Heat and Power (CHP) Applications in Low-Energy Houses

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Department of Energy
and Process Engineering

EPT-M-2013-5

# MASTER THESIS

for

Stud.techn. Kristin Alfstad

Spring 2013

***Performance evaluation of combined heat and power (CHP) applications in low-energy houses***

*Evaluering av ytelse for systemer for kombinert varme og kraft produksjon i lavenergihus*

**Background and objective**

Combined heat and power system (CHP) are systems that produce electricity and heat simultaneously and could cover a large portion of heating in low energy buildings. Until the recent focus on Net-ZEB, the potential advantage provided by electricity production of CHP was largely untapped and was considered as a by-product during energetic evaluation. Within Net-ZEB concept, CHP systems are now considered as a potential solution that could dampen or even avert the strong electrical exchanges that result in case of an all-electric Net-ZEB building. Nevertheless, the potential offered by these systems is strongly dependent on their suitable integration with the building heat loads that in-turn depends on the CHP nominal capacity as well as building heating load profile.

The objective of this work is therefore to evaluate the applicability challenges of such system within low-energy buildings and investigate the energetic and interactive performance of such system in the Net-ZEB context.

This assignment is closely related to The Research Centre on Zero Emission Building at NTNU and SINTEF (FME ZEB) that has the vision to eliminate the greenhouse gas emissions caused by buildings. The main objective of FME ZEB is to develop competitive products and solutions for existing and new buildings that will lead to market penetration of buildings that have zero emissions of greenhouse gases related to their production, operation and demolition.

**The following tasks are to be considered:**

1. Extension and improvement of existing implementation of external and internal combustion CHP models
2. Simulation setup of two implemented models in whole building simulation setup
3. Evaluation of energetic performance of two systems in whole-building simulation

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Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

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☐ Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
☐ Field work

Department of Energy and Process Engineering, 14. January 2013

Olav Bolland
Department Head

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Academic Supervisor

Research Advisors: PhD candidate Usman Ijaz Dar

# Preface

This Master's Thesis was prepared during the spring of 2013 at the Norwegian University of Science and Technology (NTNU), Department of Energy and Process Engineering. The work is closely related to The Research Centre on Zero Emission Buildings.

I would like to thank Usman Ijaz Dar for great help and support. Your time and effort has been very much appreciated. I would also like to thank all my good friends for an exceptional time in Trondheim and my family for perpetual support. Finally, I would like to thank Espen for all inspiration and for being supportive at all times.

# Abstract

The Research Centre on Zero Emission Buildings has a vision to eliminate the greenhouse gas emissions caused by buildings related to their production, operation and demolition. The concept of Zero Energy Building (ZEB) has gained wide international attention during the last few years and the government in Norway has agreed that passive house standard is to be required for new buildings from 2015 and nearly ZEBs as a standard from 2020.

Combined heat and power (CHP), also known as cogeneration, is an emerging technology associated with the potential to reduce primary energy consumption and associated greenhouse gas emissions through the concurrent production of electricity and heat from the same fuel source. Until the recent focus on Net-ZEB, the heat provided by electricity production of CHP was considered as a by-product during energetic evaluation. Within the Net-ZEB concept, CHP systems are considered as a potential energy supply solution for buildings.

As CHP systems have large thermal output and the heating needs of buildings are getting decreased with super insulated envelops, the integration of the CHP systems becomes challenging. The potential offered by these systems is strongly dependent on their suitable integration with the building heat loads.

A simulation model is used to investigate the performance of CHP systems supplying a residential building. Analysis of the simulation results indicate that increasing the size of the storage tank does not improve the performance of the system as the heat losses becomes greater. Having less stringent requirements to the thermal comfort will improve the operation of the CHP unit, but the comfort must be maintained at an acceptable level.

By adding an auxiliary gas boiler to the system, covering the heating needs outside the heating season, a system efficiency of 80% is achieved when supplying a passive house and 81% when supplying a low energy building. Compared to the systems only using CHP, these efficiencies became 78% and 79% for the passive house and low energy building, respectively. When supplying the low energy building a higher efficiency is achieved. The low energy building has higher heating needs which are a more favorable condition for the operation of the CHP. Nevertheless, the system supplying the low energy building will emit more $CO_2$ which is not desirable in a net-ZEB context.

The amount of $CO_2$-production for different energy supply systems are calculated and compared showing that the CHP systems are more favorable when the $CO_2$-production factor for electricity is high. Taking into account that the $CO_2$-production factor for electricity is expected to increase over the years, as the electricity production in the world

becomes greener, the CHP-technology will need further development in order to retain its position as a favorable energy supply solution in a net-ZEB context.

# Sammendrag

Zero Emission Buildings (ZEB) er et forskningssenter med en visjon om å eliminere klimagassutslipp forårsaket av bygninger relatert til utslipp ved produksjon av bygget, utslipp fra energi til drift og utslipp ved rivning. Begrepet Zero Energy Building (ZEB) har fått stor internasjonal oppmerksomhet i løpet av de siste årene, og regjeringen i Norge har vedtatt at nye bygg skal ha passivhus-standard fra 2015 og nesten nulllenergibygg fra 2020.

Kombinert varme- og kraftproduksjon, også kalt kogenerering, er en teknologi forbundet med å redusere primærenergibruk og tilhørende klimagassutslipp. Dette oppnås ved simultant å produsere varme og elektrisitet fra samme energikilde. Før fokuset ble rettet mot nullenergibygg ble overskuddsvarmen ved elektrisitetsproduksjon kun ansett som et biprodukt. ZEB-konseptet anser derimot kombinert varme- og kraftproduksjon som en potensiell løsning som energiforsyning til bygg.

Hus som bygges i dag er godt isolert og har dermed redusert oppvarmingsbehov. I og med at kogenereringsenher har stor varmeeffekt blir integreringen utfordrende. Ettersom potensialet til disse systemene er sterkt avhengig av integrasjonen med varmelasten til bygget det skal forsyne er det viktig at det oppdages gode løsninger for hvordan integreringen kan og bør utføres.

En simuleringsmodell brukes for å undersøke ytelsen til et kombinert varme- og kraftproduksjonssystem som leverer varme til en enebolig. Analysen av simuleringsresultatene indikerer at økt størrelse på varmtvannstanken ikke forbedrer ytelsen til systemet ettersom varmetapet blir større. Etter at kravene til den termiske komforten ble gjort mildere viste det seg at driften av systemet ble forbedret.

Ved å innføre en gasskjel til å dekke varmebehovet utenfor fyringssesongen ble systemvirkningsgraden 80 % da systemet forsynte et passivhus og 81 % da et lavenergihus ble forsynt. Til sammenligning oppnådde systemene virkningsgrader på henholdsvis 78 % og 79 % uten gasskjelen, kun forsynt av kogenereringsenheten.

Når kogeneringssystemet forsyner et lavenergibygg oppnås en høyere virkningsgrad enn ved forsyning av passivhuset. Lavenergibygget har et større varmebehov hvilket medfører mer gunstige forhold for kogenereringsenheten. Likevel vil systemet avgi mer $CO_2$ når det forsyner lavenergihuset, noe som ikke er gunstig i henhold til ZEB-konseptet.

Til sammenligning med andre energiforsyningssystemer viser beregninger at kogenereringssytemene er gunstige når det kommer til $CO_2$-utslipp, spesielt når $CO_2$-faktoren for elektrisitet er høy. Tatt i betraktning at denne vil reduseres med årene, ettersom elektrisitetsproduksjonen i verden blir mer miljøvennlig, må kogenereringsteknologien utvikles videre for å kunne beholde sin status i ZEB-sammenheng.

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# 1. Introduction

## 1.1 Background and motivation

The Research Centre on Zero Emission Buildings has a vision to eliminate the greenhouse gas emissions caused by buildings related to their production, operation and demolition. The concept of Zero Energy Building (ZEB) has gained wide international attention during the last few years and is now seen as the future target for the design of buildings. The concept is considered as a realistic solution for the mitigation of $CO_2$ - emissions and/or the reduction of energy use in the building sector (1). The government in Norway has agreed that passive house standard is to be required for new buildings from 2015 and nearly ZEBs as a standard from 2020 (2).

The term ZEB is used commercially, in policies and national targets, without a common understanding. The Net-ZEB term defined by Satori et al is the definition applied in this thesis. It says that Net-ZEBs are connected to an energy infrastructure and have a balance between energy taken from and supplied back to the energy grid (3).

Approximately 40 % of the total energy use in Norway is being consumed by the building sector and the building stock is highly dependent on electricity for heating purposes. More than 95% of the electricity production in Norway comes from renewable hydropower (4). However, in a global perspective, renewable electricity production only represents a small part of the total production. The open economy makes Norway strongly dependent on the rest of the world. Likewise, our actions can have great international influence (5). As a consequence, reducing the electricity use or increasing the production of renewable electricity in Norway indirectly poses a positive effect on the global climate, replacing non-renewable electricity production somewhere else.

Combined heat and power (CHP), also known as cogeneration, is an emerging technology associated with the potential to reduce primary energy consumption and associated greenhouse gas emissions through the concurrent production of electricity and heat from the same fuel source. Until the recent focus on Net-ZEB, the heat provided by electricity production of CHP was considered as a by-product during energetic evaluation. Within the Net-ZEB concept, CHP systems are considered as a potential energy supply solution for buildings (6, 7). The CHP unit can run on renewable fuels, but fossil fuels are most widely used. Even though the CHP is fueled with fossil fuels as natural gas, the technology is considered low-carbon as it will contribute in utilizing the limited fossil resources more efficient (8). Norway is the 7th largest producer of natural gas worldwide hence natural gas is a superior fuel of choice with great potential in the Norwegian context (9).

As CHP systems have large thermal output and the heating needs of buildings are getting decreased with super insulated envelops, the integration of the CHP systems becomes challenging. The potential offered by these systems is strongly dependent on their suitable integration with the building heat loads. In order to study and evaluate the performance of different systems in different load environments, an accurate simulation model of the micro-CHP device is needed making it possible to investigate the benefit of CHP versus the problems that arise due to dynamics.

## 1.2 Purpose and objectives of this report

The main purpose of this study is to analyze the performance of selected cogeneration systems in passive house and low-energy buildings in the Norwegian building context.

The objectives of the performance assessment study are to:

1. Quantify the performance in terms of energy and emissions.
2. Study the temporal characteristics of the CHP unit.

## 1.3 Method and tool

This study uses a simulation program to perform dynamic simulations. The building model is implemented using the international standard ISO 13790 (10) where the model for CHP is implemented using a mathematical model presented in the International Energy Agency's Energy Conservation in Buildings and Community Systems (IEA ECBCS) Annex 42 (11-13). Both model implementations are done in MATLAB as a part of a PHD work carried out at NTNU (14).

# 2. Nomenclature and symbols

## 2.1 Abbreviations and indices

| | |
|---|---|
| boil | Boiler |
| CHP | Combined heat and power (=cogeneration) |
| $CO_2$ | Carbon dioxide |
| Cond | Condensing |
| cw | Cooling water |
| DH | District heating |
| DHW | Domestic hot water |
| El | Electricity, electric |
| eng | Engine |
| exh | Exhaust |
| Fuel | Delivered fuel |
| GB | Gas boiler |
| gen | Generation |
| HX | Heat exchange |
| ICE | Internal combustion engine |
| ICE | Internal combustion engine |
| LEB | Low energy building |
| LHV | Lower heating value |
| max | Maximum |
| MFAB | Multi-family apartment building |
| NG | Natural gas |
| nom | Nominal |
| Norm | Normal |
| Prod factor | Production factor ($CO_2$) |
| SE | Stirling engine |
| SF | Single family |
| SFH | Single family house |
| SFP | Specific fan power |
| SH | Space heating |
| SP | Space heating |
| ss | Steady state |
| ZEB | Zero emission building |

# 3. Micro-CHP

## 3.1 Technologies

CHP-technology makes it possible to supply residential buildings with both electricity and heat. The electricity can be exported to the energy grid or be used to cover the electricity needs in the house. The heat is used for space heating and domestic water heating. Micro-scale CHP systems for single- or multi-family dwellings are typically designed to provide electricity less than 10kW and thermal heat less than 25kW (6). Different types of micro-CHP technologies are available or under research and development. These include micro-turbine based systems, fuel cell-based systems, reciprocating internal combustion engine-based systems and Stirling engine-based systems. This study is further based on CHP systems with Stirling engine (SE) and internal combustion engine (ICE) technology. Hence, only these technologies are further presented.

Reciprocating internal combustion engines are well proven, robust and reliable. They are available over a wide range of sizes and can be fired on a broad variety of fuels. Usually, reciprocating engines use natural gas or diesel oil as fuel, but bio-oils and biomass are also under research (7). Owing to moving parts, the engines need service regularly. The ICE has a high operating noise levels. Placed in a dwelling, this noise level is unacceptable and needs to be reduced. By housing the engine in a soundproofed enclosure, the noise levels are reduced to those of a standard refrigerator(15). Emissions from the ICE are typically the highest of any CHP-technology owing to the combustion conditions within the engine. Reciprocating ICE systems are usually the prime mover of choice for small-scale cogeneration applications due to their well-proven technology, robust nature and reliability (16).

The SE differs from the ICE in that the fuel combustion occurs outside the cylinders. External combustion is continuous and more controllable than internal combustion. It allows a wide range of energy sources and it is possible to change fuels during operation, with no need to stop or make adjustments on the engine. Fossil fuels such as oil or gas, and renewable energy sources like solar and biomass are all applicable. The external combustion also results in low vibration and noise level. The SE is well developed and has good performance at partial load. It also has a general prospect of high efficiency, but with moderate electrical efficiency. Due to fewer moving parts, they are expected to prove more durable and have long maintenance free operation periods compared to reciprocating ICEs. A SE are highly applicable to residential buildings, especially because the electricity/heat ratio is suitable (6, 7).

Although the SE technology for residential applications is not fully developed yet and is in very limited use, it shows a strong potential due to its high efficiency, fuel flexibility, low emissions and noise/vibration levels and good performance at partial load. A number of

Stirling engine developers for micro-CHP applications are available and manufactures are worldwide dispersed (17).

## 3.2 Integration of the CHP

The residential CHP- system can be integrated to satisfy both the electrical and thermal demands, the electrical demand and part of the thermal demand, the thermal demand and part of the electrical demand or, most commonly, satisfy part of the electrical demand and part of the thermal demand. Some practical and economic constraints need to be considered in each system design (6).

1. **The cogeneration unit is designed to fully meet the electrical demand:** If the cogeneration plant is producing more heat than the thermal demand, the plant can either start working under part load conditions or switch on and off trying to meet the demand. Another option is to dump surplus heat to the atmosphere or to store it in a thermal storage device. On the other hand, if the heat demand is higher than the cogeneration capacity, a secondary heat source to “top-up” the heat demand is used.

