Corresponding author:
Tymofii Tereshchenko
tymofii.tereshchenko@ntnu.no
Phone: +47 735 98381

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

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.

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
$\dot{E}_\iota$ (kW) - power rate
ΔE (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
$Fuel_{in}$ (kWh) - total primary fuel energy consumed in the cogeneration plant
n $(-)$ - intensity of GHG emissions of production unit
Q_{net} (kWh) - thermal energy output from cogeneration plant
Q_{del} (kWh) - the heat energy delivered to the border of the supplied building
$\dot{Q}_\iota$ (kW) - heat effect
T (K) - temperature of the medium
T_0 (K) - mean ambient temperature of 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
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
τ_i (h) - operation time of the power plant
$\Delta\tau_i$ (h) - duration of the heat or electricity load
η_c $(-)$ - Carnot efficiency
υ_p $(-)$ - degree of process quality

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. 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:

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

$$f_E = E/(Q + E) \quad (2)$$

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:

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

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

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 EN15316-4-5:2007, 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].

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

The thermal energy balance is given by:

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

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

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

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:

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

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

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:

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

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:

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

Consequently, the heat exergy can be defined as:

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

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

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

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

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:

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

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

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:

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

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

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:

$$\Delta \boldsymbol{E} = \boldsymbol{Q} \cdot \boldsymbol{\eta_c} \cdot \boldsymbol{v_p} \quad (20)$$

where

$$\boldsymbol{\eta_c} = \mathbf{1} - \boldsymbol{T_{cond}}/\boldsymbol{T_{out}} \quad (21)$$

and the maximum electricity production without heat extraction is:

$$\boldsymbol{E} = \boldsymbol{E_{del}} + \Delta \boldsymbol{E} \quad (22)$$

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:

$$\boldsymbol{f_Q = \Delta E / E} \quad (23)$$

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

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.

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	$f_Q = \frac{\Delta E}{E}$	$f_E = \frac{E - \Delta E}{E}$

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:

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

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

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

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

where $E_{del,j}$ is the total electricity use at the primary side of a building, $\dot{E}_\iota$ 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 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.

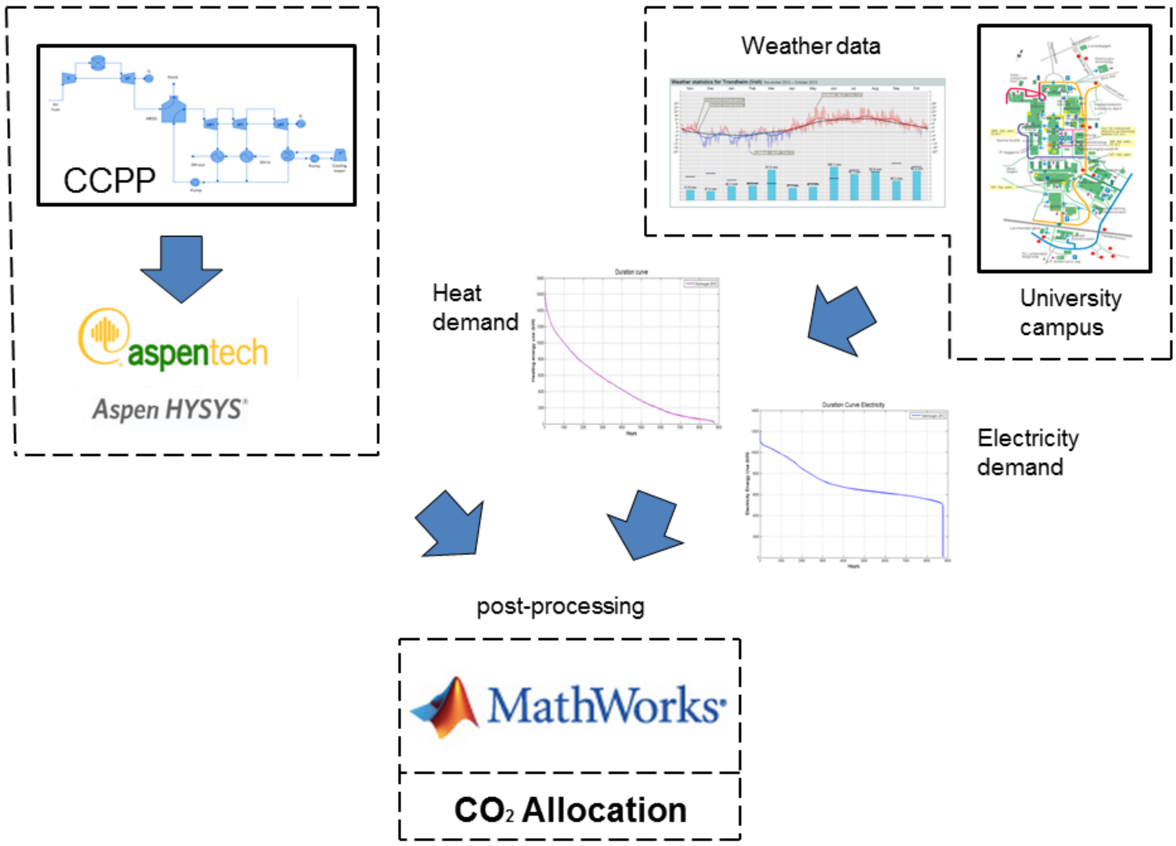

Fig. 1 The flowchart represents steps of analysis done in this paper

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.

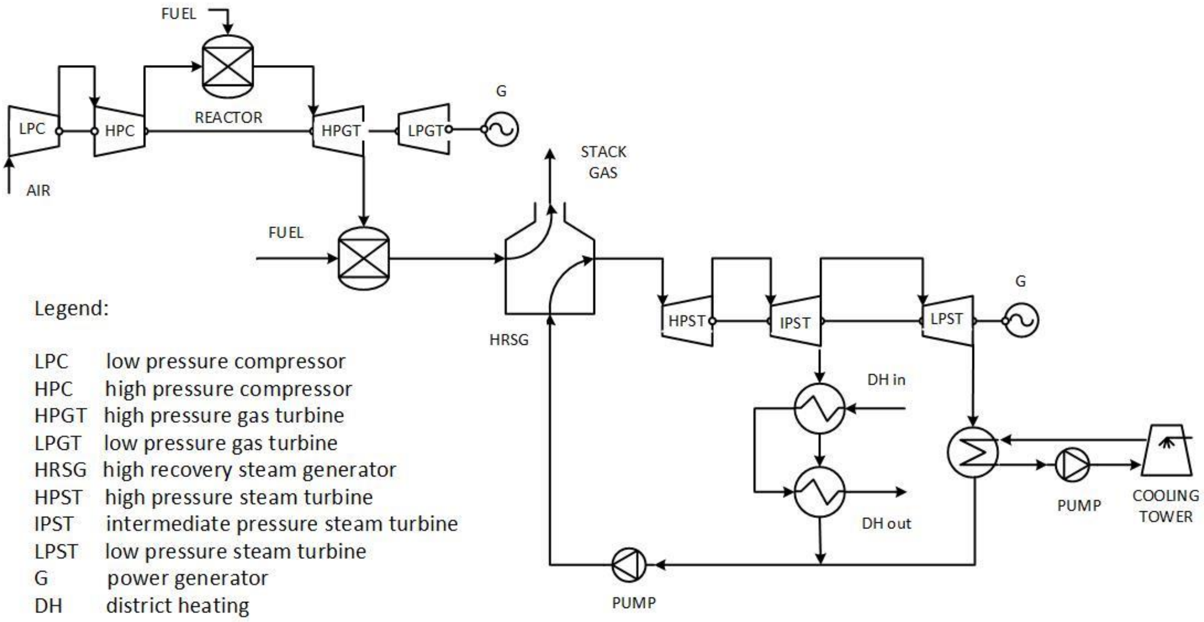

Fig. 2 Schematic of CCPP

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 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, 5MW.

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.

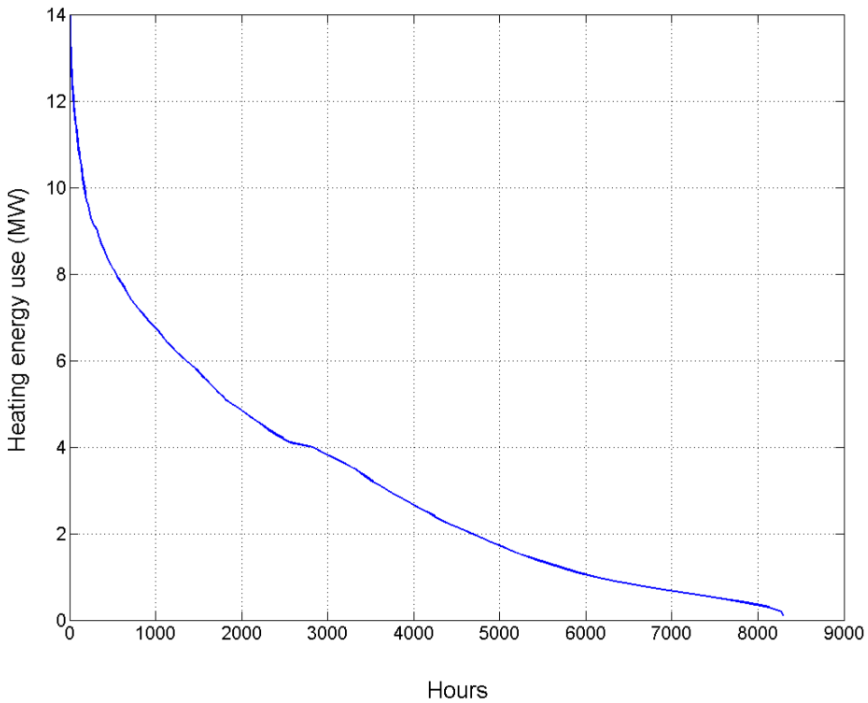

Fig. 3 The heat duration curve of the analyzed campus

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.

