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Thesis for the degree of Philosophiae Doctor

Trondheim, May 2016

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PREFACE

The doctoral work presented in this thesis was carried out in the period from September 2012 to December 2015 at the Norwegian University of Science and Technology (NTNU) in Trondheim, Norway. The work was performed under the supervision of Assoc. Prof. Natasa Nord and Prof. Ivar Ståle Ertesvåg.

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Finally, special thanks go to my sister, Olga, and her husband, Huber, and to my parents, Viktor and Iryna, and to my girlfriend Valentyna for providing support, love and understanding, giving me the extra motivation for the continuation of my studies and the completion of my PhD project.

ABSTRACT

The successful development of a district heating (DH) system requires deep understanding of operation issues. This includes the integration of new energy conversion technologies, control strategies, and economic issues. Different energy sources can be utilized as a primary energy input in the DH systems. Nowadays, the focus is on reduction of the use of fossil fuels and a shift toward renewable alternatives. New developments in the building sector emphasize the application of new design forms and materials, trying to reach the desired lower certified level of energy use. This is corroborated by European Directive 2010/31/EU, stating that, by the end of 2020, all new buildings should be nearly Zero-Energy Buildings (nZEB). The directive pushes society toward the rational use of energy in the building sector. In its deeper analysis, it can be concluded that the DH companies experience the reduction in heat demands. Furthermore, much discussion has taken place regarding the lowering of temperature levels in the DH network, allowing better energy utilization and the application of low temperature excess heat. In this context, DH systems and energy units are becoming more complex and sophisticated; therefore, the need for profound knowledge of DH operation arises.

This thesis discussed different issues associated with the operation of energy production units integrated to DH systems. Therefore, the studies presented in this thesis shed light on operation of DH systems under the three main points. The first concentrates on customers’ impact on DH operation. Hence, the reduction in heat demand, different temperature levels, and available control strategies were analyzed. Next, debates were held about the investment decisions that DH companies face when there is a need to extend or develop energy production units. This included the analysis of units’ sizes, heat load fluctuations, fuel price volatility, mutual effects, and technical limitations. The third research point demonstrated how DH operation could question existing legislation guidelines.

In this thesis, Aspen HYSYS process simulation software was employed for the simulation of energy units. Data post-processing was carried out by MATLAB. Sensitivity analyses of the performed studies were performed under the annual heat and electricity loads obtained from the energy monitoring system of the university campus.

ABSTRACT

The results found that effective plant operation was highly dependent on heat load profile. The operation of a Combined Cycle Power Plant (CCPP) connected to low energy building stock was rather difficult. This means that the CCPP is suitable for high-density heat areas, while it has poor energy performance indicators in low heat density areas.

Further, the analysis of possible solutions for supplying the DH system with several energy supply technologies found that proper evaluation of all the risks associated with the choice of installation and investment could lead to significant savings in a long-term operation of a DH system. This is highly relevant due to changes in heat load profiles, legislation amendments, and improvements in energy saving measures.

The existing method for heat supply optimization, which is based on the methodology of finding the optimal generation mix in some target year, is found to be a simple way to deal with the costs and operation issues. A number of additional important factors affecting plant operation are missing.

Analysis of the allocation factors found that the allocation of fuel, emissions, and operation expenses in Combined Heat and Power (CHP) plants, performed according to standard EN15603 was sensitive when annual operation was considered. Therefore, the decision regarding allocation methods should be carefully analyzed before implementation in the standards, pricing models, and different policies. Mistaken allocation could disable benefits from cogeneration technology and distribution systems. The results of the allocation analysis presented in this work could be used by designers of CHP systems and policy makers, as a tool for developing an emission trading system for CHP plants and for the pricing of heat and power.

The literature review of different factors leading to the premature breakup of the distribution network showed that it is very important to be aware of existing degradation mechanisms and prevent them in good time.

Operation of the DH system with the various energy sources, following different control strategies, is a rather complicated process. In addition, legislation amendments put an extra pressure on DH companies. Based on the process simulation and feasibility studies, the presented information fits well within the issues associated with the operation of DH systems. Further, the performed studies provided valuable information, applicable for operation analysis, control strategy development, and growth of DH networks.

SAMMENDRAG

Vellykket utvikling av fjernvarmesystemer krever en dyp forståelse av driften av systemene. Dette omfatter blant annet følgende problemstillinger: integrasjon av nye energiforsyningsteknologier, styringsstrategier og økonomiske spørsmål. Ulike energikilder kan benyttes som primær energikilder i fjernvarmesystemer. I dag er det fokus på reduksjon av bruken av fossile brensel og skifte til fornybare alternativer. Den nye utviklingen i bygningssektoren legger vekt på bruk av nye teknologier og materialer som skal føre til lavere energibruk i bygninger. Dette samsvarer med det europeiske direktivet 2010/31/ EU som krever at ved utgangen av 2020 skal alle nye bygninger være nesten nullenergibygninger, nearly Zero-Energy Buildings (nZEB). Direktivet krever at samfunnet skal bruke energi i bygninger på en rasjonell måte, samt at en signifikant andel skal være fornybar energi. I en videre analyse kan det konkluderes med at fjernvarmesystemer står overfor en reduksjon av varmebehovet i bygninger. Videre er det nå mye diskusjoner om å senke temperaturnivået i fjernvarmesystemer, slik at bedre utnyttelse og anvendelse av lavtemperatur spillvarme blir mulig. De ovennevnte utfordringer gjør fjernvarmesystemer og energiforsyningsanlegget mer kompliserte, og dermed oppstår behovet for bedre kunnskap om fjernvarmedriften.

Denne avhandlingen analyserer ulike problemstillinger knyttet til drift av energiproduksjonsanlegg i fjernvarmesystemer. Tre ulike problemstillinger er analysert i avhandlingen. Den første er relatert til hvordan brukere (kunder) påvirker fjernvarmedriften. Derfor ble reduksjon av varmebehovet, forskjellige temperaturnivåer samt styringsstrategier analysert. Den andre problemstillingen behandlerinvesteringsbeslutningen som fjernvarmeselskaper står overfor når utvidelse eller utvikling av varmeproduksjonsanlegg skal skje. Dette omfatter analyse av følgende parametere: anleggets størrelse, varmebelastningsprofiler, energipriser og tekniske begrensninger. Den siste problemstillingen behandler hvordan driften av fjernvarmesystemene påvirkes av eksisterende regelverk. Hver en av disse forskningsproblemstillingene er publiser i ulike journal- og konferanseartikler og følgelig er det mulig å lese dem separat.

I denne avhandlingen er simuleringsverktøyet Aspen HYSYS benyttet for å simulere energiproduksjonsanlegg. Databehandlingen er utført i MATLAB. Følsomhetsanalyse er utført

for ulike årlige varme- og elektrisitetsbelastninger målt ved energioppfølgingssystem for universitetsområdet.

Resultatene viser at driften av varmeproduksjonsanlegget har vært svært avhengig av varmebelastningsprofiler. Drift av kombi kraft- varmeanlegg, Combined Cycle Power Plant (CCPP), knyttet til lavenergibygninger er lite gunstig. Det betyr at CCPP er egnet for områder med høy varmetetthet, fordi dette alternativet gir ugunstige ytelsesindikatorer i områder med lavt varmebehov.

Analyse av mulige varmeforsyningsteknologier for fjernvarmesystemer har avslørt at en riktig evaluering av alle risikoene forbundet med valget av installasjon og investeringer i varmeforsyningsteknologier kan føre til betydelige besparelser i driften av fjernvarmesystemer i et langsiktig perspektiv. Dette er svært relevant på grunn av endringen av varmelastprofiler, regelverket og forbedringer i energisparende tiltak i bygninger.

Den nåværende metoden for optimalisering av varmeproduksjon er funnet å være for enkel for å ta hensyn til alle kostnadene og driftsutfordringene. En rekke viktige faktorer som påvirker driften av systemene mangler.

Allokeringsfaktorer for brensel, utslipp, og driftskostnader i et kraft-varmeanlegg, Combined Heat and Power (CHP), beregnet på basis av standarden NS EN15603 gav stor variasjon når hensynet til hele den årlige driften er tatt med. Beslutningen om allokeringsfaktorer bør derfor være godt analysert før implementering i standarder, prismodeller og ulike regelverk. Feil beslutning om allokeringsfaktorer kunne ødelegge fordeler av kraft-varmeteknologi og distribusjonssystemet. Resultatene om allokeringsfaktorene kan benyttes ved prosjektering av CHP systemer og ved utvikling av regelverk. I tillegg kan resultatene benyttes som et verktøy for utvikling av systemer for utslippskvotehandel for kraft-varmeanlegg og for prissetting av varme og strøm.

Litteraturstudiet av ulike faktorer som påvirker pålitelighet av distribusjonsnett viser at det er svært viktig å være klar over eksisterende degraderingsmekanismer av rør og at man bør forsøke å hindre de på forhånd.

Drift av fjernvarmesystemer med ulike energiforsyningsteknologier som benytter ulike styringsstrategier er en ganske komplisert prosess. I tillegg, setter endringer i regelverket et ekstra press på fjernvarmeselskaper. På basis av simuleringsresultater og analyser i denne avhandlingen, kan det konkluderes med at den framlagte informasjonen passerer godt innenfor de

problemstillinger som er knyttet til drift av fjernvarmesystemer. Videre gir resultatene verdifull informasjon om både drift og styring, samt forplanlegging og utvidelse av fjernvarmesystemer.

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ABBREVIATIONS

AO	– adjust operator
ASW	– Aspen Simulation Workbook
CCPP	– combined cycle power plant
CHP	– combined heat and power plant
COP	– coefficient of performance
DEPC	– diesel engine power cogeneration plant
DH	– district heating
EU	– European Union
GHG	– greenhouse gas
GTC	– gas turbine cycle
HOB	– heat only boiler
HP	– heat pump
HRSG	– high recovery steam generator
HTHP	– high temperature heat pump
IPST	– intermediate pressure steam turbine
LCA	– life cycle assessment
LCOE	– levelized cost of energy
LHV	– lower heating value
LTDH	– low temperature district heating
MHP	– mechanical heat pump
MeLHA	– method for heat load analysis
nZEB	– nearly zero energy buildings
NVE	– Norwegian Water Resources and Energy Directorate
O&M	– operation and maintenance
PC	– plant combination
PE	– PolyEthylene
PEF	– primary energy factor
PFD	– process flow diagram
RH	– relative humidity
PHR	– power to heat ratio
SAM	– simulation adjust manager
STC	– steam turbine cycle
TES	– thermal energy storage
4GDH	– fourth generation district heating

LIST OF SYMBOLS AND INDEXES

Since the studies presented in different chapters of this thesis are self-contained and not continued in the following chapter, there is some degree of overlap in the use of symbols and indexes. Therefore, symbols for these chapters are listed separately. In addition, clear explanations are provided for each symbol, in the text.

List of symbols in Chapter 3

$F_t\ (-)$ – correction factor

List of symbols in Chapter 4

$\dot{Q}_h^{bhs} (kW)$ – heat rate at the beginning of the heating season

$\dot{Q}_h^d\ (kW)$ – design value of the heat rate at the minimum outdoor temperature

$\dot{Q}_h\ (kW)$ – heat rate

$Q_{h\,max}\ (kW)$ – maximum heat rate

$f\ (-)$ – load factor, average heat load during heating season

$f_0\ (-)$ – load factor at the beginning of a heating season

t_{in}(°C) – design indoor temperature

t_{ex}(°C) – actual outdoor temperature

t_{ex}^{bhs}(°C) – outdoor temperature at the beginning of the heating season

t_{ex}^d(°C) – design outdoor temperature

$\tau\ (h)$ – duration of the heating season

$\tau_{max}\ (h)$ – utilization time

$\tau_i\ (h)$ – the ith value of heating hours

$\lambda\ (-)$ – exponential function

List of symbols in Chapter 5

$k\ (-)$ – radiator-type coefficient

$n\ (h)$ – operation hours of CCPP

$P_{comp,i}\ (GWh)$ – power production in CCPP with outdoor temperature compensation control

$P_{const,i}\ (GWh)$ – power production in CCPP with constant temperature control

T_{ex}^{d} (°C) – design outdoor temperature

T_{ex} (°C) – outdoor temperature

T_{in} (°C) – indoor temperature

T_1 (°C) – supply temperature to DH system

T_2 (°C) – return temperature from DH system

T_{2d} (°C) – DH design temperature in return line

T_3 (°C) – supply temperature to hydronic heating system

T_{3d} (°C) – DH design temperature in supply line

$u_m\ (-)$ – mixing coefficient

List of symbols in Chapter 6

$C(EUR)$ – total annual cost

$E(MW)$ – electrical production

$F(EUR)$ – fuel cost

$I(EUR)$ – investment cost

$LCOE\ (EUR/kWh)$ – levelized cost of energy

$M(EUR)$ – operations and maintenance cost

$P(MW)$ – installed heat power capacity for each plant

$Q(MWh)$ – annual thermal production

$c(EUR/kW)$ – specific total cost per capacity unit

n *(years)* – system's lifetime

r (%) – discount rate

$\tau(hours)$ – operation time

$\tau_{n,m}(hours/year)$ – break-even operation time for two energy units

η (%) – efficiency

f_Q $(-)$ – fraction of cogeneration emissions allocated to heat generation

Subscript/Superscript

CHP – combined heat and power plant

Elb – electric boiler

HOB – heat only boiler

HP – heat pump

fix – fixed O&M cost

$fuel$ – fuel cost

t $(-)$ – year

var – variable O&M cost

List of symbols in Chapter 7

E_{el} (kWh) – electricity from cogeneration plant

$E_{F,i}$ (kWh) – fuel input to cogeneration plant

Ex_E (kWh) – net output of electrical exergy from cogeneration

Ex_Q (kWh) – net output of thermal exergy from cogeneration

$E_{P,in}$ (kWh) – primary energy input

E_{del} (kWh) – electricity energy generated in the cogeneration plant

$\dot{E}_\iota$ (kW) – power rate

ΔE (kWh) – electricity losses in cogeneration plant due to heat extraction

f_Q $(-)$ – fraction of cogeneration emissions allocated to heat generation

f_E $(-)$ – fraction of cogeneration emissions allocated to electricity production generation plant

$f_{P,dh}$ $(-)$ – primary energy factor of the DH system

$f_{P,F,i}$ $(-)$ – primary energy factor of the fuel for cogeneration plant

$f_{P,el}$ $(-)$ – the primary energy factor of replaced electrical energy

$n\ (-)$	– intensity of GHG emissions of production unit
$Q_{del}\ (kWh)$	– the heat energy delivered to the border of the supplied building
$T\ (K)$	– temperature of the medium
$T_0\ (K)$	– mean ambient temperature of the heating period
$T_s\ (K)$	– supply temperature in DH system
$T_r\ (K)$	– return temperature in DH system
$T_{cond}\ (K)$	– condensing temperature in the cogeneration plant
$T_{out}\ (K)$	– temperature of extracted steam in the cogeneration plant
$\Delta T\ (°C)$	– temperature difference between supply and return lines in DH system
$\eta_{alt\_heat}\ (-)$	– heat production efficiency of producing thermal energy via alternative heat generation plant
$\eta_{alt\_elec}\ (-)$	– power production efficiency of producing power energy via alternative power
$\eta_c\ (-)$	– Carnot efficiency
$v_p\ (-)$	– degree of process quality

List of symbols in Chapter 8

$F(t)$	– unreliability function
$R(t)$	– reliability function of an item
$T(-)$	– time to failure
$f(t)$	– failure probability density function
$t(-)$	– time when unit was put in operation
$\lambda(t)$	– failure rate function
$\Delta t(-)$	– time interval

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1. INTRODUCTION

1.1. Motive for this thesis

District heating (DH) is a service that provides heat to customers in order to satisfy their needs in respect of space heating, hot water preparation, supplying heat to ventilation systems, and industrial purposes. The historical development of DH has its origins in the past, with the first DH system based on a geothermal heat source being developed in the 14th century in France [1]. This is the oldest DH system, which is still in operation, and is located in Chaudes-Aigues, a small town in the central district of France. The commercial use of DH services started in the 19th century and, to date, three generations of DH distribution technologies have been successfully developed and are already in operation.

The first generation of DH technology, established in the 1880s in the USA, was labeled “Steam”, since this was the main heat carrier used at that time. This generation is characterized by high operating temperatures, using only a supply line and quite often no return. Nowadays this distribution technology is considered outdated due to high heat losses and for safety reasons. However, several huge DH systems are still in operation (New York, Manhattan, Paris, and in parts of Copenhagen) mainly due to the high population densities and customers’ requirements.

This era of the technology ended in the 1930s, when the second generation of DH distribution technology was established in the USSR. This second generation used pressurized water with temperatures above 100°C to provide heat. The technology was labeled “Soviet DH technology” and was used extensively in different countries. This new solution allowed the utilization of pipes in concrete ducts and the distribution of heat via huge substations. These systems emerged in the 1930s and dominated all new systems until the 1970s.

The third generation of DH distribution, known as “Scandinavian DH technology” was introduced in the 1970s. Water as a carrier remained the same; however, the temperature levels decreased. New solutions for connection, such as compact substations with brazed heat exchangers, prefabricated pre-insulated pipes and generally high quality components, were introduced.

Today, the research society is moving in the direction of fourth generation district heating (4GDH) [2], also named low temperature district heating (LTDH). Different energy sources can be utilized as primary energy in the DH systems when lower distribution temperatures can be provided. LTDH systems employ assembly-oriented components and flexible pipes for heat distribution. The primary goal of all new developments is the reduction in energy use and savings of primary energy. Therefore, experts and researchers all over the world are trying to reduce the use of fossil fuels, decrease the negative environmental impact, and improve security of supply.

DH service is quite flexible and allows different types of energy sources and various energy conversion technologies to be employed. There are five distinct strategic heat generation technologies used in DH systems. They have high level of potential available, much higher than that used in DH systems today. These strategic heat sources can be identified as Combined Heat and Power (CHP), waste incineration, industrial surplus heat, geothermal heat, and renewables such as biomass.

According to [3], fuel and heat supplies to DH systems are dominated by the use of condenser heat from CHP plants, corresponding to 68% of all district heat generated. Renewables constitute 14% of the district heat supply, which is higher than the corresponding fraction of 7% in the total primary energy supply. Hereby, the European DH systems have together succeeded in fulfilling the EU ambition of a 12% renewable share by 2010. The total share of renewables and heat retrieved from other activities amounted to 78% for all heat generated, proving that the European DH systems are, in general, successful in avoiding direct heat from fossil fuels. The share of renewable energy sources varies greatly by country. The highest shares are found in Sweden, Norway, Denmark and Finland (between 30 and 50%). Iceland, with a 97% geothermal supply, stands out as being almost fully renewable [4].

Further, DH has a huge potential to limit the warming of the planet by the reduction of carbon dioxide intensities. At present, DH alone is responsible for avoiding at least 113 million tonnes of CO_2 emissions per year. This corresponds to 2.6% of total European CO_2 emissions [5]. DH technology plays a major role in achieving at least an 80% reduction in total European greenhouse gas (GHG) emissions, a 50% energy efficiency improvement in the European energy system and a 60% share of renewable energy in total European energy use [6]. In addition, DH is a competitive and cost-effective technology. Although initial investment costs in the systems are high, taking the lifetime costs and energy system benefits into account, very good value for money is achieved [5].

1. INTRODUCITON

Over recent years, European countries have shown both growth and decline in district heat deliveries [3]. High growth rates in Portugal (20%), the Netherlands (16%), Belgium (8%), and Finland (6%) can be explained by more industrial heat deliveries from CHP plants. High growth rates for ordinary DH systems can be found in Italy (8%), Norway (7%), and Austria (6%). Lower growth rates in Sweden and Denmark (2% each) are a consequence of the fact that DH has a high market share in these countries. Germany and France are examples of old, but immature DH countries, with unchanged heat sales over the last 11 years. The highest decreases appeared in Romania (-11%), Bulgaria (-10%), Estonia (-9%), Latvia (-7%), Lithuania (-6%), and Poland (-6%). The main explanation for these high annual decreases is lost deliveries to industrial heat consumers. The decrease to residential and other consumers has been limited. The DH systems in Hungary, Croatia, and Slovenia seem to have managed the transition to a market economy very well, with almost unchanged heat sales over the last 11 years.

As can be seen, DH technology has great potential in Europe and the greater world. The increase in annual heat deliveries, application of new highly efficient technologies, and substitution of fossil fuels by renewable resources makes DH technology competitive in the market and attractive to new customers. This will not only provide a reduction of emissions to the environment but also lead to a high degree of flexibility in the heat supply.

1.2. Aims

With the improvements in energy conversion technologies, the operation of DH systems became more complicated. In addition, climate change, global warming, and legislation amendments are factors affecting the use of heat. Today, DH companies face many challenges. The ability to find the solution to adapting to market changes and technological developments is the key to success for the promotion and development of DH services. For this reason, this thesis focuses on relevant topics associated with the actual operation of DH systems today. The aim of this thesis is to cover four main research points; a short introduction to each of these is presented below.

European Directive 2010/31/EU [7] stated that, by the end of 2020, all new buildings should be nearly Zero-Energy Buildings (nZEB). Since such buildings require a small quantity of energy for heating, they can utilize energy from the return line of the DH system. Further, these new types of buildings can successfully be integrated into the fourth generation of heat distribution technology such as LTDH, which is the new trend in the DH industry. On the other hand, the existing building stock has a service lifetime of around 50 years, indicating that the required supply temperature in the DH system cannot be lowered beneath a certain level. Hereafter, together with new types of buildings and different policies, this could change heat use. The above mentioned situation in the energy used by building will lead to changes in heat load profiles and unavoidably influence the performance indicators of energy conversion units. Since combined heat and power (CHP) plants constitute the largest share in the heat market, their operation under the highlighted conditions requires deep analysis.

Another research point that requires investigation is the decision-making associated with investments in energy production units when a new DH system is under consideration. Nowadays, there are a number of energy conversion technologies available for employment in the DH systems. High operation efficiencies and low level of emissions makes them even more attractive for installation in the DH systems. However, decisions about investment and the assignment of the energy unit require profound knowledge of technology, economics and the range of operation limits. Furthermore, with the fluctuations in users' heat loads, fuel price volatility and high requirements for security of supply, the investment in a single energy production unit capable of fulfilling the full range of the DH load is rather high. Due to changeable heat load patterns from year to year and difficulties in heat load prediction, the operation of a single energy production unit can become inefficient and quite often unprofitable. Good practice states that it is of greater benefit to employ the energy unit able to cope with part of the DH load. In such circumstances, the unit's operation efficiency and heat load factor increases. However, the DH system's flexibility and security of supply decreases. As can be seen, the question regarding identification of a set of energy production units is still on the agenda. How to identify the best plant combination, which of the available units should be operated as a base load plant and which as intermediate and peak load plants: these are the questions that should be answered. In addition, more light should be shed on units' sizes, operation strategies, and the impacts caused by heat load and fuel price variations.

One important issue that still has plenty of questions surrounding it is the allocation of synergy benefits in joint generation processes. Different approaches, describing how to solve this problem, were presented and discussed a number of times, but still there is no common solution available. Different allocation methods are known and are in use. Some methods are based on thermodynamic principles, others, on economic approaches. However, the allocation methods do not consider some important issues of plant operation. The effects of change in plant performance on allocation factors and the further influence on DH generation cost have put in doubt the annual allocation factors for all existing methods and should be investigated. For this reason, it is necessary to perform an uncertainty analysis of allocation methods in order to answer these questions. In addition, it would be of interest to identify the difference in results between existing allocation methods and the conditions of their application.

The last research question included in this thesis covers issues associated with heat distribution. It is well known that DH systems are rarely developed from scratch, and huge DH networks are the result of extension and mergers. The future trend in DH technology is LTDH, which is a promising technology able to decrease the negative environmental impact and lead to sustainability. However, without a reliable distribution system, it is difficult to utilize the ideas of LTDH and stay competitive in the energy market. Hence, it is highly desirable that old DH pipes provide reliable operation and do not influence heat distribution due to unpredicted failures. In this regard, discussions should take place regarding the factors leading to pipe deterioration processes and the solutions for how to prevent these in a timely manner.

1.3. Limitations

This thesis deals with the planning of future DH systems with various energy sources. The presented work was executed with the help of commercially available process simulation software, Aspen HYSYS data; post-processing was performed by MATLAB.

This thesis is based on analyses performed on yearly heat and electricity energy use at the university campus. The studies primarily focused on small-size energy production units and DH systems with a maximum heat rate of 14 MW. Large distribution networks were omitted from this thesis, mainly due to difficulties in obtaining operation heat load data from the DH provider. Further, the studies did not consider real energy units' operation data for the same reason; nevertheless, simplified models of the energy supply technologies were developed based on the thermodynamics models in HYSYS.

Electricity production in the CHP plant and mismatch with DH generation has been excluded from this project. The operation of an heat pump (HP) model was based on constant source temperature and did not consider scenarios with various sources, their temperatures, and fluctuations. Further, the correspondence between simulated data and existing plant operation data was left outside this thesis. The evaluation and design of plant models was based on literature data. The studies did not consider detailed grid operation.

The executed work in this thesis is not continuous and looked at the problem of DH system planning and operation from different angles. This means that the studies can be read separately in three main chapters. A short introduction is provided at the start of each chapter. The chapters in this thesis present the most important findings; the full studies can be found in the corresponding journal publications attached at the end of this document.

1.4. Thesis organization

This thesis constitutes eight main chapters. The content of each chapter is indicated below:

- Chapter 2 provides an introduction to relevant topics associated with the research questions examined in this thesis.
- Chapter 3 introduces a comparison between two process simulation tools available for system modeling and the fulfillment of research aims.
- Chapter 4 introduces heat duration curves that have been used in this thesis; further, the methodology for the creation of an analytical heat duration curve used for comparison is presented.
- Chapter 5 discusses the influence of changes in customers' heat load patterns and operation strategies in a DH system on the operation of a CHP plant with Combined Cycle Power Plant (CCPP) technology.
- Chapter 6 debates the economic issues and technical aspects of energy conversion technologies employed in the DH systems. The information presented in this chapter examines the best set of production units, corresponding dimension sizes and operation strategies. In addition, an economic comparison of the technologies is provided.
- Chapter 7 provides an uncertainty analysis of the allocation factors for heat and electricity in a CCPP. A case study includes different technical methods for the allocation of synergy benefits in CHP plants. In addition, the effects of change in design and off-design parameters of a CHP, due to yearly operation, are introduced.
- Chapter 8 presents a comprehensive literature review on factors resulting in the degradation of a distribution network in a DH system. The section concludes with calculation techniques available for pipe accident prevention and ideas for database creation.
- Chapter 9 offers a summary of the work performed in the PhD project and suggestions for future work.

1.5. Publications

The papers published, accepted or submitted during the PhD project are listed below. The papers are co-authored. The author of the thesis executed the major work in writing these papers such as: creating methodology and models, simulations, interpretation of results and discussion. The contribution of the main supervisor in writing these papers has been discussions, suggestions and critical review.

Journal publications:

- **Tereshchenko T**, Nord N. Uncertainty of the allocation factors of heat and electricity production of combined cycle power plant. *Applied Thermal Engineering*, 2015. 76(0): p. 410-422.
- **Tereshchenko T**, Nord N. Implementation of CCPP for energy supply of future building stock. *Applied Energy*, 2015. 155(0): p. 753-765.
- **Tereshchenko T**, Nord N. Energy planning of district heating for future building stock based on renewable energies and increasing supply flexibility. *Elsevier Energy*. (Accepted).
- **Tereshchenko T**, Nord N. *The 8th International Cold Climate HVAC 2015 Conference, CCHVAC 2015*. Importance of increased knowledge on reliability of district heating pipes. Elsevier Procedia Engineering 2016 (Accepted).

Conference papers:

- **Tereshchenko T**, Nord N. The allocation factors of heat and electricity production of combined cycle power plant. *The 9th Conference on Sustainable Development of Energy, Water and Environment Systems*. September 20 - 27, 2014, Venice-Istanbul.

2. BACKGROUND

This chapter provides an introduction to different issues affecting DH. Firstly, the load problem and different temperature levels in the DH network are presented. Then, an overview of heat energy supply technologies is given. Further, the complexities of the employment of different energy conversion technologies for heat energy generation are clarified. The penultimate section discusses the importance and awareness of reliability issues. Finally, the last section provides a review of the development and current status of existing allocation methods. The more specific issues are explained separately in each section.

2.1. Challenges of future DH demands

The estimation of heat demand is a complex task, especially for large-scale systems involving many heat consumers and consumer types [8]. There are many parameters, which could have an effect on heat load prediction in a DH system. Different authors have implemented algorithms based on yearly observations for heat load prediction. In [9], Werner described a model based on physical theory. Different additive elements, for example wind speed and global radiation, were added to the heat load model. Aronsson in [10] created a model which was based on Werner's work but with improvements. He formed the groups that shared the total heat demand load in a DH system. In [11], Arvaston concluded that, together with the social behavior of customers, outdoor temperature has the greatest effect on heat demand, while different additive elements investigated by the mentioned researchers play a secondary role. Gadd and Werner in [12] mentioned that heat load can be split into social and physical components. Heat loads that depend on temperature difference and level of insulation belong to physical heat load. Distribution heat losses caused by pipe insulation can also be included in this category.

The retrofit of a DH system can affect heat load variation, since physical components such as pipe insulation or distribution pipes play an important role in the overall heat balance of a DH system. As mentioned in [1], typical relative heat losses in ordinary DH systems are 8 – 15% in Western and Northern Europe. The corresponding level is about 12 – 15% in Eastern Europe. Errors and deviations in customer substations and internal heating systems in buildings have a significant impact on the operation and load of heat supply plants. At the same time, our industrialized society always tries to automatize the monitoring processes in different parts of DH systems. One of the future trends in the DH industry is smart systems. The smart DH will allow all the substations to be monitored automatically without great labor input. This can lead to smart load control and consequently to load decrease.

European Directive 2010/31/EU [7] stated that, by the end of 2020, all new buildings should be nZEB and member states should achieve cost-optimal levels by ensuring minimum energy performance requirements for buildings [13]. The change in the heat duration curve for the heat energy supply unit is inevitable.

Currently, the entire building sector cannot consist of nZEB and passive houses. Therefore, the penetration of these buildings into the building stock will show an effect on use patterns in the future. The modernization of existing buildings has decreased the heat losses in European Union (EU) countries, reducing the share of heat use for space heating purposes [14]. This process has already been accomplished in Western Europe, leading to increased effectiveness in heat use for consumers and decreasing heat consumption throughout the year [15]. Werner and Olsson in [16] described the possibility of reducing the heat load variation for peak demand by using buildings connected to the DH system as a means of heat storage. In this study the authors assumed that the maximum time for heat storage discharge for different permitted changes in indoor temperature and different induced changes in the outdoor temperature should be 100 hours. Measurements were performed on different types of buildings (wooden, stone, tower blocks, and old brick buildings). The conclusion was that the estimated time constants were often well above the assumed 100 hours for all types of buildings. Applying this strategy, an immediate increase in heat load during daytime temperature variation can be avoided for peak load energy units. The possibility of optimizing and reducing peak loads in DH systems, applying remote meters and control strategies, was described by Drysdale in [17].

However, it is not only the residential sector that can be connected to the DH system. With the increase in electricity prices, the industrial sector can shift from electrical heating to DH. Difs et al. in [18] investigated the possibility of integrating the industrial sector into existing DH systems. In this study the Method for Heat Load Analysis (MeLHA) was applied to 34 industries, located in various regions of Sweden and from different trade sectors. If industries use only DH services for space heating and hot tap water, then the integration effect will result in an additional load to the base load plant. The conclusion from this study was that industrial processes can be successfully integrated into the DH systems, with benefits to base load plants such as CHP systems.

Different heat load patterns on the customer side, together with climate change and global warming [19, 20], can significantly decrease the profitability of energy supply units in DH systems. As stated in [21], a good practice consists of designing the CHP plant according to the minimum heat demand. However, in the case of DH networks, the minimum heat demand is very low and does not justify the installation of a CHP plant. Then the simple question emerges: How should DH companies react in a situation in which the CHP unit is already installed but the heat demand profile shows significant variation throughout the year?

2.2. Temperature level trends in the DH system

Since the beginning of the DH age, three generations of DH distribution technology have been developed [1]. In the earliest systems, steam was used as a heat carrier. Later on, water became the heat carrier. The materials used in the distribution system propagated different temperature and pressure levels. Nowadays, DH systems are predominantly built according to third generation principles. However, different countries have different requirements for supply and return temperatures in the DH system. In Sweden, for example, for many years the temperatures in hydronic systems were 80°C – 60°C, while in Germany, these values were higher and sometimes reached more than 100°C in the supply line; in Eastern Europe it could even reach 150°C. With the third generation of DH distribution technology, a reduction in distribution heat losses took place. Together with new building codes, these led to a decrease in the supply and return temperatures in the DH network for areas with new types of buildings.

Considering different references [22-24], it can be noticed that different types of buildings have different requirements regarding temperature levels. Authors in [25] showed that, even in non-renovated houses in Denmark, it is enough to supply DH water at a temperature of 67°C. International studies [25-28] showed that there is an over-sizing of around 20 – 30% of DH systems and also of radiator systems, since designers want to be sure that the system provides enough heat. This offers the possibility of further reductions in DH temperatures.

Future grids, with the fourth generation of DH technology, may use low-temperature heat distribution networks with normal distribution temperatures of 50°C – 20°C as an annual average [2].

However, in reality, it is not an easy task to implement ideas regarding low temperature levels in DH systems, when combining them with heat energy units like CHP. Different publications devoted to low temperature DH mostly deal with future buildings and not the existing building stock which, due to the long lifetime of buildings, is expected to constitute the major part of the heat demand for many decades to come [2]. This means that, without prepared infrastructure, it is almost impossible to bring ideas of LTDH to life. Different customers have different heat load characteristics and it is therefore sometimes difficult to satisfy all customers' demands with one temperature level lower than 80°C in the supply line of a DH system. One should also take into account the different types of structures being built during recent decades, as well as buildings at random stages of renovation [24, 25, 29, 30]. At the same time, a DH system should be competitive and cost-effective.

Nevertheless, the situation is different with the return temperature levels in the DH system. For certain types of CHP systems, a high return temperature in the DH network could lead to a decrease in plant efficiency or it could be economically inefficient, depending on power and heat outputs and the configuration of the plant. A higher return temperature results in higher heat losses, less energy stored in thermal storage, if that is used, and lower efficiency of heat generation. These facts make DH less attractive [31]. For these reasons, it was considered that, for DH systems connected to CHP units, a reduction in return water temperature should be implemented, leading to an increase in the temperature difference between supply and return lines. One of the ways to perform this is to implement the "temperature cascading" [32] principle in the return line of the DH system, suggested by researchers in [33]. This idea implies the connection of customers with low heat use to the return pipes, which is relevant for passive houses and nZEB buildings [34, 35]. Applying the temperature cascading principle and new substation schemes, as in [36], it is possible to obtain 20°C in the return line of a DH system and, with future improvements in buildings, insulation properties and distribution systems, even 15°C.

As can be seen, there are a number of obstacles associated with a decrease in temperature levels in the DH network. This complicated and controversial process could lead to variations in the operation parameters and efficiency of CHP plants. Therefore, profound analysis is required.

2.3. Energy supply technologies

Different energy supply plants are available for employment in the DH system. However, it is not an easy task to decide which of these should be installed in a particular situation. Due to technological complexity and limitations in operation, their applicability decreases. Therefore, the following section focuses on the pros and cons of the analyzed energy supply plants.

2.3.1 Biomass combined heat and power plant

CHP technology was first publicized more than a century ago [21] and today is well known and proven to be reliable. According to [37, 38], CHP systems can be classified into topping and bottoming cycles. Further, different exploitation regimes can be indicated; these are: heat-match mode, electricity-match mode, mixed-match mode, and stand-alone mode. In general, the heat-match mode results in the highest fuel utilization rate and perhaps in the best economic performance for cogeneration in the industrial and building sectors [39]. CHP is efficient because it avoids the large amounts of waste heat produced in typical power generation plants [40]. In comparison with other energy conversion technologies used today, CHP has one of the highest indicators. The total efficiency of such plants can reach up to 90% [41]. The attractive property of a CHP plant connected to a DH network is the possibility to massively include renewable sources of energy into energy systems at a reasonable cost [21]. Biomass CHP plants are often seen as an efficient way to reduce GHG emissions due to their very low CO_2 emissions levels [42, 43]. Further, today's CHP plants have quite high conversion efficiencies, leading to better utilization of primary energy.

One of the known CHP modifications employs combined cycle technology and is named the Combined Cycle Power Plant (CCPP). Such plants reach a higher average fuel utilization of about 80%. In addition, this technology allows primary energy savings of between 9% and 20% to be achieved, in comparison with the separate generation of power and heat [44]. Nowadays, more attention is devoted to the appliance of renewable fuels in energy production plants. For this reason, bioethanol, or ethanol derived from biomass, has been recognized as a potential alternative to fossil fuels [45, 46]. Bioethanol driven CCPP reaches significantly higher reductions in emissions, when compared with carbon-intensive fossil fuel technology [44].

However, there are several drawbacks associated with biomass CHP. Some biomass resources, in particular straw, contain aggressive components such as chlorine. These can lead to slagging and corrosion that reduces the security of supply for DH customers. Further, biomass fuel has a great variety of composition and, therefore, different lower heating values (LHV) can affect the efficiency of CHP plants and their output [47]. These place limitations on plant operation, for example when the peak load should be covered. The slow startup of this technology requires a startup load and extra operation hours. Further, most CHP plants designed for DH purposes are characterized by very low power to heat ratio (PHR) [48], which decreases the total energy efficiency. In addition, biomass-based CHP plants are widely used in regions that have ample fuel wood resources, forestry or agricultural residues. A business plan including the cost of the biomass resource collection and logistics is needed to ensure that CHP or power generation from solid biomass is economically viable [49].

2.3.2 Biomass heat only boiler

Nowadays, the modern heat only boilers (HOBs) are biomass based. Type of fuel determines which equipment should be installed for the best fuel utilization. The main advantage of such systems is their high efficiency, especially when energy recovery technology is applied. If the moisture content of the fuel is above 30 – 35%, such as with forest wood chips, flue gas condensation should be employed. Flue gas condensation can improve the overall maximum efficiency of the plant by up to 30%, depending on fuel type and the temperature of the DH water [50]. Thereby, the thermal efficiency usually exceeds 100%, based on LHV. For plants firing wood chips with 45 – 55% moisture content, the thermal efficiency exceeds 110%. Some plants are equipped with cooling devices for full flue gas condensation. Therefore, thermal efficiencies of more than 120% are reached [51]. A biomass HOB provides the possibility to maximize CO_2 savings and potentially eliminate all emissions from fossil fuel systems.

This technology requires lower total capital costs' investment than other options when it is used as a base load plant. In peak load mode, the biomass HOB can demonstrate lower unit installed costs (MEUR/MW) due to economies of scale. Further, HOB does not require back-up of the conventional plant in a peak load mode, which is not the case for the base load plant. The costs of biomass fuels are typically lower than those of fossil fuels, and such systems can therefore provide significant operational savings. Cheaper fuel translates into lower running costs and, hence, annual savings, which reduces the payback period [52].

The drawback of such systems is the high complexity, which requires highly trained operation staff. Further, such systems are sensitive to biomass fuel, which can vary depending on its composition. Higher combustion temperatures can lead to high temperature corrosion, soot, and wear-out of the equipment. Further, there are several high temperature corrosion mechanisms that can occur in HOBs utilizing chlorine-containing fuels [53]. Biomass heating systems generally have a higher initial capital cost than fossil fuel systems of equivalent rated capacity. In addition, if there is a need to run at low load conditions for extended periods, there are also potentially higher maintenance costs [52].

2.3.3 Vapor compression heat pump

HP systems offer economical alternatives to recovering heat from different sources for use in various industrial, commercial, and residential applications. In general, HP technology reduces the use of oil and gas and decreases air pollution, since it consumes less primary energy than other conventional heating units [54]. A DH system is a promising energy-saving measure for high-density cities, and HP systems play an essential role in such large-scale systems [55, 56]. Further, DH systems with HP technology have demonstrated significant reductions in annual energy bills [57]. Today, the most advanced technical developments in the HP field provide the opportunity to deliver heat at a temperature of 110°C [58-60]. According to [57, 61], large-scale HP applications can be successfully applied in DH systems. These HPs are based on mechanical vapor compression and absorption closed cycle principles.

A general advantage of HP technology is its ability to utilize energy at a low temperature level. In addition, the HP is flexible concerning the use of renewable energy, waste, and surplus heat. Compared with traditional heating technologies, HPs are more complex and have high investments costs. However, this is counterbalanced by considerable savings in operation costs [47]. HP allows high operational flexibility throughout a varying load profile, by maintaining high coefficient-of-performance (COP) values.

The main drawback associated with HP technology is high electricity use. This is particularly relevant when the electricity prices in local conditions are quite high. At the same time, the use of large HPs can be called into question due to the high carbon content in the marginal or incremental electricity generation in most industrialized regions and countries [1]. The investment cost of high temperature HP is typically the same for the different technologies, when only the HP itself is considered [47]. Economically, a simple payback period for industrial HP applications is between two and five years [61]. However, in order to stay cost-efficient, low running costs have to lead to required payback periods of less than two years. This drawback can be solved if the HP is integrated wisely. This requires a low temperature lift between heat source and sink, a simultaneous replacing of heating (sink side) and cooling (source side) equipment and long running periods [62].

2.3.4 Electric boiler

Even though undesirable in new requirements, electric boilers are sometimes necessary as an energy supply to cover extreme operation situations and as a back-up plant. Therefore, some information about their operation, benefits, and drawbacks is provided. Electric boilers for DH are used to some extent in countries where electricity is occasionally available at a low price, for example in Sweden and Norway [1]. Due to its very simple design, the electric boiler is extremely reliable and easy to maintain. With no built-in complex components, which may impede operation and maintenance, the boiler has a quick startup and is easy to control. It requires no fuel feeding systems or stack. However, as it uses electricity as an energy input, the investment costs can be high compared to other energy supply technologies. Further, the operating costs are very dependent on the size of the boiler. Thus, heat production from electric boilers can only compete with other heat production units at low electricity prices [47]. If necessary, an electric boiler can also be operated as a peak load plant, even though this may be problematic from the perspective that in many countries there is a tendency for peak heat demand to coincide with the peak in electric power demand [1].

2.4. Economic evaluation of heat energy generation

The economy of DH companies is highly dependent on heat sales. The rule is quite simple: the more heating energy is consumed by customers, the more profitable are the DH companies. This tendency was clearly explained by the authors in [1]. Today, with the new building codes and standards, much attention is devoted to efficient energy use in buildings and the reduction of heat loss [63, 64].

DH service is quite flexible and allows the employment of different energy conversion technologies for heat energy generation. When a technology is considered for use, many issues should be considered. One scenario is when the energy production plants are already installed and in operation. Then, it is fundamental to find a solution to how the existing plants can be operated with the lowest possible annual costs. On the contrary, when planning a new DH system, the heat demands of the different target areas and the possible future development of these should be analyzed; available heat sources should also be investigated. Finally, it is an important task to determine the optimal generation units' combination, optimum configuration of the DH network, and the optimum water temperature levels [65]. In addition, economics, energy saving, and environmental impact have become more important criteria for system design and operation, more heavily burdening designers [66].

DH production units are chosen based on the scale and variation of heat demand, the local availability, costs of energy sources, and the investment cost of each technology [67]. Hence, for optimal utilization of the renewable energy and for economic reasons, the thermodynamic performance of energy production units is of major interest [68]. If the simulation approach has a significant influence on operation results, the economically optimal investment may not be derived; thus, the cost of utility for society and the revenue for the investor will be influenced [69]. This means that the decision to employ different technologies has to be based on proper evaluation with the aid of relevant simulation software. In turn, this must include the variability of the system parameters, aiming to find the best performance obtainable from the match between production plants and users [70].

In liberalized energy markets, the installed utility technologies are optimized in an effort to reduce the total production cost for each individual hour of production [68] and to find the cheapest unit commitment and load dispatch satisfying the given heat, power and reserve demands using the given units [71]. This makes economy of production, together with the technical aspects of technology, the main parameters that should be investigated before the final verdict is handed down.

When the combination of energy supply plants is under consideration, capital investment, operation and maintenance (O&M) costs should be carefully examined for each production unit. The main idea here is that different fuels can be utilized, depending on the available primary energy. In addition, electricity rates should be considered. According to [72], electricity rates affect the operation of CHP as well as of heat pumps (HP) and electric boilers. At the same time, the plant running costs put extra pressure on economic decisions when annual operation is considered. The appropriate sizing of production plants is vital to achieve good levels of utilization, to ensure suitable performance for chosen systems, and to enable effective integration with existing or new DH systems [52]. Further, it should be noted that, in most cases, plant operation becomes inefficient if the energy production unit operates under a low plant load [72, 73]. The plant load indicates the lower bound of the plant operation region. When the DH load drops below it, the plant efficiency decreases and plant operation becomes unstable. Given the large costs of installation and the tight energy saving constraints to which these plants are subjected, an incorrect predictive analysis can result in investment unsustainability, either in economic or environmental terms [74, 75].

Ultimately, a possible change in heat load profiles should be taken into account. According to [76], it is expected that, in the medium term, heat load patterns can demonstrate fluctuations. The main reasons for that are: improved insulation of buildings, installation of ventilation systems with heat recovery, creation of heat islands due to growth of cities and global warming [19, 20], and legislation amendments. The aforementioned facts facilitate a decrease in customers' heat load profiles. On the other hand, the rise in population [77] and housing comfort levels [78] will contribute to an increase in the load to be heated. Thus, the levelling and size of the future DH demand will influence future DH operation and local DH system development [72].

The existing heat energy planning method considers a plant with the highest investment cost as a base load plant. In turn, this gives lower specific heat cost and higher plant efficiency [1]. This means that economy-of-size takes place and denotes energy plants with lower cost at higher production volumes as the main driving force. However, these arguments are no longer as strong, since more efficient technologies for providing heat are available. Identifying a set of new energy production units is a rather complicated process and, therefore, more insight should be devoted to it.

2.5. Flexibility and availability

DH is an energy service based on moving heat from available heat sources directly to customers for immediate use. In order to stay competitive in the energy market, DH should provide a reliable heat supply to customers throughout the year. In reality, this is not an easy task. Different malfunctions and accidents associated with the operation of DH systems and the distribution of heat carrier lead to a decrease in security of supply. The possibility of losing the heat supply is particularly dangerous during the winter season in countries with extremely low outdoor temperatures.

Accidents in DH networks are inevitable and occur for various reasons: wear and tear, equipment failures, and pipeline breach [79]. Accidents lead to financial and capital losses for repair and restoration of the network. Failures reduce the reliability of the network due to lowering of the pressure or due to interruption of the DH supply, which ultimately leads to dissatisfaction among customers. Sensitive customers such as industrial centers, government buildings, and hospitals are most likely to be affected [80]. One of the serious problems in the DH supply is deterioration of the distribution network due to different reasons. Pipe deterioration can lead to pipe breaks and leaks, which may result in a reduction in the water-carrying capacity of pipes and lead to substantial repair costs [81]. Pipe breaks incur large direct and indirect economic and social costs, such as water and energy loss, repair costs, traffic delays, and factory production loss due to inadequate DH service. Unfortunately, breaks in the pipe network are hard to locate because most parts of the pipes are buried underground and inaccessible [82]. Component failures in flow networks lead to the disappearance of flow capacity and the expected magnitude of the throughput flow may not be guaranteed. As a result, the quality of service received from the network can be seriously affected [83].

The reliability of a district heat supply system is a very important factor. According to ISO 8402, the reliability of a system is defined as the ability of the system to perform a required function, under given environmental and operational conditions and for a stated period of time [84]. Different techniques and concepts for evaluating DH network performance are available today. Reliability as a concept has been used in the context of engineering systems for more than 60 years [79].

Nowadays, in Norway, electricity is the primary by energy source. The introduction of alternative energy carriers in the energy system, such as bio energy and DH, is supported by the government and is expected to increase in the future [85]. Currently, three generations of DH distribution technology are in use. The research society is moving towards a fourth generation of LTDH. However, the development of LTDH is impossible without a reliable distribution system. It is well known that DH systems are rarely developed from scratch, and huge DH networks are a result of extension and mergers. Therefore, it is highly desirable that old DH pipes provide reliable operation and do not influence heat distribution due to unpredicted failures. Piping failures can be prevented through reliability measures and DH network is subject to improvement.

2.6. Allocation of products in joint generation processes

The reduction in CO_2 emissions is a challenge for the coming decade, especially with the implementation of the Kyoto protocol. Besides transport, heating is responsible for a large share of the total GHG emissions [86, 87]. One way to decrease the emissions generated by energy services (heating, hot water, electricity), is to increase the efficiency of the different energy conversion technologies that provide these services, by combining them in a polygeneration energy system. A polygeneration energy system is one that generates more than just one single energy service. In the case of DH for instance, polygeneration systems could save over 60% of the energy resources and emissions compared with conventional solutions [88-91]. The simplest example of such a system is the CHP plant. Today, the benefits and potential of cogeneration technology are well known and proven. The following authors discussed this technology in detail [92-95]. When DH is generated in highly efficient CHP plants, it is a reasonable and well-established measure for increasing energy efficiency and promoting the resource-saving use of primary energy carriers [96].

The EU has recognized the importance of CHP technology in combination with DH systems. The benefits of CHP arise from a higher efficiency, which leads to fuel savings and consequently emission reductions. The improved efficiencies and fuel flexibility of CHP provide significant benefits in terms of security of energy supply systems. Directive 2004/08/EC [97] promotes cogeneration technology. The guidelines from the directive allow the benefits of expanding CHP in district heating systems to be made visible [98]. The EU has set targets to reduce energy use by 20% and CO_2 emissions by at least 20% by 2020. DH can greatly contribute to achieving the global policy objectives. Doubling sales of DH by 2020 will reduce Europe's primary energy supply, its dependency on imports from other countries, and CO_2 emissions [99].

In CHP plants, heat and electricity are generated simultaneously. Consequently, it is difficult to precisely distribute the primary energy input, emissions or operating costs to each of these energy outputs. In order to address this problem, different allocation methods have been developed [96]. The allocation method is the methodology, which can provide information on how to share the benefits and drawbacks from joint generation. Today, the main strategy for CHP plants is to be more environment-friendly and energy efficient. DH technology can provide the possibility of decreasing pollution in combination with CHP plants. Unfortunately, not all CHP plants use renewable energy sources like biofuel or municipal waste for producing heat and power. This is one of the reasons why allocation methods should be used in CHP plants in order to allocate CO_2 emissions. The allocation methods could also indicate the economic share for heat and power.

The CHP plant produces electricity and heat, while the delivery of these two products is performed by different companies. The method for emissions' allocation is needed to ensure that each part is credited with its appropriate share of the emissions from the system. In addition, having a meaningful allocation method allows the sources of CO_2 and other emissions to be better understood and, where appropriate, reduced [100]. The choice of allocation method will have a great effect on energy pricing and CO_2 allocation in CHPs. The most recognizable method of fuel allocation is the power bonus method given in standard EN 15316:2007 [101]. This method is well known and accepted by the European Commission of Life Cycle Assessment (LCA) [102].

Limited work has been carried out on developing methods for allocating CO_2 emissions from cogeneration. One of the first reports on allocation methods comes from Strickland and Nyboer [103, 104]. These researchers discussed several methods which could be used for the allocation of products from CHP plants. Their work was based on methods mentioned previously by Phylipsen et al. [105], with some simplifications. The following authors had performed analysis in their research based on these methods. Graus and Worrell in their study [106] employed different allocation methods to calculate the CO_2-intencities from CHP production. Abusoglu and Kanoglu in [107] performed analysis on a Diesel Engine Power Cogeneration plant (DEPC). They studied the allocation of emissions from a DEPC plant based on six methods. In [108] Aldrich et al. investigated GHG emissions in CHP systems applying exergy method with improvements. Wang and Lior in [109] analyzed fuel allocation in a combined steam-injected gas turbine applying seven methods, three of which were thermoeconomics-based. Holmberg et al. studied allocation of fuel and CO_2 emissions in a CHP plant integrated with a pulp and paper mill [110]. In [111], Rosen reported that the exergy method is the most accurate method for allocating CO_2 emissions from CHP systems. Dittmann et al. in [112] concluded that the Dresden method, which was proposed by Zscherning and Sander [113], is the best because it is based on the laws of thermodynamics. In their research devoted to energy systems, the World Energy Council [114] proposed different allocation schemes in the context of using LCA, but still there is no generally accepted one [115].

Many studies have been devoted to investigating the design conditions of CHP plants. The focus so far has been on describing the thermodynamic principles of combined cycles at design point and practical design considerations. However, it must be realized that the operating conditions change, and the system should be able to operate at conditions far from design point. Off-design theory is about predicting how the system reacts to parameter changes. In design and off-design of the CHP plant, the actual geometry of the components remains constant, but operational parameters can undergo changes. The CHP plant may operate for prolonged times at off-design conditions, depending on power demand, ambient condition, and other considerations. This will have a significant impact on the plant performance and, consequently, ensure the system performs not only at design conditions but also at off-design conditions [116].

As can be seen, in their work, researchers have carried out different analyses on allocation methods and parametric studies of CHP systems. However, proper information was not found on how different operational and off-design parameters of CHP systems can have an effect on the allocation between heat and power production. Therefore, the need grows for analysis and comparison of design and off-design parameters of the CHP plant in combination with the allocation methods.

3. PROCESS MODELING

In this thesis, the modeling procedure was based on thermodynamic principles and performed in Aspen HYSYS [117] process simulation software. Aspen HYSYS simulator offers a comprehensive thermodynamics foundation for the accurate calculation of physical properties, transport properties, and phase behavior for the oil and gas and refining industries [117]. This software is commercial and available for purchase. Different authors [117-120] have performed analyses in this software, and the simulation results were found to be in good agreement with the operating data. In this thesis, the obtained results were post processed in MATLAB [121]. The detailed simulation models are explained separately in each chapter.

Based on Chapters 1 and 2, the main objective of this thesis was to simulate different energy supply sources for the purpose of obtaining reliable plants' operation data that could be coupled with DH load, economic evaluation, and allocation issues. Therefore, the need to find a robust tool to simulate these plants was crucial. Different authors used a variety of software for this purpose; however, there is no common judgement regarding which tool performs better. Some authors created models based on thermodynamic equations; others used more sophisticated software packages such as GT Pro and Steam PRO, provided by Thermoflex [122], EBSILON Professional [123], Modelica [124], IPSEpro [125], SIVAEL [126], PRO/II [127], and Aspen HYSYS [117]. At the moment of writing this thesis, the only two available process simulation tools were PRO/II and Aspen HYSYS. Therefore, before the final decision on employing the simulation software was made, several attempts to construct and analyze an energy plant model were executed. The following text presents a comparative analysis of two process simulation software packages and conclusions about their use.

3.1. Comparison of simulation tools PRO/II and Aspen HYSYS

In general, all the computer programs available for flow-sheeting in process design can be classified into two basic types [128]:

- Full simulation programs, which require powerful computer facilities.
- Simple material balance programs requiring only a relatively small core size.

Both PRO/II and Aspen HYSYS belong to full steady-state simulation programs. They can perform simultaneous heat and transfer material balances and carry out sizing of equipment. Since this thesis focused on operational issues of energy supply sources, which belonged to process simulation, both software packages were found convenient for this objective.

In order to evaluate the mentioned simulation tools, a simplified model of a CHP plant connected to a DH system was developed and tested.

3.1.1 CHP simulation in PRO/II

This software is commercial and available for purchase. PRO/II is quite flexible for the construction of different kinds of systems, mainly for chemical process engineering. It can perform steady-state mass and energy balance calculations for modeling continuous processes [128]. It contains a number of components in pallets, which can be easily adjusted for the simulation of energy conversion systems in a steady state mode. With the help of this pallet, the model of a CHP with an integrated absorption chiller was developed for DH and district cooling purposes.

The standard fluid package was used for simulation of the fluid system. The well-known Peng-Robinson equation of state was employed with thermodynamic properties of water and steam (IAPWS-IF97) [129]. In this model of the CHP plant, natural gas was chosen as a fuel. It was supplied to the combustion chamber with air excess of 5%. For simplification, the methane was treated as natural gas and air was chosen as an oxidizer. The combustion process was assumed to be complete. The gases were treated as ideal. The schematic of the CHP plant with absorption chiller is shown in Fig. 3.1.

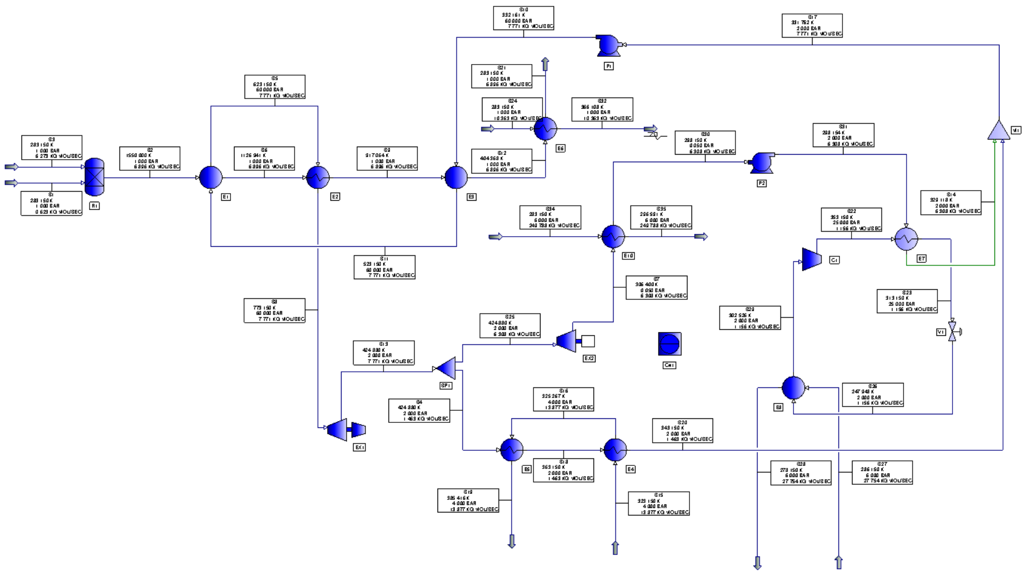

Fig. 3.1. Schematic of CHP with absorption chiller built in PRO/II

PRO/II allows the user to choose between different chemical reactor types for solving the heat and material balances of the components in the reaction. These reactors can be either adiabatic or isothermal [130]. In spite of the fact that the software is quite simple to use, its application revealed uncertainties, which questioned further employment of this software in this thesis.

Firstly, it was a complicated procedure to introduce the correct composition in the reactor. The system indicated that the incorrect number of reactions was specified. This means that it was impossible to calculate the chemical reaction with several components simultaneously. In the case in which it is difficult to specify combustion reaction properly, another option is a link between reaction and LHV of fuel. The LHV of fuels can be found in the PRO/II database and is similar to the values in an average thermodynamic book. When attempts to establish complex combustion reaction failed, it was decided to use a simplified reaction set of methane and oxygen and apply values of LHV. However, with LHV, the obtained combustion temperature was out of realistic values and reached 4000°C, when the real value should be close to 1500°C. The found result was confusing. Since the main idea of this simulation was not the investigation of combustion processes, but the testing of software and identification of CHP performance data, eventually, the flue gas temperature was set manually.

Finally, the DH system was implemented by the means of heat exchangers to the CHP plant. During the simulation, it was noted that heat exchangers exhibited different behavior. The heat exchanger's coldest stream was heated up to the temperature exceeding the hottest stream. This means that PRO/II violated the principles of thermodynamics, which is not possible in reality. Further, PRO/II did not display any warning, and simulation continued. However, the graphical illustration of the heating process found that the temperature cross took place. Nevertheless, the final report showed that there were some inconsistencies with the energy balance in heat exchangers. For the advanced user who is familiar with the PRO/II software, this is of no great consequence. In the case of a beginner, this could drastically affect simulation results, as well as lead to time wasting.

The required model for the analysis was created and tested. However, the application of this process simulation tool found a number of uncertainties that called into question its further employment.

3.1.2 Aspen HYSYS

The second available process simulation software was Aspen HYSYS, which is a process simulation environment designed to serve many process industries, especially oil, gas, refining, and others. With Aspen HYSYS, one can create rigorous steady-state and dynamic state models for plant design, performance monitoring, troubleshooting, operational improvement, business planning, and asset management. Through a completely interactive Aspen HYSYS interface, one can easily manipulate process variables and unit operation topology, as well as fully customize simulation using its customization and extensibility capabilities. The process simulation capabilities of Aspen HYSYS enable engineers to predict the behavior of a process using basic engineering relationships such as mass and energy balance, phase and chemical equilibrium, and reaction kinetics [131].

The simulation process in Aspen HYSYS is similar to that in PRO/II. Firstly, the thermodynamic system should be defined. For the simulation, the same equation of state was used. The process flow diagram (PFD) provides the information on the flowsheet created for the simulation. Building the model is a quite intuitive process. The software is flexible and it is not complicated to extend the model from basic to complex, using sophisticated control systems. The schematic of the CHP model is shown in Fig. 3.2.

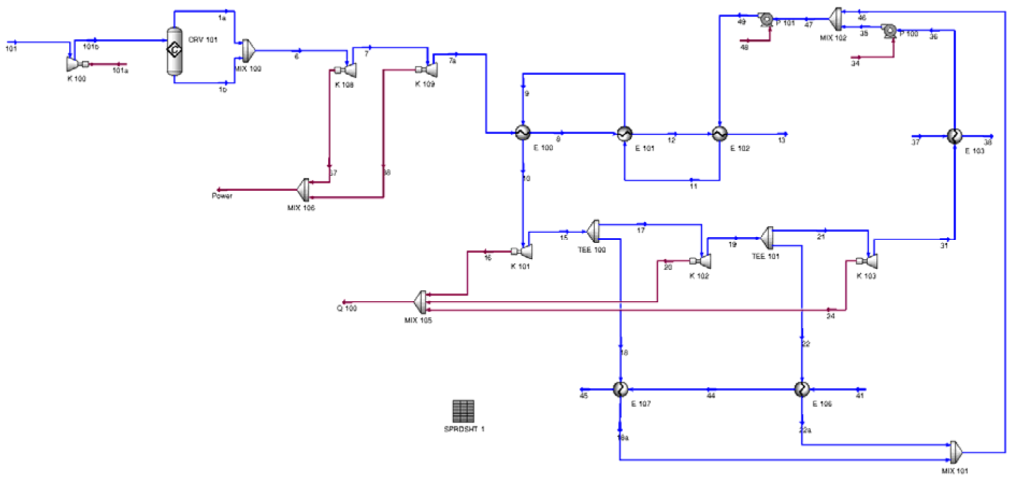

Fig. 3.2. Schematic of CHP built in Aspen HYSYS

The model can be adjusted to automatic calculations of the whole system, changing it by one variable in the process flowsheet. All steams and components can be chosen from the pallet and, using drag-drop technique, set to PFD. It is simple to position and specify process streams; the most challenging thing is to avoid over specification. The system works in such a way that it calculates values automatically for the next stream. If one specifies two streams in the loop cycle with different parameters, the system will provide an error message. For comparison, the PRO/II neglects different input stream data. It calculates values based on flowsheet data in spite of input values added as a supplement.

The most challenging task during modeling of the CHP plant was to avoid a low correction factor (F_t) when the heat exchangers were simulated. The CHP plant employed a number of heat exchangers used for a high recovery steam generator (HRSG) model, district heating connection model and others. The correction factor is used to screen alternative designs for the exchangers before resorting to detailed design calculations. In all the criteria proposed for calculating the number of shells in the heat exchanger, the implicit constraint is that F_t should not fall below a certain value (0.75 or 0.8) [132]. During the modeling, the heat exchangers' F_t values dropped a number of times to the point of 0.2, and the system reacted immediately with an error message. If the mass flow rate of the water carrier passing through one side of the heat exchanger is low, this means that there is not enough heat transfer between the two flows. This is one of the reasons for low F_t . However, the warning does not prevent a solution being obtained; it just informs that the exchanger configuration might not be the optimal one for the heat transfer required. If one changes the shell type of heat exchanger to another configuration, then the warning will disappear. Nevertheless, this is only relevant where geometry is a critical concern in flowsheet calculations [133].