2. **The cogeneration unit is designed to fully meet the thermal demand:** If the electrical demand in the building is less than the electrical output, the plant can be throttled back. If the plant is not throttled back, the surplus electricity can either be exported to the utility grid or stored in an electrical storage device such as batteries or capacitors. If the cogeneration plant is not producing enough electricity to cover the demand of the building, electricity can be imported from the utility grid.

3. **The cogeneration unit is designed to fully meet both electricity and thermal demands:** With this design, it is usually necessary to install systems that are oversized in both their electrical and thermal outputs. Such systems will have a decrease in the units running time, due to an insufficient load being available. The reduction in run hours will make the economics of the system poorer. For this reason, cogeneration devices are usually sized to meet only a part of the electrical and thermal need (6).

4. The residential building sector is characterised by highly variable demands and the technical challenges are large. In a way of meeting the potential mismatch between production and demand of heat and electricity, systems usually include thermal storage and connection to the electrical grid. Battery storage is impractical and expensive at present (15, 18).

## 3.3 Operational strategies

The efficiency of a cogeneration system is measured as the fraction of the fuel input that can usefully be recovered as power and heat. The remaining energy is lost within the exhaust gases and in the engine and generator by radiation and convection. The maximum energy efficiency of a system is reached when the energy delivered by the cogeneration unit equals the energy requirement of the building. The overall efficiency depends on several factors: The prime mover, the size of the plant and the temperature at which the recovered heat can be utilized. Also, conditioning and operating regime of the cogeneration unit plays a vital part.

Control strategies are shown to have a significant effect on energy and environmental system performance (19). Cogeneration units can be run at base-load and load-following mode. At base-load mode, the unit operates at constant power. At the load-following modes, it follows either the thermal or electrical load. A challenge with the power control is that some micro-CHP units only have on/off operation available at the present. Long start-up and shutdown periods may also be required for some of the cogeneration technologies. Lower efficiency at part load and substantial fuel demand during start-up phase are also challenges associated with the power control.

For systems that cycle on and off, operating patterns have the most notable impact. Electrical, as well as total efficiencies are strongly affected by the length of operating periods. Micro-CHP systems therefore tend to be undersized compared to the peak demands of the house, allowing for longer operating periods at high loads (15). The systems are also required to have high annual usage and extensive periods of almost continuous operation in order to be profitable (6).

# 4. CHP modelling

Recognizing the importance of study micro-CHP systems, a generic model for combustion – based cogeneration systems has been developed within the IEA/ECBS Annex 42 (11). The main objective of the model is to accurately predict the thermal and electrical outputs of a CHP unit. This model can be coupled with other plant components as heat storage, other heating devices and the thermal and electrical demands of a building. The interaction between the CHP unit and the building is then to be investigated.

## 4.1 Model control volumes

Testing work within Annex 42 of the IEA ECBCS indicate that a "grey-box" modelling approach is appropriate for the simulation of micro-CHP devices. The model is divided into three blocks, each representing a control volume modelling the time-varying thermal characteristics (see Figure 1). For a detailed mathematical description of the model see Appendix A.

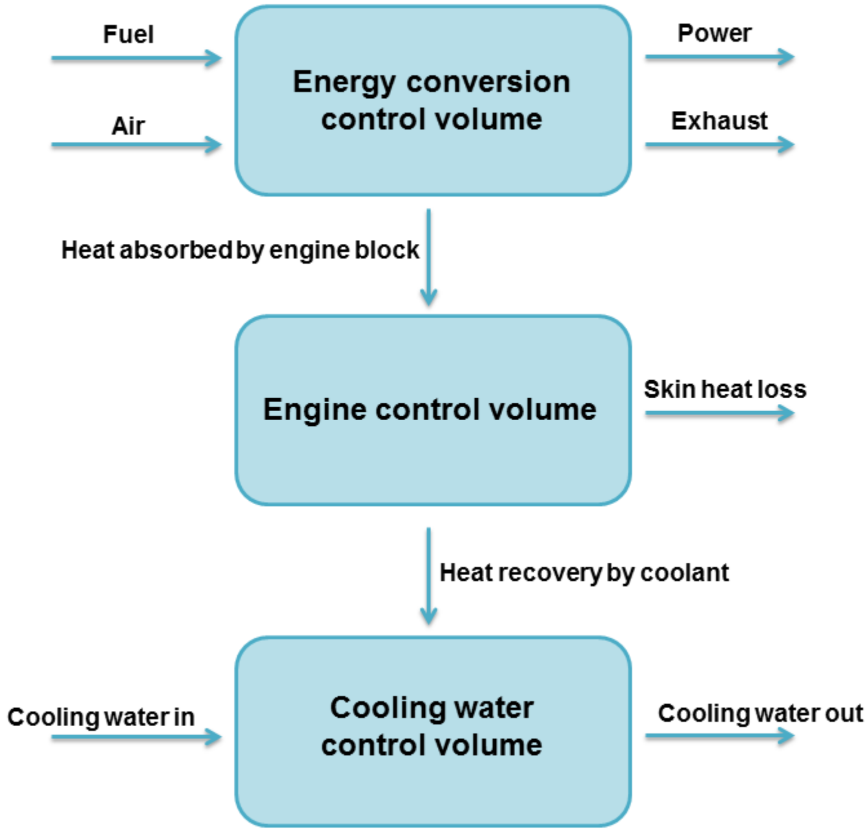


**Figure 1: Control volumes of the generic micro-CHP model.**

## 4.2 Model implementation

An implementation of the generic micro CHP-model is used to perform simulations on different micro-CHP systems. Empirical data from calibration and validation work of one SE and one ICE is then applied in order to perform simulations testing different CHP devices.

When the calibration parameters for the ICE were implemented in the model, the model did not work as expected. In fact, the model was not running at all, unable to produce any result. Hence, further considerations on the ICE will not occur in this thesis except a presentation of the calibration and validation work that were used.

By further inspection, the problem seems to lay in the energy balances of the engine and the cooling water control volumes (eq. 12 and eq. 14 in Appendix A). These equations are used to solve the dynamics of the system by iteration with the Crank-Nicholson method. When the first iteration process is conducted, the value of the engine temperature becomes extremely negatively low and the cooling water temperature, out of the engine, becomes extremely high. These temperatures are causing other parameters to become infinite, making it impossible for the model to complete the simulation process.

The four parameters$[MC]_{eng}$, $[MC]_{cw}$, $UA_{Hx}$ and $UA_{loss}$ are possible suspects of causing the errors. They might be incorrectly specified. It is also possible that the dynamics of the model used in this work is not capable of handling the calibration parameters.

# 5. Description and characteristics of system components

This chapter describes the model components of the domestic energy system.

## 5.1 System components

The domestic energy systems were modeled using a system model mainly consisting of the following parts:

- A cogeneration model based on the mathematical model developed within IEA Annex 42 (11).
- Necessary calibration details for the micro CHP-devices.
- A stratified hot water storage tank model (20).
- A 5 resistance, single capacitance building model containing all necessary data for energy calculation within the building based on ISO 13790 (10). The energy calculations are performed according to NS-EN 15603(21).
- Demand load profiles.
- A gas boiler with an efficiency curve.

## 5.2 CHP devices

The SE model is based on the calibration work done on a WhisperGen Stirling Engine device within the IEA Annex 42 (12). The ICE model is based on the calibration work done on an AISIN SEIKI device at the Built Environment Control Laboratory of Seconda Univerità degli studi di Napoli (22).

The technical data of the units can be seen in Table 1. All calibration parameters can be seen in Appendix B.

| Technical data | SE | ICE |
|---|---|---|
| Electrical power output | 698 W | 6 kW |
| Thermal output | 7288 W | 11,7 kW |
| Electrical efficiency | 0,0929 | 0,27 |
| Thermal efficiency | 0,97 | 0,69 |

**Table 1: A selection of technical data on the micro-CHP devices.**

Natural gas is used as fuel for the micro-CHP units in all the simulations. The lower heating value is obtained from the Norwegian Water Resources and Energy Directorate (NVE) and is set to 37,38 MJ/m$^3$ (23).

## 5.3 Thermal and electrical storage

A stratified cylindrical tank is used for hot water storage. The tank is divided into 12 nodes. Hot water for space heating is tapped directly from the tank while the domestic water is heated by a heat exchanger placed inside the tank. An insulation thickness of 15 cm was assumed for all the tank sizes.

The following tank sizes are considered:

1. Small tank: 300 liters
2. Medium tank: 500 liters
3. Large tank: 800 liters

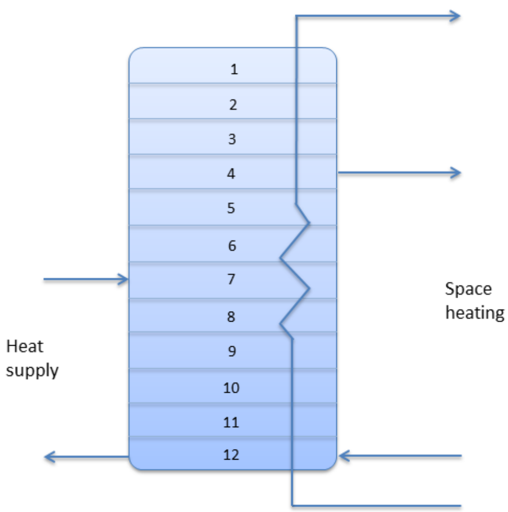


**Figure 2: Stratified storage tank with 12 nodes, inlets and outlets.**

Unless otherwise is stated, a storage tank of 300 liters is used.

No electric storage is considered in this study. The electricity surplus generated by the cogeneration system is directly delivered to the grid..

# 5.4 Gas boiler

A gas boiler (GB) is used in some simulation cases as replacement heater for the CHP unit, outside the heating season (HS). The GB effect is 5 kW and is fuelled with natural gas. The GB is connected in parallel with the CHP unit and the storage tank.

Two different GBs are considered:

### 1. Regular GB

The regular GB has an assumed constant efficiency of 0,86 according to NS-EN 3031 (24).

### 2. Condensing GB

The condensing GB has an assumed efficiency curve as shown in Figure 3. The efficiency depends on the inlet water temperature. The condensing GBs can achieve a high efficiency by using the waste heat in the flue gasses to pre-heat the cold water entering the boiler. The water vapour produced during combustion is condensed into water.

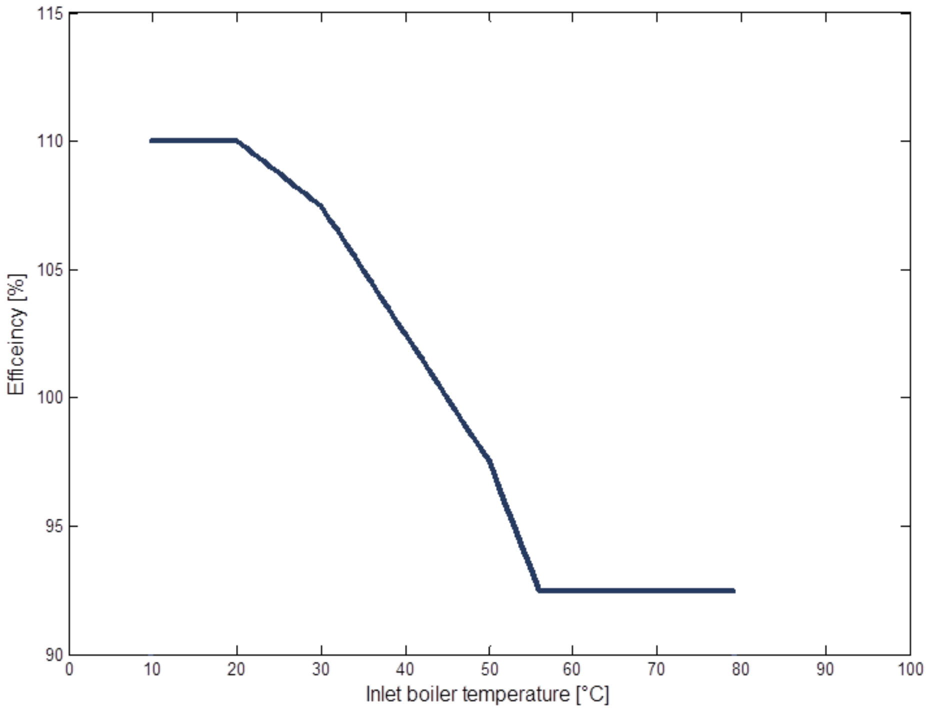

**Figure 3: Condensing gas boiler efficiency curve.**

# 5.5 Control

A heat-demand-following control mode is used in all cases analyzed. The storage tank temperature level is controlled with a sensor placed inside the tank. The control system has a dead band of 10°C, telling the CHP unit when it needs to supply the tank with heat. The temperature control is set to lie between 55°C and 65°C.

A sensor placed inside the house controls the space heating demand and ensures thermal comfort. The sensor control has a set temperature and a dead band. The dead band tells the sensor how many degrees the indoor temperature is allowed to decrease below the set temperature.

The control system has to comply with a number of requirements:

- Make sure that the determined temperature-level in the storage tank is satisfied.
- Comply with the thermal comfort requirements of the building.
- Minimize the number of stop cycles in order to achieve high efficiency.
- Maximize the run time of the CHP-device.

# 5.6 Building

The CHP system is analysed supplying a single-family house (SFH) within both the Norwegian standard NS3700 for passive houses (PH) and according to the standard for low energy buildings (LEB), class one (25) . These standards have requirements to U-values, thermal bridges, heat recovery, specific fan power (SFP) and infiltration rates which will affect the energy demand level. The values used within the simulations can be seen in Appendix D along with other building-specific parameters.

The space heating demand is time-dependent and influenced by many internal and external factors. It is calculated in the building model within the simulation. This is an important parameter, making it possible to investigate the interaction between the CHP system and the building. It is assumed that there is no space heating demand outside the heating season and cooling needs are not taken into account.

The heating of the building is distributed using low-temperature radiators with supply temperature of 40 °C. A standard domestic hot water consumption profile for a SFH with a demand of 170 liters per day is used. The supply temperature is controlled to be 45°C where the cold district water temperature is assumed fixed at 10°C throughout the year. Standard load profiles are used for electrical appliances, lighting and occupants.

100 % of the heat from electrical appliances, lighting and occupants is assumed as internal gains. The storage tank is assumed located inside the heating zone hence heat losses from the tank are added as internal gains. External heat gains from the sun and shading considerations are calculated according to NS 3031 (26).

The model is using climate details from Oslo, having design temperature of -25 °C and mean annual temperature of 6.3 °C.

