The usage of natural gas HHV from small cogeneration systems implemented in a 3rd generation DH plant

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Compared with households, Serbian district heating (DH) companies pay higher natural gas and electricity prices. These facts together with the lack of incentives for DH connection and relatively mild winters encourage customers to disconnect and use split systems. To reduce the heating cost and retain customers, DH companies try to reduce operating costs. A promising way is to implement small cogeneration systems that meet their electricity demand. For a case study DH system, the paper analyzes the usage of heat from passive condensation and oil cooling in natural gas-fired cogeneration systems with capacities in the range of 100 kW_e to 2 MW_e. The available heat from cogeneration systems is modeled based on the manufacturers' data. The case study is a boiler room with a total installed capacity of 37.73 MW and a nominal temperature regime 130/75 °C. As the plant operates with approximately constant water flow and variable supply and return water temperatures, the average temperature regime is 69.4/49.5°C for the average outdoor temperature of 5.4°C. The average regime and mild weather enable the usage of oil cooling and flue gas condensation from the cogeneration plant. The heat recuperation for variable minimum temperature differences from parallel and in-series connections of an oil cooler (OC) and a flue gas condenser (FGC) based on meteorological data are simulated. Paradoxically, the system is the least efficient during the coldest weather, when the heat from oil cooling is wasted, and FGC operates as a dry flue gas cooler.

Keywords: District heating, Modelling, Cogeneration, Higher heating value, Heat exchanger

1. INTRODUCTION

Lund et al. [1] classified district heating systems into four generations based on the historical development, temperature, heat carriers, and characteristics of the applied technologies. Additionally, low-temperature systems of the fifth generation have emerged [2] [3]. This progression aligns with the Second Law of Thermodynamics and focuses on reducing the temperature of the medium and improving the exergetic efficiency of district heating systems.

Taking into account the water temperature, district heating systems in the Republic of Serbia operate during one heating season as second, third, and fourth-generation systems. In fact, they predominantly function as third and fourth-generation systems. There are two reasons for this: (i) the climate, characterized by significant fluctuations in outdoor temperature [4], and (ii) the control based on an approximately constant flow of water in district heating systems. Inland, in smaller towns, these systems typically operate for 180 days per year and are not used for heating domestic hot water. There are no incentives for connecting new users. Compared to households, district heating systems face higher prices for natural gas and electricity. As a result, heat utilities are losing customers. To reverse these negative trends, companies are making every effort to increase efficiency levels and implement more cost-effective technologies. Examples include the use of solar power plants to reduce pumping costs [5], the application of heat pumps and cogeneration plants. A feasibility study [6] has shown that cogeneration systems, where the electrical power covers its own consumption, are profitable for district heating company Toplana Kraljevo. Due to the relatively low number of operating hours (below 3000 h/year) and the declining efficiency of gas engines at lower power levels, the optimal electrical power range for the examined heat utility was found to be between 0.8 and 1.2 MWe. The aim of this study is to determine whether it is economically viable to directly utilize the high heat value of flue gas and the heat released by engine cooling oil in a typical district heating plant in Serbia. The case study is JEP Toplana Kraljevo but the outcome could be generalized to the majority of Serbian DH companies. Figure 1 illustrates the supply and return water temperatures in relation to the outdoor temperature. For the average outdoor temperature during the heating season of 5.4°C, the supply and return water temperatures are approximately 69.4°C and 49.5°C, respectively, with slight variations depending on wind intensity. As can be seen, the average return water temperature is suitable for utilizing waste heat from engine cooling oil and the high heat value of the flue gas. The cooling oil typically exits the gas engine at a temperature of 55°C, which is often the dew point temperature of the flue gas resulting from the combustion of natural gas. The mentioned heat can be utilized through active and passive condensation systems [7] [8]. Active systems involve the use of additional energy for utilizing the high heat value of the flue gas through heat pumps [3], while passive systems involve direct heat exchange.

