Comparative study of gas engines and gas turbines in cogeneration (CHP), using the example of a typical public district heat distribution network

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A comparative study of gas engines and gas turbines in combined heat and power generation for a typical public heat supply network

Tobias Vogel, Gerd Oeljeklaus, Thomas Polklas, Christian Frekers and Klaus Görner

Abstract

A comparative study of gas engines and gas turbines in combined heat and power generation for a typical public heat supply network

Besides the reinforced expansion of renewable energies one other central target in the German Energiewende is the efficiency enhancement using fossil energy. Due to the German climatic conditions, the combined heat and power (CHP) generation is a suitable instrument to achieve this goal. Therefore, it is part of the new energy concept of the German Federal Government. But owing to the changing market associated to the Energiewende, also other technologies move into spotlight, like large, stationary, high-efficient gas engines.

As modules of a power plant network these engines can provide thermal energy to a district heating network alternatively to a gas turbine combined cycle power plant (CCPP) in CHP-mode. The engine's waste heat originating from cooling water and exhaust gas can be used either directly for heat supply or in a water-/steam cycle with an extraction back-pressure turbine and following heating condensers (HeaCo). These three systems (CCPP, engine, engine + HeaCo) have been modeled, simulated and evaluated based on annual data of a representative district heating network.

All systems comply the values for primary energy saving and fuel utilization ratio required by the German KWK-act. Regarding the annual exergetic utilization ratio the engine systems offer an advantage of 1-2 %-points. Due to the higher power production the system engine + WSC is economically favorable. In addition, these engines have an enormous ability for residual load management with load transients of up to 33 % $MW_{el,inst}$/min.

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Introduction

The transformation of energy systems currently underway in Germany is showing an increasingly noticeable impact on the existing energy industry. Due to the long-term expansion targets of e.g. 80% renewable energy being used in electricity generation by 2050 [3], it can be assumed that the demands from fossil power generation will change and that especially time flexibility will gain even more significance than today in order to shoulder the growing demand for balancing energy. According to the recently-published BMWi White Book [3], 25% of the fossil share is expected to be covered by plants based on the combined heat and power generation principle (CHP). This emphasizes the significance of CHP within thermal power plant technology, although the absolute amount of electricity provided by thermal power plants, and therefore CHP, will decrease in the long term with a concurrent increase of renewable energies. Due to their lower greenhouse gas emissions, gas-powered plants offer significant advantages. The main differentiation to be made in CHP application is between industrial and public heat supply. A typical example for the latter are municipal power utilities with district heating networks, which have been selected for this study due to their comparability. Many municipal power utilities are already using gas-powered CCPP plants as thermal power stations, which offer lower greenhouse gas emissions and higher efficiency levels compared to CHP plants powered by solid fuels.
Modern and highly efficient CCPP plants, as shown i.a. by the example of Irsching 5, can currently not be operated profitably in Germany due to the low electricity prices and low prices for CO_2 emissions. The CHP approach, with its privileges such as revenue from heat sales, CHP bonus and CHP electricity input priority, offers an opportunity to improve profitability, a notion which is also supported by the planned construction projects for thermal power plants in Düsseldorf (Lausward Block Fortuna) and Cologne (thermal power station Niehl 3).

Within the area of CHP applications, modern and highly efficient gas engines arranged in combined power plants, can present an alternative to CCPP plants. As well as delivering highly efficient CHP capabilities, these engines can participate in the balancing energy market due to their high flexibility, a factor which can further increase profitability. Furthermore, their modular construction with unit sizes of approximately 10 MW_{el} allows an operation that is tailored to requirements and simultaneously offers high efficiency across the entire load range. Hence the following will describe a study based on the possibility of providing full coverage of all energy supply requirements with engine-based power plants as an alternative to an existing CCPP system. Apart from employing plants that operate purely with engines, it is also possible to equip the engines with a downstream located water-/steam cycle, with the goal of maximising the electricity yield whilst simultaneously fulfilling the CHP requirements.

Specification of the scope of the investigation

Considering the background of the current situation on the German electricity market, it seems interesting to compare CHP plants based on stationary gas engines with currently utilised systems in order to gain an evaluation. The main areas of application for CHP plants in the multi-digit megawatt range are industrial CHPs as well as the supply of district heating. Industrial CHP plants usually need to adhere to industry-specific process heat restrictions. Conversely, the energy utilisation pathway for district heating takes place within a more uniform environment, which is why this area of application was chosen for this case study. Due to the variable heat requirements throughout the year, the review needs to be carried out based on annual yield calculations. Therefore as initial step within the thermodynamic modelling the design layoutand the partial-load operation concept was worked out.

Example application and boundary conditions

For this study, the district heating network of a mid-sized city was chosen as a specific example, representing a typical average district supply system in a moderate climatic area of Germany.

The weather profile used for the location was taken from the Meteonorm 7.0 software in hourly time resolution. At this location, the average annual temperature is +10.5 °C, with the daily mean temperature varying between -6.7 °C and +26.3 °C. Using degree-day numbers in conjunction with daily mean temperatures in accordance with VDI 2067, the duration of the heating period can be determined. In the reference year, with a heating threshold of +15 °C, this amounts to 262 days.