One good feature of Aspen HYSYS is that developers of this software created an application, which is integrated into the Microsoft environment. The name of this tool is Aspen Simulation Workbook (ASW). It allows system analysis to be performed for many different scenarios. This tool is integrated into Microsoft Excel spreadsheet. Different input and output variables could be simulated and, further, the results could be stored in a tabular form. However, in order to use this application properly, there should be a stable workable model in Aspen HYSYS. With the help of this tool, a significant amount of time could be saved in comparison with manual simulation, while performing parametric studies. ASW was very important, since, with its help, a large amount of data was simulated and the impact of various input parameters on CHP plant operation was investigated.

In order to work in ASW, it is better to use controllers in the model and automatize the simulation process. This can be implemented by applying the Adjust operator (AO) that is included in the component pallet of Aspen HYSYS. The AO performs variation of the value of one stream variable to meet a required value or specification in another stream or operation. In a flowsheet, a certain combination of specifications may be required that cannot be solved directly. These types of problems must be solved using trial-and-error techniques. To quickly solve flowsheet problems that fall into this category, the AO can be used to automatically conduct the trial-and-error iterations for you. In addition, the AO is extremely flexible. It can be used to solve for the desired value of just a single dependent variable, or multiple AOs can be installed to solve for the desired values of several variables simultaneously [134].

Observation of the AO operation then provided conclusions regarding its applicability for modeling of CHP systems. The process simulation became complicated when there were multiple AOs in the flowsheet that triggered contradictory calculations. This is usually the case when multiple AOs calculations might become looped and when one makes them work simultaneously. This should ensure that results for a single AO would hold true when another AO is making changes to the flowsheet. In simultaneous mode the AOs' calculations will run together to find a more global solution. In such cases the simultaneous solution should be set in the AO's parameter toolbar [133]. In relation to the analyzed CHP plant, the mentioned observation led to a decrease in the total AOs used to operate simultaneously, which made the model more stable.

When several AOs performs simultaneous calculations, it is convenient to use simultaneous adjust manager (SAM), which is integrated in the adjust toolbar. The convergence of calculations could be observed afterwards. By modifying the adjust settings in SAM, the overall system simulation became more stable. However, the change of mass flow rate in the DH system by various step sizes affected the iteration of SAM operation. When the mass flow rate was changed gradually, all the values were calculated immediately. At the same time, the large input intervals induced complexities in the AOs' operation. In the case of the analyzed CHP model, the input value was presented by the DH load. The rapid change of customer demand induced instability in model operation. However, further studies showed that this can be avoided by the employment of control systems.

3.2. Conclusions

The analysis of the two process simulation tools revealed their strong and weak sides. The practical experience of application of these two tools showed that, in comparison with PRO/II, Aspen HYSYS software is more reliable for modeling and simulation of energy supply units and for the purpose of this thesis. Performing a parametric study in Aspen HYSYS seems to be very convenient.

The performed test revealed that Aspen HYSYS has a better interface and numerous supplementary options. Furthermore, continuous software development and the introduction of additional modules available with every new version hold out the hope that all defects will be eliminated in the nearest future.

4. HEAT DURATION CURVES

Even though the studies presented in this thesis are separate from each other, the employed heat duration curves had the same characteristics and were based on measurements at the university campus. In all three studies the heat duration curve under Case 1 was used as a reference, while Case 2 presented the heat duration curve under a higher occupancy level and lower outdoor temperature, and Case 3 showed the situation for future energy use. In order to give some critics for the analyzed curves, the analytical heat duration curve was also introduced. Both analytical and measured duration curves are introduced in this chapter.

4.1. Analytical heat duration curve

The heat duration curves used for estimating energy use in DH are individual for each DH system. The number of heating hours needed to supply customers depends on geographical location, climatic conditions, outdoor temperature when the heating season begins, and building types connected to the DH. Further, the construction of a heat duration curve is the major operational problem for DH companies. It is not fully possible to plan and predict the heat supply to customers and the fuel needed for the CCPP during an operational year [1]. Therefore, an analytical expression can be used for the calculation of the heat duration curve, applying the methodology presented in [135].

Total heat use can be estimated as follows:

$$\dot{Q}_h = f(\tau) \tag{4.1}$$

The heat rate needed at the beginning of the heating season can be evaluated as:

$$\dot{Q}_h^{bhs} = \dot{Q}_h^d \cdot \frac{t_{in-}t_{ex}^{bhs}}{t_{in-}t_{ex}^d} \tag{4.2}$$

where $\dot{Q}_h^{bhs}$ is heat rate needed to satisfy heat demand at the beginning of the heating season, $\dot{Q}_h^d$ is the design value of heat rate for the minimum outdoor temperature, t_{ex}^{bhs} is the outdoor temperature when the heating season begins (it is also called the threshold temperature and this value is in the range of 8 – 15°C [1]), t_{ex}^d, t_{in} are design outdoor temperature and indoor temperature respectively.

4. HEAT DURATION CURVES

Total heat use, measured at the primary side of the consumer substation, can be estimated as:

$$Q_{h\,max} = \int_0^{\tau} \dot{Q}_h \cdot (\tau) d\tau = \dot{Q}_h^d \cdot \tau \cdot f \tag{4.3}$$

where τ is the duration of a heating season, f is a load factor (average relative heat load during the heating season). In Eq. (4.3), the load factor f presents the ratio between utilization time and duration of a heating season.

$$f = \frac{\tau_{max}}{\tau} \tag{4.4}$$

τ_{max} presents the number of hours when the customer could reach yearly heat use in the case of constant maximum heat rate provided.

After approximation of Eq. (4.3) with an exponential function, the following is obtained:

$$\frac{\dot{Q}_h}{\dot{Q}_h^d} = 1 - m_1 \cdot \left(\frac{\tau_i}{\tau}\right)^{\lambda} \tag{4.5}$$

Let us define values of m_1 and λ from the following conditions: $\tau = \tau_i$; $\dot{Q}_h = \dot{Q}_h^{bhs} = \dot{Q}_h^d \cdot f$

$$m_1 = 1 - \frac{\dot{Q}_h^{bhs}}{\dot{Q}_h^d} = 1 - f_0 \tag{4.6}$$

where f_0 is a load factor at the beginning of a heating season. Further, it can be estimated from the next equation:

$$f_0 = \frac{t_{in-}t_{ex}^{bhs}}{t_{in-}t_{ex}^{d}} \tag{4.7}$$

Combining Eq. (4.7) and Eq. (4.5), we get:

$$\frac{\dot{Q}_h}{\dot{Q}_h^d} = 1 - (1 - f_0) \cdot \left(\frac{\tau_i}{\tau}\right)^{\lambda} \tag{4.8}$$

Substituting $\dot{Q}_h$ from Eq. (4.8) to Eq. (4.3), we get the following:

$$\int_0^{\tau} \dot{Q}_h^d \cdot \left[1 - (1 - f_0) \cdot \left(\frac{\tau_i}{\tau}\right)^{\lambda}\right] d\tau = \dot{Q}_h^d \cdot \tau \cdot f \tag{4.9}$$

After integration, both sides of Eq. (4.9) will have the following form:

$$Q_h^d \cdot \left[\tau - (1 - f_0) \cdot \frac{{\tau_i}^{\lambda+1}}{\lambda + 1} \cdot \frac{1}{\tau^{\lambda}}\right] = Q_h^d \cdot \tau \cdot f \tag{4.10}$$

Considering that: $\tau = \tau_i$, it can be rewritten as:

$$Q_h^d \cdot \tau \cdot \left[-(1 - f_0) \cdot \frac{1}{\lambda + 1}\right] = Q_h^d \cdot \tau \cdot f \tag{4.11}$$

In Eq. (4.11), the information inside the brackets can be transformed into:

$$1 - \frac{1 - f_0}{\lambda + 1} = f \tag{4.12}$$

Consequently,

$$\lambda = \frac{f - f_0}{1 - f} \tag{4.13}$$

Finally, the ratio between the current and the design heat rate can be written as:

$$\frac{\dot{Q}_h}{\dot{Q}_h^d} = 1 - (1 - f_0) \cdot \left(\frac{\tau_i}{\tau}\right)^{\frac{f - f_0}{1 - f}} \tag{4.14}$$

This expression is called Rossander's equation. As described in [135], it allows different heat duration curves to be built for different data sets.

4.2. Different scenarios of heat load variation

Three heat demand profiles were considered to illustrate the heat use in the DH system. The analyzed duration curves are depicted in Fig. 4.1.

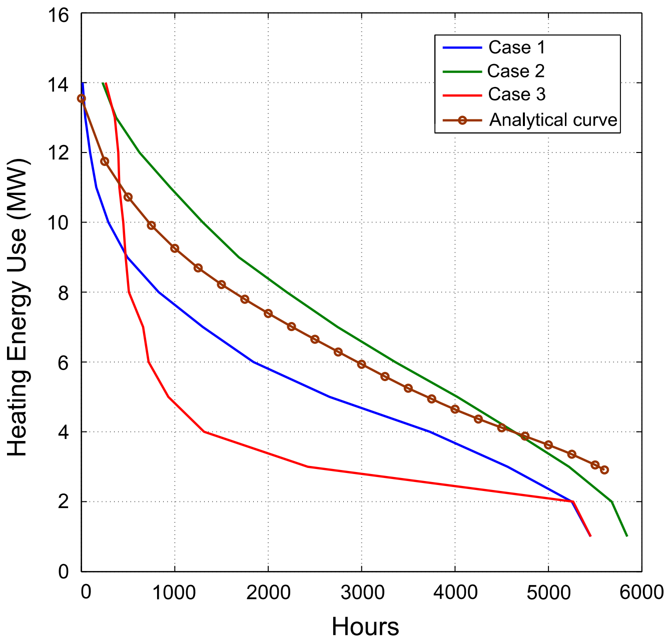

Fig. 4.1. Heat duration curves used for analyses

In this work, it was assumed that the design outdoor temperature was equal to -19°C, while the threshold temperature or beginning of the heating season was assumed when the outdoor temperature was equal to +10°C.

Case 1 presents the heat duration curve during a normal year in the analyzed location and is used as a reference year. Case 2 presents the heat duration curve under a higher occupancy level and lower outdoor temperature. The heat duration curves in Case 1 and Case 2 are the result of measurements, which were carried out at our university campus. Case 3 represents the situation for future building stock, taking into account newly-built passive houses and nZEB with low-heat energy use throughout the year and high peaks during the wintertime. Case 3 is the result of an assumption and is characterized by a decrease in heating energy use of almost 30% in comparison with the reference year. In order to justify selected duration curves, Rossander's analytical heat duration curve was built for predefined conditions (see Fig. 4.1). The heat load characteristics of the analyzed cases are summarized in Table 4.1.

Table 4.1. Heat load characteristics

	Rossander's curve	**Case 1**	**Case 2**	**Case 3**
Heating energy use (GWh)	37.1	24.36	36.43	17.18
Average DH load (MW)	6.63	4.47	6.22	3.15
Heat rate under maximum hours' frequency (MW)	3	4	5	2
Duration of maximum hours' frequency (hours)	756	1072	664	2840
Heat rate under minimum hours' frequency (MW)	13	14	13	11
Duration of minimum hours' frequency (hours)	147	14	146	12
Utilization time (hours)	2650	1740	2459	1227

From Fig. 4.1 and Table 4.1, it can be seen that the analytical duration curve has the highest values of calculated heating energy use and average DH load. At the same time, the analytical curve gave values close to Case 2, which represents the scenario of higher occupancy and lower outdoor temperature. Rossander's equation was developed in the 20th century, when climatic conditions were more severe and the length of the heating season was longer. Hence, the analytical curve shows the maximum possible heat energy use for the analyzed region. Later on, with the development of building codes and new energy policies, the heat energy use in buildings decreased. This is the situation presented by Case 1. At the same time, the heat energy use is expected to decrease in the future for all new building types. This is shown by Case 3, where the value of heating energy use is less than twice that of the analytical duration curve.

As can be noticed from Fig. 4.1 and Table 4.1, there is a tendency towards a reduction in heating energy use in building stock. Hence, the analyzed duration curves under Case1, Case 2 and Case 3 are a good fit with the analytical duration curve.

One additional and very important characteristic used for the description of heat energy use is the frequency diagram. It shows the frequency of heat load hours in the DH system throughout the year. The heat load frequency diagram for analyzed cases of DH load is depicted in Fig. 4.2.

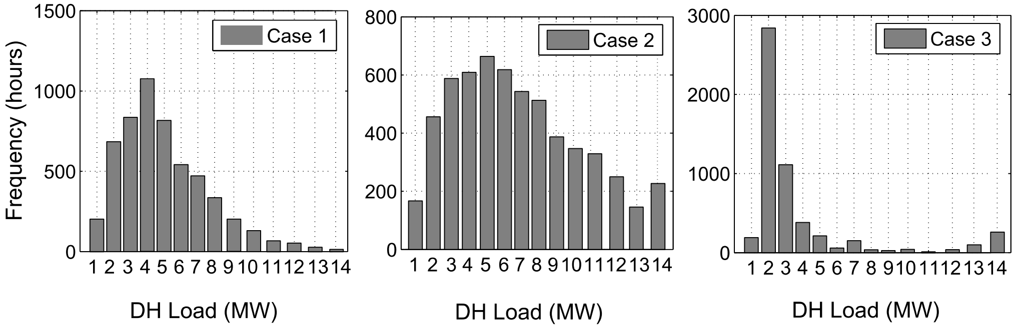

Fig. 4.2. Frequency of occurrence of heat load hours in DH system

In Fig. 4.2, it can be seen that Case 1 had maximum heating hours' frequency at 4 MW of DH load. Further, with the increase in DH load, the heating hours' frequency decreased. In Case 2, the heating hours were evenly distributed throughout the load interval, while Case 3 showed a pattern, which did not follow either Case 1 or Case 2. Case 3 had maximum heating hours at 2 MW of DH load, while the rest of the DH load is sporadically distributed. The information depicted in Fig. 4.2 was very useful as an input for supply plant performance analysis.

The duration curves shown in Fig. 4.1 were applied in the studies devoted to operation of CCPPs and allocation factors. The heat duration curves showed a period of around 6000 hours. However, for the study of combined operation of energy supply units, the heat duration curves with 8760 operation hours were employed.

Table 4.2 shows the summary of the heat load characteristics.

Table 4.2. Heat load characteristics based on prolonged heat duration hours

	Case 1	**Case 2**	**Case 3**
Heating energy use (GWh)	27.48	40.06	21.39
Average DH load (MW)	3.14	4.57	2.44
Heat rate under maximum hours' frequency (MW)	1	1	2
Duration of maximum heat rate (hours)	2465	1887	3547
Heat rate under minimum hours' frequency (MW)	14	16	11
Duration of minimum heat rate (hours)	14	38	12
Utilization time (hours)	1962	2861	1528

5. ENERGY SUPPLY OF FUTURE BUILDING STOCK USING BIO-BASED CCPP

This chapter introduces a study into the development of CCPP for future building areas. The main idea of this study was to investigate the operation of an ethanol-based CCPP under changeable heat demand profiles, together with the possibilities of lowering temperature levels in a DH system.

These days the most advanced plants are fed with liquid bio fuels. The CHP plants with CCPP technology are proved to be the most efficient technology in terms of utilization of primary energy. For this reason the study focused on an investigation of the most energy-efficient generation technology for DH systems. Further, this technology is based on renewable bio-based fuel.

Due to strict energy requirements on new buildings, energy units connected to supply DH systems, such as CCPPs, can demonstrate significant fluctuations in performance indicators. This is especially relevant because of the low energy buildings that are already connected, or will be connected, to the DH in the future. Due to low annual heat demand and high heat demand peaks during extreme outdoor conditions for such buildings, more insight needs to be devoted to this problem of the operation of CCPPs. The main research questions addressed were the following:

- Is the change in performance indicators of the CCPP plant due to different temperature levels and control strategies in the DH system?
- Is CCPP a suitable energy supply technology for low energy building stock, and can it be employed as an energy source for LTDH?

The challenges regarding the heat load prediction and decrease in temperature levels in the supply and return lines of the DH system were described in Chapter 2 and Chapter 4 of this thesis. This section shows the most important findings of this study. More details on the methodology can be found in journal article II, attached at the end of this thesis. However, light is shed on the most relevant information about plant configuration, the calculation method for supply temperature control, and some results in this section.

5.1. Supply temperature control

There are different control options for supply and return temperatures in DH systems [1]. Such methods can be based on a constant supply temperature combined with local flow control, or a constant flow rate in combination with control of the supply temperature, or both. The control of the flow or supply temperature can be based on the feedback (indoor temperature) or the feed forward (outdoor temperature) control approach [136]. In this study, two control options were investigated: constant supply temperature from the energy unit and outdoor temperature compensation.

In the case of the constant temperature control strategy, the supply temperature to the DH is set in the heat energy unit. Supply temperature remains constant during operation, while the control is performed by the adjustment of the mass flow rate of water to the customers. In the case of the outdoor temperature compensated curve, the highest supply temperature is reached at the design outdoor temperature. The return temperature is the result of the control strategy in customer substations and overall mixing of flows from all substations.

Different methods are known for creating outdoor temperature compensated curves in the DH network. In the current study the outdoor temperature compensation was evaluated based on the methodology presented in [137]. In this methodology, the expected supply temperature in the DH network T_1 can be estimated as:

$$T_1 = (1 + u_m) \cdot T_3 - u_m \cdot T_2 \tag{5.1}$$

where T_2 and T_3 are the expected DH return temperature and the expected supply temperature in the hydronic heating system in a building, respectively. u_m is the mixing coefficient. Then, the expected DH return temperature can be estimated as:

$$T_2 = T_3 - (T_{3d} - T_{2d}) \cdot \left(\frac{T_{in} - T_{ex}}{T_{in} - T_{ex}^d} \right) \tag{5.2}$$

where T_{ex}^d, T_{ex}, T_{in} are design outdoor temperature, outdoor temperature, and indoor temperature in the building, respectively. T_{2d} and T_{3d} are DH design temperatures in the return and supply lines, respectively.

The mixing coefficient can be evaluated as:

$$u_m = \frac{T_{3d} - T_1}{T_1 - T_2} \tag{5.3}$$

The expected supply temperature for the hydronic heating system can be estimated as:

$$T_3 = T_{in} + 0.5 \cdot (T_{3d} - T_{2d}) \cdot \frac{T_{in} - T_{ex}}{T_{in} - T_{ex}^d} + 0.5 \cdot (T_{3d} + T_{2d} - 2 \cdot T_{in}) \cdot \left(\frac{T_{in} - T_{ex}}{T_{in} - T_{ex}^d}\right)^{\frac{1}{1+k}} \tag{5.4}$$

where k is a radiator-type coefficient. The k value for the most common type of radiators is 0.25.

Based on this methodology, Fig. 5.1 shows outdoor temperature compensated curves, shaped for a temperature level in the DH network of 100°C – 45°C.

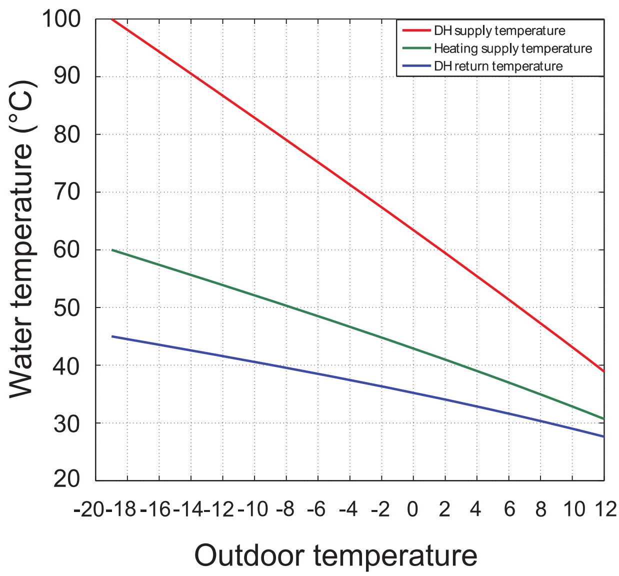

Fig. 5.1. Outdoor temperature compensation curves

5.2. Ethanol driven CCPP under different heat loads and temperature levels

In this study a small-scale DH system was analyzed, employing a CHP system with CCPP technology as an energy source for the DH system. The configuration of the CCPP system was an ethanol-driven gas turbine cycle (GTC), using exhaust heat recovery to drive a bottoming steam cycle (STC), with steam extraction for DH.

Ethanol-based CCPP is a well-known technology, and different authors have performed studies on such systems [138-140]. The schematic layout of the system is presented in Fig. 5.2 and design parameters are summarized in Table 5.1.

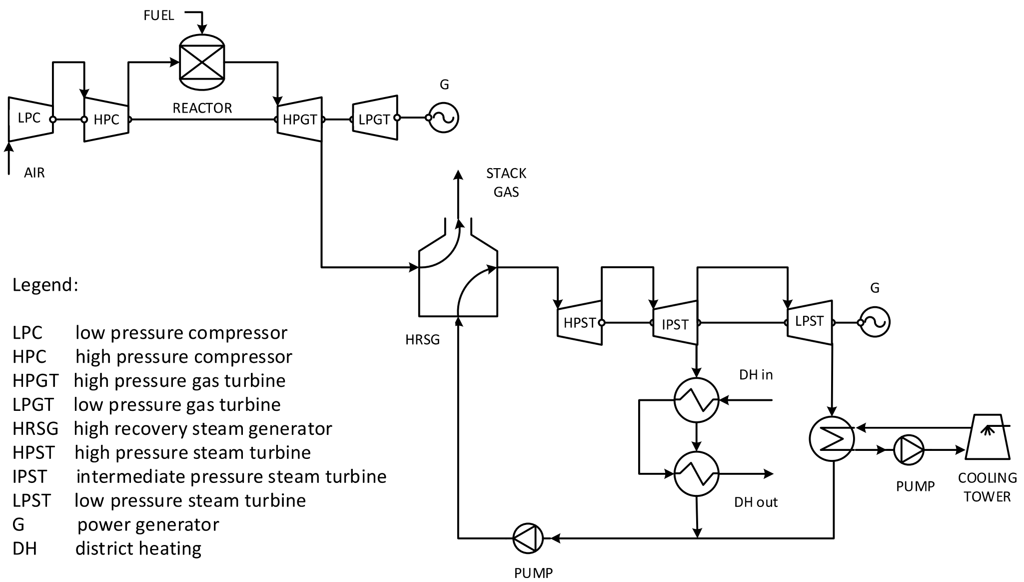

Fig. 5.2. Schematic of the CCPP

5. ENERGY SUPPLY FOR FUTURE BUILDING STOCK

Table 5.1. Design point parameters of the CCPP

Parameter	Value
Ambient pressure	101 kPa
Air relative humidity	60%
Ambient air temperature	+15°C
Pump pressure	100 bar
Steam turbine inlet temperature	+540°C
Condensing pressure	0.05 bar
Air excess in air-fuel mixture	4.0
Fuel temperature	+15°C
Gas turbine adiabatic efficiency	0.9
Steam turbine adiabatic efficiency	0.9
Compressor adiabatic efficiency	0.9
Gas turbine inlet temperature	+1096°C
Supply temperature in DH system	+100°C
Return temperature in DH system	+45°C

Since the aim of this study was to analyze how the CCPP could be implemented for future buildings, a range of different supply and return temperatures was considered. The temperature levels were chosen based on a review of the DH generations. A detailed explanation about the choice of temperature levels is provided in Section 2 of this thesis. Taking into account different studies, conclusions, and the recommendations from these studies regarding temperature levels in DH systems, the temperatures presented in Table 5.2 were studied.

Table 5.2. Analyzed temperature levels in DH network

Explanation	Supply temperature in DH network (°C)	Return temperature in DH network (°C)
2nd generation of distribution technology with medium, low, and ultra-low return temperature	100	45 30 15
3rd generation of distribution technology with medium, low, and ultra-low return temperature	90	45 30 15
3rd generation of distribution technology with medium, low, ultra-low return temperature	80	45 30 15

The temperature levels in Table 5.2 do not fit to the DH generations; however, this is the result of different types of buildings (new and existing) being part of future building stock. In addition, some of them may be connected to the return line of the DH.

5.3. Results of performance analysis of CCPP

The analysis of CCPP operation was performed under the heat loads introduced in Section 4.2 and Table 4.1. The results are presented in three separate sections. The results clarify issues associated with energy conversion, performance, and fuel energy use in the heat production unit under different temperature and heat load levels.

5.3.1 Energy conversion in CCPP under different heat loads

Power production for two temperature control strategies in the DH system are shown in Fig. 5.3. The shortcut “const” shows values for constant temperature control, while “comp” shows values for outdoor temperature compensated control.

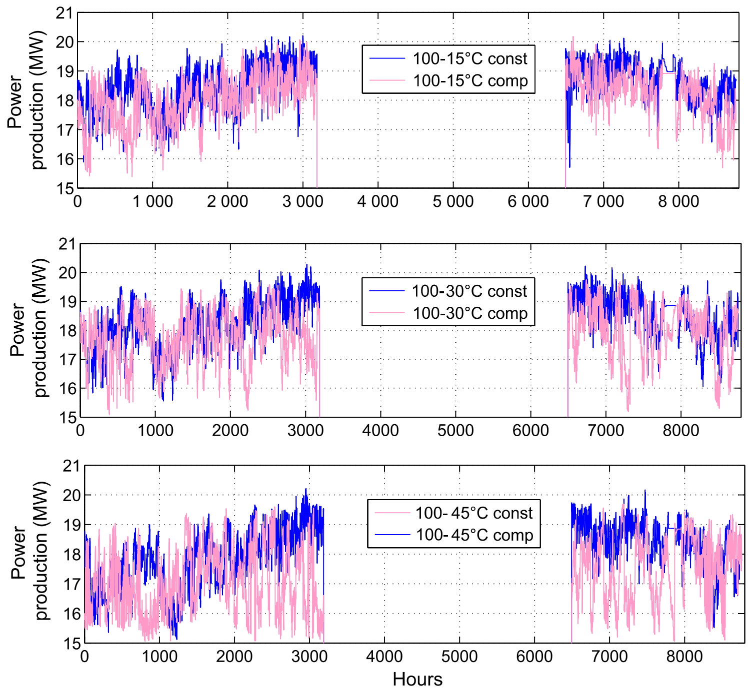

Fig. 5.3. Power production in the CCPP for different control strategies and return temperatures in the DH system

In Fig. 5.3 it can be seen that the difference in the power production of the CCPP, due to changes in control strategies with different supply and return temperatures, was not significant. Therefore, due to the enormous computational time needed for CCPP simulation over the entire year under two control strategies and at different temperature levels, it was decided to focus on only one control option. To identify the difference in the CCPP power production, the relative deviations between obtained results were calculated as:

$$\Delta P = \frac{1}{n} \cdot \sum_{i=1}^{n} \frac{\left(P_{comp,i} - P_{const,i}\right)}{P_{const,i}} \tag{5.5}$$

where $P_{const,i}$ and $P_{comp,i}$ are the values of power production in the CCPP with constant temperature control and outdoor temperature compensation strategies at time step i. n is the number of operating hours of the CCPP.

Fig. 5.4 presents the relative annual deviation between the power production in the CCPP for two different control strategies, with respect to the constant control strategy.

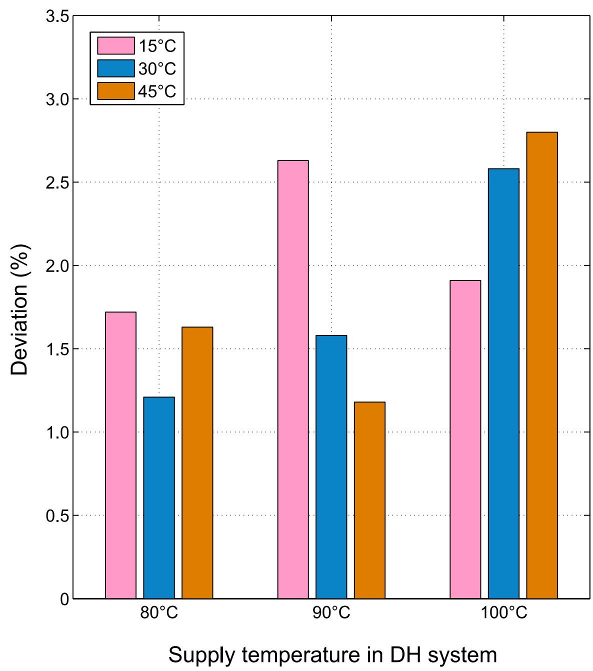

Fig. 5.4. Average deviation between two data sets

The difference between power productions for different temperature levels in DH systems is relatively small, under 3%, as shown in Fig. 5.4. The smallest deviation occurred in the temperature range of 80°C – 30°C, while the largest was in the 100°C – 45°C range.

Due to the relatively small difference in power production in the CCPP between the two control strategies applied in the DH system, see Fig. 5.4, only the constant supply temperature strategy was analyzed in the further analysis. Therefore, all future results are related to the constant supply temperature strategy.

Fig. 5.5 shows heat efficiency in the analyzed CCPP depending on frequency of heat load hours in the DH system. Fig. 5.5 shows minimum, maximum, and median values of heat efficiency for the analyzed scenarios.

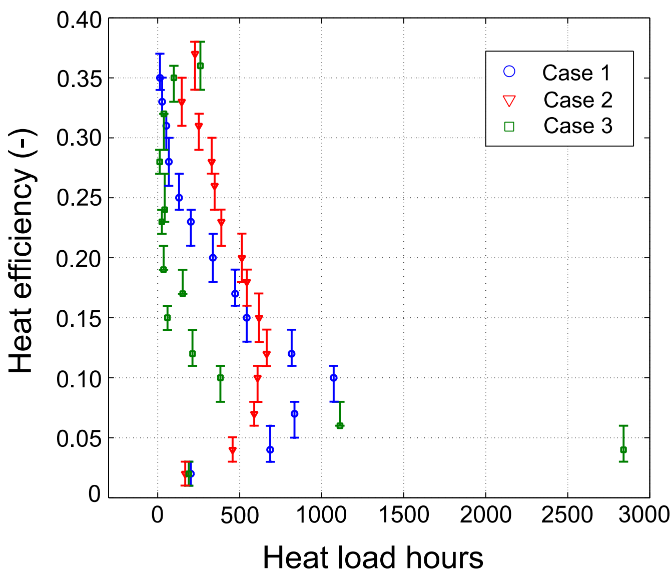

Fig. 5.5. Heat efficiency in the CCPP

Despite the fact that, in Case 3 there is the highest number of hours (2840 hours) with a heat load of 2 MW, see Fig. 4.2, the heat efficiency of the CCPP is quite low. The highest heat efficiency is found for the maximum heat load of 14 MW for all the cases. The higher the DH load, the higher the plant capacity utilization and heat efficiency.

5.3.2 CCPP performance under different heat loads and temperature levels

Fig. 5.6 introduces average system performance characteristics for the analyzed CCPP, such as power efficiency, heat efficiency, and energy efficiency.

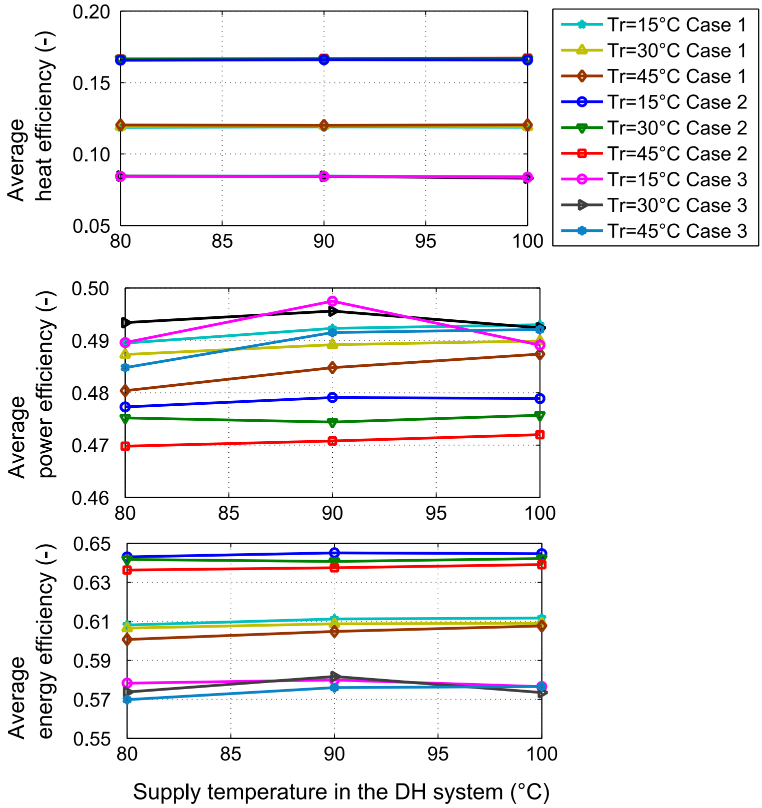

Fig. 5.6. Average heat, power, and energy efficiencies

Fig. 5.6 reveals that the results of average efficiencies in the CCPP varied among cases due to different load distribution. The efficiencies were highly dependent on the DH heat load distribution. Fig. 5.6 shows that uniform distribution of heat load resulted in better plant operation throughout the year. The average heat efficiency for Case 2 is higher than for Cases 1 and 3. However, for different levels of supply and return temperatures, for all the cases the change in power efficiency and energy efficiencies was in the range of 1 – 2%, which is quite small. The maximum average energy efficiency was in the range of 57 – 65% for all the cases. As can be seen, the plant operation under Case 3 showed that it was poorly loaded by the DH system throughout the year. This information should be considered while running the CCPP with the DH system.

The power efficiency for Case 2 was sensitive to temperature difference in the DH system because, with the increase in temperature difference, there is an increase in the power generation of the plant. Further, it was found that the difference in the energy efficiencies was negligible for different DH loads and the analyzed temperature levels. However, in the case of continuous hour-by-hour operation of the CCPP, the deviation in energy efficiency was in the range of 2 – 10% between minimum and maximum values. This can be explained by rapid change in the DH load, which results in an immediate response in fuel input and power production within the CCPP.

Of special interest is Case 3, since it reflects one of the possible scenarios in the future when low energy buildings will constitute a certain part of the building stock. Low heating energy use makes such buildings unattractive for supply by large heat production units.

The analysis showed that the temperature difference had a positive influence on power production within the STC under Case 3. The difference in average values of the power efficiency and energy efficiency between cases was not very large. The deviation between minimum and maximum values of efficiencies varied from 2% to 10%, depending on heat load rate. Meanwhile, the CCPP was sensitive to change in the DH load, especially if a long operation period is considered. The main conclusion could be that it was beneficial to have a high heat load, while running the CCPP.

The values found in this study for heat efficiency, power efficiency and energy efficiency are different in comparison with design conditions. One of the reasons was that design values were given at the maximum DH load and fixed reference point. In reality, it is quite complicated to run a CCPP based on full DH load due to variable heat load characteristics and high seasonal variations. Further, different elevations above sea level, ambient temperature, and air pressure may cause adjustments to plant operation.

5.3.3 Fuel use

Finally, Fig. 5.7 represents the fuel input within the analyzed CCPP.

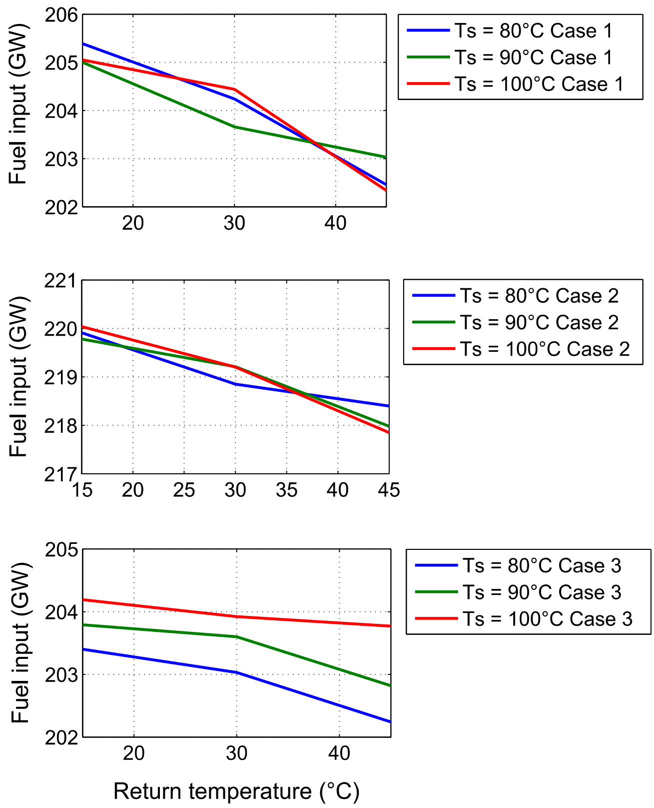

Fig. 5.7. Amount of fuel input in the CCPP

It can be seen that the reduction in return temperature shows a negative tendency in terms of fuel use. Cases 1 and 2 showed a gradual reduction in fuel input when the return temperatures increased. With the increase of temperature difference in the DH system, the fuel use increased; see Fig. 5.7. This happened because more energy input was required to heat up water in the DH system per 1K. However, for Case 3, the fuel energy input did not follow uniform increase with respect to temperature level used. This could be explained by rapid change in the DH load in the CCPP. Further, the load factor given in Eq. (5.6) shows plant capacity utilization in terms of heating energy production.

The load factor is the ratio of average load to the maximum load in the supply system [141].

$$\text{Load factor} = \frac{Average\ load}{Maximum\ load} = \frac{Energy\ consumed\ during\ a\ period}{Maximum\ demand \cdot operation\ time} \tag{5.6}$$

Table 5.3 gives the values of the load factor for the analyzed cases.

Table 5.3. Heat load factor for analyzed cases

	Heat load factor (-)
Case 1	0.32
Case 2	0.45
Case 3	0.23

From Table 5.3 it can be seen that the load factor for Case 3 is the lowest. This indicates that the plant operates sporadically following the heat load during an operational year. The higher the load factor, the cheaper the heat energy for the customer. In reality, it is very difficult to achieve a high load factor due to variable load characteristics from year to year.

The analysis of different temperature levels applied in the DH system indicated that the energy efficiency had negligible variation due to temperature levels in the DH system when running the CCPP. The reason for this is the high power production that takes place in the GTC. The analysis found that heat load distribution plays a crucial role in plant performance operation. Low heat load distribution leads to poor overall plant performance indicators. This provides incentives to run the plant for power production only. For this reason, when there is a need to select the DH supply and return temperatures for higher electricity production, the most effective method is to choose lower DH supply and return temperatures. Nevertheless, if we cannot change both of them, lowering the supply temperature is of greater benefit [142].

Based on this study, it was concluded that it was rather difficult to operate a CCPP connected to low-energy building stock. Such buildings should be supplied from low temperature energy sources specially designed for this purpose. However, when high-grade heat is required, the CCPP can be used to produce additional heating energy. This means that the CCPP is suitable for high-density heat areas, while it operates poorly in low heat density areas. For future building stock, it means that the CCPP could be successfully implemented if the buildings were grouped in one place, rather than spread over a large area.

5.4. Conclusions from CCPP operation

The results showed that the power production in the CCPP was not influenced significantly by the supply temperature control. The change in the power production was between 1.2% and 2.8% due to change in the temperature control. Therefore, the focus in the study was on the constant supply temperature in the DH system.

The analysis of the change in the DH load showed that the average heat efficiency was highest for the uniform load distribution and lowest for very non-uniform load distribution. The average power efficiency was dependent on different temperature levels in the supply and return lines of the DH system. The results showed that the highest power efficiency was obtained for the temperature levels of 100°C – 15°C and the lowest for 80°C – 45°C, for Case 1 and Case 2. This indicated that a large temperature difference between the supply and return lines of the DH system resulted in higher power production in the CCPP. The results found that decrease in supply temperature had a low impact on energy efficiency. However, decreasing supply temperature to the DH system can lead to an increase in the service pipeline's lifetime, which is beneficial for the DH system. Another important conclusion is that the CCPP performance indicators are highly dependent on the heat load distribution in the DH system during the year. When DH load distribution had a uniform pattern throughout the operation year, as in Case 2, this resulted in better plant performance in comparison with Case 3. In the case of non-uniform heat load distribution, as in Case 3, plant performance was poor, indicating that the plant was poorly loaded. The results on load factor confirmed that fact, showing that in Case 2 the best possible heat load pattern for CCPP operation was obtained, while Case 3 presented the worst possible situation. However, in the current CCPP, GT technology was employed, which utilized the benefits of the low DH load by increasing power production. Analysis of all the CCPP performance indicators versus the DH load showed negligible variation for all the temperature levels applied in the DH system. The difference was in the range of 2 – 3% between cases. The change in the overall fuel energy input showed that fuel use increased with increase in temperature difference between the supply and the return lines in the DH system.

The results obtained in this study point out an inevitable decrease in plant profitability while operating the CCPP under low and non-uniform heat demand profiles. This observation provides incentives to shut down the heat supply to DH systems and run CCPP at full load, producing as much electricity as possible. Low energy building stock should be connected to specially designed low-grade temperature sources under a prepared infrastructure. However, CCPP could be used if low energy buildings were located close to each other to increase the heat density. The CCPP could also be used during the peak energy demand. This will have a positive result on plant operation, since the CCPP will operate at its maximum heat load output, increasing its performance indicators.

6. OPTIMAL COMBINATION OF RENEWABLE HEAT PRODUCTION PLANTS IN DH SYSTEMS

This chapter focuses on the technical and economic aspects as prerequisites for employing different energy production units to provide heat to a DH system. The study presented in this part of the thesis sheds light on the situation in which there is a need in construction for a set of energy units for existing or new DH systems. The issue of security of supply is very important for energy providers and also for the customers connected to the DH. Therefore, proper evaluation of technical and economic aspects is seriously required before the decision to make a major investment is accepted in favor of one or the other technology.

Three highly energy-efficient energy conversion technologies, such as bio-based CHP, bio-based HOB, and large mechanical HP, were considered for the analysis. In addition, an electric boiler was employed for extreme operation situations and as a back-up plant. Further, change in heat load profiles and fuel price volatility were analyzed. The main research questions addressed were the following:

- What is the best combination of energy conversion technologies and their heat capacities to cover the DH load?
- What is the difference between new and existing methods for heat supply optimization?
- What will be the economic consequences of change in heat load profiles and fuel price volatility for the chosen production units?

This chapter provides the most important results found during the study. The full description of the methodology and problem formulation can be found in journal article III, attached at the end of this thesis.

6.1. Economic appraisal and economy issues

This section focuses on various economic issues associated with the installation of energy production units. The presented information is based on a literature review. The aim was to identify the available economic data associated with capital investment and O&M cost for each technology. In addition, fuel prices and electricity rates were considered.

Several issues should be considered when making a decision on the installation of an energy production unit. Firstly, the technology should meet customer requirements in providing heat to the DH system. At this point, it can be noted that different customers can use a wide range of temperatures due to their various requirements. Further, heat load patterns should be taken into account. Due to changeable climate characteristics and continuous improvements in building codes and standards, the heat load patterns can show variations from year to year. Finally, the employed energy conversion technology should be environmentally friendly and certainly display positive economy under long-term operation. Therefore, a detailed feasibility study should be carried out considering the installation of a particular system.

Normally, three economic key points should be analyzed before making an investment in a particular technology. These are: capital investment costs, fixed and variable O&M costs, and fuel costs. The capital investment costs include costs for the system, costs required for its installation, land, feasibility studies, rental of different equipment and machines, building costs, and various miscellaneous costs. In the annual operation costs, the following can be included: insurance and property taxes, spare parts of equipment, electricity, O&M labor, contingencies, and other miscellaneous costs. The contingencies can be assumed to equal 10% of the O&M costs [143]. Biomass systems can require more regular maintenance than fossil fuel boilers. For example, a biomass system can require between 0.5 and 1.5 days per month of attendance time [52].

For this study, it was necessary to find cost and performance data for each technology. It was difficult to find data organized in a consequent way. However, a huge literature review was performed. The cost data found for each technology are summarized in tables and presented in Appendix I. These tables show the data organized separately for each technology. The tables provide information about energy efficiency, capital investment cost, variable and fixed O&M cost with respect to the plant's heat output. The cost data is collected for renewable based heat generation technologies. However, the most important information selected for the analysis is illuminated in the further text.

Fig. 6.1 - Fig. 6.8 show analyzed information from Table A1 to Table A4 in Appendix I. The figures shows dependency of investment costs and efficiencies versus heat rate of different heat supply technologies. The idea behind Fig. 6.1 to Fig. 6.8 was to identify whether there are relationships between plant performance data, investment, and heat rate. Analyses were made for bio-based HOB, biomass CHP, electric boiler and HP because these technologies will be analyzed altogether in Section 6.3. The investment cost in Fig. 6.1 – Fig. 6.4 is given in EUR/kWh.

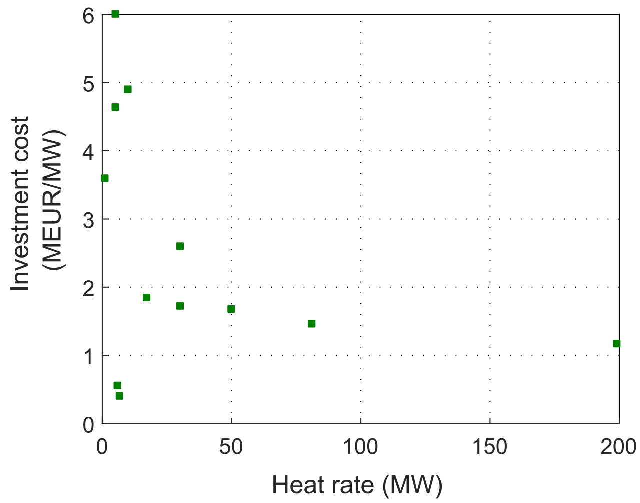

Fig. 6.1. CHP’s investment cost versus heat rate

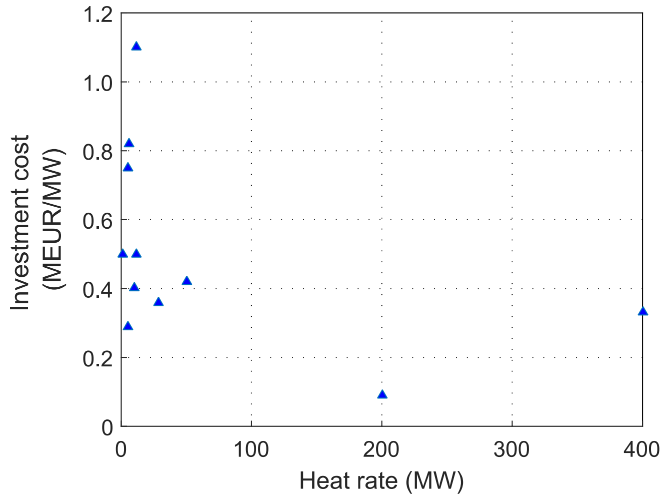

Fig. 6.2. HOB's investment cost versus heat rate

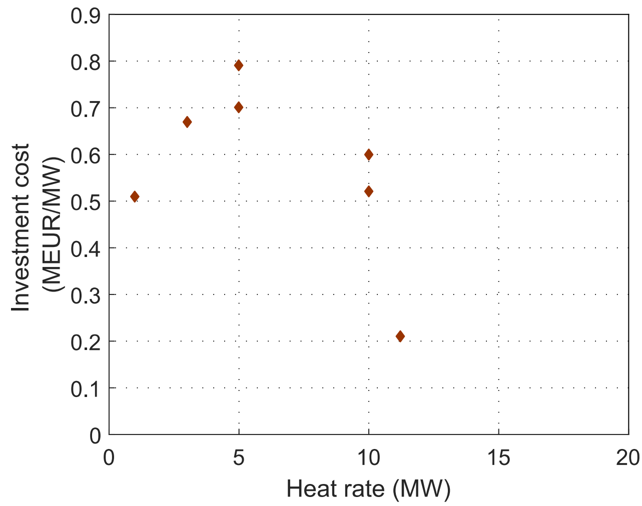

Fig. 6.3. HP's investment cost versus heat rate

From Fig. 6.1 and Fig. 6.2, it can be noticed that most of the references dealt mainly with small-scale CHP and HOB systems with heat capacities in the range of 1 – 50 MW. The HPs' data goes up to 12 MW; see Fig. 6.3.

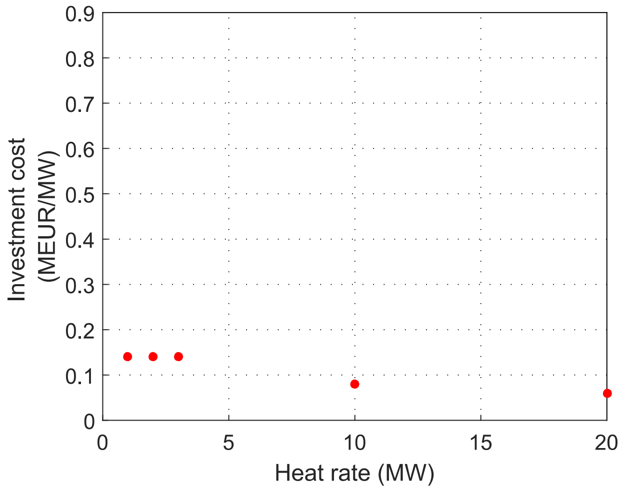

Fig. 6.4. Electric boiler's investment cost versus heat rate

Heat output from the electric boiler ranges from 2 MW to 20 MW. In comparison with CHP, HOB, and HP investment data, the investment cost for the electric boiler per unit of heat output is the lowest.

The analysis of plant efficiencies showed that the best CHPs could reach efficiency up to 110%; see Fig. 6.5. This is because flue gas condensation technology is employed in such plants.

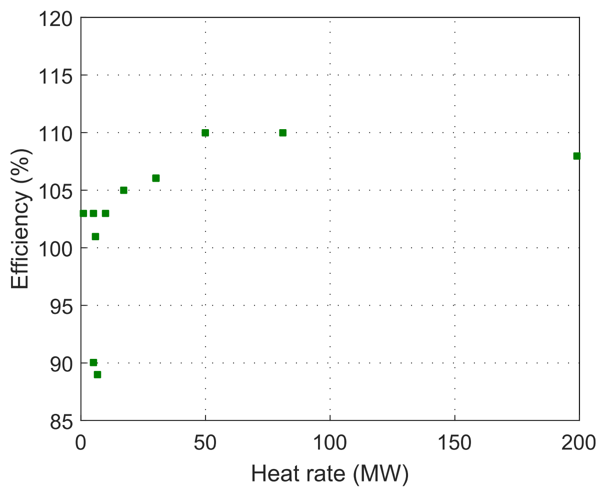

Fig. 6.5. CHP's efficiency versus heat rate

In the case when HOB is equipped with a flue gas condensation system, its heat efficiency could reach 110%, while simplified solutions provided a value of 90%; see Fig. 6.6. Most of the references for biomass HOB show high boiler efficiency, higher than 85%.

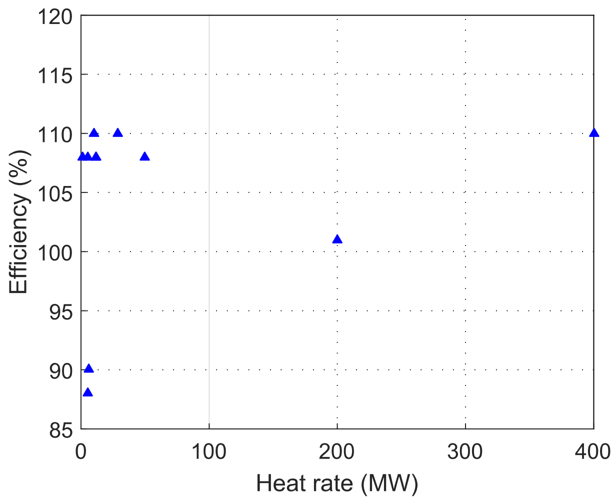

Fig. 6.6. HOB's efficiency versus heat rate

Fig. 6.7 shows COP for HP units.

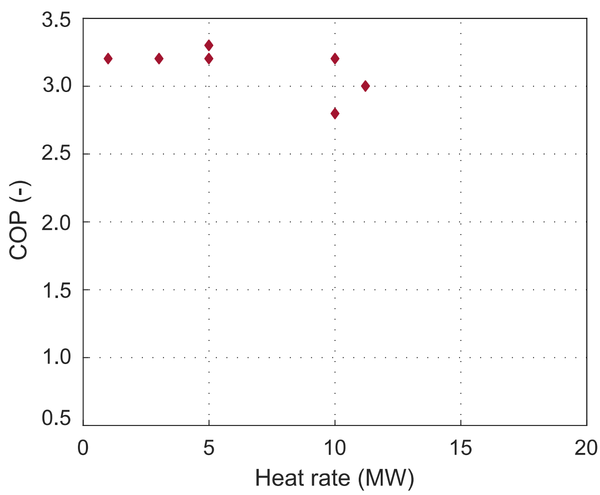

Fig. 6.7. HP's COP versus heat rate

Fig. 6.8 shows heat efficiencies for electric boiler technology. From Fig. 6.8 it can be seen that investigated electric boilers have efficiencies around 100%, while for the HP units the COP value is in the range of 2.8 – 3.3.

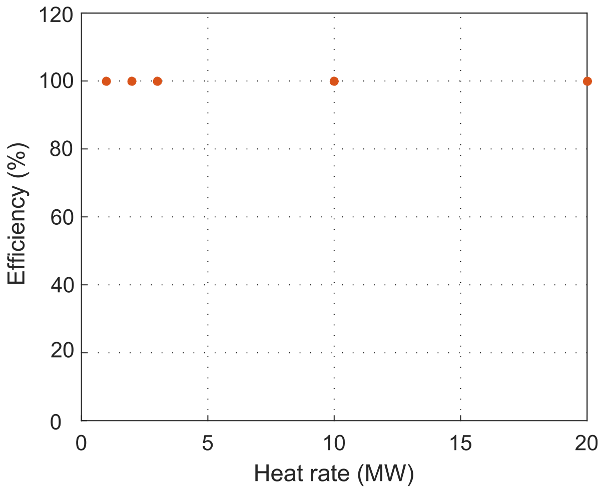

Fig. 6.8. Electric boiler's efficiency versus heat rate

The electric boilers do not have heat losses through a chimney. Therefore, the efficiency ranges from 95% to 100%. The lower values are for units installed outdoors because they have greater jacket heat loss. However, this is not the case found in Fig. 6.8.

6. OPTIMAL COMBINATION OF ENERGY PRODUCTION PLANTS

The comprehensive economic feasibility of the heat production units is impossible to gauge without fuel prices. In this study both CHP and HOB systems utilized biomass as a fuel. At the same time, electricity was required for HP operation. Hence, Table 6.1 summarizes the fuel prices for these technologies found in the literature for EU countries.

Table 6.1. Prices for fuel and electricity

Fuel type	**Price**	**Comment**	**Reference**
Electricity	0.120 EUR/kWh	Annual consumption level: 500 MWh – 2000 MWh; EU-28 in 2013	[144]
	0.127 EUR/kWh	Annual consumption level: 500 MWh – 2000 MWh; Euro Area (EA-17) in 2013	[144]
Wood chips	40 EUR/tonne		[145]
	70 EUR/tonne		[146, 147]
	56 EUR/tonne	Croatia, 2014	[148]
	58 EUR/tonne	Romania, 2014	
	136 EUR/tonne	Ireland, 2014	
	132 EUR/tonne	Austria, 2014	
	113 EUR/tonne	Germany, 2014	

6.2. Energy supply plants' models

For the purpose of feasibility, detailed plant models were necessary. In order to carry out economic and plant analysis, simplified models were developed based on detailed HYSYS models. The simplified models showed the relation between fuel inputs and heat plant output. Some examples of application are mentioned in [149-151]. The application of polynomial models was necessary due to the high number of plant variables and economic parameters. After the models were developed, the system simulations under different heat loads were executed.

6.2.1 Biomass based CHP models, detailed and simplified

The analyzed biomass CHP plant is shown in Fig. 6.9. The CHP plant utilized a steam cycle for generation of heat and power energies. The DH needs were satisfied by an intermediate pressure steam turbine (IPST). The temperature level in the DH system was 105-50°C.

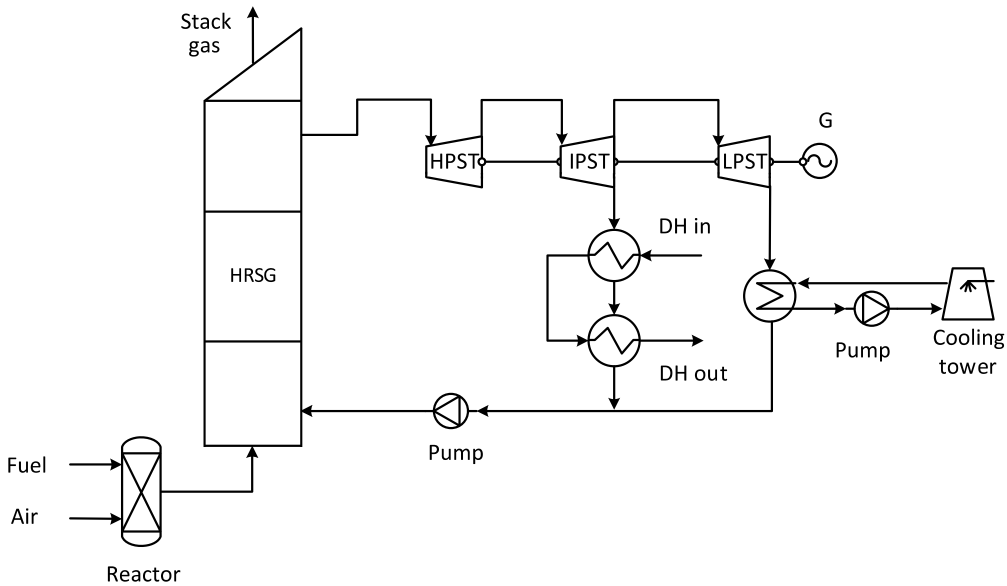

Fig. 6.9. Schematic of the biomass based CHP

The dynamic behavior of modern CHP plants is characterized by the short startup time and quick load change capability [152]. In order to ensure that operation of a CHP plant is realistic, the startup and standstill intervals were considered in the analysis. It was assumed that the CHP plant did not operate (was in standstill mode) if the DH load was low for longer than 72 hours.

Therefore, three startup modes [152] were applied when the condition of plant operation was satisfied:

- Hot start after 8 hours' standstill: 40 – 60 minutes;
- Warm start after 48 hours' standstill: 80 – 120 minutes;
- Cold start after 120 hours' standstill: 120 – 170 minutes.

According to [52], from both a technical and an economic point of view, a biomass plant is best operated relatively continuously at between 30% and 100% of its rated output. Biomass plants do not generally respond well to rapidly varying loads or long periods at low load conditions below a minimum modulating range. Therefore, the lower bound of a CHP's heat capacity applied in this study was equal to 30% of full plant capacity.

The heat capacities of the energy supply models were chosen based on 20%, 40%, and 60% of full DH load. Therefore, the heat capacities were 2.8 MW, 5.6 MW, and 8.4 MW. After the model simulation was conducted in Aspen HYSYS for different heat load, sufficient data points for defining the simplified mathematical model were obtained. Fig. 6.10 shows the relationship between power production and heat load in a CHP plant. Fig. 6.11 shows the dependencies between fuel consumption and heat load in a CHP plant.

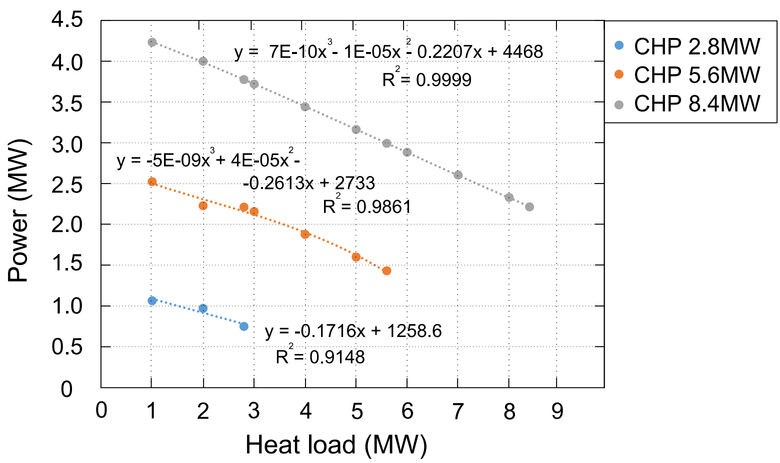

Fig. 6.10. Power production versus heat load in CHP plant

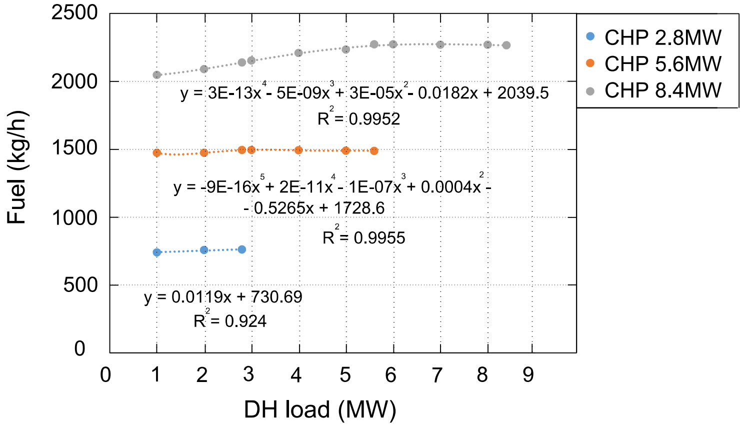

Fig. 6.11. Fuel consumption versus heat load in CHP plant

Finally, based on the model data, it was possible to calculate the CHP energy efficiency as a function of the heat load.

Fig. 6.12 shows the dependence of energy efficiency on heat load in CHP plant.

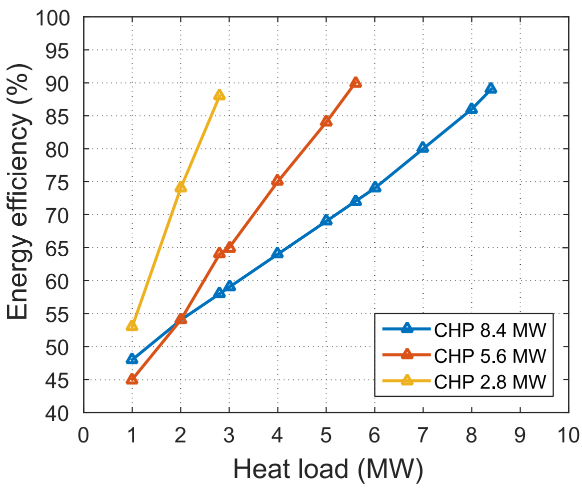

Fig. 6.12. Heat load versus CHP plant's energy efficiency

As can be seen from Fig. 6.12, the maximum energy efficiency of the CHP model is close to 0.9, for all three CHP sizes. The maximum efficiency was reached for the maximum heat load. Hence, the CHP's found energy efficiencies fit well with data shown in Fig. 6.5. Similarity in the energy efficiency values in Fig. 6.5 and Fig. 6.12 proved the high degree of quality of the applied CHP models.

6.2.2 Biomass HOB models, detailed and simplified

Nowadays, the most advanced HOBs are designed with the heat recovery of the flue gases. This innovation has led to improved efficiency. Fig. 6.13 shows a layout of a biomass HOB with flue gas heat recovery.