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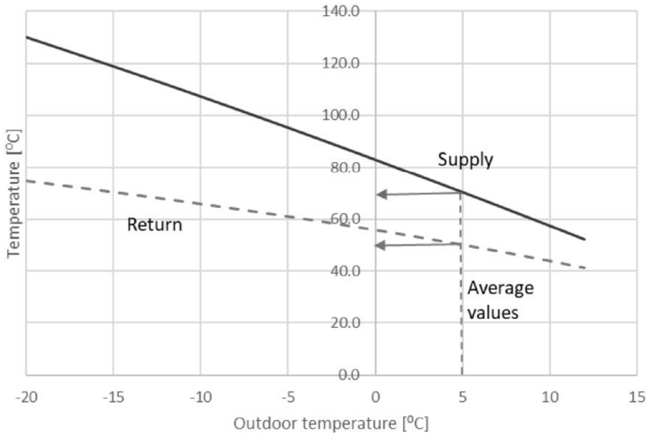

Figure 1: Supply and return temperatures in the examined DH system depending on the outdoor temperature.

2. EXAMINED SYSTEM

Figure 2 shows the examined system. In contrast to conventional systems, this study examines the utilization of an oil cooler (plate heat exchanger) and a flue gas condenser (FGC). Initially, the goal was to analyse both parallel and series connections and determine the optimal dynamics of these connections. However, this approach was abandoned due to the fact that even at the highest operating temperature of the JEP Toplana, the heat loads on the oil cooler and FGC are below 8%. In gas engines, it is customary to cool the oil using air coolers (AC), while the flue gas exits the engine at temperatures above 100°C. In the analysed case, whenever the return temperature would be below 52°C - Δt_{min}, the oil cooler would operate, while the FGC would always operate but with reduced efficiency at higher DH return temperatures compared to the dew point temperature of the flue gas.

Subsequently, the study focuses on determining the optimal design parameters of the FGC to achieve an optimal system configuration.

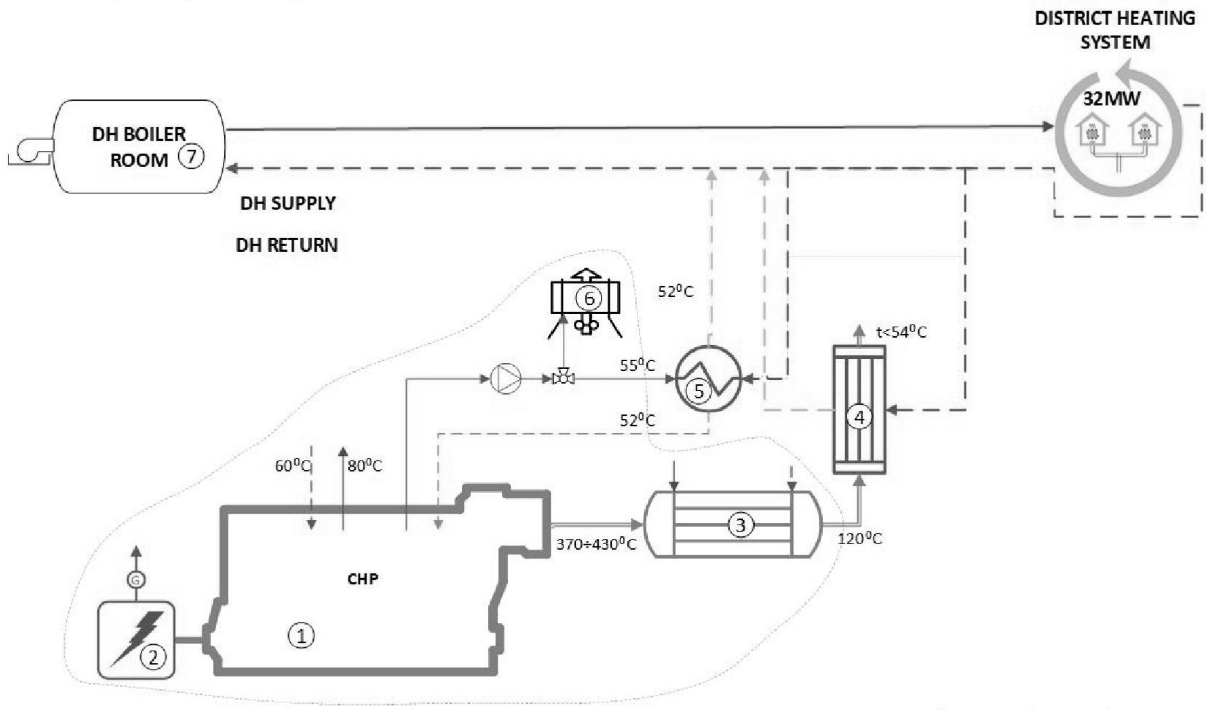

Figure 2: The examined system. 1 – gas engine, 2 – power generator, 3- flue gas dry cooler, 4 – flue gas condenser (FGC), 5 – oil cooler, 6 – air cooler (AC), 7 – boiler room