For the district heating supply, a requirement profile representing the dependence of the necessary feed temperature as well as the required district heat load on the ambient temperature has to be established. With reference to published examples, the following constellation was assumed. Depending on the ambient temperature Tamb, the flow temperature varies continuously between + 130 °C (T_{amb} < -10 °C) and 80 °C (T_{amb} > +15 °C), with the return temperature remaining constant at +60 °C. The district heating load to be supplied varies continuously between 161 MW_{th} (T_{amb} = -15 °C) and 12 MW_{th} (T_{amb} = +30 °C). This demand profile was transferred to the reference year, and the district heating annual cycle was determined (also see figure 3). The district heating demand for the reference year and the chosen location was determined to be 587.8 GWhth/a under the given assumptions.

Power plant systems

All power plant systems were modelled and simulated using the commercially available power plant modelling programme Ebsilon®Professional (in short: Ebsilon) in version 10.05. The baseline was formed by a modern CCPP system in a medium power range (approx. 100 MW_{el}). The gas engine systems were compared on the basis of this reference plant. As a preliminary point, this section provides a detailed introduction to the technical system configurations.

CCGT plant (reference plant)

For CCPP plants in the medium power segment that are being operated as thermal power plants, typical configurations consist of two gas turbines plus a heat recovery steam generator with auxiliary firing and a downstream located back-pressure extraction turbine. In addition to that, an auxiliary boiler is usually installed to cover district heat peak loads. The plant schematic selected for the reference plant is shown in figure 1, along with key process parameters and output data for district heating supply temperatures of +130 °C and +80 °C.

The hot gas turbine exhaust gas (approx. 500 °C) is further heated in the heat recovery steam generator with auxiliary firing as required, and subsequently used for superheating, evaporation and preheating of the feed water in the water-steam cycle. Following, a further part of the waste heat contained within the gas turbine exhaust gas is then used to provide district heating. The in the HRSG generated live steam then flows into an back-pressure extraction turbine. Here, the steam is expanded to low pressure level (LP), with partial steam extraction taking place at intermediate pressure level (IP). Extraction steam as well as exhaust steam is used and therby condensed in two heating condensers (HeaCo) to supply the district heating system. In order to cover the peak load, a district heating auxiliary heater is coupled downstream of the IP heating condenser on the district heating side. The operating concept of the CCPP plant is stated to be as follows: The CCPP runs at full load for maximum district heating supply temperature. If the required district heating supply flow temperature drops, only the added thermal power from the auxiliary heating is initially reduced. This is then deactivated above approximately +107 °C. A further load reduction is carried out by reducing the auxiliary firing in the gas turbine exhaust gas. The final load reduction can be achieved by shutting down one gas turbine, with a further load reduction by operating the remaining gas turbine at part load being avoided. For this reason, partial heat removal from the provided district heat takes place via the re-cooler station during times of very low district heating demands, which is not shown in figure 1 but is considered in the calculation.

Gas engine systems

The gas engine used for this study is the MAN Diesel & Turbo SE 20 V 35/44 G in the CHP version as well as the GCC version (GCC = waste heat recovery using water-/steam cycle), both of which are optimised for the respective application. Both engine versions provide the same 10.6 MW_{el}, but with slight differences in the electrical efficiencies of 45.5 % (CHP) and 45.1 % (GCC). All figures in this section apply for one engine respectively. Independent of the engine version, waste heat accrues at a total of three temperature levels (exhaust gas, high temperature (HT) and low temperature (LT) cooling water), which are subsequently to be used further for energy efficiency purposes, for example for district heating or electricity production.

For the CHP version in this study, the waste heat is transferred directly to the district heating network using heat exchangers. At full engine load this lies at a constant at 9.47 MW_{th} per engine across the entire range of the district heating supply flow temperature (+130 to +80 °C). This configuration is described as "Engine(CHP)" system.

In the GCC version, the engine's exhaust gas has a temperature of 395 °C at full load. The use of such a high temperatures for providing low-temperature heat is exergetically inefficient. Efficiency can be improved by coupling a water-/steam cycle for electricity generation downstream of the engine exhaust path. Depending on the application, various layouts

technical data

parameter	value (130°C)	value (80°C)	unit
Q_f_DH-supplementary heater	51,65	0,00	MW_{th}
Q_f_total	299,35	81,89	MW_{th}
P_el_net	101,40	38,13	MW_{el}
Q_DH	159,00	32,00	MW_{th}
Exhaust gas temperature (inlet HRSG)	875,00	503,00	°C
live steam mass flow	45,27	11,46	kg/s
live steam temperature	525,00	394,31	°C
live steam pressure	80,00	18,73	bar
back-pressure IP-HeaCo	1,59	0,53	bar
back-pressure LP-HeaCo	0,60	0,31	bar
DH-flow line temperature	130,00	80,00	°C
DH-return line temperature	60,00	60,00	°C