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	71310 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	218703 kg/h

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.

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

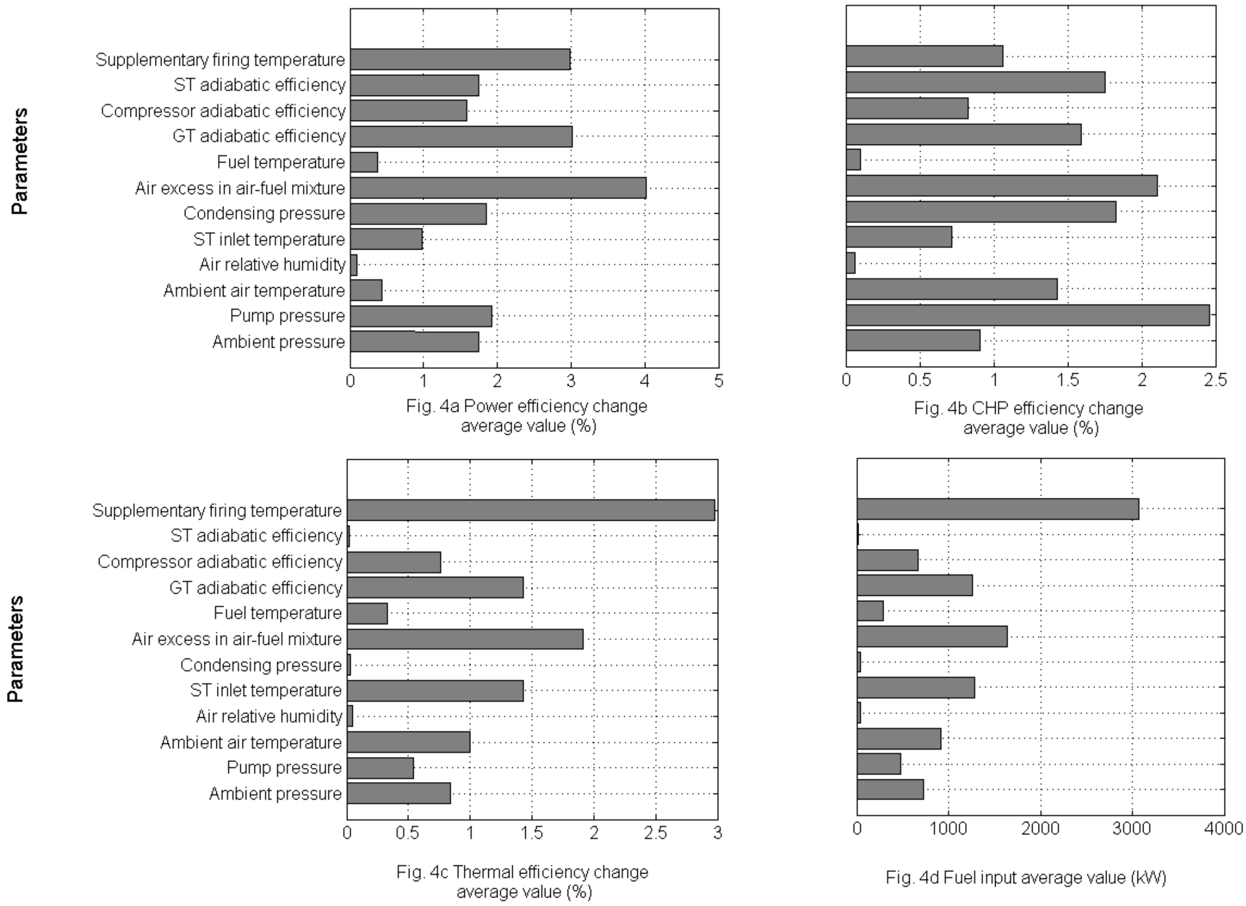

Fig. 4 (that includes 4a-4b-4c-4d).