3. MODELLING

3.1. DH company JEP Toplana Kraljevo

The application of the system was analysed on a cogeneration plant that would be installed in the Central Boiler Room, which has a total installed power of 37.727 MW: 3 boilers, each with the nominal power of 12 MW, and 3 dry flue gas recuperators each with the capacity 575.76 kW. The supply and return water temperatures are shown in Figure 1, while Figure 3 illustrates the variation in heat load depending on the ambient temperature. The presented curve is obtained based on average measured values, which vary depending on the wind. The city is situated in a valley and is characterized by moderate winds. DH company operates intermittently with operating hours between 5 AM and 9 PM during the whole heating season.

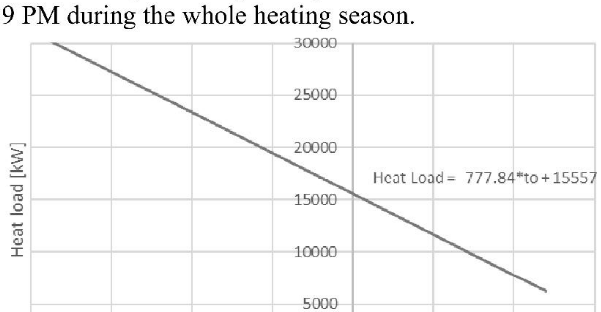

Figure 3: The average heat load depending on the environment temperature.

3.2. Weather Conditions

The heating season of 180 days long has an average outdoor air temperature of 5.4°C. The external design temperature is projected to be -14.7°C. Most of the systems connected to the district heating (DH) network were designed based on an external design temperature of -20°C, which was determined using data from the period 1961-1990. In this study, daily temperature profiles were calculated based on average monthly temperatures (minimum, maximum, and average temperatures) [9] and used in the simulations [10].

3.3. CHP

Gas engines have been modelled based on literature [11] and manufacturer data [12][13][14]. Depending on the nominal electrical power Pel, the electrical efficiency levels of gas engines for various nominal powers are provided in Table 1.

Table 1: Electrical efficiency of gas engines depending on the nominal capacity

P_{el}	η_{el} [%]
< 300 kWe	34%
300÷1000 kWe	η_{el}=0.001 P_{el}+38.784
1000÷2000 kWe	η_{el}=0.0031 P_{el}+38.784

Table 2 displays the flue gas exit temperatures at the outlet of the gas engine.

Table 2: Flue gas exit temperature depending on the nominal capacity of a gas engine

P_{el}	t_{FG}
< 600 kWe	370 °C
600÷1200 kWe	t_{FG} =0.132 P_{el}+3289.19
>1200 kWe	430 °C

Table 3 shows the ratio of heat contained in the flue gas and the engine’s cooling water.

Table 3: The ratio of heat contained in the cooling water and the flue gas

P_{el}	$Q_{flue\ gas}/Q_{cooling\ water}$
< 600 kWe	0.53
600÷1000 kWe	1/(0.0008 P_{el}+0.0075)
>1000 kWe	1.07

The amount of heat released to the surroundings by the engine oil varies from 4.4% to 6.2% of the nominal electrical power of the engine. In this study, the average value was considered, which is Q_{oil} = 0.0565 Pel.

Manufacturers provide all details for combustion with 5% oxygen in dry flue gas. In this case, the gas obtained from the national distributor [15] was used, with the following volumetric composition: N_2 = 0.99%, CO_2 = 0.471%, CH_4 = 96.184%, C_2H_6 = 1.624%, C_3H_8 = 0.508%, C_4H_{10} = 0.175%, C_5H_{12} = 0.049%. By combusting the given gas with an excess air coefficient of 1.28, the volumetric composition of the flue gas in wet products is obtained as follows: CO_2 = 7.673%, N_2 = 73.053%, O_2 = 4.25% (5% in dry products), H_2O = 15.024%, with a dew point temperature of 54.02 °C.