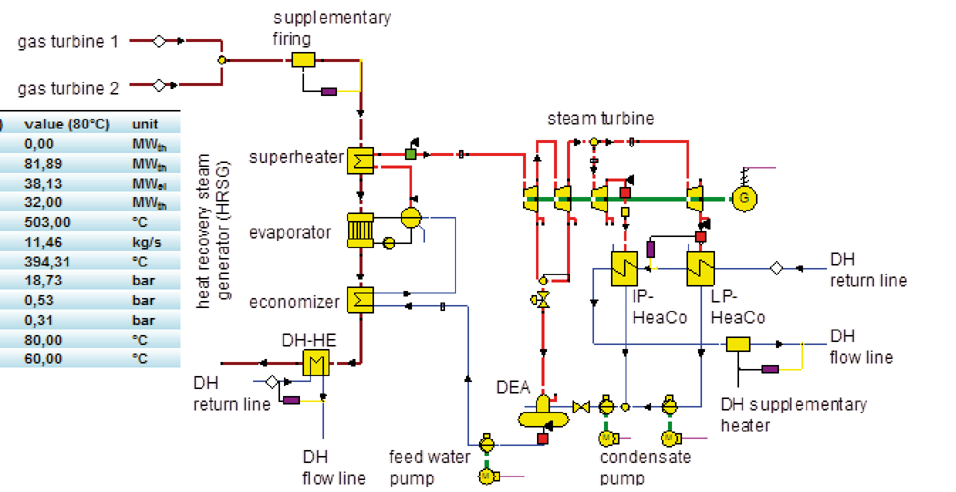


Image 1. Process flow diagram with main process parameters of the reference plant.

are possible for this. It must be differentiated as to whether the requirement is to provide the highest-possible electricity generation capacity (application: condensing turbine) or to provide low temperature heat with concurrent electricity production (application: back-pressure turbine). For the application considered here, the use of a back-pressure turbine similar to the CCPP reference plant seems promising, since the exhaust steam can also provide district heating. Figure 2 shows the respective process flow diagram of such a module, consisting of a gas engine (GCC version) coupled with a downstream located water-/steam cycle, composed of a single-stage heat recovery steam generator (HRSG), a back-pressure extraction turbine and two heating condensers. In the following, this will be referred as „engine (GCC)+HeaCo"system. Due to the variable feed temperature in the district heating, the main process parameters for both limiting cases (district heating supply flow temperature of + 130 and +80 °C) are displayed in the table to give an impression of the range of values.

First, the engine exhaust gas is routed to a heat recovery steam generator (HRSG), consisting of superheater, evaporator with drum, and preheater. As a result of the heat transfer from the engine exhaust gas to the water-/steam cycle, live steam at 380 °C and approx. 20 bar is generated. The live steam is then expanded in a back-pressure extraction turbine, whereby the extraction serves the supply of the deareator (DEA) with bleed steam and thereby ensures degassing. The majority of the steam, however, is still fully expanded to the level of possible back-pressure depending on the district heating supply flow temperature. The turbine exhaust steam condenses in the heating condenser, thereby transferring the heat released in the process into the district heating network. After this, the condensed water is fed to the deareators by the condensate pump, from where it then returns to the heat recovery steam generator via the feed water pump.

In this process flow, the district heating supply is provided in two ways. On the one hand by directly using the engine exhaust heat in the form of cooling water and the residial engine exhaust heat, and on the other hand indirectly via the heating condenser. When the engine exhaust heat is used directly, the water returning from the district heating with a temperature of 60 °C is routed to the heat exchanger LT-CW-DH-HE. Here, the water is heated for the first time by using the waste heat in the low-temperature cooling water. After this, it flows to the HT-CW-DH-HE, where the HT cooling water further heats the water (on the district heating side). A further temperature increase is provided by the residual heat in the engine exhaust gas. Depending on the required temperature level of the district heating supply flow, different mass-flow overlays occur, which is why bypasses and additional (district heating return) feeds are provided, which will not be discussed here further. In the case of indirect provision, the water from the district heating return is routed back to the heating condenser and is heated up before then being added to the water which has been directly heated. A gas-fired peak load boiler ensures that the maximum district heating supply flow temperature can be met.

For both concepts, Engine(CHP) and Engine (GCC)+HeaCo, a modular structure up to the level of a large combined power plant is possible. This means that the supply of district heating networks, such as in this example, can be covered in a similar manner as with the CCPP concept. In this context, the following section describes an evaluation of the reference year carried out with the developed calculation models.

Evaluation parameters

With regard to the evaluation parameters, it necessary to differentiate between energetic is and exergetic parameters, as well as between considerations based on specific points in time and time periods. For a specific point in time, the efficiency represents the ratio between useful power and total power. Analogously, a mean efficiency can be calculated over a period of time, which is then described as degree of utilisation, representing the relationship between the target energy output and the energy expended. Since CHP plants are operated under different boundary conditions, the time-based evaluation approach is of great importance, hence why it is used here.

Fuel utilisation factor ω

When evaluating the energy characteristics of a CHP plant in the form of an efficiency, i.e. the ratio of useful power output to energy expenditure, the two qualities of the different target energy types, electrical energy W and heat Q, are considered to be equal. In order to make this distinction, the efficiency of the CHP plant is described according to VDI 4608 as the fuel utilisation factor ω), which is calculated as per equation 1:

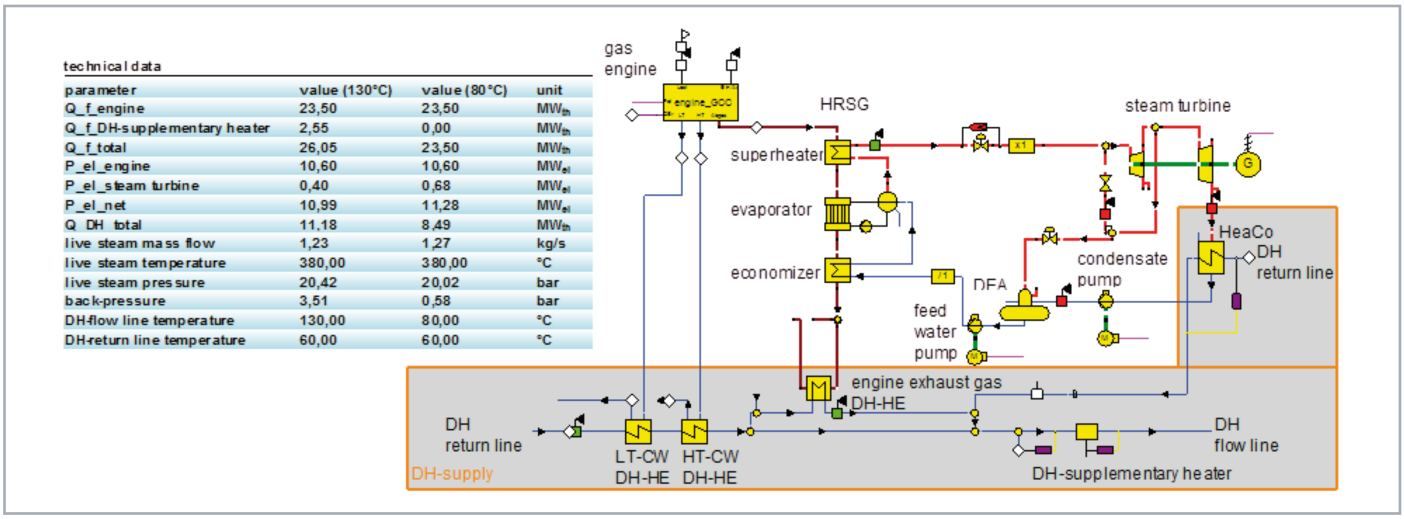
Table 1. Parameters for economic assessment.

Parameter	Unit	high price scenario (HPS)	low price scenario (LPS)
Natural gas price	€/MWh$_{th}$	21	21
Electricity price	€/MWh$_{el}$	60	28
Heat price	€/MWh$_{th}$	50	50
CHP bonus electricity	€/MWh$_{el}$	18	18
CO2 costs	€/tCO2	7.5	7.5

$$\omega = \frac{W + Q}{Q_{Br}} = \frac{\sum_{i=1}^{365}(W_i + Q_i)}{\sum_{i=1}^{365}(m_{f,i} \cdot LHV)}. \qquad (1)$$

Since heat is also partially provided from uncoupled generation from the CHP plant, for example to cover peak demand loads, this proportion must be subtracted out of the CHP evaluation. In addition to the simple fuel utilisation factor, we therefore introduce the fuel utilisation factor of the coupled production of electricity and heat in addition to equation 1, ω_{CHP}, with the modified values flowing into the evaluation in each case:

$$\omega_{CHP} = \frac{W_{CHP} + Q_{CHP}}{Q_{f,CHP}} = \frac{\sum_{i=1}^{365}(W_{CHP,i} + Q_{CHP,i})}{\sum_{i=1}^{365}(m_{f,CHP,i} \cdot LHV)}. \qquad (2)$$


technical data

parameter	value (130°C)	value (80°C)	unit
Q_f_engine	23,50	23,50	MW$_{th}$
Q_f_DH-supplementary heater	2,55	0,00	MW$_{th}$
Q_f_total	26,05	23,50	MW$_{th}$
P_el_engine	10,60	10,60	MW$_{el}$
P_el_steam turbine	0,40	0,68	MW$_{el}$
P_el_net	10,99	11,28	MW$_{el}$
Q DH total	11,18	8,49	MW$_{th}$
live steam mass flow	1,23	1,27	kg/s
live steam temperature	380,00	380,00	°C
live steam pressure	20,42	20,02	bar
back-pressure	3,51	0,58	bar
DH-flow line temperature	130,00	80,00	°C
DH-return line temperature	60,00	60,00	°C

Image 2. Process flow diagram with main process parameters of the engine (GCC)+heating condenser system.

CHP coefficient σ

The CHP coefficient σ_{CHP} of a CHP plant in equation 3 according to [5] represents the relationship between the electrical energy W_{CHP} provided and the heat Q_{CHP} provided:

$$\sigma_{CHP} = \frac{W_{CHP}}{Q_{CHP}}. \tag{3}$$

Primary Energy Savings PES

The Primary Energy Saving (PES), defined according to [5], represent the percentage fuel saving through the coupling of heat and power for a CHP application versus separate generation using suitable reference systems. Its calculation is shown in equation 4:

$$PES = \left(1 - \frac{1}{\frac{\eta_{el,CHP}}{\eta_{el,ref}} + \frac{\eta_{th,CHP}}{\eta_{th,ref}}}\right) \cdot 100\,. \tag{4}$$

For the efficiencies of the reference systems, the harmonised efficiency figures given in [4] for natural gas were used. With the separate generation of electricity, the harmonised efficiency reference value for construction years 2012-2015 is 52.5%, whereas a harmonised efficiency reference value for the separate generation of heat in the form of steam or hot water is reckoned to be 90%.