# 6. Description of systems

## 6.1 Energy supply system

The total energy supply system for the SFH is illustrated in Figure 4. This system consists of one CHP unit in parallel connection with a GB and a cylindrical hot water storage tank. DHW is heated by a heat exchanger inside the storage tank. Space heating water for the radiators is tapped directly from the storage tank. A controller decides the operation between the CHP unit and the GB.

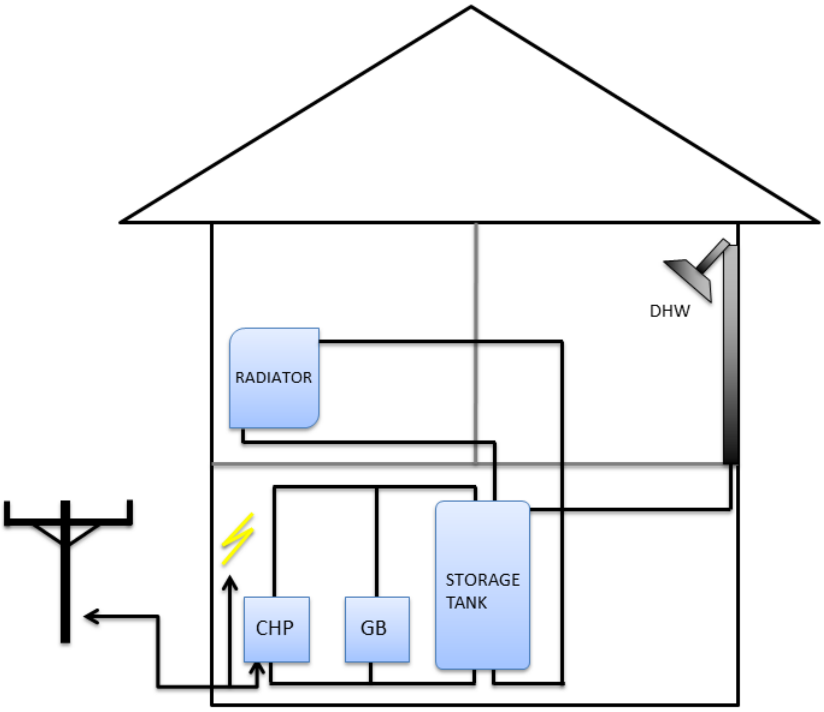


**Figure 4: Illustration of the energy supply system.**

## 6.2 System configurations

Figure 5 and Figure 6 illustrates the schematics of the system configurations analysed in this thesis:

**A.** Micro-CHP as heat and power generator.

**B.** Micro-CHP as heat and power generator in parallel with a gas boiler as heat generator.

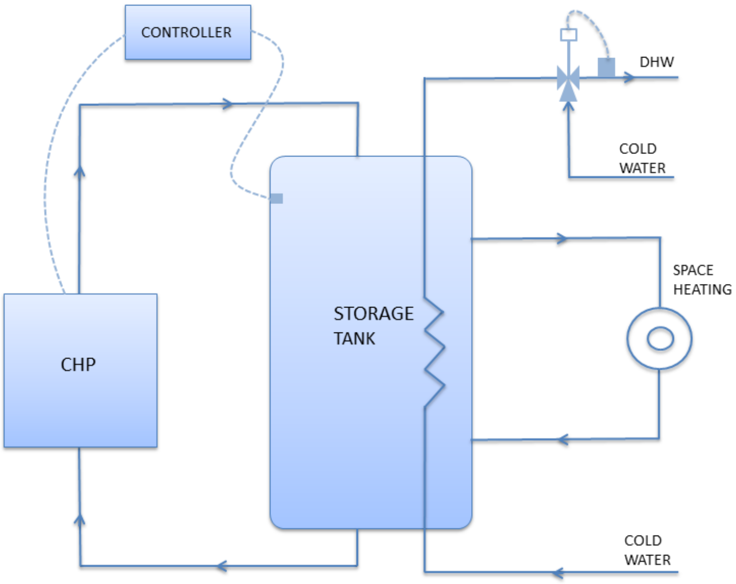


**Figure 5: Configuration A, only CHP.**

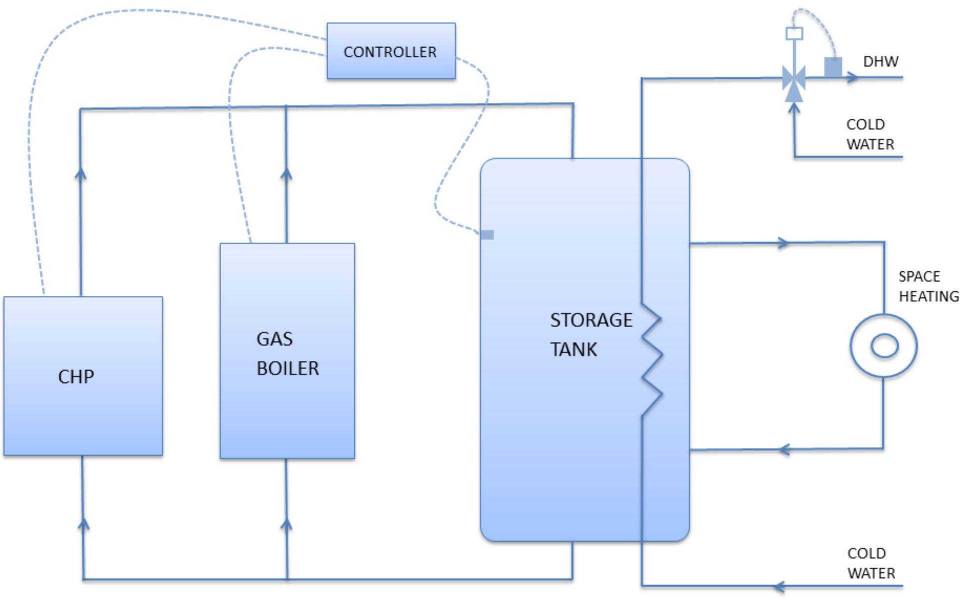


**Figure 6: Configuration B, CHP and GB.**

# 7. Performance assessment method

## 7.1 General

### 7.1.1 Types of performance assessments

The following aspects are evaluated within this thesis:

- Energy performance
- The temporal characteristics of the CHP system
- $CO_2$ emission analysis

### 7.1.2 Performance assessment procedure

The simulation program produces values for net power, generated heat, delivered energies for the building, losses etc. From these values, energy efficiencies are derived. The simulation program also produces information on the temporal characteristics of the CHP unit. The energy efficiencies and the temporal characteristics are further used to evaluate the performance of the various systems. $CO_2$ - emissions are calculated from the fuel flow values.

### 7.1.3 Evaluation period and time step

The simulation period is one year (Jan to Dec). The simulation time step is set to 6 minutes (see Appendix D). However, some components use smaller internal time steps within the simulations.

## 7.2 Energy analysis

### 7.2.1 Energies

The energies of the system are defined as follows:

1. Energy demand: Energy needed to fulfill the building's requirements for space heating, domestic hot water, and for electric lighting and appliances.
2. Energy gains: Part of the energy demand that is covered by internal and external heat gains.
3. Net energy: Part of the energy demand that is covered by the CHP system.
4. Exported energy: Surplus electricity generated from the CHP engine that is exported to the electricity grid.
5. Delivered energy: Energy that is delivered to the building from the CHP system either as heat of electricity.
6. Gross energy: Energy delivered to the energy supply system as fuel.

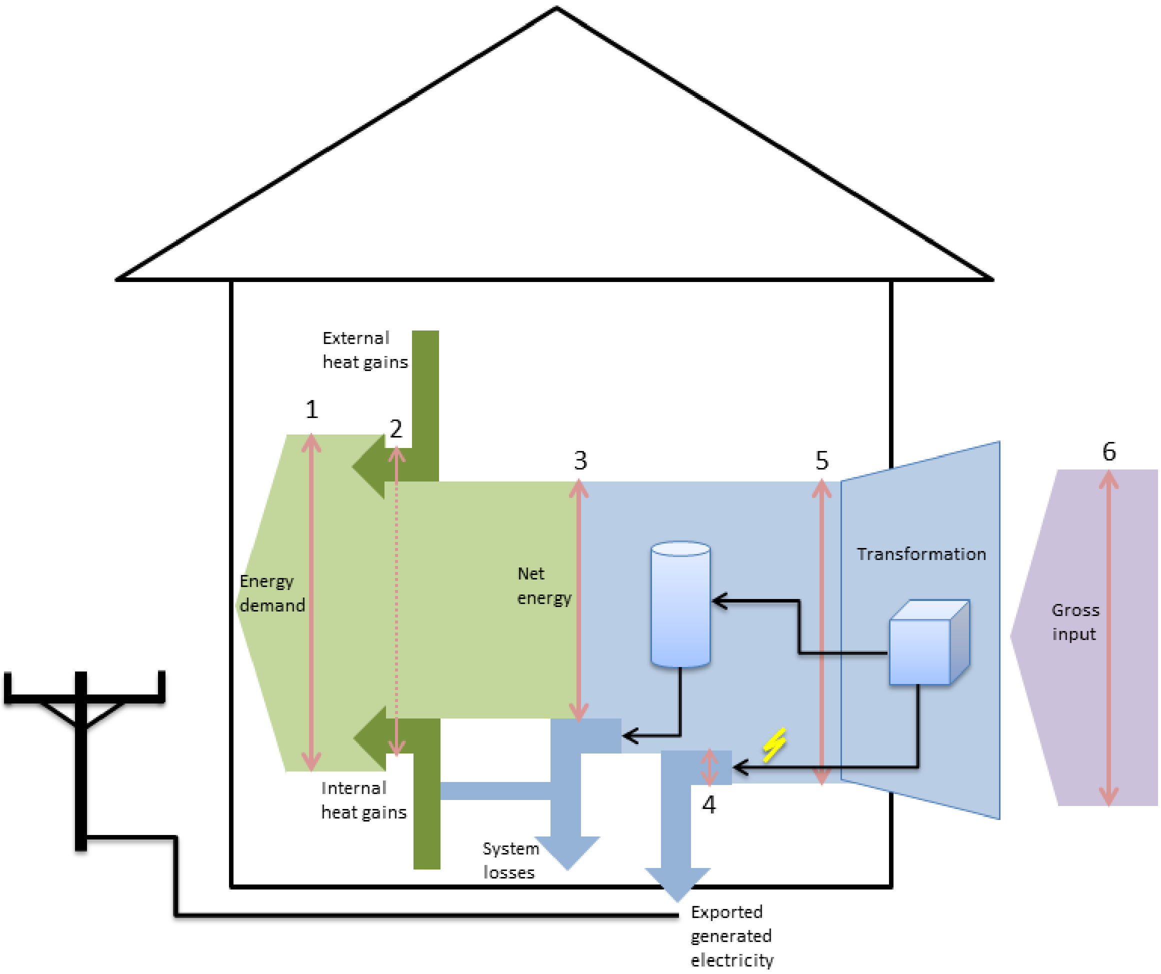


**Figure 7: Energies of the CHP- system.**

### 7.2.2 Control volumes

Two different control volumes and types of energy balances are used for the energy analysis.

1. The first control volume is used in order to analyze the cogeneration unit alone, considering the heat and power production and the gross heat input.
2. The second control volume considers the total heat of the system. Both the Stirling Engine and the internal combustion engine have a large heat production compared to the electricity production, making it interesting to consider only the heat flow of the systems.

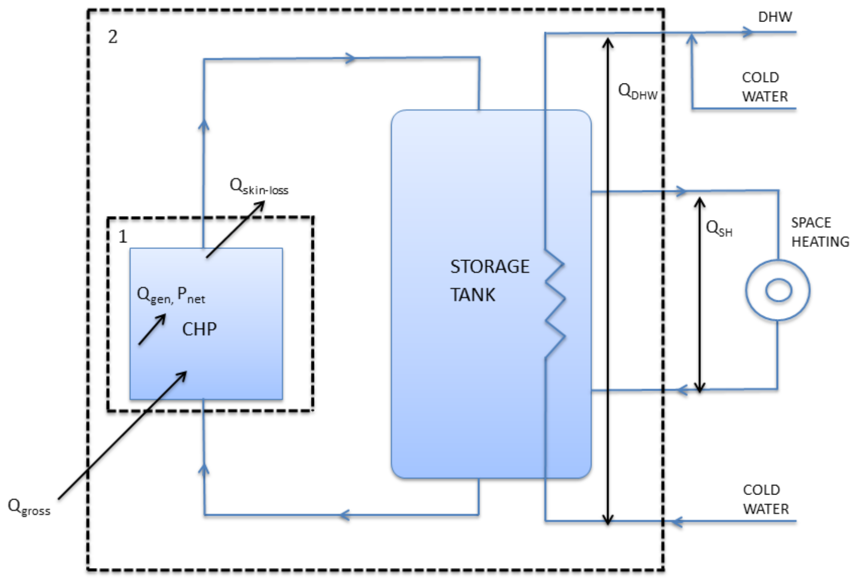


**Figure 8: Control volumes of the CHP system.**

### 7.2.3 Efficiencies considered

The following three efficiencies are used to evaluate the performance of the systems.

1. CHP efficiency

$$\eta_{CHP} = \frac{P_{net} + Q_{tank}}{Q_{gross}} \tag{1}$$

2. System efficiency

$$\eta_{system} = \frac{Q_{DHW} + Q_{SH}}{Q_{gen} + Q_{boil}} \tag{2}$$

3. Boiler efficiency

$$\eta_{boiler} = \frac{Q_{DHW}}{Q_{gross}} \tag{3}$$

The condensing boiler efficiency is determined from the efficiency curve in Figure 3. An average annual efficiency is then calculated.

## 7.3 Temporal characteristics

There is a wide range of possible topics for temporal characteristic evaluations and assessments. This thesis focuses on criteria that have a relation to, or an impact on, the energy and emission performance.

The following temporal characteristics were considered:

- Number of cycles
- Average length on periods the engine operates at full load (normal mode) and in standby mode.
- Longest and shortest period at full load operation (normal mode).

## 7.4 Environmental performance

The performance criterion regarding emissions is the amount of $CO_2$ emitted by the CHP unit during the simulation period.

$$CO_2[kg] = \dot{m}_{fuel} * LHV_{fuel} * CO_{2_{prod\ factor,NG}} - El_{exported} * CO_{2_{prod\ factor,el}} \tag{4}$$

### 7.4.1 Performance assessment in an net-ZEB context

The Research Centre on Zero Emission Buildings has defined $CO_2$ – production factors for different energy sources supplying buildings in an outline of a full ZEB – definition (27). These can be seen in Table 2 along with the factor for the European UCTE electricity mix (28). These factors are used to account the amount of $CO_2$ emitted by different energy supply systems. By using an example, only considering the heating needs as described in Table 3, the expected amount of $CO_2$ – production for different energy supply systems can be calculated. As an illustration; the amount of $CO_2$ emitted by micro-CHP supply systems can then be compared with the amount of $CO_2$ emitted by other conventional energy supply systems in Norway.