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

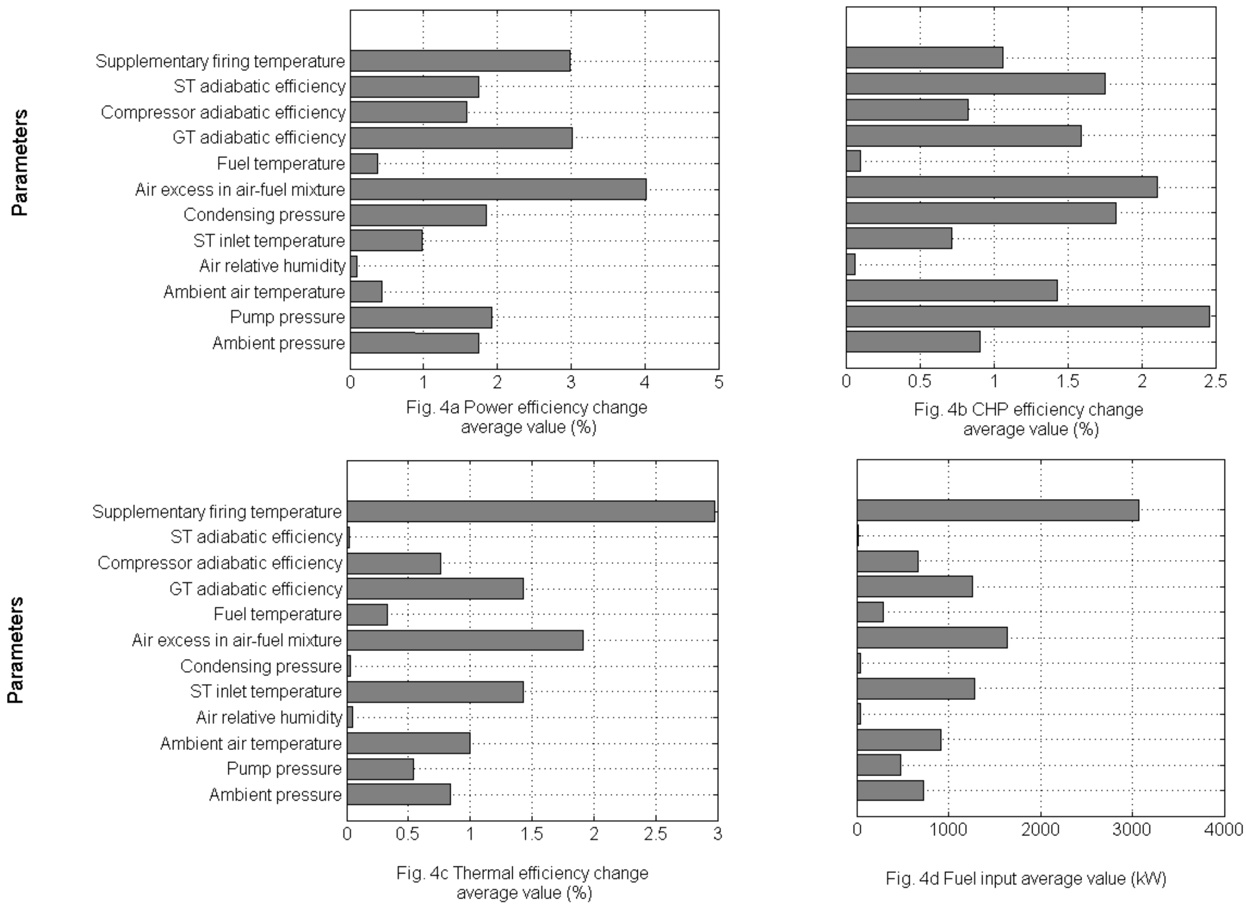

Fig. 4a. The maximum value for CHP efficiency change was 2.46 %; see

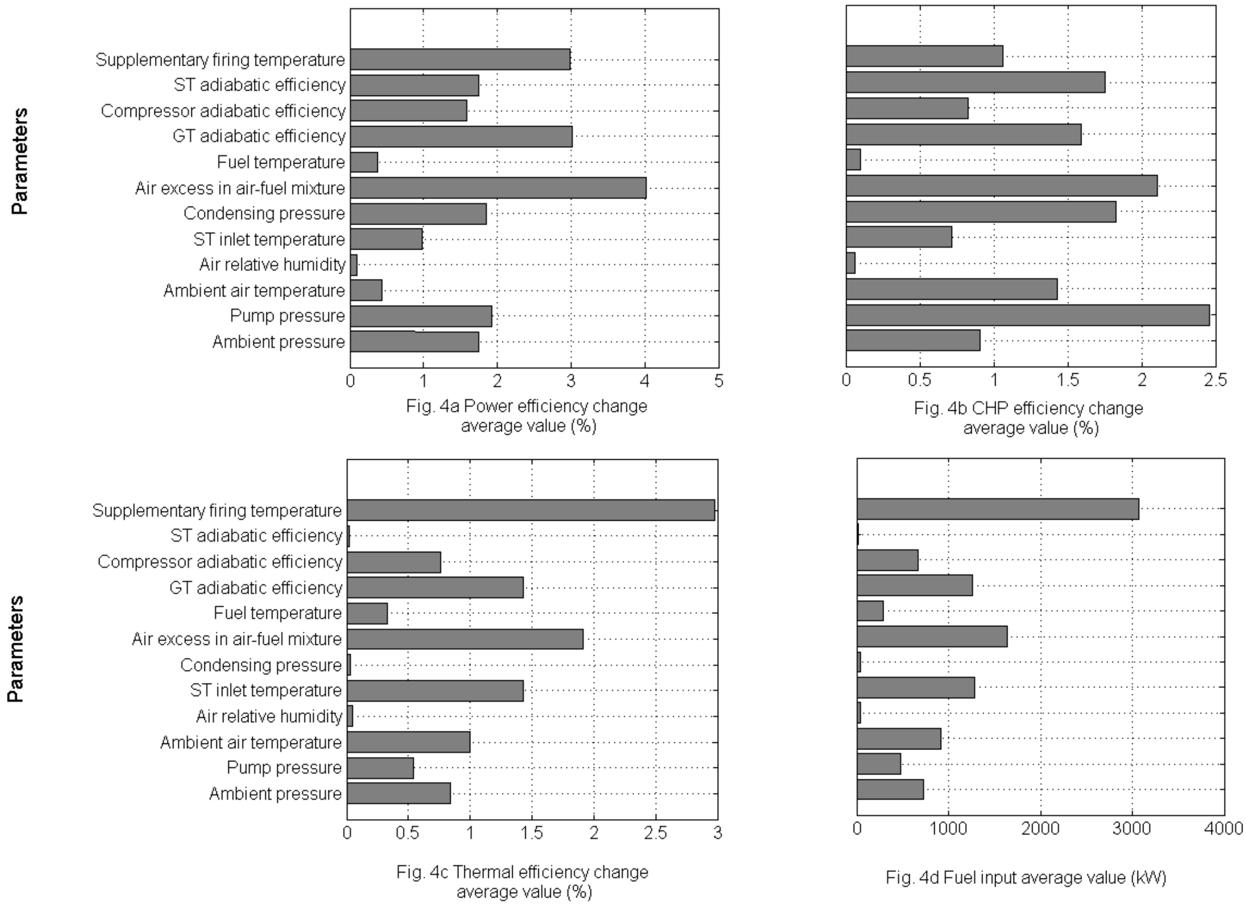

Fig. 4a Power efficiency change average value (%)

Fig. 4b CHP efficiency change average value (%)

Fig. 4c Thermal efficiency change average value (%)

Fig. 4d Fuel input average value (kW)

Fig. 4b. In terms of thermal efficiency, the maximum change was 2.98 %; see

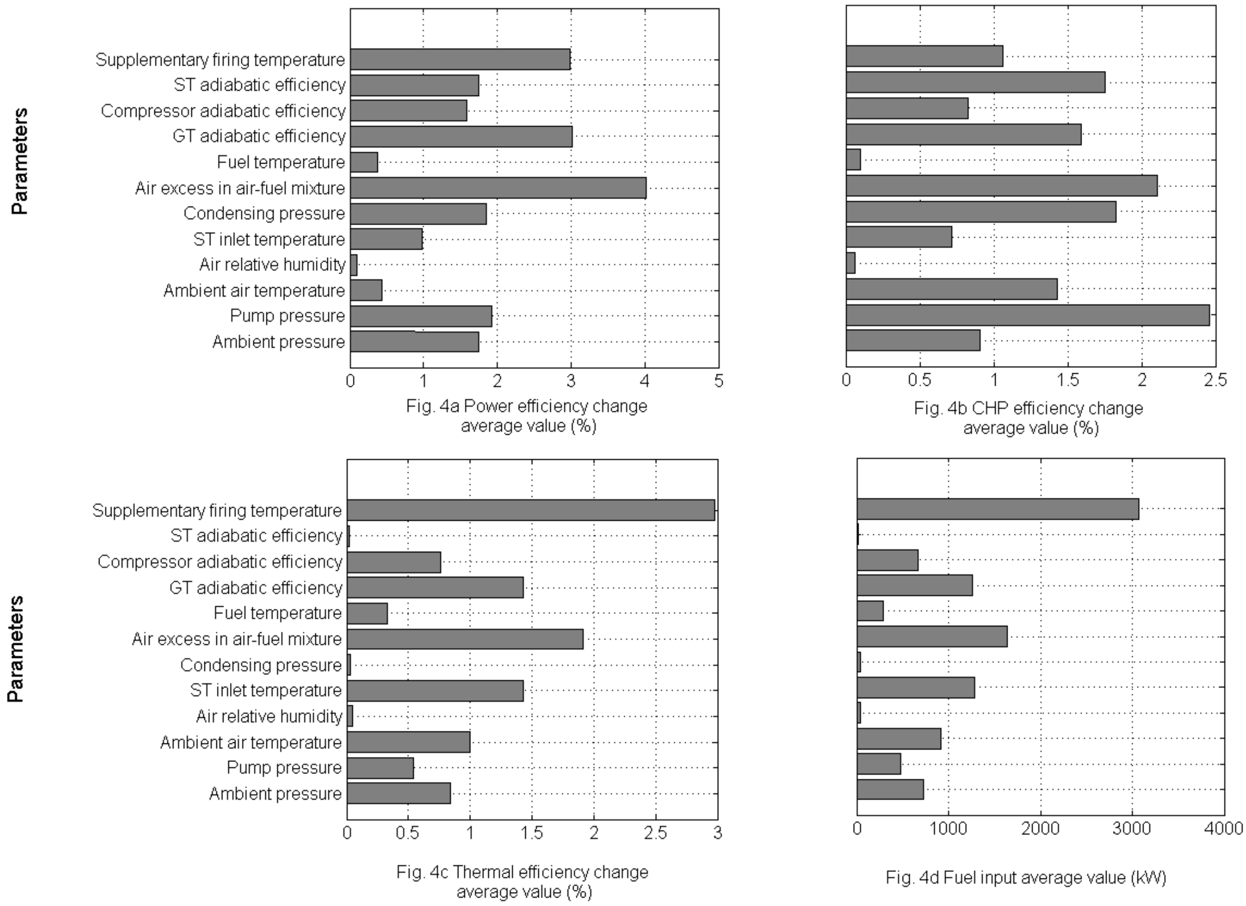

Fig. 4c. The maximum change in the fuel input rate was 3075 kW, as shown in

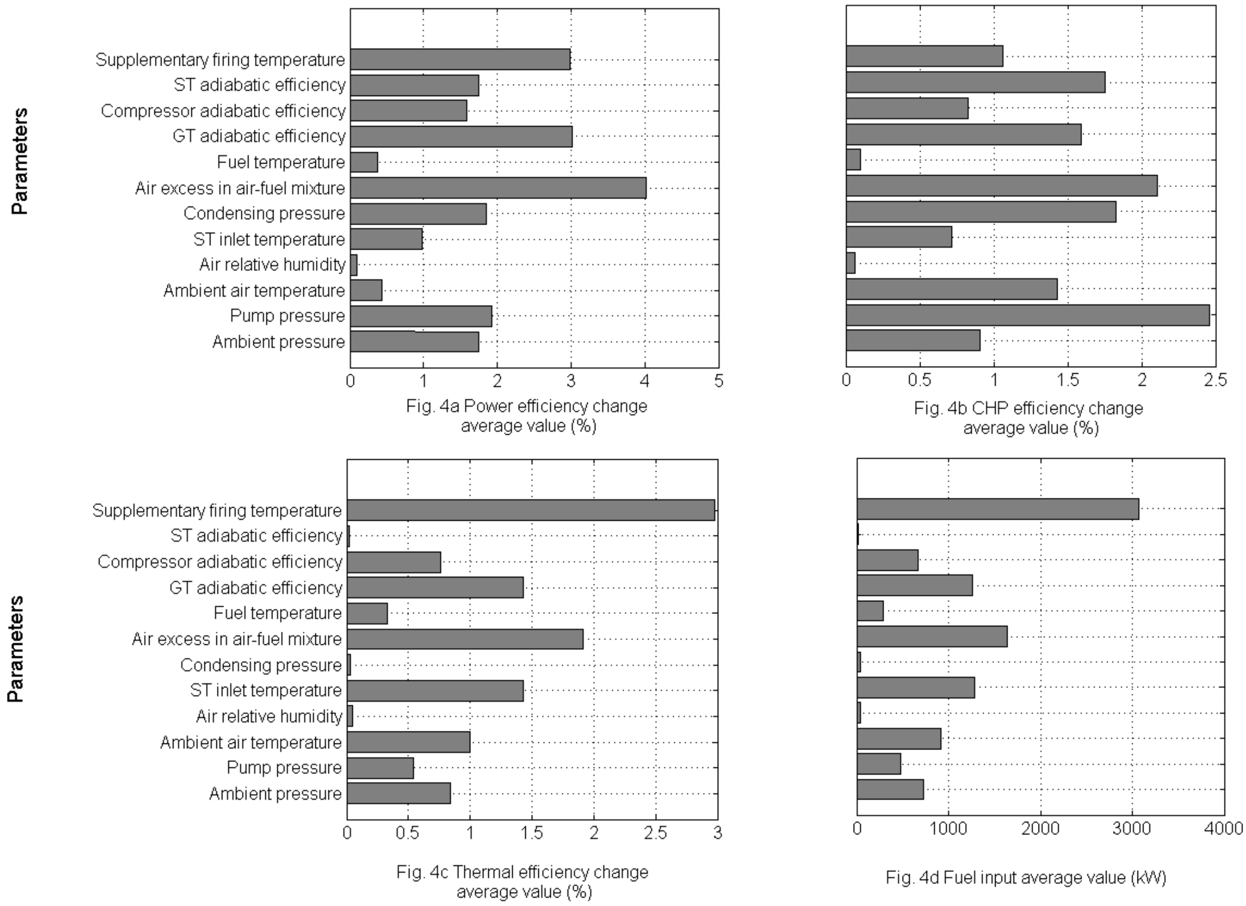

Fig. 4a Power efficiency change average value (%)

Fig. 4b CHP efficiency change average value (%)

Fig. 4c Thermal efficiency change average value (%)

Fig. 4d Fuel input average value (kW)

Fig. 4d, due to a change in the supplementary firing temperature.