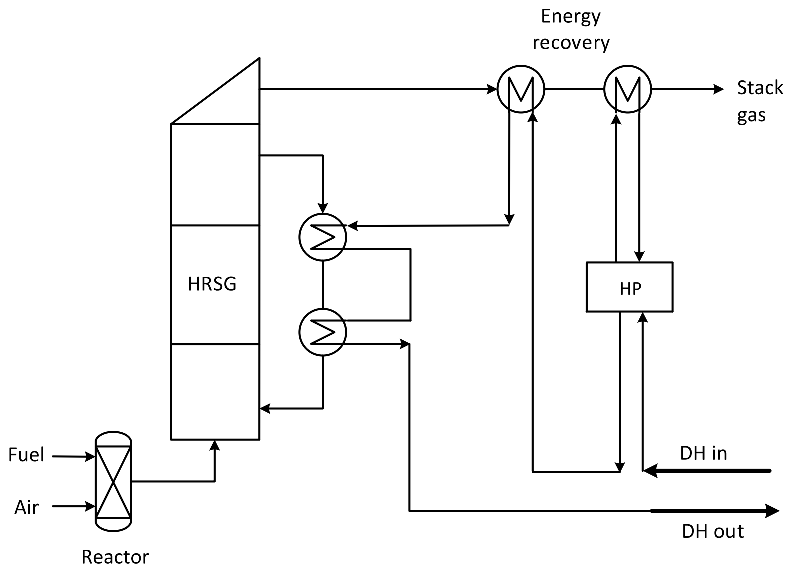

Fig. 6.13. Schematic of bio-based HOB with flue gas condensation

Fig. 6.13 shows a schematic of a biomass HOB with a two-stage condensing system for maximum energy conversion. In the first stage, the incoming DH water was preheated by an absorption HP, while in the second, it was after-heated and then supplied to the HRSG of the HOB. Fig. 6.14 shows the HOB that operated simultaneously with a small HP unit for increasing energy utilization. The absorption HP was driven by high-pressure steam with ammonia as a working liquid and water as an absorbent. In the condensing system the temperature of the flue gases decreased to 35°C and most of the water vapor was condensed to water. The supplied water temperature to the HRSG after the condensing system constituted 80°C. The return DH water from consumers had a temperature of 50°C and, after warming up in the HOB, the temperature reached 105°C.

Normally, the typical wood-fired HOB plants are controlled in the interval of 25 – 100% of full capacity, without violating emission standards. The best technologies can be controlled at 10 – 100% with fuel not exceeding 35% moisture content [47]. Therefore, the lower bound of the

HOB's heat capacity applied in this study was equal to 25% of full plant capacity. The HOB plant's ability to cover the heat load was based on three dimension sizes, marked 20%, 40% and 60% of the full DH load.

Fig. 6.14 shows polynomial models for the HOB in Fig. 6.13.

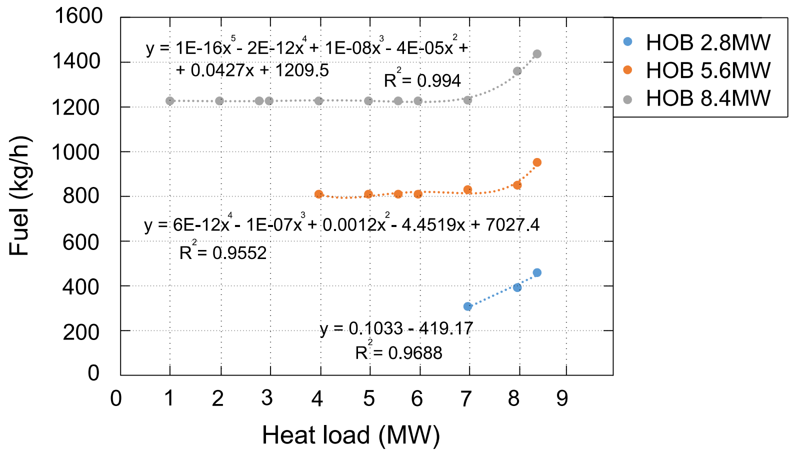

Fig. 6.14. Fuel consumption versus DH load in HOB

Fig. 6.15. shows dependence of HOB heat efficiency on heat load in DH system

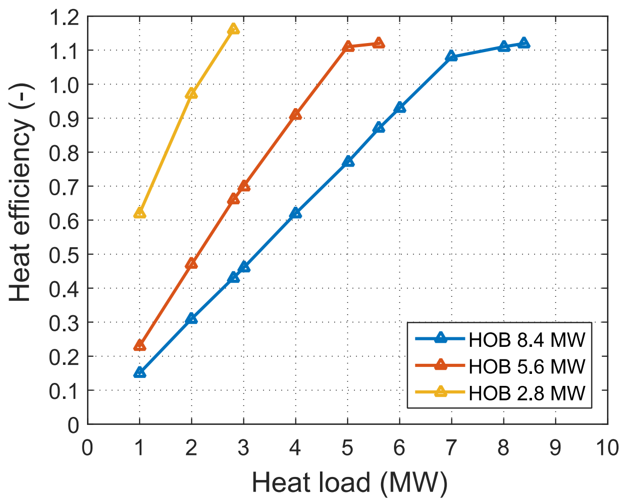

Fig. 6.15. Heat load versus HOB's heat efficiency

As can be seen from Fig. 6.15, the developed HOB models have maximum heat efficiencies of 1.12 – 1.16. This is mainly because flue gas condensation technology was used.

The heat efficiencies presented in Fig. 6.15 showed a match with the existing literature; see Fig. 6.6. This proved that the introduced HOB models were good and reliable for further analysis.

6.2.3 Vapor compression HP, detailed and simplified models

The mechanical heat pump (MHP) was based on the vapor compression principle and utilized ammonia as a working fluid. The scheme of the two-stage MHP is given in Fig. 6.16.

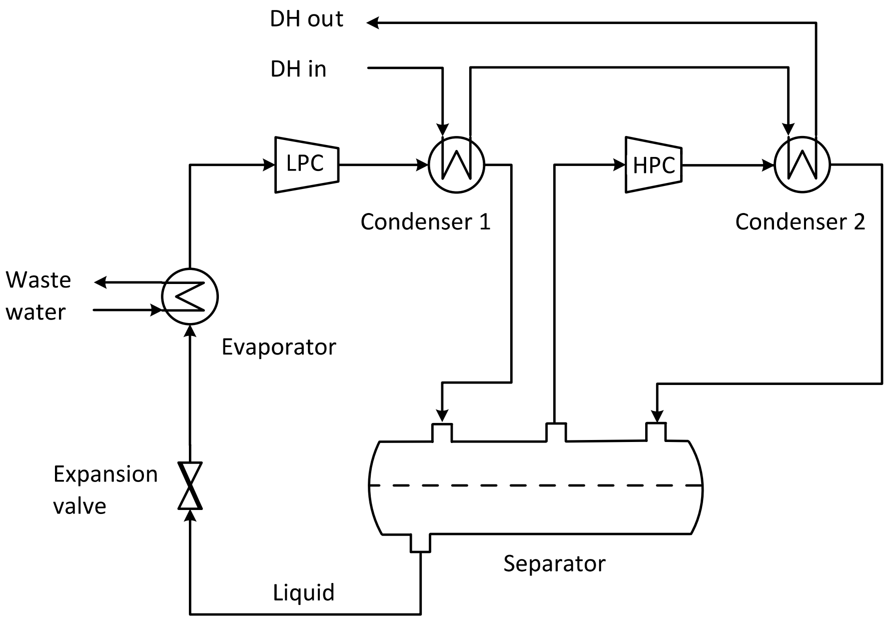

Fig. 6.16. MHP with two-stage compression and separation vessel

The main issue associated with the use of HP technology in DH systems is ensuring that the desired supply temperature is satisfied. This HP used Vilter's single-screw compressor [60]. This technology is referred as a high temperature heat pump (HTHP) used for industrial installations.

In the MHP system, the four main components of the HP, the evaporator, compressor, condenser, and expansion valve, were connected to a closed circuit. The MHP contained a separator vessel. The function of this vessel is to separate the refrigerant in the liquid and vapor. In the analyzed model the MHP was assumed to upgrade heat from residual waste water. The incoming temperature of residual water to the evaporator was 27°C. The HP plant's ability to cover the heat load was based on three dimension sizes: 2.8 MW, 5.6MW and 8.4 MW.

The polynomial model of the HP is depicted in Fig. 6.17.

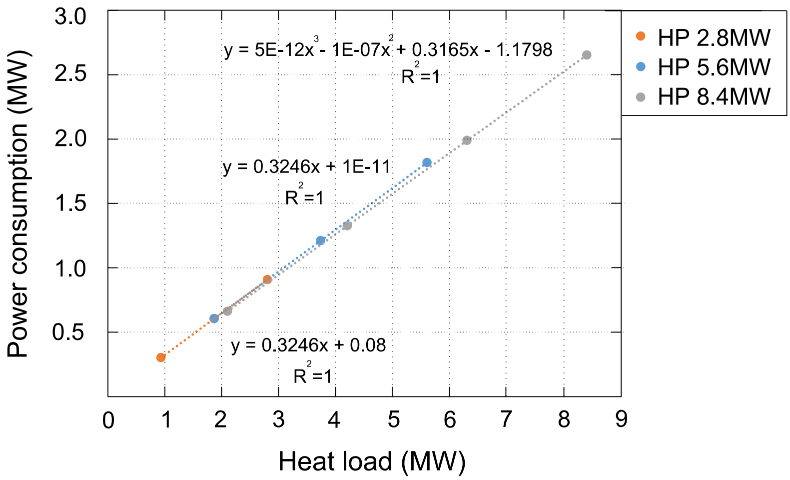

Fig. 6.17. Power consumption versus DH load in HP

Due to low variation in heat source temperature, the COP of the analyzed HP was almost equal to 3.3. Similar values were found in the literature; see Fig. 6.7 and Appendix I.

6.2.4 Electric boiler

The employed electric boiler model was described by linear dependency. The boiler control ability was adjusted between 10 – 100% [47] and had efficiency of $\eta = 99\%$, similar to that found in the literature; see Fig. 6.8 and Appendix I.

6.3. Methodology for analysis of the energy supply plants

6.3.1 Existing method of heat supply optimization

The existing method that is used by DH companies today is based on the methodology for finding the optimal generation mix in some target year. This method was developed primarily for electrical energy planning and is explained in detail in [153]. Further, the method was adjusted to DH needs and presented by authors in [1].

6. OPTIMAL COMBINATION OF ENERGY PRODUCTION PLANTS

The basic cost function for the heat generation optimization can be expressed as:

$$C = C_{fix} + C_{var} \tag{6.1}$$

where C is a total annual cost, which consists of an annual fixed cost, C_{fix}, and a variable operating cost C_{var}.

The corresponding cost for each heat capacity unit will be:

$$c = c_{fix} + c_{var} \cdot \tau \tag{6.2}$$

where c is a specific total cost per capacity unit, c_{fix} is a specific capacity cost per capacity unit, c_{var} is a variable cost per heat unit, and τ is operation time.

The specific capacity cost per capacity unit can be found as:

$$c = C/P \tag{6.3}$$

where P is the installed heat power capacity for each plant.

The specific capacity cost per capacity unit is found as:

$$c_{fix} = C_{fix}/P \tag{6.4}$$

Thus, the variable cost per heat unit can be expressed as:

$$c_{var} = C_{var}/Q \tag{6.5}$$

where Q is annual heat supply.

The break-even times of plants' operation can be found for a number of different energy production units that are included in the optimization process. Eq. (6.6) and Eq. (6.7) show a situation where three energy production plants are optimized in order to find the lowest annual total cost.

The break-even times $\tau_{1,2}$ and $\tau_{2,3}$ are obtained using the basic optimization condition that stipulates that the total cost should be equal for two competing plants at each intersection:

$$\tau_{1,2} = (c_{fix,2} - c_{fix,1})/(c_{var,1} - c_{var,2}) \tag{6.6}$$

$$\tau_{2,3} = (c_{fix,3} - c_{fix,2})/(c_{var,2} - c_{var,3}) \tag{6.7}$$

6.3.2 The new methodology for analysis of the energy supply plants

In order to combine the plants properly, there is a need to identify the total number of combinations. Therefore, the basic formula for the number of possible combinations of k objects from a set of n objects can be written as:

$$\binom{n}{k} = \frac{n \cdot (n-1) \dots (n-k+1)}{k \cdot (k-1) \dots 1} = \frac{n!}{k! \cdot (n-k)!} \tag{6.8}$$

Eq. (6.8) allows the total number of possible plants' sets with three elements in each to be found.

The method implied the use of plant capacities in the proportion of 20%, 40%, and 60% of the full DH load, which makes it easier to develop combination sets. In this study heat generation units were combined in three dimension sizes: 2.8 MW, which corresponds to 20% of the full DH load, 5.6 MW, equal to 40% of the full DH load, and 8.4 MW, equal to 60% of the DH load. One of the conditions is that a combination set should employ different technologies without repetitions. Another is that three plants should not have a total heat capacity of more than 100% of the DH load, e.g. 14 MW. Therefore, under these conditions, the number of generated plant combinations (PCs) by Eq. (6.8) was limited to 36.

Fig. 6.18 shows how the plants were combined. The PCs are based on the plant's ability to satisfy base load. When one technology is chosen for the base load, other technologies cover the rest of the load as intermediate and peak load plants.

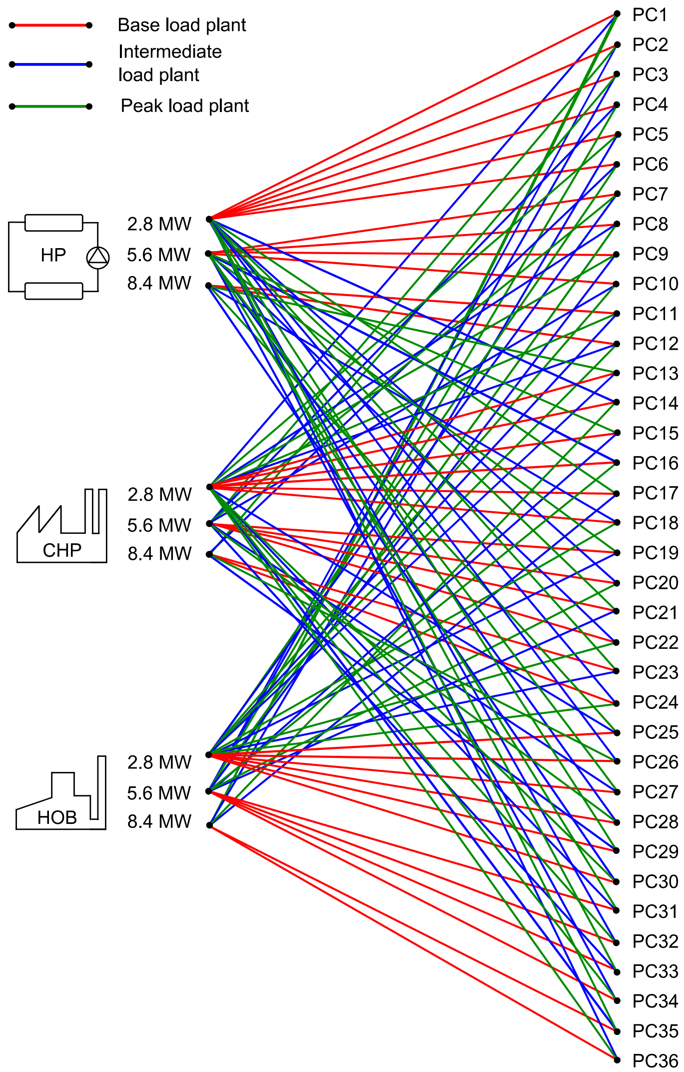

Fig. 6.18. Analyzed combinations of energy supply sources

Fig. 6.18 shows three energy generation technologies with different heat outputs developed in combination sets. In total there are 36 plant combinations marked PC. The colored lines indicate the plant's attachment to base load, intermediate load or peak load. The electric boiler was not included in Fig. 6.18; however, each combination has an electric boiler of 3 MW of heat output to cover extreme operation situations and as a back-up plant.

Fig. 6.19 introduces the information flowchart for the new method used in this analysis.

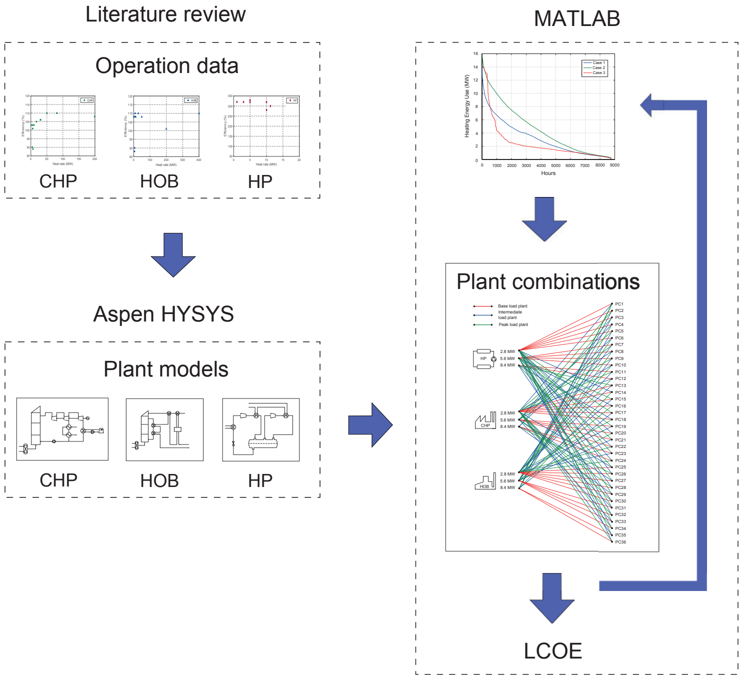

Fig. 6.19. Flowchart showing analysis steps for the new method

6.3.3 Economic evaluation

In Section 6.1 the overview of the cost data for technologies and fuel prices was presented. This section introduces a methodology for the cost analysis of heat generation. In this study, the levelized cost of energy (LCOE) [154] approach was used to compare PCs. The LCOE of a given technology is the ratio of lifetime costs to lifetime energy generation, both of which are discounted back to a common year using a discount rate that reflects the average cost of capital [155]. The LCOE allows alternative technologies to be compared with different scales of operation, different investment and operating time or both [154].

The energy system's operation cost consists of two parts: firstly, the O&M cost (labor, spare parts, external costs, management cost, insurance plant cost) and, secondly, the fuel cost [156]. Therefore, the total costs can be estimated as a sum of capital investments, fixed and variable O&M costs, and the fuel consumption costs. Hence, the LCOE can simply be presented as:

$$LCOE = \frac{Total\ Life\ Cycle\ Cost}{Total\ Lifetime\ Energy\ Production} \tag{6.9}$$

The total life cycle cost in Eq. (6.9) includes capital investment cost, O&M cost and fuel cost. The capital investment cost can be estimated as:

$$I_t = I_{CHP} + I_{HOB} + I_{HP} + I_{Elb} \tag{6.10}$$

where I_{CHP}, I_{HOB}, I_{HP}, I_{Elb} are investment costs for the installation of CHP, HOB, HP, and electric boiler.

The fixed share of O&M includes all costs which are independent of how the plant is operated, e.g. administration, operational staff, planned and unplanned maintenance, payments for O&M service agreements, network use of system charges, property tax, and insurance. Re-investments within the scheduled lifetime are also included, whereas re-investments to extend the life are excluded. Variable O&M costs included consumption of auxiliary materials (water, lubricants, fuel additives), treatment and disposal of residuals, output related repair and maintenance, and spare parts (not, however, costs covered by guarantees and insurance) [47]. Therefore, the O&M costs can be calculated as:

$$M_t = C_{var}^{CHP} + C_{var}^{HOB} + C_{var}^{HP} + C_{var}^{Elb} + C_{fix}^{CHP} + C_{fix}^{HOB} + C_{fix}^{HP} + C_{fix}^{Elb} \tag{6.11}$$

where C_{var}^{CHP}, C_{var}^{HOB}, C_{var}^{HP}, C_{var}^{Elb} are variable O&M costs, and C_{fix}^{CHP}, C_{fix}^{HOB}, C_{fix}^{HP}, C_{fix}^{Elb} are fixed O&M costs for CHP, HOB, HP and electric boiler.

The fuel consumption cost was evaluated as a sum of biomass fuel consumed by the CHP and HOB, and electricity needed for the operation of the electric boiler and HP:

$$F_t = C_{fuel}^{CHP} + C_{fuel}^{HOB} + C_{el}^{HP} + C_{el}^{Elb} \tag{6.12}$$

where C_{fuel}^{CHP} and C_{fuel}^{HOB} represent the fuel cost for operation of the CHP and HOB, and C_{el}^{HP} and C_{el}^{Elb} represent electricity cost for the HP and electric boiler.

The allocation of the CHP's fuel cost between thermal production and electrical production was based on an energy method [157] that defines heat allocation as:

$$f_Q = Q/(Q + E) \tag{6.13}$$

where Q and E represent thermal and electrical production. There are many other allocation methods available. They are discussed and analyzed in Chapter 7 and journal article I at the end of this thesis. Finally, including all the costs, Eq. (6.9) can be rewritten as:

$$LCOE = \frac{\sum_{t=1}^{n} \frac{I_t + M_t + F_t}{(1+r)^t}}{\sum_{t=1}^{n} \frac{Q_t}{(1+r)^t}} \tag{6.14}$$

where I_t is investment expenditure in the year t; M_t is O&M expenditure in the year t; F_t is fuel expenditure in the year t; Q_t is heat generation in the year t; r is a discount rate; and n is the lifetime of the system. In this work only heat costs were analyzed. The influence of the CHP's power generation was not treated. Hence, allocation was used as in Eq. (6.13). It should be noted that different allocation methods exist (see Chapter 7) and may provide various allocations between heat and power products. In turn, this could lead to an increase in LCOE for DH.

The discount rate is meant to reflect the loss of utility from deferred consumption and the degree of systematic risk of the project (i.e. the degree to which the economic properties of the project are aligned with the economy as a whole) [158]. The discount rate to be used in socio-economic analyses in the energy sector in Norway is determined by the Norwegian Water Resources and Energy Directorate (NVE) [159], based on instructions from the Ministry of Finance. DH is normally considered an investment with low economic risks [1]. Therefore, NVE has analyzed the energy sector with respect to the amount of systematic risk of various projects and stated that a discount rate of 4.0% – 6.5% should be applied for bio-based DH systems [160, 161].

The technical life of technologies can be adopted from [47, 49, 162]; for biomass CHP it is typically 20 – 25 years, for biomass HOB, large-scale vapor compression HP, and electric boiler, this value is 20 years [47].

Table 6.2 shows a summary of costs used for this study.

Table 6.2. Investment and O&M costs used in the analysis

Plant type	**Plant capacity (MW)**	**Investment costs (MEUR/MW)**	**Fixed O&M cost (EUR/MWhfuel)**	**Variable O&M cost (EUR/MWhfuel)**
CHP	2.8	3.0	2.0	2.6
	5.6	2.6	2.0	2.6
	8.4	2.3	2.0	2.6
Biomass HOB	2.8	0.8	2.1	2.0
	5.6	0.6	2.1	2.0
	8.4	0.5	2.1	2.0
HP	2.8	0.25	6.0	0.2
	5.6	0.42	6.0	0.2
	8.4	0.6	6.0	0.2
Electric boiler	3.0	0.15	1100 EUR/MW/year	0.5 EUR/MWh

The plant capacities in Table 6.2 were chosen based on considerations described in Section 6.3.2. After an evaluation of different prices of biomass fuel and electricity rates presented in Table 6.1, the biomass fuel price was chosen as 75 EUR/tonne and the electricity price as 0.12 EUR/kWh.

6.4. Renewable plant combinations for DH based on existing and new methods

To discuss and analyze the renewable plant combinations, firstly, an explanation of the existing method of heat supply optimization is provided. Then, the findings from the utilization of the new method are shown.

6.4.1 Results of the existing method

The main idea behind different optimization methods for energy supply solutions is to find technologies that best satisfy DH operation from both technical and economic viewpoints. Therefore, the existing method for heat supply optimization balances operation cost and investment cost to achieve the lowest total annual cost. Fig. 6.20 introduces the plant optimization method in which the following assumptions are made: constant energy price; 0 – 100% control range of the plant capacities; no influence of plant size on the investment cost; constant plant efficiency regardless of the plant load. Table 6.3 introduces data for the existing method of heat supply optimization. These data are based on the literature review and are similar to data used in the new, suggested method. This makes it easier to compare the results of these two methods.

Table 6.3. Economic and performance data used for the analysis

	CHP	HP	HOB	Electric boiler
Investment cost (EUR/kW)	2600	420	630	150
Energy cost (EUR/kWh)	0.01	0.12	0.01	0.2
Plant efficiency (-)	0.88	3.0	1.0	0.99
Service lifetime (years)	25	20	20	20
Interest rate (%)	5	5	5	5

Fig. 6.20 shows results found for heat load under Case 1, introduced in Section 4.2. A summary of the heat load characteristics of Case 1 can be found in Table 6.4.

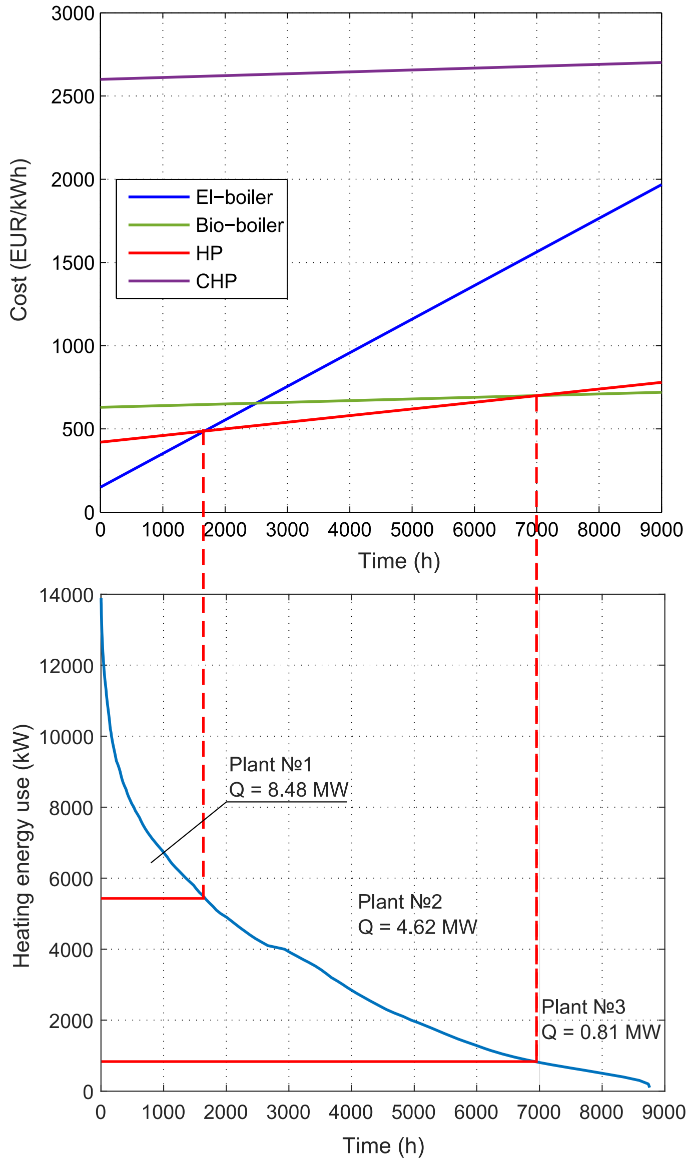

Fig. 6.20. Duration diagram showing linear cost characteristics for three plant models (upper diagram) and corresponding optimal division of plant capacities (lower diagram)

Fig. 6.20 shows that the electric boiler has the lowest investment cost and, therefore, it was beneficial to operate it in the interval between 0 – 1760 hours. The intermediate load should be covered by HP and the base load by HOB. Further, it can be seen that CHP is not a viable plant because it was too expensive to operate. In practice, it is well known that CHP is a reliable provider of heat and it is beneficial to operate it as a base load plant. EU-28 total gross production of derived heat in 2013 was 2.45 million TJ [163]. In Fig. 6.20, the plant capacities could be distributed as follows: for the peak load plant, an electric boiler of 8.48 MW; for the intermediate load plant, HP of 4.62 MW; and for the base load plant, HOB of 0.81 MW.

A sensitivity analysis of the existing optimization method, see Fig. 6.20, was performed in order to estimate the robustness of the method regarding the change in heat load. Table 6.4 shows the results found under the heat duration curves introduced in Section 4.2 and Table 4.2. Table 6.4 introduces DH cost under different scenarios of heat load, heat capacities, heat energy use of heat generation technologies, and operation hours.

Table 6.4. Sensitivity of the current optimization method to different load profiles

		Electric boiler	**HP**	**HOB**
Case 1 DH cost – 0.109 EUR/kWh	Heat capacity (MW)	8.48	4.62	0.81
	Heat energy use (MWh)	1352	12899	13216
Case 2 DH cost – 0.104 EUR/kWh	Heat capacity (MW)	8.22	7.13	1.03
	Heat energy use (MWh)	304	18510	21232
Case 3 DH cost – 0.083 EUR/kWh	Heat capacity (MW)	11.05	2.08	0.87
	Heat energy use (MWh)	1458	7902	12005
Operation hours		1166	5334	1 760

From Table 6.4 it can be seen that, with the change in heat load profiles, the optimal plant heat capacities show significant variation. For the electric boiler, the change was between 1% and 23%, for HP the change was 55% and 70%, and for HOB between 7% to 22% due to load change. This shows that this method is sensitive to changes in heat load profile. This means that the chosen plant combination is firmly connected to the observed load. In turn, this can lead to a low load factor for operated plants and further affect the DH cost in the case of load change. The number of plant operational hours remained the same under different loads. Further, the cost of heat generation in the DH system showed that a decrease in heat load profiles induced a minor decrease in DH cost. However, this is not always the case due to possible mismatches in plant operation. This means that more operating hours could require the same DH load to be fulfilled and, therefore, an increase in DH cost is inevitable.

Table 6.4 showed that the existing method of heat supply optimization is not applicable for long-term planning of DH systems considering variations in heat load profiles and is based on normalized DH load. However, the method is easy to use for rough estimations of production.

Fig. 6.21 shows deviation in DH cost due to variation in investment cost.

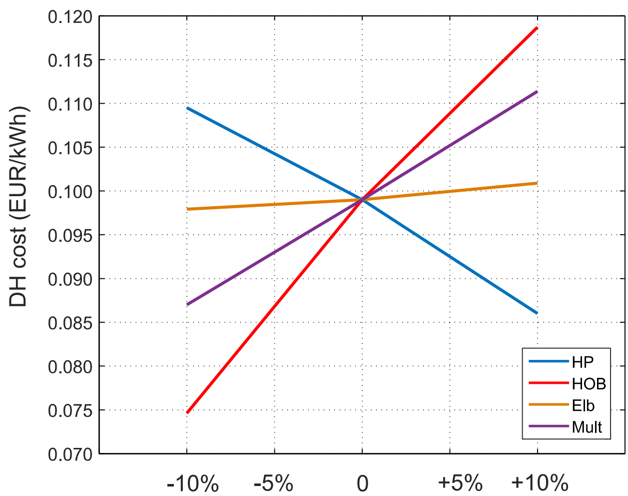

Fig. 6.21. Deviation in DH cost due to variation in investment cost

The change in the HP's investment cost in the range of ±10% induced a change of +9.6% – -12% in DH cost. In turn, the variation in investment cost for HOB has even larger consequences of -22% – +18%, while the electric boiler showed minor changes of less than ±1% for both increase and decrease in investment cost. The multiple uncertainties simultaneously showed a change of -11% – +11.4% on DH cost or ±0.01 EUR/kWh for both reduction and increase in investment cost. In comparison with HOB and electric boiler, HP showed the opposite tendency due to variation in investment cost.

Next, the sensitivity of DH cost to variation in energy cost was investigated and the results are shown in Fig. 6.22.

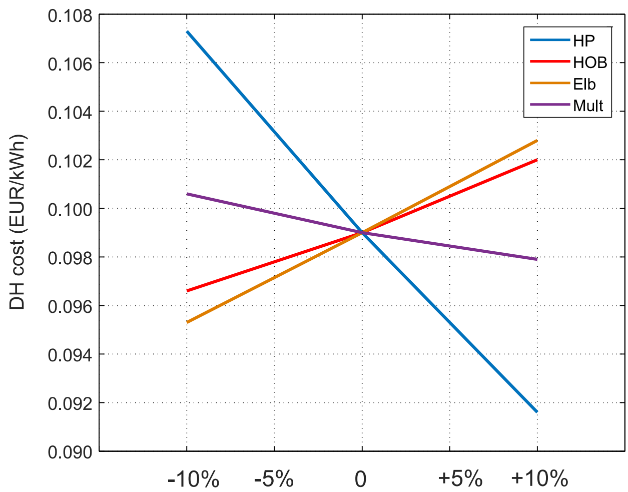

Fig. 6.22. Deviation in DH cost due to variation in energy cost

Fig. 6.22 shows the sensitivity of the existing method when the change in energy cost was induced for the analyzed technologies. The largest change in DH cost was induced by HP (+7.61% – -6.79%). For HOB the induced change in DH cost was in the range of -2.2% – +2.7%, and for electric boiler these values were -3.4% – +3.5%. The multiple uncertainties simultaneously showed change in DH cost by ±1% for both increase and decrease in energy cost. The energy price for HP and electric boiler operation has the largest impact on DH cost. As can be seen from Fig. 6.21 and Fig. 6.22, the change in the investment cost induced a larger change in DH cost than in energy cost. This means that the existing method of heat supply optimization is sensitive to change in the investment costs of the analyzed technologies.

The conclusion from the above analysis was that the existing method was sensitive to variations in heat load and investment cost. However, some expensive technologies such as CHP might be excluded from the optimization method due to the high investment cost. Further, this optimization is very simplified with respect to operation time and real performance. In addition, it does not show how DH should be equipped and operated over a long term when change in load appears [1].

6.4.2 Results of the new method

According to [155], the cost for electricity generation in Europe varies from a low of 0.06 EUR/kWh to a high of 0.19 EUR/kWh, depending on technology and local conditions. Therefore, Fig. 6.23 shows the LCOE results for the analyzed PCs that are competitive with the power generation cost and, consequently, with direct electric heating. The results are presented for PCs with LCOE lower than 0.2 EUR/kWh. The PCs with LCOE higher than 0.2 EUR/kWh can be found in journal article III at the end of this thesis.

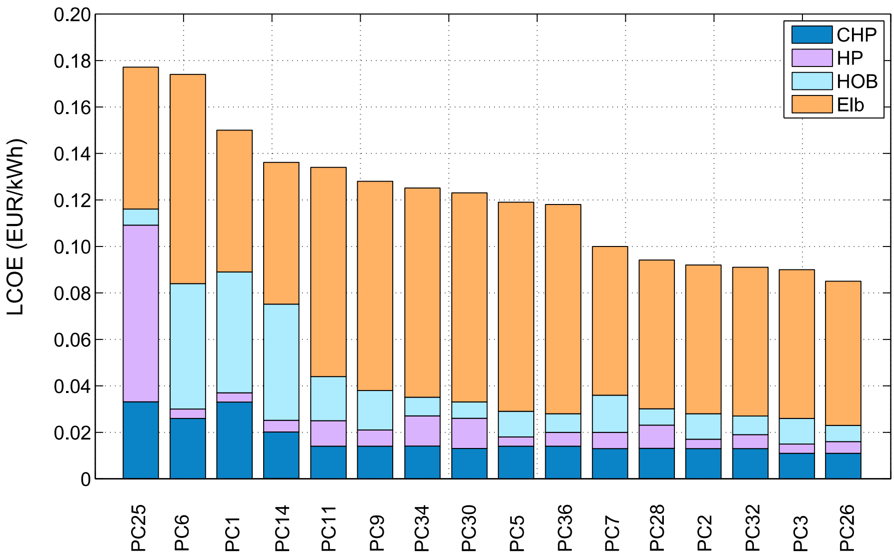

Fig. 6.23. Low LCOE

In this study, it was assumed that the electric boiler would be used to cover the heat load in the DH system due to limitations in the combined operation of HP, CHP, and HOB and during extreme operation situations. From Fig. 6.23 it can be seen that heat energy produced by electric boiler constitutes a high portion of the LCOE. Due to the high value of O&M cost, the operation of the electric boiler makes DH uncompetitive in comparison with direct electric heating. Further, in some combinations, its annual operation constituted only 38 hours (PC5, PC9, PC11, PC30, PC34, PC36); in spite of this fact, the found portion of the LCOE was high. This showed that operation of the electric boiler is expensive, even though its maximum heat output was only 3 MW of delivered heat. Next, it can be seen that HP's contribution to the LCOE was relatively low for the presented combinations. From this point, it can be concluded that the presented heat capacities of HP technology fit well with the analyzed PCs. The exception was combination PC25, where the 8.4 MW HP was operated as a peak load plant. This means that HP should not be used as a peak load plant with a high installed heat rate.

As was highlighted earlier, the electric boiler was used during extreme outdoor conditions. Fig. 6.24 shows the combined operation of energy supply plants based on PC28, in which the electric boiler operation is indicated.

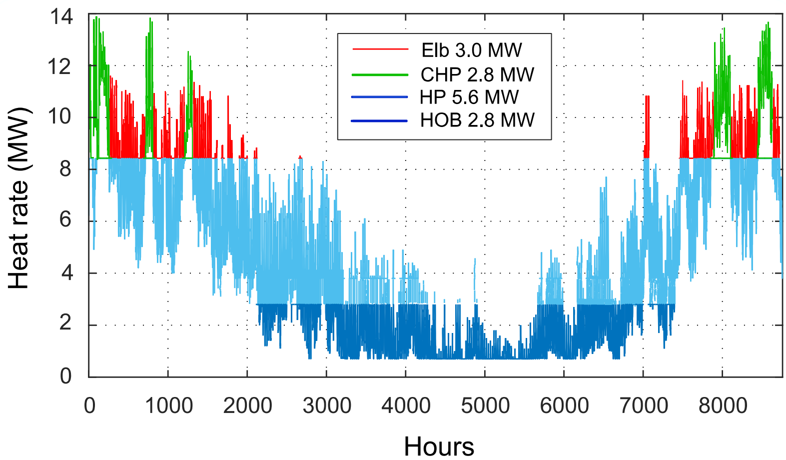

Fig. 6.24. Hourly heat rate distribution

From Fig. 6.24 it can be seen that, due to limitations in CHP operation, see Section 6.2.1, the electric boiler was used to cover DH load when CHP was in standstill mode. In general, running an electric boiler is convenient due to its simplicity and lack of limitations in operational regimes. However, in a long-term operation this can lead to an increase in DH price, which existing and new customers consider impermissible. Therefore, its operation should be limited.

Fig. 6.25 presents the contribution of fuel costs in LCOE for PCs shown in Fig. 6.23.

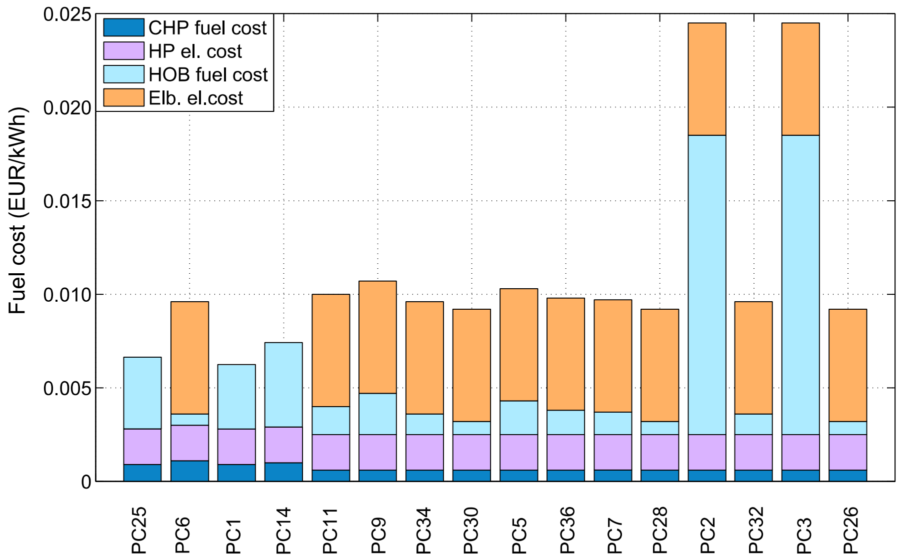

Fig. 6.25. Contribution of fuel cost in LCOE

Fig. 6.25 again shows that the highest fuel cost of each combination was due to operation of the electric boiler. The exceptions were PC2 and PC3, where the HOB was operated as an intermediate load plant. In addition, PC1, PC14, and PC25 operated without the electric boiler. Due to the high COP of the HP for these three combinations, the electricity use was low in comparison with the total LCOE value presented in Fig. 6.23. In countries with low electricity prices, such as in Scandinavia, the employment of HP for heat supply purposes is a good option for efficient heat-energy supply. The fuel use of the CHP was low, even for the configuration where its heat load share was 60%. A similar trend was found for the operation of HOB.

The next step of the analysis involved evaluating changes in LCOE due to different heat load patterns. The analysis was performed for combinations with low LCOE.

Fig. 6.26 presents the LCOE for different heat load patterns and different combinations of energy supply technologies. To recall, Case 2 introduces the scenario where the heat duration curve was under high occupancy and lower outdoor temperatures, Case 3 shows the scenario in which the heat duration curve is constructed for future building stock.

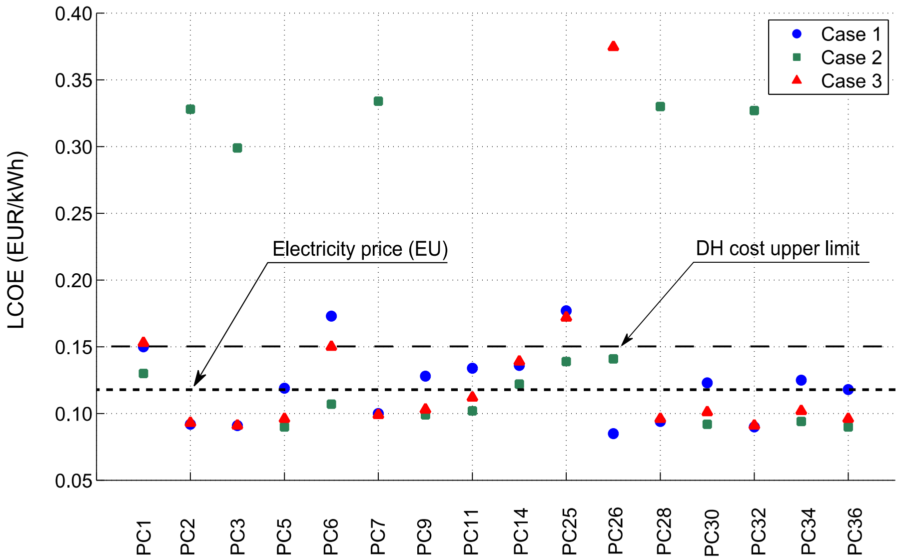

Fig. 6.26. LCOE values for analyzed scenarios

In order to stay competitive in the energy market, the heat generation cost should be lower than the alternatives. At this point, this means that the heat generation cost should be lower than the electricity production, to avoid switching to direct electric heating. As can be seen from Fig. 6.26, several combinations could be highlighted as being competitive in a long-term perspective. These combinations were: PC5, PC30, PC34, and PC36. Four additional combinations, PC1, PC9, PC11, and PC14, could be underlined as alternatives with LCOE values lower than 0.15 EUR/kWh. It can be noticed that all these combinations have a small CHP as a peak load plant. The exception is combination PC14, where a large HOB was utilized for this purpose. Further, in comparison to all the PCs presented in Fig. 6.23, the above-mentioned combinations found the lowest LCOE values under the duration curve of Case 2. This means that the heat load factor increased, which provided better energy utilization in the aforementioned combinations. The found plant sizes fitted perfectly with the required DH loads.

Among eight PCs (PC1, PC5, PC9, PC11, PC14, PC30, PC34, PC36), only one employed CHP as a base load plant. In addition, its heat capacity was only 2.8 MW. At the same time different sizes of HOB and HP were utilized for the base load plant. For intermediate load plants, the trend was similar, while for peak load plants most of the combinations employed the small CHP. This trend for peak load plants is due to the application of CHP's allocation method. In the current study the energy method was applied with the heat allocation value of $f_Q = 0.54$. However, the possible deviations in LCOE might be present due to the application of different allocation methods.

The summary of LCOE values under different heat load profiles can be seen in Table 6.5.

Table 6.5. Heat generation cost under different load profiles

Combination	Case 1 (EUR/kWh)	Case 2 (EUR/kWh)	Case 3 (EUR/kWh)
PC1	0.150	0.130	0.153
PC5	0.119	0.090	0.096
PC9	0.128	0.099	0.103
PC11	0.134	0.102	0.112
PC14	0.136	0.122	0.139
PC30	0.123	0.092	0.101
PC34	0.125	0.094	0.102
PC36	0.118	0.090	0.096

Table 6.5 shows that the variation in heat generation cost due to the change in heat load patterns was in the range of 12.2% – 25.2% or 0.017 – 0.031 EUR/kWh of heat. The lowest differences were found for the combinations PC14 and PC30. Hence, it could be concluded that these two combinations are the best solution for customers due to the smallest change in DH cost under different heat loads. However, combinations PC5 and PC36 should be highlighted as those, which showed generation cost reduction for both an increase and decrease in DH load. In PC36, 8.4 MW HOB was employed for the base load plant, 2.8 MW CHP covered the intermediate load and 2.8 MW HP was used for the peak load. Meanwhile, PC5 has the following results: HP of 2.8 MW for the base load plant, HOB of 8.4 MW for the intermediate load and CHP of 2.8 MW for the peak load. The combinations presented in Table 6.5 showed the lowest LCOE for different heat load profiles among all the 36 combinations. This is very important, since employing these

combinations means that DH customers would not experience large changes in the heat cost due to the load change, while obtaining the low heat cost.

Finally, Fig. 6.27 shows relation between the LCOE and system efficiency for different PCs under different heat load profiles.

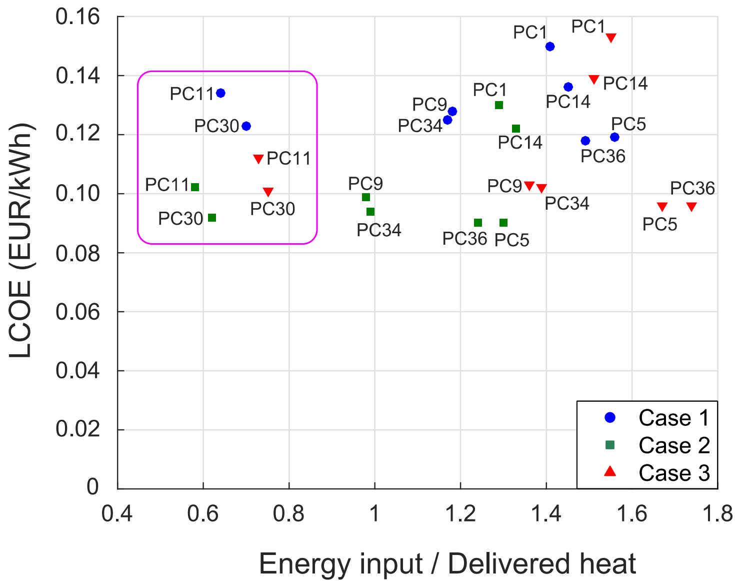

Fig. 6.27. LCOE and system efficiency for different heat supply options under three heat loads

Fig. 6.27 shows that plant combinations PC11 and PC30 are more energy-efficient under different heat loads than other combinations. As previously found, PC30 and PC14 had the lowest difference in values of LCOE under different heat loads; see Table 6.5. However, Fig. 6.27 shows that, in terms of energy input per delivered heat, PC30 is more efficient than PC14. Apart from PC30, the low value was found for combination PC11. The reason for this is that both PC11 and PC30 employed large HP for base load and intermediate load.

6.5. Discussion on methodology of choosing heat supply plants

The existing method of heat supply optimization is found to be a simple way to deal with all the costs and operation issues. It was found that the method is sensitive to change in heat load profiles. In turn, this could lead to a low load factor for operated plants and further increase the DH price. It is very simplified with respect to real sizes, operation times, and actual heat capacities. In comparison with the existing methodology, the new method suggested in this thesis is more sophisticated and involves deeper analysis.

The analysis of results from the new method showed that the operation of electric boilers can be avoided and DH companies should eliminate this technology from the DH system. In all the analyzed combinations, electric boiler operation constituted from 38 to 790 hours of intermittent operation at full heat capacity. It was shown that its employment was mainly due to the operational limitations of the other technologies. As an alternative to this, thermal energy storage (TES) could be considered. In addition, employing TES could lead to an increase in the heat load factor for intermediate and peak load plants.

All the PCs showed high LCOE values due to electric boiler operation. The LCOE remained high even when the electric boiler was not put in operation. The reason for this might be the high value of the fixed O&M cost used in the analysis. This value was adopted from a technical report [47] with reference to 2012. The meaningfulness of this value can be doubted. However, the report dated two years earlier showed this value in the same range, which makes the adopted value appear reasonable. Therefore, some changes in this value might alter the results of the study. However, any decrease in this high value of the fixed O&M cost would give a decrease in the DH cost.

Combinations with LCOE higher than 0.2 EUR/kWh can be treated from a viewpoint of future development and extension of DH systems. The found results are mainly due to actual operation and the low heat load factor facing those PCs.

The fuel was allocated between heat and power production by the energy method. In turn, this made the CHP operation highly efficient due to the subsidizing of low heat load by electricity load and further fuel allocation to power production. This resulted in CHP operation being efficient as a peak load plant. However, a number of technical allocation methods were developed and used in different countries. Therefore, the possible deviations in LCOE might be present due to the application of different allocation methods. The difference between allocation methods and results can be found in Chapter 7 of this thesis and journal article I.

The example of the existing method of heat supply optimization found that it was inappropriate to utilize CHP due to its high investment cost. However, the new method showed the opposite result; small CHP plants could be employed for peak load operation. This is a good observation, since it concurs with Directive 2004/8/EC [97] on the promotion of highly-efficient cogeneration. The more CHP is used, the more primary energy is saved, and the higher the security of the energy supply.

If one considers the four technologies discussed in this study, modern HOBs are shown to be very effective. In comparison with other technologies, their linear cost characteristic could show a decrease with an increase in operation hours. This provides the possibility to employ a single HOB for annual operation. However, the employment of a single plant decreases security of supply in the DH systems. To avoid this, the need arises for several heat production units. Hence, the cost difference of utilizing four plants would always be greater than with three or two. Therefore, it can be concluded that the heat generation cost increases with the increase in the DH's flexibility and reliability of supply.

6.6. Conclusion on choosing renewable plant combinations

The results from the new method found that the operation of an electric boiler led to a high value of LCOE, in spite of the fact that it was operated sporadically and the maximum heat output was 3 MW of heat. Next, one should consider electricity rates, since few countries have cheap electricity as is the case in Norway and Sweden. This revealed that the operation of an electric boiler is rather expensive and should be kept to the minimum. In addition, policy makers should provide legislative framework to limit the use of this technology for DH.

The study identified 16 PCs with the LCOE under 0.2 EUR/kWh. However, not all of these were found to be insensitive to changes in heat load profiles. Further, eight PCs were selected as those with low sensitivity to heat load variation and LCOE under 0.15 EUR/kWh (PC1, PC5, PC9, PC11, PC14, PC30, PC34, and PC36). It was noted that six of those have a small CHP as a peak load plant. Among the eight combinations, only one employed CHP as a base load plant with a heat capacity of 2.8 MW. It was concluded that the operation of a large HP for the peak load should be avoided due to the low load factor and high investment cost.

The analysis on system efficiency found that combinations PC11 and PC30 showed the most rational utilization of energy input under different heat loads. The main reason for this is that a large HP was used in these combinations to satisfy the base load and intermediate load.

The uncertainty in fuel price meant that the biggest variation in the total LCOE was found in combinations in which the HOB was operated as an intermediate load plant. This means that an increase in fuel price will have a negative effect on LCOE for this technology employed for intermediate load. The total deviation in LCOE values for the presented combinations due to fuel price variation was in the range of 1.6% – 3.6% or 0.002 – 0.005 EUR/kWh. The consequences of the price variation for the HP were smaller than for the CHP and the HOB in the analyzed range. One of the reasons for this was that the cost foundation is different for electricity production and for wood chip collection. However, in some countries electricity rates are rather high, and a normal trend is for this to increase over time. In turn, this can lead to an additional portion of O&M cost when the HP technology is chosen for operation.

The uncertainty in PCs due to changes in investment cost in the range of ±20% has an effect of 14 – 20% on LCOE. Hence, underestimation of investment cost can lead to significant changes in LCOE values for these technologies.

The uncertainty due to model quality meant that the HP model has a larger effect on LCOE in comparison with CHP and HOB. The deviation in the range of ±10% induced a change in LCOE of 1.42% – 4.7%. In the case of the HOB and the CHP models, the consequences were smaller, around 1%. The impact of multiple uncertainties simultaneously found changes in the range of 4% – 6%. The conclusion is that the presented models and the analysis approach proved to be sufficiently accurate for the purpose of this study. Consequently, the results and conclusions might be treated as reliable.

7. ALLOCATION FACTORS

This section deals with the influence of the CHP plant's different design and off-design parameters on allocation factors. The majority of CHP plants are constructed based on parameters designed for local conditions. However, their regular operation is mainly executed in off-design conditions. The efficiency indicators of the CCPP technology are found to be the highest among other technologies based on the CHP platform [1]. Therefore, changes in allocation factors for heat and electricity were investigated in a natural gas based CCPP.

There are many different methods for the allocation of synergy benefits in joint generation processes. The choice of allocation method has a great effect on energy pricing and emission allocation in CHP plants. These days, the power bonus method is one of the main methods used for these purposes in the EU and is given as a standard. However, aside from this method, six different technical allocation methods and a number of economic based methods are known and in use in different countries. The question of how to set true cost-based prices for a CHP plant has been discussed within the energy industry and by energy market planners for almost 100 years, but still no general solution is available [1].

This chapter aims to provide more insight into the features of available allocation methods and their influence on product allocation in a CHP plant. Discussions are provided on sensitivity issues and difficulties in the data quality for each method. Further, the influence of a CHP plant's different design and off-design parameters on allocation factors was discussed. The main research questions addressed in this study were:

- How could change in off-design conditions affect the allocation factors?
- What is the sensitivity of different allocation methods?
- Which method can properly deal with the temperature levels that are necessary for a transition to LTDH?

This section provides a brief summary of the allocation issues and results. The complete study can be found in journal article I and conference paper IV, attached at the end of this thesis.

7.1. Overview of allocation methods

This section provides an overview of technical-based allocation methods used in the study. The economic-based allocations were not investigated since such methods are prone to being misleading and fluctuating markedly with price swings for fossil fuels. The economic-based allocations are easily influenced by decision and policy makers [100].

7.1.1 The energy method

The *energy method* is most widely used because of its simplicity. This is an example of physical allocation. The primary energy use is allocated between the heat and electricity produced in the CHP plant. If the amount of electricity produced in the CHP plant is 70% and the amount of heat is 30%, this means that 70 units of energy are allocated for power production and 30 for heat production. This means that, in the *energy method,* the allocation factors can be expressed as:

$$f_Q = Q/(Q+E) \tag{7.1}$$

$$f_E = E/(Q+E) \tag{7.2}$$

where f_Q and f_E denote fractions of emissions allocated to heat and electricity production, respectively. In Eq. (7.1) and Eq. (7.2), Q and E represent thermal and electrical production, respectively. This method does not take any energy quality aspects into account, allocating a lower impact to electricity than to the other methods [142]. Consequently, it can be argued that it underestimates the share of the emissions allocated to electricity production [111].

7.1.2 The alternative generation method

The *alternative generation method* was developed by the Finnish District Heating Association [164]. In the alternative generation method, the share of CO_2 emissions is beneficial for both the heat and the power production in the CHP plant. The method allocates emissions and resources to the heat and power production in proportion to the fuel needed to produce the same amount of heat or power in separate plants. These alternative plants use the same fuel as the CHP plant [165]. Consider a CHP plant, which consumes 100 units of energy, while producing 30 units of electricity and 60 units of heat. Alternative production in two separate plants, a heat only plant and a condensing plant, will depend on their efficiencies, η_{heat} and η_{elec} respectively. In order to produce the same amount of electricity and heat, the separate plants will consume more fuel, because of lower separate efficiencies in comparison with cogeneration. The allocation of heat and electricity will be based on the amount of fuel needed if separate production plants had been used [142]. From the following example, the allocation factor can be expressed as:

$$f_Q = \left(\frac{Q}{\eta_{alt\_heat}}\right) \Big/ \left(\frac{Q}{\eta_{alt\_heat}} + \frac{E}{\eta_{alt\_elec}}\right) \tag{7.3}$$

$$f_E = \left(\frac{E}{\eta_{alt\_elec}}\right) \Big/ \left(\frac{Q}{\eta_{alt\_heat}} + \frac{E}{\eta_{alt\_elec}}\right) \tag{7.4}$$

where η_{alt_heat} and η_{alt_elec} are the heat and power production efficiencies of producing thermal and power energy via an alternative generation plant. This allocation method, therefore, shares the emissions among the products in a particular format and treats one or the other product as the primary one [111].

7.1.3 The power bonus method

The *power bonus method* is the most recognizable method for energy allocation, because it is promoted by European standard EN 15613-4-5:2007 [101] and is widely used today. In this method the heat is the main product, while all power is considered as a bonus. The primary energy is allocated to the electricity produced in the CHP plant. The total primary energy used by the CHP plant includes all energy used in the production of heat and electricity. This includes the primary energy related to fuel handling and combustion as well as the primary energy needed for the production of additives, handling of ashes, construction, and dismantling of the CHP plant, etc. In accordance with EN15316-4-5:2007, the performance of the DH system and the produced heat in the CHP plant can be rated by evaluating the primary energy factor (PEF) $f_{P,dh}$ of the specific DH system. The PEF is defined as the primary energy input $E_{P,in}$ to the system divided by the heat Q_{del} delivered at the border of the supplied building [101].

$$f_{P,dh} = E_{P,in}/Q_{del} \tag{7.5}$$

The thermal energy balance is given by:

$$f_{P,dh} \cdot \sum_j Q_{del,j} + f_{P,el} \cdot E_{el} = \sum_i f_{P,F,i} \cdot E_{F,i} \tag{7.6}$$

From Eq. (7.6), the PEF of the DH system can be expressed as:

$$f_{P,dh} = \left(\sum_i f_{P,F,i} \cdot E_{F,i} - f_{P,el} \cdot E_{el}\right) \Big/ \sum_j Q_{del,j} \tag{7.7}$$

where $f_{P,dh}$ is the PEF of the DH system, $f_{P,F,i}$ is the PEF of the fuel for the cogeneration plant, $f_{P,el}$ is the PEF of replaced electrical power, E_{el} is the electricity from the cogeneration plant, Q_{del} is the delivered heat at the border of the supplied building, and $E_{F,i}$ is the fuel input to the cogeneration plant.

Finally, in the *power bonus method,* the allocation of primary energy can be expressed as:

$$f_Q = f_{P,dh} \cdot Q_{del}/(Q_{del} + E_{del}) \tag{7.8}$$

$$f_E = 1 - f_{P,dh} \cdot Q_{del}/(Q_{del} + E_{del}) \tag{7.9}$$

This method promotes cogeneration technology instead of the separate production of heat and electricity. It also promotes the usage of different renewables like municipal waste, pellets, biofuels, etc. Today, the *power bonus method* is one of the most efficient methods for promoting DH technology; as power is counted as a bonus, the largest part of CO_2 emissions is allocated to power production.

7.1.4 The exergy method

The *exergy method* represents allocation from a thermodynamic point of view. This is an example of physical allocation; it defines the quality of energy. The exergy is the maximum amount of work which can be obtained from a system when it interacts with the environment in a reversible way. For exergy analysis, the characteristics of the reference environment must be specified completely. This is commonly done by specifying the temperature, pressure, and chemical composition of the reference environment. The results of the exergy analyses, consequently, are relative to the specified reference environment, which, in most applications, is modeled after the actual local environment. The exergy of a system is zero when it is in equilibrium with the reference environment [111]. A number of authors have carried out exergy analysis in their research for different purposes [38, 166-168].

From the thermodynamic point of view, electricity consists of 100% exergy, and consequently the exergy of electricity is defined as:

$$Ex_E = E \tag{7.10}$$

According to the exergy method, the heat allocation can be calculated based on the following equation:

$$Ex_Q = (1 - T_0/T) \cdot Q \tag{7.11}$$

where Ex_E and Ex_Q are net output of electricity and thermal exergy from cogeneration, and T and T_0 are the medium and mean ambient temperatures of the heating period. When the heat is transferred at a sliding temperature, Eq. (7.11) is not valid. In that case, the temperature T should be replaced by the logarithmic mean temperature of the temperatures at which the heat is transferred. In the case of the DH system, these temperatures are the supply and return temperatures of the DH network, T_s and T_r [169], and then the temperature of the medium can be defined as:

$$T = (T_s - T_r)/ln(T_s/T_r) \tag{7.12}$$

Consequently, the heat exergy can be defined as:

$$Ex_Q = \left[1 - \frac{T_0}{(T_s - T_r)/ln(T_s/T_r)}\right] \cdot Q \tag{7.13}$$

Finally, the allocation factors for the heat and electricity based on the *exergy method* become:

$$f_Q = Ex_Q/(Ex_Q + Ex_E) \tag{7.14}$$

$$f_E = Ex_E/(Ex_Q + Ex_E) \tag{7.15}$$

The application of this method requires profound knowledge of thermodynamics and power plant processes and is therefore rather complicated for practical use. However, it is judged to be the fairest method, from a thermodynamic point of view, for dividing the benefits of the CHP production between electricity and heat [170] and can be carried out relatively simply because the necessary data can be measured directly in the plant. Thermodynamically, however, the method is not really “clean” because the losses of exergy caused by the heat exchange from the cogeneration process to the heating system are not allocated to the heat [112]. Consequently, compared to the energy allocation method, the exergy method avoids the difficulties associated with the allocations based on energy values. Such methods are problematic, especially for cogeneration systems, because the two main products are of significantly different quality and usefulness [100, 111].

7.1.5 The 200% method

The *200% method* uses 200% efficiency for heat production. This means that all emissions are left to power production. This method, which was established by the Danish Energy Agency [171], is similar to the power bonus method, where all electricity is counted as bonus. It is well known in Denmark, where there are large-scale CHP plants, which primarily produce power, and small-scale CHP plants for producing heat. The Danish Energy Authority has stipulated that energy efficiency of 200% has to be used when allocating the fuel costs of the CHP to the heat production in the energy and emission statistics. This means that, in order to produce two units of heat energy, one unit of real fuel has to be used and the other unit will be recovered from the heat otherwise directed to the turbine condenser. In the condenser, the heat unit would be wasted to the environment if not recovered to DH [172]. Finally, in this method, the allocation factor for heat and electricity can be defined as:

$$f_Q = Q/(2 \cdot Fuel_{in}) \tag{7.16}$$

$$f_E = 1 - Q/(2 \cdot Fuel_{in}) \tag{7.17}$$

where $Fuel_{in}$ is the total primary fuel energy consumed in the cogeneration plant. This method assumes that the heat is produced with fixed efficiency, which is chosen as a general average between the energy and exergy methods [142].

7.1.6 PAS 2050

The publicly available specification *PAS 2050* [173] is the British standard, which explains the calculation of GHG of goods and services. The allocation of emissions in the CHP is between the heat and power produced, multiplied by the intensity of the GHG emissions of the production unit. The special coefficient specifies the emissions released from fuel combustion used in the system. For boiler-based CHP systems (coal, wood, solid fuel), the coefficient is 2.5, while for turbine-based CHP systems (natural gas, landfill gas), the coefficient is 2.0.