Exergetic efficiency ζ

Based on VDI 4608 Sheet 1, the exergetic efficiency ζ for ne year in daily increments is defined in equation 5 as:

$$\zeta = \frac{W_{el,net,a} + E_{DH,a}}{E_{add,f,a}} = \frac{\sum_{i=1}^{365} W_{el,net,i} + \sum_{i=1}^{365}\left(\left(\frac{T_i - T_{amb,i}}{T_i}\right)\cdot Q_i\right)}{\sum_{i=1}^{365}\left(m_{f,i}\cdot f_{ex}\cdot LHV\right)}. \tag{5}$$

With T_i being the temperature of the district heating supply, and according to equation 6 equates to the thermodynamic mean temperature of flow and return temperatures:

$$T_i = \frac{T_{fl,i} - T_{rl,i}}{\ln\left(\frac{T_{fl,i}}{T_{rl,i}}\right)}. \tag{6}$$

For methane, the fuel exergy proportion based on LHV according to [1] is 0.95. In contrast to the energetic efficiency, here only the exergetic proportion of the heat provided DH, year flows into the system evaluation, and therefore avoids the shortcomings of the energetic approach, resulting from the different qualities of the two target energy types. In addition, the heat supplied for district heating is multiplied with the applicable "Carnot factor". As well as the temperature of the provided district heating supply T_i, the mean ambient temperature Tamb also flows into the calculation. A discussion is currently ongoing within the field of CHP application regarding the convention of selecting the mean ambient temperature, which is summarised in [2]. Within this study, the mean ambient temperature of each heat requirement $T_{amb,i}$ is used for their evaluation, which in line with VDI 4608 Sheet 1.

Cost effectiveness

As well as the investment, the economic efficiency of a CHP plant is determined by the revenue situation. For this reason, an estimation of the revenue situation is also provided, with the assumptions stated in table 1 being taken as a basis. Due to the variable compensation situation for electricity, two scenarios with high price scenario (HPS) and low price scenario (LPS) were used. The HPS represents roughly the price which a municipal utility can generate for electricity distribution within its own network, whereas the LPS is based on the CHP-index of the EEX electricity exchange.

In order to get the CHP bonus, it is necessary to target an annual fuel utilisation factor of more than 80%, as well as a PES of more than 10% [5].

Results of the annual yield simulation and evaluation

In the following, an example evaluation for the three technical system configurations:

- CCPP plant,
- Engine(CHP) and
- Engine (GCC)+HeaCo

Based on the developed calculation models, an exemplary assessment was carried out for the supply of the example application with district heating and electricity based on the reference year. The plants were all operated in heat-optimised mode. For reasons of comparability, the temperature dependence of the engine and gas turbine were neglected.

Starting situation of the CCPP plant

Figure 3 shows the sorted annual profile line for the district heating demand within the supply area, as well as key performance data (district heating load, re-cooling power, net electrical power and fuel requirements) of the CCPP plant in the reference year.

The profile of the district heating demand in the reference year shows that the required district heating load lies between 142.7 MW_{th} and 14 MW_{th}. But demands of more than 120 MW_{th} only occur on 14 days in the year. Furthermore, the district heating demand profile shows that even in the midsummer, there is still a baseload demand within the district heating network.

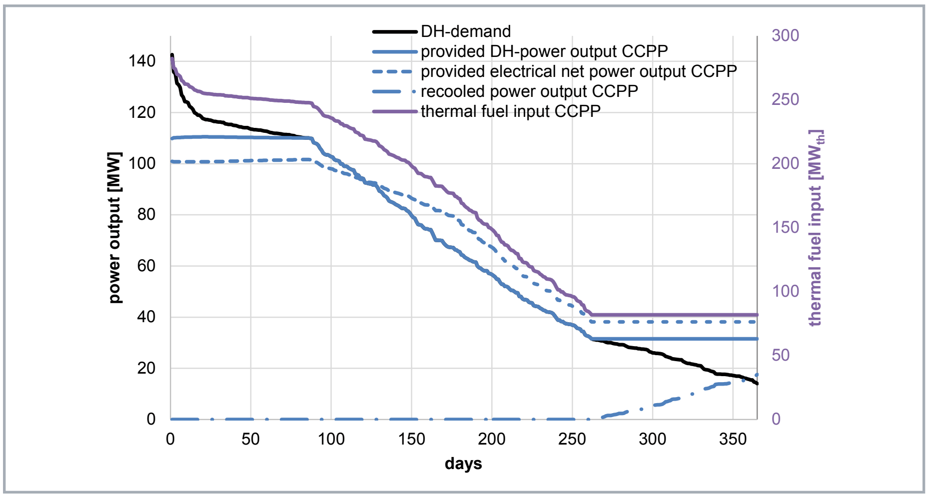


Image 3. Sorted annual profile line for the district heating demand within the supply area, as well as key performance data (district heating load, re-cooling power, net electrical power and thermal fuel power) of the CCGT plant in the reference year.

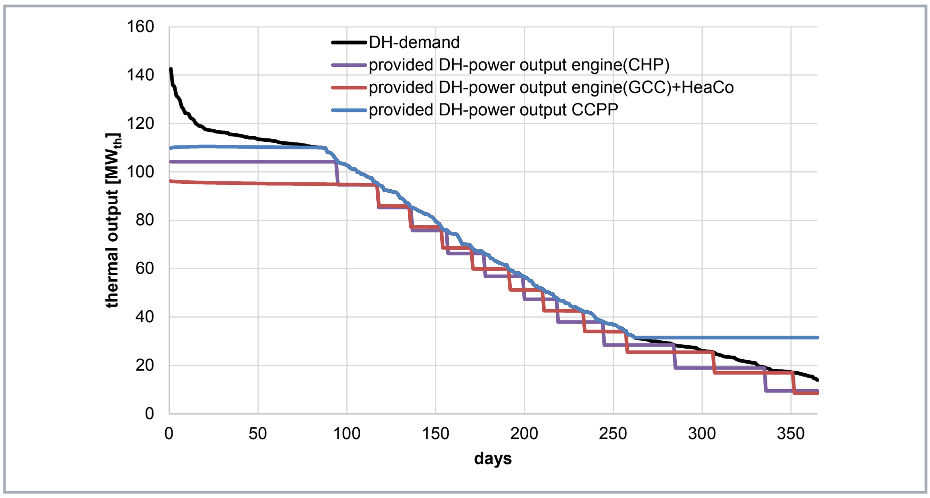


Image 4. Sorted annual profile line for the district heating demand within the supply area as well as the heat load supplied by the three system configurations (excluding auxiliary boiler) for the reference year.