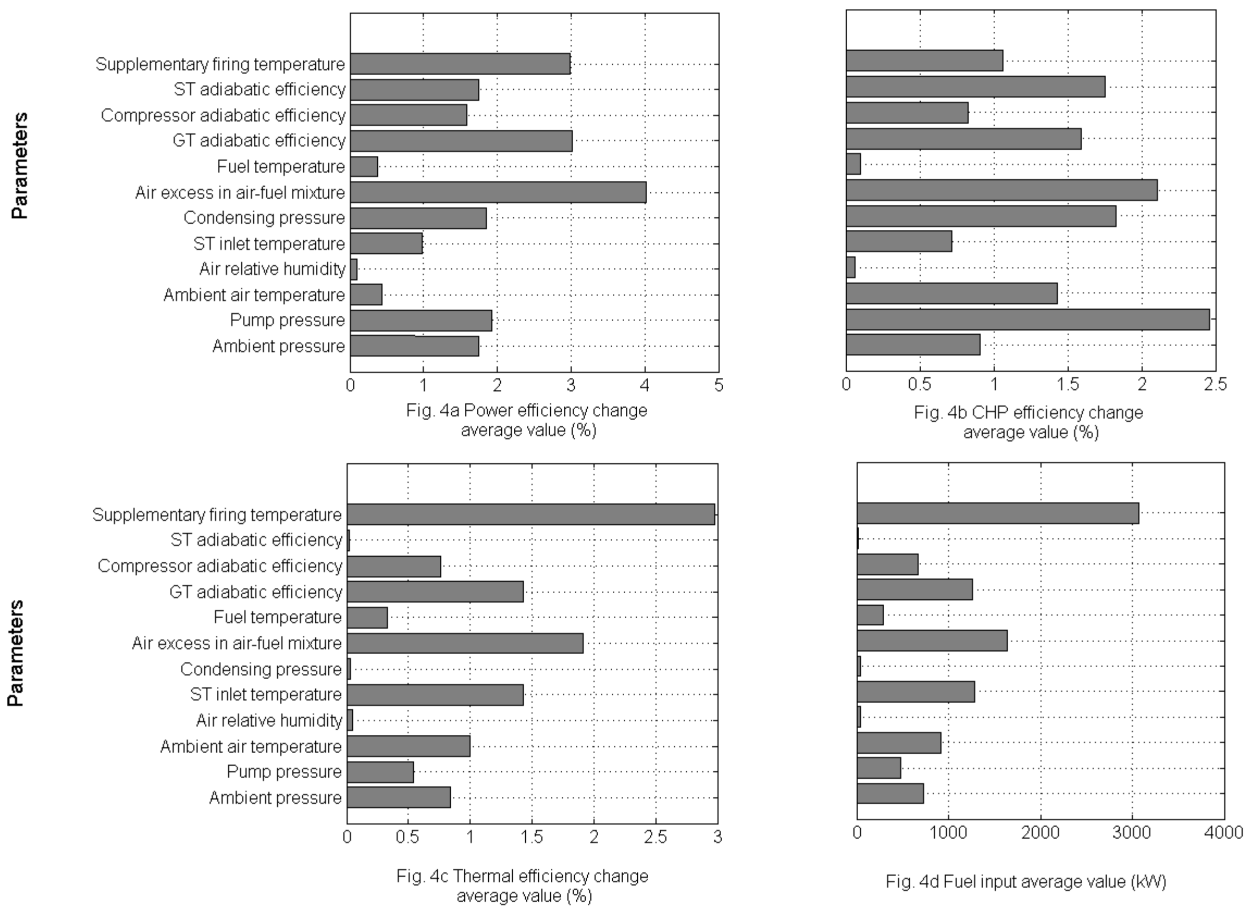

Fig. 4 Change in CCPP behavior based on analyzed parameter

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

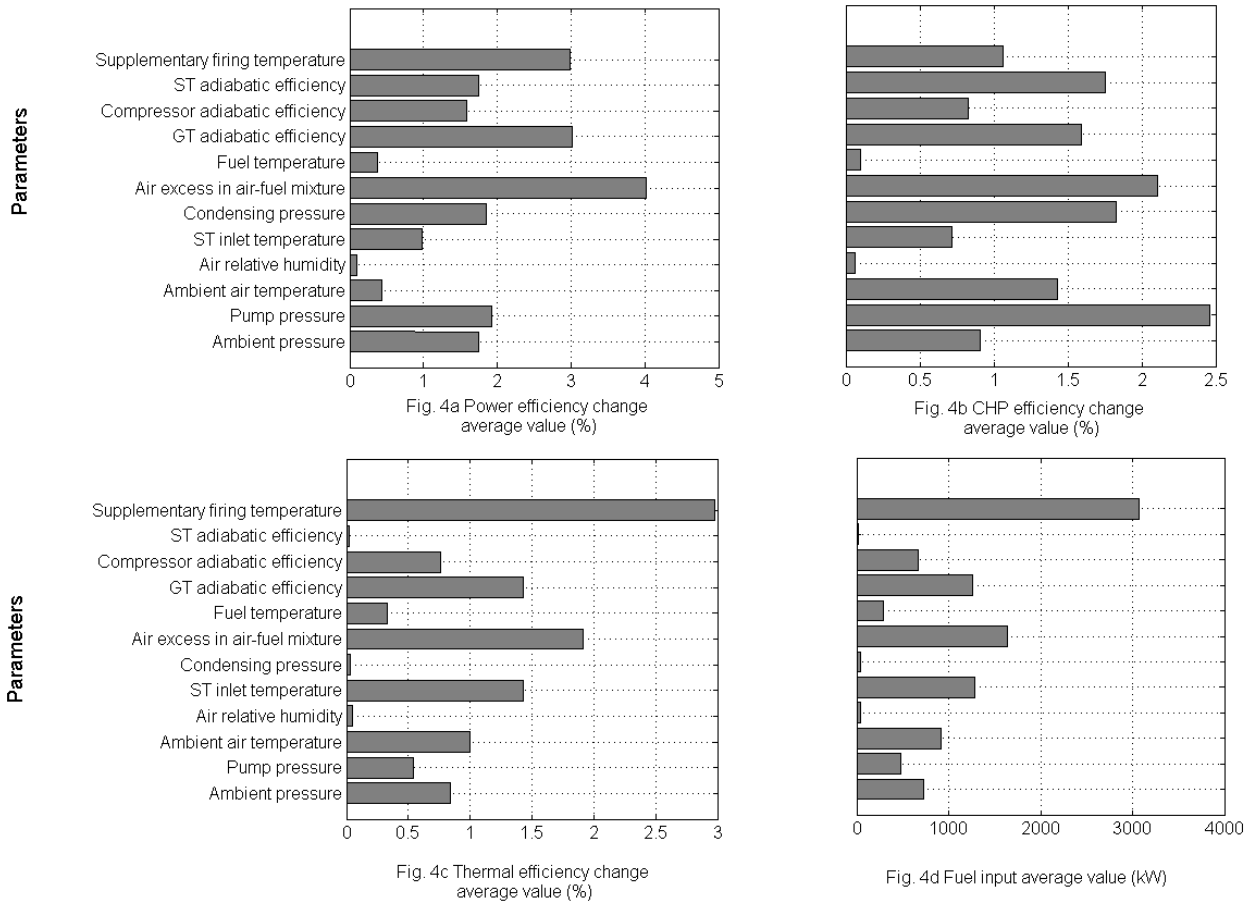

Fig. 4a. The air excess coefficient provided an increase in the mass flow rate of the flue gases through the GTC; this led to an increase in power production in the CCPP by 4.2 % in comparison with the design case. At the same time, the fuel input to the system had decreased. The reduction in fuel input 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