Finally, the allocation factors in this method can be expressed as:

$$f_Q = Q/(n \cdot E + Q) \tag{7.18}$$

$$f_E = (n \cdot E)/(n \cdot E + Q) \tag{7.19}$$

where n is the intensity of the GHG emissions of the production unit. It is important to note that these ratios apply to 1 MJ of energy produced. In most situations more energy of one type than of another will be produced. The allocation of emissions to heat and electricity arising from the CHP relies on the process-specific ratio of heat to electricity from each CHP system. For example, where a boiler-based CHP system delivers useful energy in the PHR 1:6, 2.5 units of emissions would be allocated to each unit of electricity and one unit of emissions would be allocated to each unit of heat delivered by the CHP system. This means that the CHP system has a useful PHR of 1:6; the corresponding GHG emissions ratio is 2.5:6. These results will change with the different heat and electricity characteristics of the CHP system [174].

7.1.7 The Dresden method

The *Dresden method*, which was proposed by Zschernig and Sander [113], is based on exergy assessment. In power plants all primary energy is related to electricity production. At the same time in the CHP plants, one part of primary energy is consumed for thermal energy production. The *Dresden method* describes how to evaluate the electricity loss caused by the heat extraction (water steam condensation) in the CHP plant. The electricity losses due to heat extraction in the CHP plant can be evaluated as:

$$\Delta E = Q \cdot \eta_c \cdot v_p \tag{7.20}$$

where

$$\eta_c = 1 - T_{cond}/T_{out} \tag{7.21}$$

and the maximum electricity production without heat extraction is:

$$E = E_{del} + \Delta E \tag{7.22}$$

where ΔE is electricity loss due to heat extraction in the CHP plant, E is electricity energy generated in CHP plant including electricity losses (maximum electricity production without heat extraction). E_{del} is electricity energy generated in the CHP plant when heat extraction occurred. η_c is Carnot efficiency; T_{cond} and T_{out} are condensing temperature and temperature of extracted steam in the CHP plant, respectively. Mainly in smaller heat and power stations, where the determination of the heat losses is complicated, the exergy of the heat rated by a real degree of process quality v_p can be used as an equivalent of the electricity loss [112]. The fuel in the cogeneration plant can be allocated by this method according to the following equations:

$$f_Q = \Delta E/E \tag{7.23}$$

$$f_E = (E - \Delta E)/E \tag{7.24}$$

The results in the exergy assessment are comparable with evaluation of the delivered heat, because heat exchange efficiency has the same value as the degree of process quality in the Dresden method [112].

7.1.8 Summary of allocation methods

The above introduced allocation methods are summarized in Table 7.1.

Table 7.1. Allocation methods

Method	Allocation factor heat	Allocation factor electricity
Energy method	$f_Q = \frac{Q}{Q+E}$	$f_E = \frac{E}{Q+E}$
Alternative generation method	$f_Q = \frac{\frac{Q}{\eta_{alt\_heat}}}{\frac{Q}{\eta_{alt\_heat}} + \frac{E}{\eta_{alt\_elec}}}$	$f_E = \frac{\frac{E}{\eta_{alt\_elec}}}{\frac{Q}{\eta_{alt\_heat}} + \frac{E}{\eta_{alt\_elec}}}$
Power bonus method	$f_Q = \frac{f_{P,dh} \cdot Q_{del}}{Q_{del} + E_{del}}$	$f_E = 1 - \frac{f_{P,dh} \cdot Q_{del}}{Q_{del} + E_{del}}$
Exergy method	$f_Q = \frac{Ex_Q}{Ex_Q + Ex_E}$	$f_E = \frac{Ex_E}{Ex_Q + Ex_E}$
200% method	$f_Q = \frac{Q}{2 \cdot Fuel_{in}}$	$f_E = 1 - \frac{Q}{2 \cdot Fuel_{in}}$
PAS 2050	$f_Q = \frac{Q}{n \cdot E + Q}$	$f_E = \frac{n \cdot E}{n \cdot E + Q}$
Dresden method	$f_Q = \frac{\Delta E}{E}$	$f_E = \frac{E - \Delta E}{E}$

7.2. Description of plant model

In this study the analysis was performed on a small-sized DH system with an annual heat load of around 27 GWh. The analysis was based on the heat duration curve marked Case 1, introduced in Section 4.1 and Table 4.1. The energy source for DH was the CCPP with supplementary firing technology. The system consisted of GTC, STC, HRSG, two combustion chambers, fed with natural gas, and other components.

The schematic layout of the system is represented in Fig. 7.1, and the design parameters are summarized in Table 7.2.

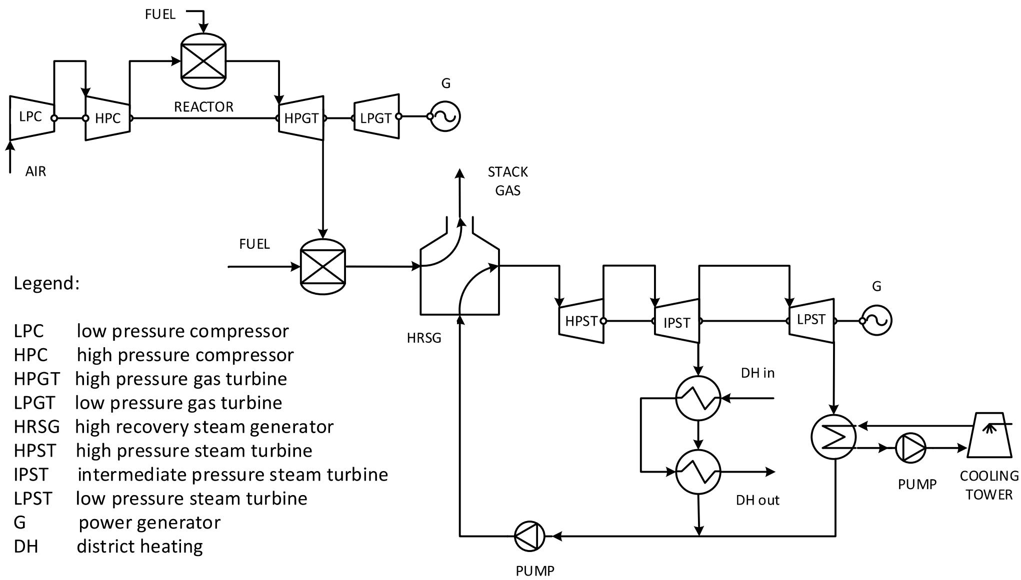

Fig. 7.1. Schematic of CCPP

Table 7.2. Design parameters of CCPP

Parameter	Value
Ambient pressure	101 kPa
Air relative humidity	60 %
Ambient air temperature	+15°C
Pump pressure	60 bar
Steam turbine inlet temperature	+500°C
Condensing pressure	0.05 bar
Air excess in air-fuel mixture	3.2
Fuel temperature	+15°C
Gas turbine adiabatic efficiency	0.9
Steam turbine adiabatic efficiency	0.9
Compressor adiabatic efficiency	0.9
Supplementary firing temperature	+900 °C

7.3. Off-design model assumptions

A number of assumptions were made concerning plant operation in design and off-design conditions. The assumptions were based on a literature study. The following assumptions are common to all the solutions examined:

- for the simplicity of calculation, methane was treated as natural gas;
- no pressure drop in heat exchanger units;
- the plant operates all year;
- the maximum heat demand in DH was equal to 14 MW;
- the electricity grid purchased all the electricity produced in the CCPP;
- heat losses in the system were neglected.

Since the aim of this study was to evaluate the effects of off-design theory on the investigated CCPP plant, Table 7.3. provides a summary of off-design parameters and shows the scale of investigation for each.

Table 7.3. Off-design parameters of CCPP

Parameter	**Value**
Ambient pressure	75 kPa -101 kPa
Air relative humidity	20% - 80%
Ambient air temperature	-20°C - +15°C
Pump pressure	40 bar - 80 bar
Steam turbine inlet temperature	+475°C - +540°C
Condensing pressure	0.05 bar - 0.2 bar
Air excess in air-fuel mixture	3.0 - 4.0
Fuel temperature	+15°C - +200°C
Gas turbine adiabatic efficiency	0.8 - 0.9
Steam turbine adiabatic efficiency	0.8 - 0.9
Compressor adiabatic efficiency	0.8 - 0.9
Supplementary firing temperature	+700°C - +900°C

7.4. Design and off-design system performance

Off-design operational analysis provides valuable information on the operation of the components and system, particularly on its range of applicability. Therefore, it is necessary to analyze the amount of electricity and heat produced by the CHP system, in terms of size, under the part-load characteristics [175]. Fig. 7.2 shows changes observed in CHP behavior.

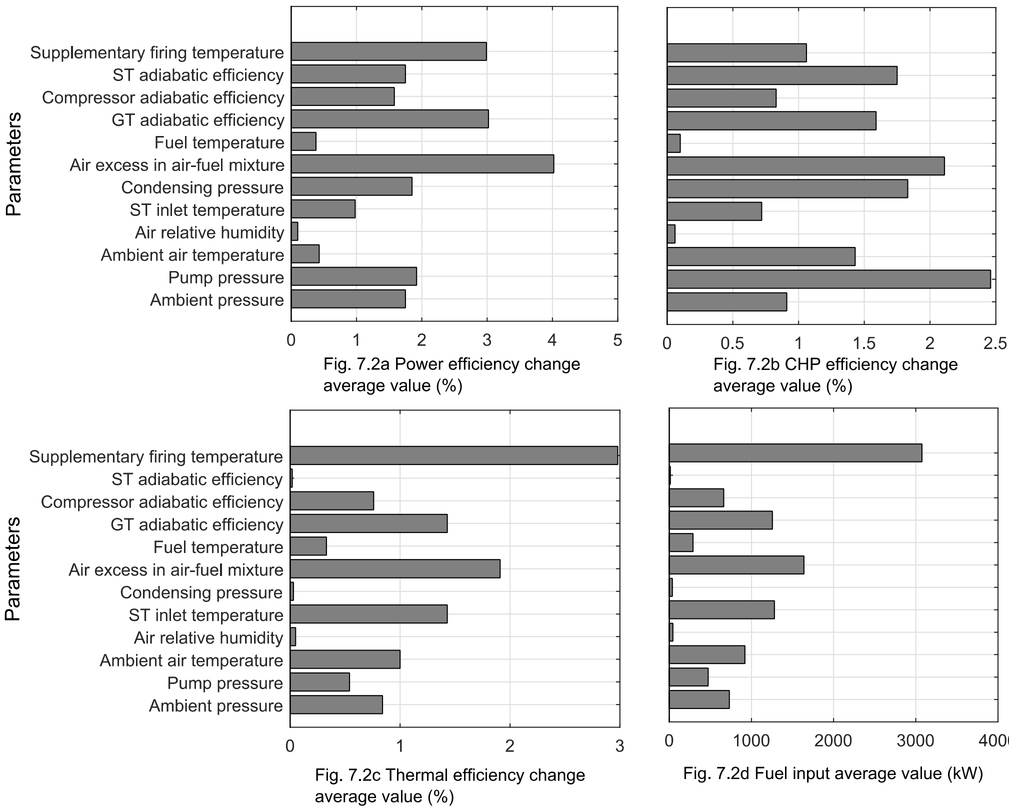

Fig. 7.2. Change in CCPP behavior based on analyzed parameters

The analysis showed that power efficiency, energy efficiency, thermal efficiency, and fuel input varied depending on the analyzed load in the DH system. For example, the obtained values for power efficiency, analyzing the possible range for air excess coefficient ($\alpha = 3.0 - 4.0$), were 27.85% and 32.45% for 14 MW heat load; for 1 MW heat load these values constituted 43.80% and 47.27%. Thus, taking into consideration the entire DH load, the average value for power efficiency change was 4.02 %; see Fig 7.2a. The maximum value for the CHP efficiency change was 2.46%; see Fig 7.2b. In terms of thermal efficiency, the maximum change was 2.98%; see Fig 7.2c. The maximum change in the fuel input rate was 3 075 kW, as shown in Fig 7.2d, due to a change in the supplementary firing temperature.

The thermal efficiency of the CCPP showed the maximum change of 2.98% when the supplementary firing temperature was set to +1000°C. The supplementary firing provided additional energy input to the STC, which resulted in better energy utilization and system flexibility, when shifting from the base load to the high peak. Based on heat flow – temperature diagrams shown in Fig. 7.3 and Fig. 7.4, we can conclude that the high temperature of flue gases before the HRSG does not indicate the best energy utilization.

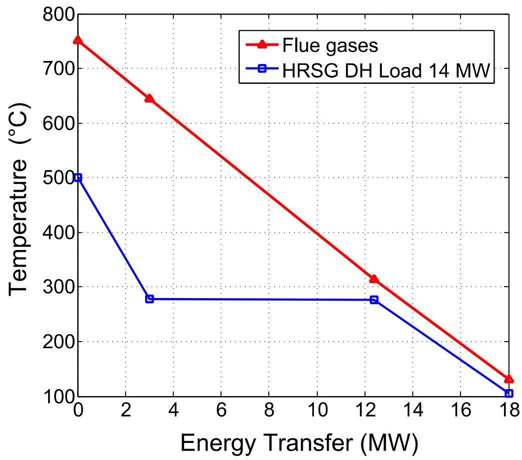

Fig. 7.3. Energy utilization in the HRSG where the temperature of flue gases is +750°C

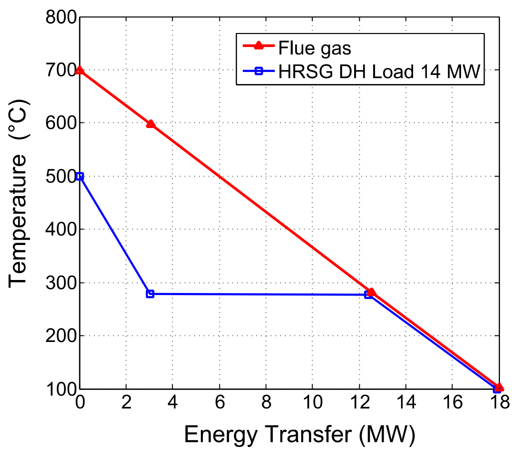

Fig. 7.4. Energy utilization in the HRSG where the temperature of flue gases is +700°C

In general, the temperature of the HRSG should be +200°C higher than the medium leaving temperature of the superheater. The higher temperature at entry provides lower energy utilization in the HRSG and increases the exergy losses. The space between the curves marked in blue and red, as presented in Fig. 7.3 and Fig. 7.4, shows exergy losses. The highest temperature in the HRSG of +900°C had an effect on fuel input in the CCPP. This was the maximum value during simulation and resulted in a change of 3 075 kW of the fuel input; see Fig 7.2d.

The supplementary firing temperature also affected the power efficiency of the CCPP; see Fig 7.2a. This could be explained by increasing the mass flow rate of air-fuel mixture through the GTC.

The minimum change in thermal efficiency occurred due to variation in the following parameters: ST adiabatic efficiency, condensing pressure, and the air relative humidity (RH). This can be seen in Fig 7.2c. The variation in condensing pressure had most effect on power production. The condensing pressure in the CCPP affected the temperature of the water-steam mixture leaving the low pressure steam turbine. The water (compressed liquid) entering the pump before the economizer should not contain any steam fraction; see Fig. 7.1. The water-steam mixture should be fully condensed up to the saturation temperature. This means that the temperature after the low pressure steam turbine remained constant in all cases.

The simulation of the CCPP showed that the operational and design parameters have a significant influence on plant performance. This is valuable information since it is important to provide a reliable heat and power supply to customers, while shifting from the base load to the peak load and vice versa. The change in the operational parameters of the CCPP has a significant influence on the values of the allocation factors due to the different amounts of produced heating and electrical energies.

7.5. Results of allocation factors regarding different operating conditions

The choice of allocation method is more important than the size of the plant, properties of the distribution network, plant technology and even more important than which fuel is used. When analyzing the environmental performance of the CHP, it is important that the reader is aware of the effects related to the allocation method used [142].

In this study different allocation methods have been analyzed in order to investigate the effect of fuel allocation between the heat and the electricity produced in the CCPP. Allocation methods were combined with the parametric studies of the CCPP given in Section 0 and the annual heat energy use at the university campus; see Case 1, introduced in Section 4.2. More details can be found in journal publication I at the end of this thesis. Operating and design parameters were analyzed, and the results were combined to estimate the effect on choice of allocation method. A sensitivity analysis of the different allocation methods was performed for the CCPP under annual heat and electricity loads.

The results presented in Table 7.4 show the values of the CO_2 allocation factors for heat in the design phase.

Table 7.4. Heat allocation factor in the design phase

Method	**Design value Heat allocation factor**
200%	0.0608
Alternative generation	0.3830
Energy method	0.2162
PAS 2050	0.1212
Power bonus method	0.2226
Exergy method	0.1507
Dresden method	0.0834

It might be noticed that different allocation methods produce different results in Table 7.4. For example, the fuel allocation for heat for the alternative generation method was 38.3%, while using the 200% method this value was 6%, and for the power bonus method it was 22.3%. These values could be explained by different calculation methods; see Table 7.1.

Fig. 7.5 presents the effects on allocation factors, when the operation conditions introduced in Table 7.3 and Section 0 were analyzed.

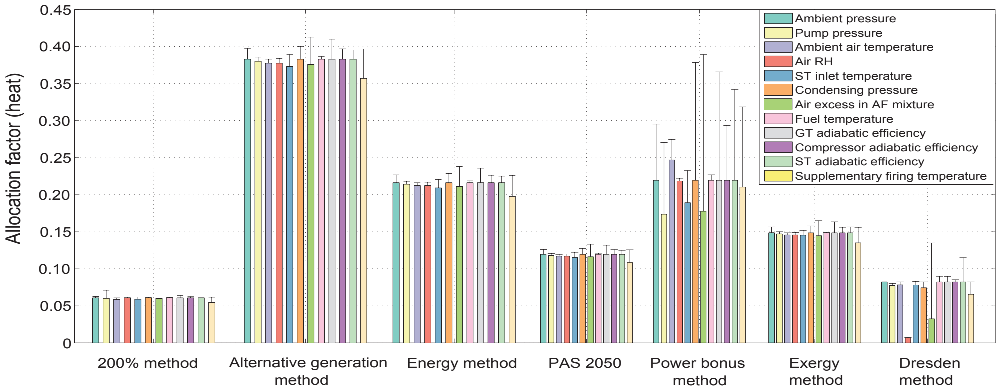

Fig. 7.5. Heat allocation factors for the analyzed methods

The change in heat allocation factors for design and operational conditions showed a small variation. This can be noticed by comparing Table 7.4 and Fig. 7.5. The method which was most sensitive to the change in operation parameters was the power bonus method. The alternative generation method offered the biggest share in the heat allocation, while the smallest share for heat was shown by the 200% method. The heat allocation factor based on the power bonus method changed by 0.16 units due to the change in condensing pressure; see Fig. 7.5. The air excess coefficient in the air-fuel mixture resulted in a change of 0.22 units. The change in the steam turbine adiabatic efficiency and supplementary firing temperature resulted in 0.12 and 0.11 units of heat allocation factor. The changes in the parameters described above have the greatest influence on power production in the CCPP, and hence on the allocation factor based on the power bonus method.

Finally, for the different allocation methods, Fig. 7.6 shows the sensitivity in the allocation factors to heat and electricity production.

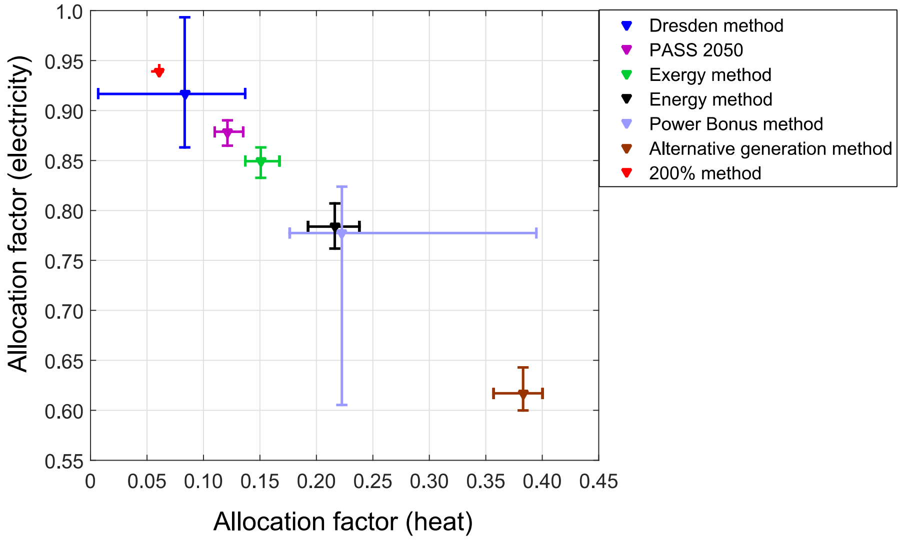

Fig. 7.6. Sensitivity of allocation factors to heat and electricity production

As Fig. 7.5 and Fig. 7.6 show, the power bonus method was the most sensitive compared with other methods. The 200% method showed the smallest change in the analyzed parameters, resulting in a beneficial share of emissions' allocation for DH between heat and power production. The PAS 2050 and exergy methods also had good results and showed that the operational and design parameters did not have a significant influence on allocation factors for both heat and electricity. The change in operation parameters gives a variation in the heat allocation in the CCPP that should be taken into consideration while applying the power bonus method. When the efficiencies of the CCPP vary significantly with load, or are varied to match the demand, the calculated CO_2 emissions are clearly not fixed and could not be constant under any convention. For practical purposes, it would be sensible to define efficiency values, perhaps seasonal averages, as a basis for nominal intensities [176]. As an alternative to the power bonus method, other methods with small variation under variable loads should be considered, such as the 200% method, the PAS 2050 method, or the exergy method. In general, the allocation of the main products is a problematic task, especially in cogeneration systems, since heat and electricity are products of significantly different quality usefulness [100].

The current analysis focused on CHP with CCPP technology. Therefore, the results are relevant for the CHP with the CCPP technology for the same configuration but different operation data. In the case of CHP without supplementary firing technology or gas turbine cycle technology, the final result presented in Fig. 7.6 might be different. Since the CCPP technology has theoretically the highest energy efficiency, the results might be treated as an upper limit for the allocation factors. Thereby, they may be used for energy planning.

7.6. Conclusions on allocation factors

The different methodologies for the allocation of CO_2 emissions for heat and power production in the CCPP have been presented and analyzed. The allocation methods were combined with a parametric study of the CCPP, and this showed that different allocation methods produce different results. For example, the fuel allocation for heat at design conditions for the alternative generation method was 38.3%, while using the 200% method this value was 6%, and for the power bonus method was 22.3%. This indicated that the choice of allocation method is very important for the development of cogeneration technology in relation to heat and power distribution systems. The 200% method gives the lowest CO_2 allocation for heat, indicating that the heat produced in the CCPP is the most environmentally friendly. On the other hand, the alternative generation method allocates a higher amount of emission to heat, which is not beneficial from a DH point of view. Among all the presented methods, the most sensitive was the power bonus method, which is promoted as the main method for emissions' allocation in the EU. The results showed the highest variance in allocation factors for both electricity and heat, ranging from 11% to 21% compared to the design case. In other methods, the variation was negligible: around 1 – 3%. All these indicated that it was difficult to estimate the CO_2 allocation under the annual heat and electricity load variations. Therefore, we can conclude that emissions allocated with the power bonus method cannot be fixed continuously as is stated in standard EN 15316. The solution can be seasonal average efficiency values as a basis for nominal intensities or methods with small variation.

The previous comparisons of different carbon allocation methods in the literature have mostly argued their merits or inadequacies, using fundamental thermodynamic conceptual analysis. This analysis might be criticized for being applicable to only one specific CHP design. However, this criticism is countered by the fact that the resulting allocations are vastly different for the seven investigated methods. This is one of the major conclusions from this study. Further, the results show that the allocation factors for each method are affected by the CCPP design parameters (turbine adiabatic efficiencies, design pressures, excess air, etc.). This shows that the results are applicable not only to a very specific CCPP design, but they also can be used as a generalized tool for CHP applications with different configurations.

This study showed that the decision regarding the choice of the allocation method should be carefully analyzed for implementation in the standards and different policies. It is important to enable a proper allocation of CO_2 emissions and the promotion of environmental benefits from cogeneration technology for DH and power distribution systems.

7.7. Modifications in plant design – Effects of DHs' temperature levels on allocation factors

This section deals with the analysis of the effects of different supply and return temperatures in the DH systems and their effect on allocation factors. The current analysis was implemented in an ethanol-based CHP plant with CCPP technology. The plant configuration is given in Section 7.2 of this thesis.

The current CCPP was ethanol-based; therefore the PEF $f_{P,dh}$ was chosen to be 1.06 (similar to wood shaving), since no proper information was found in [177] for this type of fuel. The decision about PEF was based on Table E.1 of the same document that provided the values for renewable types of fuel like wood shavings and different types of logs.

The results from the power bonus method are presented separately, since they showed a negative trend. As stated in [101], the value can be negative due to high electricity efficiency and it should be replaced by zero. The investigated range of supply and return temperatures was previously introduced in Table 5.2.

7.8. Allocation factors regarding different DH temperatures

In this study, two control strategies were investigated: constant supply temperature control and outdoor temperature compensation control. The details of strategies are discussed in Section 5.1 of this thesis. Fig. 7.7 – Fig. 7.10 present heat allocation factors for the analyzed ethanol-based CCPP.

Fig. 7.7 displays values applying the power bonus method. Results based on a constant temperature strategy in the DH system are marked green, while those with outdoor temperature compensation strategy are yellow. As can be seen, all values in Fig. 7.7 are negative. The following pattern is the result of high electricity efficiency and correspondingly high power production in the CHP plant.

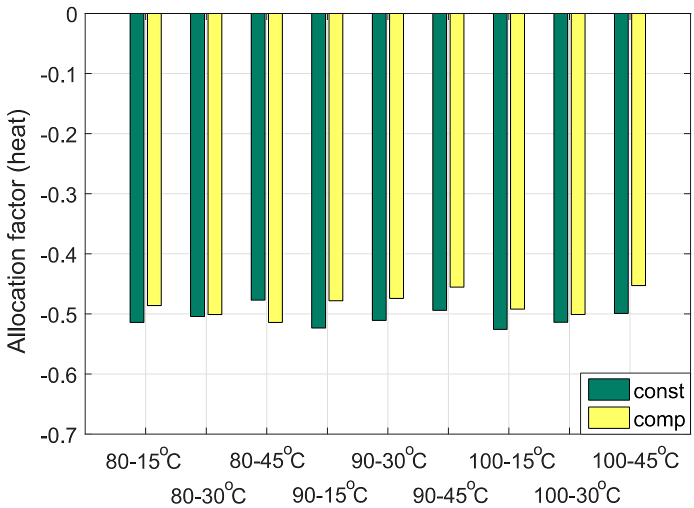

Fig. 7.7. Effect of different temperature levels on power bonus method

In spite of the fact that results from the power bonus method were not similar to those from other allocation methods, some conclusions can be highlighted. From Fig. 7.7 it can be seen that the constant supply strategy allocated more CO_2 emissions to electricity production in comparison with the outdoor temperature compensation strategy. However, these values could be misleading, since they are only valid for renewable fuels such as ethanol or wood shavings. For fossil fuels, the results, most likely, would be positive, and different tendencies could take place. The power bonus method was developed for the promotion of cogeneration technology and the usage of different renewable fuels, and these results confirmed this fact.

Fig. 7.8 provides information on how different supply temperatures affect the choice of allocation method. From Fig. 7.8 it can be seen that different supply temperature levels affected the alternative generation method, energy method, and exergy method. However, for the 200% method, Dresden method, and PAS 2050, the variation was negligible. In comparison with all investigated methods, the exergy method showed the largest variation for all analyzed temperature levels.

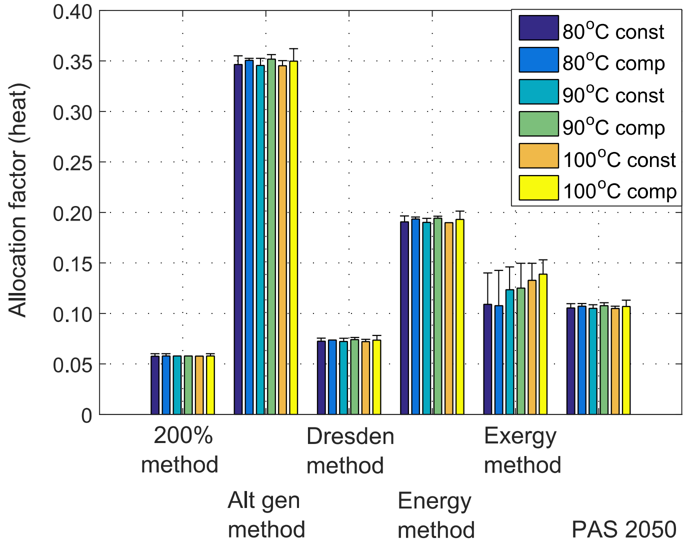

Fig. 7.8. Effect of different supply temperatures in the DH system on heat allocation factor

Fig. 7.8 reveals that lower supply temperatures in the DH system positively affect the heat allocation factor when applying the exergy method. The lowest heat allocation factors were 0.109 and 0.1076 and corresponded to a supply temperature of 80°C. The variation between the lowest and the highest values in the heat allocation factor for the same temperature and constant supply strategy was 27.5%, while with outdoor temperature compensation strategy the difference constituted 31.2%. At the same time for a supply temperature of 100°C, the difference between the lowest and the highest values with constant supply was 10.5% and with outdoor temperature compensation strategy, 18.1%. The Dresden method and the 200% method allocated equally for all the temperature levels and control options. These methods showed the smallest variation in analyzed supply temperature range, which resulted in a beneficial share of emissions' allocation for the heat side. The reason for this is that there is no temperature in the definition of allocation factors.

Fig. 7.9 displays the results of the return temperature level on the choice of allocation method.

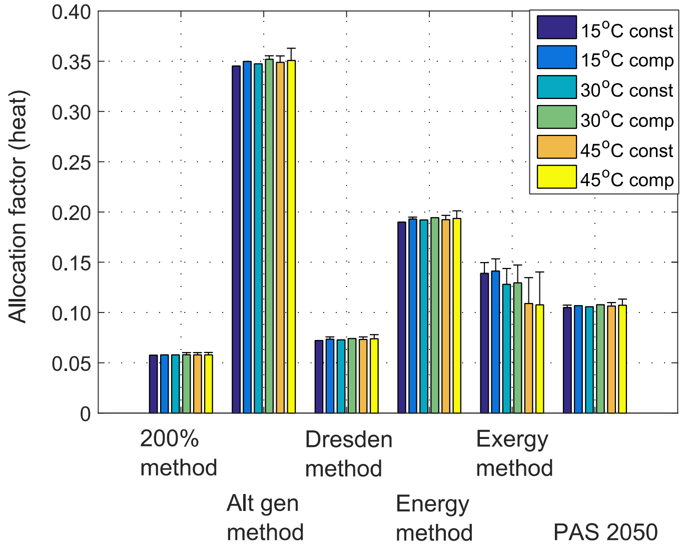

Fig. 7.9. Effect of different return temperatures in the DH system on heat allocation factor

The analysis of the return temperatures indicated similar trends found for supply temperatures; see Fig. 7.8 and Fig. 7.9. The most affected method was the exergy method. However, it was observed that, in order to obtain a lower heat allocation factor, the return temperature should be the highest. The heat allocation factors for return temperature of 45°C were 0.109 and 0.1076, while for 15°C these values were 0.139 and 0.141 for the two control strategies, respectively. At the same time, the difference between the highest and lowest heat allocation for 45°C constituted 21.8% with constant supply strategy and 29.1% with outdoor temperature compensation strategy. For return temperature of 15°C, the difference between the highest and lowest allocation values was 6.84% and 6.94%. The alternative generation method and the energy method showed negligible variations, while the Dresden method, the 200% method, and PAS 2050 method showed that, in spite of the fact that different return temperatures were applied, the heat allocation factor was almost constant. This is a good observation, since these methods are not sensitive to temperature change in the DH system during annual operation.

Further, Fig. 7.10 presents how temperature difference in the supply and return lines of the DH system (ΔT) affects the heat allocation factor. The results shown in Fig. 7.10. are marked red for outdoor temperature compensation strategy and blue for constant temperature strategy.

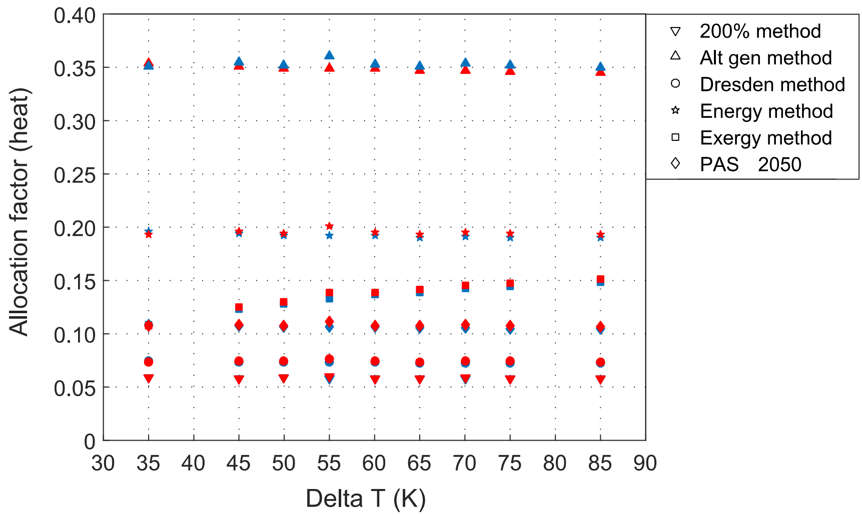

Fig. 7.10. Effect of different ΔT on heat allocation factor

The results found that, for the energy method, alternative generation method, PAS 2050, and exergy method, the control strategy with outdoor temperature compensation showed higher heat allocation factors than when constant supply and return temperature strategy were applied. This means that the outdoor temperature compensation strategy in the DH system allocates more fuel and emissions to heat production in the CCPP. This is a drawback of this control method, since these methods do not promote outdoor compensation control.

Further, it was noticed that, among all methods presented in Fig. 7.10, the exergy method showed the biggest change due to temperature difference. The exergy method showed a tendency to increase the heat allocation factor with an increase in the difference between the supply and return temperatures in the DH network. This observation was valid for both control strategies in the analyzed CCPP. For the outdoor temperature compensation strategy, the variance between ΔT = 35°C and ΔT = 85°C constituted 26.6%, while for constant temperature strategy this value was 21.2%; see Fig. 7.10.

The 200% method did not show any variation in both cases. The values remained constant for all analyzed temperature levels. A similar observation was found for the Dresden method, in which the variation in results was negligible for both strategies. The alternative generation method showed a tendency to decrease the heat allocation factor with the increase in temperature difference. The difference was of 2.45% between ΔT = 35°C and ΔT = 85°C; see Fig. 7.10.

The results for the energy method showed that, applying the constant supply temperature strategy, the value of the heat allocation factor decreased by 3.0%. For the outdoor temperature compensation strategy, the found tendency was negative and less than 1.0 %. For PAS 2050, the outdoor temperature compensation strategy allocated less, 3.3% for ΔT = 85°C, than for ΔT = 35°C. In the case of constant supply temperature, the tendency was less than 1.0% during the whole temperature interval.

As was previously discussed, the different methodologies produce different allocation results. It was found that the exergy method was the most sensitive to different temperature levels in the DH system. The results showed that, with the decrease in supply temperature, the heat allocation factor also decreases, while the opposite situation was the case for the return temperatures. The analysis of the return temperature levels found that the lowest allocation factor for heat was provided by a temperature of 45°C and the highest by 15°C. The analysis revealed that, with the increase in ΔT between the supply and return temperatures in the DH system, the heat allocation factor applying the exergy method had a tendency to increase. The opposite tendency was found for the alternative generation method and energy method. However, these values were not very steep in comparison with the exergy method.

For all the analyzed methods, the outdoor compensation strategy showed higher values than the constant supply strategy. The Dresden method, the 200% method, and the energy method showed that different temperature levels have the smallest effect on the heat allocation factor. This is a good observation, since it shows that these methods will not be affected by yearly temperature variation that could occur in the DH system. The 200% method allocated the lowest amount of CO_2 to the heat, produced in the CCPP, indicating that it is the most environmentally friendly method, from the DH point of view.

The power bonus method showed negative results, since renewable fuel was used.

The study showed that, for those who apply the exergy method, attention should be paid to temperature levels in the DH system, since the heat allocation factor will affect the competiveness of the DH business and future development of the DH system. For the DH customers, the increase in the heat allocation factor could affect the price development and this, in turn, could lead to shifting to other technologies such as HPs, which is not beneficial for DH companies.

Currently it seems that only the exergy method can properly manage temperature levels. However, the exergy method does not motivate a high temperature difference, which is necessary for the DH transition to the LTDH.

8. RELIABILITY ISSUES

This chapter deals with the security of supply in the DH systems. The aim was to define the underpinning data necessary to understand and describe the reliability of the DH network. The main idea behind this chapter was to estimate the reliability of the heat supply in a DH network. However, due to limited and fragmented historical records of pipe accidents and maintenance information for the DH network, the study was not put into execution. The information acquisition showed that DH companies do not report and organize the necessary data on pipe and maintenance in a systematic way that would enable a good analysis. Further, statistical data on heat losses, delivered heat, and DH network properties provided by the branch organizations and at the national level are usually not coherent and differ a lot from each other. The main reason for this is that there is no generic system to report data on the DH network operation, such as temperature levels, pressure levels, frequencies of accidents at specific conditions, delivered heat, age of pipes, type of pipes, etc. It was found that the quality of the data on DH network operation differs from country to county. Frequently, different DH organizations do not follow strict requirements associated with data collection, which in turn leads to poor data quality and fragmented data.

Recently, the IEA DHC Annex X report was released. The project aimed to develop methods and tools for the estimation of the technical lifetime of pre-insulated bonded DH pipes in operation. However, the fundamental information about causes leading to pipe deterioration processes was missing. Therefore, the literature review presented in this section and the identification of important data about DH pipe reliability are a valuable contribution to this report. Further, the methodology for the calculation of reliability indexes and suggestions for a suitable database structure are given at the end of this chapter. This chapter is based on conference paper V, attached at the end of this thesis.

8.1. Factors affecting pipe reliability

The main causes leading to pipe deterioration were identified by different authors in [178-180]. Al-Barqawi and Zayed in [181] acknowledged the factors resulting in pipe degradation. Three groups were identified and classified as shown in Table 8.1.

Table 8.1. Factors leading to water system deterioration

Physical factors	**Environmental factors**	**Operational factors**
Pipe age and material	Pipe bedding	Internal water pressure
Pipe wall thickness	Trench backfill	Leakage
Pipe vintage	Soil type	Water quality
Pipe diameter	Groundwater	Flow velocity
Type of joints	Climate	Backflow potential
Thrust restraint	Pipe location	Operational and maintenance practices
Pipe lining and coating	Disturbances	
Dissimilar metals	Stray electrical currents	
Pipe installation	Seismic activity	
Pipe manufacture		

Many of the factors listed in Table 1 are not readily measurable or quantifiable. Physical mechanisms that lead to pipe breakage are often very complex. Moreover, the quantitative relationships between these factors and pipe failure are often not completely understood [182, 183]. The most commonly assumed factors for a DH distribution network are described below.

Age and installation period. DH technology was first put forward in the middle of the 19th century in the US and in the early 20th century in Europe. The differing launch periods for DH systems in Europe indicate that the pipes used in the systems could have been installed at different periods. In France, for example, first generation DH systems with steam as a carrier are still in use. It has been found that the construction period can affect pipe durability [184]. Different types of pipes are used and sometimes older pipes are less likely to fail than their younger counterparts. Further, only several pipes can remain under the ground after four repairs, since pipe records are unable to provide accurate ages of pipes [185]. When the age of piping exceeds 30 years, the frequency of damage increases, and the technical conditions of the pipeline become the main reason for the formation of defects [186].

Corrosion is the main reason for pipe replacements [187] and structural deterioration [188]. As indicated in [186], the failures of pipelines due to corrosion mechanisms constitute 30% – 40% of all damage to these pipelines. Metal pipe corrosion pitting is a continuous and variable process. Under certain environmental conditions, metal pipes can become corroded based on the properties of the pipe, soil, liquids and stray electric currents [189]. Corrosion deterioration mechanisms can be divided into two types: internal and external. Internal corrosion is caused by different characteristics of transported water. Poor water quality can cause internal corrosion of the pipelines and substations and may block and weaken the functioning of the controlling and metering devices in the entire DH system [190]. Different levels of water pH-value, oxygen content or bacteria can be the reasons for this process. Meanwhile, external corrosion occurs with pipes sensitive to soil composition, moisture and aeration. These can be described as aggressive environmental conditions. Corrosion occurs at apparently random locations on a pipe [191], weakening it by decreasing the material's thickness and by creating stress concentrations [192]. However, not all pipes used in DH are exposed to corrosion. For example, flexible pipes [1] made of polymer material such as PolyEthylene (PE) are corrosion resistant. Nevertheless, they are sensitive to the high temperatures used in DH systems. Therefore, manufacturers normally limit the maximum supply temperatures in order to extend the pipe's service life.

If the service pipe fails, the escaping hot water may cause extensive damage to the surrounding area. Repairs will then include replacing part of the service pipe, resulting in an interruption to service, which will leave end users without a hot water supply. Aside from the inconvenience caused to end users, the repair costs will be high. This is especially so in the case of larger diameter pipes [193].

Diameter. The failures associated with pipe diameters can be explained by the thickness of pipes. Small pipes have lower wall thickness, resulting in reduced pipe strength. The high probability of pipe breakage was found in a network of small pipes [194]. In the study related to damage caused by earthquakes [192], pipe diameter was identified as an influence on the number of breaks and failures in pipelines. Pipes with small diameters experienced more damage than those with large diameters. The influence of pipe diameter determines the total area from which a failure point may occur. The pipe may be thought of as being developed from a simple plate of area πDL [195]. Smaller pipe sizes, of lesser safety significance, have much higher failure rates [196].

Pipe length. The dependency of failure probability on pipe length was also acknowledged [195]. The failure probability increases directly in proportion to length; for example, a 1.0 m length of pipe bears a 10-times greater failure probability than a 0.1 m pipe. This assumption was based on uniform distribution of weak spots along the pipe. The number of weak spots such as bends, junctions, welds, and flaws increases proportionally with the increase in pipe length.

Pipe material. It is of no surprise that pipe service life is dependent on the pipe material used for hot water distribution. A study of the historical development of DH systems in the world indicates that the use of different types of distribution pipes was due to different materials being available during certain time periods. DH distribution technology can be classified in three generations [1]; the first generation of DH technology used steam as a carrier, while the second and third generations employed water as a carrier to deliver heat. Nowadays, the temperature level in DH networks has decreased; however, systems based on first generation principles are still in operation. Therefore, it is possible to distinguish the different pipe types used in DH systems. The traditional metal for pipes is steel. Pre-insulated rigid steel pipes have the largest share in DH systems, and a number of publications are devoted to this type of pipe [197, 198]. New developments in the DH field have introduced pre-insulated rigid polymer pipes and pre-insulated flexible PE pipes with a life span of more than 30 years [199]. At the same time, copper pipes are also in use in customer substations [200].

Dissimilar metals. In general, dissimilar metals could be employed when DH systems of different ages are connected. Welded dissimilar metal joints can have a remarkable effect on the plant’s availability and safety and lead to leakages and pipe cracking [201]. Dissimilar metal joints can be installed in pipes with large diameters. The study devoted to the safety of nuclear plants found that the probability of cracked welding occurring is fairly high [202].

Seasonal variation. Accidents mainly occur in DH systems during the. The main reason is that the largest heat demand associated with DH occurs at this time of the year, while extreme outdoor temperatures weaken the pipes, particularly if they are exposed to the cold without adequate insulation.

Soil conditions affect external corrosion rates and play an important role in pipe degradation [203]. The rate of corrosion is affected by the properties of the soil, in particular its pH content, redox potential, the existence of sulfides, and the water resistance table [204], and by the soil type: clay, sand or peat soil [205]. This is relevant for DH pipes of the third generation of distribution technology [1], which are buried in the ground.

Previous failures. The number of previous failures is a significant factor in predicting future failures [206]. Pipes in the same location often have the same age and materials and are laid with the same construction and joining methods. Pipes in the same location are also likely to be exposed to the same external and internal corrosion conditions [203].

Nearby excavation, together with seismic activities, affects pipe integrity. A detailed description of these processes was provided in [207, 208]. Contact during excavation usually occurs when an individual piece of operational excavating equipment breaks the pipe. It was found that the risk due to corrosion is significantly less than the risk due to third-party intervention [178, 205]; research work carried out in the UK showed that third-party activities have a high probability of causing pipe failure [209].

Pressure. The pressure change due to breakage depends on the ratio of flow rate through the pipe versus the flow rate through the break. If the loss of water is very small, compared with the mainstream through the pipe, then pressure fluctuations due to the break would be negligible [210]. The possibilities of pipe ruptures due to high pressure in the water distribution networks, together with the hammer effect, were acknowledged in [211]. The analysis found a high risk of pipe damage that leads to water supply interruptions. Another analysis of the DH network was executed applying RELAP5 code. RELAP5 is a generic transient analysis code for thermal-hydraulic systems using a fluid that may be a mixture of steam, water, noncondensables, and a nonvolatile solute [212]. The analysis on the DH network found that, under some conditions, pressure peak could exceed the value used in hydraulic tests, but the possibility of pipe damage still remained high [213]. The pressure head in the distribution network has a direct influence on the frequency of pipe failure [214]. Further, water hammering due to an immediate change in the velocity of the carrier in DH systems directly affects joints [1, 178]. This situation can occur when distribution pumps tend to be opened rapidly against the closing of valves located on pump outages or when predefined valve opening time is ignored [1]. The researchers in [215] identified the possibility of pipe ruptures depending on operational pressure in a DH network. According to the results, the rupture probability increases linearly up to a pipe thickness of 3 mm. The plastic deformation will occur from thicknesses of 3 mm to 1 mm. In the case of thickness being less than 1 mm, the rupture probability increases rapidly.

Land use. Traffic areas, residential areas and commercial areas are used as a substitute for external loads on pipes [203]. The stresses occurring in DH pipes are complex and originate from a number of sources, including soil loading, ground surface loading (due to traffic) and as a result of temperature changes. The pipe failure occurs either when the stress level exceeds the nominal pipe material strength or when a critical defect develops and leads to degraded pipe strength [191].

Temperature levels. Improper temperature levels used in DH systems can cause mechanical stresses and thermal strain in distribution pipes. When a carrier pipe is made of steel, which is always the case with rigid pipes, normal operation temperatures are too low to cause any significant creep deformation. However, when carrier pipes are made of polymeric material, as they sometimes are in flexible pipes, creep and thermal expansion are the major issues [1]. Thus, when designing and installing a buried pipeline, which is both pressure- and temperature loaded, one should pay special attention to the extreme temperature variations and hence to the stresses and movements that the pipeline will have to withstand [216]. The joints are highly affected by mechanical stresses due to the large temperature differences whenever the distribution network is in operation or shut off [217]. Temperature fatigue, occurring due to high temperature levels in the DH system, results in high failure probabilities [218, 219]. Moreover, different pipe manufacturers limit the maximum supply temperature used in DH systems to 120 °C.

Welding. According to the study [220] performed by EuroHeat & Power, the largest number of failures in polymeric DH pipes occurs in joints. One of the reasons is bad welding procedure. The annual frequency of failure due to the joints of the outer tube is the highest, compared with damage caused by medium tube joints and damage found by the leakage detection system [217].

8.2. Suggestions for comprehensive DH pipe database

As can be seen, there are many different factors affecting the DH pipe integrity. In order to stay competitive in the market and provide a reliable service to customers, the reliability issues should be carefully analyzed. However, there is a lack of sufficient and well organized data on this issue. For this reason, it would be wise to collect relevant information about accidents associated with the DH distribution system. As was mentioned, DH technology has been developed and extended over a long time; therefore, collecting all the data is challenging. Nevertheless, due to new technology and its development, the DH companies can start to develop much better knowledge-based databases.

Fig. 8.1 shows a suggested database structure.

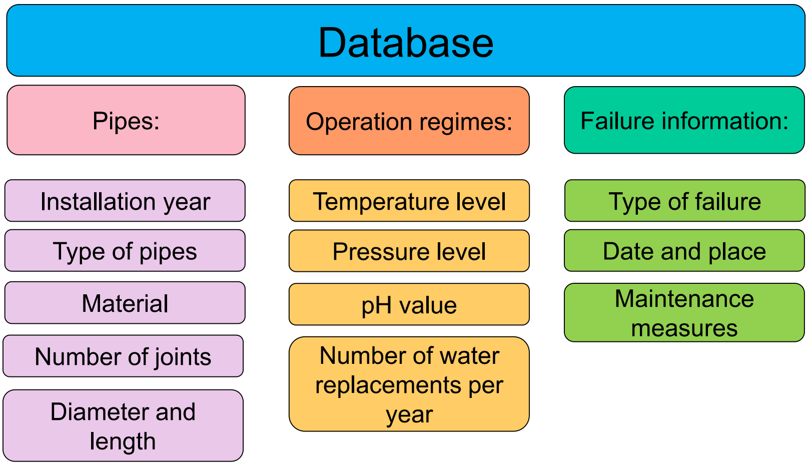

Fig. 8.1. Suggested database structure

A comprehensive database has to include different information that allows pipe failure accidents to be predicted by analytical and statistical methods. Hereafter, a good database should include several categories of information: first, information about pipes: installation year, type of pipes and material, diameters, length of pipe section, number of joints; next, the operating regimes of the DH system, e.g. temperature and pressure levels, pH-value of the heat carrier, number of water replacements during the year; the final part should include: type of failure, date and place, and maintenance measures. The structure presented in Fig. 8.1 is simple to follow, understand, and search for relevant information. It can easily be extended and developed into a more sophisticated information source.

8.3. Statistical method for pipe reliability calculation

This section provides information on how to perform a reliability calculation of the DH network based on a statistical methodology. The presented method allows the probability density function, pipe failure rate and unreliability function to be calculated, and, further, it allows conclusions to be drawn about network reliability.

The pipe failure rate denotes the rate of failure per time unit. The probability that an item will fail in the interval $(t, t + \Delta t]$ when the item was functioning at the time t, can be defined based on the following equation [221]:

$$\Pr(t \leq T + \Delta t \mid T > t) = \frac{\Pr(t < T \leq t + \Delta t)}{\Pr(T > t)} = \frac{F(t + \Delta t) - F(t)}{R(t)} \tag{8.1}$$

where, T is a time to failure, Δt is a time interval, t is a time when the unit was put in operation. It is assumed that, at the beginning of the deterioration process, t = 0, while, $R(t)$ is a reliability function of an item.

This can be rewritten as a failure that occurs during operational time in the DH system and has the following form:

$$Failure\ rate = \frac{Total\ failure\ occurence}{Total\ operation\ time} \tag{8.2}$$

The unreliability function expresses the period of time when the DH system is not able to perform its main function to deliver heat to the customers. This function can be expressed as:

$$F(t) = \Pr(T \leq t) = 1 - R(t) \tag{8.3}$$

Based on unreliability function $F(t)$, presented in Eq. (8.3), we can find the failure probability density function $f(t)$:

$$f(t) = \frac{d}{dt} \cdot F(t) = \frac{d}{dt} \cdot (1 - R(t)) \tag{8.4}$$

Further, the failure rate function, $\lambda(t)$, can be estimated based on probability density function $f(t)$ and reliability function $R(t)$ from Eq. (8.2) and Eq. (8.4):

$$\lambda(t) = \frac{f(t)}{R(t)} = \frac{(-dR(t)/dt)}{R(t)} \tag{8.5}$$

8.4. Conclusions

Unfortunately, it is quite difficult to find good literature on the DH pipe reliability issue. Research on this topic is quite rare and requires profound knowledge in reliability theory. Further, a comprehensive database is required for the analysis.

This chapter intended to shed light on the factors affecting DH pipe reliability. The collected information leads to a better understanding of pipe degradation mechanisms and can be used as a tool for pipe failure analysis. In addition, for proper operation of the DH systems, it is highly desirable to collect the maintenance information about pipe accidents. A comprehensive database allows pipe failure accidents to be predicted by the analytical and statistical methods. A good database can provide an instant start for distribution system analysis and help in pipe model creation based on statistical data.

The failure rate function presented in Eq. (8.5) should be calculated for each DH system taken from the database introduced in Fig. 8.1. The failure rate function shows the extent of system wear and tear. This value should be used for the prevention of accidents and to increase the security of supply.

9. CONCLUSIONS AND FUTURE WORK

The main objective of this work was to look into the issues associated with the energy planning of DH systems when different and mostly renewable energy sources were included. The possibilities of decreasing temperature levels in existing DH systems and transition to LTDH were studied.

The studies presented in this thesis were performed with the help of commercially available process simulation software, Aspen HYSYS. The process simulation was employed for model development and investigation of the behavior of energy supply sources. Later on, the found results were thoroughly analyzed and the major findings presented.

In the following section, the most important observations from the executed work are delivered; suggestions for future work are also provided. Conclusions in greater detail can be found at the end of each chapter and in the journal and conference articles.

9.1. Main conclusions

The results on the ethanol-based CCPP showed that power production in the ethanol driven CCPP was not influenced significantly by supply temperature control. The analysis of the change in the DH load showed that average heat efficiency was highest for the uniform distribution load and lowest for very non-uniform loads. The average power efficiency was dependent on different temperature levels in the supply and return lines of the DH system. This indicated that a large temperature difference between the supply and return lines of the DH system resulted in higher power production in the CCPP. This is very beneficial for the transition of the DH systems to the low temperature systems or fourth generation systems. This means that ethanol-based CCPP should be a good renewable energy source for future DH systems. In addition, the results found that a decrease in supply temperature had a low impact on energy efficiency.

Another important conclusion is that the CCPP performance indicators are highly dependent on the heat load distribution in the DH system during the year. However, the analysis of all the CCPP performance indicators versus the DH load showed negligible variation for all the temperature levels applied in the DH system. The change in the overall fuel energy input showed that fuel use increases with the increase in temperature difference between supply and return lines in the DH system.

Further, the found results point out an inevitable decrease in plant profitability, while operating the CCPP under low and non-uniform heat demand profiles. This observation provides incentives to shut down the heat supply to DH systems and run CCPP at full load, producing as much electricity as possible. Low energy building stock should be connected to specially designed low-grade temperature sources under a prepared infrastructure. However, CCPP could be used if low energy buildings were located close to each other to increase the heat density. The CCPP could also be used during the peak energy demand. This will have a positive result for plant operation, since the CCPP will operate at its maximum heat load output, increasing its performance indicators.

A new method of heat energy planning was suggested instead of the existing method of heat supply optimization. The new method implied employment of plant performance data rather than constant efficiencies, as in the existing method.

The analysis of energy planning of future building stock including renewables showed that that the operation of an electric boiler was quite expensive and should be kept to the minimum. In addition, policy makers should provide a legislative framework to exclude this technology from DH. The study identified eight PCs with low sensitivity to heat load variation and LCOE under 0.15 EUR/kWh. However, the uncertainty in PCs due to changes in investment cost leads to significant changes in LCOE values for these technologies.

It is found to be simple to manage all the costs and operation issues in the existing method for energy planning of heat supply optimization. A number of additional important factors affecting plant operation are missing. It was found that the method is sensitive to change in heat load profiles and variation in investment cost. In addition, it is very simplified with respect to real sizes, operation times, and actual heat capacities. In comparison with the existing method of heat supply optimization, the new method suggested in this thesis is more sophisticated and involves deeper analysis.

In the suggested new method for energy planning, the fuel was allocated between heat and power production by the energy method. In turn, this made the CHP operation highly efficient due to the subsidizing of low heat load by electricity load and further fuel allocation to power production. This showed that CHP operation is efficient as a peak load plant. However, a number of technical allocation methods have been developed and used in different countries. Therefore, possible deviations in LCOE might occur due to the application of different allocation methods.

This study showed that an increase in the DH's flexibility and reliability of supply increased heat generation cost.

The yearly operation of a natural gas driven CCPP plant was considered in order to investigate the sensitivity of existing technical allocation methods. As was expected, the different allocation methods produced different results. This demonstrated that the choice of allocation method is very important for the development of cogeneration technology in relation to heat and power distribution systems.

Among all the presented methods, the most sensitive was the power bonus method, which is promoted as the main method for emissions' allocation in the EU. Due to the high sensitivity of this method, the estimation of CO_2 allocation under the annual heat and electricity load variations becomes more complex and uncertain. Therefore, it can be concluded that emissions allocated with the power bonus method cannot be fixed continuously, as is stated in standard EN 15316.

The performed study on allocation factors found that the exergy method was the most sensitive due to changes in temperature levels in the DH system. The results showed that, with the decrease of the supply temperature, the heat allocation factor decreases. However, for the return temperatures, the opposite pattern was observed. Further, the analysis revealed that, with the increase in temperature difference between supply and return temperatures in the DH system, the heat allocation factor applying the exergy method had a tendency towards increase. The opposite tendency was the case for the alternative generation method and the energy method. However, these tendencies were not steep in comparison with the exergy method. For all analyzed methods, the outdoor compensation strategy showed higher values than the constant supply strategy. Currently, it seems that only the exergy method can properly manage temperature levels. However, the exergy method does not motivate high temperature difference, which is necessary for the DH transition to LTDH.

The literature review of different factors leading to premature breakup of the distribution network showed that it is very important to be aware of existing degradation mechanisms and prevent them in good time. Hence, it is desirable to collect the maintenance information about pipe accidents in the DH systems. Therefore, a comprehensive database has to include information that allows pipe failure accidents to be predicted with the analytical and statistical methods. A good database can provide an instant start for the analysis of the distribution system, help in pipe model creation based on statistical data, lead to accident prevention and increase security of supply. Currently, it is not possible to find all the necessary data. Therefore, in this work a literature review alone was made on this issue.

9.2. Suggestions for future work

Chapter 6 of this thesis debated different plant combinations depending on their heat outputs, technical, and economic constraints. As discussed, there are a number of pitfalls, which need to be considered. However, finding a suitable solution is only half of the issue. Proper operation of the chosen units is of high importance. Therefore, it would be of interest to make a research on the optimized operation of these energy units. The optimal scheduling of energy generation depending on heat requirements is a key to effectiveness and profitability. The optimization should deal with customers' profiles based on short-term planning and consider hydronic balancing of applicable systems. In addition, the applicability of thermal energy storage should be investigated.

Further, this thesis focused on a detailed simulation of energy production units, considering the operational effects of the DH system. However, for better understanding, the interactions between the DH's constituent elements need to be extended. As it is known, the DH system consists of four main components: energy production unit, distribution network, customers' heat load and customers' substation. The need for a detailed simulation model, which includes all four elements, is crucial for understanding the interactions between customers, production of heat, and heat delivery. In turn, a mismatch between customer needs and DH providers can be eliminated. As an extension of this idea, the interaction between heat and power production should be examined. Quite often, there is a discrepancy between productions of these two goods. Day and night tariffs, peak and off-peak intervals, weather dependency, technical limitations, customer loads: all these make the production mechanisms complex. At this point, it is desirable to look into the generation of power and heat as an integrated mechanism. Perhaps, case studies should also be included.

Another area relevant for future research deals with the reliability issues. As was discussed, the reliability of the DH systems is of high importance. This statement is valid for both DH companies and customers. One of the ideas behind the current thesis was the implementation of a reliability study of the distribution network. The methodology is described and presented in the corresponding section of the current thesis. However, due to lack of historical data about pipe accidents, this research was not put into execution. As can be seen, the necessary information should be handled with the help of companies involved in the DH business. Then, the study can be performed and further developed in different research topics.

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ERRATA LIST

Page 53 – 56	Fig. 6.3. ~~Dependence of investment cost versus heat rate for HP~~ HP's investment cost versus heat rate
	Fig. 6.4. ~~Dependence of investment cost versus heat rate for HP~~ Electric boiler's investment cost versus heat rate
	Fig. 6.7. ~~Dependence of efficiency versus heat rate for HP~~ HP's COP versus heat rate
	Fig. 6.8. ~~Dependence of efficiency versus heat rate for electric boiler~~ Electric boiler's efficiency versus heat rate
Page 73	Fig. 6.20. ~~The linear cost characteristics for three plant model is shown in the upper diagram and the corresponding optimal division of plant capacities are shown in the lower duration diagram~~ Duration diagram showing linear cost characteristics for three plant models (upper diagram) and corresponding optimal division of plant capacities (lower diagram)
Page 82	Fig. 6.27. ~~Primary energy use MWh/MWh_heat~~ Energy input/delivered heat
Page 86	Improved introduction to Chapter 7
Page 112	Improved introduction to Chapter 8

Appendix I – Cost data for energy supply technologies

Table A1- Table A4 provide a summary of different costs for the following technologies: biomass HOB, CHP, HP and electric boiler. The data is presented based on the LHV of fuels.

Table A1 Investment and O&M costs for biomass HOB

Heat output	**Efficiency (%)**	**Investment costs (MEUR/MW)**	**Fixed O&M cost (EUR/kW)**	**Variable O&M cost**	**Reference**
1 MW	108	0.5	Total O&M 5.4 EUR/MWh		[47]
5 MW	108	0.75	Total O&M 5.4 EUR/MWh		[47]
5 MW	88	0.29	Total O&M 278180 EUR		[143]
5.8 MW	90	0.82	Operational costs 1110 kEUR/year		[222]
10.3 MW	110	0.4	2 EUR/MWhfuel	2 EUR/MWhfuel	[223]
12 MW	108	0.5	10	2 EUR/MWhfuel	[67]
12 MW	108	1.1	Total O&M 5.4 EUR/MWh		[47]
28.5 MW	110	0.36	2 EUR/MWhfuel	2 EUR/MWhfuel	[223]
50 MW	108	0.42	8.3	2 EUR/MWhfuel	[224]
200 MW	101	0.09	3.3	-	[225]
400 MW	110	0.33	2 EUR/MWhfuel	2 EUR/MWhfuel	[223]

Appendix I – Cost data for energy supply technologies

Table A2 Investment and O&M costs for biomass CHP

Heat/power output	Efficiency (%)	Investment costs (MEUR/MW)	Fixed O&M cost (EUR/kW)	Variable O&M cost	Reference
1 MW	heat - 78 electric - 25	3.6 of heat	3-4 % of investment per year		[47]
5 MW	heat - 78 electric - 25	4.64 of heat	3-4% of investment per year		[47]
5 MW	total - 90	6.0 of heat	Total O&M 0.055 EUR/kWh		[93]
0.5 MWel 5.5 MWheat	electric - 18 total - 83	0.56 of heat 4.71 of electric	0.128 EUR/kWel 0.0367 EUR/kWheat		[226, 227]
1.0 MWel 5.8 Mwheat	heat - 65 electric - 24	4.2 of electric 0.4 of heat	Total O&M 0.032 EUR/kW		[222]
10.3 MW	electric - 25 total - 105	3.9 of electric	2 EUR/MWhfuel	2.6 EUR/MWhfuel	[223]
10 MW	heat - 78 electric - 25	4.9 of heat	3-4% of investment per year		[47]
17 MW	heat - 81 electric - 24	1.85 of heat	41	2.4 EUR/MWhfuel	[67]
5 MWel 18 MWheat	electric - 22 total - 104	6.49 of electric	157	2.3 EUR/MWhfuel	[162]
28.5 MW	electric - 27 total - 110	2.3 of electric	2 EUR/MWhfuel	2.6 EUR/MWhfuel	[223]
30 MW	heat - 77 electric - 29	2.6 of heat	29	3.9 EUR/MWh	[47]
30 MW	heat - 79.5 electric - 26.5	1.72 of heat	35.2	2.9 EUR/MWhfuel	[224]
10 MWel 28 MWheat	electric – 27 total - 105	5.15 of electric	116	2.3 EUR/MWhfuel	[162]
50 MW	heat - 81 electric - 29	1.68 of heat	34	kEUR/MW year 24.1	[225]
80 MW	electric - 30 total - 110	1.7 of electric	2 EUR/MWhfuel	2.6 EUR/MWhfuel	[223]
81 MW	heat - 81 electric - 29	1.47 of heat	24.8	3 EUR/MWhfuel	[224]
30 MWel 75 MWheat	heat - 60 electric - 30	3.0 of electric	2.1 EUR/MWhfuel	2.5 EUR/MWhfuel	[228]
30 MWel 76 MWheat	electric - 28 total - 105	4.06 of electric	77	2.3 EUR/MWhfuel	[162]
36 MWel 72 MWheat	electric - 30 heat - 60	1.5 of electric	37 EUR/kWel	4.5 EUR/MWh el	[229]
199 MW	heat - 77 electric - 31	1.18 of heat	17.6	3.1 EUR/MWhfuel	[224]

80 MWel 195 MWheat	electric - 31 total - 106	3.23 of electric	55	2.3 EUR/MWhfuel	[162]
479 MW	electric - 34 total - 110	1.3 of electric	2 EUR/MWhfuel	2.6 EUR/MWhfuel	[223]

Table A3 Investment and O&M costs for HP

Heat output	**COP**	**Investment costs (MEUR/MW)**	**Fixed O&M cost (EUR/kW)**	**Variable O&M cost**	**Reference**
1 MW	3.2	0.51	4.2 EUR/kW		[47]
3 MW	3.2	0.67	5.9 EUR/kW		[47]
5 MW	3.2	0.79	7.3 EUR/kW		[47]
5 MW	3.3	0.7	7.0 EUR/kW		[225]
10 MW	3.2	0.6	0.5	0.7 EUR/MWhfuel	[223]
10 MW	2.8	0.52	3.7	0.2 EUR/MWhfuel	[67]
11.2 MW	3.0	0.21	8.9 EUR/kW		[230]

Table A4 Investment and O&M costs for electric boiler

Technology	**Efficiency (%)**	**Investment costs (MEUR/MW)**	**Fixed O&M cost (EUR/kW)**	**Variable O&M cost**	**Reference**
1 - 3 MW	electric - 99	0.14	1.1	0.5 EUR/MWh	[47]
10 MW	electric - 99	0.08	1.1	0.5 EUR/MWh	[47]
20 MW	electric - 99	0.06	1.1	0.5 EUR/MWh	[47]

Appendix II – Papers

Paper I Tereshchenko T, Nord N. Uncertainty of the allocation factors of heat and electricity production of combined cycle power plant. *Applied Thermal Engineering*, 2015. 76(0): p. 410-422.