Table 4 presents the potentials in the flue gas and oil for selected nominal powers of gas engines and the previously mentioned flue gas. In the table, "40 °C" refers to the thermal energy released when the flue gas is cooled down from 120 °C to 40 °C.

Table 4: The available heat flow rate in the flue gas and oil depending on the installed capacity

P_{el} [kW]	B [m_N^3/s]	Q_{fg} 40°C [kW]	Q_{fg} 50°C [kW]	Q_{fg} 60°C [kW]	Q_{fg} 70°C [kW]	Q_{oil} [kW]
100	0.008542	31.9	18.1	9.4	7.8	5.7
500	0.037	138.1	78.6	40.6	33.8	28.3
800	0.0587	219.1	124.7	64.4	53.6	45.3
1000	0.073	272.5	155.1	80.0	66.7	56.6
1200	0.0814	303.9	172.9	89.3	74.4	67.9
1500	0.0996	371.8	211.6	109.2	91.0	84.9
2000	0.1283	478.9	272.5	140.7	117.2	113.2

3.4. Heat exchangers

A plate heat exchanger is intended for cooling the oil and heating the water of the district heating system. The overall heat transfer coefficient for these exchangers ranges from 350 to 1200 W/m^{2}K [16]. In this study, the surface area of the exchanger was calculated assuming a heat conduction coefficient of 850 W/m^{2}K and a minimum temperature difference of 3 K in the exchanger. Considering that the oil is cooled in the 55/52°C regime, this exchanger does not operate when the return water temperature of the district heating system exceeds 49°C. In such cases, air cooling is required (see position 6 in Figure 2). The price of the exchanger is calculated based on manufacturer quotes and approximated by the expression:

$$C_{HE} = 627.34 \cdot A + 131.51,$$

where C_{HE} [€] is the price and A [m^2] is the surface area of the exchanger.

The calculation of the FGC is somewhat more complex since it can also operate as a dry flue gas cooler.

Figures 4 and 5 show the modeling procedure and applied algorithms, whereas Table 5 explains the applied nomenclature. Figure 4 shows how the nominal FGC is dimensioned. Based on the nominal condition, i.e. desired performance of the FGC, its surface area is calculated. This is done by calculating the overall heat transfer coefficient by [17] for film condensation of a binary mixture with an inert gas. In the case when the FGC functions as a dry flue gas cooler, the overall heat transfer coefficient decreases and is calculated by [18]. The obtained values were checked by [16] and [19] and agree with the theory as the overall heat transfer coefficient is 1.5 to 2 times higher when the condensation of the flue gas takes place. The previous explains why there are three branches in both algorithms. One is for the FGC, the other for FGC operating as a dry cooler, and the third is for the FGC with two segments: one with and the other without the condensation.

To design the nominal FGC (see Figure 4) the following parameters are required: minimal temperature difference in the FGC Δt_{min}, water inlet t_{wNul} and outlet temperatures t_{wNizl}, and natural gas consumption in the CHP, B_{CHP}. To calculate the actual performance of the FGC (Figure 5) the model requires the following input parameters: overall heat transfer coefficients in the dry k and the condensation zone k_k, natural gas consumption in the CHP B_{CHP}, surface area for the heat exchange A, water flow rate $\dot{m}_w$, and water inlet temperature t_{wul}. Flue gas enthalpies are calculated by [20].