The selected layout of the CCPP plant allows sensible deployment of the district heating. It is evident that the peak demand is not covered completely by the CCPP plant. Therfore peak times are covered by the supporting auxiliary boiler. This increases the full load operation of the CCPP plant. At around 110 MW_{th} the nominal district heating decoupling of the CCPP plant takes place, which is required for 87 days in the reference year. Up to this point, the auxiliary boiler is still in partial load operation and provides the residual district heating load, whereas at all other times of the year it is shut down. If the ambient temperature increases, then the district heating supply demand drops. The supplied district heating load is reduced analogously to this by the power plant controls. With the selected plant design, the minimum district heating load which can be provided from the CCPP plant is reached above a temperature of +15 °C. Since the district heating demand still continues to reduce further with the temperature increase, a re-cooling station needs to be utilised since more heat than required is decoupled from the CCPP plant. The electrical net power shows two plateaus in winter and summer, similar to the district heating supply, with around 101 MW_{el} in the winter and 38.1 MW_{el} in the summer. During the transitional period, the supplied electrical power drops continuously with increasing ambient temperature.

Alternative representation of the supply situation with gas engines and gas engine combined power plants

If the supply of a district heating network is provided using gas engines (Engine(CHP)) or gas engine combined power plants (Engine (GCC)+HeaCo) instead of the CCPP plant, then the first requirement is to create a comparable base layout. As well as the technical plant configuration (see above), the number of engines in a combined power plant provides a further degree of freedom. The goal of the study was to create a technically comparable starting point and then to vary the number of the model Engines (CHP) in order to provide a similar district heating baseload in terms of output and duration as with the reference plant. Based on a pre- study, 11 engines were therefore selected for the combined engine power plant, this is applicable for both engine systems. Furthermore, it was assumed that the engines are continuously operated at baseload and shut down in a modular manner, i.e. stepwise, in order not to exceed the required district heating demand. The auxiliary boiler provides the residual district heating load in each case.

The comparison of the three technical configurations is first carried out time-based for the reference year, using the sorted annual cycle lines for heat load (Figure 4) as well as the net electrical power output (Figure 5), before the annual yield values are finally summarised in Table 2 and the assessment is carried out based on the selected evaluation parameters.

The district heating demand as well the thermal power output provided from the CCPP plant follow the profile shown in Figure 3.

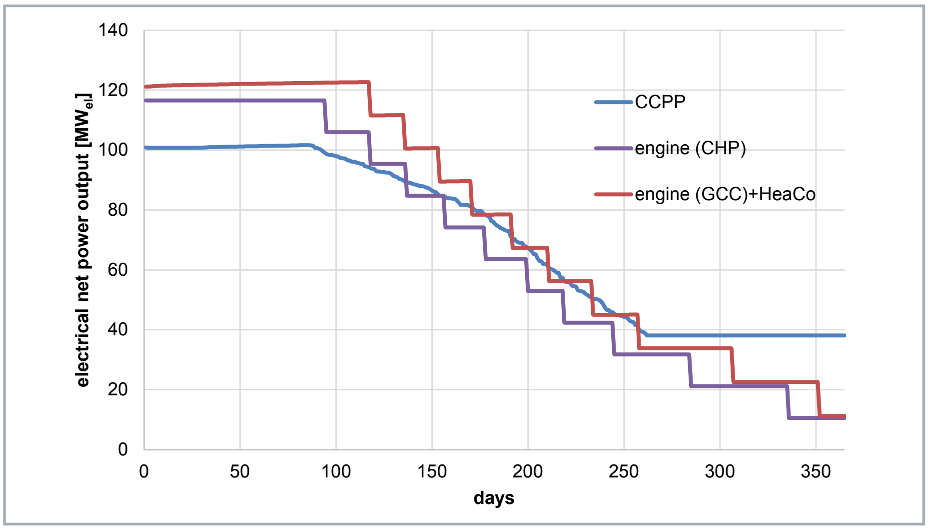


Image 5. Sorted annual profile line for the net electrical power supplied by the three system configurations in the reference year.

With the Engine(CHP) system, during coupled generation in winter, a thermal power output of 104.2 MW_{th} is provided, whereby the residual heat load is secured by the auxiliary boiler.