Paper II Tereshchenko T, Nord N. Implementation of CCPP for energy supply of future building stock. *Applied Energy*, 2015. 155(0): p. 753-765.

Paper III Tereshchenko T, Nord N. Energy planning of district heating for future building stock based on renewable energies and increasing supply flexibility. *Elsevier Energy*. (Accepted).

Paper IV Tereshchenko T, Nord N. The allocation factors of heat and electricity production of combined cycle power plant. *The 9th Conference on Sustainable Development of Energy, Water and Environment Systems*. September 20 - 27, 2014, Venice-Istanbul.

Paper V Tereshchenko T, Nord N. *The 8th International Cold Climate HVAC 2015 Conference, CCHVAC 2015*. Importance of increased knowledge on reliability of district heating pipes. Elsevier Procedia Engineering 2016 (Accepted).

Paper I Tereshchenko T, Nord N. Uncertainty of the allocation factors of heat and electricity production of combined cycle power plant. *Applied Thermal Engineering*, 2015. 76(0): p. 410-422.

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Research paper

Uncertainty of the allocation factors of heat and electricity production of combined cycle power plant

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HIGHLIGHTS

- Combined cycle power plant was modeled for design and off-design conditions.
- Seven emissions' allocation methods were analyzed.
- Power bonus method was the most sensitive method compared to the others.
- The results might be used for development of emission trading system for CHP plants and pricing of heat and power.

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ABSTRACT

There are many different methods for the allocation of CO_2 emissions in Combined Heat and Power plants. The choice of allocation method has a great effect on energy pricing and CO_2 allocation in Combined Heat and Power plants. The power bonus method is the main method used for the allocation of CO_2 emissions between heat and power production in the European Union and given as a standard. Aside from this method, six different allocation methods were tested on the Combined Cycle Power Plant in this study. Operational and design parameters of the Combined Cycle Power Plant were taken into consideration during analysis. The District Heating system, with an annual heat load of 27 GWh and maximum heat effect requirement of 14 MW, was chosen for the simulation model. This load was represented by the university campus. The energy source for District Heating was a Combined Cycle Power Plant with supplementary firing technology and natural gas as a fuel. The modeling of the system was carried out by the simulation software Aspen HYSYS, while data post-processing was done by MATLAB. Sensitivity analysis of the different allocation methods was performed for the Combined Cycle Power Plant under a yearly heat and electricity load. It was noted that different allocation methods produce different allocation factors. The differences between heat allocation factors for design and operational conditions were small. The most sensitive method was the power bonus method. The study showed that the decision regarding allocation method should be carefully analyzed before implementation in the standards and different policies, because benefits from cogeneration technology and distribution systems should be enabled. The results obtained in this study can be used by designers of Combined Heat and Power systems and policy makers, as a tool for developing an emission trading system for Combined Heat and Power plants and for the pricing of heat and power.

1. Introduction

The reduction of CO_2 emissions is a challenge for the coming decade, especially with the implementation of the Kyoto protocol. Beside transport, heating is responsible for a large share of the total greenhouse gas emissions [1,2]. One way to decrease the emissions generated by energy services (heating, hot water, electricity), is to increase the efficiency of the different energy conversion

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Nomenclature

$E_{el}(kWh)$ electricity from cogeneration plant
$E_{F,i}(kWh)$ fuel input to cogeneration plant
$E_{net}(kWh)$ electricity energy output from cogeneration plant
$Ex_E(kWh)$ net output of electrical exergy from cogeneration
$Ex_Q(kWh)$ net output of thermal exergy from cogeneration
$E_{P,in}(kWh)$ primary energy input
$E_{del}(kWh)$ power energy generated in the cogeneration plant
$\dot{E}i(\mathrm{kW})$ power rate
$\Delta E(kWh)$ electricity losses in cogeneration plant due to thermal production
$f_Q(-)$ fraction of cogeneration emissions allocated to heat generation
$f_E(-)$ fraction of cogeneration emissions allocated to electricity production generation plant
$f_{P,dh}(-)$ primary energy factor of the DH system
$f_{P,F,i}(-)$ primary energy factor of the fuel for cogeneration plant
$f_{P,el}(-)$ the primary energy factor of replaced electrical power
$Fuel_{in}(\mathrm{kWh})$ total primary fuel energy consumed in the cogeneration plant
$n(-)$ intensity of GHG emissions of production unit
$Q_{net}(\mathrm{kWh})$ thermal energy output from cogeneration plant
$Q_{del}(\mathrm{kWh})$ the heat energy delivered to the border of the supplied building
$\dot{Q}_i(\mathrm{kW})$ heat effect
$T(\mathrm{K})$ temperature of the medium
$T_0(\mathrm{K})$ mean ambient temperature of heating period
$T_s(\mathrm{K})$ supply temperature in DH system
$T_r(\mathrm{K})$ return temperature in DH system
$T_{cond}(\mathrm{K})$ condensing temperature in the cogeneration plant
$T_{out}(\mathrm{K})$ temperature of extracted steam in the cogeneration plant
$\eta_{alt\_heat}(-)$ heat production efficiency of producing thermal energy via alternative heat generation plant
$\eta_{alt\_elec}(-)$ power production efficiency of producing power energy via alternative power
$\tau_i(h)$ operation time of the power plant
$\Delta\tau_i(h)$ duration of the heat or electricity load
$\eta_c(-)$ Carnot efficiency
$\nu_p(-)$ degree of process quality

technologies that provide these services, by combining them in a polygeneration energy system. A polygeneration energy system is one that generates more than just one single energy service. In the case of District Heating (DH) for instance, polygeneration systems could save over 60% of the energy resources and emissions compared to conventional solutions [3–6]. The simplest example of such a system is the Combined Heat and Power (CHP) plant. Today, the benefits and potential of cogeneration technology are well-known and prove. The following authors discussed this technology in detail [7–10]. When DH is generated in highly efficient CHP plants, it is a reasonable and well-established measure to increase energy efficiency and to promote the resource saving use of primary energy carriers [11].

The European Union has recognized the importance of CHP technology in combination with DH systems. The benefits of CHP arise from a higher efficiency, which leads to fuel savings and consequently emission reductions. The improved efficiencies and fuel flexibility of CHP provide significant benefits in terms of security of energy supply systems. The Directive 2004/08/EC [12] promotes cogeneration technology. The guidelines from the directive allow the benefits of expanding CHP in district-heating systems to be made visible [13]. The European Union has set targets to reduce energy use by 20% and CO_2 emissions by at least 20% by 2020. DH can greatly contribute to achieving the global policy objectives. Doubling sales of DH by 2020 will reduce Europe's primary energy supply, import dependency on other countries, and CO_2 emissions [14].

In CHP plants, heat and electricity are generated simultaneously. Consequently, it is difficult to precisely distribute the primary energy input, emissions or operating costs to each of these energy outputs. In order to address this problem, different allocation methods have been developed [11]. The allocation method is the methodology which can provide information how to share benefits and drawbacks from joint generation. The main strategy for CHP plants today it is to be more environment-friendly and energy efficient. The DH technology can provide the possibility of decreasing pollution in combination with CHP plants. Unfortunately not all CHP plants use renewable energy sources like biofuel or municipal waste for producing heat and power. This is one of the reasons why allocation methods should be used in CHP plants in order to allocate CO_2 emissions. The allocation methods could also indicate the economic potential of technology. When less fuel is consumed, less pollution is released; this means that technology is environmentally-friendly.

The CHP plant produces electricity and heat, while the delivery of these two products is performed by different companies. The method for emissions' allocation is needed to ensure that each part is credited with its appropriate share of the emissions from the system. In addition, having a meaningful allocation method allows the sources of CO_2 and other emissions to be better understood and, where appropriate, reduced [15]. The choice of allocation method will have a great effect on energy pricing and CO_2 allocation in CHP. The most recognizable method of fuel allocation is the power bonus method given in the standard EN 15316:2007 [16]. This method is well known and accepted by the Life Cycle Assessment society (LCA) [17].

Limited work has been carried out on developing methods for allocating CO_2 emissions from cogeneration. One of the first records about allocation methods belongs to Strickland and Nyboer [18,19]. These researchers have mentioned several methods which could be used for allocation products from CHP plants. Their work was based on methods mentioned previously by Phylipsen et al. [20] with some simplifications. The following authors had performed analysis in their research based on these methods. Graus and Worrell in their study [21] employed different allocation methods to calculate the CO_2-intencities from CHP production. Abusoglu and Kanoglu in [22] performed analysis on Diesel Engine Power Cogeneration (DEPC) plant. They studied allocation of emissions from a DEPC plant based on six methods. In [23] Aldrich et al. investigated Greenhouse Gas (GHG) emissions in CHP systems applying exergy method with improvements. Wang and Lior in [24] analyzed fuel allocation in a combined steam-injected gas turbine (STIG) applying seven methods, three of them were thermoeconomics-based. Holmberg et al. studied allocation of fuel and CO_2 emissions in CHP plant integrated with pulp and paper mill [25]. Rosen in [26] reported that the exergy method is the most accurate method for allocation CO_2 emissions from CHP systems. Dittmann et al. in [27] concluded that Dresden method which was proposed by Zscherning and Sander [28] is the best one because it is based on laws of thermodynamics. World Energy Council (WEC) [29] in their research devoted to energy systems proposed different allocation schemes in the context with using Life Cycle Assessment (LCA), but still there is no generally accepted one [30].

The economic-based allocations are not investigated in this paper since such methods are prone to be misleading and fluctuate markedly with price swings for fossil fuels. The economic-based allocations are easily influenced by decision and policy makers [15].

Many studies have been devoted to investigating the design conditions of CHP plants. The focus so far has been on describing the thermodynamic principles of combined cycles at design point and practical design considerations. However, it must be realized that the operating conditions change, and the system should be able to operate at conditions far from design point. Off-design theory is about predicting how the system reacts to parameter changes. In design and off-design of the CHP plant, the actual geometry of the components remains constant but operational parameters can undergo changes. The CHP plant may operate for prolonged times at off-design conditions, depending on power demand, ambient condition, and other considerations. This will have a significant impact on the plant performance and, consequently, ensure the system performs not only at design conditions, but also at off-design conditions [31]. Therefore, the need increases for analysis and comparison of design and off-design parameters of the CHP plant in combination with the allocation methods.

Nowadays Combined Cycle Power Plants (CCPP) are receiving major attention throughout the world as one of the most effective options among the various energy conversion technologies. This technology is well developed and has been widely accepted in fossil-fired power plants due to its higher efficiency [32]. In this paper, CCPP has been analyzed and the results presented focus on a CCPP integrated in a DH system.

Different analyzes had been carried out on allocation methods and parametric studies of CHP systems by researchers in their work. However the authors did not found proper information how different operational and design parameters of CHP systems can effect on allocation between heat and power production. The proposed methods give constant yearly values for fuel and CO_2 emissions allocation. Therefore, the authors feel that uncertainty analysis of allocation methods is necessary in order to see yearly variations. In addition much research is needed in this area.

The aim of this paper is to investigate the effects of the different parameters which the system undergoes during the year. The goal was to compare system operation in design conditions with off-design conditions and to see how these different conditions would affect the choice of the allocation method. The modeling of the system was carried out by the simulation software Aspen HYSYS [33], while the data post-processing was done in MATLAB [34].

Aspen HYSYS simulator offers a comprehensive thermodynamics foundation for accurate calculation of physical properties, transport properties, and phase behavior for the oil & gas and refining industries [33]. The research carried out on CHP systems in [35,36] showed that the simulation results were found to be in good agreement with the operating data.

This paper is divided into the following sections: Section 2 introduces the methodology for the calculation of the allocation methods; Section 3 described the model and details of the process in the CCPP. Section 4 presents the off-design model assumptions. Results from parametric studies of the CCPP and the allocation methods are described in Section 5. The final section offers a conclusion on the results from Section 5 and remarks on the possibilities for future work.

2. Methodology

Firstly, the allocation methods were introduced. To calculate the allocation factors, it was necessary to calculate total electricity and heat energy production in a CHP plant. Dependence between heat and electricity use from the customer side and the power plant side was described afterwards.

2.1. Allocation methods

The principle of energy allocation is widely used when heat and power are produced simultaneously in a CHP plant. Seven different allocation methods were analyzed in this paper. The methods are given in the following text.

The *energy method* is most widely used because of its simplicity. This is an example of physical allocation. The primary energy consumption is allocated between heat and electricity produced in the CHP plant. If the amount of electricity produced in the CHP plant is 70% and the amount of heat is 30%, this mean that allocation is 70 units of energy which is consumed for power production and 30 for heat production. The emissions released in the environment are allocated as 70% from power production and 30% from heat production. This means that, in the *energy method,* the allocation factors can be expressed as:

$$\boldsymbol{f_Q} = \boldsymbol{Q}/(\boldsymbol{Q}+\boldsymbol{E}) \tag{1}$$

$$\boldsymbol{f_E} = \boldsymbol{E}/(\boldsymbol{Q}+\boldsymbol{E}) \tag{2}$$

where f_Q and f_E denote fractions of emissions allocated to heat and electricity production, respectively. In Equations (1) and (2), Q and E represent thermal and electrical production, respectively. This method does not take any energy quality aspects into account, allocating lower impact to electricity than to the other methods [37]. Consequently, it can be argued that it underestimates the share of the emissions allocated to electricity production [26].

The *alternative generation method* was developed by the Finnish District Heating Association [38]. In the alternative generation method, *the* share of CO_2 emissions is beneficial for both the heat and the power production in the CHP plant. The method allocates emissions and resources to the heat and power production in proportion to the fuel needed to produce the same amount of heat or power in separate plants. These alternative plants use the same fuel as the CHP plant [39]. Consider a CHP plant, which consumes 100 units of energy, while producing 30 units of electricity and 60 units of heat. Alternative production in two separate plants, a heat only plant and a condensing plant, will depend on their efficiencies, η_{heat} and η_{elec} respectively. In order to produce the same amount of electricity and heat, the separate plants will consume more fuel, because of lower separate efficiencies in comparison with cogeneration. The allocation of heat and electricity will be based on the amount of fuel needed if separate production plants had been used [37]. From the following example, the allocation factor can be expressed as:

$$\boldsymbol{f_Q} = \left(\frac{\boldsymbol{Q}}{\eta_{\boldsymbol{alt\_heat}}}\right) \Big/ \left(\frac{\boldsymbol{Q}}{\eta_{\boldsymbol{alt\_heat}}} + \frac{\boldsymbol{E}}{\eta_{\boldsymbol{alt\_elec}}}\right) \tag{3}$$

$$\boldsymbol{f_E} = \left(\frac{\boldsymbol{E}}{\eta_{\boldsymbol{alt\_elec}}}\right) \Big/ \left(\frac{\boldsymbol{Q}}{\eta_{\boldsymbol{alt\_heat}}} + \frac{\boldsymbol{E}}{\eta_{\boldsymbol{alt\_elec}}}\right) \tag{4}$$

where η_{alt_heat} and η_{alt_elec} are the heat and power production efficiencies of producing thermal and power energy via an alternative generation plant. This allocation method therefore shares the emissions among the products in a particular format and treats one or the other product as the primary one [26].

The *power bonus method* is the most recognizable method for energy allocation, because it is promoted by the European standard EN 15613-4-5:2007 [16] and is widely used nowadays. In this method the heat is the main product, while all power is considered as a bonus. The primary energy is allocated to the electricity produced in the CHP plant. The total primary energy used by the CHP plant includes all energy used in the production of heat and electricity. This includes the primary energy related to fuel handling and combustion as well as primary energy needed for the production of additives, handling of ashes, construction, and dismantling of the CHP plant, etc. In accordance with EN 15316-4-5:2007, the performance of the DH system and produced heat in the CHP plant can be rated by evaluating the primary energy factor $f_{P,dh}$ of the specific DH system. The primary energy factor is defined as the primary energy input $E_{P,in}$ to the system divided by the heat Q_{del} delivered at the border of the supplied building [16].

$$f_{P,dh} = E_{P,in} \Big/ Q_{del} \tag{5}$$

The thermal energy balance is given by:

$$f_{P,dh} \cdot \sum_j Q_{del,j} + f_{P,el} \cdot E_{el} = \sum_i f_{P,F,i} \cdot E_{F,i} \tag{6}$$

From Equation (6) the primary energy factor of the DH system can be expressed as:

$$f_{P,dh} = \left(\sum_i f_{P,F,i} \cdot E_{F,i} - f_{P,el} \cdot E_{el} \right) \Big/ \sum_j Q_{del,j} \tag{7}$$

where $f_{P,dh}$ is the primary energy factor of the DH system, $f_{P,F,i}$ is the primary energy factor of the fuel for the cogeneration plant, $f_{P,el}$ is the primary energy factor of replaced electrical power, E_{el} is the electricity from the cogeneration plant, Q_{del} is the delivered heat at the border of the supplied building, and $E_{F,i}$ is the fuel input to the cogeneration plant.

Finally, in the *power bonus method,* the allocation of primary energy can be expressed as:

$$f_Q = f_{P,dh} \cdot Q_{del} \Big/ \left(Q_{del} + E_{del} \right) \tag{8}$$

$$f_E = 1 - f_{P,dh} \cdot Q_{del} \Big/ \left(Q_{del} + E_{del} \right) \tag{9}$$

This method promotes cogeneration technology instead of the separate production of heat and electricity. It also promotes the usage of different renewables like municipal waste, pellets, biofuels, etc. Today, the *power bonus method* is one of the most efficient methods for promoting DH technology; as power is counted as a bonus, the largest part of CO_2 emissions is allocated to power production.

The *exergy method* represents allocation from a thermodynamic point of view. This is an example of physical allocation; it defines the quality of energy. The exergy is the maximum amount of work which can be obtained from the system when it interacts with the reference state. For exergy analysis, the characteristics of the reference environment must be specified completely. This is commonly done by specifying the temperature, pressure, and chemical composition of the reference environment. The results of the exergy analyses, consequently, are relative to the specified reference environment, which, in most applications, is modeled after the actual local environment. The exergy of a system is zero when it is in equilibrium with the reference environment [26]. Many authors have carried out exergy analysis in their research for different purposes [40–43].

From the thermodynamic point of view, electricity consists of 100% exergy, and consequently the exergy of electricity is defined as:

$$Ex_E = E \tag{10}$$

According to the exergy method, the heat allocation can be calculated based on the following equation:

$$Ex_Q = \left(1 - \frac{T_0}{T} \right) \cdot Q \tag{11}$$

where Ex_E and Ex_Q are net output of electricity and thermal exergy from cogeneration, T and T_0 are the medium and mean ambient temperatures of the heating period. When the heat is transferred at a sliding temperature, Equation (11) is not valid. In that case, the temperature T should be replaced by the logarithmic mean temperature of the temperatures at which the heat is transferred. In the case of the DH system, these temperatures are the supply and return temperatures of the DH network, T_s and T_r [44], and then the temperature of the medium can be defined as:

$$T = (T_s - T_r) / ln(T_s / T_r) \tag{12}$$

Consequently, the heat exergy can be defined as:

$$Ex_Q = \left[1 - \frac{T_0}{(T_s - T_r)/ln(T_s/T_r)} \right] \cdot Q \tag{13}$$

Finally, the allocation factors for the heat and electricity based on the *exergy method* become:

$$f_Q = Ex_Q / (Ex_Q + Ex_E) \tag{14}$$

$$f_E = Ex_E / (Ex_Q + Ex_E) \tag{15}$$

The application of this method requires profound knowledge of thermodynamics and power plant processes and is therefore rather complicated for practical use. However, it is judged as the fairest method, from a thermodynamic point of view, for dividing the benefits of the CHP production between electricity and heat [45] and can be carried out relatively simply because the necessary data can be measured directly on the plant. Thermodynamically, however, the method is not really “clean” because the losses of exergy caused by the heat exchange from the cogeneration process to the heating system are not allocated to the heat [27]. Consequently, compared to the energy allocation method, the exergy method avoids the difficulties associated with the allocations based on energy values. Such methods are problematic especially for cogeneration systems because the two main products are of significantly different quality and usefulness [15,26].

The *200% method* uses 200% efficiency for heat production. This means that all emissions are left to power production. This method, which was established by the Danish Energy Agency [46], is similar to the power bonus method, where all electricity is counted as bonus. It is well known in Denmark where there are large-scale CHP plants, which primarily produce power, and small-scale CHP plants for producing heat. The Danish Energy Authority has stipulated that energy efficiency of 200% has to be used when allocating the fuel costs of the CHP to the heat production in the energy and emission statistics. This means that, in order to produce two units of heat energy, one unit of real fuel has to be used and the other unit will be recovered from the heat otherwise directed to the turbine condenser. In the condenser, the heat unit would be wasted to the environment if not recovered to district heating [47]. Finally, in this method, the allocation factor for heat and electricity can be defined as:

$$\boldsymbol{f_Q} = \boldsymbol{Q}/(2\cdot\boldsymbol{Fuel_{in}}) \tag{16}$$

$$\boldsymbol{f_E} = 1 - \boldsymbol{Q}/(2\cdot\boldsymbol{Fuel_{in}}) \tag{17}$$

where $Fuel_{in}$ is the total primary fuel energy consumed in the cogeneration plant. The method assumes that the heat is produced with fixed efficiency, which is chosen as a general average between the energy and exergy methods [37].

The publicly available Specification *PAS 2050* [48] is the British standard, which explains the calculation of Greenhouse Gas Emissions (GHG) of goods and services. The allocation of emissions in the CHP is between the heat and power produced, multiplied by the intensity of the GHG emissions of the production unit. The special coefficient specifies the emissions released from fuel combustion used in the system. For the boiler-based CHP systems (coal, wood, solid fuel), the coefficient is 2.5, while for the turbine-based CHP systems (natural gas, landfill gas), the coefficient is 2.0. Finally the allocation factors in this method can be expressed as:

$$\boldsymbol{f_Q} = \boldsymbol{Q}/(\boldsymbol{n}\cdot\boldsymbol{E} + \boldsymbol{Q}) \tag{18}$$

$$\boldsymbol{f_E} = (\boldsymbol{n}\cdot\boldsymbol{E})/(\boldsymbol{n}\cdot\boldsymbol{E} + \boldsymbol{Q}) \tag{19}$$

where n is the intensity of GHG emissions of the production unit. It is important to note that these ratios apply to 1 MJ of energy produced. In most situations more energy of one type than of another will be produced. The allocation of emissions to heat and electricity arising from the CHP relies on the process-specific ratio of heat to electricity from each CHP system. For example, where a boiler-based CHP system delivers useful energy in the power to heat ratio 1:6, 2.5 units of emissions would be allocated to each unit of electricity and one unit of emissions would be allocated to each unit of heat delivered by the CHP system. This means that the CHP system has useful power to heat ratio of 1:6; the corresponding GHG emissions ratio is 2.5:6. These results will change with different heat and electricity characteristics of the CHP system [49].

The *Dresden method*, which was proposed by Zschernig and Sander [28], is based on exergy assessment. In power plants all primary energy is related to electricity production. At the same time in the CHP plants, one part of primary energy is consumed for thermal energy production. The *Dresden method* describes how to evaluate the electricity loss caused by the heat extraction (water steam condensation) in the CHP plant. The electricity losses due to heat extraction in the CHP plant can be evaluated as:

$$\Delta\boldsymbol{E} = \boldsymbol{Q}\cdot\boldsymbol{\eta_c}\cdot\boldsymbol{v_p} \tag{20}$$

where

$$\boldsymbol{\eta_c} = 1 - \boldsymbol{T_{cond}}/\boldsymbol{T_{out}} \tag{21}$$

and the maximum electricity production without heat extraction is:

$$\boldsymbol{E} = \boldsymbol{E_{del}} + \Delta\boldsymbol{E} \tag{22}$$

where ΔE is electricity loss due to heat extraction in the CHP plant, E is electricity energy generated in CHP plant including electricity losses (maximum electricity production without heat extraction). E_{del} is electricity energy generated in the CHP plant when heat extraction occurred. η_c is Carnot efficiency; T_{cond} and T_{out} are condensing temperature and temperature of extracted steam in the CHP plant. Mainly in smaller heat and power stations, where the determination of the heat losses is complicated, the exergy of the heat rated by a real degree of process quality v_p can be used as an equivalent of the electricity loss [27]. The fuel in the cogeneration plant can be allocated by this method according to the following equations:

$$\boldsymbol{f_Q} = \Delta\boldsymbol{E}/\boldsymbol{E} \tag{23}$$

$$\boldsymbol{f_E} = (\boldsymbol{E} - \Delta\boldsymbol{E})/\boldsymbol{E} \tag{24}$$

The results in the exergy assessment are comparable with evaluation of the delivered heat, because heat exchange efficiency has the same value as the degree of process quality in the Dresden method [27].

The above introduced allocation methods are summarized in Table 1.

2.2. Heat and power production in CCPP

The methodology presented in this section describes the calculation of heat and power demand in the campus and future implementation in the simulation model.

Total heat use, measured at the primary side of the consumer substation, can be estimated as:

$$\boldsymbol{Q_{del,j}} = \int \dot{\boldsymbol{Q}}\boldsymbol{d\tau} = \lim_{\tau\to 0}\sum_{\boldsymbol{i}} \dot{\boldsymbol{Q}}_{\boldsymbol{i}}\cdot\Delta\boldsymbol{\tau_i} \tag{25}$$

where $Q_{del,j}$ is total heat energy use at the primary side of customer substation, $\dot{Q}_i$ is heat effect required during $i-th$ hour, $\Delta\tau_i$ is the duration, $\dot{Q}_i$ the heat load.

The electricity use of the university campus can be calculated as:

$$\boldsymbol{E_{del,j}} = \int \dot{\boldsymbol{E}}\boldsymbol{d\tau} = \lim_{\tau\to 0}\sum_{\boldsymbol{i}} \dot{\boldsymbol{E}}_{\boldsymbol{i}}\cdot\Delta\boldsymbol{\tau_i} \tag{26}$$

where $E_{del,j}$ is the total electricity use at the primary side of a building, $\dot{E}_i$ is power rate demand, and $\Delta\tau_i$ is duration of the electricity load.

The CCPP was simulated based on the required heat energy use; the details of the simulation model are described in the next section. The input in the simulation model was thermal energy and the outputs were: power produced and fuel input in CCPP.

The fuel consumption for power production in the CCPP can be evaluated by using the relationship between thermal and power energy produced in the CCPP:

$$\boldsymbol{F_{in}} = \boldsymbol{f}\left(\dot{\boldsymbol{Q}}_{\boldsymbol{net}}, \dot{\boldsymbol{E}}_{\boldsymbol{net}}\right)\cdot\boldsymbol{\tau_i} \tag{27}$$

where Q_{net} and E_{net} are outputs of thermal and power energy from the CCPP, τ_i is the operation time. In order to evaluate the fuel input

Table 1
Allocation methods.

Method	Allocation factor heat	Allocation factor electricity
Energy method	$f_Q = \frac{Q}{Q+E}$	$f_E = \frac{E}{Q+E}$
Alternative generation method	$f_Q = \frac{\frac{Q}{\eta_{alt\_heat}}}{\frac{Q}{\eta_{alt\_heat}}+\frac{E}{\eta_{alt\_elec}}}$	$f_E = \frac{\frac{E}{\eta_{alt\_elec}}}{\frac{Q}{\eta_{alt\_heat}}+\frac{E}{\eta_{alt\_elec}}}$
Power bonus method	$f_Q = \frac{f_{P,dh}\cdot Q_{del}}{Q_{del}+E_{del}}$	$f_E = 1 - \frac{f_{P,dh}\cdot Q_{del}}{Q_{del}+E_{del}}$
Exergy method	$f_Q = \frac{Ex_Q}{Ex_Q+Ex_E}$	$f_E = \frac{Ex_E}{Ex_Q+Ex_E}$
200% method	$f_Q = \frac{Q}{2\cdot Fuel_{in}}$	$f_E = 1 - \frac{Q}{2\cdot Fuel_{in}}$
PAS 2050	$f_Q = \frac{Q}{n\cdot E+Q}$	$f_E = \frac{n\cdot E}{n\cdot E+Q}$
Dresden method	$f_Q = \frac{\Delta E}{E}$	$f_E = \frac{E-\Delta E}{E}$

for power production in the CCPP, data post-processing was performed in the MATLAB.

The information flow for the methodology used in this study is given in Fig. 1.

3. Case study

A small-sized DH system with an annual heat load around 27 GWh was analyzed in this paper. The load was represented by the university campus. The heat load values were collected over five years. The coldest year was taken as a starting point for plant design. The system was modeled with Aspen HYSYS simulation software. The property package was modeled with the Peng-Robinson equation of state. The ambient temperature at the design point was +15 °C, ambient pressure was 1.013 bar and air Relative Humidity (RH) was 60%.

The energy source for DH was the CCPP with supplementary firing technology. The system consisted of gas turbine cycle (GTC), steam turbine cycle (STC), heat recovery steam generator (HRSG), two combustion chambers, fed with natural gas and other components. The schematic layout of the system is represented in Fig. 2, and design parameters are summarized in Table 2.

In this simulation, natural gas was used as a fuel. The lower heating value (LHV) of the gas was 50.03 MJ/kg. The air and fuel are supplied to the reactor after a two-stage compression system. The adiabatic efficiency of the compression system was assumed to be 90%. The low pressure compressor (LPC) provides pressure of 6 bar, while the high pressure compressor (HPC) compresses up to 13 bar (Fig. 2). The air excess coefficient α was set to be 3.2 in the air-fuel mixture.

The air excess provides the dilution of the temperature before the GTC. The GTC was represented by two units; one is a high pressure gas turbine (HPGT) and the other is a low pressure gas turbine (LPGT); see Fig. 2. In the design stage, the temperature before the GTC was assumed to not exceed +1100 °C. The temperature of flue gases entering the gas turbine after conducting simulation was set to be 1086 °C. The entering pressure of flue gases in the HPGT was 13 bar. The pressure before the LPGT was 6 bar. The leaving pressure was 1.5 bar, which is slightly higher than ambient conditions. The nominal power of the GT generators was 14 MW and that of the compressor units, 5 MW.

In the CCPP with supplementary firing technology, the supplementary firing provided additional energy input to the steam cycle. In this way the flue gas temperature was increased. The fuel was added after the GTC. The combustion of supplementary fuel was accomplished by the air excess leaving the gas turbine in flue gases. The fuel was mixed with flue gases and burned in duct burners in the HRSG. There was no need for an air supply to the HRSG, because enough oxygen content was left after combustion in the reactor. In the design case, the temperature of the exiting flue gases was set to +900 °C.

The HRSG was modeled as three stages or heat exchangers; see Fig. 2. These are an economizer, an evaporator and a superheater. The HRSG has one steam pressure level. The parameters of the live steam entering the steam cycle were: $T = +500$ °C, $p = 60$ bar. The STC represented three units. The first was a high pressure steam turbine (HPST), the next was an intermediate pressure steam turbine (IPST), and the last was a low pressure steam turbine (LPST). The entering parameters of the working medium in the IPST were pressure of 12 bar and temperature +278 °C. In the LPST, the steam condenses up to a pressure of 0.05 bar. The adiabatic efficiency of the STC was assumed to be 90%.

The STC is with one extraction for DH purposes. The mass flow rate of water from the DH is satisfied by means of heat transfer connected with the heat exchange units. The DH system was fed from the IPST. The steam extraction occurred at a pressure of 10 bar.

The temperature of supply water in the DH system was +105 °C and the return water temperature was +50 °C. The CCPP had a two-stage heat exchanger system for satisfying the DH heat demand. The first stage heated return water to a temperature of +90 °C and the second stage heated up to +105 °C.

The heat duration curve (see Fig. 3), was obtained based on measurements in the university campus. The maximum heat load was 14 MW. The part load operation of the modeled CCPP plants was simulated by changing the mass flow rate in the DH system. The minimum heat load in the DH system in part load simulations was 1 MW, while the maximum was 14 MW. The DH load under 1 MW was covered by an electric boiler and was not included in the CCPP heat production calculation. The total heat consumption covered by the electric boiler was 2 GWh of delivered heat during the year.

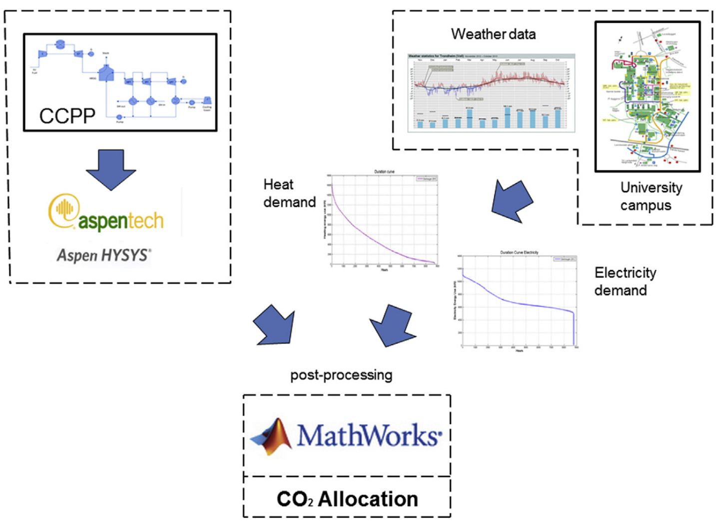

Fig. 1. The flowchart represents steps of analysis done in this paper.

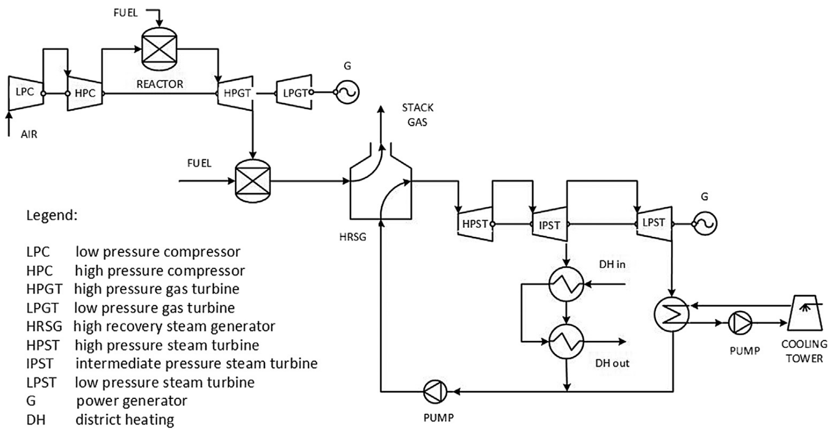

Fig. 2. Schematic of CCPP.

In CHP plants, the part load operation usually covers large periods of the total plant operation time and depends on DH heat demand [35]. From Fig. 3, we can notice that maximum load occurs only for a few hundred hours during the year, while the average load constitutes 29% of the maximum load covered by CCPP. The average load in CCPP corresponds to 48% or half of the all plant operational time. The performance parameters of analyzed CCPP at 100% DH load are summarized in Table 3.

4. Off-design model assumptions

A number of assumptions were made concerning plant operation in design and off-design conditions. The assumptions were based on a literature study. The following assumptions are common to all the solutions examined:

- for the simplicity of calculation, methane was treated as natural gas;
- no pressure drop in heat exchanger units;
- the plant operates all through the year;
- the maximum heat demand in DH was equal to 14 MW;
- the electricity grid purchased all the electricity produced in the CCPP;
- heat losses in the system were neglected;

In the CHP design, energy supply companies use different standards and directives in order to achieve a stable system with the best economic and environmental characteristics. The standardized data collected from many sources and research reports provides guidelines on how to achieve the best performance. The following text gives an overview of different operating conditions that have an impact on plant performance. The operation and design conditions which were analyzed are described below.

Ambient air temperature has a great effect on CCPP performance. It is known that CCPP is designed for optimal parameters of ambient air. This value is regulated by ISO 2314 [50] and is +15 °C for the design case. However, this value cannot stay the same throughout the year. When it comes to CCPP exploitation, the parameters of intake air affect not only the GTC but also the supply fuel quality and products of stack gases. When air temperature rises, the GT may swallow the same volume of air, but that air weighs less with increasing atmospheric temperature. In this case the density of the air reduces. Less air mass means less fuel mass is required to be ignited with that air and consequently lower power is developed in the GT output [51]. As a result, the main performance characteristics of the CCPP, such as power performance, fuel consumption, etc., change significantly. Most of the time, the CCPP works in off-design conditions. Therefore, in this study the outdoor air temperature was simulated for the coldest period of the year, which corresponds to −20 °C, transition period −10 °C, 0 °C, +5 °C and for the design case +15 °C.

Table 2
Design parameters of CCPP.

Parameter	Value
Ambient pressure	101 kPa
Air relative humidity	60%
Ambient air temperature	+15 °C
Pump pressure	60 bar
Steam turbine inlet temperature	+500 °C
Condensing pressure	0.05 bar
Air excess in air-fuel mixture	3.2
Fuel temperature	+15 °C
Gas turbine adiabatic efficiency	0.9
Steam turbine adiabatic efficiency	0.9
Compressor adiabatic efficiency	0.9
Supplementary firing temperature	+900 °C

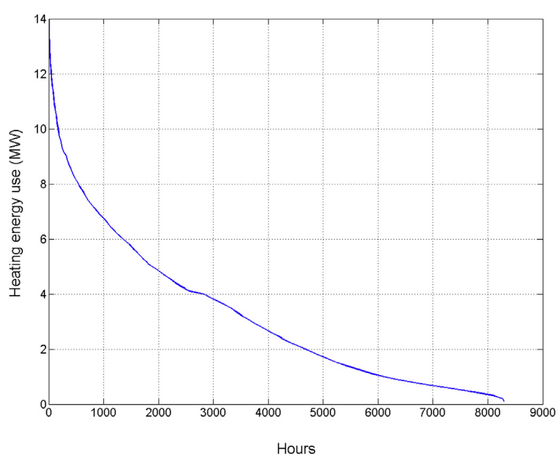

Fig. 3. The heat duration curve of the analyzed campus.

Table 3
Performance parameters of CCPP at 100% DH load.

Parameter	Value
Power production in HPGT	5.1 MW
Power production in LPGT	7.2 MW
Power production in HPST	2.5 MW
Power production in IPST	0.24 MW
Power production in LPGT	0.25 MW
Power consumption of LPC	4.4 MW
Power consumption of HPC	2.8 MW
Primary fuel input	1270 kg/h
Supplemental fuel input (flue gas temperature before HRSG is +900 °C)	587 kg/h
Air mass flow rate	71,310 kg/h
Air temperature after LPC	+228.6 °C
Air temperature after HPC	+360.9 °C
Flue gas temperature after superheater	+765.7 °C
Flue gas temperature after evaporator	+348.5 °C
Flue gas temperature after economizer	+116.7 °C
Water temperature before economizer	+100 °C
Water temperature before evaporator	+277 °C
Steam temperature before superheater	+278 °C
Steam temperature after superheater	+500 °C
Steam temperature after HPST	+278.2 °C
Steam temperature after IPST	+256.6 °C
Steam temperature after LPST	+33.15 °C
Steam-water mixture temperature after the first stage of heat exchange unit in DH system	+180 °C
Water temperature after the second stage of heat exchange unit in DH system	+110.6 °C
Mass flow rate of water in DH system	218,703 kg/h

Ambient pressure for the CCPP should be 1 bar. This is based on ISO 2314 conditions and corresponds to the pressure at sea level. The ambient pressure can vary depending on sea level variation and atmospheric conditions. In this study the ambient pressure was changed in the range from 101 bar to 75 bar which corresponded to the elevation change at sea level from 0 to 2743 m.

Ambient relative humidity (RH) mostly affects the CCPP power output. When all parameters remain stable, a change of the RH to a higher value can increase the efficiency of the plant. This is because at higher levels of RH there will be higher content in the working medium of the gas cycle. This results in a better GT enthalpy drop and more exhaust gas energy entering the HRSG [52]. The higher energy transfer in the HRSG leads to a change of pinch point temperature approach. The pinch method is a methodology for minimizing energy use and for better energy utilization of steam flows. Applying this method increases the area of energy transfer between flue gases and the working medium in the economizer. This gives better energy utilization in the HRSG and respectively increases the efficiency of the unit. The off-design simulation can show the consequences of different operational parameters if changes take place during exploitation. A change in operation conditions was performed for air RH in the range of 20% to 80%.

Supplementary firing provides additional energy input to the system. In the CCPP, supplementary firing increases the temperature in the HRSG and stabilizes the parameters of generated steam, providing a system which is more flexible than the traditional one. This provides better energy utilization of flue gas from an exergy point of view. On the other hand, with the development of GT technologies, the requirement for such an option decreases, because contemporary GTs have higher inlet temperatures and respectively higher exhaust temperatures too. Nevertheless, the increased operating and fuel flexibility of the combined cycle with supplementary firing may be an advantage in special cases, particularly in installations used for cogeneration. This arrangement makes it possible to control the electrical and thermal outputs independently [52]. In the design case the temperature of flue gases was set to +900 °C. This value was arrived at based on the HRSG maximum inlet temperature in the design conditions. In off-design simulation the value was changed from +700 °C to +1000 °C. The high temperature or supplementary firing does not mean that the HRSG will have the best performance characteristics. In order to determine the best energy utilization in the HRSG, pinch point analysis was applied and the results are presented in Section 5 of this paper.

Change in the *pump pressure* has the main effect on power production in the plant. When pump pressure increases, the STC undergoes an additional portion of steam extraction in the steam turbine (ST) in comparison with the design point. However, an increase in pressure in the STC leads to additional use of electricity. In this study the pressure in the STC after the pump system was simulated ranged from 40 to 80 bar.

Air excess coefficient in the air-fuel mixture is an important factor affecting the flue gas flow rate. This is the ratio of the excess combustion air, which defines the total combustion air flow. The change in the ratio of excess air also had a strong impact on the production of the CHP plant [53]. Based on stoichiometric coefficients for combustion reaction (natural gas with air), the temperature of flue gases might be +1900 °C. The air excess coefficient regulates the temperature dilution before the GTC. Every manufacturer of GT equipment provides detailed information stating that the GT inlet temperature cannot be above a certain limit. During the development of GT technology, the temperature limit gradually increased in comparison with the first exploited GTs. Nowadays we can divide them into five generations [52]. The inlet temperature of flue gases in the last generation can reach the limit of more than +1350 °C. The temperature of the flue gases before the GT cycle affects the parameters of the flue gases after the GT cycle. This has an effect on steam production in the HRSG and consequently power production in the STC. If we assume that changes might be made to the GT in future, resulting in better operational parameters such as inlet temperature of flue gases, then the need for simulation of air-fuel ratio increases. In this analysis the air access coefficient in the air-fuel mixture supplied to the GTC was simulated in the range of 3.0 to 4.0.

The *fuel temperature* affects the burning process in the reactor. The gaseous fuel is supplied directly to the CHP plant by means of pipes. It cannot be stored near the plant because of its properties. After treatment and pressure regulation, it is supplied to the reactor for further burning. The pressure of the supplied gaseous fuel depends on its temperature and density, and on the ambient conditions. Preheated fuel provides a stabilized burning process in the reactor. Therefore, it is important that fuel is preheated before reaching the reactor. The temperature of the preheated fuel is regulated by standards at a value of +15 °C. However, in some cases this temperature can also be preheated up to +250 °C. In this study, the off-design analysis had to deal with temperatures in the range from +50 to +200 °C.

The steam turbine inlet temperature affects the thermal efficiency of the CHP plant. When the vapor expands in the ST, the temperature drops and energy is released. The higher the temperature in the ST cycle, the higher the useful energy for heat production in the CHP plant. During analysis the inlet steam pressure had variations from +475 °C to +540 °C.

The condensing pressure mostly affects power production in the CHP plant and the total CHP efficiency. The condensing pressure of the LPST varied from 0.05 to 0.2 bar. The simulations of reduced components' efficiencies were performed by changing the *adiabatic efficiencies* for the GT, ST, and compressors separately. The efficiencies were reduced to 80%, having been 90% at the design point. The summary of the off-design parameters is given in Table 4.

Table 4
Off-design parameters of CCPP.

Parameter	Value
Ambient pressure	75 kPa–101 kPa
Air relative humidity	20%–80%
Ambient air temperature	−20 °C–+15 °C
Pump pressure	40 bar–80 bar
Steam turbine inlet temperature	+475 °C–+540 °C
Condensing pressure	0.05 bar–0.2 bar
Air excess in air-fuel mixture	3.0–4.0
Fuel temperature	+15 °C–+200 °C
Gas turbine adiabatic efficiency	0.8–0.9
Steam turbine adiabatic efficiency	0.8–0.9
Compressor adiabatic efficiency	0.8–0.9
Supplementary firing temperature	+700 °C–+900 °C

5. Results and discussion

5.1. Design and off-design system performance

Off-design operational analysis provides valuable information on the operation of the components and system, particularly on its range of applicability. Therefore, it is necessary to analyze the amount of electricity and heat produced by the CHP system, in terms of size, under the part-load characteristics [54].

Firstly, the parametric studies of the CCPP plant shown in Fig. 2 were carried out in order to see any variation in the plant performance under changeable operational conditions. The simulations were performed for the DH load given in Fig. 3. The change in CCPP behavior is represented in the Fig. 4 (that includes 4a–4d).

The analysis show that power efficiency, CHP efficiency, thermal efficiency and fuel input varied depending on analyzed load in DH system. For example, the obtained values for power efficiency, analyzing the possible range for air excess coefficient ($\alpha = 3.0$–4.0), were 27.85% and 32.45% for 14 MW heat load; for 1 MW heat load these values constituted to be 43.80% and 47.27%. Thus, taking into consideration all simulation steps for DH load, the average value for power efficiency change was 4.02%, see Fig. 4a. The maximum value for CHP efficiency change was 2.46%; see Fig. 4b. In terms of thermal efficiency, the maximum change was 2.98%; see Fig. 4c. The maximum change in the fuel input rate was 3075 kW, as shown in Fig. 4d, due to a change in the supplementary firing temperature.

The biggest variation in the power efficiency occurred when the air excess coefficient was changed from the design value $\alpha = 3.0$ to $\alpha = 4.0$, while the smallest was when the air RH had been analyzed; see Fig. 4a. The air excess coefficient provided an increase in the mass flow rate of the flue gases through the GTC; this led to an increase in power production in the CCPP by 4.2% in comparison with the design case. At the same time, the fuel input to the system had decreased. The reduction in fuel input can be explained by the fuel dilution, increasing the mass flow rate of air and fuel to the system. In terms of the CHP efficiency, this also had a positive effect. The CHP efficiency increased to 2.11%, according to Fig. 4b.

The air RH had brought about a decrease of the fuel input to the system, while the CHP efficiency, the power efficiency, and the thermal efficiency continued with no variation. The higher air RH provided higher levels of humidity and consequently a higher content in the working medium of the gas cycle. This had a positive effect on the HRSG. The higher enthalpy drop in the GTC resulted in more exhaust gas energy released in the HRSG.

The thermal efficiency of the CCPP showed the maximum change of 2.98% when the supplementary firing temperature was set to +1000 °C. The supplementary firing provided additional energy input to the STC, which resulted in better energy utilization and system flexibility, when shifting from the base load to the high

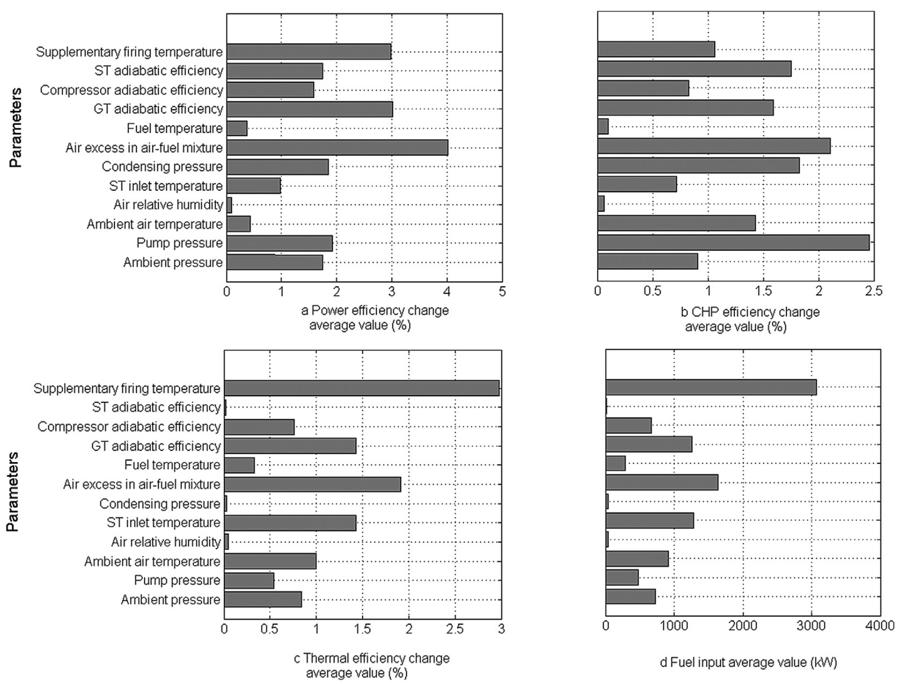

Fig. 4. Change in CCPP behavior based on analyzed parameter.

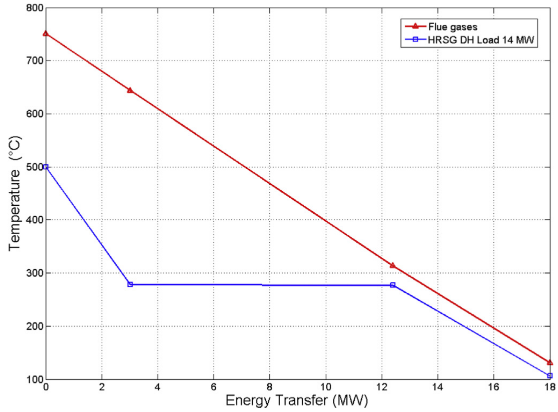

Fig. 5. Energy utilization in the HRSG where the temperature of flue gases is +750 °C.

peak. Based on heat flow–temperature diagrams shown in Fig. 5 and Fig. 6, we can conclude that the high temperature of flue gases before the HRSG does not indicate the best energy utilization. In the observed CCPP, the HRSG had one pressure stage.

This means that the pressure entering the economizer is the same as the one leaving the superheater. In general, the temperature of the HRSG should be +200 °C higher than the medium leaving temperature of the superheater. The higher temperature at entry provides lower energy utilization in the HRSG and increases the exergy losses. The space between the curves marked in blue or red, as presented in Figs. 5 and 6, shows exergy losses. The highest temperature in the HRSG had an effect on fuel input in the CCPP. This was the maximum value during simulation and resulted in a change of 3075 kW of the fuel input; see Fig. 4d.

The supplementary firing temperature also affected the power efficiency of the CCPP; see Fig. 4a. The maximum change in the power efficiency was 2.99% when the minimum supplementary firing temperature was set. This could be explained by increasing the mass flow rate of air-fuel mixture through the GTC. The CHP efficiency showed a negligible variation of 1.06%, see Fig. 4b, due to the supplementary firing temperature. The minimum change in the thermal efficiency occurred due to variation in the following parameters: ST adiabatic efficiency, condensing pressure, and the air RH. This can be seen in Fig. 4c. The variation in condensing pressure had most effect on power production. The condensing pressure in the CCPP affected the temperature of the water-steam mixture leaving the LPST. The water (compressed liquid) entering the pump before the economizer should not contain any steam fraction; see Fig. 2. The water-steam mixture should be fully condensed up to the saturation temperature. This means that the temperature after the LPST remains constant in all cases.

The biggest influence on CHP efficiency was the change in pump pressure, which increased by 2.46% in comparison with the smallest value for pump pressure in the analyzed range; see Fig. 4b. The maximum change occurred when the pressure in the STC was increased to 80 bar. The higher the pressure, the higher the amount of electricity produced in the STC. Power production increased by 1.92% in comparison with the design case when the pump pressure was set to maximum; see Fig. 4a. The thermal efficiency did not show any particular changes due to the constant vapor temperature level in the STC.

The lowest influence on the fuel input in the CCPP had ST adiabatic efficiency, condensing pressure and air RH, while the supplementary firing had the highest; see.Fig. 4d. These parameters

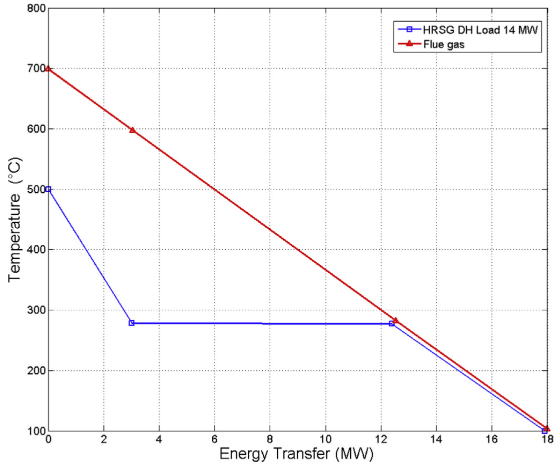

Fig. 6. Energy utilization in the HRSG where the temperature of flue gases is +700 °C.

can change electricity production in the plant, but both the fuel input and the thermal efficiencies remained constant.

The simulation of the CCPP showed that the operational and design parameters have a significant influence on plant performance. This is valuable information since it is important to provide a reliable heat and power supply to customers while shifting from the base load to the peak load and vice versa.

5.2. Results on allocation methods

In Section 2, different methods for allocating CO_2 emissions for cogeneration systems were introduced. The choice of allocation method is more important than the size of the plant, properties of the distribution network, plant technology and even more important than which fuel is used. When analyzing the environmental performance of the CHP, it is important that the reader is aware of the effects related to the allocation method used [37].

In this study different allocation methods have been analyzed in order to investigate the effect of fuel allocation between the heat and the electricity produced in the CCPP. Allocation methods were combined with the parametric studies of the CCPP and annual heat energy use at the university campus. Operating and design parameters were analyzed, and the results were combined to estimate the effect on choice of allocation method as shown in Section 5.1. A sensitivity analysis of the different allocation methods was performed for the CCPP under annual heat and electricity load. Based on the DH load and parametric studies of the CCPP given in Section 5.1, results were obtained for various allocation methods.

The results represented in the Table 5 show the values of the CO_2 allocation factors for heat in the design phase.

Table 5
Allocation factor heat in the design phase.

Method	Design value allocation factor heat
200%	0.0608
Alternative generation	0.3830
Energy method	0.2162
PAS 2050	0.1212
Power bonus method	0.2226
Exergy method	0.1507
Dresden method	0.8340

It might be noticed that different allocation methods produce different results in Table 5. For example, the fuel allocation for heat for the *alternative generation method* was 38.3%, while using the *200% method* this value was 6% and for the *power bonus method* was 22.3%.

Fig. 7 presents the effect on allocation factors depending on analyzed parameters introduced in Table 4 and Section 4.

The change in heat allocation factors for design and operational conditions showed a small variation. This can be noticed by comparing Table 5 and Fig. 7. The most sensitive method due to the change in operation parameters was the power bonus method. The alternative generation method offered the biggest share in the heat allocation, while the smallest share for heat was shown by the 200% method. The heat allocation factor based on the power bonus method changed by 0.16 units due to the change in condensing pressure; see Fig. 7. The air excess coefficient in the air-fuel mixture resulted in a change of 0.22 units. The change in the steam turbine adiabatic efficiency and supplementary firing temperature resulted in 0.12 and 0.11 units of heat allocation factor. The changes in the parameters described above have the greatest influence on power production in the CCPP.

Finally, for the different allocation methods, Fig. 8 shows the maximum sensitivity in the allocation factors for heat and electricity production.

As Figs. 7 and 8 show, the power bonus method was the most sensitive compared to other methods. The 200% method showed the smallest change in the analyzed parameters, resulting in a beneficial share of emissions' allocation for DH between heat and power production. The PAS 2050 and exergy methods also had good results and showed that the operational and design parameters did not have a significant influence on allocation factors for both heat and electricity. The change in operation parameters gives a variation in the heat allocation in the CCPP that should be taken into consideration while applying the power bonus method. When the efficiencies of the CCPP vary significantly with load, or are varied to match the demand, the calculated CO_2 emissions are clearly not fixed and could not be constant under any convention. For practical purposes, it would be sensible to define efficiency values, perhaps seasonal averages, as a basis for nominal intensities [55]. As an alternative to the power bonus method, other methods with small variation under variable loads should be considered such as the 200% method, the PAS 2050 method, or the exergy method. In general, the allocation of the main products is a problematic task, especially in cogeneration systems, since heat and electricity are products of significantly different quality usefulness [15].

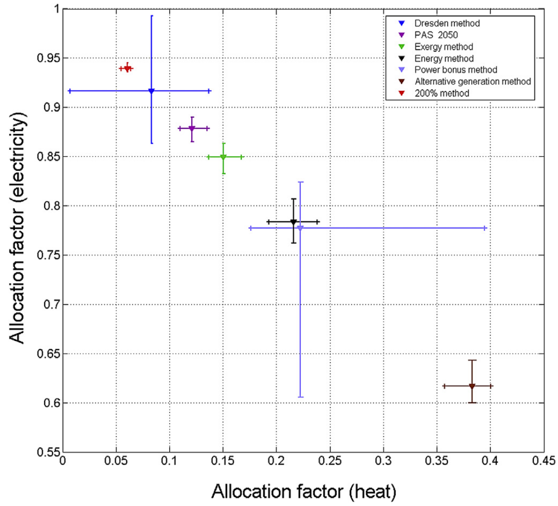

Fig. 8. Sensitivity of allocation factors for heat and electricity production.

The current analysis was focused on CHP with CCPP technology. Therefore, the results are relevant for the CHP with the CCPP technology for the same configuration, but different operation data. In the case of CHP without supplementary firing technology or gas turbine cycle technology, the final result presented in the Fig. 8 might be different.

6. Conclusions

The different methodologies for the allocation of CO_2 emissions for heat and power production in the CCPP have been presented and analyzed. The allocation methods were combined with a parametric study of the CCPP and this showed that different allocation methods produce different results. For example, the fuel allocation for heat at design conditions for the alternative generation method was 38.3%, while using the 200% method this value was 6%, and for the power bonus method was 22.3%. This indicated that the choice of allocation method is very important for the development of cogeneration technology in relation to heat and power distribution systems. The 200% method gives the lowest CO_2 allocation for heat, indicating that the heat produced in the CCPP is

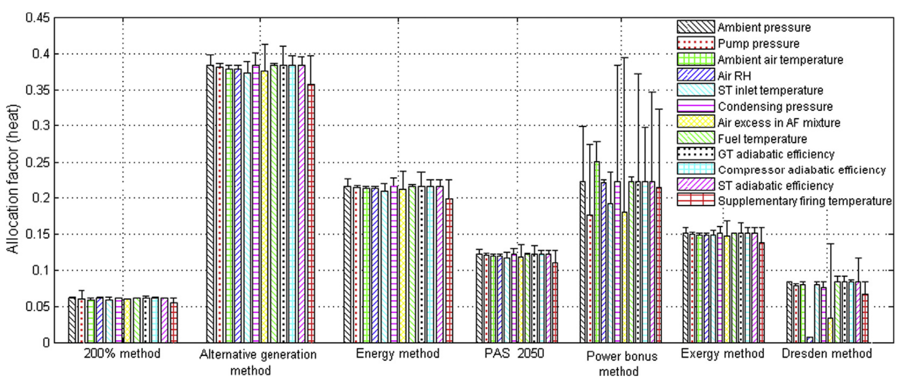

Fig. 7. Heat allocation factors for analyzed methods.

the most environmentally friendly. On the other hand, the alternative generation method allocates a higher amount of emission to heat, which is not beneficial from a DH point of view. Among all the presented methods, the most sensitive was the power bonus method, which is promoted as the main method for emissions' allocation in the EU. The results showed the highest variance in allocation factors for both electricity and heat, ranging from 11% to 21% compared to the design case. In other methods, the variation was negligible: around 1–3%. All these indicated that the CO_2 allocation was difficult to estimate under the annual heat and electricity load variations. Therefore, we can conclude that emissions allocated with the power bonus method cannot be fixed continuously as is stated in standard EN 15316. The solution can be efficiency values, seasonal averages as a basis for nominal intensities or methods with small variation. This study showed that the decision regarding choosing the allocation method should be carefully analyzed for implementation in the standards and different policies. It is important to enable a proper allocation of CO_2 emissions and the promotion of environmental benefits from cogeneration technology for DH and power distribution systems. The results obtained in this study can be used by the designers of CHP systems and policy makers as a tool for developing an emission trading system for CHP plants and for the pricing of heat and power.

Appendix A. Supplementary data

Supplementary data related to this article can be found at http://dx.doi.org/10.1016/j.applthermaleng.2014.11.019.

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Implementation of CCPP for energy supply of future building stock

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HIGHLIGHTS

- Sensitivity analysis was performed under changeable heat demand profiles.
- Results found high dependency of plant performance from heat load in DH system.
- CCPP as the most energy efficient conversion plant is suitable for heat supply of high heat density areas.
- CCPP operate with difficulties in energy supply of low heat density areas.
- The power production in CCPP is not influenced by the supply temperature control.

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ABSTRACT

European Directive 2010/31/EU stated that by the end of 2020 all new buildings should be nearly Zero-Energy Buildings (nZEB). Since such buildings require a low quantity of energy for heating, they can utilize energy from the return line of a District Heating (DH) system. Further, this new type of buildings can successfully be integrated into the fourth generation of heat distribution technology, which is the new trend in the DH industry. On the other hand, existing building stock has a service lifetime of around 50 years, indicating that the required supply temperature in the DH system cannot be lowered beneath a certain level. Hereafter, together with new types of buildings and different policies, this could lead to changes in heat demand profiles of the DH system.