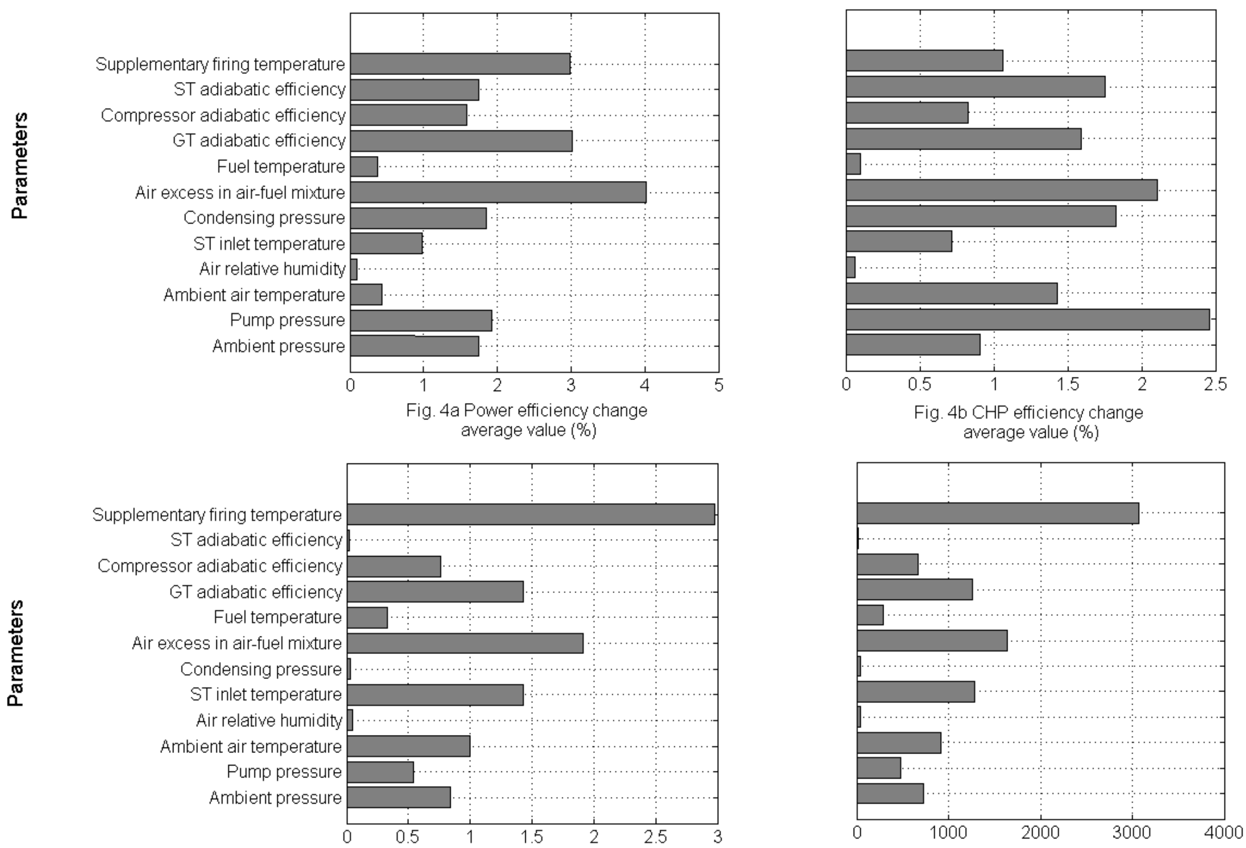

Fig. 4a Power efficiency change average value (%)

Fig. 4b CHP efficiency change average value (%)

Fig. 4c Thermal efficiency change average value (%)

Fig. 4d Fuel input average value (kW)

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 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.

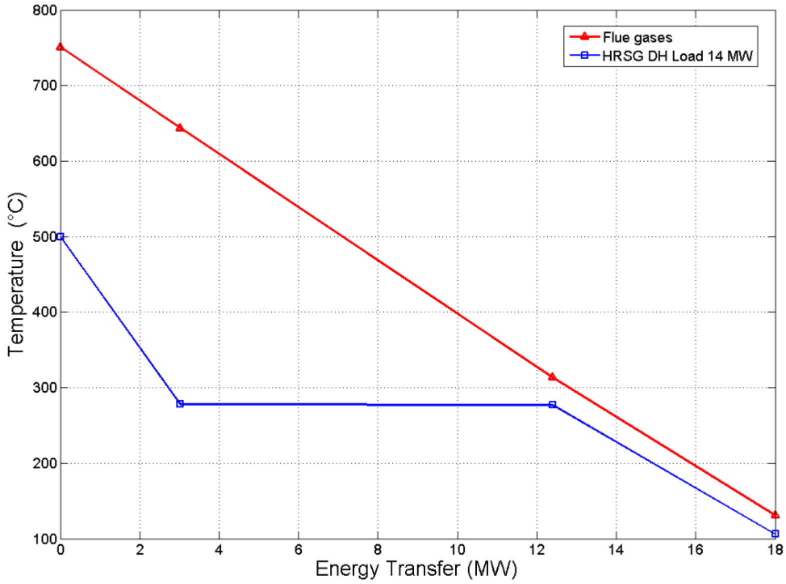

Fig. 5 Energy utilization in the HRSG where the temperature of flue gases is +750°C

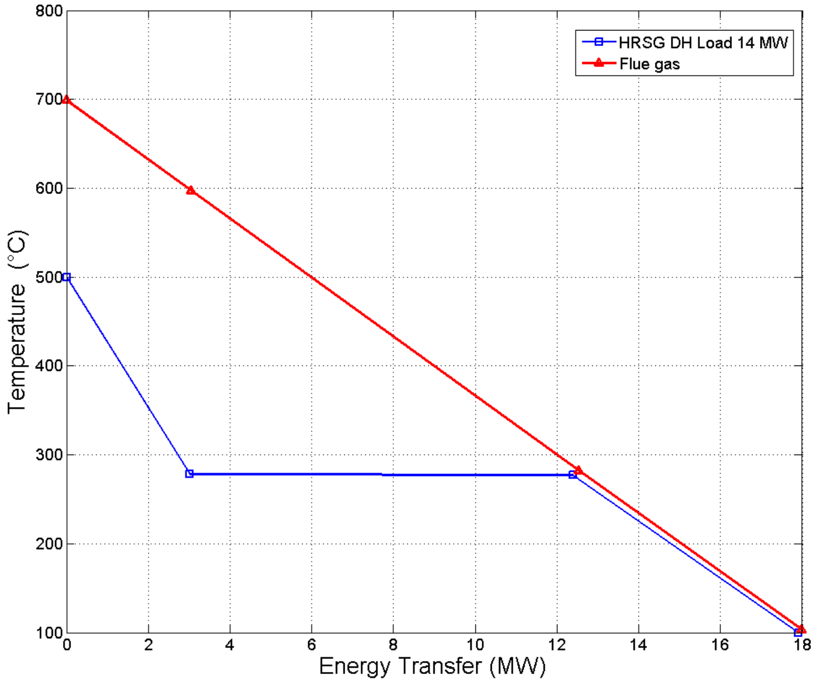

Fig. 6 Energy utilization in the HRSG where the temperature of flue gases is +700°C

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 Fig. 5 and Fig. 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

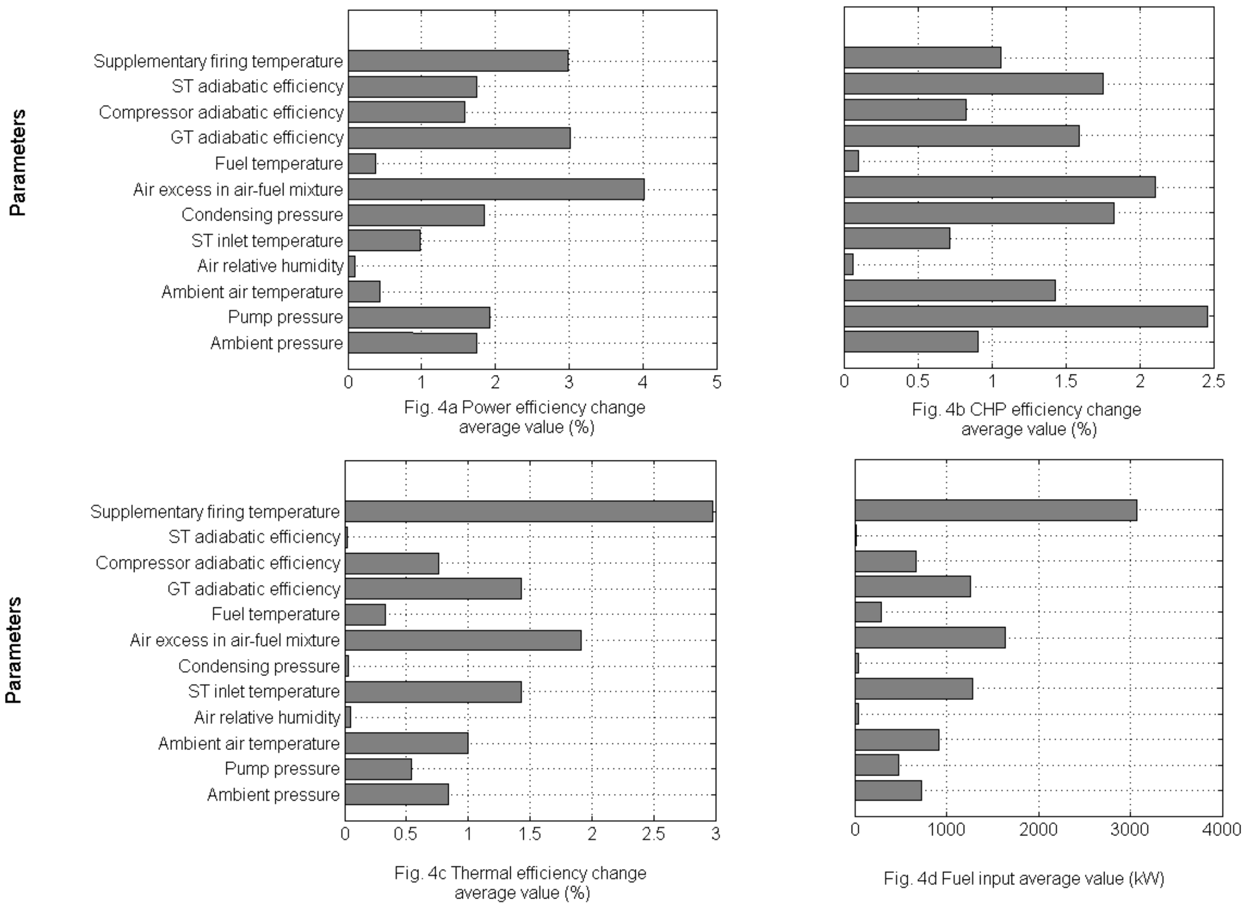

Fig. 4d.