System efficiencies are taken from NS 3031 (26) as can be seen in Table 4. Natural gas is the assumed fuel for both the CHP unit and the gas boiler. Hence, the $CO_2$-production factor for natural gas is used in calculations where heat or electricity is delivered from these devices.

| **Delivered energy** | **$CO_2$ – production factor [g $CO_2$/kWh]** | |
|---|---|---|
| Electricity from the grid | Net-ZEB | UCTE |
| | 278 | 563.4 |
| District heating | 231 | |
| Natural gas | 211 | |

**Table 2: $CO_2$–production factors**

| **Heating needs** | |
|---|---|
| DHW needs | 2530 kWh |
| SH needs | 4000 kWh |
| Tot | 10963 kWh |

**Table 3: Heating needs for the example case.**

| **System efficiencies** | |
|---|---|
| Electric heaters | 0.98 |
| Electrical water heater | 0.98 |
| District heating | 0.88 |

**Table 4: System efficiencies**

# 8. System cases and configurations

This chapter outlines the cases selected for the performance assessment. Only a limited number of cases are investigated, as the parameter space for these systems are extremely large. Table 5 outlines all system cases.

| Case | Building type | Storage tank size | Boiler type | Strategy | Thernal comfort requirements | System configutration |
|---|---|---|---|---|---|---|
| **1** | PH | 300 | - | CHP all year | Normal | A |
| **2** | | 500 | - | CHP all year | Normal | A |
| **3** | | 800 | - | CHP all year | Normal | A |
| **4** | LE | 300 | - | CHP all year | Normal | A |
| **5** | | 500 | - | CHP all year | Normal | A |
| **6** | | 800 | - | CHP all year | Normal | A |
| **7** | PH | 300 | - | CHP all year | Strict | A |
| **8** | | 300 | - | CHP all year | Less stringent | A |
| **9** | | 300 | Normal | Boiler all year | Normal | B |
| **10** | | 300 | Normal | CHP for HS + boiler outside HS | Normal | B |
| **11** | | 300 | Condensing | Boiler all year | Normal | B |
| **12** | | 300 | Condensing | CHP for HS + boiler outside HS | Normal | B |
| **13** | LE | 300 | Normal | Boiler all year | Normal | B |
| **14** | | 300 | Normal | CHP for HS + boiler outside HS | Normal | B |
| **15** | | 300 | Condensing | Boiler all year | Normal | B |
| **16** | | 300 | Condensing | CHP for HS + boiler outside HS | Normal | B |

**Table 5: System cases and configurations.**

In case 1 – 6 the performance of the systems are investigated with different storage tank sizes. Further, in case 7 and 8, the effect of changing the thermal comfort requirements are investigated. In case 7, having strict requirements, the dead band of the space heating controller is only 1°C while the set temperature is 20.5°C. In case 8, the requirements are more tolerant having an extended dead band of 3°C and a set temperature of 21.5°C.

In case 9 – 16 the performance of the systems are investigated and attempted rectified by adding an auxiliary GB. Cases having the GB to supply the entire heat demand throughout the year are also simulated.

## 8.1 Emissions analysis in a net-ZEB context

The energy supply systems to be compared are:

Case I: A micro – CHP unit supplying the entire heating needs.
Case II: A combination of CHP and a condensing GB supplying the heating needs.
Case III: A condensing GB to cover the entire heat demand.
Case IV: Electric heaters to cover the entire heat demand.
Case V: District heating (DH) to cover the entire heat demand.

The aim is to investigate and compare the amounts of $CO_2$ - emissions.

# 9. Results and discussion

This chapter presents the simulation results of the different system cases. A table containing results regarding the energy performance is presented first. This table also contains information on $CO_2$-emissions. Secondly, a table containing temporal characteristics of the CHP unit is presented. This includes the cycling frequency and information on how the unit operates in different modes.

## 9.1.1 Dynamic behavior

This subchapter illustrates the dynamic behaviour of the model. The illustrations are used to illustrate the validity of the model and also to increase knowledge about the behaviour.

Figure 9 illustrates the simultaneously production of heat and power as the engine moves between different operational modes (1-4). The power/heat ratio can be seen as quite large. The green line illustrates that the generated heat is assumed constant in both warm-up and normal mode, while the power production, represented by the blue line, is depending on the engine temperature during the warm-up period.

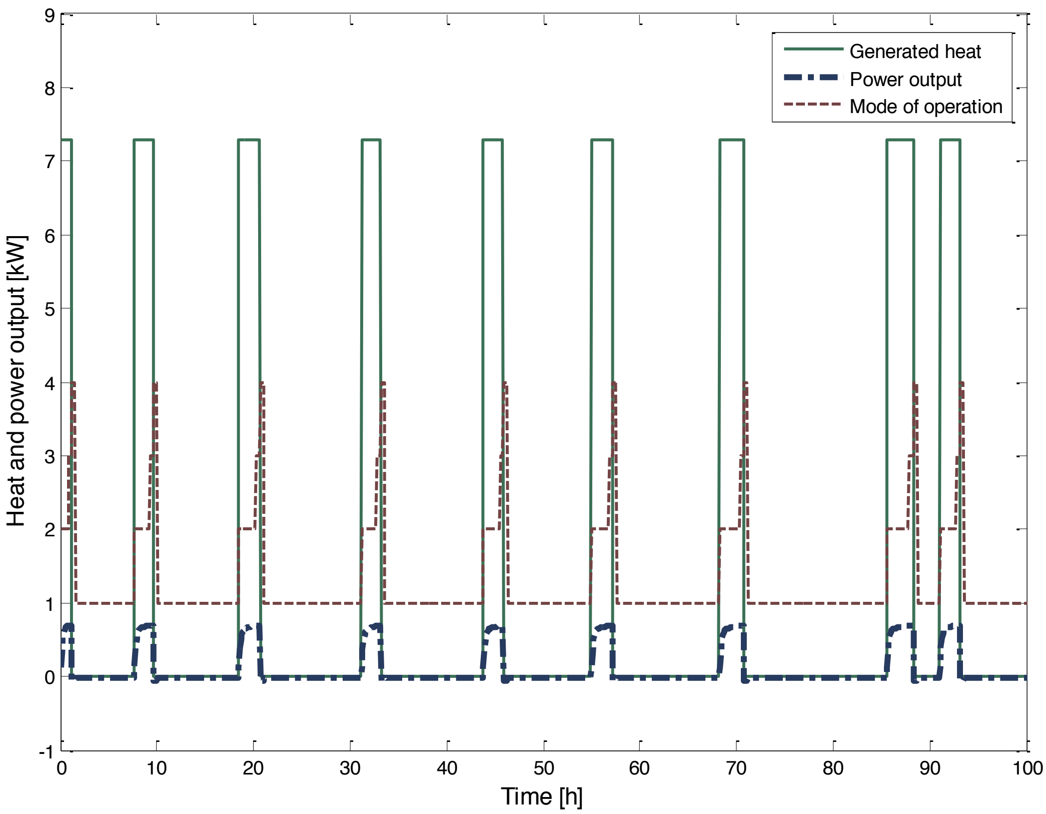


**Figure 9: Dynamic performance of the Stirling Engine.**

Figure 10 and Figure 11illustrates the dynamic behaviour of the engine and the storage tank temperature along with the operation of the CHP unit. Figure 10 includes a period where the warmest outdoor temperature occurs, while Figure 11 includes a period where the lowest outdoor temperature occurs during the year. As can be seen, the engine cycles more frequent when the climate is cold.

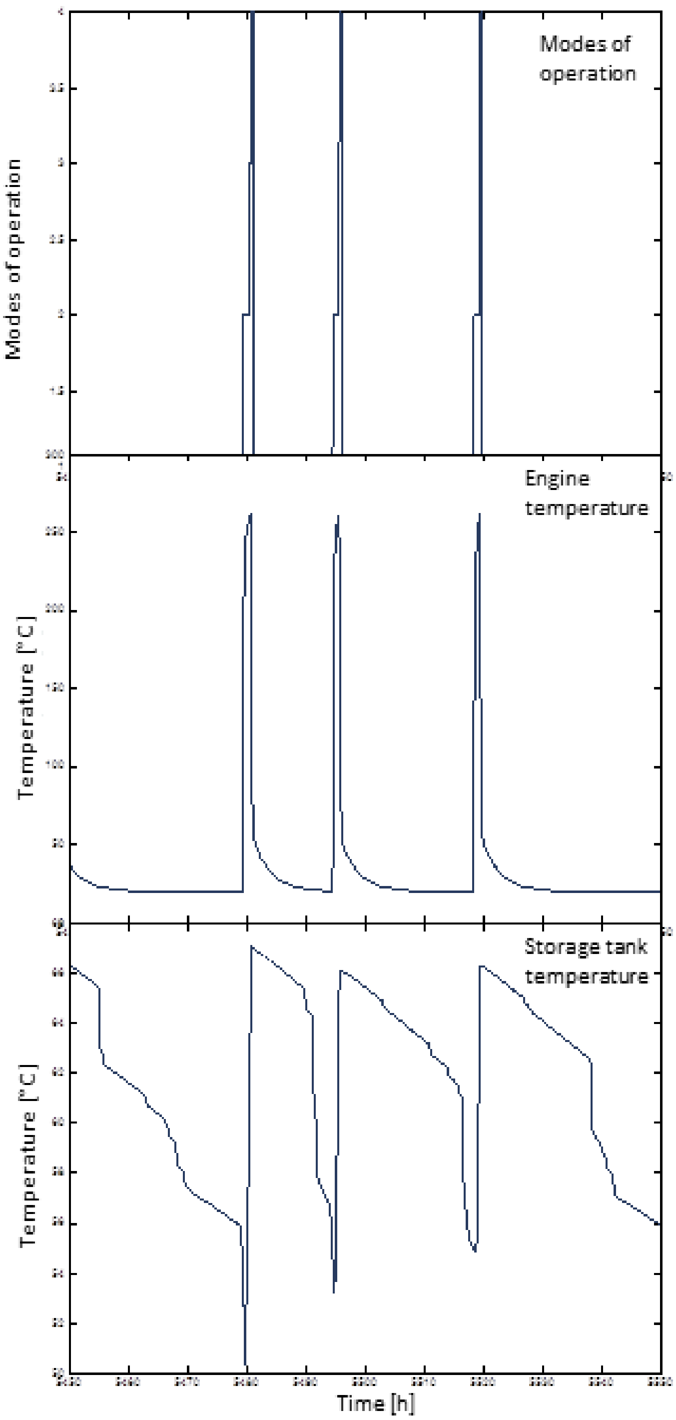


**Figure 10: Dynamic behaviour during a warm period where the outdoor temperature reaches its maximum.**

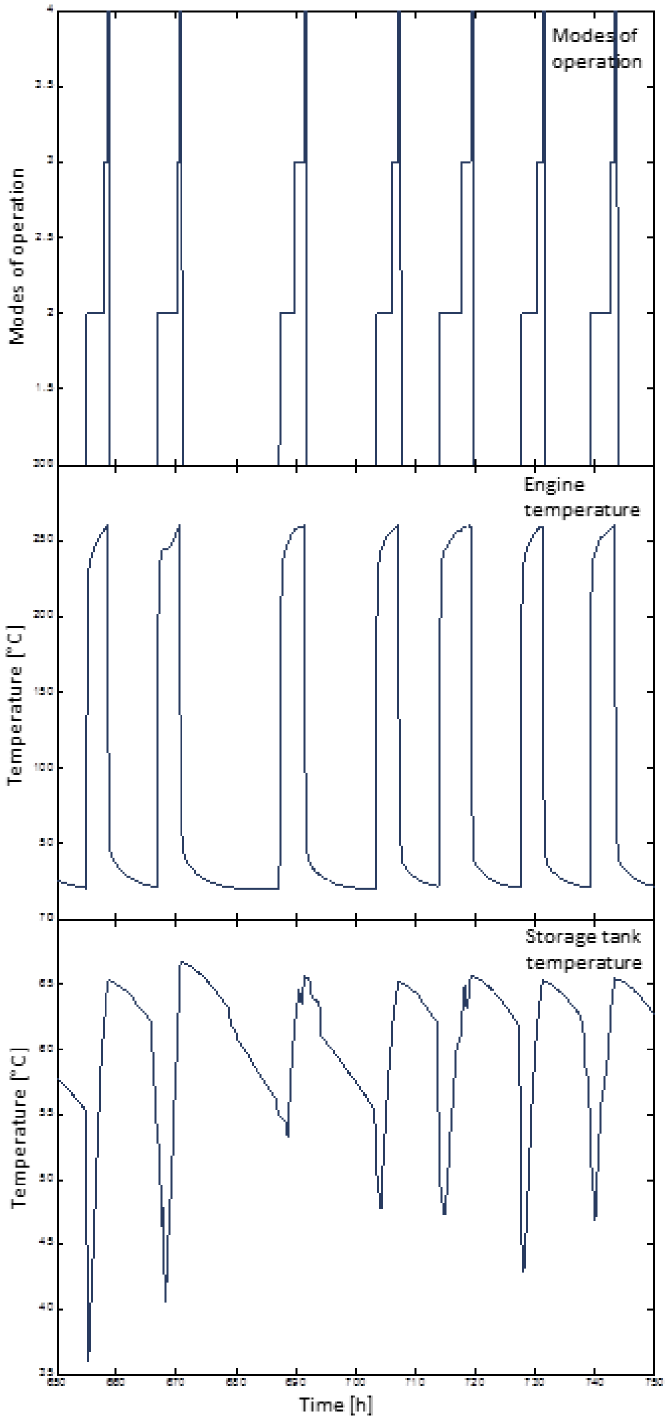


**Figure 11 Dynamic behaviour during a cold period where the outdoor temperature reaches its minimum.**

### Import and export with the electricity grid

When the CHP unit produces more power than what is requested by the building, the electricity can be exported to the electricity grid. When the requested electricity amount is larger than what is produced and when the CHP unit is in stand-by, electricity is imported from the grid.

Figure 12 and Figure 13 illustrates the dynamic import and export curve of electricity during one day with warm outdoor temperature and one day with cold outdoor temperature. The electricity amount above zero is the exported amount, while the negative area of the curve is what needs to be imported. No electricity is exported during the warm day, while some electricity is exported during the cold day due to the engine running more hours. The imported part is much greater than the exported part meaning that the amount of electricity exported at one moment is reimported at some other time.