In the sorted annual profile line this full load period extends to 94 days, which correlates well with the 87 days full load operation of the CCPP plant. If the ambient temperature increases to above +5.6 °C, one of the 11 engines shuts down due to the reducing supply required for the district heating demand. This continues with increasing temperatures so that the typical stepped profile for modular concepts emerges. The resulting residual heat load between stepped profile and district heating load is covered in each case by the auxiliary boiler. In contrast to the CCPP plant and particularly at very low district heating loads, the modular construction of the engine combination power plant allows for better matching to the district heating demand so that at the minimum only one engine remains in operation. In addition to this stepped operating mode, it is of course also possible to provide a continuously variable combined operation of the engines. Here, with decreasing heat demand, one engine is operated at part load for example, so that no operation of the auxiliary boiler is necessary. However, in this first case study it was decided not to apply this method, and will be covered in further studies. The heat output provided by the Engine (GCC)+HeaCo model runs at altogether a lower power level than with the Engine(CHP). This is due to the fact that with the Engine (GCC)+HeaCo, part of the waste heat is converted into electricity and is therefore no longer available for heat supply. Due to the roughly constant heat output per engine, this is more noticeable at higher numbers of engines than at lower numbers. As a result of the lower heat output provided in the Engine (GCC)+HeaCo model, this enables a higher utilisation period resp.number of full load operating hours. The winter plateau with all engines in operation amounts to 118 days for the Engine (GCC)+HeaCo.

Table 2. Comparison of the yields and evaluation parameters for the three systems for the reference year.

Parameter	Unit	CCPP plant	Engine(CHP)	Engine (GCC)+HeaCo
Annual yield values				
Q	GWh_{th}	587.78	587.78	587.78
Q_{CHP}	GWh_{th}	575.45	528.51	513.57
W	GWh_{el}	617.79	591.48	666.24
W_{CHP}	GWh_{el}	617.79	591.48	666.24
Q_{Fu}	GWh_{th}	1,445.70	1,362.34	1,481.54
$Q_{Fu,CHP}$	GWh_{th}	1,432.72	1,299.96	1,403.43
Evaluation parameters				
ω_{CHP}	%	83.28	86.16	84.07
σ_{CHP}	-	1.07	1.12	1.30
PES	%	21.11	24.15	23.71
ζ	%	49.77	50.94	51.78
Revenue costs HPS	Million €	45.05	44.87	48.02
Revenue costs LPS	Million €	25.28	25.94	26.70

The provided net electrical power output from the engine systems also shows a stepped layout across the sorted annual distribution. Furthermore, it should be noted that the highest electrical net power output is provided by the Engine (GCC)+HeaCo system. The district heating supply temperature reduces with increasing ambient temperature, which is why the back-pressure reduces and the net power output increases. As a result of this behaviour, there is no plateau within the steps for Engine (GCC)+HeaCo at supply temperatures above 80 °C. For this reason, the net power output within one step increases from left to right. The maximum provided net power output during the annual cycle is 122.6 MW_{el}. To this effect, a constant electrical net power output is produced in one step with the Engine(CHP) system. During the winter plateau, the net electrical power output is 116.6 MW_{el}. The lowest maximum electrical net power output is provided by the CCPP plant, although there is also a small rise to the right (higher ambient temperature) due to the behaviour of the back-pressure turbine and heating condenser. The minimum power of the CCPP plant is substantially higher than that of the engines, which can be explained by the lower number of gas turbines. Overall, there is a power advantage for the Engine (GCC)+HeaCo system during the winter plateau, lying at 5.1% compared to the Engine (GCC)+HeaCo system and 20.7% compared to the CCPP plant.

All three systems provide the required heat load of 587.78 GWh_{th} demanded by the district heating network, with the highest heat component being delivered in CHP by the CCPP plant with 97.9%, excluding the re-cooling power. This high component is primarily due to the auxiliary firing of the waste heat boiler, which was not used with the engine systems. Furthermore, the maximum CHP heat output of the engine models is always below that of the CCPP plant, as already shown in Figure 4. During electricity production, it can be observed that the entire electricity production is descended from CHP. The greatest quantity of electricity is provided by the Engine (GCC)+HeaCo system at 666.24 GWh_{el}, which represents an increased yield of 7.8% compared with the CCPP plant, and 12.6% compared with the Engine (CHP) system. The picture for fuel consumption is very similar to that for electricity production. In this case, the highest demand also lies with Engine (GCC)+HeaCo, followed by CCPP plant and then the Engine(CHP).

In order to carry out a comprehensive evaluation of the systems based on the yield values stated, the evaluation parameters introduced previously were used. The highest CHP fuel utilisation factor is shown by the Engine (CHP) system with 86.16%, but with both engine systems comparing favourably with the CCPP plant. Furthermore, all three systems are above the evaluation benchmark of 80% stated in the CHP regulations for the CHP bonus. The highest CHP coefficient is offered by the Engine (GCC)+HeaCo system, at 1.3, followed by the Engine(CHP) with 1.12 then the CCPP plant with 1.07. As a result, the Engine (GCC)+HeaCo system is especially interesting from the point of view of the CHP bonus, since it provides a favourable ratio of electrical energy to heat. To take into account the different thermodynamic qualities of heat and electrical energy, an evaluation of the primary energy saving as well as the exergetic utilisation factor were also used. The Engine(CHP) has the highest primary energy saving with 24.15%, followed by the Engine (GCC)+HeaCo with 23.71% and the CCPP plant with 21.11%. Thus all three systems lie above the threshold value of 10% required in [5]. Due to its higher electricity production, the Engine (GCC)+HeaCo system has the highest exergetic utilisation factor of 51.78%. The CCPP plant has the lowest exergetic utilisation factor with 49.77%.