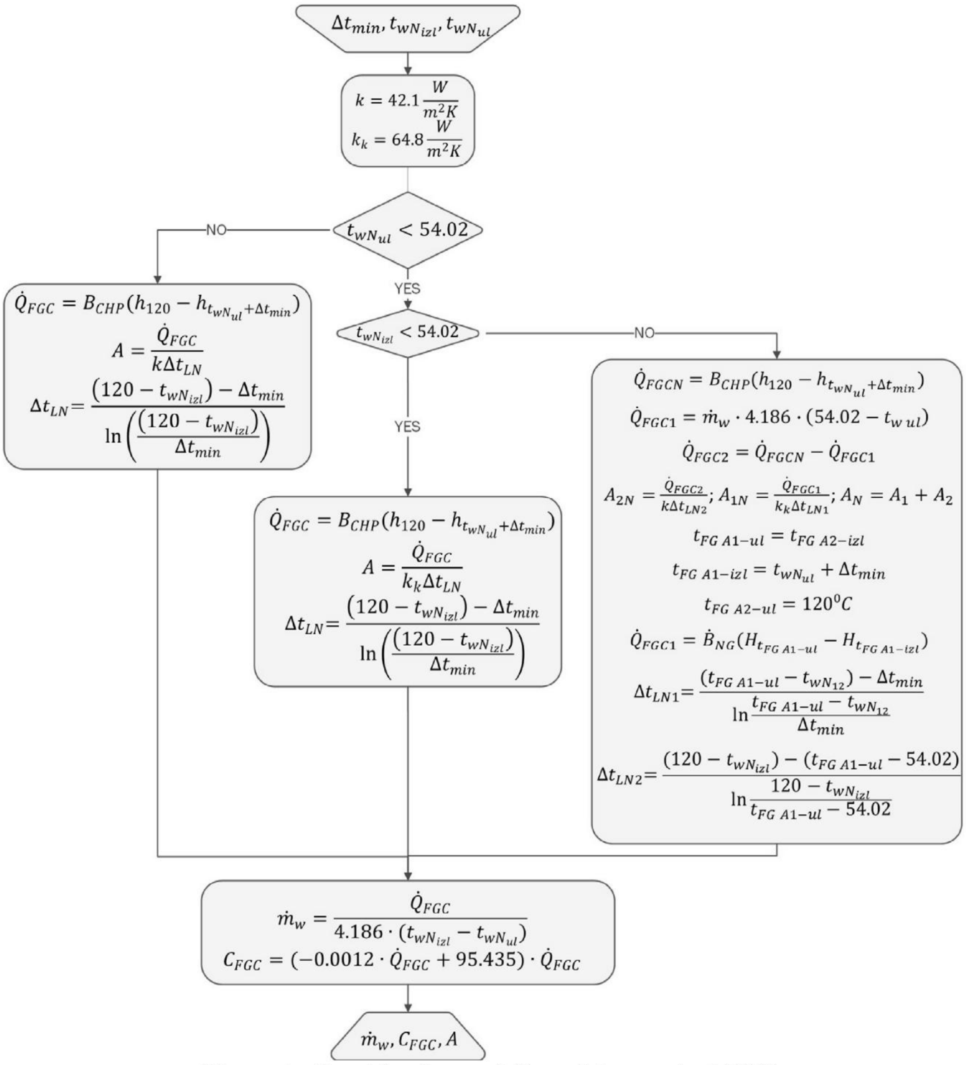

Figure 4. Algorithm for modeling of the nominal FGC

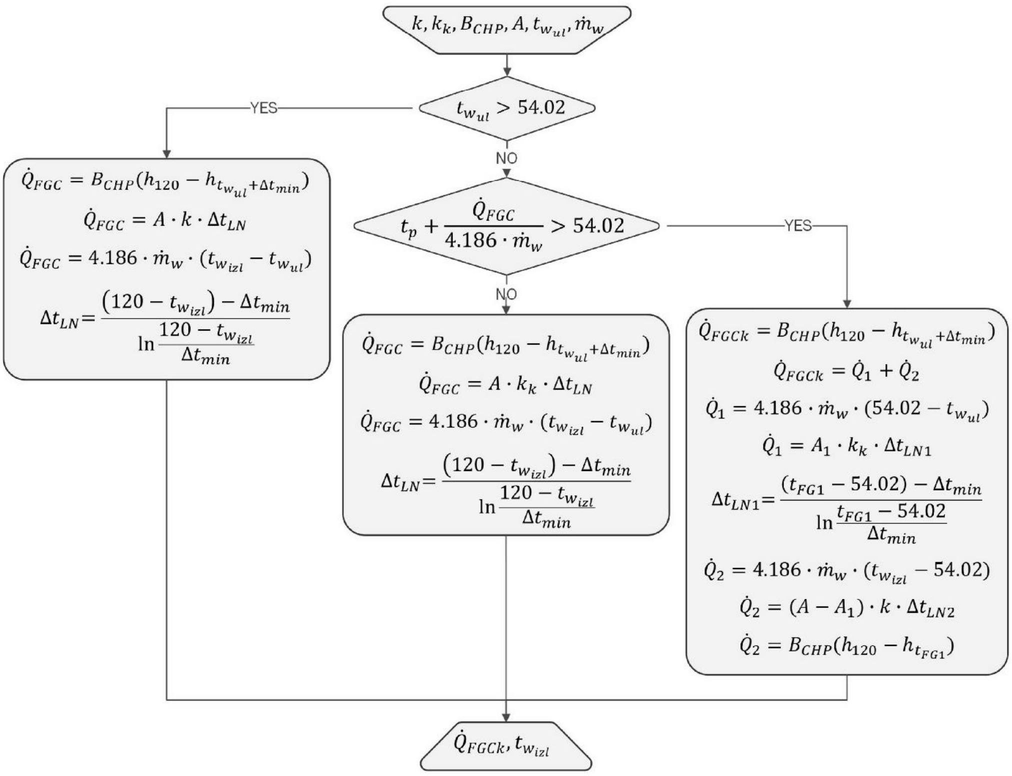

Figure 5. Algorithm for modeling actual performance of the FGC

Table 5: Nomenclature for algorithms shown in Figures 4 and 5.

Terms	Unit and quantity
Δt_{min}	[°C] minimal temperature difference
t_{wNizl}	[°C] nominal water temperature at the outlet of the FGC
t_{wNul}	[°C] nominal water temperature at the inlet of the FGC (DH return)
k	[W/m²K] overall heat transfer coefficient (dry cooling of the flue gas) in the FGC
k_k	[W/m²K] overall heat transfer coefficient in the FGC
A	[m²] surface area
Δt_{LN}	[°C] logarithmic temperature difference
h_{120}	[kJ/m$_N^3$] enthalpy of the flue gas at 120°C per mN3 of natural gas
$h_{t_{wNul}+\Delta t_{min}}$	[kJ/m$_N^3$] enthalpy of the flue gas at the outlet of FGC
B_{CHP}	[m$_N^3$/s] fuel consumption of the FGC
$\dot{m}_w$	[kg/s] water flow rate in FGC
Q_{FGC}	[kW] nominal heat flow rate in the FGC
Q_{FGC1}	[kW] heat flow rate in the part of FGC with condensation
Q_{FGC2}	[kW] heat flow rate in the part of FGC without condensation
Q_{FGCN}	[kW] designed power of the FGC
A_{1N}	[m²] surface area of the FGC where condensation take place