The above-mentioned situation in the building market will lead to changes in heat load profiles and unavoidably influence the performance indicators of energy conversion units. Therefore, the focus in this paper was devoted to operation analysis of ethanol-based a Combined Heat and Power (CHP) plant with combined cycle technology under changeable demand conditions and different temperature levels in the DH system.

In this paper different temperature levels of the DH system together with two temperature control strategies were considered. Further, the analysis included different heat demand profiles. The analysis was performed in Aspen HYSYS simulation software and data post processing was made by MATLAB. The results found that effective plant operation is highly dependent on heat load profile. Temperature control strategies did not induce a significant change in overall power production in CHP. The decrease in the supply temperature did not show a significant impact on plant performance. However, increase in temperature difference between supply and return lines led to higher power production and better overall plant performance. Further, it was concluded that it is rather difficult to operate CCPP connected to low energy building stock. This means that the CCPP is suitable for high-density heat areas, while it has poor energy performance indicators in low heat density areas.

1. Introduction

Biomass Combined Heat and Power (CHP) plants are often seen as an efficient way to reduce greenhouse gas emissions due to their very low CO_2 emissions [1,2]. One of the known CHPs' modifications employs combined cycle technology and named as Combined Cycle Power Plant (CCPP). Such plants reaches a higher average fuel utilization of about 80%. In addition, this technology allows to reach the primary energy savings between 9% and 20% in comparison to the separate generation of power and heat [3]. Nowadays, more attention is devoted to appliance of renewable fuels in energy production plants. For this reason, bioethanol, or

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Nomenclature

$f(-)$	load factor, average heat load during heating season
$f_0(-)$	load factor at the beginning of a heating season
$k(-)$	radiator-type coefficient
$n(\mathrm{h})$	operation hours of CCPP
$P_{comp,i}(\mathrm{GWh})$	power production in CCPP with outdoor compensation temperature control
$P_{const,i}(\mathrm{GWh})$	power production in CCPP with constant temperature control
$\dot{Q}_h^d(\mathrm{kW})$	design value of the heat load
$\dot{Q}_h(\mathrm{kW})$	heat rate
$T_{ex}^d(^\circ\mathrm{C})$	design outdoor temperature
$T_{ex}(^\circ\mathrm{C})$	outdoor temperature
$T_t^d(^\circ\mathrm{C})$	threshold temperature
$T_{in}(^\circ\mathrm{C})$	indoor temperature in the building
$T_r(^\circ\mathrm{C})$	return temperature in the DH network
$T_s(^\circ\mathrm{C})$	supply temperature in the DH network
$T_1(^\circ\mathrm{C})$	supply temperature to DH system
$T_2(^\circ\mathrm{C})$	return temperature from DH system
$T_{2d}(^\circ\mathrm{C})$	DH design temperature in return line
$T_3(^\circ\mathrm{C})$	supply temperature to hydronic heating system
$T_{3d}(^\circ\mathrm{C})$	DH design temperature in supply line
$u_m(-)$	mixing coefficient
$\tau(\mathrm{h})$	duration of heating season
$\tau_i(\mathrm{h})$	the ith value of heating hours
$\Delta T(^\circ\mathrm{C})$	temperature difference between supply and return lines in DH system

ethanol derived from biomass, has been recognized as a potential alternative to fossil fuels [4,5]. Bioethanol driven CCPP reaches significantly higher emissions reductions in comparison to the carbon-intensive fossil fuel technology [3].

The aim of this study was to investigate the operation of an ethanol-based CCPP, under changeable heat demand profiles, together with the possibilities of lowering temperature levels in a DH system. Due to strict energy requirements on new building constructions, energy units connected to supply DH systems, such as CCPPs, can demonstrate significant fluctuations in performance indicators. This is especially relevant because of the low energy buildings that are already connected, or will be connected, to the DH in the future. Due to low annual heat demand and high heat demand peaks during extreme outdoor conditions for such buildings, more insight needs to be devoted to this problem of the operation of CCPPs. Since the aim of this work was to analyze the performance of a CCPP at different loads and temperatures, a brief overview on building load trends and temperatures in DH is given in the following text.

The estimation of heat demands is a complex task, especially for large-scale systems involving many heat consumers and consumer types [6]. There are many parameters, which could have an effect on heat load prediction in a DH system. Different authors implemented algorithms based on yearly observations for heat load prediction. Werner in [7] described a model based on physical theory. Different additive elements, for example wind speed and global radiation, were added to the heat load model. Aronsson in [8] created a model which was based on [7] but with improvements. Arvaston in [9] and Gadd and Werner in [10] concluded that social and physical parameters of the customers has the greatest effect on heat demand, while different additive elements investigated by mentioned researches play a secondary role.

Retrofit of a DH system can affect heat load variation, since physical components such as pipe insulation or distribution pipes play an important role in the overall heat balance of a DH system. As mentioned in [11], typical relative heat losses in ordinary DH systems are 8–15% in Western and Northern Europe. The corresponding level is about 15–12% in Eastern Europe. Errors and deviations in customer substations and internal heating systems in buildings have a significant impact on the operation and load of heat supply plants. One of the future trends in the DH industry is smart DH systems [11]. This concept is based on the smart monitoring of DH system that will allow all the substations to be monitored automatically without enormous labor input. This can lead to smart load control and consequently to load decrease. However, it is not only the residential sector can be connected to the DH system. With the increase in electricity prices, the industrial sector can shift from electrical heating to DH. If industries use only DH services for space heating and hot tap water, then the integration effect will result in an additional load to base load plant, since the summer heat load is less than the plant's minimum operating heat load. Consequently, industrial processes can be successfully integrated into the DH systems, with benefits to base load plants such as CHP systems [12].

European Directive 2010/31/EU [13] stated that by the end of 2020 all new buildings should be nearly Zero-Energy Buildings (nZEB) and Member States should achieve cost-optimal levels by ensuring minimum energy performance requirements for buildings [14]. The change in the heat duration curve for the heat energy supply unit is inevitable with more such buildings being connected to DH.

However, the entire building sector cannot consist of nZEB and passive houses. Therefore, the penetration of these buildings into the building stock will show an effect on use patterns in the future. The modernization of existing buildings has decreased the heat losses in EU countries, reducing the share of consumption of heat for space heating purposes [15]. This process has been already accomplished in Western Europe, leading to an increased effectivity in the heat supply for consumers and decreasing heat consumption throughout the year [16]. Werner and Olsson in [17] described the possibility of reducing the heat load variation for peak demand by using buildings connected to the DH system as a means of heat storage. The conclusion was that the estimated time constants were often well above the assumed 100 hours for all types of buildings. Applying this strategy, an immediate increase in heat load during daytime temperature variation can be avoided for peak load energy units. The possibility of optimizing and reducing peak loads in DH systems, applying remote meters and control strategies, was described by Drysdale in [18].

From the beginning of the DH age in the world, three generations of DH distribution technology were developed [11]. In the earliest systems, steam was used as a heat carrier. Later on, water became the heat carrier. The materials used in the distribution system propagated different temperature and pressure levels. Nowadays, DH systems are predominantly built according to third generation principles. However, different countries have different requirements for supply and return temperatures in the DH system. In Sweden, for example, for many years the temperatures in hydronic systems were 80–60 °C, while in Germany, these values were greater and sometimes reached higher than 100 °C in the supply line; in Eastern Europe it could even reach 150 °C. With the third generation of DH distribution technology, the reduction of distribution heat losses took place. Together with new building codes, these led to a decrease in supply and return temperatures in the DH network for areas with new types of buildings.

Considering different Refs. [19–21], it can be noticed that for different types of buildings there are different requirements for temperature levels. Authors in [22] showed that even in the non-renovated houses in Denmark, it is enough to supply DH water at a temperature of 67 °C. International studies [22–25] showed that there is an over-sizing of around 20–30% of DH systems and also of radiator systems, since designers want to be sure that the system provides enough heat. This can be the reason for further reductions in DH temperatures.

Future grids, with the fourth generation of DH technology, may use low-temperature heat distribution networks with normal distribution temperatures of 50–20 °C as an annual average [26].

However, in reality, it is not an easy task to implement the ideas regarding low temperature levels in DH systems, when combining them with heat energy units like CHP. Different publications devoted to low temperature DH mostly deal with future buildings and not the existing building stock, which is expected to constitute the major part of the heat demand for many decades to come [26]. This means that without prepared infrastructure, it is almost impossible to bring ideas of low temperature DH to life. Different customers have different heat load characteristics and it is therefore sometimes rather complicated to satisfy all customers' demands with one temperature level lower than 80 °C in the supply line of a DH system. One should also take into account the different types of structures being built during recent decades, as well as buildings at random stages of renovation [21,22,27,28]. At the same time, a DH system should be competitive and cost-effective.

Nevertheless, the situation is different with the return temperature levels in the DH system. For certain types of CHP systems a high return temperature in the DH network could lead to a decrease in plant efficiency or it could be economically inefficient, depending on power and heat outputs and the configuration of the plant. A higher return temperature results in higher heat losses, less energy stored in thermal storage, if that is used, and lower efficiency of heat generation. These facts make DH less attractive [29]. For these reasons, the authors considered that for DH systems connected to CHP units, a reduction in return water temperature should be implemented, leading to an increase in the temperature difference between supply and return lines. One of the ways to perform this is by the implementation of the "temperature cascading" [30] principle, suggested by researchers in [31]. This idea implies the connection of customers with low heat consumption to the return pipes, which is relevant for passive houses and nZEB buildings [32,33]. Applying the temperature cascading principle and new substation schemes, as in [34], it is possible to obtain 20 °C in the return line of a DH system and, with future improvements in buildings, insulation properties and distribution systems, even 15 °C.

Different heat load patterns from the customer side together with climate change and global warming [35,36] can significantly decrease the profitability of energy supply units in DH systems. As stated in [36], a good practice consists of designing the CHP plant according to the minimum heat demand. However, in the case of DH networks, the minimum heat demand is very low and does not justify the installation of a CHP plant. Then the simple question emerges: How should DH companies react in the situation when the CHP unit is already installed, but the heat demand profile shows significant variation throughout the years? Therefore, the need for operation analysis of CHP systems with integrated DH systems and changeable heat demand profiles arises.

Different studies have been carried out on CHP plants and DH systems. However, not many of them focuses on detail operation of energy production unit. The problem of reduction of heat demand is vital in CCPP applications, particularly facing the improvements in building energy saving measures. Therefore, the innovation of this paper is increased knowledge on CCPP plant operation considering operation issues: temperature levels in DH system, temperature control strategy and different heat load patterns of DH system. CCPP plant is considered to have the highest energy efficiency. However, due to unfavorable operation conditions induced by the future building energy saving trends, the performance of a CCPP can be significantly changed. Hence, the innovation of this paper is information on the changed performance of a CCPP.

This paper is divided into the following sections: Section 2 introduces the methodology for the calculation of the temperature control strategy and heat duration curve in the DH system and process simulation of a CCPP; Section 3 describes the model and details of the process in the CCPP. Results of the analysis are discussed in Section 4. The last two sections outline conclusions on the results from Section 4 and remarks on the possibilities for future work.

2. Methodology

The methodology presented in this section describes calculation techniques for the heat duration curve, an outdoor temperature compensation strategy within a DH system and a simulation method for the CCPP.

2.1. Estimation of heat duration curve

The heat duration curves used for estimating energy consumption in DH are individual for each DH system. The number of heating hours needed to supply customers depends on geographical location, climatic conditions and outdoor temperature when the heating season begins, and building types connected to DH. Further, an issue with the construction of heat duration curve represents the major operation problem for DH companies. It is not fully possible to plan and predict heat supply to the customers and the fuel needed for the CCPP during an operation year [11].

Therefore, an analytical expression can be used for the calculation of the heat duration curve, applying the methodology presented in [37]. The final equation has the following representation:

$$\frac{\dot{Q}_h}{\dot{Q}_h^d} = 1 - (1 - f_0) \cdot \left(\frac{\tau_i}{\tau}\right)^{\frac{f - f_0}{1 - f}} \tag{1}$$

where f is a load factor (average relative heat load during the heating season) and f_0 is a load factor at the beginning of a heating season; τ and τ_i are duration of the heating season and the ith value of heating hours; $\dot{Q}_h$ is the heat rate, and $\dot{Q}_h^d$ is the design value of the heat rate for the minimum external temperature.

This expression is called Rossander's equation. As described in [37], it allows different heat duration curves to be built for different data sets.

In the current analysis, the authors assumed that the design external temperature is equal to $T_{ex}^d = -19$ °C, while the threshold temperature or beginning of the heating season is assumed when the outdoor temperature is equal to $T_t^d = +10$ °C. Rossander's analytical heat duration curve was built for these temperature conditions and is depicted in Fig. 2.

In this paper three duration curves are used for analysis. Two are based on measurements and one is the product of the authors' assumption. In order to justify selected duration curves, an analytical heat duration curve for predefined climatic conditions was used for comparison.

2.2. Supply water temperature control

There are different control options for supply and return temperatures in DH systems [11]. Such methods can be based on a constant supply temperature combined with local flow control, or a constant flow rate in combination with the control of supply temperature, or both. The control of the flow or supply temperature can be based on the feedback (indoor temperature) or the feed forward (outdoor temperature) control signal [38]. In this study, two control options were investigated: constant supply temperature from energy unit and outdoor temperature compensation.

In the case of the constant temperature control strategy, the supply temperature to the DH is set in the heat energy unit. Supply temperature remains constant during operation, while the control is performed by the adjustment of mass flow rate of water carrier to the customers. In the case of the outdoor temperature compensated curve, the highest supply temperature is reached at the design outdoor temperature. The return temperature is the result of the control strategy in customer substations and overall mixing of flows from all substations.

Outdoor temperature compensation in the DH network can be evaluated based on the methodology presented in [39]. In this methodology, the expected supply temperature in the DH network T_1 can be estimated as:

$$T_1 = (1 + u_m) \cdot T_3 - u_m \cdot T_2 \quad (2)$$

where T_2 and T_3 are expected DH return temperature and the expected supply temperature in the hydronic heating system in a building. u_m is the mixing coefficient. Then, the expected DH return temperature can be estimated as:

$$T_2 = T_3 - (T_{3d} - T_{2d}) \cdot \left(\frac{T_{in} - T_{ex}}{T_{in} - T_{ex}^d}\right) \quad (3)$$

where T_{ex}^d, T_{ex}, T_{in} are design outdoor temperature, outdoor temperature, and indoor temperature in the building, respectively. T_{2d} and T_{3d} are DH design temperatures in the return and supply lines, respectively.

The mixing coefficient can be evaluated as:

$$u_m = \frac{T_{3d} - T_1}{T_1 - T_2} \quad (4)$$

The expected supply temperature for the hydronic heating system can be estimated as:

$$T_3 = T_{in} + 0.5 \cdot (T_{3d} - T_{2d}) \cdot \frac{T_{in} - T_{ex}}{T_{in} - T_{ex}^d} + 0.5 \cdot (T_{3d} + T_{2d} - 2 \cdot T_{in}) \cdot \left(\frac{T_{in} - T_{ex}}{T_{in} - T_{ex}^d}\right)^{\frac{1}{1+k}} \quad (5)$$

Based on this methodology, Fig. 1 shows curves for the water temperatures in DH. The calculation was performed for a temperature level in the DH network of 100–45 °C.

2.3. CCPP simulation

In this paper heat and electricity production in the CCPP were simulated. The simulation process represents a transient calculation with a step size of one hour. The model of the CCPP was based on thermodynamic principles and performed in the Aspen HYSYS process simulation software. This commercial software is available for purchase. Different authors have performed analyses in this software and their publications validated the accuracy of the models being built in this software [40–43]. Data post processing was carried out using by MATLAB [44].

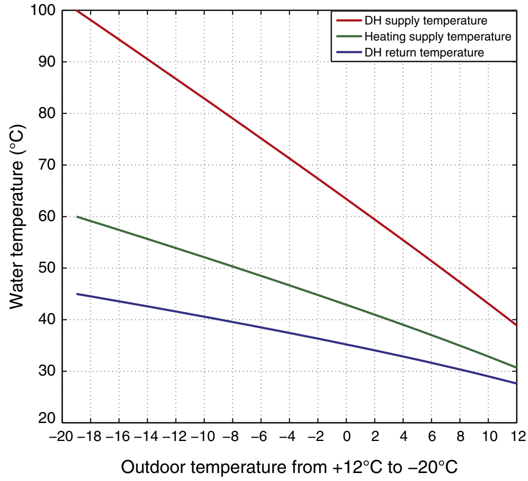

Fig. 1. Outdoor temperature compensated curves.

3. Case study

In this paper a small-scale DH system with the maximum heat load of 14 MW was analyzed. Three heat demand profiles were considered to illustrate the heat use in the DH system. The analyzed duration curves are depicted in Fig. 2.

Case 1 presents the heat duration curve during a regular year in the analyzed location and is used as a reference year. Case 2 presents the heat duration curve under a higher occupancy level and lower outdoor temperature. The heat duration curves in Case 1 and Case 2 are the result of measurements, which were carried out with the help of facilities of Norwegian University of Science and Technology (NTNU). Case 3 represents the situation for future energy consumption, taking into account newly-built passive houses and nZEB with low heat energy use throughout the year and high peaks during the winter time. Case

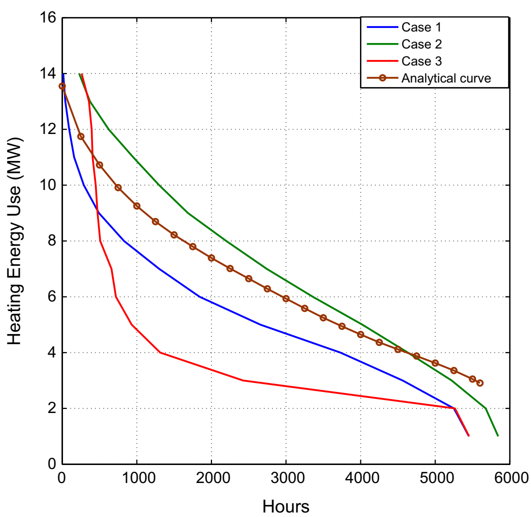

Fig. 2. Heat duration curve.

Table 1
Heat load characteristics.

	Rossander's curve	Case 1	Case 2	Case 3
Heating energy use (GWh)	37.1	24.36	36.43	17.18
Average DH load (MW)	6.63	4.47	6.22	3.15
Heat rate under maximum hour's frequency (MW)	3	4	5	2
Duration of maximum hour's frequency (hours)	756	1072	664	2840
Heat rate under minimum hour's frequency (MW)	13	14	13	11
Duration of minimum hour's frequency (hours)	147	14	146	12
Utilization time (hours)	2650	1740	2459	1227

Table 2
Design point parameters of the CCPP.

Parameter	Value
Ambient pressure	101 kPa
Air relative humidity	60%
Ambient air temperature	+15 °C
Pump pressure	100 bar
Steam turbine inlet temperature	+540 °C
Condensing pressure	0.05 bar
Air excess in air–fuel mixture	4.0
Fuel temperature	+15 °C
Gas turbine adiabatic efficiency	0.9
Steam turbine adiabatic efficiency	0.9
Compressor adiabatic efficiency	0.9
Gas turbine inlet temperature	+1096 °C
Supply temperature in DH system	+100 °C
Return temperature in DH system	+45 °C

3 is the result of an assumption and is represented by a decrease in heating energy use of almost 30% in comparison with the reference year. The heat load characteristics of the analyzed cases are summarized in Table 1.

From Fig. 2 and Table 1, it can be seen that the analytical duration curve has the highest values of calculated heating energy use and average DH load. At the same time, the analytical curve gave values close to Case 2, which represents the scenario of higher occupancy and lower outdoor temperature. Rossander's equation was developed in the twentieth century, when climatic conditions were more severe and the length of the heating season was longer. Hence, the analytical curve shows the maximum possible heat energy use for the analyzed region. Later on, with the development of building codes and new energy policies, the heat energy use in buildings decreased. This observation is very important, since heat energy units should be capable of withstanding the heat load decrease coming from customers.

The CHP system with CCPP technology was employed as an energy source for the DH system. The configuration of the CCPP system was an ethanol-driven gas turbine cycle (GTC), using exhaust heat recovery to drive a bottoming steam cycle (STC), with steam extraction for DH. Ethanol-based CCPPs are well known, and different authors have performed studies on such systems [45–47]. The schematic layout of the system is presented in Fig. 3, and design parameters are summarized in Table 2.

The lower heating value (LHV) of the ethanol is 26.45 MJ/kg. The air and fuel were supplied to the reactor after a two-stage compression system, that included a low pressure compressor (LPC) and the high pressure compressor (HPC). The air excess coefficient, α, was set to be 4.0 in the air–fuel mixture.

The GTC was represented by two units; one is a high pressure gas turbine (HPGT) and the other is a low pressure gas turbine (LPGT); see Fig. 3. The temperature of the flue gases entering the gas turbine was set 1096 °C. The entering pressure of flue gases in the HPGT was 15 bar. The pressure before the LPGT was 6 bar. The leaving pressure was 1.5 bar, which is slightly higher than ambient conditions.

The heat recovery steam generator (HRSG) has one steam pressure level. The parameters of the live steam entering the steam cycle were: $T = 540$ °C, $p = 100$ bar. The STC involved a high pressure steam turbine (HPST), an intermediate pressure steam turbine (IPST), and a low pressure steam turbine (LPST). The entering parameters of the working medium in the IPST were pressure of 12 bar and temperature 245 °C. In the LPST, the steam condenses up to a pressure of 0.05 bar.

There is one extraction in the STC for DH purposes. The mass flow rate of water from the DH is satisfied with the heat exchange units. The DH system was fed from the IPST. The steam extraction occurred at a pressure of 10 bar.

Fig. 4 and Fig. 5 show changes in the plant performance due to change in air temperature, heat load and elevation. These curves were used for comparison with yearly operation values in the CCPP.

Fig. 4a shows the relative efficiency of the steam process, power process, and combined-cycle process as a function of ambient air

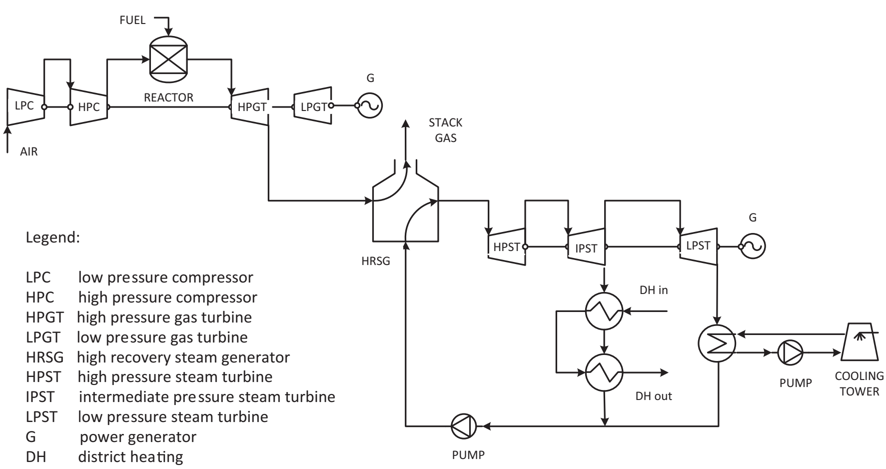

Fig. 3. Schematic of the CCPP.

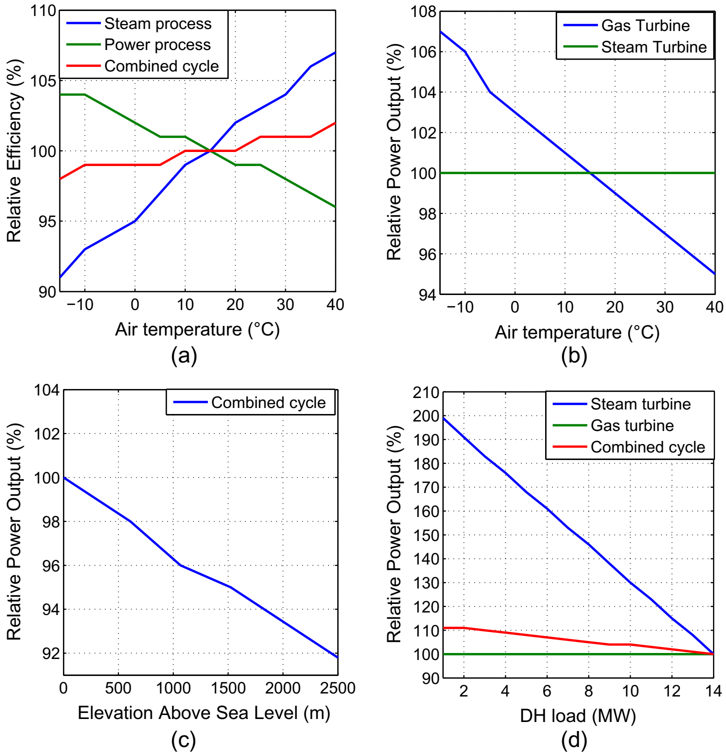

Fig. 4. CCPP design conditions.

temperature, while other ambient conditions and condenser pressure remain unchanged. The curves presented on the figures were based on a full DH load of 14 MW. The reference value of the ambient temperature was fixed at +15 °C [48] for the design conditions. The reference elevation level is 0 m, which corresponds to ambient pressure of 1.013 bar. However, these values are changeable during the year and have a significant impact on plant performance.

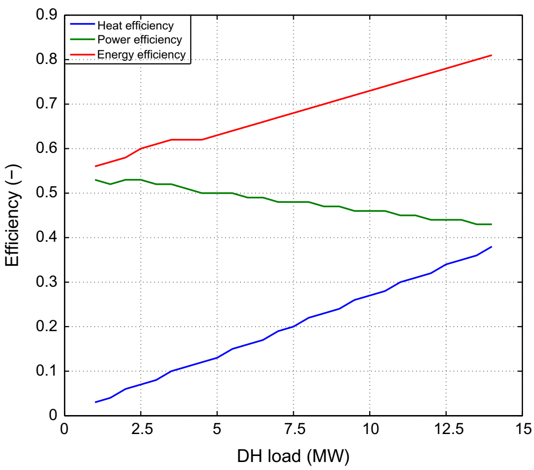

Fig. 5. Energy efficiencies of the analyzed CCPP.

Fig. 4b shows the relative power output of a gas turbine and steam turbine in the CCPP. The relative power output of the steam turbine remains constant, since there is no change in the DH load. Fig. 4c presents the relative power output of the combined process due to the elevation above sea level. As can be seen, with an increase in elevation, the power output of the CCPP decreases due to the change in air density. Fig. 4d illustrates relative power output versus heat load in the DH system. The gas turbine cycle remains constant, since the change is only made to heat output in the range of 1 MW to 14 MW.

Finally, Fig. 5. presents the heat, power and energy efficiencies of the CCPP for different heat loads in the DH system.

It is important to have design plant characteristics, as presented in Fig. 4 and Fig. 5 Such curves show the theoretical maximum and minimum of possible plant performance. However, these values are given for a certain reference point as previously discussed). In reality, when CHP plants operate under changeable seasonal heat loads, these parameters are far from the design point. The comparison with yearly operation values will shed light on the issue of variation in heat load profiles.

Since the aim of this study was to analyze how the CCPP could be implemented for future building areas, a range of different supply and return temperatures was considered.

The temperature levels were chosen based on the review of the DH generations. A detailed explanation about the choice of temperature levels is provided in Section 1 of this paper. Taking into account different studies, conclusions, and the recommendations from these studies about temperature levels in DH systems, the temperatures presented in Table 3 were studied.

Table 3
Analyzed temperature levels in DH network.

Explanation	Supply temperature in DH network Ts (°C)	Return temperature in DH network Tr (°C)
2nd generation of distribution technology with medium return temperature	100	45
2nd generation of distribution technology with low return temperature	100	30
2nd generation of distribution technology with ultra-low return temperature	100	15
3rd generation of distribution technology with medium return temperature	90	45
3rd generation of distribution technology with low return temperature	90	30
3rd generation of distribution technology with ultra-low return temperature	90	15
3rd generation of distribution technology with medium return temperature	80	45
3rd generation of distribution technology with low return temperature	80	30
3rd generation of distribution technology with ultra-low return temperature	80	15

4. Results

4.1. Energy conversion in CCPP under different heat loads

Power productions for two temperature control strategies in the DH system are shown in Fig. 6. The shortcut "const" shows values for constant temperature control, while "comp" shows values for outdoor temperature compensated control.

In Fig. 6 it can be seen that the difference in power production of the CCPP, due to changes in control strategies with different supply and return temperatures, was not significant. Therefore, due to the enormous computational time needed for CCPP simulation over the entire year under two control strategies and at different temperature levels, the authors decided to focus only on the most relevant control option. For this reason, in order to identify the difference in the CCPP power production, the relative deviations between obtained results were calculated as:

$$\Delta P = \frac{1}{n} \cdot \sum_{i=1}^{n} \frac{(P_{comp,i} - P_{const,i})}{P_{const,i}} \qquad (6)$$

where $P_{const,i}$ and $P_{comp,i}$ are the values of power production in the CCPP with constant temperature control and outdoor temperature compensation strategies at time step i. n is the number of operation hours of the CCPP.

Fig. 7 represents the relative deviation between the power productions in the CCPP for two different control strategies, with respect to the constant control strategy.

The difference between power productions for different temperature levels in DH systems is relatively small, under 3%, as shown in Fig. 7. The smallest deviation occurred in the temperature range of 80–30 °C, while the largest in 100–45 °C.

Due to different control strategies applied in the DH system, the annual amount of generated electricity in the CCPP also differs. Fig. 8 presents the annual power production for two strategies and different temperatures.

Due to different control strategies, the yearly difference for the temperature range of 80–30 °C was 1.35 GWh and for 100–45 °C this value was 4.31 GWh of produced electricity.

For constant temperature control, the power production increases with the increase in supply temperature in the DH

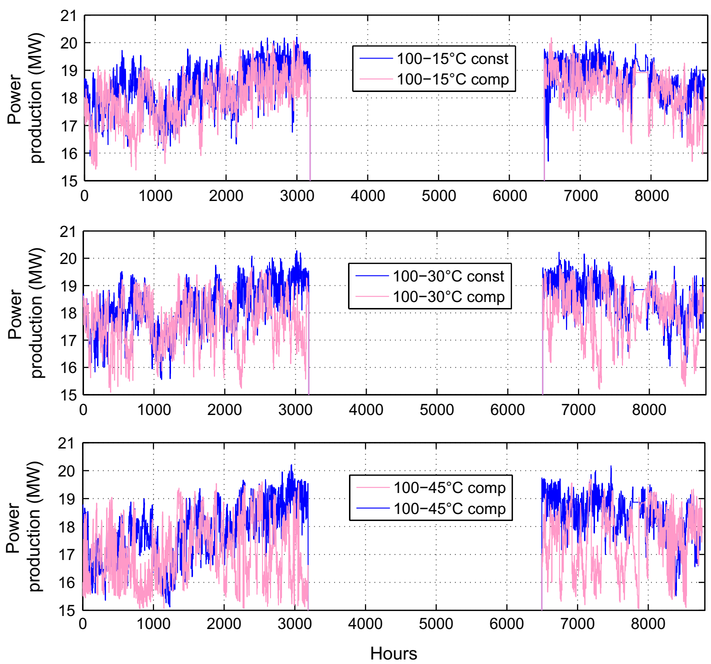

Fig. 6. Power production in the CCPP, for different control strategies and return temperatures in the DH system.

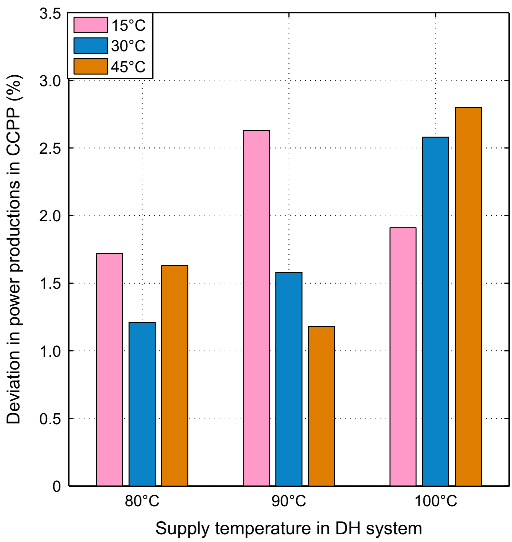

Fig. 7. Average deviation between two data sets.

system for the same return temperature, while for temperature compensated control, the power production decreases with the increase in supply temperature in the DH system; see Fig. 8.

Fig. 8 reveals the fact that power production in the CCPP is sensitive to change in supply temperature in the DH system and also to the difference in change in temperature between supply and return lines. The greater the temperature difference in the DH, the more power is produced, regardless of which temperature control option is used.

Due to the relatively small difference in power production in the CCPP between the two control strategies applied in the DH system, see Fig. 7 and Fig. 8, in the further analysis only the constant supply temperature strategy was analyzed. Therefore, all future results are related to the constant supply temperature strategy.

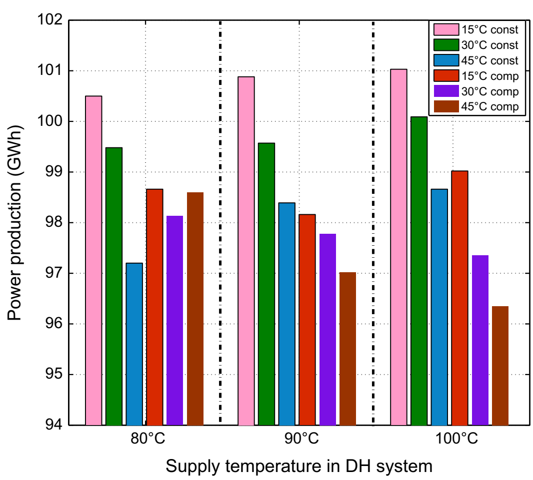

Fig. 8. Comparison of yearly power production in the CCPP for different supply and return temperatures and two control strategies in the DH system.

Fig. 9 presents annual power production under different load profiles and temperature levels applied in the DH system.

As can be seen from Fig. 9 and Table 2, Case 2 has a uniform distribution of heating hours between DH load, which resulted in higher power production in comparison with Cases 1 and 3. Further, a different trend was observed for the power production for Case 3 due to a change in the temperature levels; see Fig. 9. Different amounts of generated electricity during the year for Cases 1–3 could be explained by the different variation of heat load in the DH system.

For the supply temperature of 90 °C, power production was decreased due to a decrease in the return temperature from 45 °C to 15 °C. However, in situations where the supply temperature was equal to 80 °C and 100 °C, the trend was different. For Case 3, the difference in power production for the supply temperature of 100 °C was negligible for all return temperatures, while for 80 °C the highest value was obtained for a return temperature of 30 °C. Further, the highest power production was obtained for the temperature level of 100–15 °C in Cases 1 and 2, indicating that the greatest temperature difference induced high power production. In the case of an increase in DH return temperature, the internal vapor pressure of the heat exchanger also increases. This phenomenon can cause steam pressure ascension, and the high steam energy is used for more heat output [49].

Fig. 10 shows heat efficiency in the analyzed CCPP depending on frequency of heat load hours in the DH system. Fig. 10 shows minimum, maximum, and median values of heat efficiency for the analyzed scenarios. The diagram is the result of hour-by-hour simulation of CCPP operation under given load in the DH system (presented by Cases 1–3).

Despite the fact that, in Case 3 there is the highest number of hours (2840) with the heat load of 2 MW, see Table 2, the heat efficiency of the CCPP is quite low. The highest heat efficiency is found for the maximum heat load of 14 MW for all cases. The higher the DH load, the higher the plant capacity utilization and heat efficiency.

4.2. CCPP performance under different load and temperature levels

Fig. 11 presents average system performance characteristics for the analyzed CCPP, such as power efficiency, heat efficiency, and energy efficiency.

Fig. 11 reveals that the results of average efficiencies in the CCPP varied among cases due to different load distribution. The calculated efficiencies were highly dependent on the DH heat load

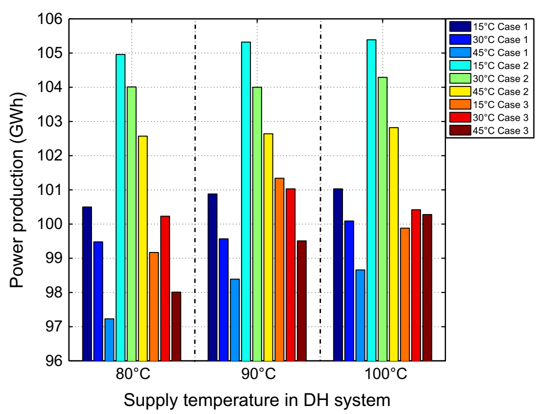

Fig. 9. Power production in the CCPP throughout the year.

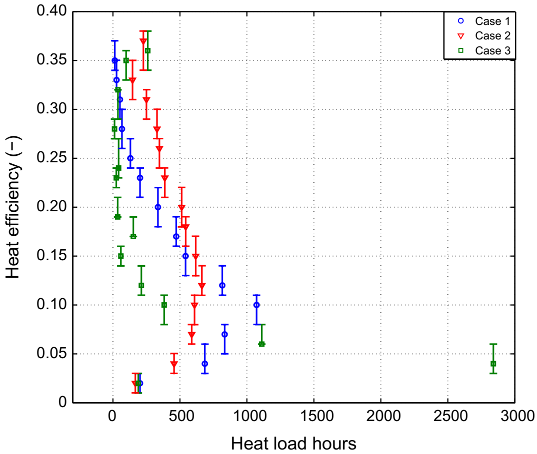

Fig. 10. Heat efficiency in the CCPP.

distribution. Fig. 11 shows that uniform distribution of heat load resulted in better plant operation throughout the year. The average heat efficiency for Case 2 is higher than for Cases 1 and 3. However, for different levels of supply and return temperatures, for all cases the change in power efficiency and energy efficiencies was in the range of 1–2%, which is quite small. The maximum average energy efficiency was in the range of 57–65% for all cases. The obtained operation values were rather different from the design conditions; see Fig. 4 and Fig. 5. This observation indicated that plant was poorly loaded by the DH system throughout the year. This information should be considered while running the CCPP with the DH system.

Fig. 12 shows the change in power efficiency and DH load for Case 2 due to different heat load and different supply and return temperatures. Further, variations in the energy efficiencies are also shown. The figure shows minimum, median, and maximum values obtained during simulations for corresponding heat loads.

Recalling Fig. 2 and Table 1, for Case 2, the highest number of heating hours occurred for the load equal to 5 MW, corresponding to 664 hours during the operation year, while the minimum of heating hours occurred at 13 MW (146 hours). The median value of power efficiency for the DH load of 13 MW was equal to 0.44. This value was the same for all analyzed temperature levels. For the 5 MW of DH load, the median power efficiency was in the range of 0.47–0.49, depending on temperature level used in the DH system. The diagram shows that power efficiency was sensitive to temperature difference in the DH system: the higher the temperature difference the higher the power efficiency.

Fig. 13 shows the distribution of the energy efficiency throughout the analyzed DH load interval for Case 2.

For the DH load of 5 MW, the energy efficiency was 0.60–0.61. Meanwhile, for the 13 MW DH load, the energy efficiency showed a value of 0.77–0.78, depending on different temperature levels in the DH system. Fig. 13 reveals that the difference in energy efficiencies was negligible for separate DH loads and analyzed temperature levels within subplots. However, in the case of continuous hour-by-hour operation of the CCPP, the deviation in energy efficiency is in the range of 2–10% between minimum and maximum values. This can be explained by rapid change of DH load, which results in an immediate response in fuel input and power production within the CCPP.

Of special interest is Case 3, since it reflects one of the possible scenarios in the future when the low energy buildings will share a certain part of the building stock. Low heating energy use makes

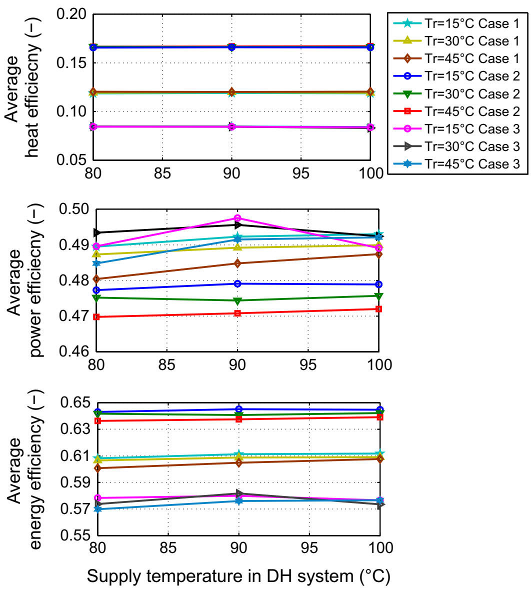

Fig. 11. Average heat, power, and energy efficiencies.

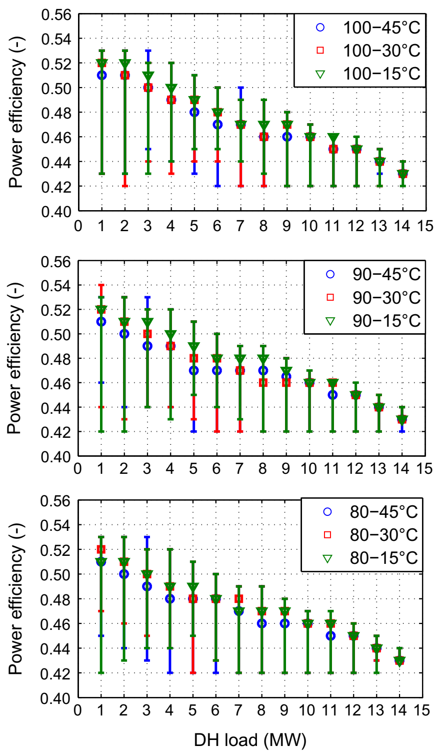

Fig. 12. Power efficiency for Case 2.

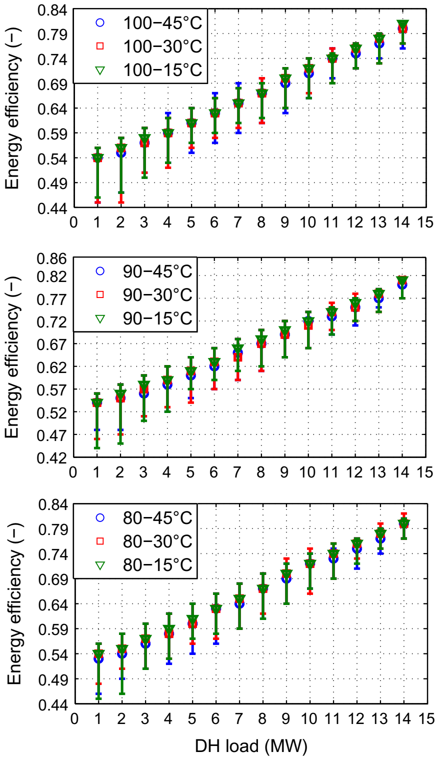

Fig. 13. Energy efficiency of the CCPP for Case 2.

such buildings unattractive for supply by large heat production units. This is the main drawback that heat production units must be capable of overcoming.

The power efficiency for 11 MW of DH load was 0.45 for supply temperature of 80 °C and return temperatures of 30 °C and 45 °C, while for 15 °C this value was 0.46. For supply temperatures of 90 °C and 100 °C and corresponding return temperatures of 45 °C and 30 °C, the power efficiency was 0.44, while for 15 °C this value was 0.45. The analysis showed the value of power efficiency equal to 0.5 for 2 MW of DH load, with supply temperatures of 100 °C and 90 °C and return of 45 °C. For the same level of supply temperatures and for 30 °C and 15 °C of return temperatures, the power efficiency increased by 2%. This indicated that temperature difference had a positive influence on power production within the STC. For a supply temperature of 80 °C, the maximum power efficiencies of 0.5 were obtained for return temperatures of 30 °C and 15 °C. For 45 °C, the power efficiency decreased by 2% and constituted 0.49.

The deviations between median values of energy efficiencies in analyzed DH load range were higher in comparison with Case 2, with respect to temperature levels in the DH system. For 11 MW of DH load and supply temperature of 100 °C, the change in return temperature from 45 °C to 30 °C and then 15 °C, resulted in 0.70, 0.72 and 0.73 of energy efficiency. For a supply temperature of 90 °C, these values were in the range of 0.75–0.77. However, for 2 MW of the DH load and supply temperature of 100 °C, the energy efficiency was equal to 0.55 for all return temperatures. For a supply temperature of 90 °C, results showed a value of 0.55 for return temperatures of 45 °C and 30 °C and 0.56 for 15 °C. For a supply temperature of 80 °C and a return of 45 °C, the energy efficiency was 0.54, while for 30 °C and 15 °C it increased and constituted 0.56.

The difference in median values of power efficiency and energy efficiency between cases was not very large. The deviation between minimum and maximum values of efficiencies varied from 2% to 10% depending on heat load rate. Meanwhile, the CCPP is sensitive to change in the DH load, especially if a long operation period is considered. The main conclusion that can be drawn is that it is beneficial to have a high heat load, while running the CCPP.

Table 4 introduces results found for energy efficiency and power efficiency. The table shows minimum, median and maximum values based on annual plant operation in the whole DH load range.

As it can be seen, the median values of energy efficiency are significantly higher in Case 2 in comparison to Case 1 and Case 3. This clearly shows that annual plant operation under low DH load is unfavorable. At the same time, Case 2 has reduced median values of power efficiency due to higher heat demand in the DH system. The highest median values of power efficiency were obtained for Case 3.

The values found in Fig. 12 and Fig. 13, and Table 4 are different in comparison with Fig. 4 and Fig. 5. One of the reasons is that design values were given at the maximum DH load and fixed reference point. In reality, it is quite complicated to run a CCPP based on full DH load due to variable heat load characteristics and high seasonal variations. Further, different elevations above sea level, ambient temperature and air pressure cause adjustments to plant operation.

4.3. Fuel use

Finally, Fig. 14 represents the fuel input within the analyzed CCPP.

It can be seen that the reduction in return temperature shows a negative tendency in terms of fuel use. Cases 1 and 2 showed a gradual reduction in fuel input when the return temperatures increased. With the increase of temperature difference in the DH system, the fuel use increased; see Fig. 14. This happens because more energy input was required to heat up water in the DH system per 1 K. However, for Case 3, the fuel energy input did not follow uniform increase with respect to temperature level used. This could be explained by rapid change in the DH load in the CCPP. Further, the load factor given in Eq. (7) shows plant capacity utilization in terms of heating energy production. The load factor is the ratio of average load to the maximum load in the supply system [50].

$$\text{Load factor} = \frac{\text{Average load}}{\text{Maximum load}} = \frac{\text{Energy consumed during a period}}{\text{Maximum demand} \cdot \text{operation time}} \quad (7)$$

Table 5 gives the values of the load factor for the analyzed cases.

From Table 5 it can be seen that the load factor for Case 3 is the lowest. This indicates that the plant operates sporadically following the heat load during an operation year. The higher the load factor the cheaper the heat energy for the customer. In reality, it is very difficult to achieve a high load factor due to variable load characteristics from year to year.

5. Discussion

The analysis of different temperature levels applied in the DH system indicated that the energy efficiency had negligible variation due to temperature levels in the DH system when running the

Table 4
Results on energy and power efficiencies.

	Temp. level	Energy efficiency			Power efficiency		
		Minimum	Median	Maximum	Minimum	Median	Maximum
Case 1	100–45 °C	0.37	0.60	0.80	0.35	0.49	0.53
	100–30 °C	0.45	0.60	0.81	0.42	0.49	0.54
	100–15 °C	0.45	0.60	0.81	0.42	0.49	0.54
	90–45 °C	0.46	0.60	0.82	0.42	0.49	0.54
	90–30 °C	0.49	0.60	0.81	0.42	0.49	0.54
	90–15 °C	0.43	0.60	0.81	0.42	0.49	0.53
	80–45 °C	0.49	0.59	0.80	0.42	0.44	0.54
	80–30 °C	0.50	0.60	0.81	0.42	0.49	0.54
	80–15 °C	0.44	0.60	0.80	0.42	0.49	0.54
Case 2	100–45 °C	0.45	0.63	0.81	0.42	0.47	0.53
	100–30 °C	0.45	0.63	0.81	0.42	0.47	0.53
	100–15 °C	0.46	0.63	0.81	0.42	0.48	0.53
	90–45 °C	0.48	0.63	0.81	0.42	0.47	0.53
	90–30 °C	0.46	0.63	0.81	0.42	0.47	0.54
	90–15 °C	0.44	0.63	0.81	0.42	0.48	0.53
	80–45 °C	0.46	0.63	0.81	0.42	0.47	0.53
	80–30 °C	0.48	0.63	0.82	0.42	0.48	0.53
	80–15 °C	0.45	0.63	0.81	0.42	0.48	0.53
Case 3	100–45 °C	0.48	0.56	0.81	0.42	0.50	0.54
	100–30 °C	0.45	0.56	0.81	0.42	0.50	0.53
	100–15 °C	0.46	0.56	0.81	0.42	0.50	0.54
	90–45 °C	0.45	0.56	0.81	0.42	0.50	0.54
	90–30 °C	0.48	0.56	0.81	0.42	0.50	0.53
	90–15 °C	0.46	0.56	0.81	0.42	0.51	0.53
	80–45 °C	0.47	0.55	0.81	0.42	0.49	0.53
	80–30 °C	0.49	0.56	0.81	0.42	0.50	0.53
	80–15 °C	0.46	0.56	0.81	0.42	0.50	0.53

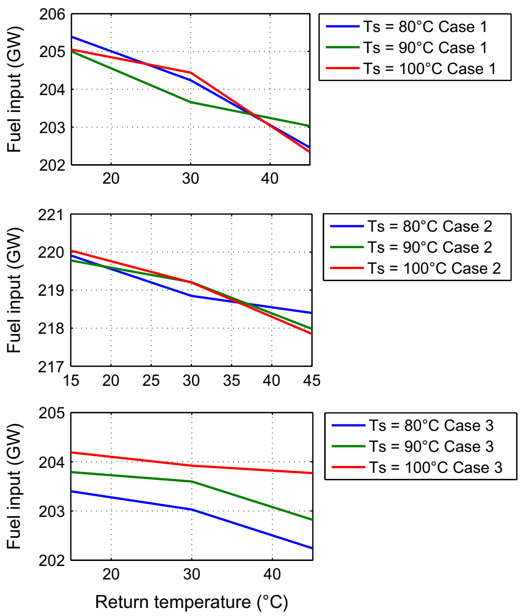

Fig. 14. Amount of fuel input in the CCPP.

Table 5
Heat load factor for analyzed cases.

	Heat load factor (–)
Case 1	0.32
Case 2	0.45
Case 3	0.23

CCPP. The reason for this is the high power production that takes place in the GTC. The analysis found that heat load distribution plays a crucial role in plant performance operation. Low heat load distribution leads to poor overall plant performance indicators. This gives incentives to run the plant for power production only. For this reason, when there is a need to select the DH supply and return temperatures for higher electricity production, the most effective method is to choose lower DH supply and return temperatures. Nevertheless, if we cannot change both of them, lowering the supply temperature is of more benefit [49].

Based on this study, it was concluded that it was rather difficult to operate a CCPP connected to low-energy building stock. Such buildings should be supplied from low temperature energy sources specially designed for this purpose. However, when high-grade heat is required, the CCPP can be used to produce additional heating energy. This means that the CCPP is suitable for high-density heat areas, while it operates poorly in low heat density areas. For future building stock, it means that the CCPP could be successfully implemented if the areas were grouped at one place, rather than spread over a large area.

The information depicted within the different plots in this study could be used as a tool for plant behavior prediction if the further reduction of supply temperature in the DH network is considered.

6. Conclusion

In this paper, the performance of the ethanol-based CHP with CCPP technology was investigated in the DH system. The focus was on different temperature levels which could occur in today's and near-future DH systems. The two different temperature control strategies in the DH system were analyzed to estimate the effects on plant operation. Three possible scenarios of the DH load and different supply and return temperatures in the DH system were considered.

The results showed that the power production in the CCPP was not influenced significantly by the supply temperature control. The change in the power production was between 1.2% and 2.8%. Therefore, the focus in the study was on the constant supply temperature in the DH system.

The analysis of the change in DH load showed that average heat efficiency was highest for the uniform distribution load and lowest for very non-uniform load. The average power efficiency was dependent on different temperature levels in the supply and return lines of the DH system. The results showed that the highest power efficiency was obtained for the temperature levels of 100–15 °C and the lowest for 80–45 °C, for Case 1 and Case 2. This indicated that a large temperature difference between the supply and return lines of the DH system resulted in higher power production in the CCPP. The results found that decrease in supply temperature had a low impact on energy efficiency. However, decreasing supply temperature to the DH system can lead to an increase in the service pipeline's lifetime, which is beneficial for the DH system. Another important conclusion is that the CCPP performance indicators are highly dependent on the heat load distribution in the DH system during the year. When DH load distribution had a uniform pattern throughout the operation year, as in Case 2, this resulted in better plant performance in comparison with Case 3. In the case of non-uniform heat load distribution, as in Case 3, plant performance was poor, indicating that the plant was poorly loaded. The comparison of three load factors provided the similar conclusion. This indicated that in the Case 2 the best possible heat load pattern for CCPP operation was obtained, while the load pattern for Case 3 was opposite. However, in the current CCPP, GT technology was employed, which utilized the benefits of the low DH load by increasing power production. Analysis of all the CCPP performance indicators versus the DH load showed negligible variation for all the temperature levels applied in the DH system. The difference was in the range of 2–3% between cases. The change in the overall fuel energy input showed that fuel use increases with increase in temperature difference between supply and return lines in the DH system.

The results obtained in this study point out an inevitable decrease in plant profitability while operating the CCPP under low and non-uniform heat demand profiles. This observation provides incentives to shut down the heat supply to DH systems and run CCPP at full load, producing as much electricity as possible. Low energy building stock should be connected to specially designed low-grade temperature sources under a prepared infrastructure. However, CCPP could be used if low energy buildings were located close to each other to increase the heat density. The CCPP could also be used during the peak energy demand. This will have a positive result on plant operation, since the CCPP will operate on its maximum heat load output, increasing its performance indicators.

The results obtained in this study can be used by designers of CHP systems, operators of DH systems, and legislators.

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Abstract: This paper discussed factors associated with the decisions on energy supply plants in new or existing district heating (DH) systems. Three highly efficient energy conversion technologies were considered. The study focused on assessment of the heat supply units considering economic aspects and technical limitation of the technologies. Further, risks associated with the changes in heat load profiles and fuel price volatility were investigated. The existing method for heat supply optimization was compared with a new method, suggested in this paper. The new method was based on detailed performance simulation models developed in Aspen HYSYS software and data post-processing in MATLAB. The results showed that the existing method for the heat supply optimization cannot show all the advantages of highly efficient conversion technologies. The study on the new method examined 36 plant combinations and identified eight with levelized cost of energy (LCOE) under 0.15 EUR/kWh. The results showed that increase in flexibility of DH provided better reliability of heat supply, while increasing the heat cost. The total deviation in LCOE due to fuel and electricity price volatility was in the rage of 1.6% -3.6%. Further, a change of 20% in the plant investment costs induced almost the same variation in LCOE.

Energy planning of district heating for future building stock based on renewable energies and increasing supply flexibility

Nomenclature:

$C(EUR)$	– total annual cost
$c(EUR/kW)$	– specific total cost per capacity unit
$\tau(hours)$	– operation time
$P(MW)$	– installed heat power capacity for each plant
$Q(MWh)$	– annual thermal production
$\tau_{n,m}(hours/year)$	– break-even operation time for two energy units
$F(EUR)$	– fuel cost;
$I(EUR)$	– investment cost;
$LCOE\ (EUR/kW)$	– levelised cost of energy;
$M(EUR)$	– operations and maintenance cost;
$P_{CHP}\ (MW)$	– power production in a CHP plant;
$P_{HP}\ (MW)$	– power needed for HP operation;
$E(MW)$	– electrical production;
$R^2(-)$	– goodness of fit;
$a_i\ (-)$	– model coefficients for the CHP power production;
$b_i\ (-)$	– model coefficients for CHP fuel input;
$c_i\ (-)$	– model coefficients for HOB fuel input;
$d_i\ (-)$	– model coefficients for HP power use;
n *(years)*	– system's lifetime;

r (%) – discount rate;

η (%) – efficiency;

Subscript/Superscript

CHP – combined heat and power plant;

el – electricity cost;

Elb – electric boiler;

HOB – heat only boiler;

HP – heat pump;

fix – fixed O&M cost;

$fuel$ – fuel cost;

t $(-)$ – year;

var – variable O&M cost.

Abstract

This paper discussed factors associated with the decisions on energy supply plants in new or existing district heating (DH) systems. Three highly efficient energy conversion technologies were considered. The study focused on assessment of the heat supply units considering economic aspects and technical limitation of the technologies. Further, risks associated with the changes in heat load profiles and fuel price volatility were investigated. The existing method for heat supply optimization was compared with a new method, suggested in this paper. The new method was based on detailed performance simulation models developed in Aspen HYSYS software and data post-processing in MATLAB. The results showed that the existing method for the heat supply optimization cannot show all the advantages of highly efficient conversion technologies. The study on the new method examined 36 plant combinations and identified eight with levelized cost of energy (LCOE) under 0.15 EUR/kWh. The results showed that increase in flexibility of DH provided better reliability of heat supply, while increasing the heat cost. The total deviation in

LCOE due to fuel and electricity price volatility was in the rage of 1.6% – 3.6%. Further, a change of 20 % in the plant investment costs induced almost the same variation in LCOE.

1. Introduction

Economy of district heating (DH) companies is highly dependent on heat sales. The rule is quite simple: the more heating energy is consumed by the customers, the higher the profitability of district heating DH companies. This tendency was good explain by authors in [1]. Today, with the new building codes and standards, a lot of attention is devoted to efficient energy use in buildings and reduction of heat losses [2, 3].

DH service is quite flexible and allows to employ different energy conversion technologies for heat energy generation. When the question is which technology to use, many issues should be considered. One scenario is when the energy production plants are already installed and in operation. Then, it is fundamental to find a solution how the existing plants can be operated with the lowest possible annual costs. On country, when planning a new DH system, the heat demands of the different target areas and the possible future development of these should be analyzed, as well as available heat sources should be investigated. Finally, an important task in decision on optimal generation units' combination, optimum configuration of DH network, and the optimum water temperature levels arises [4]. In addition, economics, energy saving, and environmental impact have become more important criteria for system design and operation, which designers have been burdened more heavily [5].

DH production units are chosen based on the scale and variation of heat demand, the local availability, costs of energy sources, and the investment cost of each technology [6]. Hence, for optimal utilization of the renewable energy and for economic reasons, the thermodynamic performance of energy production units is of major interest [7]. If the simulation approach has significant influence on operation results, then the cost of utility for society and the revenue for the investor will be also influenced by quality of simulation model [8]. This means that the decision on different technologies has to be based on proper evaluation by the help of relevant simulation models. In turn, this have to include the variability of the system parameters, aiming to find the best performance obtainable from the matching between production plants and users [9].

In liberalized energy markets, the installed utility technologies are optimized in an effort to reduce total production cost for each individual hour of production [7], to find the cheapest unit commitment and load dispatch satisfying given heat, power and reserve demands using given units [10]. These makes economy of production together with technical aspects of technology to be the main parameters that should be investigated before the final verdict is handed down.

When the combination of energy supply plants is under consideration, capital investment and operation and maintenance (O&M) costs should be carefully examined for each production unit. The main idea here is that different fuels can be utilized depending on their availability and cost. In addition, electricity rates should be considered. According to [11], electricity rates affects the operation of combined heat and power (CHP) plants as well as heat pumps (HP), and electric boilers. At the same time, the plant running costs put extra pressure on economic decision when annual operation is considered. Appropriate sizing of production plants is vital to achieve good levels of utilization, to ensure suitable performance for chosen systems, and to enable effective integration with existing or new DH systems [12]. Further, it should be noticed, that in most cases the plant operation becomes inefficient if the energy production unit operates under a low plant load [11, 13]. Given the high costs of installation and the tight energy saving constraints at which these plants are subjected, an incorrect predictive analysis can result in investment unsustainability either in economic or environmental terms [14, 15].

Ultimately, possible change in heat load profiles should be taken into account. According to [16], it is expected that in the medium term the heat load patterns can demonstrate fluctuations. The main reasons for that are: improved insulation of buildings, installation of ventilation systems with heat recovery, creation of heat islands due to growth of cities and global warming [17, 18] and legislation amendments. The mentioned facts facilitates change in customers' heat load profiles. However, the rise in population [19] and housing comfort levels [20], will contribute to the increase of the load to be heated. Thus, the levelling and size of the future DH demand will influence future DH operation and local DH system development [11].

The existing method of heat supply optimization that DH companies use currently is based on methodology on construction of optimal generation mix [21]. This method implies an energy unit with the highest investment cost be employed as a base load plant. In turn, this gives lover specific heat cost and higher plant efficiency [1]. This means that economy-of-size takes place that denotes energy plants with lower cost at higher production volumes be the main driving force. However, these arguments are no longer as strong, since more efficient heat generation technologies are available. Unfortunately, this method does not provide clear explanation which plant should be used by DH companies in various situations. Further, the energy efficiency of energy production units is treated as constant regardless of the load change. As mentioned before, the energy production unit operates inefficiently under a low plant load [11, 13].

Low DH price and ability to withstand energy efficient stand-alone heat generation solutions are the key factors that would make DH companies profitable in a long term. Therefore, this work aimed to propose a methodology that allows to identify the best combination of energy supply plants employing renewable energies and decreasing DH generation cost. The new method considers different input variables and operation constraints that makes it robust tool for heat energy planning.

The economic and technical aspects of heat generation were considered as well as yearly operation. In addition, the study provided information on consequences due to change in heat load patterns and fuel price volatility. In comparison to existing literature, this paper shed light on how to combine a few energy supply technologies including significant economic data. In addition, the models used in the analysis were based on detail thermodynamic models that made the results reliable.

2. Relevant energy supply technologies

Different energy supply plants are available for employment in the DH system. However, it is not an easy task to make a decision, which of them should be installed in particular situation. Due to technological complexity and limitations in operation, their applicability decreases. Therefore, the following section focusses on pros and cons of analyzed energy supply plants.

2.1 Biomass combined heat and power plant

CHP technology is well known and proved to be reliable nowadays. This technology was put forward more than a century ago [22]. According to [23, 24] CHP systems can be classified into topping and bottoming cycle with different exploitation regimes such as heat-much mode, electricity-much mode, mixed-much mode, and stand-alone mode [25]. CHP is efficient because it avoids the large amounts of waste heat produced in typical power generation plants [26]. In comparison to other energy conversion technologies used today, CHP has one of the highest indicators and its energy efficiency can reach up to 90% leading to better utilization of primary energy [27]. The attractive property of a CHP plant connected to a DH network is the possibility to massively include renewable sources of energy into energy systems at a reasonable cost [28].

Biomass CHP plants are often seen as an efficient way to reduce greenhouse gases emissions due to their very low CO_2 emissions level [29, 30].

However, there are several drawbacks associated with biomass CHP. Some biomass resources, in particular straw, contain aggressive components such as chlorine. These can lead to slagging and corrosion that reduces security of supply of DH customers. Further, biomass fuel has great variety of composition and therefore, different lower heating values (LHV) can effect efficiency of CHP plants and it outputs [31]. These put limitations on plant operation, for example when the peak load should be covered. Slow start up of this technology requires startup load and extra operation hours. Further, most CHP plants designed for DH purposes are characterized by very low power to heat ratio [32]. In addition, biomass-based CHP plants are widely used in regions that have ample fuel wood resources, forestry or agricultural residues. A business plan including the cost of the biomass resource collection and logistics is needed to ensure that CHP or power generation from solid biomass is economically viable [33].