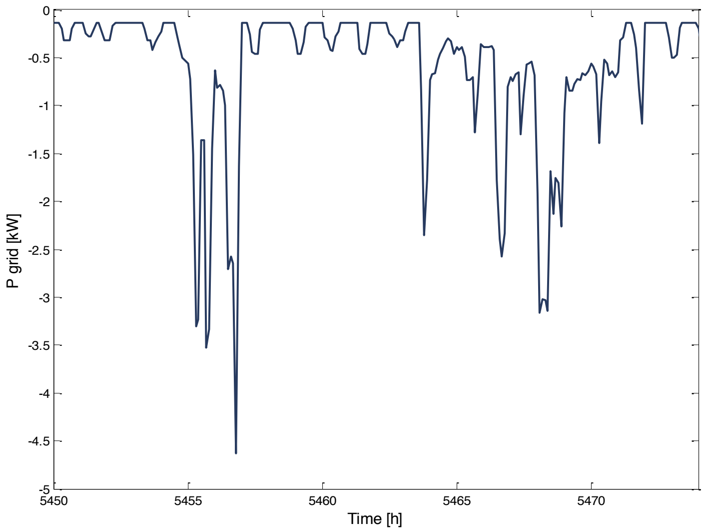


**Figure 12: Import and export of electricity during a warm period where the outdoor temperature reaches its maximum.**

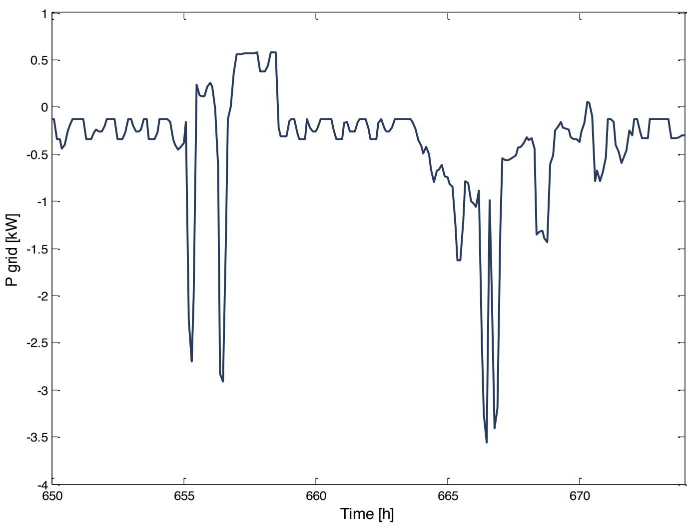


**Figure 13: Import and export of electricity during a cold period where the outdoor temperature reaches its minimum.**

## 9.2 Improving the performance using thermal mass/tank size

| Case | Gross input [kWh] | SH [kWh] | DHW [kWh] | Losses [kWh] | Net power [kWh] | $\eta_{CHP}$ [-] | $\eta_{system}$ [-] | $CO_2$ [kg] |
|---|---|---|---|---|---|---|---|---|
| **1** | 8341 | 3982 | 2289 | 341 | 592 | 0.876 | 0.775 | 1736 |
| **2** | 8432 | 3951 | 2335 | 471 | 620 | 0.886 | 0.768 | 1740 |
| **3** | 8551 | 3917 | 2347 | 622 | 644 | 0.892 | 0.755 | 1756 |
| **4** | 11588 | 6678 | 2228 | 325 | 883 | 0.886 | 0.792 | 2409 |
| **5** | 12158 | 7031 | 2285 | 451 | 955 | 0.894 | 0.790 | 2499 |
| **6** | 12302 | 7031 | 2296 | 597 | 986 | 0.899 | 0.782 | 2518 |

**Table 6: Energy and emission performance with different storage tank sizes supplying a PH and a LEB.**

| Case | Number of cycles | Time in normal/ warm-up [%] | Time in standby/ cool-down [%] | Longest period in normal mode [min] | Shortest period in normal mode [min] | Average duration normal mode [min] | Average duration standby [min] |
|---|---|---|---|---|---|---|---|
| **1** | 490 | 13 | 87 | 222 | 12 | 29 | 911 |
| **2** | 358 | 13 | 87 | 114 | 24 | 45 | 1253 |
| **3** | 270 | 13 | 87 | 150 | 48 | 63 | 1670 |
| **4** | 546 | 18 | 82 | 162 | 12 | 38 | 769 |
| **5** | 416 | 18 | 81 | 234 | 24 | 57 | 1006 |
| **6** | 296 | 19 | 81 | 186 | 48 | 85 | 1415 |

**Table 7: Temporal characteristics with different storage tank sizes supplying a PH and a LEB.**

The results show some correlations due to the storage tank size. The number of operational cycles decreases with increasing tank size, while the average duration of the normal mode periods increases. Due to a greater amount of water, the control system of a large storage tank will be less sensitive to heat load changes hence lower the unit cycling frequency. The CHP efficiency increases along with increasing tank sizes due to longer power-producing periods and decreasing cycling frequency.

Increased tank size is related to increased heat losses due to larger surface areas. Greater losses results in higher heat demand following more energy is generated by the CHP. Figure 14 illustrates that the heat loss becomes a greater part of the generated heat as the tank size increases. This is causing a small reduction of the system efficiencies along with the increasing tank sizes. The CHP efficiencies are larger than the system efficiencies due to losses in the system. Choosing a tank of 300 litres would be beneficial considering the system heat efficiency.

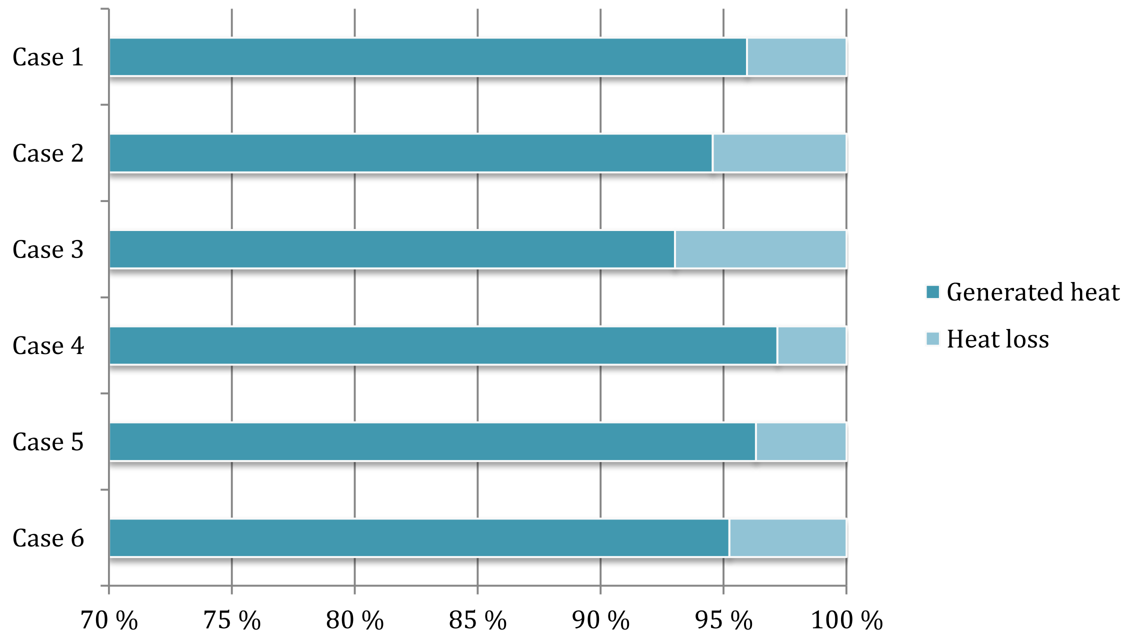


**Figure 14: Tank heat loss of generated heat with different tank sizes supplying a PH and a LEB.**

$CO_2$-emissions increases in accordance with the energy production resulting in lower $CO_2$-emissions when the tank is small. Figure 15 illustrates that the emissions constitute a smaller part of the total power production when the power production is high, as the tank size is large. The number of operational cycles and their length directly affects the net power output.  Hence, a system with steady production and few operational cycles is preferable in order to achieve low $CO_2$-emissions.

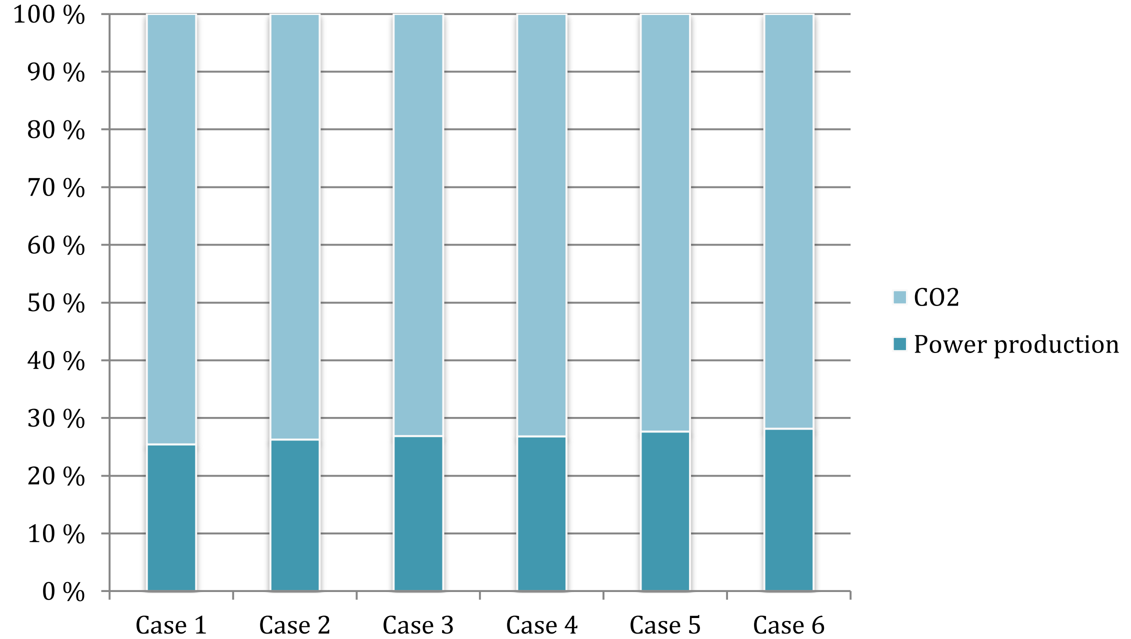


**Figure 15: $CO_2$ - emissions of total power production with different tank sizes supplying a PH and a LEB.**

### Heat demand correlations

Both the CHP efficiency and the system efficiency become higher when supplying the LEB. The temporal characteristics show that when supplying the LEB, the engine operates more minutes in normal and cool-down mode. It tends to operate for longer average periods at full power but it also cycles more often.

When heavily loaded, the engine needs to operate for longer periods of full power in order to stay in between the temperature-range of the storage tank. With a lighter heat load, when supplying the PH, the engine reaches its operating conditions more quickly and cycles only to stay within the temperature range.

The CHP unit produces more power when connected to the LEB because of the larger heating needs. Following, these systems are able to export a greater amount of electricity to the power grid. This will add a positive effect in the net-ZEB context, reducing the need for electricity import. Figure 16 illustrates that the amount of exported electricity poses a larger portion of the total electricity need as the heat demand and power production increases within the system cases. On the other hand, when supplying the LEB the total emitted amount of $CO_2$ are larger than when supplying the PH although the exported amount of electricity is greater.

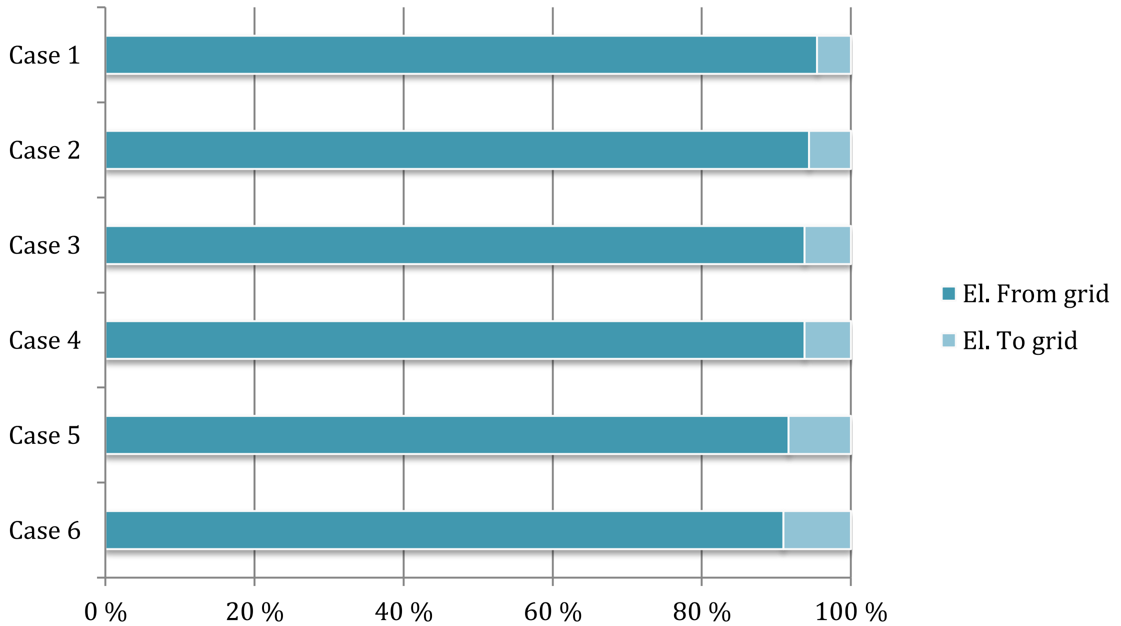


**Figure 16: Percentage import/export with different tank sizes supplying a PH and a LEB.**

# 9.3 Loosening the requirements to thermal comfort

| Case | Set temp/ dead band | Gross input [kWh] | SH [kWh] | DHW [kWh] | Tank losses [kWh] | Net power [kWh] | $\eta_{CHP}$ [-] | $\eta_{system}$ [-] |
|---|---|---|---|---|---|---|---|---|
| **7** | 20,5/1 | 8463 | 4096 | 2260 | 321 | 594 | 0.871 | 0.776 |
| **8** | 21,5/3 | 8341 | 3982 | 2289 | 341 | 592 | 0.876 | 0.777 |

**Table 8: Energy performance with strict and less stringent requirements to the thermal comfort.**

| Case | Set temp/ dead - band | Number of cycles | Time in normal/ warm-up [%] | Time in standby/ cool-down [%] | Longest period in normal mode [min] | Shortest period in normal mode [min] | Average duration normal mode [min] | Average duration standby [min] |
|---|---|---|---|---|---|---|---|---|
| **7** | 20,5/1 | 540 | 13 | 87 | 114 | 12 | 25 | 823 |
| **8** | 21,5/3 | 490 | 13 | 87 | 222 | 12 | 29 | 911 |

**Table 9: Temporal characteristics with strict and less stringent requirements to the thermal comfort.**

It can be seen in Figure 17 and Figure 18 that the space heating signal changes more rapidly when the requirements to the thermal comfort is strict, compared to when the requirements are less stringent. This has a negative impact on the rest of the system, causing the CHP unit to cycle more frequent and reducing the duration of the periods in normal mode. Loosening on the requirements results in somewhat higher CHP efficiency while the system efficiency seems almost unchanged. The strongest argument for choosing a space heating control with a larger dead band is to avoid the control being unstable, which may occur with a very small dead band.