A_{2N}	[m²] surface area of the FGC where condensation does not take place
t_{FGA2ul}	[°C] flue gas temperature between the zones with and without condensation
$t_{FGA2izl}$	[°C] flue gas temperature at the exit of the dry zone
t_{FGA1ul}	[°C] flue gas temperature at the entrance of the dry zone
$t_{FGA1izl}$	[°C] flue gas temperature at the exit of the condensation zone
t_{wN12}	[°C] nominal water temperature between the condensation and dry zones
Δt_{LN1}	[°C] logarithmic temperature difference in the condensation part of the FGC
Δt_{LN2}	[°C] logarithmic temperature difference in the dry part of the FGC
t_{FG1}	[°C] flue gas temperature between the zones with and without condensation
t_{wizl}	[°C] water temperature at the outlet of the FGC
Q_{FGCK}	[kW] thermal power of the FGC

4. RESULTS

A test was conducted on an oil cooler operating in the 49/52°C regime. In this operating mode, the cooler utilizes 47.1% of the available energy for oil cooling, which corresponds to $0.471 \cdot 180 \cdot 16 \cdot 0.0565\ P_{el}$ in kWh/year.

The performance of a total of 6 different nominal designs of the FGC was investigated: 40/42°C (nominal water inlet and outlet temperatures), 40/50°C, 48/50°C, 50/55°C, and 52/55°C.