The supplementary firing temperature also affected the power efficiency of the CCPP; see

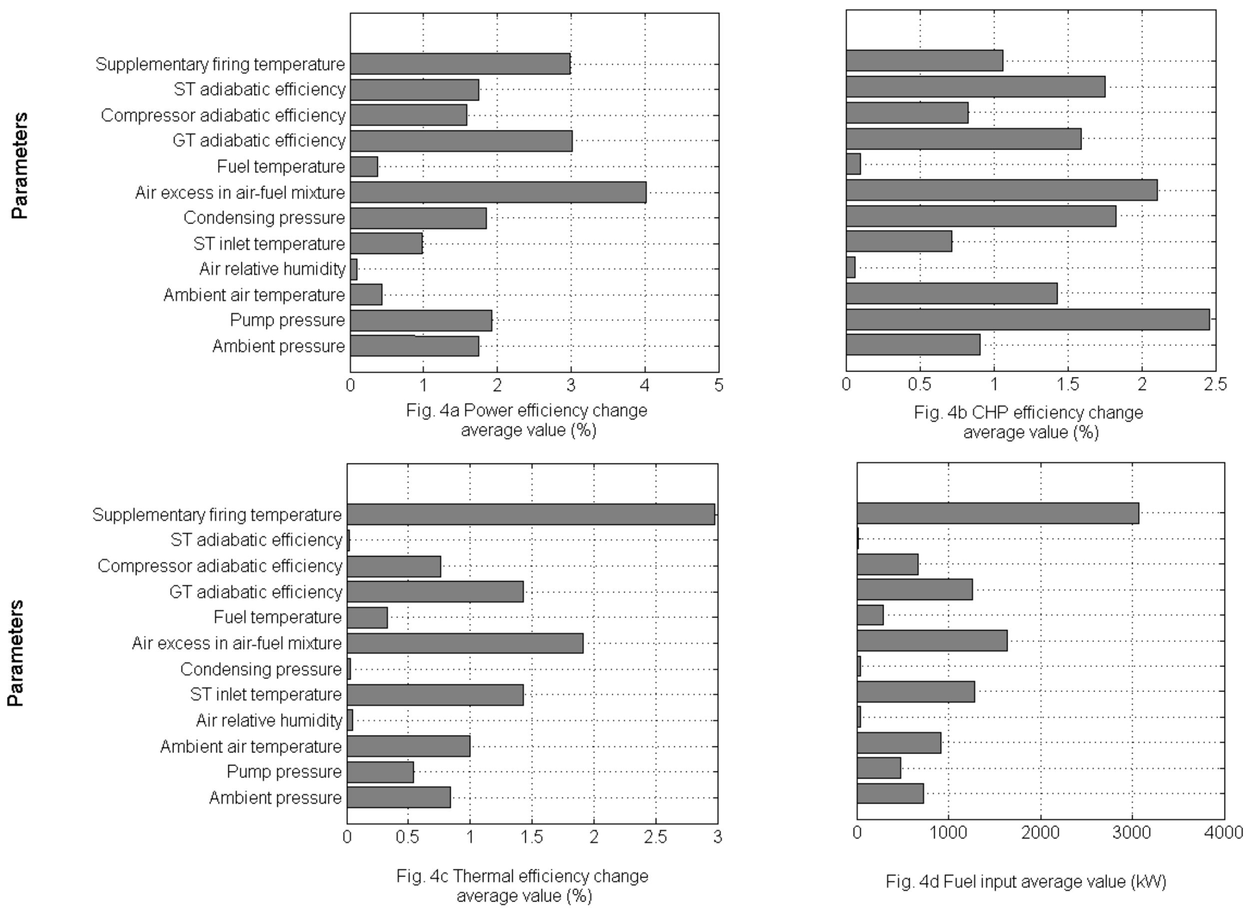

Fig. 4a. The maximum change in the power efficiency was 2.99 % when the minimum supplementary firing temperature was set. This could be explained by increasing the mass flow rate of air-fuel mixture through the GTC. The CHP efficiency showed a negligible variation of 1.06 %, see

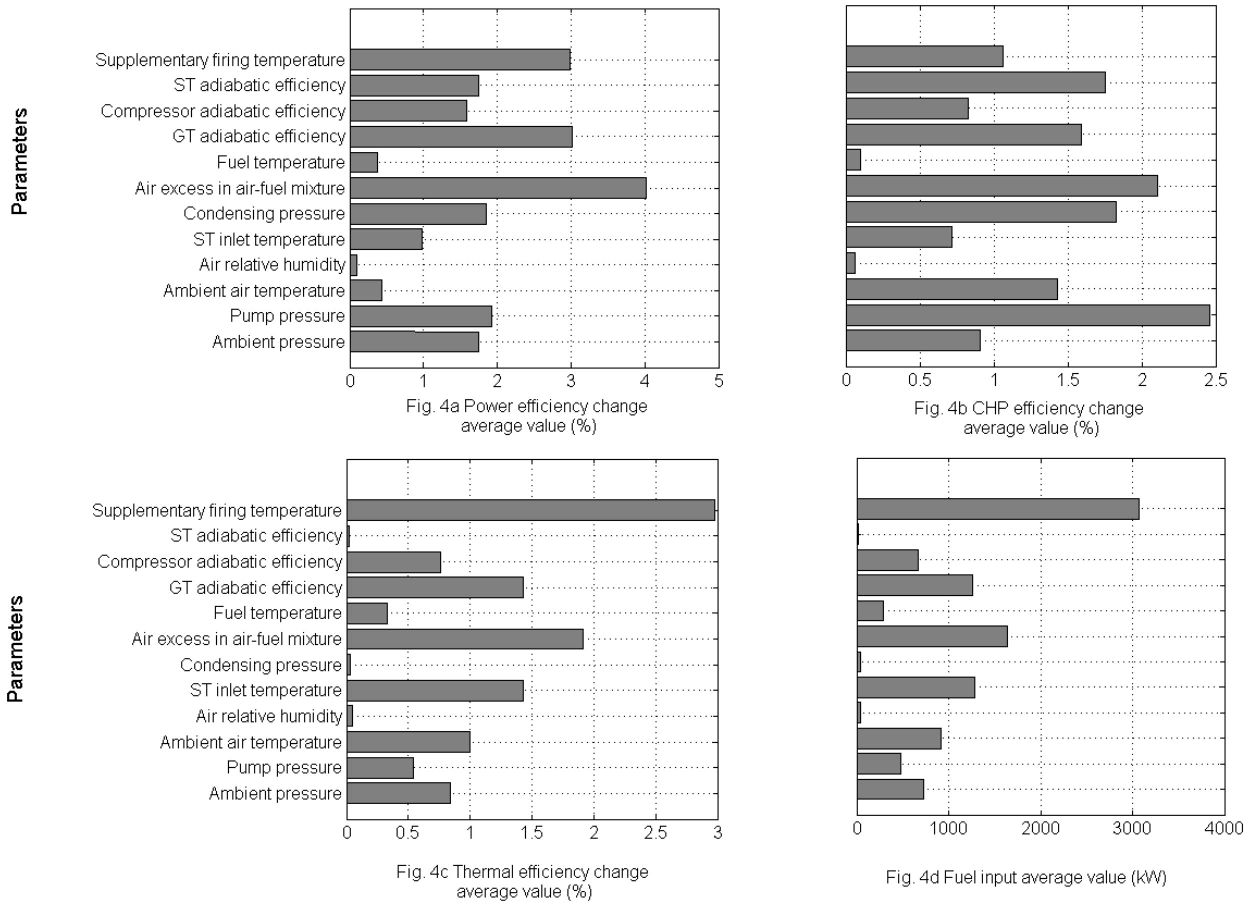

Fig. 4b, due to the supplementary firing temperature. The minimum change in the thermal efficiency occurred due to variation in the following parameters: ST adiabatic efficiency, condensing pressure, and the air RH. This can be seen in