It can be seen in Figure 18 that although the signal switches less frequent when having less stringent requirements, the on-periods are still very short. This means that the air temperature reaches the set temperature very quickly after the system has started to deliver heat to the room. One possible solution to avoid it is to further expand the dead band. This may cause a notable reduction of the thermal comfort in the room, as humans are very sensitive to temperature changes.

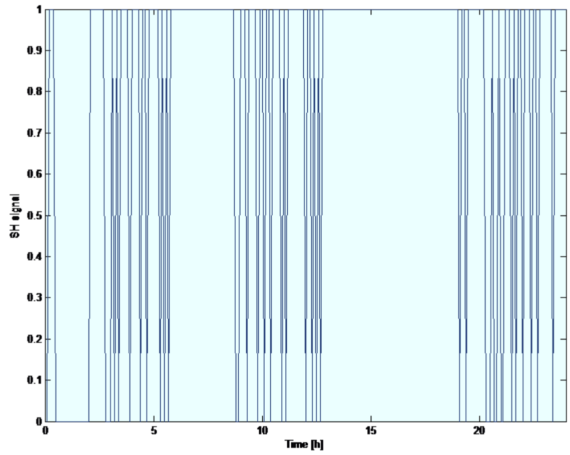


**Figure 17: Space heating signal with 3 °C dead band**

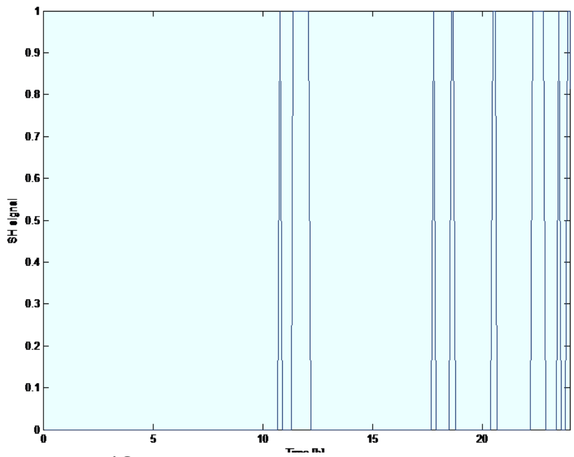


**Figure 18: Space heating signal with 1 °C dead band**

# 9.4 Improving the performance with an auxiliary gas boiler

| Case | Heat from CHP [%] | Heat from GB [%] | SH [kWh] | DHW [kWh] | $\eta_{CHP}$ [-] | $\eta_{boiler}$ [-] | $\eta_{system}$ [-] | $CO_2$-emissions [kg] |
|---|---|---|---|---|---|---|---|---|
| **9** | 0 | 100 | 4036 | 2327 | - | 0.86 | 0.93 | 1638 |
| **10** | 83 | 17 | 3977 | 2292 | 0.88 | 0.86 | 0.80 | 1707 |
| **11** | 0 | 100 | 4040 | 2321 | - | 0.105 | 0.93 | 1341 |
| **12** | 83 | 17 | 3976 | 2294 | 0.88 | 0.104 | 0.80 | 1663 |
| **13** | 0 | 100 | 7426 | 2295 | - | 0.86 | 0.95 | 2453 |
| **14** | 88 | 12 | 6678 | 2227 | 0.89 | 0.86 | 0.81 | 2378 |
| **15** | 0 | 100 | 7424 | 2290 | - | 0.105 | 0.94 | 2018 |
| **16** | 88 | 12 | 6676 | 2228 | 0.89 | 0.104 | 0.81 | 2331 |

**Table 10: Energy and emission performance of systems with CHP and/or GB supplying a PH and a LEB.**

| Case | Number of cycles | Time in normal/ warm-up [%] | Time in standby/ cool-down [%] | Time switched off [%] | Longest period in normal mode [min] | Shortest period in normal mode [min] | Average duration normal mode [min] | Average duration standby [min] |
|---|---|---|---|---|---|---|---|---|
| **9** | - | - | - | 100 | - | - | - | - |
| **10** | 338 | 11 | 49 | 40 | 222 | 12 | 33 | 745 |
| **11** | - | - | - | 100 | - | - | - | - |
| **12** | 337 | 11 | 49 | 40 | 234 | 12 | 33 | 748 |
| **13** | - | - | - | 100 | - | - | - | - |
| **14** | 392 | 15 | 45 | 40 | 162 | 12 | 44 | 577 |
| **15** | - | - | - | 100 | - | - | - | - |
| **16** | 391 | 15 | 45 | 40 | 162 | 12 | 45 | 578 |

**Table 11: Temporal characteristics of the CHP unit in systems having both CHP and GB supplying a PH and a LEB.**

The total efficiency of the condensing GB is calculated to be 104% for both the PH and the LEB case. When the GB is supplying the entire annual heating needs the efficiency becomes 105%. These efficiencies are much higher than the efficiency of the regular GB at 86%. Although the condensing GB has much greater efficiency, the system efficiencies seem unaffected. However, the boiler efficiency affects the $CO_2$-emissions, achieving lower emissions when using a condensing GB.

| Case | Net power [kWh] | Heat from CHP [%] | Heat from GB [%] | $\eta_{CHP}$ [-] | $\eta_{boiler}$ [-] | $\eta_{system}$ [-] | $CO_2$-emissions [kg] |
|---|---|---|---|---|---|---|---|
| **1** | 592 | 100 | 0 | 0.88 | - | 0.78 | 1736 |
| **10** | 525 | 83 | 17 | 0.88 | 0.86 | 0.80 | 1707 |
| **12** | 525 | 83 | 17 | 0.88 | 0.104 | 0.80 | 1663 |
| **4** | 883 | 100 | 0 | 0.89 | - | 0.79 | 2409 |
| **14** | 815 | 88 | 12 | 0.89 | 0.86 | 0.81 | 2378 |
| **16** | 815 | 88 | 12 | 0.89 | 0.104 | 0.81 | 2331 |

**Table 12: Energy and emission performance of three different systems supplying a PH and a LEB.**

| Case | Number of cycles | Time in normal/ warm-up [%] | Time in standby/ cool-down [%] | Longest period in normal mode [min] | Shortest period in normal mode [min] | Average duration normal mode [min] | Average duration standby [min] |
|---|---|---|---|---|---|---|---|
| **1** | 490 | 13 | 87 | 222 | 12 | 29 | 911 |
| **10** | 338 | 18 | 83 | 234 | 12 | 33 | 745 |
| **12** | 337 | 18 | 83 | 234 | 12 | 33 | 748 |
| **4** | 546 | 18 | 82 | 162 | 12 | 38 | 769 |
| **14** | 392 | 26 | 75 | 162 | 12 | 44 | 577 |
| **16** | 391 | 26 | 75 | 162 | 12 | 45 | 578 |

**Table 13: Temporal characteristics of the CHP unit in three different systems supplying a PH and a LEB.**

Table 12 and Table 13 are created in order to evaluate and compare the systems containing different auxiliary GBs with the systems only supplied by CHP. The first three systems are supplying a PH while the three cases at the bottom supply a LEB.

Switching off the CHP unit outside the heating season will obviously lead to shorter time of operation and the power production becomes lower. The CHP efficiency remains almost unchanged, while the system efficiency becomes greater when having both a CHP unit and a GB.

It can be seen from the results that the number of cycles decreases when a GB is present. The average duration of the periods in normal mode increases and the average duration in standby mode decreases. As the GB is added to the system, the CHP unit is made unavailable outside the heating season operating 18% and 26% of the time available. When the CHP unit operates alone, throughout the year, the time of operation gets reduced to 13% and 18%. The numbers represents PH and the LEB cases, respectively.

The effect of having longer average periods in normal mode and lower cycling frequency can be seen from the emission results, emitting less $CO_2$ when a GB is present.

## 9.5 Emissions analysis in an net-ZEB context

Figure 19 illustrates the amount of $CO_2$ emitted by different energy supply systems using both the $CO_2$-production factor within the net-ZEB definition and the $CO_2$-production factor using the European UCTE electricity mix.

When the total heating demand is covered by electricity from the grid, the emissions become the greatest. Emissions from the systems including CHP depends on the electricity mix, hence the emissions become lower when the $CO_2$-production factor for electricity is large. When using the UCTE electricity mix, the system only consisting of CHP emits the least $CO_2$.

The $CO_2$ factor for electricity is expected to decrease over the years as the electricity production is expected to become greener (27). The emission factor for natural gas will remain unchanged. This may lead to poorer ratings of the CHP systems regarding $CO_2$-emissions in the future.

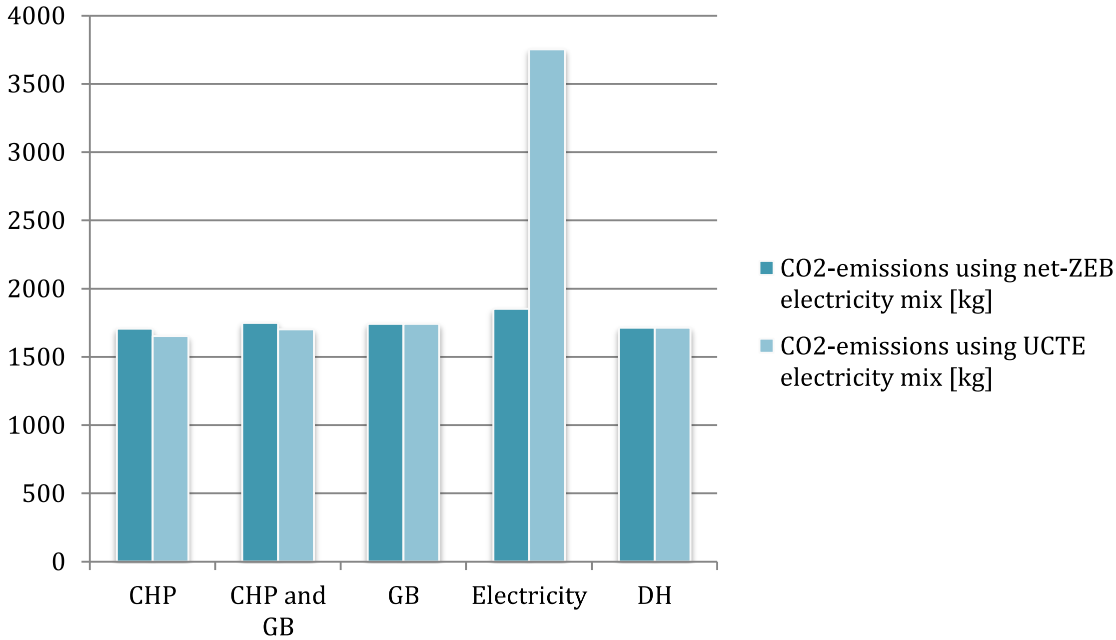


**Figure 19: $CO_2$-emissions for different energy supply systems using different $CO_2$-production factors for the electricity**

# 10. Conclusions

This thesis investigates the performance of micro-CHP systems supplying a PH and a LEB by computer simulations.

The results show that the system efficiency becomes grater when the heat demand is higher as all the systems supplying the LEB obtained a greater efficiency compared to the systems supplying the PH. The CHP unit operated more hours at full power production and was able to export more electricity to the power grid, which is favorable in a net-ZEB context.

A system having a large storage tank will experience a larger heating demand and the operation of the CHP unit becomes more favorable. But due to larger surface areas, a large tank will also experience larger heat losses. These losses constitute a greater part of the stored heat as the tank size increases. This effect results in poorer system efficiency for the systems having larger storage tanks. Hence, increasing the thermal mass of the system does not improve the performance of the system. The system with the smallest storage tank of 300 liters, supplying the LEB, obtained the highest system efficiency of 79.2%.

The requirements to the thermal comfort are shown to have an impact on the operation of the CHP unit. Having less stringent requirements is beneficial for the operation of the CHP unit, but the comfort in the room must be ensured. A control that ensures thermal comfort in the room and at the same time does not need to switch on and off so often is preferable for the system efficiency.

$CO_2$-emissions increases along with the power output. Hence, supplying the LEB leads to greater $CO_2$-emissions than when supplying the PH. What should be noted is that the emissions do not necessarily increase linearly with the power output, depending on the duration of the periods at maximal production and the number of operational cycles. When supplying the LEB, the $CO_2$-emissions constitute a smaller part of the total power output compared to when supplying the PH. The conclusion to be drawn is that a system with steady production is preferable in order to achieve low $CO_2$-emissions; producing power at high efficiency and at the same time increases the possibility of exporting electricity to the grid.

In order to improve the system efficiency, an auxiliary GB was added to the system. The CHP unit experienced longer average operating time in normal mode, as well as less operational cycling. When adding a condensing boiler, the system obtained an efficiency of 80% when supplying the PH and 81% when supplying the LEB. These results indicate that the CHP systems benefit from having an auxiliary heater supplying the periods of low heating needs.

The system became more efficient when supplying a LEB compared to a PH, but the energy consumption and emissions are larger. In a net-ZEB context a PH is more favorable due to the low heat consumption and low emissions. In order to achieve an efficient and well-functional system, even with low heating needs, the system should include an auxiliary heater and the control of the system need to be correctly implemented, making the CHP unit operate at long hours at full power production with low cycling frequency.

## 10.1 Evaluation in an net-ZEB context

The results from $CO_2$ - emission calculations performed on five different energy supply systems indicate that the use of electricity causes high emissions and is not a preferable choice when it comes to heating of buildings in a net-ZEB context. The CHP systems are more beneficial when the $CO_2$-production factor for electricity is high. When using the UCTE electricity mix the supply system only consisting of CHP had the lowest emissions. Hence, the results show that domestic CHP systems fuelled by natural gas can be seen as attractive solutions in a net-ZEB context. Taking into account that the $CO_2$-production factor for electricity will increase over the years, the CHP-technology will need further development in order to retain its position in a net-ZEB context.