2.2 Biomass heat only boiler

Nowadays, the modern heat only boilers (HOBs) are biomass based. Type of fuel propagates which equipment should be installed for the best fuel utilization. The main advantage of such systems is their high efficiency, especially when energy recovery technology is applied. If a moisture content of the fuel is above 30 – 35%, as with forest wood-chips, flue gas condensation should be employed. Flue-gas condensation can improve the overall maximum efficiency of plant up to 30% depending on fuel type and the temperature of the DH water [34]. For plants firing wood-chips with 45 – 55% moisture content, the thermal efficiency of more than 100% could be reached based on LHV [35]. Biomass HOB provides possibility to maximize CO_2 savings and potentially eliminate all emissions from fossil fuel systems. The costs of biomass fuels are typically lower than the fossil fuels and such systems can therefore provide significant operational savings, which reduces the payback period [12].

The drawback of such systems is high complexity that required highly trained operation staff. Higher combustion temperatures can lead to high temperature corrosion, soot, and wear out of equipment [36]. Biomass heating systems generally have higher initial capital cost than fossil fuel systems of equivalent rated capacity. Although biomass systems have higher upfront costs than fossil fuel boilers. If there is a need to run at low load conditions for extended periods, potentially higher maintenance cost appears [12].

2.3 Heat pump

Heat pump (HP) systems offer economical alternatives of recovering heat from different sources for use in various industrial, commercial, and residential applications [37]. A DH system is a promising energy-saving measure for high-density cities and HP systems play an essential role in such large-scale system [38, 39]. Further, DH systems with HP technology has demonstrated significant reduction in annual energy bill [40]. Today, the most advanced technical developments in the HP field provides opportunity to deliver heat at a temperature of 110°C [41-43]. According to [40, 44], the large-scale HP applications based on mechanical vapor compression and absorption closed cycle principles can be successfully applied in the DH systems.

A general advantage of HP technology is ability to utilize energy at a low temperature level. In addition, the HP is flexible concerning use of renewable energy, waste, and surplus heat. Compared with traditional heating technologies, the HPs are more complex and have high investments costs. However, this is counterbalanced by considerable savings in operation costs [31].

The main drawback associated with HP technology is electricity use. This is particularly relevant when the electricity prices in local conditions are rather high. At the same time, the use of large HPs can be called into question due to high carbon content in the marginal or incremental electricity generation in most industrialized regions and countries [1]. Investment cost of high temperature HP is typically the same for the different technologies, when only the HP itself is considered [31]. Economically, simple payback period for industrial HP applications is between 2 and 5 years [44].

2.4 Electric boiler

Even though nondesirable in new requirements, electric boilers are sometimes necessary for energy supply to cover the extreme operation situations and as a back-up plant. Electric boilers for DH are used to some extent in countries where electricity is occasionally available at a low price, for example in Sweden and Norway [1]. Due to its very simple design, the electric boiler is extremely undependable and easy to maintain. The operating costs are very dependent on the size of the boiler. Thus, heat production from electric boilers can only compete with other heat production units at low electricity prices [31]. If necessary, an electric boiler can also be operated as a peak load plant, even though this may be problematic from the perspective that in many countries there is a tendency that peak heat demand coincides with the peak in electric power demand [1].

3. Economic data on energy supply technologies

This section focuses on various economic issues associated with the installation of energy production unit. The presented information is based on literature review. The aim was to identify available economic data associated with capital investment and O&M values for each technology. In addition, fuel prices and electricity rates were considered.

Several issues should be considered when one does a decision about installation of energy production unit. First, the technology should meet customer requirements in providing heat to the DH system. At this point, it can be noted that different customers can use wide range of temperatures due to their various purposes. Further, heat load patterns should be taken into account. Due to changeable climate characteristics and continuous improvements in building codes and standards, the heat load patterns can show variation from year to year. On the other hand, employed energy conversion technology should be environmentally friendly and certainly display positive economy under its long term operation. Therefore, a detailed feasibility study should be carried out considering installation of certain system.

Normally, three economic key-points should be analyzed before doing investment in certain technology. These are following: capital investment cost, fixed O&M costs, variable O&M cost, and fuel costs.

Due to significant amount of found cost data for each technology, the corresponding tables are presented in Appendix. The data in Appendix are organized in tables, for each technology separately. However, the most important information selected for the analysis is listed further in Section 4.7.

The comprehensive economic feasibility of heat production units is impossible without fuel prices. In this study both CHP and HOB systems utilized biomass as a fuel. At the same time, electricity was required for HP operation. Hence, Table *1* summarizes the fuel prices for these technologies found in the literature for EU countries.

Table 1. Prices for biomass fuel and electricity

Fuel type	Price	Comment	Reference
Electricity	0.120 EUR/kWh	Annual consumption level: 500 MWh – 2000 MWh; EU-28 in 2013	[45]
	0.127 EUR/kWh	Annual consumption level: 500 MWh – 2000 MWh; Euro Area (EA-17) in 2013	[45]
Wood chips	40 EUR/tonne		[46]
	70 EUR/tonne		[47, 48]
	56 EUR/tonne	Croatia, 2014	[49]
	58 EUR/tonne	Romania, 2014	
	136 EUR/tonne	Ireland, 2014	
	132 EUR/tonne	Austria, 2014	
	113 EUR/tonne	Germany, 2014	

4. Methodology

In this section, the methodology for analysis of energy supply technologies and economic evaluations are described. In this study, three state of the art technologies have been chosen for the analysis. In addition, electric boiler was considered for heat supply during extreme operation situations. For the feasibility purpose, the detailed plant models are necessary. Therefore, the simulation of energy supply sources was done in Aspen HYSYS [50] simulation software. The Aspen HYSYS simulation software is well known in process simulation and gives possibility to include different components. Some examples of application are mentioned in [51-53]. For the purpose of this study, simplified plant models were developed based on detailed HYSYS models. The simplified, polynomial models were necessary to enable easier link between different plant performance data and heat load data. Detail explanation on the new method is given in Section 4.6. In addition, the analysis considered three scenarios of heat load patterns. The heat duration curves are introduced in Section 5. Based on the polynomial plant models and heat load data, the methodology for plant analysis was developed in MATLAB software [54].

4.1 Biomass based CHP models, detailed and simplified

The biomass CHP plant is shown in Fig. 1.

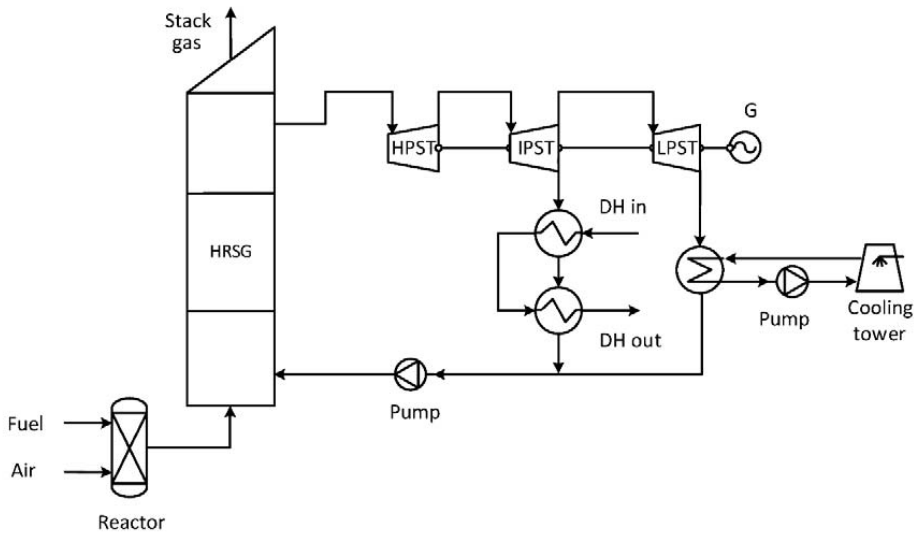


Fig. 1. Schematic of the biomass based CHP

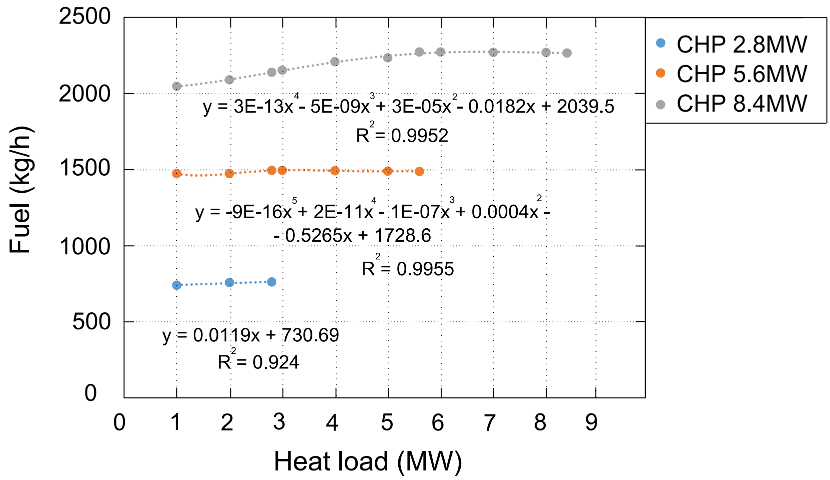
The LHV of biomass fuel was assumed 19 MJ/kg with a moisture content of 40%. The ambient conditions were 15°C and 1.015 bar. After fuel combustion in the reactor, the flue gases with the temperature of 880°C flew in a high recovery steam generator (HRST) where the pressurized water carrier was heated up to 540°C. The HRSG was modeled as three stages heat exchangers. These are an economizer, an evaporator, and a superheater. The steam turbine cycle (STC) contained high pressure steam turbine (HPST), intermediate pressure steam turbine (IPST), and low pressure steam turbine (LPST). The live steam flowing from HRSG expanded in HPST from 540°C and 100 bar to 259°C and 12 bar. The expansion continued in the IPST to 239°C and 10 bar. IPST was with one extraction for DH purposes. The DH was satisfied based on required values of heat energy from consumers. Finally, in the LPST the steam expanded to 33°C and 0.05 bar after the condenser, the water was pumped back to HRSG. The total efficiency of CHP plant operation was 88%.

The dynamic behavior of modern CHP plants is characterized by the short startup time and quick load change capability [55]. In order to ensure that operation of CHP plant is realistic, the startup and standstill intervals were considered in the analysis. It was assumed that the CHP plant did not operate (was in standstill mode) if DH load was low for longer than 72 hours. Therefore, the three startup modes [55] were applied when the condition of plant operation was satisfied:

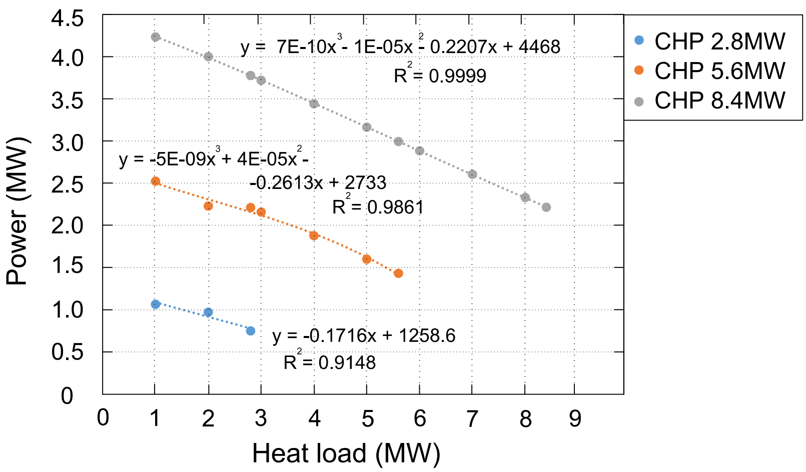
- Hot start after 8 hours standstill: 40 – 60 minutes;
- Warm start after 48 hours standstill: 80 – 120 minutes;
- Cold start after 120 hours standstill: 120 – 170 minutes.

According to [12], from both technical and economic points of view, a biomass CHP plant is best operated relatively continuously at between 30% and 100% of its rated output. Biomass plants do not generally respond well to rapidly varying loads, or long periods at low load conditions below a minimum modulating range. Therefore, the lower bound of CHP's heat capacity applied in this study was equal to 30% of full plant capacity.

After the model simulation was conducted in Aspen HYSYS, enough data points for defining the simplified model were obtained. Fig. 2. shows relationship between power production and DH load, and fuel consumption and DH load in CHP plant. The plant performance for three different sizes of the heat load are given in Fig. 2. These three sizes were chosen based on the maximum heat demand, see Section 5.



a) power production versus DH load in CHP plant



b) fuel consumption versus DH load in CHP plant

Fig. 2. Operational characteristics of three CHP plants with various heat capacities

From Fig. 2a the power production of a CHP plant can be described by using heat output as:

$$P_{CHP}(Q_{CHP}) = a_3 \cdot Q_{CHP}^3 + a_2 \cdot Q_{CHP}^2 + a_1 \cdot Q_{CHP} + a_0 \qquad (1)$$

where Q_{CHP} is required heat output to the DH system and a_3, a_2, a_1, a_0 are model coefficients. Further, the dependencies between fuel consumption and DH load in CHP plant can be described as fifth-polynomial model for fuel input, as a function of heat output:

$$F_{CHP}(Q_{CHP}) = b_5 \cdot Q_{CHP}^5 + b_4 \cdot Q_{CHP}^4 + b_3 \cdot Q_{CHP}^3 + b_2 \cdot Q_{CHP}^2 + b_1 \cdot Q_{CHP} + b_0 \qquad (2)$$

where Q_{CHP} is required heat output to the DH system and $b_5, b_4, b_3, b_2, b_1, b_0$ are model coefficients. The accuracy of the curve fitting and future model ability can be measured by R^2 value. The closer R^2 value to 1, the better the model.

Finally, based on the model data, it was possible to calculate the CHP energy efficiency as a function of the heat load. The maximum energy efficiency of CHP model was close to 0.9, for all three CHP sizes. The maximum efficiency was reached for the maximum heat load. Hence, the found CHPs' energy efficiencies fits well with data presented in Appendix, which proved the high degree of quality of the applied CHP models.

4.2 Biomass HOB models, detailed and simplified

Nowadays, the most advanced HOB are designed with the heat recovery of the flue gases that leads to improved efficiency.

Fig. 3 shows a layout of biomass HOB with energy recovery.

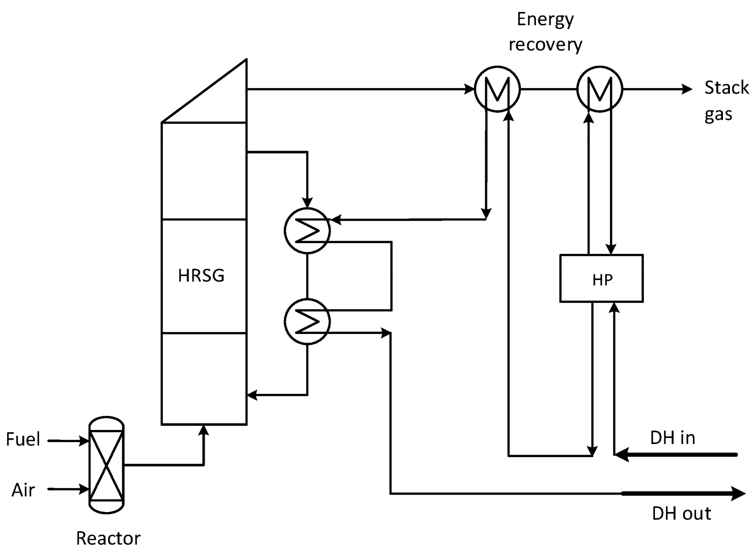


Fig. 3. Schematic of HOB

The fuel with the air were supplied to the reactor where the combustion process took place. Further, the heat was released to heat up the DH water in the HRSG. In this study, the model of biomass HOB was constructed in two stage flue gas condensing system for maximum energy conversion. In the first stage the incoming DH water was preheated by absorption HP, while in the second was after heated and then supplied to HRSG of HOB. The absorption HP was driven by high-pressure steam with ammonia as a working liquid and a water as an absorbent. In the condensing system the temperature of flue gases decreased to 35°C and the most of water vapor was condensed to water. The supplied water temperature to HRSG after condensing system constituted 80°C. In this study, both HOB and absorption HP were evaluated as a single unit. The return DH water from consumers had temperature of 50°C and after warming up in the HOB the temperature of 105°C was reached. Normally, the typical wood fired HOB plants are regulated in the interval of 25 – 100% of full capacity, without violating emission standards. The best technologies can be controlled 10 – 100% with fuel not exceeding 35% moisture content [31]. Therefore, the lower bound of HOB's heat capacity applied in this study was equal to 25% of full plant capacity.

In the HOB model the main interest was relationship between fuel use and DH load. Therefore, Eq. (3) presents a simplified model of the HOB based on detailed HYSYS model.

$$F_{HOB}(Q_{HOB}) = c_5 \cdot Q_{HOB}^5 + c_4 \cdot Q_{HOB}^4 + c_3 \cdot Q_{HOB}^3 + c_2 \cdot Q_{HOB}^2 + c_1 \cdot Q_{HOB} + c_0 \qquad (3)$$

where Q_{HOB} is required heat output to the DH system; $c_5, c_4, c_3, c_2, c_1, c_0$ – model coefficients.

Fig. 4. shows polynomial models for the HOB in Fig. 3.

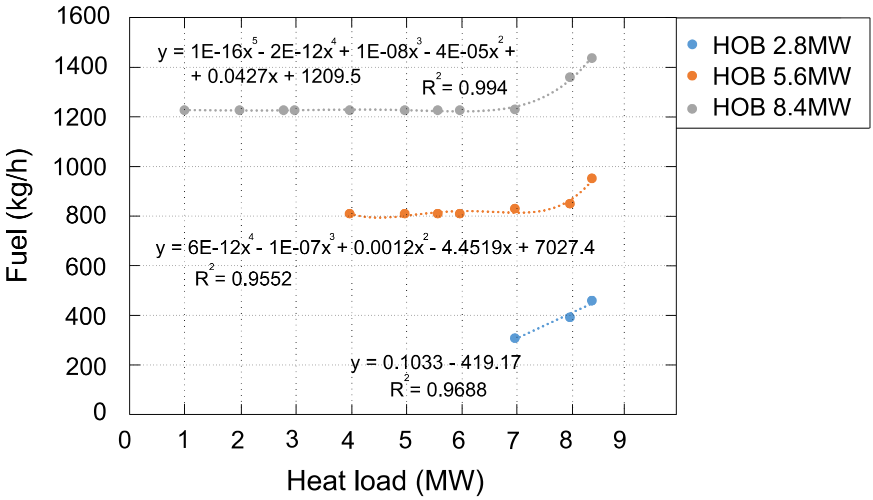


Fig. 4. Fuel consumption versus DH load in HOB

The developed HOB models showed maximum heat efficiencies of 1.12 - 1.16. This is mainly because flue gas condensation technology was used. The heat efficiencies showed match with existing literature, see Appendix, which proved that the introduced HOB models were good and reliable for further analysis.

4.3 Vapor compression HP, detailed and simplified models

The main issue associated with the use of HP technology in DH systems is to ensure that desired supply temperature is satisfied. This HP modification uses NH_3 (ammonia/ R717) as a working fluid and Vilter's single-screw compressor [43]. This technology is referred as high temperature heat pump (HTHP) used for industrial installations.

In this study, a large mechanical heat pump (MHP) was considered for the analysis. The MHP was based on vapor compression principle and utilized ammonia as a working fluid. The scheme of two stage MHP presented in Fig. 5.

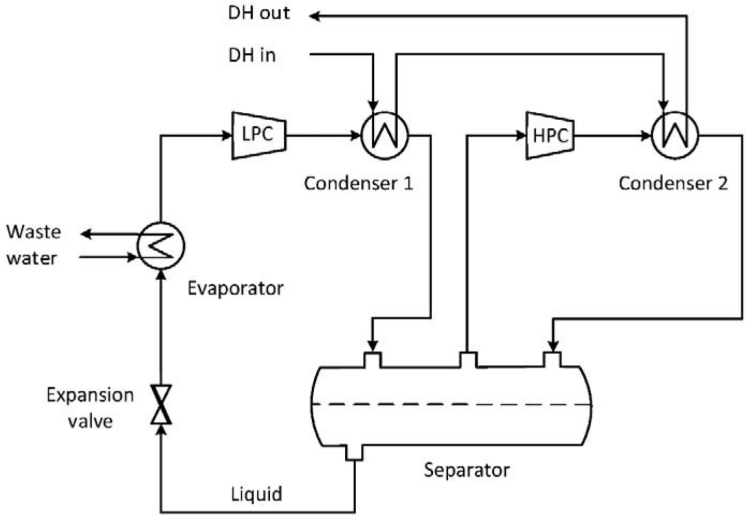


Fig. 5. MHP with two stage compression and separation vessel

In the MHP system, four main components of HP such as evaporator, compressor, condenser, and expansion valve were connected to a closed circuit. The MHP contained a separator vessel. The function of vessel is to separate the refrigerant in liquid and vapor. In the analyzed model, the MHP was assumed to upgrade heat from residual waste water. The incoming temperature of residual water to the evaporator was 27°C. After releasing heat in the evaporator, the temperature dropped to 24°C. Further, the ammonia vapor was compressed in the low-pressure compressor (LPC) from 7 bar and 15°C to 30 bar and 167°C. The refrigerant in the gaseous state flowed to Condenser 1 where the water from the DH greed preheated up to 70°C. After Condenser 1, the mixture of fluid and gaseous refrigerant flowed to the separator vessel. In the separation vessel the refrigerant was separated into two fractions. The liquid fraction was forward to the evaporator via expansion valve and gaseous refrigerant continued be compressed in the high-pressure compressor (HPC). The HPC compressed working fluid from 30 bar and 66°C to 76 bar and 172°C. Further, the hot vapor flowed the condenser 2 and DH water was after-heated up to the temperature of 105°C. Finally, the high-pressure refrigerant flowed back to the separator and cycle continued.

The simplified model of HP's power use can be expressed as:

$$P_{HP}(Q_{HP}) = d_3 \cdot Q_{HP}^3 + d_2 \cdot Q_{HP}^2 + d_1 \cdot Q_{HP} + d_0 \quad (4)$$

where Q_{HP} is required heat output to the DH system, d_3, d_2, d_1, d_0 are the model coefficients.

The polynomial model of the HP is depicted on the Fig. 6..

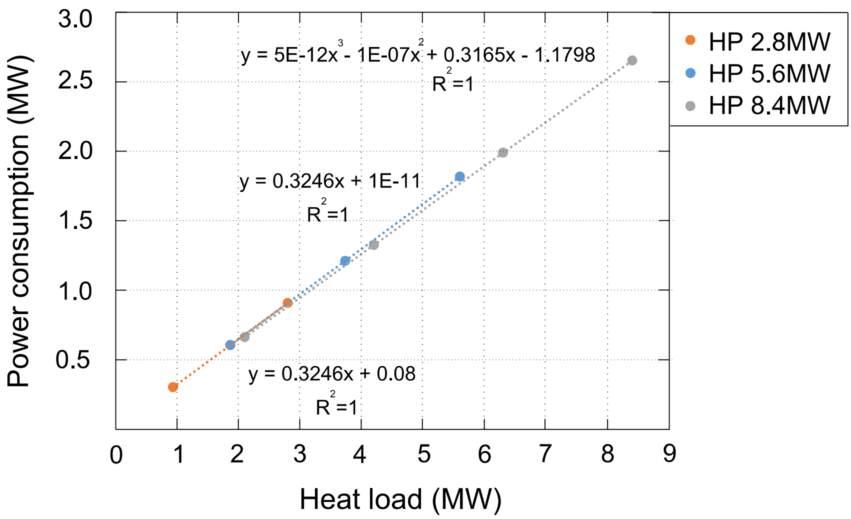


Fig. 6. Power consumption versus DH load in HP

Due to low variation of heat source temperature, the COP of the analyzed HP was almost equal to 3.3. Similar valued were found in the literature for the HP performance.

4.4 Electric boiler

The employed electric boiler model was described by linear dependency. The boiler control ability was adjusted between 10 – 100% [31] and had efficiency of $\eta = 99\%$.

4.5 Existing method for heat supply optimization

In this paper the new, suggested, method is compared to the existing method of heat supply optimization. The existing method implies the following assumptions: constant energy price; 0 – 100% control range of the plant capacities; no influence of plant size on investment cost; constant plant efficiency regardless of the plant load. This method was developed primarily for electrical energy planning and explained in details in [21]. Further, the method was adjusted to DH needs [1].

The total cost for the heat generation can be expressed as:

$$C = C_{fix} + C_{var} \tag{5}$$

where C is a total annual cost which consists of an annual fixed cost, C_{fix}, and a variable operating cost C_{var}.

The specific cost for each heat unit will be:

$$c = c_{fix} + c_{var} \cdot \tau \tag{6}$$

where c is a specific total cost per capacity unit, c_{fix} is a specific investment cost per installed heat unit, c_{var} is a variable cost per heat unit, τ is operation time.

The specific total cost per installed heat unit can be found as:

$$c = C/P \tag{7}$$

where P is installed heat rate for each plant.

The specific investment cost can be found as:

$$c_{fix} = C_{fix}/P \tag{8}$$

Thus, the variable cost per heat unit can be expressed as:

$$c_{var} = C_{var}/Q \tag{9}$$

where Q is annual heat supply.

The break-even times of plants operation can be found for a various number of energy production units that are taken in optimization process. Eq. (10) and Eq. (11) shows situation where three energy production plants are optimized in order to find the lowest annual total cost. The break-even times $\tau_{1,2}$ and $\tau_{2,3}$ are obtained using the basic optimization condition that stipulates that the total cost should be equal for two competing plants at each intersection:

$$\tau_{1,2} = (c_{fix,2} - c_{fix,1})/(c_{var,1} - c_{var,2}) \tag{10}$$

$$\tau_{2,3} = (c_{fix,3} - c_{fix,2})/(c_{var,2} - c_{var,3}) \quad (11)$$

4.6 The suggested methodology for analysis of the energy supply plants

In order to combine the plants properly, there is a need to identify the total number of combinations. Therefore, the basic formula for the number of possible combinations of k objects from a set of n objects can be written as:

$$\binom{n}{k} = \frac{n \cdot (n-1) \ldots (n-k+1)}{k \cdot (k-1) \ldots 1} = \frac{n!}{k! \cdot (n-k)!} \quad (12)$$

The Eq. (12) applied in this study allows finding the total number of possible plants' sets with three elements in each of them.

The method implied to use plant capacities in proportion of 20%, 40%, and 60% of the maximum DH load (see Section 5), which makes it easier to develop combinations sets. In this study heat generation units were combined in three dimension sizes: 2.8 MW that corresponds to 20% of the full DH load, 5.6 MW equal to 40% of the full DH load, and 8.4 MW equal to 60% of the DH load. One of the conditions is that a combination set should employ different technologies in it without repetitions. Another is that three plants should not have total heat capacity more than 100% of the DH load e.g. 14 MW. Therefore, under these conditions a number of generated plant combinations (PCs) by Eq. (12) was limited to 36.

Fig. 7. shows how the plants were combined. The PCs are based on plant ability to satisfy base load. When one technology is chosen for the base load, other technologies cover the rest of the load as an intermediate and peak load plants.

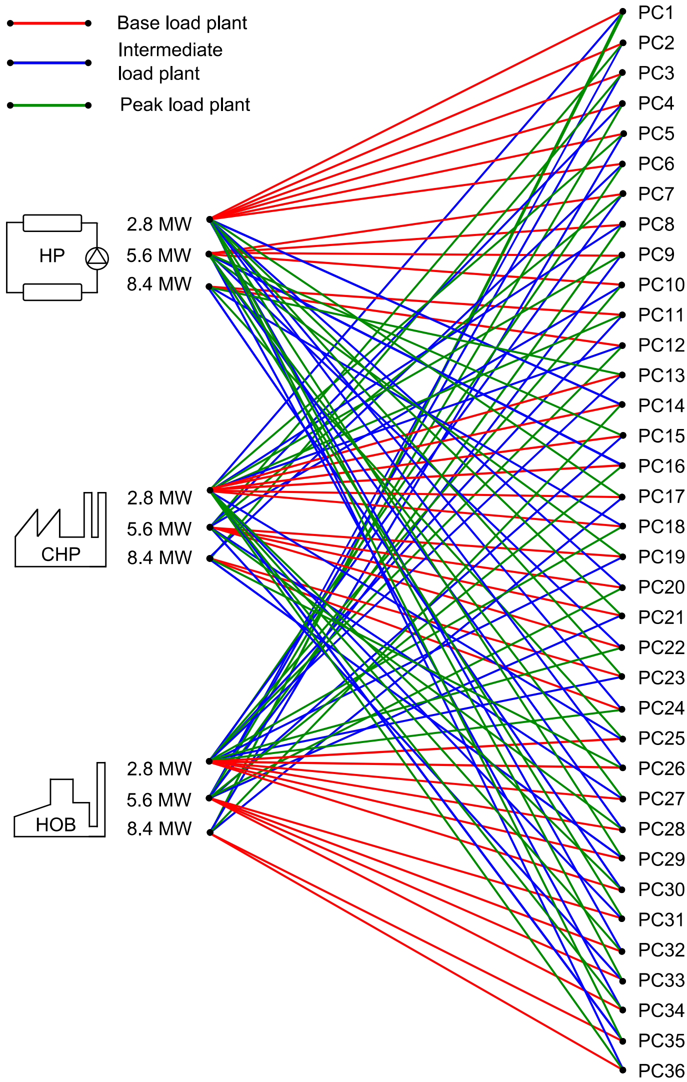


Fig. 7. Analyzed combinations of energy supply sources

Fig. 7. shows three energy generation technologies with different heat outputs developed in combination sets. The color lines indicates plant's attachment to base load, intermediate load or peak load. The electric boiler was not included in Fig. 7, however, each combination has an electric boiler of 3 MW of heat output to cover extreme operation situations and as a back-up plant. Fig. 8. introduces the information flowchart for the new method used in this analysis.

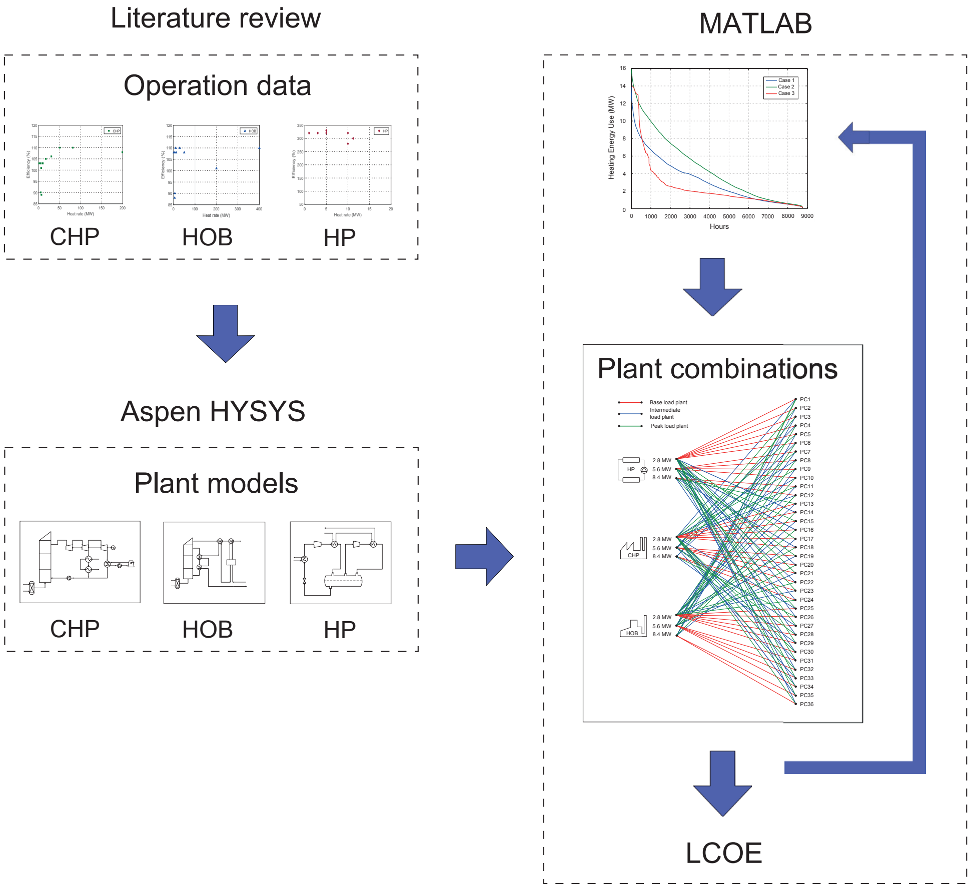


Fig. 8. Information flowchart for the new method for energy planning

4.7 Economical evaluation

In Section 3 the overview of the cost data for technologies and fuel prices was presented. This section introduces technique for performing the cost analysis. In this study, the levelized cost of energy (LCOE) [56] approach was used to compare PCs. The LCOE of a given technology is the ratio of lifetime costs to lifetime energy generation, both of which are discounted back to a common year using a discount rate that reflects the average cost of capital [57]. The LCOE allows alternative technologies to be compared when different scales of operation, different investment and operating time periods, or both exist [56].

The LCOE can simply be presented as:

$$LCOE = \frac{Total\ Life\ Cycle\ Cost}{Total\ Lifetime\ Energy\ Production} \tag{13}$$

The total life cycle cost in the Eq. (13) includes capital investment cost, O&M cost, and fuel cost. The capital investment cost can be estimated as:

$$I_t = I_{CHP} + I_{HOB} + I_{HP} + I_{Elb} \tag{14}$$

where, I_{CHP}, I_{HOB}, I_{HP}, I_{Elb} are investment costs for installation of CHP, HOB, HP and electric boiler.

The fixed share of O&M includes all costs, which are independent of how the plant is operated, e.g. administration, operational staff, planned and unplanned maintenance, payments for O&M service agreements, network use of system charges, property tax, and insurance. Re-investments within the scheduled lifetime are also included, whereas re-investments to extend the life are excluded. While variable O&M costs included consumption of auxiliary materials (water, lubricants, fuel additives), treatment and disposal of residuals, output related repair and maintenance, and spare parts (however not costs covered by guarantees and insurance) [31]. Therefore, the O&M costs can be found as:

$$M_t = C_{var}^{CHP} + C_{var}^{HOB} + C_{var}^{HP} + C_{var}^{Elb} + C_{fix}^{CHP} + C_{fix}^{HOB} + C_{fix}^{HP} + C_{fix}^{Elb} \tag{15}$$

where, C_{var}^{CHP}, C_{var}^{HOB}, C_{var}^{HP}, C_{var}^{Elb} are variable O&M costs, and C_{fix}^{CHP}, C_{fix}^{HOB}, C_{fix}^{HP}, C_{fix}^{Elb} are fixed O&M for CHP, HOB, HP, and electric boiler.

The fuel consumption cost was evaluated as a sum of biomass fuel consumed by CHP, and HOB, and electricity needed for operation of electric boiler and HP:

$$F_t = C_{fuel}^{CHP} + C_{fuel}^{HOB} + C_{el}^{HP} + C_{el}^{Elb} \tag{16}$$

where, C_{fuel}^{CHP}, C_{fuel}^{HOB} present the fuel cost for operation of CHP, HOB, HP and electric boiler. The allocation of CHP's fuel cost between thermal production and electrical production was based on an energy method [58]:

$$f_Q = Q/(Q + E) \tag{17}$$

where, Q and E represent thermal and electrical production.

Finally, including all the costs, Eq. (13) can be rewritten as:

$$LCOE = \frac{\sum_{t=1}^{n} \frac{I_t + M_t + F_t}{(1+r)^t}}{\sum_{t=1}^{n} \frac{Q_t}{(1+r)^t}} \tag{18}$$

where, I_t is investment expenditures in the year t; M_t is O&M expenditures in the year t; F_t is fuel expenditures in the year t; Q_t is heat generation in the year t; r is a discount rate; and n is life of the system.

The discount rate is meant to reflect the loss of utility from deferred consumption and the degree of systematic risk of the project [59]. The discount rate used in various analyses in the energy sector in Norway is determined by the Norwegian Water Resources and Energy Directorate (NVE) [60], based on instructions from the Ministry of Finance. Since DH is normally considered as investment with low economic risks [1], the NVE has stated to apply discount rate of 4.0 – 6.5% for bio-based DH systems [61, 62].

The technical life of technologies can be adopted from [31, 33, 63], for biomass CHP is typically 20 – 25 years, for biomass HOB and large scale vapor compression HP and electric boiler this value is 20 years [31].

Based on literature review presented in Section 3, the investment and O&M costs given in Table 2 were selected for this analysis. However, some uncertainty in these values could take place. Therefore, in order to evaluate consequences due to inaccurate cost data, the uncertainty analysis is presented in Section 6.

Table 2. Investment and O&M costs used in the analysis

Plant type	Plant capacity (MW)	Investment costs (MEUR/MW)	Fixed O&M cost (EUR/MWhfuel)	Variable O&M cost (EUR/MWhfuel)
CHP	2.8	3.0	2.0	2.6
	5.6	2.6	2.0	2.6
	8.4	2.3	2.0	2.6
Biomass HOB	2.8	0.8	2.1	2.0
	5.6	0.6	2.1	2.0
	8.4	0.5	2.1	2.0
HP	2.8	0.25	6.0	0.2
	5.6	0.42	6.0	0.2
	8.4	0.6	6.0	0.2
Electric boiler	3.0	0.15	1100 EUR/MW/year	0.5 EUR/MWh

After evaluation of different prices of biomass fuel and electricity rates presented in Table *1*, the biomass fuel price was chosen as 75 EUR/tonne and electricity price 0.12 EUR/kWh.

5. Case study

The analysis of different combinations of energy supply technologies was based on heat energy demand measured in the university campus. The required supply and return temperature levels in the DH system were assumed 105 – 50°C. In this study, three heat demand profiles were considered to illustrate influence of different load distribution. The analyzed duration curves are depicted in Fig. 9..

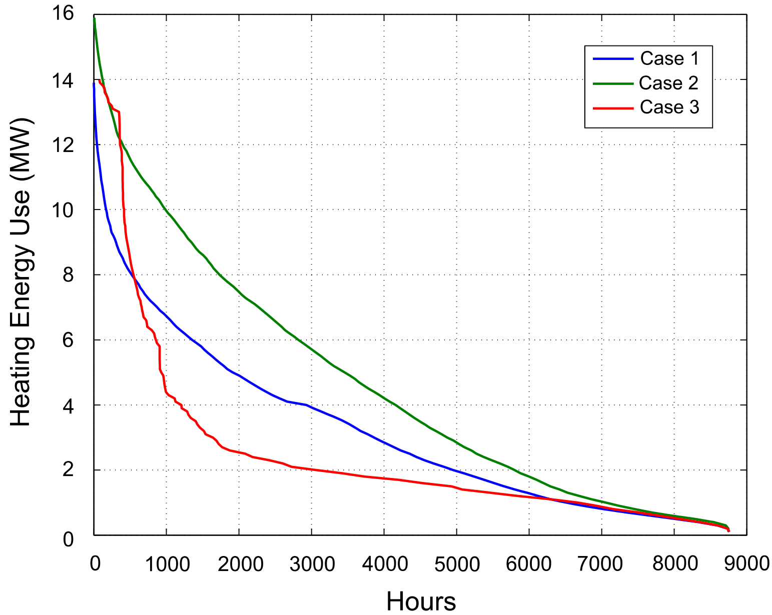


Fig. 9. Heat duration curves

Case 1 in Fig. 9. presented the heat duration curve during a regular year in the analyzed location and was used as a reference year. Case 2 presented the heat duration curve under a higher occupancy level and lower outdoor temperature. The heat duration curves in Case 1 and Case 2 were measured at the university campus. Case 3 presents the situation for future energy use, taking into account newly-built passive houses and nearly zero energy buildings (nZEB) with low heat energy use throughout the year and high peaks occasionally. Case 3 is the result of an assumption and is characterized by a decrease in heating energy use of 22.17% in comparison with the reference year. The heat load characteristics of the analyzed cases are summarized in Table 3.

Table 3. Heat load characteristics

	Case 1	Case 2	Case 3
Heating energy use (GWh)	27.48	40.06	21.39
Average DH load (MW)	3.14	4.57	2.44
Heat rate under maximum hours' frequency (MW)	1	1	2
Duration of maximum heat rate (hours)	2465	1887	3547
Heat rate under minimum hours' frequency (MW)	14	16	11
Duration of minimum heat rate (hours)	14	38	12
Utilization time (hours)	1962	2861	1528

6. Results

Energy planning results by using the existing method of heat supply optimization are shown first. Afterwards, the findings from the new method of energy planning are shown.

6.1 Results on the existing method

The main idea of different optimization techniques is finding the best solution that satisfies DH operation from both technical and economical points. Therefore, the existing method for heat supply optimization balances operation cost and investment cost for achieving the lowest total annual cost. This method is explained in Section 4.5. Fig. 10. introduces the existing plant optimization method.

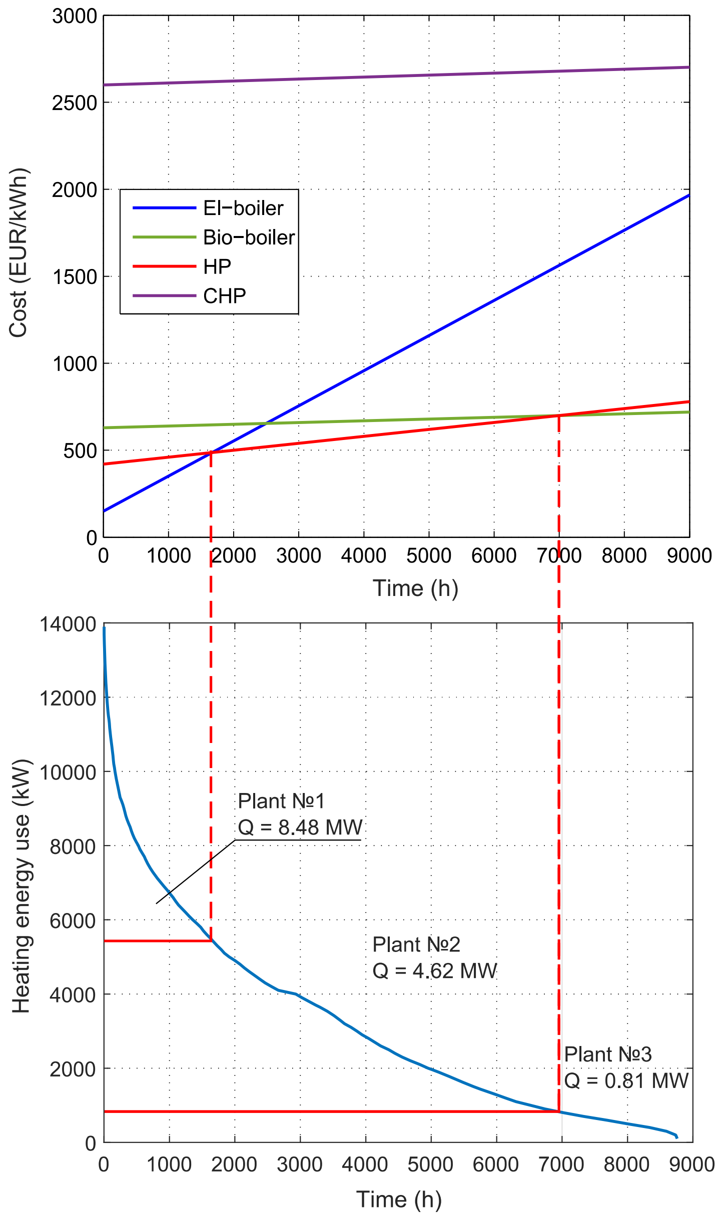


Fig. 10. The linear cost characteristics for three plant model is shown in the upper diagram and the corresponding optimal division of plant capacities are shown in the lower duration diagram

Fig. 10. shows that the electric boiler has lowest investment cost and therefore, it is beneficial be utilized as a peak load plant from 0 – 1760 hours. The intermediate load should be covered by the HP and the base load by HOB. Further, it can be noted that CHP is not a relevant plant according to the existing method, because the investment is too high. In reality, it is well known that CHP is reliable provider of heat supply and it is beneficial to run it as a base load plant. In Fig. 10., the plant capacities could be distributed as follows: for the peak load plant an electric boiler of 8.48 MW maximum rate, for the intermediate load plant HP of 4.62 MW, and for the base load plant HOB of 0.81 MW.

The sensitivity analysis of the current optimization method (Fig. 10) was performed in order to estimate robustness of the method due to change in heat load. Table 4 shows sensitivity results.

Table 4. Sensitivity of the current optimization method due to different load profiles

		Electric boiler	HP	HOB
Case 1 DH cost – 0.109 EUR/kWh	Heat capacity (MW)	8.48	4.62	0.81
	Heat energy use (MWh)	1352	12899	13216
Case 2 DH cost – 0.104 EUR/kWh	Heat capacity (MW)	8.22	7.13	1.03
	Heat energy use (MWh)	304	18510	21232
Case 3 DH cost – 0.083 EUR/kWh	Heat capacity (MW)	11.05	2.08	0.87
	Heat energy use (MWh)	1458	7902	12005
Operation hours		1166	5334	1 760

From Table 4 it can be seen that change in heat load induced significant variation in the plant heat capacities . For the electric boiler the induced change was between 1% and 23%, for HP was 55% and 70%, and for HOB was between 7% and 22% due to load change. This showed that this method is very sensitive to changes in heat load profile. In turn, this can lead to low load factor for operated plants and further effect the DH price.

The uncertainty due to change in investment cost in the range of ±10% showed that electric boiler was not sensitive, which lead to negligible change in DH price of less than 1%. However, HP and HOB were more effected. The change in HP's investment cost induced +9.6% –

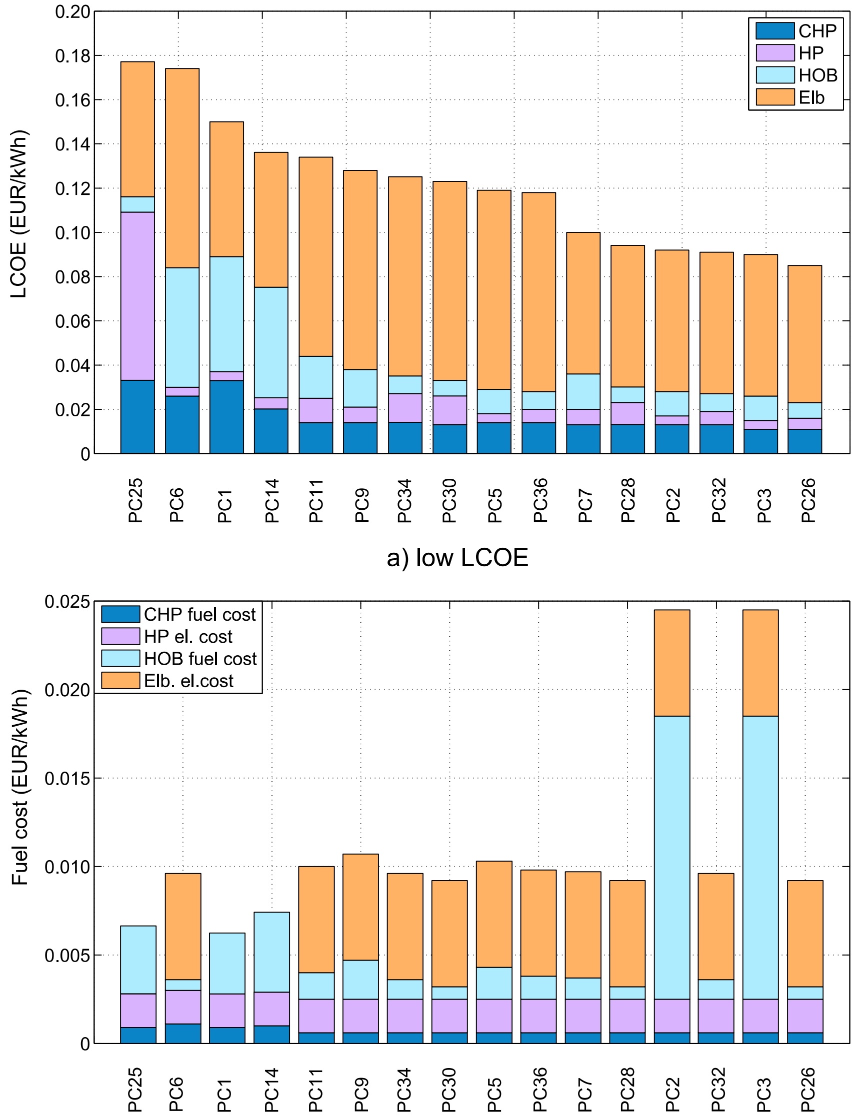
-12% change in DH cost. For HOB these values were even higher and constituted -22% – +18%. The effect due to multiple uncertainty induced change of -11% – +11.4% on DH cost or ±0.01 EUR/kWh for both reduction and increase in investment cost. In addition, these lead to change in heat capacities of selected plants. Hence, the method is also sensitive to variation in investment cost.

The uncertainty due to change in energy cost for chosen plants was carried out in the range of ±10%. The largest change in DH cost induced the HP (+7.61% – -6.79%). For the HOB these values were in the range of -2.2% – +2.7%, while for the electric boiler -3.4% – +3.5%. However, the impact due to multiple uncertainty showed 1% change in DH cost. As it can be seen, the change in the investment cost induced larger change in DH cost than change in energy cost. This means that existing method of heat supply optimization is sensitive to change in investment cost of analyzed technologies.

The conclusion from the above analysis was that the existing method was sensitive to variations in heat load profiles. This meant that any future change in heat demand would influence the heat cost. Further, some expensive technologies such as CHP might be excluded due to high investment cost. Finally, it does not show how DH should be equipped and operated over a long term in order to minimize the annual cost of heat supply [1].

6.2 Results on the new method

The entire approach for the new method was introduced in Section 4.6. and 4.7. The most relevant results are presented here. Fig. 11. present LCOE for different combinations of energy supply technologies, based on heating load profile marked with Case 1. Under the reference year, the LCOE varied from 0.085 – 2.554 EUR/kWh. Therefore, for the purpose of better representation and further analysis, the found values were sorted in two categories: lower than 0.2 EUR/kWh and higher than 0.2 EUR/kWh. According to [57], the cost for electricity generation in Europe varies from low 0.06 EUR/kWh to high 0.19 EUR/kWh depending on technology and local conditions. Therefore, Fig. 11. shows the LCOE results for the analyzed PCs that are competitive with power generation cost and consequently, with the direct electric heating.



a) low LCOE

b) Contribution of fuel cost in LCOE

Fig. 11. Low LCOE and fuel cost in these plant combinations

In this study, it was assumed that electric boiler would be used to cover heat load in the DH system due to limitations in combined operation of the HP, the CHP, the HOB, and during extreme operation situations. From Fig. 11.a it can be seen, that heat energy produced by electric boiler constitutes a high portion of the LCOE. Due to high value of O&M cost, the operation of electric boiler makes DH not competitive in comparison to direct electric heating. Next, it can be noticed that the HP's contribution to the LCOE was relatively low for presented plant combinations. From this point, it can be concluded that presented heat capacities of the HP fits well to the analyzed PCs. The exception was combination PC25, where the 8.4MW HP was operated as a peak load plant. This means that the HP should not be used as a peak load plant with a high installed heat rate.

Fig. 11b shows again that the highest fuel cost of each combination was due to operation of electric boiler. The exceptions were PC2 and PC3, where the HOB was operated as an intermediate load plant. In addition, PC1, PC14, and PC25 operated without electric boiler. Due to high COP of the HP, the electricity use was low in comparison to total LCOE value presented in Fig. 11.a. In the countries with low electricity prices, like in Scandinavia, the employment of the HP for heat supply purpose is a good option of efficient heat energy supply. The fuel use for the CHP was low, even for configuration where its heat load share was 60 %. The similar trend was found for the HOB operation.

As it was highlighted earlier, the electric boiler was used during extreme operation situations. Fig. 12 shows combined operation of energy supply plants based on PC28, where the HOB was used as base load plant covering 20% of the maximum heat demand, the HP was used to cover the intermediate load covering 40 % of the maximum load, and the CHP was utilized to cover the peak load with 20 % of the maximum load.

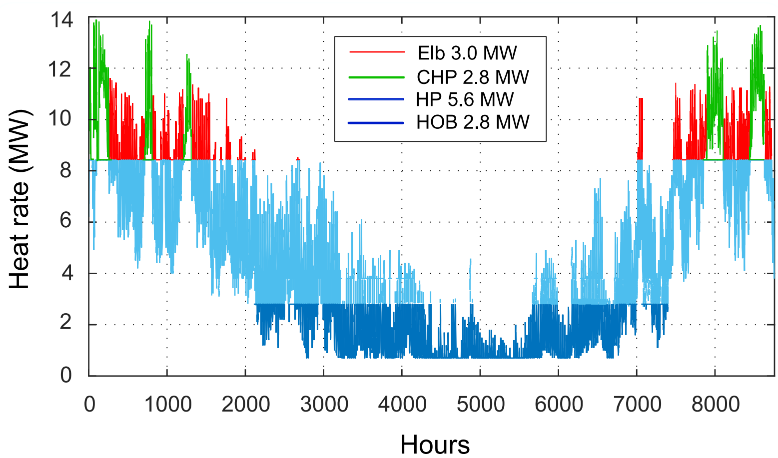


Fig. 12. Hourly heat rate distribution for the PC28

From Fig. 12 it can be seen that due to limitations in CHP operation, see Section 4.1, the electric boiler was used to cover DH load when CHP was in standstill mode. In general, to run electric boiler is convenient due to simplicity and no limitations in operation regimes. However, in a long-term operation this can lead to an increase in DH price, which existing and new customers consider impermissible.

The second part of PCs consist of combinations where the LCOE values were higher than 0.2 EUR/kWh, see Fig. 13.

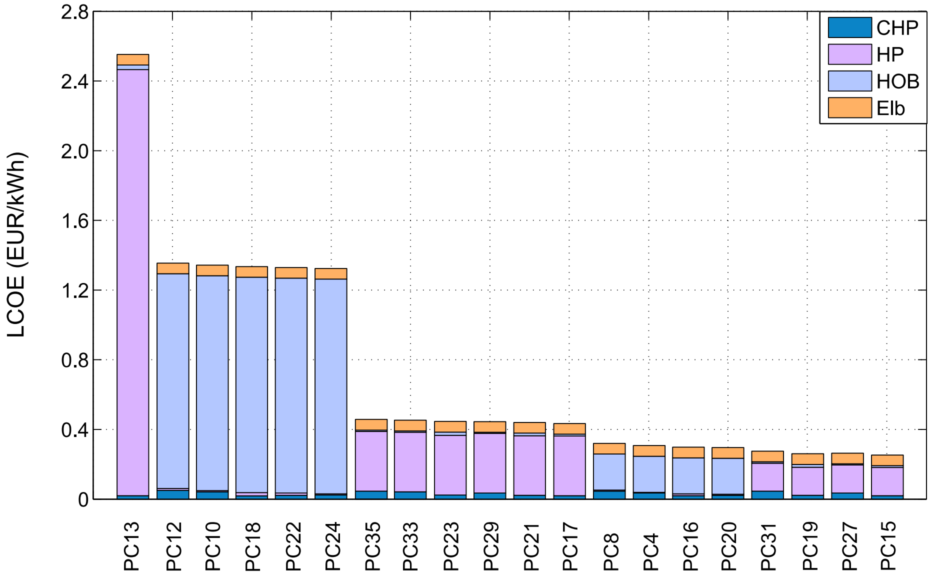


Fig. 13 High LCOE

It was found that the contribution of the electric boiler to LCOE was equal in all combinations. This meant that it was not operated. These values present the investment cost for this technology. Next, the high values of the LCOE for the HOB and the HP were due to low heat load factor. However, in the case of the CHP, the low heat load factor was substituted by electricity production and corresponding heat allocation factor of utilized fuel. Therefore, there was no high discrepancy between the presented CHPs' cost in the LCOE and it was very low. PC13 showed the highest value of LCOE. The reason for this is the same as for the PC25, where the HP with the high heat capacity was operated as the peak load plant.

Changes in the LCOE due to different heat load patterns were also investigated. The analysis was performed for combinations that have low LCOE and introduced in Fig. 11.

Fig. 14 present the LCOE for different heat load patterns and different combinations of energy supply technology. To recall, Case 2 introduces the scenario where the heat duration curve was under high occupancy and lower outdoor temperatures, Case 3 shows scenario where heat duration curve is constructed for future building stock.

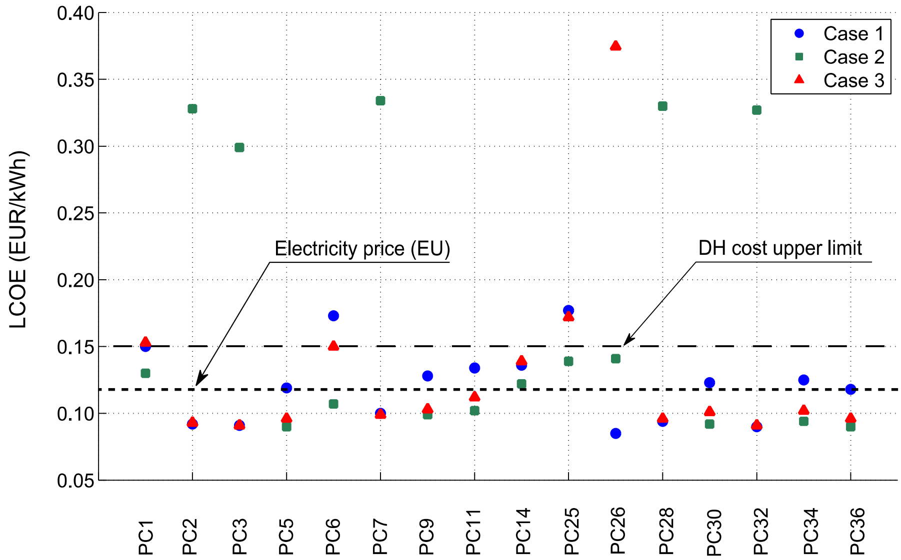


Fig. 14 LCOE values for analyzed scenarios

In order to stay competitive on the energy market, the heat generation cost should be lower than alternatives. At this point, this means that heat generation cost should be lower than the electricity production, to avoid switching to the direct electric heating. As it can be seen from Fig. 14, several combinations could be highlighted to be competitive in a long-term perspective, because the gave the low heat cost regardless of the heat load change These combinations were: PC5, PC30, PC34, and PC36. Four additional combinations PC1, PC9, PC11, and PC14 could be underlined as an alternatives with the LCOE values lower than 0.15 EUR/kWh. It can be noticed, that all these combinations have a small CHP as a peak load plant. The exception is combination PC14, where a large HOB was utilized for this purpose. Further, in comparison to all the PCs presented in Fig. 7., the above-mentioned combinations found the lowest LCOE values under the duration curve of Case 2. This means that the heat load factor increased, which provided better energy utilization in mentioned combinations. The found plant sizes fitted perfectly to required DH loads.

Among eight PCs (PC1, PC5, PC9, PC11, PC14, PC30, PC34, PC36) only one employed the CHP as a base load plant. In addition, its heat capacity was only 2.8 MW. At the same time different sizes of the HOB and the HP were utilized for the base load plant. For the intermediate load plants the trend was similar, while for peak load plants the most of combinations employed the small CHP. The found trend for peak load plants was found due to application of CHP's allocation method.

The summary of the LCOE values under different heat load profiles can be seen in Table 5.

Table 5. Heat generation cost under different load profiles

Combination	Case1 (EUR/kWh)	Case 2 (EUR/kWh)	Case 3 (EUR/kWh)
PC1	0.150	0.130	0.153
PC5	0.119	0.090	0.096
PC9	0.128	0.099	0.103
PC11	0.134	0.102	0.112
PC14	0.136	0.122	0.139
PC30	0.123	0.092	0.101
PC34	0.125	0.094	0.102
PC36	0.118	0.090	0.096

Table 5 shows that the variation in the heat generation cost due to change in heat load patterns was in the range of 12.2 – 25.2% or 0.017 – 0.031 EUR/kWh of heat. The lowest differences were found for the combinations PC14 and PC30. At this point it could be concluded that these two combinations were the best solution for customers due to smallest change in DH cost under different heat loads. However, combinations PC5 and PC36 should be highlighted, because they showed generation cost reduction for both increase and decrease of the DH load. In PC36, a 8.4 MW HOB was employed for the base load plant, a 2.8 MW CHP covered intermediate load, and a 2.8 MW HP was used for peak load. PC5 included the following plants: a HP of 2.8 MW for the base load plant, a HOB of 8.4 MW for the intermediate load, and a CHP of 2.8 MW for the peak load. The combinations presented in Table 5 showed the lowest LCOE for different heat load profiles among all the 36 combinations. This is very important, since employing these combinations DH customers would pay upon consumed heat based on best matched operation of heat production units.

Fig. 15 shows relation between the LCOE and system efficiency for different PCs under different heat load profiles.

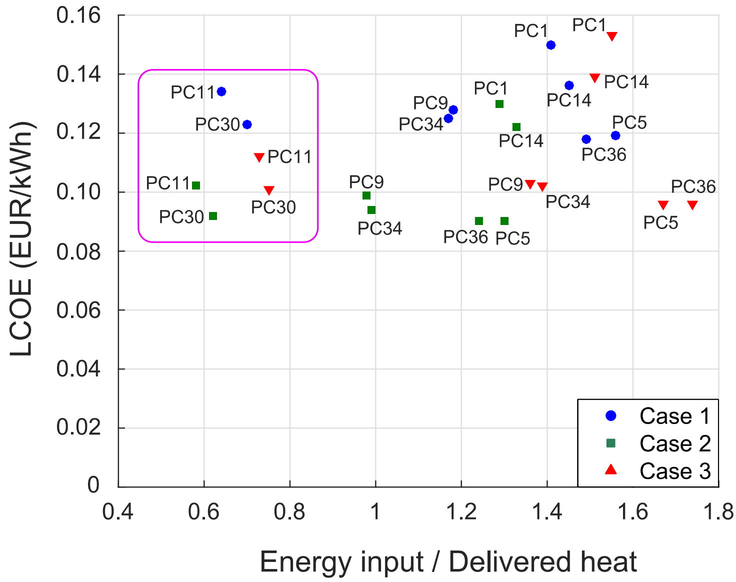


Fig. 15 LCOE and system efficiency for different heat supply options under three heat loads

Fig. 15 shows that plant combinations PC11 and PC30 are more energy efficient under different heat loads than other combinations. As it found before, the PC30 and PC14 had the lowest difference in values of LCOE under different heat loads, see Table 5. However, Fig. 15 shows that in terms of energy input per delivered heat, the PC30 is more efficient than PC14. Apart from PC30, the low value was found for combination PC11. The reason for this is that both PC11 and PC30 employed large HP for base load and intermediate load, respectively.

6.3 Uncertainties due to fuel price volatility, variation in investment cost, and model quality

The uncertainty analysis performed in this section was executed for eight PCs with low LCOE and showed in Table 5. The analysis was based on values from the literature review and presented in Table 1. The following fuel prices were considered: the minimum for electricity was 0.113 EUR/kWh, for wood chips was 40 EUR/kWh, while the maximum for electricity was 0.127 EUR/kWh and for wood chips 136 EUR/kWh.

The analysis found that the highest variation in total LCOE had combinations where the HOB was operated as an intermediate load plant. This mean that increase in the fuel price would have negative effect on LCOE for this technology. The total deviation in LCOE values for the presented PCs due to price volatility was in the rage of 1.6% – 3.6% or 0.002 – 0.005 EUR/kWh. The largest deviation for the CHP fuel cost was found in combinations where the CHP was operated as an intermediate load plant (PC1, PC6 PC14, and PC25), while the smallest deviation was found where the CHP was operated for the peak load. The largest deviations for the HOB fuel cost were found for the HOB operated as the intermediate load plant for small and intermediate heat capacities. Further, operation of the HOB as a base load plant showed the smallest variance in cost. In comparison to the results found for the CHP and the HOB, the consequences of the HP's price variation were minor in the analyzed range. One of the reasons for this is that the cost foundation for electricity production and wood chips collection is different.