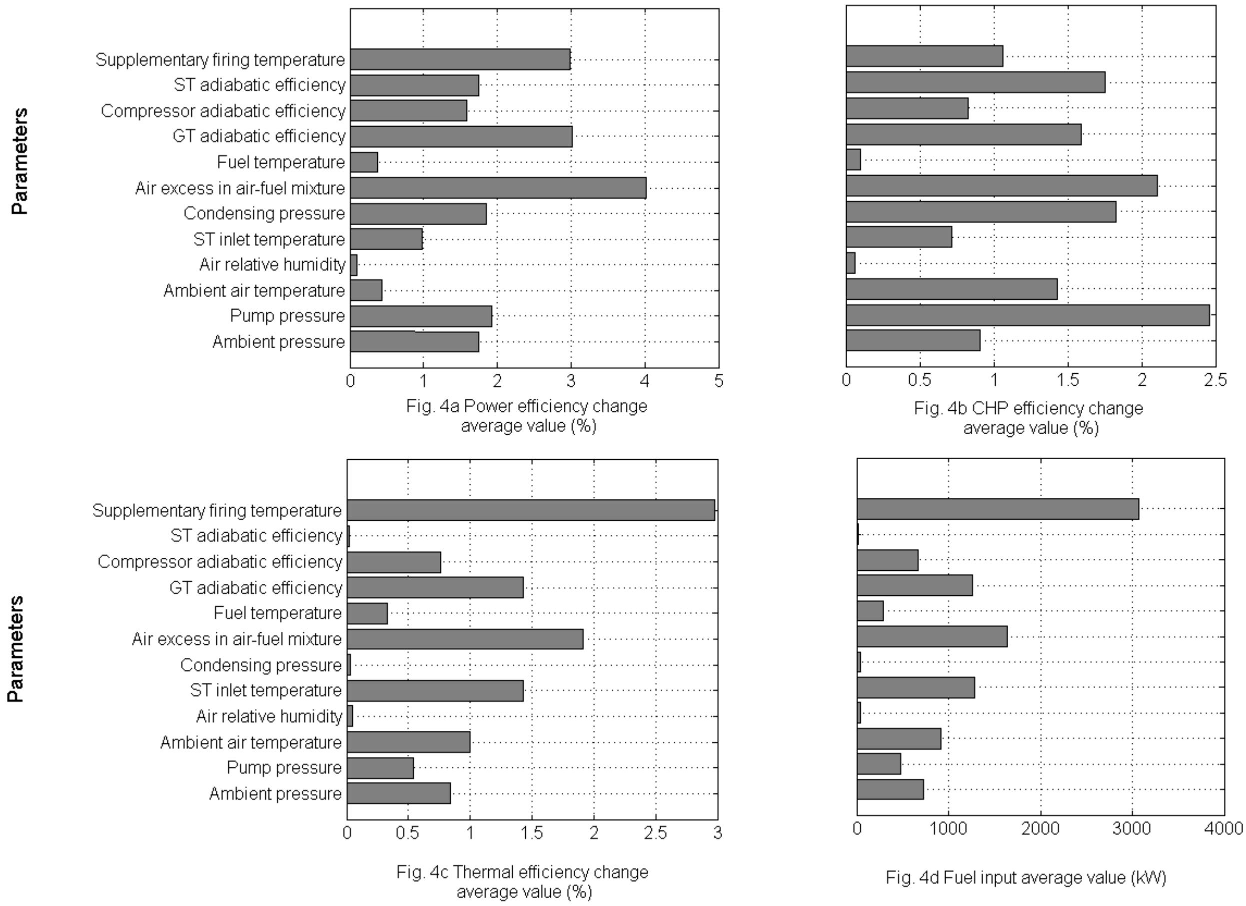

Fig. 4c. The variation in condensing pressure had most effect on power production. The condensing pressure in the CCPP affected the temperature of the water-steam mixture leaving the LPST. The water (compressed liquid) entering the pump before the economizer should not contain any steam fraction; see 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

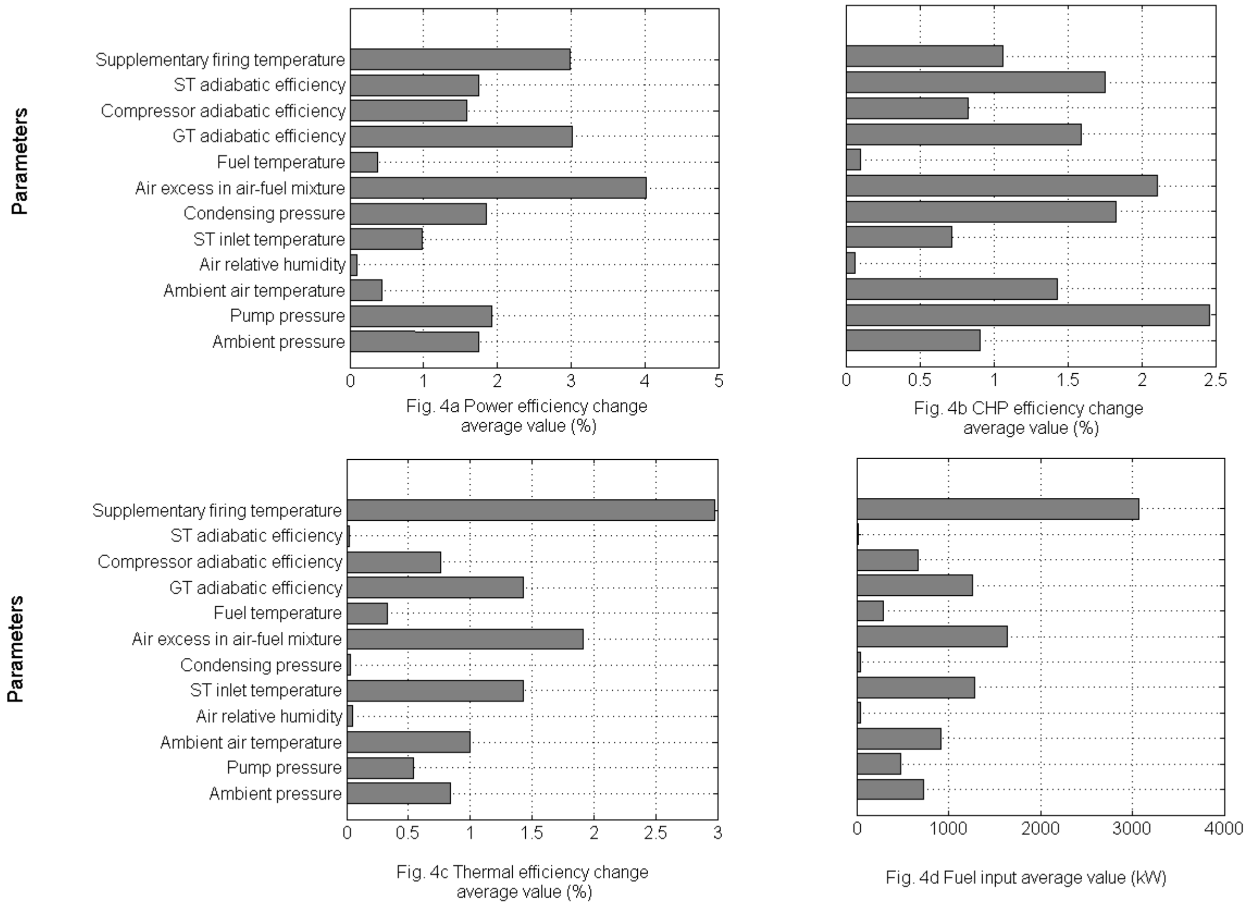

Fig. 4b. The maximum change occurred when the pressure in the STC was increased to 80 bar. The higher the pressure, the higher the amount of electricity produced in the STC. Power production increased by 1.92 % in comparison with the design case when the pump pressure was set to maximum; see

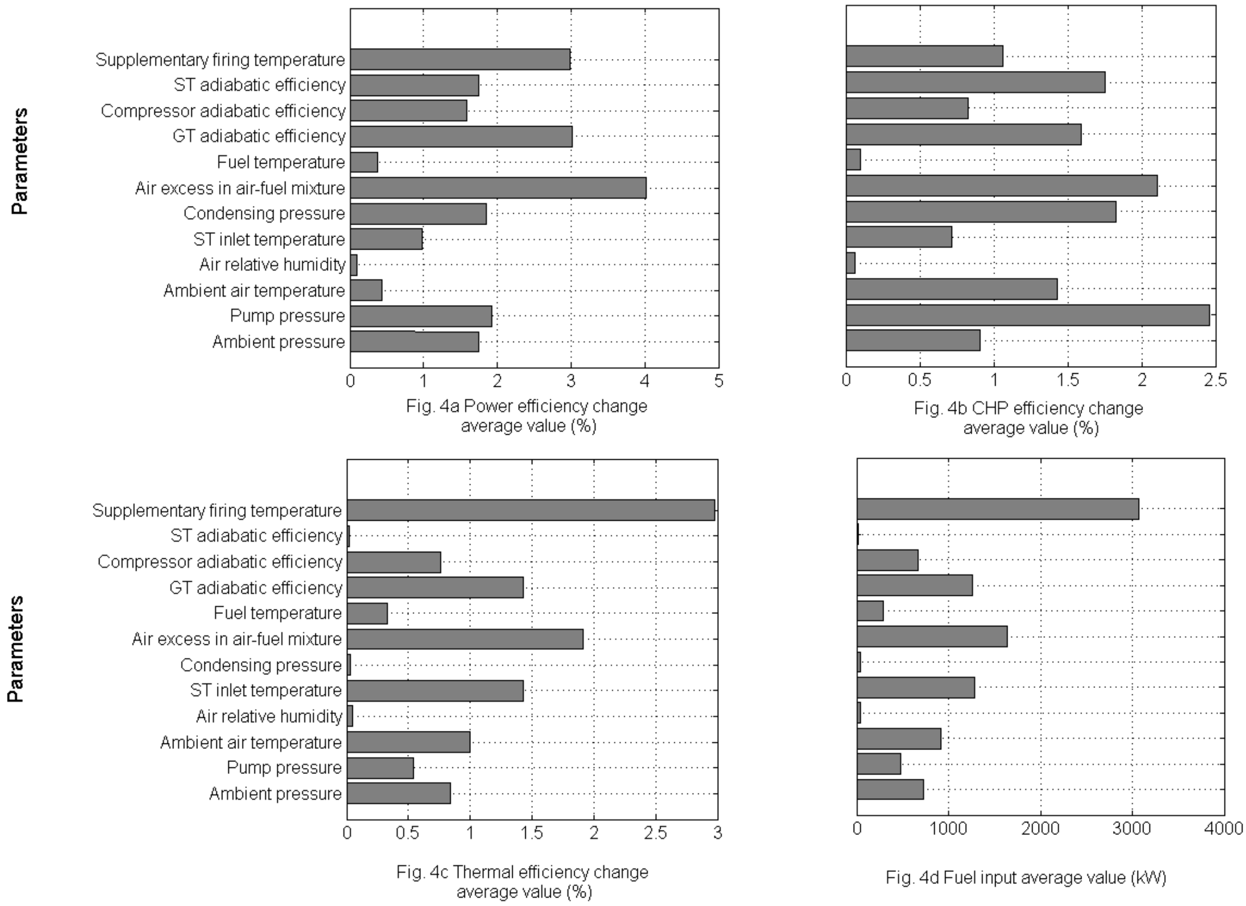

Fig. 4a. The thermal efficiency did not show any particular changes due to the constant vapor temperature level in the STC.