## 10.2 Further work

This thesis includes only a few considerations when it comes to evaluating the integration of a CHP system. There are a lot of parameters in the model that could be investigated in order to improve the performance. Also, combinations with other energy generating technology could be investigated as well as other methods of controlling the system.

As the model did not manage to work properly calibrated as an ICE, no conclusions can be drawn regarding systems supplied by this engine type. The model should be further inspected in order to determine the exact reason causing the errors. The fact that the model is generic constructed, the ability to operate as an ICE ought to be possible.

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# 12. Appendix

## Appendix A
## Mathematical model description

### Energy conversion control volume

The energy conversion control volume feeds information from the engine unit performance map into the engine model in form of a heat flux. This control volume represents the engine working fluid, combustion gasses and the engine alternator.

Steady state energy balance for the engine control volume is:

$$\dot{\boldsymbol{H}}_{\boldsymbol{fuel}} + \dot{\boldsymbol{H}}_{\boldsymbol{air}} = \boldsymbol{P}_{\boldsymbol{net,ss}} + \boldsymbol{q}_{\boldsymbol{gen,ss}} + \dot{\boldsymbol{H}}_{\boldsymbol{exh}} \tag{1}$$

$\dot{H}_{fuel}$ *is the total enthalpy of the fuel.*
$\dot{H}_{air}$ *is the total entalpy of the combustion air.*
$P_{net,ss}$ *is the rate of the steady − state electricity production.*
$q_{gen,ss}$ *is the rate steady − state heat production.*
$\dot{H}_{exh}$ *is the total entalpy of the exhaust gases.*

As a simplification, the engine’s steady-state (part load) performance is correlated to the total energy input to the system:

$$\boldsymbol{P}_{\boldsymbol{net,ss}} = \boldsymbol{\eta}_{\boldsymbol{e}}\boldsymbol{q}_{\boldsymbol{gross}} \tag{2}$$

$$\boldsymbol{q}_{\boldsymbol{gen,ss}} = \boldsymbol{\eta}_{\boldsymbol{q}}\boldsymbol{q}_{\boldsymbol{gross}} \tag{3}$$

$$\boldsymbol{q}_{\boldsymbol{gross}} = \dot{\boldsymbol{m}}_{\boldsymbol{fuel}}\,\boldsymbol{LHV}_{\boldsymbol{fuel}} \tag{4}$$

$P_{net,ss}$ *is the steady − state electrical outputof the system* $(W)$.
$\eta_e$ *is the steady − state electrical conversion effeiciency of the engine.*
$q_{gen,ss}$ *is the steady − state rate of heat generation within the engine* $(W)$.
$\eta_q$ *is the steady − state part load, thermal effeiciency of the engine.*
$q_{gross}$ *is the gross heat input into the system* $(W)$.
$\dot{m}_{fuel}$ *is the fuel flow rate* $(kg/s)$.
$LHV_{fuel}$ *is the lower heating value of the fuel* $(J/kg)$.

The system’s part load electrical and thermal efficiencies are determined using empirical correlations:

$$\boldsymbol{\eta}_{\boldsymbol{e}} = \boldsymbol{f}(\dot{\boldsymbol{m}}_{\boldsymbol{cw}}, \boldsymbol{T}_{\boldsymbol{cw}}, \boldsymbol{P}_{\boldsymbol{net,ss}}) \tag{5}$$

$$\boldsymbol{\eta}_{\boldsymbol{q}} = \boldsymbol{f}(\dot{\boldsymbol{m}}_{\boldsymbol{cw}}, \boldsymbol{T}_{\boldsymbol{cw}}, \boldsymbol{P}_{\boldsymbol{net,ss}}) \tag{6}$$

$\dot{m}_{cw}$ *is the mass flow rate of the cooling water* $(kg/s)$.
$T_{cw}$ *is the temperature of the cooling water at the inlet of the cooling water control volume* $(°C)$.

Together, these correlations constitute a “performance map” describing the cogeneration system’s steady-state behaviour under a variety of loading conditions.

The full performance map expression for electrical efficiency is:

$$\begin{aligned}\boldsymbol{\eta_e} = {} & \boldsymbol{a_0 + a_1 P_{net,ss}^2 + a_2 P_{net,ss} + a_3 \dot{m}_{cw}^2 + a_4 \dot{m}_{cw} + a_5 T_{cw}^2 + a_6 T_{cw} + a_7 P_{net,ss}^2 \dot{m}_{cw}^2} \\ & \boldsymbol{+ a_8 P_{net,ss} \dot{m}_{cw} + a_9 P_{net,ss} \dot{m}_{cw}^2 + a_{10} P_{net,ss}^2 \dot{m}_{cw} + a_{11} P_{net,ss}^2 T_{cw}^2} \\ & \boldsymbol{+ a_{12} P_{net,ss} T_{cw} + a_{13} P_{net,ss} T_{cw}^2 + a_{14} P_{net,ss}^2 T_{cw} + a_{15} \dot{m}_{cw}^2 T_{cw}^2} \\ & \boldsymbol{+ a_{16} \dot{m}_{cw} T_{cw} + a_{17} \dot{m}_{cw} T_{cw}^2 + a_{18} \dot{m}_{cw}^2 T_{cw} + a_{19} P_{net,ss}^2 \dot{m}_{cw}^2 T_{cw}^2} \\ & \boldsymbol{+ a_{20} P_{net,ss}^2 \dot{m}_{cw}^2 T_{cw} + a_{21} P_{net,ss}^2 \dot{m}_{cw} T_{cw}^2 + a_{22} P_{net,ss} \dot{m}_{cw}^2 T_{cw}^2} \\ & \boldsymbol{+ a_{23} P_{net,ss}^2 \dot{m}_{cw} T_{cw} + a_{24} P_{net,ss} \dot{m}_{cw}^2 T_{cw} + a_{25} P_{net,ss} \dot{m}_{cw} T_{cw}^2} \\ & \boldsymbol{+ a_{26} P_{net,ss} \dot{m}_{cw} T_{cw}}\end{aligned} \tag{7}$$

Similarly, for the thermal efficiency:

$$\begin{aligned}\boldsymbol{\eta_q} = {} & \boldsymbol{b_0 + b_1 P_{net,ss}^2 + b_2 P_{net,ss} + b_3 \dot{m}_{cw}^2 + b_4 \dot{m}_{cw} + b_5 T_{cw}^2 + b_6 T_{cw} + b_7 P_{net,ss}^2 \dot{m}_{cw}^2} \\ & \boldsymbol{+ b_8 P_{net,ss} \dot{m}_{cw} + b_9 P_{net,ss} \dot{m}_{cw}^2 + b_{10} P_{net,ss}^2 \dot{m}_{cw} + b_{11} P_{net,ss}^2 T_{cw}^2} \\ & \boldsymbol{+ b_{12} P_{net,ss} T_{cw} + b_{13} P_{net,ss} T_{cw}^2 + b_{14} P_{net,ss}^2 T_{cw} + b_{15} \dot{m}_{cw}^2 T_{cw}^2} \\ & \boldsymbol{+ b_{16} \dot{m}_{cw} T_{cw} + b_{17} \dot{m}_{cw} T_{cw}^2 + b_{18} \dot{m}_{cw}^2 T_{cw} + b_{19} P_{net,ss}^2 \dot{m}_{cw}^2 T_{cw}^2} \\ & \boldsymbol{+ b_{20} P_{net,ss}^2 \dot{m}_{cw}^2 T_{cw} + b_{21} P_{net,ss}^2 \dot{m}_{cw} T_{cw}^2 + b_{22} P_{net,ss} \dot{m}_{cw}^2 T_{cw}^2} \\ & \boldsymbol{+ b_{23} P_{net,ss}^2 \dot{m}_{cw} T_{cw} + b_{24} P_{net,ss} \dot{m}_{cw}^2 T_{cw} + b_{25} P_{net,ss} \dot{m}_{cw} T_{cw}^2} \\ & \boldsymbol{+ b_{26} P_{net,ss} \dot{m}_{cw} T_{cw}}\end{aligned} \tag{8}$$

$a_0 - a_{26}$ *and* $b_0 - b_{26}$ *are empirically derived coefficients.*

### Engine control volume

The engine control volume represents the dynamic thermal behaviour of the cogeneration device. This control volume represents the response of the thermal mass of the engine block and encapsulated working fluid, internal heat exchange equipment and the external heater.

The energy balance of the engine control volume is:

$$[\boldsymbol{MC}]_{eng} \frac{\boldsymbol{dT_{eng}}}{\boldsymbol{dt}} = \boldsymbol{q_{gen,ss} - q_{HX} - q_{skin-loss}} \tag{9}$$

$[MC]_{eng}$ *is the thermal capacitance of the control volume* $(W/K)$.
$q_{HX}$ *is the rate of heat transfer to the cooling water* $(W)$.
$q_{skin-loss}$ *is the rate of heat loss from the engine* $(W)$.
$T_{eng}$ *is the bulk temperature of the thermal mass control volume* $(°C)$.
$t$ *is time.*

The heat transfer between the engine and the cooling water control volume can be expressed as follows using an overall heat-transfer coefficient:

$$\boldsymbol{q_{HX} = UA_{HX}\left(T_{eng} - T_{cw,out}\right)} \tag{10}$$

$UA_{HX}$ *is the overall thermal conductance between the thermal mass and the cooling water control volumes* $(W/K)$.

It is assumed that the heat loss from the engine is proportional to the temperature difference between the engine and the surroundings and can be expressed as follows:

$$q_{skin-loss} = UA_{loss}(T_{eng} - T_{room}) \tag{11}$$

$UA_{loss}$ *is the effective thermal conductance between the engine control volume and the surroundings* $(W/K)$.
$T_{room}$ *is the surrounding temperature* $(°C)$.

Using equation (10) and (11), equation (12) can be expressed as:

$$[MC]_{eng}\frac{dT_{eng}}{dt} = q_{gen,ss} + UA_{HX}(T_{cw,out} - T_{eng}) + UA_{loss}(T_{room} - T_{eng}) \tag{12}$$

### Cooling water control volume

The cooling water control volume represents the cooling water flowing through the device and the elements of the heat exchanger in immediate thermal contact. Heat is transferred to the cooling water from the exhaust gases and the engine casing.

$$[MC]_{cw}\frac{T_{cw,out}}{dt} = [\dot{m}c_p]_{cw}(T_{cw,in} - T_{cw,out}) + q_{HX} \tag{13}$$

$[MC]_{cw}$ *is the thermal capacitance of the control volume* $(J/K)$.
$T_{cw,out}$ *is the bulk outlet temperature* $(°C)$.
$T_{cw,in}$ *is the inlet temperature* $(°C)$.
$[\dot{m}c_p]_{cw}$ *is the thermal capacity flow rate of the coling water* $(W/K)$.

Using equation (10) and (13) equation (14) can be expressed as:

$$[MC]_{cw}\frac{T_{cw,out}}{dt} = [\dot{m}c_p]_{cw}(T_{cw,in} - T_{cw,out}) + UA_{HX}(T_{eng} - T_{cw,out}) \tag{14}$$

It is assumed that the cogeneration device can regulate the cooling water flow rate in order to optimize the engine performance and heat recovery. This internal control is not directly modelled; instead the effect is represented in the system performance map through an empirical correlation:

$$\dot{m}_{cw} = c_0 + c_1P_{net,ss}^2 + c_2P_{net,ss} + c_3T_{cw}^2 + c_4T_{cw} + c_5P_{net,ss}^2T_{cw}^2 + c_6P_{net,ss}T_{cw} + c_7 + c_8P_{net,ss}^2T_{cw} \tag{15}$$

$c_0 - c_8$ *are empirically – derived coefficients*.

The airflow into the unit does not affect the model's thermal or electrical performance predictions. The air stoichiometry is regulated to manage the cogeneration unit's combustion efficiency, operating temperature and emissions. The single control volume does not provide adequate resolutions to quantify these effects. Instead, the airflow is expressed as follows:

$$\dot{m}_{air} = f(\dot{m}_{fuel}) = d_0 + d_1\dot{m}_{fuel}^2 + d_2\dot{m}_{fuel} \tag{16}$$

$\dot{m}_{air}$ *is the combustion air flow rate* $(kg/s)$.
$d_0 - d_2$ *are empirically – derived coefficients* $(-)$.

## Changing operation point

This model has constraints on the maximum rate of change permitted in the system fuel flow. Fuel flow change can be described using empirically data:

$$\frac{d\dot{m}_{fuel}}{dt} = \frac{\left|\dot{m}_{fuel,demand}^{t+\Delta t} - \dot{m}_{fuel}^{t}\right|}{\Delta t} \tag{17}$$

$\frac{d\dot{m}_{fuel}}{dt}$ *is the rate of fuel flow change* $(kg/s)$.

$\dot{m}_{fuel,demand}^{t+\Delta t}$ *is the system fuel flow rate requested* $(kg/s)$.
$t$ *is time* $(s)$.
$\Delta t$ *is the duration of the simulation time step* $(s)$.

The fuel flow rate requested from the cogeneration unit can then be adjusted to the maximum allowable rate of change:

$$\dot{m}_{fuel}^{t+\Delta t} = \begin{cases} \dot{m}_{fuel,demand}^{t+\Delta t} \; if \; \frac{d\dot{m}_{fuel}}{dt} \leq \left(\frac{d\dot{m}_{fuel}}{dt}\right)_{max} \\ \dot{m}_{fuel}^{t} \pm \left(\frac{d\dot{m}_{fuel}}{dt}\right)_{max} \; if \; \frac{d\dot{m}_{fuel}}{dt} > \left(\frac{d\dot{m}_{fuel}}{dt}\right)_{max} \end{cases} \tag{18}$$

$\left(\frac{d\dot{m}_{fuel}}{dt}\right)_{max}$ *is the empirically derived maximum rate of fuel flow change* $(kg/s)$.

The dynamic response of the cogeneration system's delivery depends on several constraints that the model does not characterize. The system's power delivery can be constrained using empirically data instead. Changes in the electrical output can be described as follows:

$$\frac{dP_{net}}{dt} = \frac{\left|P_{net,ss}^{t+\Delta t} - P_{net}^{t}\right|}{\Delta t} \tag{19}$$

$\frac{dP_{net}}{dt}$ *is the rate of change of the system'selectrical output* $(W/s)$.