Figure 6 shows the performances of the FGC 48/50°C for average days in each month during the heating season. The climate conditions dictate that this exchanger operates mostly as an FGC rather than a Dry Flue Gas Cooler (DFGC), even in January. The steeper functions on the graph represent the operation when the water temperature at the exchanger's inlet is very close to the dew point temperature of 54.02°C, resulting in condensation occurring in one part of the exchanger while not in the other. This phenomenon becomes less pronounced as the nominal range of water inlet and outlet temperatures decreases. Interestingly, the OC (Oil Cooler) cannot be used during low outdoor temperatures, and additional energy is not required for FGC operation because the temperature difference is sufficient to overcome the chimney draft. The power of the exchanger is highest at the lowest return water temperatures, specifically temperatures lower than the nominal design conditions. The most significant influence on the thermal power absorbed by the FGC is whether the return water temperature is lower than the dew point temperature of the flue gas. Considering that the convective heat transfer coefficient is 1.5 to 2 times smaller for return water temperatures above the dew point temperature [17], the FGC power is reduced accordingly for water temperatures above the dew point temperature.

Figure 7 illustrates the surface areas of the exchanger based on the nominal temperature regime and the installed power of the gas engine. Lower nominal water temperatures result in larger exchanger surfaces. This is understandable because more energy is extracted from the flue gas. The 40/50 regime has a larger surface area compared to the 40/42 regime because it has a smaller logarithmic temperature difference.

Figure 8 displays the total heat collected by the exchangers during the heating season, depending on the conditions for which they are designed. It is evident that lower nominal temperatures and smaller temperature ranges result in better exchanger performance. The 40/42 exchanger outperforms the 40/50 exchanger because the latter operates more frequently in transitional conditions, where there is no condensation in certain parts of the exchanger (refer to Fig 4).

Exchangers designed for smaller temperature differences in the water operate more frequently in conditions involving condensation. Furthermore, exchangers designed to operate just below the dew point temperature exhibit excellent performance because they achieve even better efficiency at lower water inlet temperatures (corresponding to the peak in Figure 6).

Considering the average efficiency of heat energy production in the analysed boiler plant at 92.2%, a price of 0.287 €/kWh, and the heating value of natural gas at 9.564 kWh/m³, the payback period of the investment depends not on the cogeneration plant's power but on the conditions for which it is designed. Table 6 shows the payback periods of the investment. Exchangers that are designed very close to the dew point temperature of the flue gas exhibit the most favourable ratio of surface area to total utilized energy for the given climatic conditions. The least favourable ratio for FGC design is in the middle of the expected temperature range for condensation.