The uncertainty due to variation in investment cost showed that the increase in the CHP's investment cost by 20% induced changes in the LCOE by 15 – 16% for the analyzed combinations. When the CHP's investment cost were decreased by 20%, the change in LCOE constituted around 19%. In comparison to the CHP, the change in investment cost for the HP and the HOB had similar trend. The increase and decrease in the HP's investment cost by 20% led to change in total annual cost by around 14 – 17 %. For the HOB these values were in the range of 14 – 20%. Hence, underestimation of investment cost can lead to significant changes in LCOE values for these technologies.

The introduced energy plant models presented Section 4 were simplified by using polynomial models as shown in Fig. 2., Fig. 4. and Fig. 6.. Even though the obtained goodness of fit (R^2) was high, some uncertainty could take place.

The uncertainty due to model quality showed that The HP's model had larger effect on LCOE in comparison to the CHP and the HOB model. The deviation in the HP model in the range of ±10% induced a change in LCOE by 1.42 – 4.7%. In the case of the HOB and the CHP models, the consequences were smaller, around 1%. The impact of multiple uncertainties simultaneously induced changes in the range of 1%. The conclusion is that the introduced models proved to be accurate enough for this analysis.

7 Discussion

The existing method of heat supply optimization was found to be simple to treat all the costs and operation issues. A number of additional important factors affecting plants operation are missing. It was found that the method is sensitive to change in heat load profiles. In turn, this could lead to low load factor for operated plants and further increase the DH cost. Further, the calculated DH cost showed that with the decrease of heat load, the DH cost decreases. However, it is not always the case due to possible mismatch in plants' operation. This means that more operation hours required fulfilling the same DH load and increase in DH cost is inevitable. In addition, the existing method is also sensitive to variation in investment cost, while the variation in energy cost induced minor changes to DH cost. All these causes misleading results, affecting the DH cost foundation. Further, it is very simplified with respect to real sizes, operation times, and actual plant performance. In comparison to the existing methodology, the new method suggested by the authors is sophisticated and involves deeper analysis.

The analysis of found results for the new method showed that the operation of the electric boiler could be avoided and DH companies should eliminate this technology from the DH system. In all the analyzed combinations the electric boiler operation constituted from 38 to 790 hours of intermittent operation at full heat capacity. As an alternative to this, the thermal energy storage (TES) could be considered. In addition, employing TES could lead to increase in heat load factor for intermediate and peak load plants.

All the PCs showed high LCOE values due to operation of electric boiler. The LCOE remained high even when electric boiler was not put in operation. The reason for this might be the high value of the fixed O&M cost used in the analysis. This value was adopted from technical report [31] with reference in 2012 year. It can be doubted about meaningfulness of this value. However, the report dated two years earlier showed this value in the same range that makes adopted value be reasonable. Therefore, some changes in this value might change the results of the study. However, any decrease of this high value of the fixed O&M cost would give a decrease in the DH cost.

Further, it was not appropriate to conclude that all the combinations presented in Fig. 13 were not competitive to direct electric heating. As it was discussed previously, the found values were mainly due to actual operation and low heat load factor facing those combinations. Hence, at this point, it is possible to look at those combinations at an angle of future development and extension of DH systems.

The fuel between heat and power production was allocated by the energy method. In turn, this made the CHP operation highly efficient due to substitution of low heat load by electricity load and further fuel allocation to power production. This showed that the CHP operation as a peak load plant was efficient. However, a number of technical allocation methods were developed and used in different countries. Therefore, the possible deviations in LCOE might be present due to application of different allocation methods.

The example with the existing method of heat supply optimization found that it was inappropriate to utilize the CHP due to its high investment cost. However, the new method showed opposite. The small CHP plants could be employed for peak load operation. This was a god observation, since this goes along with the Directive 2004/8/EC [64] on promotion of highly-efficient cogeneration. The more CHP used, the more primary energy is saved and the higher the security of the energy supply.

If one considers four technologies discussed in this study, it was shown that modern HOBs were very efficient. In comparison to other technologies, its linear cost characteristic could show decrease with the increase of operation hours. This provides possibility to employ a single HOB for annual operation. However, the employment of a single plant decreases security of supply in the DH systems. To avoid this, the need in several heat production units arises. Hence, the cost difference utilizing four plants would always be higher than with three or two. Therefore, it can be concluded, that with the increase of DH's flexibility and reliability of supply, the heat generation cost increases.

8 Conclusion

In this paper, the economic issues associated with the decision on heat production plant combinations were analyzed. The study focused on the situation when there is a need in construction of a set of plants for new DH system. Three heat duration curves together with three highly efficient energy conversion technologies were considered. The existing method of heat supply optimization was compared to the new method.

The results on the new method found that the operation of electric boiler led to high value of the LCOE, in spite of the fact, that it was operated sporadically and maximum heat output was 3 MW of heat. Next, one should consider electricity rates, since not many countries have cheap electricity like in case of Norway and Sweden. This revealed that operation of electric boiler was rather expensive and should be limited to minimum. In addition, policy makers should provide legislative framework to ban this technology from DH.

The study identified sixteen PCs with the LCOE under 0.2 EUR/kWh. However, not all of them were found non sensitive to change in heat load profiles. Further, eight PCs were selected as those with low sensitivity to heat load variation and the LCOE under 0.15 EUR/kWh (PC, PC5, PC9, PC11, PC14, PC30, PC34 and PC36). It was noticed that six of those had a small CHP as a peak load plant. However, it was opposite compared what the existing method suggested. Among the eight combinations only one employed the CHP as a base load plant with heat capacity of 2.8 MW. At the same time, the HOB and the HP technologies utilized all there sizes for the base load plant. For intermediate load plants the trend was similar, while for the peak load plants, most of the combinations employed a small CHP. It was concluded that the operation of a large HP for the peak should be avoided due to low heat load factor and high investment cost.

The change in heat load profiles showed that with the increase of heat use (Case 2), the mentioned eight combinations showed the lowest LCOE. This meant that the heat load factor increased that provided better energy utilization. The found plant sizes fitted perfectly to satisfy required DH loads. The lowest difference in the LCOE under different heat loads were found for the combination PC14 and PC30. These two combinations were the best solution due to smallest change in DH cost under different heat loads. The normal trend of DH cost was increase over the years due to change in heat load, however, PC5 and PC36 showed that DH generation cost could be lowered. This was a good finding for future development of DH and for customers due to protection against increase in price.

The analysis on system efficiency found the most rational utilization of energy input under different heat loads had combinations PC11 and PC30. The main reason for this is that large HP was used in these combinations to satisfy the base load and intermediate load.

The uncertainty in fuel price found that the highest variation in the LCOE had combinations where the HOB was operated as an intermediate load plant. This means that increase in fuel cost would have negative effect on the LCOE for this technology employed for intermediate load. The total deviation in the LCOE values for presented combinations due to price variation was in the rage of 1.6% – 3.6% or 0.002 – 0.005 EUR/kWh. The consequences of price variation for the HP were smaller than for the CHP and the HOB in the analyzed range. One of the reasons for this was that the cost foundation for electricity production and wood chips collection was different. However, in some countries electricity rates are rather high and a normal trend is its increase within the time. In turn, this can lead to additional portion of O&M cost when HP technology is chosen for operation.

The uncertainty in the PCs due to changes in investment cost in the range of ±20% had an effect of 14 – 20% on the LCOE. Hence, underestimation of investment cost can lead to significant changes in LCOE values for these technologies.

The uncertainty due to model quality found that the HP's model had larger effect on the LCOE in comparison to the CHP and the HOB. The deviation in the range of ±10% induced change in LCOE by 1.42 – 4.7%. In the case of the HOB and the CHP models, the consequences were smaller, around 1%. The impact of multiple uncertainties simultaneously found changes in the range of 4 – 6%. The conclusion is that presented models and the analysis approach proved to be accurate enough for the purpose of this study. Thereby the results and conclusions might be treated as reliable.

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Appendix

Table 6- Table 9 provides a summary of different costs for the following technologies: biomass HOB, CHP, HP and electric boiler. The presented data is given based on LHV of fuels.

Table 6. Investment and O&M costs for biomass HOB

Heat output	Efficiency (%)	Investment costs (MEUR/MW)	Fixed O&M cost (EUR/kW)	Variable O&M cost	Reference
1 MW	108	0.5	Total O&M 5.4 EUR/MWh		[31]
5 MW	108	0.75	Total O&M 5.4 EUR/MWh		[31]
5 MW	88	0.29	Total O&M 278180 EUR		[65]
5.8 MW	90	0.82	Operational costs 1110 kEUR/year		[66]
10.3 MW	110	0.4	2 EUR/MWhfuel	2 EUR/MWhfuel	[67]
12 MW	108	0.5	10	2 EUR/MWhfuel	[6]
12 MW	108	1.1	Total O&M 5.4 EUR/MWh		[31]
28.5 MW	110	0.36	2 EUR/MWhfuel	2 EUR/MWhfuel	[67]
50 MW	108	0.42	8.3	2 EUR/MWhfuel	[68]
200 MW	101	0.09	3.3	-	[69]
400 MW	110	0.33	2 EUR/MWhfuel	2 EUR/MWhfuel	[67]

Table 7. Investment and O&M costs for biomass CHP

Heat/power output	Efficiency (%)	Investment costs (MEUR/MW)	Fixed O&M cost (EUR/kW)	Variable O&M cost	Reference
1 MW	heat - 78 electric - 25	3.6 of heat	3-4 % of investment per year		[31]
5 MW	heat - 78 electric - 25	4.64 of heat	3-4% of investment per year		[31]
5 MW	total - 90	6.0 of heat	Total O&M 0.055 EUR/kWh		[70]

0.5 MWel 5.5 MWheat	electric - 18 total - 83	0.56 of heat 4.71of electric	0.128 EUR/kWel 0.0367 EUR/kWheat		[71, 72]
1.0 MWel 5.8 Mwheat	heat - 65 electric - 24	4.2 of electric 0.4 of heat	Total O&M 0.032 EUR/kW		[66]
10.3 MW	electric - 25 total - 105	3.9 of electric	2 EUR/MWhfuel	2.6 EUR/MWhfuel	[67]
10 MW	heat - 78 electric - 25	4.9 of heat	3-4% of investment per year		[31]
17 MW	heat - 81 electric - 24	1.85 of heat	41	2.4 EUR/MWhfuel	[6]
5 MWel 18 MWheat	electric - 22 total - 104	6.49 of electric	157	2.3 EUR/MWhfuel	[63]
28.5 MW	electric - 27 total - 110	2.3 of electric	2 EUR/MWhfuel	2.6 EUR/MWhfuel	[67]
30 MW	heat - 77 electric - 29	2.6 of heat	29	3.9 EUR/MWh	[31]
30 MW	heat - 79.5 electric - 26.5	1.72 of heat	35.2	2.9 EUR/MWhfuel	[68]
10 MWel 28 MWheat	electric – 27 total - 105	5.15 of electric	116	2.3 EUR/MWhfuel	[63]
50 MW	heat - 81 electric - 29	1.68 of heat	34	kEUR/MW year 24.1	[69]
80 MW	electric - 30 total - 110	1.7 of electric	2 EUR/MWhfuel	2.6 EUR/MWhfuel	[67]
81 MW	heat - 81 electric - 29	1.47 of heat	24.8	3 EUR/MWhfuel	[68]
30 MWel 75 MWheat	heat - 60 electric - 30	3.0 of electric	2.1 EUR/MWhfuel	2.5 EUR/MWhfuel	[73]
30 MWel 76 MWheat	electric - 28 total - 105	4.06 of electric	77	2.3 EUR/MWhfuel	[63]
36 MWel 72 MWheat	electric - 30 heat - 60	1.5 of electric	37 EUR/kWel	4.5 EUR/MWh el	[74]
199 MW	heat - 77 electric - 31	1.18 of heat	17.6	3.1 EUR/MWhfuel	[68]
80 MWel 195 MWheat	electric - 31 total - 106	3.23 of electric	55	2.3 EUR/MWhfuel	[63]
479 MW	electric - 34 total - 110	1.3 of electric	2 EUR/MWhfuel	2.6 EUR/MWhfuel	[67]

Table 8. Investment and O&M costs for HP

Heat output	COP	Investment costs (MEUR/MW)	Fixed O&M cost (EUR/kW)	Variable O&M cost	Reference
1 MW	3.2	0.51	4.2 EUR/kW		[31]
3 MW	3.2	0.67	5.9 EUR/kW		[31]
5 MW	3.2	0.79	7.3 EUR/kW		[31]
5 MW	3.3	0.7	7.0 EUR/kW		[69]
10 MW	3.2	0.6	0.5	0.7 EUR/MWhfuel	[67]
10 MW	2.8	0.52	3.7	0.2 EUR/MWhfuel	[6]
11.2 MW	3.0	0.21	8.9 EUR/kW		[75]

Table 9. Investment and O&M costs for electric boiler

Technology	Efficiency (%)	Investment costs (MEUR/MW)	Fixed O&M cost (EUR/kW)	Variable O&M cost	Reference
1 - 3 MW	electric - 99	0.14	1.1	0.5 EUR/MWh	[31]
10 MW	electric - 99	0.08	1.1	0.5 EUR/MWh	[31]
20 MW	electric - 99	0.06	1.1	0.5 EUR/MWh	[31]

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The allocation factors of heat and electricity production of combined cycle power plant

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ABSTRACT

The main objectives for this work were analysis of design and off-design conditions of the Combined Cycled Power Plants (CCPP) plant with regard to different allocation methods. Change in operation could have an effect on emissions allocation in the CCPP plant. Therefore it is important to do investigation on allocation methods. Seven allocation methods were tested on the CCPP plant in this study. The results showed which factors influenced mostly operation of CCPP and allocation factors. Sensitivity analysis of the different allocation methods was performed for CCPP under a yearly heat and electricity load. It was noted that the most sensitive method due to change in variation was power bonus method. The study showed that the decision of allocation method should be carefully analyzed before implementation in the standards and different policies, because benefits from cogeneration technology and distribution systems should be enabled.

KEYWORDS: Emissions allocation, Allocation methods, Parametric studies, District heating

1. INTRODUCTION

The reduction of CO_2 emissions is a challenge for the coming decade, especially with the implementation of the Kyoto protocol. Beside transport, heating is responsible for a large share of the total greenhouse gas emissions [1,2]. One way to decrease the emissions generated by energy services (heating, hot water, electricity), is to increase the efficiency of the different energy conversion technologies that provide these services, by combining them in a polygeneration energy system. In the case of District Heating (DH) for instance, polygeneration systems could save over 60 % of the energy resources and emissions compared to conventional solutions [3-6]. The simplest example of such a system is Combined Heat and Power (CHP) plant. When DH is generated in highly efficient CHP plants, it is a reasonable and well-established measure to increase energy efficiency and to promote the resource saving use of primary energy carriers [7].

In CHP plants, heat and electricity are generated simultaneously. Consequently, it is difficult to precisely allocate the primary energy input, emissions or operating costs to each of these energy outputs. In order to address this problem, different allocation methods have been developed [7]. The main strategy for CHP plants today it is to be more environment-friendly and energy efficient. The DH technology can provide the possibility of decreasing pollutions in combination with CHP plants. Unfortunately not all CHP plants use renewable energy sources like biofuel or municipal waste for producing heat and power. This is one of the reasons why allocation methods should be used in CHP plants in order to allocate CO_2 emissions.

The CHP plant produces electricity and heat, while the delivery of these two products is performed by different companies. The method for emissions' allocation is needed to ensure that each part is credited with its appropriate share of the emissions from the system. In addition, having a meaningful allocation method allows the sources of CO_2 and other emissions to be better understood and, where appropriate, reduced [8]. The choice of allocation method will have a great effect on energy pricing and CO_2 allocation in CHP. Therefore it is important to carry out investigation in this field. The most recognizable method of fuel allocation is Power bonus method given in the standard EN15316:2007 [9]. The Economic-based allocations are not investigated in this paper since such methods are prone to be misleading and fluctuate markedly with price swings for fossil fuels. Economic-based allocations are easily influenced by decision and policy makers [8].

Many studies have been devoted to investigation design conditions of CHP plants. The focus so far has been on describing the thermodynamic principles of combined cycles at design point and practical design considerations. However it must be realized the operating conditions change, and the system should be able to operate at conditions far from design point. Off-design theory is about predicting how the system reacts to parameter changes. The CHP plant may operate for prolonged time at off-design conditions, depending on power demand, ambient condition and other considerations. This will have significant impact on the plant performance [10]. Therefore the need increases for analysis and comparison of design and off-design parameters of the CHP plant in combination with allocation methods. Nowadays CCPPs are receiving major attention throughout the world as one of the most effective options among the various energy conversion technologies. This technology is well developed and has been widely accepted in fossil-fired power plants due to its higher efficiency [11]. In this paper CCPP has been analyzed and the results presented focus on CCPP integrated in a DH system.

The aim of this paper is to investigate the effects of the different parameters which the system undergoes during the year. The goal was to compare system operation in design conditions with off-design conditions and to see how these different conditions will effect on choice of allocation method. The modeling of the system was carried out by the simulation software Aspen HYSYS [12] and data post processing was done in MATLAB [13].

2. METHODOLOGY

Firstly, the allocation methods were introduced. In order to calculate allocation factors, it is necessary to calculate total electricity and heat energy production in CHP plant. The rest of the energy system is considered by including heat and electricity use at the customer side.

2.1. Allocation methods

Seven different allocation methods were analyzed in this paper. The methods are given in the following text.

The *energy method* is most widely used because of its simplicity. This is an example of physical allocation. The primary energy consumption is allocated between heat and electricity produced in the CHP plant. This method does not take any exergy of energy quality aspects into account, allocating lower impact to electricity than to the other methods [14].

The *alternative generation method* was developed by the Finnish District Heating Association [15]. In the alternative generation method, the share of CO_2 emissions is beneficial for both the heat and the power production in the CHP plant. The method allocates emissions and resources to the heat and power production in proportion to the fuel needed to produce the same amount of heat or power in separate plants. These alternative plants use the same fuel as the CHP plant [16].

The *power bonus method* is the most recognizable method for energy allocation because it is promoted by the European standard EN 15613-4-5 [9] and it is widely used nowadays. In this method the heat is the main product, while all power is considered as a bonus. The primary energy is allocated to the electricity produced in the CHP plant.

The *exergy method* represents allocation from a thermodynamic point of view. This is an example of physical allocation; it defines the quality of energy. For exergy analysis, the characteristics of the reference environment must be specified completely. This is commonly done by specifying the temperature, pressure and chemical composition of the reference environment [17]. The method is judged as the fairest method from a thermodynamic point of view, for dividing the benefits of CHP production between electricity and heat [18].

The *200 % method* uses 200 % efficiency for heat production. This means that all emissions are left to power production. This method, which established by The Danish Energy Agency [20] is similar to the Power bonus method, where all electricity is counted as bonus. The Danish Energy Authority has stipulated that energy efficiency of 200 % has to be used when allocating the fuel costs of CHP to the heat production in the energy and emission statistics. This means that, in order to produce two units of heat energy, one unit of real fuel has to be used and the other unit will be recovered from the heat otherwise directed to the turbine condenser. In the condenser, the heat unit would be wasted to the environment if not recovered to district heating [21].

The publicly available Specification *PAS 2050* [22] is the British standard, which explains the calculation of Greenhouse Gas Emissions (GHG) of goods and services. The allocation of emissions in CHP is between the heat and power produced, multiplied by the intensity of the

GHG emissions of production unit. The special coefficient specifies the emissions released from fuel combustion used in the system.

The *Dresden method,* which was proposed by Zschering and Sander [23], is based on exergetic assessment The Dresden method describes how to evaluate the electricity loss caused by the heat extraction (water steam condensation) in CHP plant. The electricity loses due to heat extraction in CHP plant should be calculated and taken into account.

The summary of allocation methods analyzed in this paper is presented in the Table 1.

Table 1 Allocation methods summary

Method	Allocation factor heat	Allocation factor electricity
Energy method	$f_Q = Q/(Q+E)$	$f_E = E/(Q+E)$
Alternative generation method	$f_Q = (\frac{Q}{\eta_{alt_{heat}}})/(\frac{Q}{\eta_{alt\_heat}} + \frac{E}{\eta_{alt\_elec}})$	$f_E = (\frac{E}{\eta_{alt_{elec}}})/(\frac{Q}{\eta_{alt\_heat}} + \frac{E}{\eta_{alt\_elec}})$
Power bonus method	$f_Q = f_{P,dh} \cdot Q_{del}/(Q_{del} + E_{del})$	$f_E = 1 - f_{P,dh} \cdot Q_{del}/(Q_{del} + E_{del})$
Exergy method	$f_Q = Ex_Q/(Ex_Q + Ex_E)$	$f_E = Ex_E/(Ex_Q + Ex_E)$
200% method	$f_Q = Q/(2 \cdot Fuel_{in})$	$f_E = 1 - Q/(2 \cdot Fuel_{in})$
PASS 2050	$f_Q = Q/(n \cdot E + Q)$	$f_E = (n \cdot E)/(n \cdot E + Q)$
Dresden method	$f_Q = \Delta E/E$	$f_E = (E - \Delta E)/E$

2.2. Heat and power production in CCPP

To include the entire energy system, for the consumer side real energy use at a university campus was analyzed. Total heat use, measured at the primary side of the consumer substation, can be estimated as:

$$\boldsymbol{Q_{del,j}} = \int \dot{\boldsymbol{Q}}\, \boldsymbol{d\tau} = \lim_{\boldsymbol{\tau \to 0}} \sum\nolimits_{\boldsymbol{i}} \ \dot{\boldsymbol{Q}}_{\boldsymbol{\imath}} \cdot \Delta \boldsymbol{\tau_i} \tag{1}$$

where $Q_{del,j}$ is total heat consumption at primary side of customer substation, $\dot{Q}_\imath$ is heat effect required during i - th heating hour, $\Delta\tau_i$ is the duration, $\dot{Q}_\imath$ the heat load.

The electricity use of the university campus can be calculated based on the following:

$$E_{del,j} = \int \dot{E}\, d\tau = \lim_{\tau \to 0} \sum_i \dot{E}_i \cdot \Delta\tau_i \tag{2}$$

where $E_{del,j}$ is total power consumption at primary side of dwelling, E_i is power demand, and $\Delta\tau_i$ is duration of the electricity load.

The CCPP was simulated based on the required heat energy use; the details of the simulation model are described in the next section. The input in simulation model was thermal energy and output was power produced and fuel input in CCPP. The information flow for the methodology used in this study is given in Fig. 1.

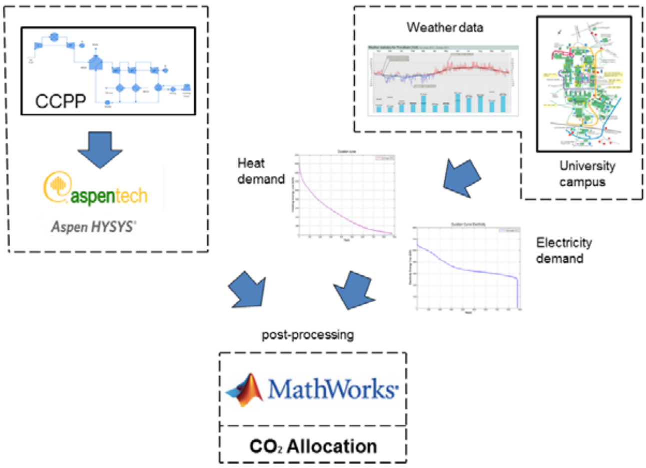

Fig. 1 The flowchart represents steps of analysis done in this paper

3. CASE STYDY

A small-sized DH system with an annual heat load around 27 GWh was analyzed in this paper. The load was represented by the university campus. The system was modeled by using Aspen HYSYS simulation software. The ambient temperature at the design point was +15°C, ambient pressure was 1.013 bar and air Relative Humidity (RH) was 60 %. The energy source for DH was CCPP with supplementary firing technology. The CCPP consisted of gas turbine cycle (GTC), steam turbine cycle (STC), heat recovery steam generator (HRSG), two combustion chambers, fed with natural gas and other components. The schematic layout of the system is represented in the Fig. 2, and summary of design parameters are summarized in Table 2. In this simulation natural gas was used as a fuel. The lower heating value (LHV) of the gas was 50.03 MJ/kg. The air and fuel are supplied to the reactor after compressing it in two stage compression system. The air was compressed up to13 bar before entering the reactor, see Fig. 2. The air excess coefficient α was set to be 3.2 in the air-fuel mixture.

In the design stage, the temperature before the GTC was assumed to not exceed +1100°C. The temperature of flue gases entering the gas turbine after conducting simulation was set to be 1086°C.

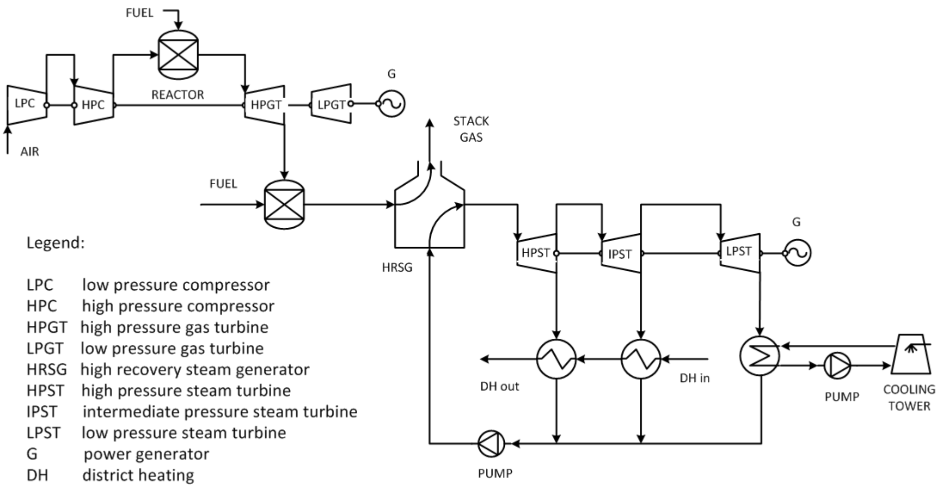

Fig. 2 Schematic of CCPP

The entering pressure of flue gases in the High Pressure Gas Turbine (HPGT) was 13 bar. The pressure before the Low Pressure Gas Turbine (LPGT) was 6 bar. The leaving pressure was 1.5 bar, which is slightly higher than ambient conditions.

Table 2 Design parameters of CCPP

Parameter	Value
Ambient pressure	101 kPa
Air relative humidity	60 %
Ambient air temperature	+15°C
Pump pressure	60 bar
Steam turbine inlet temperature	+500°C
Condensing pressure	0.05 bar
Air excess in air-fuel mixture	3.2
Fuel temperature	+15°C
Gas turbine adiabatic efficiency	0.9
Steam turbine adiabatic efficiency	0.9
Compressor adiabatic efficiency	0.9
Supplementary firing temperature	+900 °C

In the CCPP with supplementary firing technology, the supplementary firing provided additional energy input to the steam cycle. In this way the flue gas temperature was increased.

The combustion of supplementary fuel was accomplished by the air excess leaving the gas turbines. The fuel was mixed with flue gases and burned in duct burners in the HRSG. In the design case, the temperature of the exiting flue gases was set to +900°C.

The HRSG was modeled as three stages or heat exchangers; see Fig. 2. These are an economizer, an evaporator and a superheater. The HRSG had one steam pressure level. The parameters of the live steam entering the steam cycle were: T = +500°C, p = 60 bar. The STC represented three units. The first was a high pressure steam turbine (HPST), the next was an intermediate pressure steam turbine (IPST) and the last was a low pressure steam turbine (LPST). The entering parameters of working medium in the IPST were 12 bar and temperature +278°C. In the LPST, the steam condenses till the pressure of 0.05 bar. The adiabatic efficiency of the STC was assumed to be 90 %. The STC is with one extraction for DH purposes. The mass flow rate of water from the DH is satisfied by means of heat transfer connected with the heat exchange units. The DH system was fed from the IPST. The steam extraction occurred at a pressure of 10 bar.

The heat duration curve of the university campus is presented in the Fig. 3.

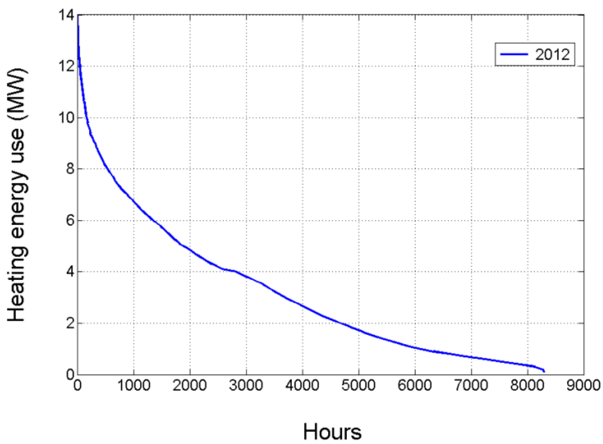

Fig. 3 The heat duration curve of the analyzed campus

The maximum of heat load was 14 MW. The temperature of supply water in DH was +105°C and return water was +50°C. The part load operation of the modeled CCPP plants was simulated by changing the mass flow rate in DH. The minimum heat load in DH system in part load simulations was 1 MW while the maximum was 14 MW.

4. OFF-DESIGN MODEL ASSUMPTIONS

To observe how the plant perform under different operational conditions, off design analysis was performed. This was important, because the plant needed to deliver energy under different conditions as explained in the previous section.

The following assumptions are common to all solutions examined:

- for the simplicity of calculation methane was treated as natural gas;
- no pressure drop in heat exchangers units;
- the plant operates all over the year;
- the maximum heat demand in DH was equal to 14 MW;
- the electricity grid was purchasing every amount of electricity produced in CCPP;
- heat loses in the system were neglected.

In the CHP design, supply companies use different standards and directives in order to achieve a stable system with the best economic and environmental characteristics. The standardized data collected from many sources and research reports gives guidelines on how to achieve best performance. The summary of the off-design parameters is given in Table 3.

Table 3 Off-design parameters of CCPP

Parameter	Value
Ambient pressure	75 kPa -101 kPa
Air relative humidity	20 % - 80 %
Ambient air temperature	-20°C - +15°C
Pump pressure	40 bar - 80 bar
Steam turbine inlet temperature	+475°C - +540°C
Condensing pressure	0.05 bar - 0.2 bar
Air excess in air-fuel mixture	3.0 - 4.0
Fuel temperature	+15°C - +200°C
Gas turbine adiabatic efficiency	0.8 - 0.9
Steam turbine adiabatic efficiency	0.8 - 0.9
Compressor adiabatic efficiency	0.8 - 0.9
Supplementary firing temperature	+700°C - +900 °C

5. RESULTS ANS DISCUSSION

5.1. Design and off-design system performance

Firstly, parametric studies of the CCPP plant shown in Fig. 2 were carried out in order to see any variation in plant performance under changeable operational conditions. The simulations were performed for DH load. The results show that variation in power efficiency was 4.02 %, see Fig. 4a, while in the maximum change in the CHP efficiency was 2.46 %; see Fig. 4b. In terms of thermal efficiency, the maximum change was 2.98 %; see Fig. 4c. The maximum change in the fuel input rate was 3075 kW, as shown in Fig. 4d, due to a change in the supplementary firing temperature. The biggest variation in the power efficiency occurred when the air excess coefficient was changed from designed $\alpha = 3.2$ to $\alpha = 4.0$, while the smallest was when air RH had been analyzed; see Fig. 4b. The air excess coefficient provided an increase in the mass flow rate of flue gases through the GTC, increasing the power production in the CCPP by 4.2 %. At the same time the fuel input to the system had decreased. In terms of CHP efficiency, this also had a positive effect. The CHP efficiency increased to 2.11 % according to Fig. 4d.

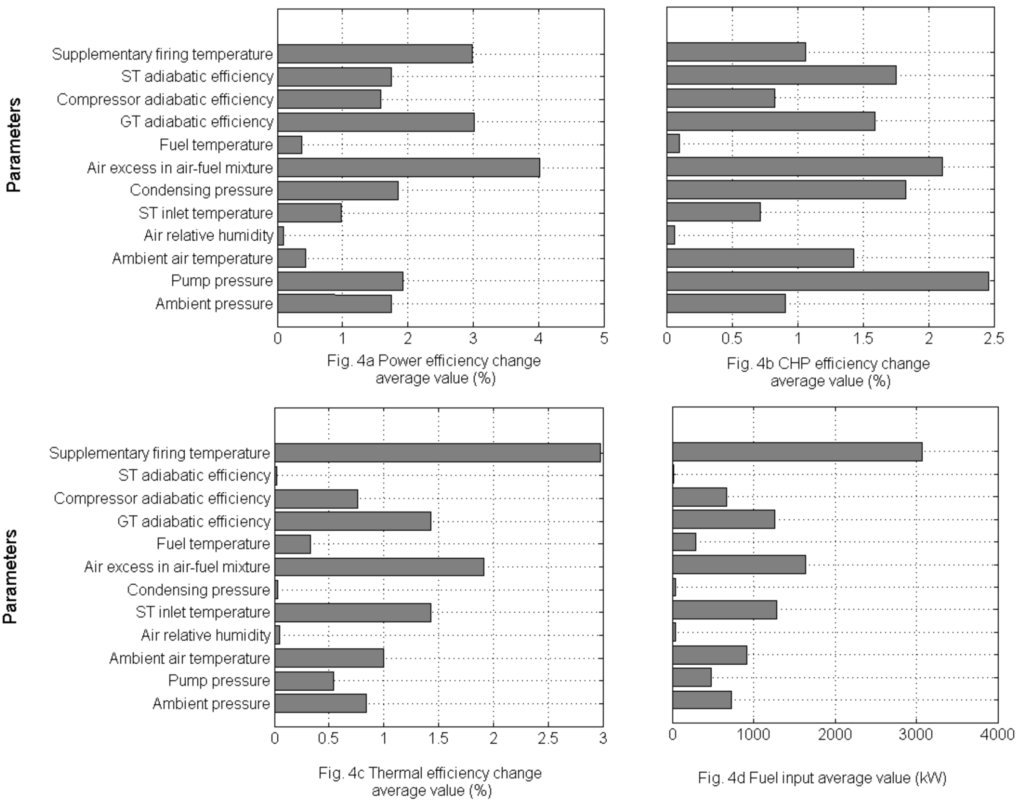

Fig. 4 Change in CCPP behavior based on analyzed parameter

The thermal efficiency of the CCPP showed the maximum change of 2.98 % when the supplementary firing temperature was set to +1000°C; see Fig. 4c. The supplementary firing provided additional energy input to the STC, which resulted in better energy utilization and system flexibility, when shifting from the base load to the high peak. Based on heat flow – temperature diagram showed in Fig. 5, we can conclude that the high temperature of flue gases before HRSG does not mean the best energy utilization. In current CCPP, the HRSG was with one pressure stage. This means that the pressure entering the economizer is the same as leaving superheater. In general it is enough to provide temperature of flue gases entering the HRSG higher than +200°C of the medium temperature leaving the superheater is enough to be provided. The higher temperature at entry provides lower energy utilization in the HRSG and increases the exergy loses. The space between the curves marked in blue or red, as presented in Fig. 5, is exergy loses. The highest temperature in the HRSG had an effect on fuel input in the CCPP. This was the maximum value during simulation and worked out in 3075 kW of fuel input; see Fig. 4d.

The supplementary firing temperature also affected the power efficiency in CCPP; see Fig. 4a. The maximum change in the power efficiency was 2.99 % when the minimum supplementary firing temperature was set. This can be explained by increasing the mass flow rate of air-fuel mixture through the GTC. The CHP efficiency showed a negligible variation of 1.06 %, see Fig. 4b, due to the supplementary firing temperature. The minimum change in the thermal efficiency occurred due to variation in the following parameters: ST adiabatic efficiency, condensing pressure, and air RH. This can be seen in Fig. 4d. The variation in condensing pressure had most effect on power production. The condensing pressure in the CCPP affected on the temperature of water-steam mixture leaving the LPST. The steam entering the pump before the economizer should not contain any water fraction; see Fig. 2. The water-steam mixture should be fully condensed up to the saturation temperature. This means that the temperature after LPST remains constant in all cases.

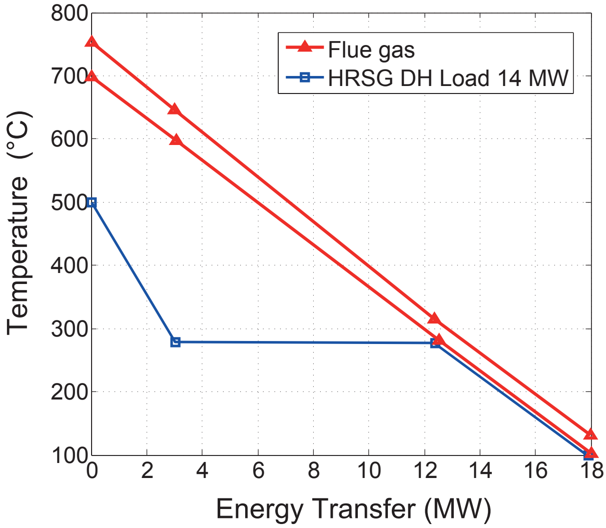

Fig. 5 Energy utilization in the HRSG where the temperature of flue gases is +700°C or +750°C

The biggest influence on the CHP efficiency was a change in the pump pressure of about 2.46 %; see Fig. 4b. The maximum change occurred when the pressure in the STC was increased to 80 bar. The higher the pressure, the higher the electricity produced in the STC. The power production increased by 1.92 % in comparison with the design case when the pump pressure was set to maximum, see Fig. 4a. The thermal efficiency did not show any particular changes due to the constant vapor temperature level in the STC.

The simulation of CCPP showed that the operational and design parameters have a significant influence on plant performance. This is valuable information since it is important to provide a reliable heat and power supply to customers while shifting from the base load to the peak load and vice versa.

5.2. Results on allocation methods

In this study different allocation methods have been analyzed in order to investigate the effect of fuel allocation between heat and electricity produced in CCPP. Allocation methods were combined with the parametric studies of CCPP and annual heat energy use at the university campus. Operating and design parameters were analyzed, and the results were combined to estimate the effect on choice of allocation method as shown in Section 5.1. Sensitivity analysis of the different allocation methods was performed for CCPP under yearly heat and electricity load. Based on the DH load and parametric studies of CCPP given in Section 5.1, results were obtained for various allocation methods.

The results represented in the Table 4 show the values of CO_2 allocation factors for heat in the design phase.

Table 4 Allocation factor heat in the design phase

Method	Design value Allocation factor heat
200 %	0.0608
Alternative generation method	0.3830
Energy method	0.2162
PASS 2050	0.1212
Power bonus method	0.2226
Exergy method	0.1507
Dresden method	0.8340

It might be noticed that different allocation methods produce different results in Table 4. For example, the fuel allocation for heat for alternative generation method was 39.8 %, while using the 200 % method this value is 6 % and for power bonus method it is 22 %.

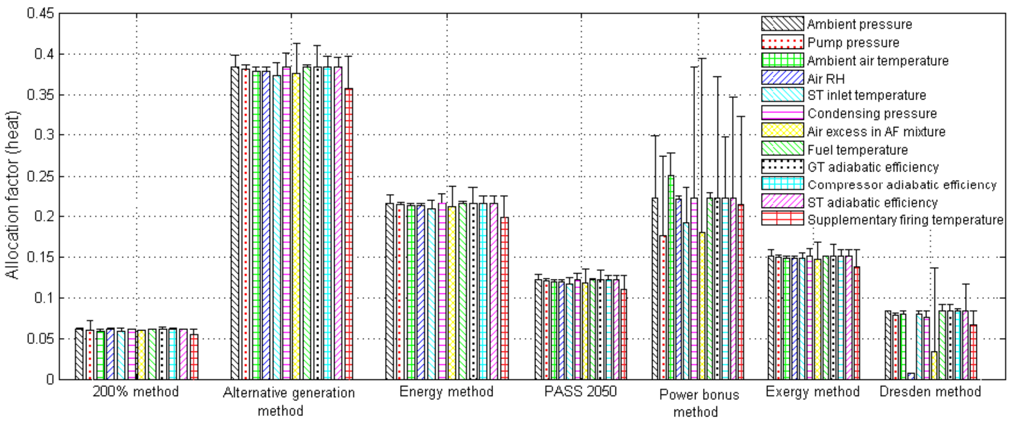

Fig. 6 Heat allocation factors for analyzed methods

Fig. 6 presents the effect on allocation factors depending on analyzed parameters introduced in Table 3 and Section 4.

The change in heat allocation factors for design and operational parameters showed a small variance. This can be noticed by comparing the Table 4 and Fig. 6. The power bonus method was the most sensitive due to change in variation. The alternative generation method offered the biggest share in the heat allocation, while the smallest share was shown by the 200 % method.

Finally, for different allocation methods, Fig. 7 shows the maximum sensitivity in the allocation factors for heat and electricity production. Based on power bonus method, the heat allocation factor changed by 0.16 units due to change in condensing pressure; see Fig. 7. The air excess coefficient in the air-fuel mixture resulted in a change of 0.22 units. The change in the steam turbine adiabatic efficiency and supplementary firing temperature resulted in 0.12 and 0.11 units of heat allocation factor. The changes in the parameters described above have greatest influence on power production in CCPP. As Fig. 6 and Fig. 7 show, the power bonus method was the most sensitive compared to other methods. The 200% method showed the smallest change in the analyzed parameters, resulting in a beneficial share of emissions' allocation for DH between heat and power production.

PASS 2050 and exergy methods also had good results; and showed that the operational and design parameters did not have a significant influence on allocation factors for both heat and electricity.

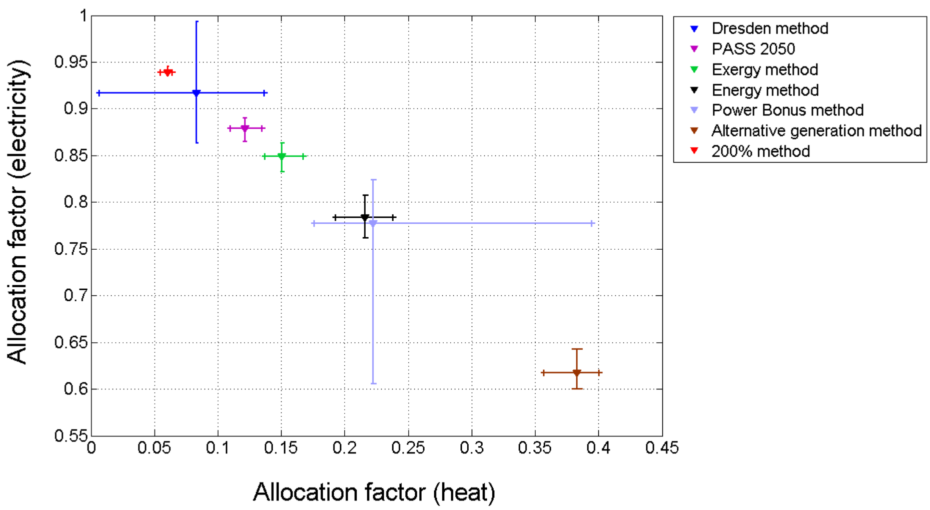

Fig. 7 Sensitivity of allocation factors for heat and electricity

Due to a change in yearly heat and electricity demand we can conclude that emissions allocated with the power bonus method cannot be fixed continuously as it stated in standard

EN 15316. The change in operation parameters gives a variation in the heat allocation in CCPP that should be taken into consideration while applying power bonus method. As an alternative to the power bonus method, other methods with small variation during the year should be considered such as the 200 % method, PASS 2050, or exergy method.

6. CONCLUSIONS

The different methodologies for the allocation of CO_2 emissions for heat and power production in the CCPP have been presented and analyzed. The allocation methods were combined with a parametric study of the CCPP and this showed that different allocation methods produce different results. This indicated that the choice of allocation method is very important for the development of cogeneration technology in relation to heat and power distribution systems. The 200 % method gives the lowest CO_2 allocation for heat, indicating that the heat produced in the CCPP is the most environmentally friendly. On the other hand, the alternative generation method allocates a higher amount of emission to heat, which is not beneficial from a DH point of view. Among all the presented methods, the most sensitive was the power bonus method, which is promoted as the main method for emissions' allocation in the EU. The results showed the highest variance in allocation factors for both electricity and heat, ranging from 11% to 21% compared to the design case. In other methods, the variation was negligible: around 1 - 3 %. All these indicated that the CO_2 allocation was difficult to estimate under the annual heat and electricity load variations. Therefore, we can conclude that emissions allocated with the power bonus method cannot be fixed continuously as is stated in standard EN 15316. The solution can be efficiency values, seasonal averages as a basis for nominal intensities or methods with small variation. This study showed that the decision regarding choosing the allocation method should be carefully analyzed for implementation in the standards and different policies. It is important to enable a proper allocation of CO_2 emissions and the promotion of environmental benefits from cogeneration technology for DH and power distribution systems.

NONENCLATURE:

f_Q	- fraction of cogeneration emissions allocated to thermal product
f_E	- fraction of cogeneration emissions allocated to electrical product
E_{el}	- electricity from cogeneration plant
Q_{del}	- the heat energy delivered to the border of the supplied building
E_{del}	- power energy generated in the cogeneration plant
$Q_{del,j}$	- total heat use at primary side of consumer substation
Q_i	- heat use required during one hour
$E_{del,j}$	- total power use at primary side of consumer
E_i	- power use required during one hour
Ex_E	- net output of electrical exergy from cogeneration
Ex_Q	- net output of thermal exergy from cogeneration
$Fuel_{in}$	- total primary fuel energy consumed in the cogeneration plant
ΔE	- electricity losses in cogeneration plant due to thermal production
η_{alt_heat}	- heat production efficiency of producing thermal energy via alternative heat - generation plant

η_{alt_elec} - power production efficiency of producing power energy via alternative power generation plant

$f_{P,dh}$ - primary energy factor of the DH system

$\Delta\tau_i$ - duration of the heat or electricity load

n - intensity of GHG emissions of production unit

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Importance of Increased Knowledge on Reliability of District Heating Pipes

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Abstract

District heating (DH) is a service that satisfies customers' demands in the areas of heating, hot water preparation and the supply of heat to ventilation systems. Three generations of DH distribution technology are already in operation; the next generation of low temperature district heating (LTDH) will soon be upon us. However, without a reliable distribution system, it is quite difficult to utilize the concept of LTDH and remain competitive in the energy market. For that reason, this paper provides a comprehensive review of pipe reliability issues associated with DH systems. In this regard, discussions have been concentrated on factors leading to pipe degradation processes. Three groups of factors, namely physical, environmental and operational, were identified and examined. Allowable heat losses in the DH network and the creation of a pipe failure database were also discussed. The information collected in this paper leads to a better understanding of pipe degradation mechanisms and can be used as a tool for pipe failure prevention.

Peer-review under responsibility of the organizing committee of CCHVAC 2015.

Keywords: District heating; distribution system; pipe reliability; pipe accidents.

1. Introduction

District heating (DH) is an energy service based on moving heat from available heat sources directly to customers for immediate use [1]. This service is flexible and allows renewable energy sources to be utilized as a primary energy input. In turn, this leads to decreasing CO_2 emissions and energy savings. Currently three generations of DH distribution technology are in use. The research society is moving towards the fourth generation of low temperature

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district heating (LTDH) [1]. The development of LTDH is impossible without a reliable distribution system. It is well known that a DH system is rarely developed from a scratch and huge DH networks are a result of extension and merging. Therefore, it is highly desirable that old DH pipes provide reliable operation and do not influence heat distribution through unpredicted failures.

In order to stay competitive in the energy market, DH should provide a reliable heat supply to customers throughout the year. In reality, this is not an easy task. Different malfunctions and accidents associated with the operation of a DH system and the distribution of heat lead to a decrease in the security of supply. The possibility of losing the heat supply is particularly dangerous during the winter season in countries with extremely low outdoor temperatures.

Accidents in the DH networks are inevitable and can occur for various reasons: wear and tear, equipment failures, pipeline breaks and so on [2]. Accidents lead to financial and capital losses, incurred by the repair and restoration of the network. Failures reduce the reliability of the network due to lowering of the pressure or due to interruption of the DH supply, which ultimately leads to customers' dissatisfaction. Sensitive customers, such as industrial centres, governmental buildings and hospitals, are most likely to be affected [3]. One serious problem in DH supply is deterioration of the distribution network; this can occur for different reasons. Pipe deterioration can lead to pipe breaks and leaks, which may result in a reduction in the water-carrying capacity of pipes and lead to substantial repair costs [4]. Pipe breaks incur large direct and indirect economic and social costs, such as water and energy loss, repair costs, traffic delays, and factory production loss due to inadequate DH service interruptions. Unfortunately, it is difficult to locate breaks in the pipe network because most parts of the pipes are buried underground and inaccessible [5]. Component failures in flow networks lead to disappearance of flow capacity, and the expected level of the throughput flow may not be guaranteed. As a result, the quality of service received from the network can be seriously affected [6].

With their further development, it is important to provide high reliability and availability of DH systems for existing and future customers. Piping failures can be prevented through reliability measures and these are subject to improvement.

Nomenclature

HL	heat losses
PL	length of DH pipelines
Q_h	heat production in the DH system
$Q_{h,f}$	heat production affected by pipe failures
Q_{loss}	heat losses in the DH system
ΔQ	decrease in heat delivery due to pipe failures
$\Delta Q_{h,f}$	relative deviation in the heat delivery due to pipe failures
a	model coefficient
b	model coefficient
f	pipe failure factor

2. Factors affecting pipe reliability

In their work, various researchers have tried to identify the main causes leading to pipe deterioration [7-9]. Al-Barqawi and Zayed [10] classified three groups of factors resulting in pipe degradation; these are presented in Table 1.

Table 1. Factors leading to water system deterioration.

Physical factors	Environmental factors	Operational factors
Pipe age and material	Pipe bedding	Internal water pressure
Pipe wall thickness	Trench backfill	Leakage
Pipe vintage	Soil type	Water quality
Pipe diameter	Groundwater	Flow velocity
Type of joints	Climate	Backflow potential
Thrust restraint	Pipe location	Operational and maintenance practices
Pipe lining and coating	Disturbances	
Dissimilar metals	Stray electrical currents	
Pipe installation	Seismic activity	
Pipe manufacture		

Many of the factors listed in Table 1 are not readily measurable or quantifiable. Physical mechanisms that lead to pipe breakage are often very complex. Moreover, the quantitative relationships between these factors and pipe failure are often not completely understood [11, 12]. The most commonly assumed factors for a DH distribution network are described below.

2.1. Age and installation period

DH technology was first put forward in the middle of the nineteenth century in the US and in the early twentieth century in Europe. The differing launching periods for DH systems in Europe indicate that the pipes used in the systems can have been installed in different periods. In France, for example, first generation DH systems with steam as a carrier are still in use. It has been found that the construction period can affect pipe durability [13]. Different types of pipes are used and sometimes older pipes have less tendency to fail than their younger counterparts. Further, only several pipes can remain under the ground after four repairs, since pipe records are unable to provide accurate ages of pipes [14]. When the age of piping exceeds 30 years, the frequency of damage increases, and the technical conditions of the pipeline become the main reason for the formation of defects [15].

2.2. Corrosion

Corrosion is the main reason for pipe replacements [16] and structural deterioration [17]. As indicated in [15], the failures of pipelines due to corrosion mechanisms constitute 30 – 40% of all damage to these pipelines. Metal pipe corrosion pitting is a continuous and variable process. Under certain environmental conditions, metal pipes can become corroded based on the properties of the pipe, soil, liquid properties and stray electric currents [18]. Corrosion deterioration mechanisms can be divided into two types: internal and external. Internal corrosion is caused by different characteristics of transported water. Poor water quality can cause internal corrosion of the pipelines and substations and may block and weaken the functioning of the controlling and metering devices in the entire DH system [19]. Different level of water pH-value, oxygen content or bacteria can be the reasons for this process. Meanwhile, external corrosion occurs with pipes sensitive to soil composition, moisture and aeration. These can be described as aggressive environmental conditions. Corrosion occurs at apparently random locations on a pipe [20], weakening it by decreasing the material’s thickness and by creating stress concentrations [21]. However, not all pipes used in DH are exposed to corrosion. For example, flexible pipes [1] made of polymer material such as PolyEthylene (PE) are corrosion resistant. Nevertheless, they are sensitive to the high temperatures used in DH systems. Therefore, manufacturers normally limit the maximum supply temperatures in order to extend the pipe’s service life.

If the service pipe fails, the escaping hot water may cause extensive damage to the surrounding area. Repairs will then include replacing part of the service pipe, resulting in an interruption to service, which will leave end users without a hot water supply. Aside from the inconvenience caused to end users, the repair costs will be high. This is especially so in the case of larger diameter pipes [22].

2.3. Diameter

The failures associated with pipe diameters can be explained by the thickness of pipes. Small pipes have lower wall thickness, resulting in reduced pipe strength. The high probability of pipe breakage was found in network of small pipes [23]. In the study related to damage caused by earthquakes [21], pipe diameter was identified as an influence on the number of breaks and failures in pipelines. Pipes with small diameters experienced more damage than those with large diameters. The influence of pipe diameter determines the total area from which a failure point may occur. The pipe may be thought of as being developed from a simple plate of area πDL [24]. Smaller pipe sizes, of lesser safety significance, have much higher failure rates [25].

2.4. Pipe length

The dependency of failure probability on pipe length was also acknowledged [24]. The failure probability increases directly in proportion to length; for example, a 1.0 m length of pipe bears a 10-times greater failure probability than a 0.1 m pipe. This assumption was based on uniform distribution of weak spots along the pipe. The number of weak spots such as bends, junctions, welds, flaws increases proportionally with the increase in pipe length.

2.5. Pipe material

It is no surprise that pipe service life is dependent on the pipe material used for hot water distribution. Study of the historical development of DH systems in the world indicates that the use of different types of distribution pipes was due to different materials being available during certain time periods. DH distribution technology can be classified in three generations [1]; the first generation of DH technology used steam as a carrier; while the second and third generations employed water as a carrier to deliver heat. Nowadays, the temperature level in DH networks has decreased; however, systems based on first generation principles are still in operation. Therefore, it is possible to distinguish different pipe types used in DH systems. The traditional metal for pipes is steel. Pre-insulated rigid steel pipes have the largest share in DH systems and a number of publications are devoted to this type of pipe [26, 27]. New developments in the DH field have introduced pre-insulated rigid polymer pipes and pre-insulated flexible PE pipes with a life span of more than 30 years [28]. At the same time, copper pipes are also in use in customer substations [29].

2.6. Dissimilar metals

In general, dissimilar metals could be employed when DH systems of different ages are connected. Welded dissimilar metal joints can have a remarkable effect on the plant's availability and safety and lead to leakages and pipe cracking [30]. Dissimilar metal joints can be installed in pipes with large diameters. The study devoted to the safety of nuclear plants found that the probability of cracked welding occurring is rather high [31].

2.7. Seasonal variation

Accidents mainly occur during the winter in DH systems. The main reason is that the largest heat demand associated with DH occurs at this time of the year, while extreme outdoor temperatures weaken the pipes, particularly if they are exposed to the cold without adequate insulation.

2.8. Soil conditions

Soil conditions affect eternal corrosion rates and play an important role in pipe degradation [32]. The rate of corrosion is affected by the properties of the soil, in particular its pH content, redox potential, existence of sulphides, water resistance table [33], and by the soil type: clay, sand or peat soil [34]. This is relevant for the DH pipes of the third generation of distribution technology [1], which are buried in the ground.

2.9. Previous failures

The number of previous failures is a significant factor in predicting future failures [35]. Pipes in the same location often have the same age and materials and are laid with the same construction and joining methods. Pipes in the same location are also likely to be exposed to the same external and internal corrosion conditions [32].

2.10. Nearby excavation

Nearby excavation, together with seismic activities, affects pipe integrity. A detailed description of these processes was provided in [36, 37]. Contact during excavation usually occurs when an individual piece of operational excavating equipment breaks the pipe. It was found that the risk due to corrosion is significantly less than the risk due to third-party intervention [7, 34]; research work carried out in the UK showed that third party activities have a high probability of causing pipe failure [38].

2.11. Pressure

The pressure change due to the breakage depends on the ratio of flow rate through the pipe versus flow rate through the break. If the losses of water are very small, compared with the mainstream through the pipe, then pressure fluctuations due to the break would be negligible [39]. The possibilities of pipe ruptures due to high pressure in the water distribution networks, together with the hammer effect, were acknowledged in [40]. The analysis found a high risk of pipe damage that leads to water supply interruptions. The analysis on the DH network found that, under some conditions, pressure peak could exceed the value used in hydraulic tests, but the possibility of pipe damage still remained high [41]. The pressure head in the distribution network has a direct influence on the frequency of pipe failure [42]. Further, water hammering due to an immediate change in velocity of the carrier in DH systems directly affects joints [1, 7]. This situation can occur when distribution pumps are tend to be open rapidly against closing of valves located on pump outages or when predefined valve opening time is ignored [1]. The researchers in [43] identified the possibility of pipe ruptures depending on operational pressure in a DH network. According to results, the rupture probability increases linearly up to a pipe thickness of 3 mm. The plastic deformation will occur from thicknesses of 3 mm to 1 mm. In the case of thickness being less than 1 mm, the rupture probability increases rapidly.

2.12. Land use

Traffic areas, residential areas and commercial areas are used as a substitute for external loads on pipe [32]. The stresses occurring in DH pipes are complex and originate from a number of sources, including soil loading, ground surface loading (due to traffic) and as a result of temperature changes. The pipe failure occurs either when the stress level exceeds the nominal pipe material strength or when a critical defect develops and leads to degraded pipe strength [20].

2.13. Temperature levels

Improper temperature levels used in DH systems can cause mechanical stresses and thermal strain in distribution pipes. When a carrier pipe is made of steel, which is always the case with rigid pipes, normal operation temperatures are too low to cause any significant creep deformation. However, when carrier pipes are made of polymeric material, as they sometimes are in flexible pipes, creep and thermal expansion are the major issues [1]. Thus, when designing and installing a buried pipeline, which is both pressure- and temperature loaded, one should pay special attention to the extreme temperature variations and hence to the stresses and movements that the pipeline will have to withstand [44]. The joints are highly affected by mechanical stresses due to the large temperature differences whenever the distribution network is in operation or shut off [45]. Temperature fatigue, occurring due to high temperature levels in the DH system, results in high failure probabilities [46, 47]. Moreover, different pipe manufacturers limit the maximum supply temperature used in DH systems to 120 °C.

2.14. Welding

According to the study [48] performed by EuroHeat & Power, the largest number of failures in polymeric DH pipes occur in joints. One of the reasons is bad welding procedure. The annual frequency of failure due to the joints of the outer tube is the highest, compared to damage caused by medium tube joints and damages found by the leakage detection system [45].

3. Discussion

As can be seen, there are many different factors affecting DH pipe reliability. In order to remain competitive in the market and provide a reliable service to customers, the reliability issues should be carefully analyzed. For this reason, it would be wise to collect relevant information about accidents associated with DH distribution system. A comprehensive database must include information that allows pipe failure accidents to be predicted by analytical and statistical methods. Therefore, a good database should include three categories of information. firstly, it should contain information about the pipes at the time of their installation: installation year, type of pipes and materials, diameters, lengths of pipe sections and number of joints. Secondly, data should be included on the operational regimes of the DH system, e.g. temperature and pressure levels, pH-value of heat carrier, and number of water replacements during the year. The last part should include type of failure, date and place, and maintenance measures.

According to Statistics of Norway [49], the heat losses in DH pipes corresponds to 14% on the national level. This can be seen from Fig. 1 and Fig. 2.

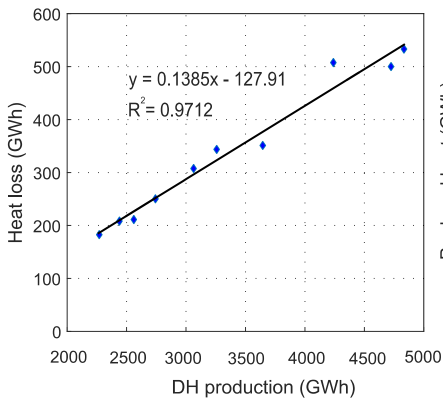

Fig. 1. Dependence of heat production versus heat losses.

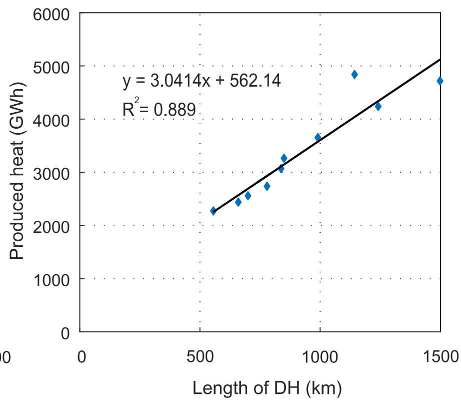

Fig. 2. Dependence of DH pipe length versus heat production.

The presented statistical data may be used to assess the allowed percentage of failures in the DH distribution system. In order to answer this question, let us examine Fig. 1 and Fig. 2. From Fig. 1 it can be seen that the heat losses during the delivery of DH services can be expressed as:

$$HL = 0.14 \cdot Q_h - 127 \tag{1}$$

where HL is heat loss and Q_h is DH heat production. At the same time, the production of heat depending on the length of DH pipelines can be found from Fig. 2 and expressed as:

$$Q_h = 3.0414 \cdot PL + 562.14 \tag{2}$$

This equation shows how much heat can be delivered to the customers, taking into consideration the length of the distribution network. Hence, a and b can be introduced as general model coefficients. Finally, it would be of interest to introduce a pipe failure factor, which has a direct influence on heat delivery.

$$Q_{h,f} = a \cdot PL \cdot (1 - f) + b \tag{3}$$

where $Q_{h,f}$ is heat production affected by pipe failures and f is a pipe failure factor. The decrease in heat delivery due to pipe failures can be estimated as:

$$\Delta Q_h = Q_h - Q_{h,f} \tag{4}$$

The relative deviation in the heat delivery due to pipe failures, can be found as:

$$\Delta Q_{h,f} = \frac{\Delta Q_h \cdot 100\%}{Q_{h,f}} \tag{5}$$

Fig. 3 shows the dependence of the pipe failure factor and the heat percentage of undelivered heat where the length of the distribution system is a parameter.

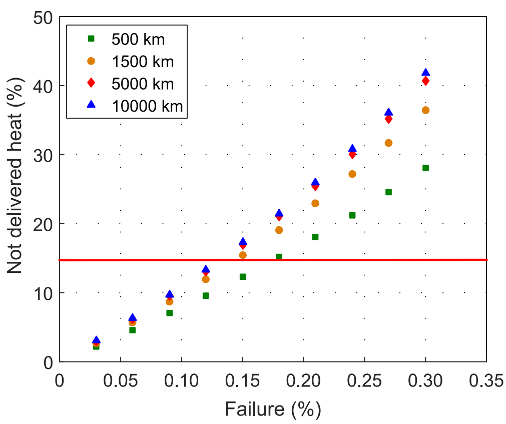

Fig. 3. Percentage of pipe failures versus relative deviation in heat delivery.

Finally, the percentage of pipe failures allowed in the DH system should be lower than the heat losses in the DH system, because if the amount of undelivered heat is too high, the transmission cost can decrease the competitiveness of the DH system:

$$\Delta Q_{h,f} \leq Q_{loss} \tag{6}$$

As it can be seen from Fig. 3, with the increase of the pipe failure factor, the undelivered heat increases. This is especially important for large DH systems, as is shown for 5000 km and 10000 km of DH pipe length. Based on this observation, it can be concluded that the percentage of failure should be less than 10% in order to maintain the decrease in heat delivery due to failures at a lower rate than the heat losses and thus prevent a decrease in the competitiveness of the DH system. That allows reliable heat delivery and security of supply to be provided.

4. Conclusions

A review of factors affecting DH pipe reliability has been carried out. The information collected in this paper leads to better understanding of pipe degradation mechanisms and can be used as a tool for pipe failure analysis. In addition, for proper operation of DH systems, it is desirable to collect the maintenance information about pipe accidents. A good database can provide an immediate start for the analysis of distribution system, help in pipe model creation based on statistical data, lead to accident prevention and increase the security of supply.

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