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

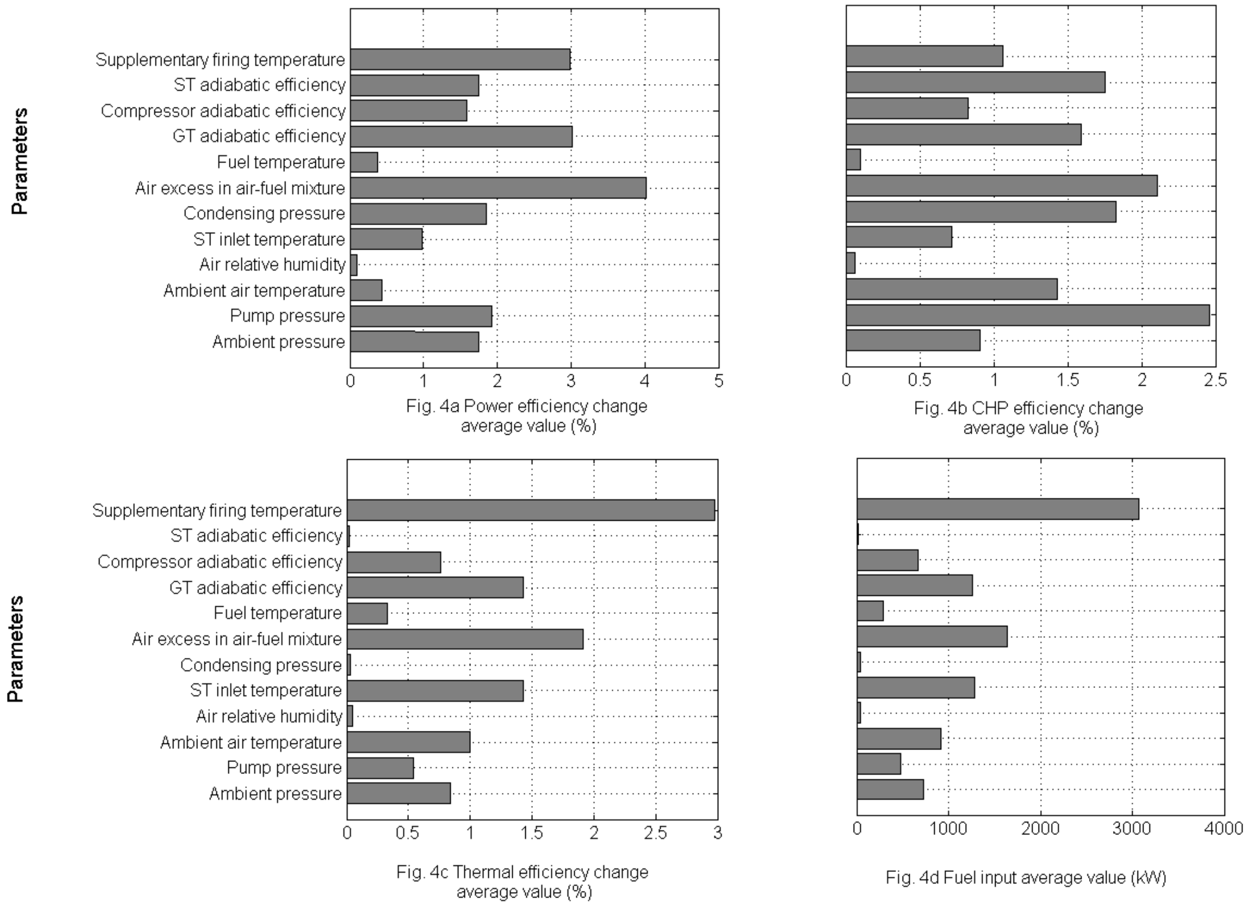

Fig. 4d. These parameters can change electricity production in the plant, but both the fuel input and the thermal efficiencies remained constant.

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.

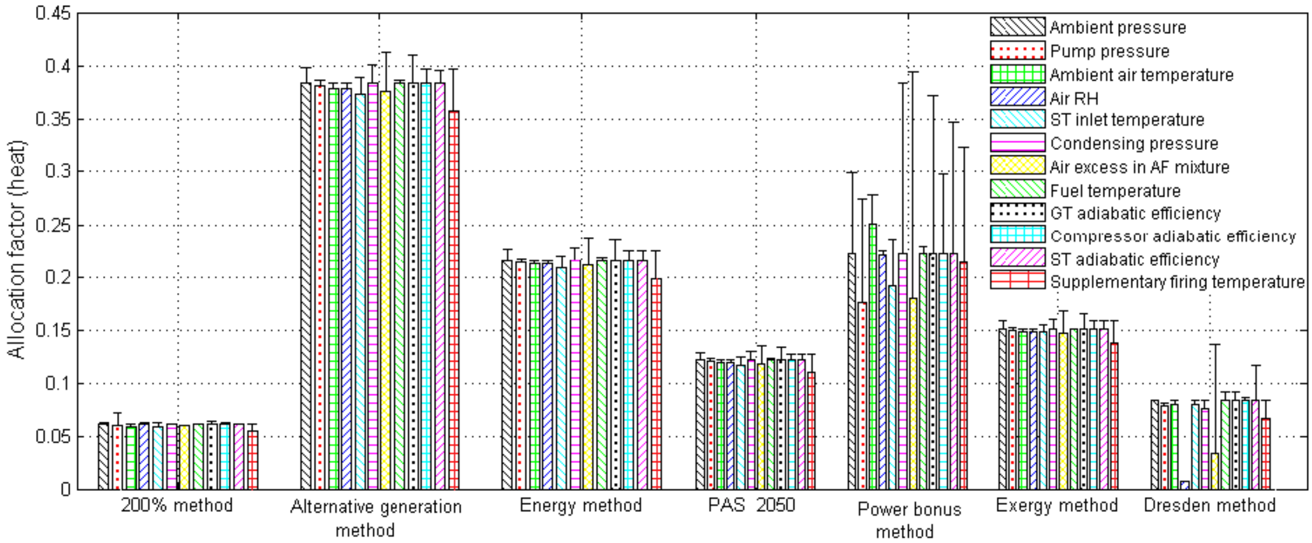

Fig. 7 Heat allocation factors for analyzed methods

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.

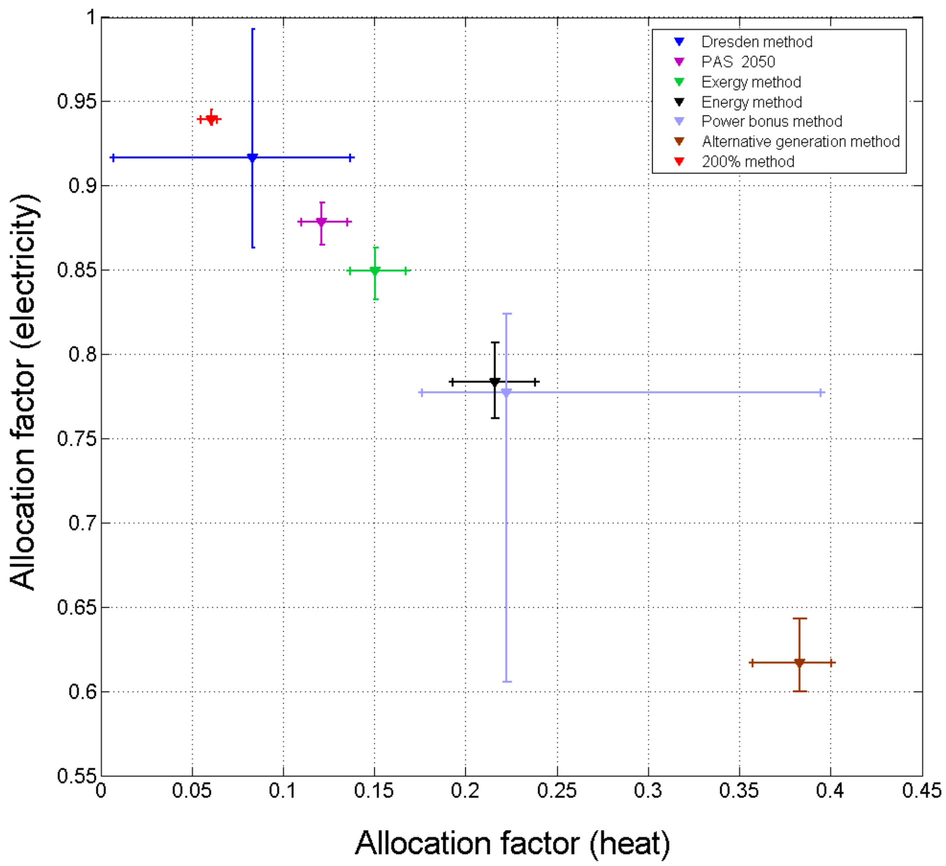

Fig. 8 Sensitivity of allocation factors for heat and electricity production

As Fig. 7 and Fig. 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].

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 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.

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