The rate of change in the cogeneration system's power output can be adjusted to the maximum rate of change derived from empirical data:

$$P_{net}^{t+\Delta t} = \begin{cases} P_{net,ss}^{t+\Delta t} \; if \; \frac{dP_{net}}{dt} \leq \left(\frac{dP_{net}}{dt}\right)_{max} \\ P_{net}^{t} \pm \left(\frac{dP_{net}}{dt}\right)_{max} \; if \; \frac{dP_{net}}{dt} > \left(\frac{dP_{net}}{dt}\right)_{max} \end{cases} \tag{20}$$

$\left(\frac{dP_{net}}{dt}\right)_{max}$ $is\ the\ maximum\ rate\ of\ change\ in\ the\ system's\ electrical\ output\ (W/s).$

# Modes of operation

Besides normal operation, cogeneration units can operate in three more operating modes. These have remarkable different characteristics and are standby, warm-up and cool-down.

### Standby

When no electric or thermal output is requested, the unit is in standby mode. The unit consumes no fuel and produces no heat or electricity. However, the electronic control system within the unit requires some electricity. During standby it is assumed as follows:

$P_{net,ss} = P_{net,standby}$
$Q_{gen,ss} = 0$
$\dot{m}_{fuel} = 0$

$P_{net,ss}$ $is\ the\ system's\ powerdelivery\ (W).$
$P_{net,stand-by}$ $is\ the\ power\ used\ by\ the\ unit's\ control\ system\ (W).$
$Q_{gen,ss}$ $is\ the\ steady\ state\ heat\ generation\ (W).$
$\dot{m}_{fuel}$ $is\ the\ fuel\ flow\ rate\ (kg/s).$

### Warm-up mode

#### a) Stirling engines

During the warm-up period, fuel flow and electric power output differ considerably from their steady-state values. The engine temperature is under operational temperature and there is not enough heat transfer to the engine to produce the requested power. The fuel flow to the heater may increase in order to raise the heater temperature as quick as possible.

Some assumptions are made about the warm-up period:

- The fuel flow value is raised to the level corresponding to the engine's maximum steady-state power output.
- This value is further increased by an amount inversely proportional to the difference between the nominal and the actual engine temperature.

The fuel flow during warm-up can be expressed as:

$$\dot{m}_{fuel,warm-up} = \dot{m}_{fuel,ss,max} + k_f \dot{m}_{fuel,ss-max} \left(\frac{T_{eng,nom} - T_{room}}{T_{eng} - T_{room}}\right) \quad (21)$$

$\dot{m}_{fuel,warm-up}$ $is\ the\ rate\ of\ fuel\ during\ warm-up\ (kg/s).$
$\dot{m}_{fuel,ss,max}$ $is\ the\ maximum\ fuel\ flow\ under\ steady-state\ conditions\ (kg/s).$
$k_f$ $is\ an\ empirical\ coefficient\ (-).$
$T_{eng,nom}$ $is\ the\ nominal\ engine\ temperature\ (°C).$

The power produced during warm-up is also correlated to the engine nominal temperature and is assumed to be as follows:

$$\boldsymbol{P_{net,warm-up} = P_{max} k_p \left(\frac{T_{eng} - T_{room}}{T_{eng,nom} - T_{room}}\right)} \quad (22)$$

$P_{net,warm-up}$ *is the rate of power generation during warm – up* $(W)$.
$k_p$ *is an empirical coefficient describing the sensitivity of electrical output to the engine temperature.*

The engine transitions to normal operation whenever

I. The engine temperature exceeds the nominal value ($T_{eng} > T_{eng,nom}$).
II. The net power produced exceeds that requested ($P_{net,warm-up} > P_{demand}$).

### b) Internal combustion engines

The start-up characteristics of internal combustion engines may include a static time delay between activation of the unit and power generation. The power generated by these devices is determined as:

$$\boldsymbol{P_{net,warm-up} = \begin{cases} 0 & if\ (t - t_0) < t_{warm-up} \\ P_{demand} & if\ (t - t_0) \geq t_{warm-up} \end{cases}} \quad (23)$$

$t$ *is the current time* $(s)$.
$t_0$ *is the time at which the engine was started* $(s)$.
$t_{warm-up}$ *is the static delay between activation and power generation* $(s)$.

The fuel flow is determined by solving equation ( 2 ), ( 3 ) and ( 4 ) and setting the steady-state power generation equal to the demand ($P_{net,ss} = P_{demand}$).

Once the static delay has passed, the warm-up period is complete and the unit switches to normal operation.

### Cool-down mode

During cool-down the engine is assumed to consume no fuel and generate no heat:

$$\boldsymbol{P_{net} = P_{net,cool-down}}$$
$$\boldsymbol{Q_{gen,ss} = 0}$$
$$\boldsymbol{\dot{m}_{fuel} = 0}$$

$P_{net,cool-down}$ *is the power used by the unit's control systems while in standby operation.*

The duration of the cool-down period, $t_{cool-down}$ (s) has to be defined.

## Switching between modes

The model must detect which operating mode the cogeneration unit is currently in and switch between modes depending on the current system state, control signals and plant boundary conditions. The model must switch in the order prescribed in **Error! Reference source not found.**.

| Current mode | Future mode |
|---|---|
| Standby | Warm-up |
| Warm-up | Normal |
| Normal operation | Cool-down |
| Cool-down | Standby |

Both the standby and the normal mode operate for indefinite periods of time until the unit is either activated or deactivated. The warm-up mode persists until the engine temperature exceeds its nominal value and the cool-down mode until the specified period lapses. There are two different configurations for the cool-down mode:

- The mandatory cool-down configuration makes the engine complete the cool-down period before it can be re-activated.
- The optional cool-down period configuration lets the engine reactivate during the cool-down period.

## Summary of equations and empirical correlations

| Equation | Description |
|---|---|
| $\eta_e = f(\dot{m}_{cw}, T_{cw}, P_{net,ss})$ | Steady-state electrical efficiency |
| $\eta_q = f(\dot{m}_{cw}, T_{cw}, P_{net,ss})$ | Steady-state thermal efficiency |
| $g_{gross} = P_{net,ss}/\eta_e$ | Gross heat input |
| $q_{gen,ss} = \eta_q q_{gross}$ | Steady state heat output |
| $\dot{m}_{fuel} = q_{gross}\,/LHV_{fuel}$ | Fuel flow rate |
| $\dot{m}_{fuel}^{t+\Delta t} = \begin{cases} \dot{m}_{fuel,demand}^{t+\Delta t}\ if\ \frac{d\dot{m}_{fuel}}{dt} \leq \left(\frac{d\dot{m}_{fuel}}{dt}\right)_{max} \\ \dot{m}_{fuel}^{t} \pm \left(\frac{d\dot{m}_{fuel}}{dt}\right)_{max}\ if\ \frac{d\dot{m}_{fuel}}{dt} > \left(\frac{d\dot{m}_{fuel}}{dt}\right)_{max} \end{cases}$ | Dynamic fuel input |
| $\dot{m}_{air} = f(\dot{m}_{fuel})$ | Combustion air flow |
| $P_{net}^{t+\Delta t} = \begin{cases} P_{net,ss}^{t+\Delta t}\ if\ \frac{dP_{net}}{dt} \leq \left(\frac{dP_{net}}{dt}\right)_{max} \\ P_{net}^{t} \pm \left(\frac{dP_{net}}{dt}\right)_{max}\ if\ \frac{dP_{net}}{dt} > \left(\frac{dP_{net}}{dt}\right)_{max} \end{cases}$ | Dynamic power output |
| $[MC]_{eng}\frac{dT_{eng}}{dt} = q_{gen,ss} + UA_{HX}(T_{cw,out} - T_{eng}) + UA_{loss}(T_{room} - T_{eng})$ <br> $[MC]_{cw}\frac{T_{cw,out}}{dt} = [\dot{m}c_p]_{cw}(T_{cw,in} - T_{cw,out}) + UA_{HX}(T_{eng} - T_{cw,out})$ | Dynamic heat recovery |

# Appendix B
# Calibration parameters

## Stirling Engine

| Model parameter | | Value | Units |
|---|---|---|---|
| Operating bounds | $P_{max}$ | 698 | W |
| | $P_{min}$ | 698 | W |
| Maximum outlet temperature | $T_{cw,out,max}$ | 85 | °C |
| Maximum rate of change in fuel flow | $\left(\frac{d\dot{m}_{fuel}}{dt}\right)_{max}$ | ∞ | kg/s |
| Maximum rate of change in power | $\left(\frac{dP_{net}}{dt}\right)_{max}$ | ∞ | W/s |
| Thermal model characteristics | $[MC]_{eng}$ | 18,5 x $10^3$ | J/K |
| | $[MC]_{HX}$ | 28,1 x $10^3$ | J/K |
| | $UA_{HX}$ | 31,8 | W/K |
| | $UA_{loss}$ | 4,64 | W/K |
| Standby mode power use | $P_{net,standby}$ | -12,5 | W |
| Warm-up characteristics | $T_{eng,nom}$ | 257,3 | °C |
| | $k_f$ | 0,04 | - |
| | $k_p$ | 1,0 | - |
| | $r_{fuel,warm-up}$ | ∞ | kg/s |
| Cool-down characteristics | $P_{net,cool-down}$ | -57,5 | W |
| | $t_{cool-down}$ | 1500 | s |
| | Cool-down mode | MC/OC | - |
| Electrical efficiency coefficients | $a_0$ | 0,0929 | - |
| | $a_1 – a_{26}$ | 0 | - |
| Thermal efficiency coefficients | $b_0$ | 0,970 | - |
| | $b_1 – b_{26}$ | 0 | - |
| Cooling water mass flow coefficients | $c_0 – c_8$ | 0 | - |
| Combustion air coefficients | $d_0 – d_2$ | 0 | - |

# Internal Combustion Engine

| Model parameter | | Value | Units |
|---|---|---|---|
| Oprating bounds | $P_{max}$ | 5596,2 | W |
| | $P_{min}$ | 0 | W |
| Maximum outlet temperature | $T_{cw,out,max}$ | $\infty$ | |
| Maximum rate of change in fuel flow | $\left(\frac{d\dot{m}_{fuel}}{dt}\right)_{max}$ | $5{,}40\times10^{-05}$ | $kg/s^2$ |
| Maximum rate of change in power | $\left(\frac{dP_{net}}{dt}\right)_{max}$ | 760 | W/s |
| Thermal model characteristics | $[MC]_{eng}$ | 82845 | J/K |
| | $[MC]_{HX}$ | 7955 | J/K |
| | $UA_{HX}$ | 3493 | W/K |
| | $UA_{loss}$ | 58 | W/K |
| Standby mode power use | $P_{net,standby}$ | -90 | W |
| Warm-up characteristics | $t_{warm\text{-}up}$ | 129 | s |
| Cool-down characteristics | Cool-down mode | MC/OC | |
| | $P_{net,cool\text{-}down}$ | 0 | W |
| | $t_{cool\text{-}down}$ | 331 | s |
| Electrical efficiency coefficients | $a_0$ | 0,0361 | - |
| | $a_1$ | $-4{,}3890\times10^{-09}$ | - |
| | $a_2$ | $6{,}6907\times10^{-05}$ | - |
| | $a_3$-$a_{26}$ | 0 | - |
| Thermal efficiency coefficients | $b_0$ | 0,9368 | - |
| | $b_1$ | $4{,}4399\times10^{-09}$ | - |
| | $b_2$ | $-6{,}9417\times10^{-05}$ | - |
| | $b_3$-$b_{26}$ | 0 | - |

# Appendix C
# Model parameters

| | | |
|---|---|---|
| **Storage tank parameters** | | |
| Number of nodes | 12 | - |
| Insulation thickness | 15 | cm |
| Conductivety due to walls | 1.44 | W/°C |
| Tank fluid thermal consuctivity | 1.44 | W/°C |
| Boiling point of tank fluid | 98 | °C |
| Heat capacity of tank | 4184 | J/kg °C |
| Density of tank fluid | 1000 | kg/m$^3$ |
| Surrounding temperature | 20 | °C |
| Mass flow cooling water CHP | 200 | kg/h |
| Mass flow SH | 360 | kg/h |
| Dead band level control | 55-65 | °C |
| Initial temperature | 40 | °C |

| | | |
|---|---|---|
| **Building parameters** | | |
| Total area | 178 | m$^2$ |
| Area roof | 100 | m$^2$ |
| Afrea doors | 2,4 | m$^2$ |
| Area floor | 89,7 | m$^2$ |
| Total air volume | 425 | m$^3$ |

| U-values | PH | LEB | |
|---|---|---|---|
| Opaque | 0,15 | 0,18 | W/m$^2$ °C |
| Glazed | 0,72 | 1,2 | W/m$^2$ °C |
| Door | 0,72 | 1,2 | W/m$^2$ °C |
| Floor | 0,11 | 0,15 | W/m$^2$ °C |
| Roof | 0,12 | 0,13 | W/m$^2$ °C |
| Thermal bridges | 0,03 | 0,04 | W/m$^2$ °C |
| Infiltration rate | 0,6 | 1 | W/m$^2$ °C |
| SFP factor ventilation | 1,5 | 2 | W/m$^2$ °C |
| Heat recovery effectiveness | 0,85 | 0,7 | W/m$^2$ °C |

| | | |
|---|---|---|
| **Fuel parameters** | | |
| Fuel source | Natural gas | |
| Lower heating value | 37,38 | MJ/m$^3$ |
| Density | 0,7 | J/m$^3$ |
| Heat capacity | 4184 | J7kg °C |

# Appendix D
# Outputs from time step decision

| Time step | Net power [kWh] | Gross heat [kWh] | Generated heat [kWh] | Skin losses [kWh] | Heat delivered to tank [kWh] | Loss env [kWh] | Heat delivered DHW [kWh] | Heat delivered SH [kWh] | Difference in energy balance tank [kWh] | Difference in energy balance tank [%] | Difference in energy balance CHP [kWh] | Difference in energy balance CHP [%] | Total difference [%] |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 721,3 | 10372,0 | 10061,0 | 1855,9 | 8220,0 | 289,1 | 3655,5 | 4115,8 | 159,6 | 1,9 | 14,9 | 0,15 | 1,04 |
| 6 | 636,4 | 9012,4 | 8742,0 | 1523,5 | 7219,6 | 248,3 | 2849,7 | 4035,6 | 86,0 | 1,2 | 1,1 | 0,01 | 0,60 |
| 12 | 674,6 | 9310,7 | 9031,4 | 1561,4 | 7241,2 | 246,2 | 2929,5 | 3980,6 | 84,9 | 1,2 | 228,8 | 2,53 | 1,85 |
| 15 | 699,7 | 9523,3 | 9237,6 | 1604,0 | 7246,6 | 249,6 | 2948,2 | 3928,0 | 120,8 | 1,7 | 387,0 | 4,19 | 2,93 |