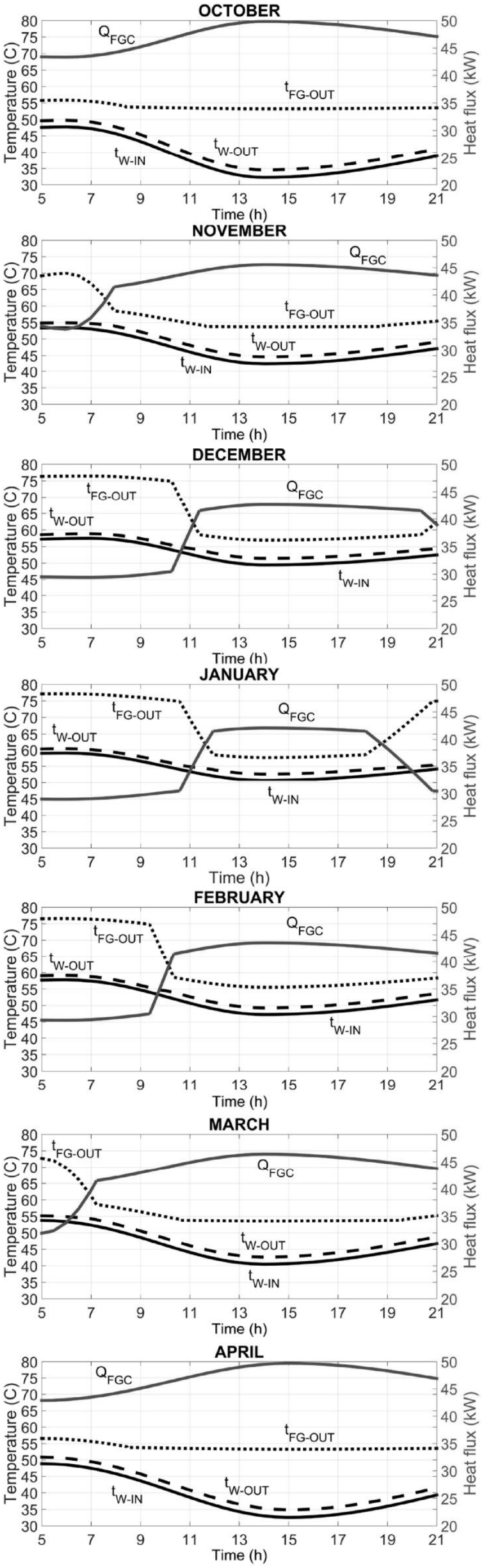

Figure 6: Performance of the FGC with nominal temp. regime 48/50ºC and Δt_{min}=8ºC throught the heating season. $t_{W\text{-}IN}$ temperature of the DH return water, $t_{W\text{-}OUT}$ water preheating temeprature, $t_{FG\text{-}OUT}$ flue gas temeprature after the FGC and Q_{FGC} thermal power of the FGC.

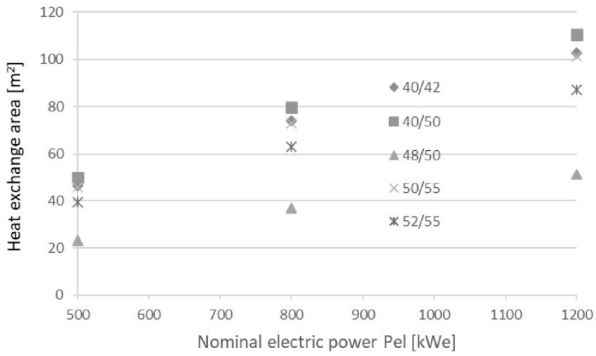

Figure 7: Required surface areas for heat exchange depending on the nominal regime and electric power of the CHP.

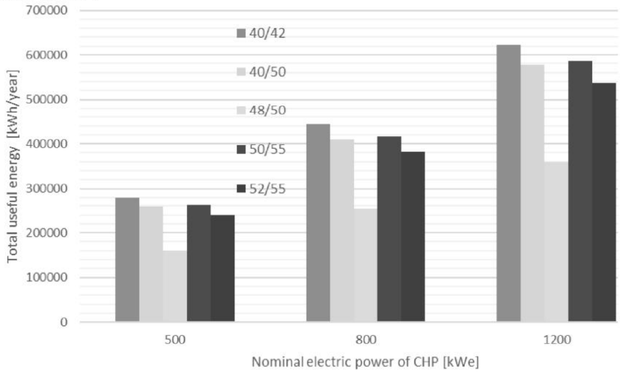

Figure 8: The total heat harvested by the FGC depending on the nominal temperature regime and electric power of the CHP.

Table 6: Simple payback periods depending on nominal temperature regimes (40/42 means water inlet and outlet temperatures are 40ºC and 42ºC, whereas the flue gas inlet and outlet temperatures are 120ºC and 48ºC)

Nominal temp. regime	40/42	40/50	48/50	50/55	52/55
Simple payback period	1.5	1.7	1.3	0.75	0.8

5. CONCLUSIONS

The main conclusions are:

- The usage of HHV from the analyzed gas engines in the case study DH system is highly profitable with a simple payback period in the range from 0.75 to 1.5 years. The real payback period would be longer because of operation and maintenance costs. Although profitable, the system technically complicates the CHP plant as it requires a new chimney and additional pumping power. In the analyzed system, the additional fan power could not be replaced by the usage of heat from the oil cooler.
- 47.1% of the heat liberated by the oil cooler can be used in the DH system.
- The nominal conditions by which the FGC is designed are the most influential factor that determines the profitability of the investment.

- The temperature rise of DH water in the FGC should be as small as possible. In that case, during shorter periods the FGC would operate as a dry flue gas cooler.
- The most profitable FGC are those that are designed to heat DH water to the dew point temperature with as smaller as possible temperature rise.
- The colder the weather the less efficient the analyzed system. At low environment temperatures, the heat from oil cooling could not be used and the FGC operates as a dry flue gas cooler with 1.5 to 2 times smaller capacity.

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