However, all systems must also enable efficient economic operation. The thermodynamic evaluation factors for system efficiency have shown that all plants are eligible under the auspices of the CHP regulations. Based on the project specific costs and revenues, a cost-revenue calculation was carried out. A positive revenue was shown for all three systems, with the highest amount shown with the Engine (GCC)+HeaCo under both scenarios (HPS and LPS).

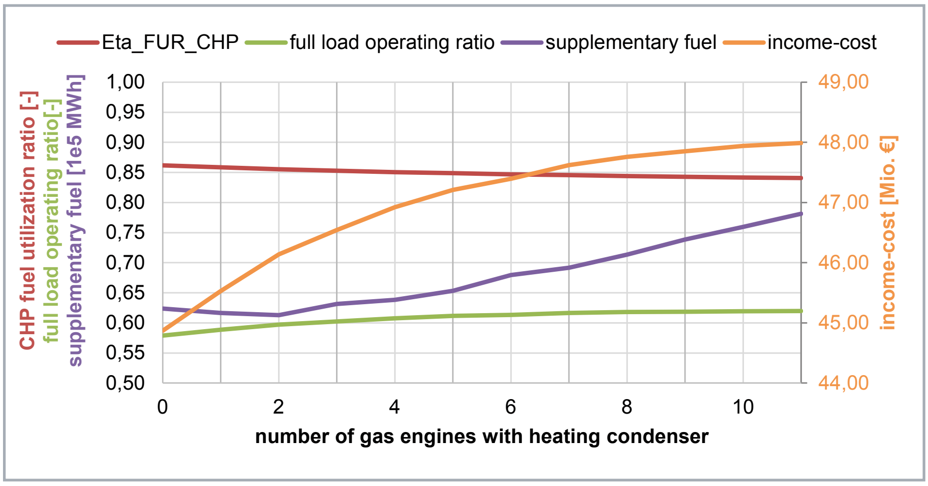


Image 6. Profile of fuel utilisation factor, fuel load amount, additional fuel requirements and revenue costs of the combined power plant consisting of 11 engines and variable number of engine (CHP) and engine (GCC)+heating condenser modules in the reference year.

In conclusion, it should therefore be noted that the engine combination systems reviewed in the chosen application may have advantages in terms of energy efficiency and economic efficiency compared with the CCPP reference plant.

Economically optimised application of the heating condenser module

For the Engine (GCC)+HeaCo system are due to the modular concept, in most cases only a part of the water-/ steam cycle modules coupled to the engines in operation. Fundamentally, however, the investment amount would be equally high even if they were operated during the whole year. A higher utilisation period of the coupled water-/ steam cycles can be achieved if not all of the engines are equipped with the waste heat capture. In this case, the engines with waste heat capture are used for district heating baseload, so that these engines and the downstream processes will have substantially higher full load hours. Ultimately, this results in a better relationship between additional revenue and additional investment. Based on this approach, a sensitivity analysis was carried out for the application being considered here. Starting with a combined power plant comprising only Engine(CHP) modules, the number of modules in the Engine (GCC)+HeaCo configuration was then increased step by step until the combined power plant was created is consisting only out of Engine (GCC)+HeaCo modules. In the intermediate steps, the combined power plant consisted of a mixture of Engine(CHP) modules and Engine (GCC)+HeaCo modules. The results of the sensitivity analysis are shown in Figures 6 and 7.

At the intersection with the Y-axes, the combined power plant consists completely out of Engine(CHP) modules (primary Y-axis) and Engine (GCC)+HeaCo modules (secondary Y-axis). Therefore their values correspond with the values in Table 2. If, starting from the Engine (CHP), the number of Engine (GCC)+HeaCo modules in the combination power plant increases, the full load operating proportion as well as the revenues also increase, whereas the CHP fuel utilisation factor drops. The fuel demand can initially be slightly reduced, being at its minimum with 2 Engine (GCC)+HeaCo modules, before it increases again with the increasing number of Engine (GCC)+HeaCo modules. For the investment decision however, the key factor is the possible advantage from the heating condenser application in the cost-revenue calculation based on the installed capacity of the steam turbine (ST), as well as the pricing basis for the steam turbine of the applicable power class. Both of these parameters are therefore shown additionally in Figure 7.

Due to the modular illustration selected, the installed ST power increases linearly with the number of Engine (GCC)+HeaCo modules in the combined power plant. It is possbile to draw from the installed ST capacity conclusions regarding the investment needs. Fundamentally, the specific investment for a steam turbine decreases with increasing ST capacity. According to this, it would be advantageous to connect multiple engine modules to one larger steam turbine of a higher power class. For exact analysis of the ST investment, manufacturer-specific information need to be used, which was not done in this study for reasons of comparability.

heating load several days in advance, the number of modules of the combined power plant which would be available for residual load management can always be identified. The gas engine used for this study, enables from standstill a fast start-up to full load within 180 s [6], which represents a load gradient of 3.5 MW_{el}/min. Based on the installed power, this means 33% $MW_{el,inst}$/Min.

In comparison, modern CCPP plants such as the Siemens H-class currently offer higher load gradients of up to 16.6 MW_{el}/min [7]. Based on the installed power, however, this only equates to 3% $MW_{el,inst}$/Min. As a result, a combined engine power plant has a clear advantage in this respect. Furthermore, it must be pointed out that for engines there is no lifetime consumption related with starting cycles and have even lower starting costs. For this reason, from the point of view of flexibility, as well as for example stated in [8], gas engines are a suitable element in a future German energy supply system.

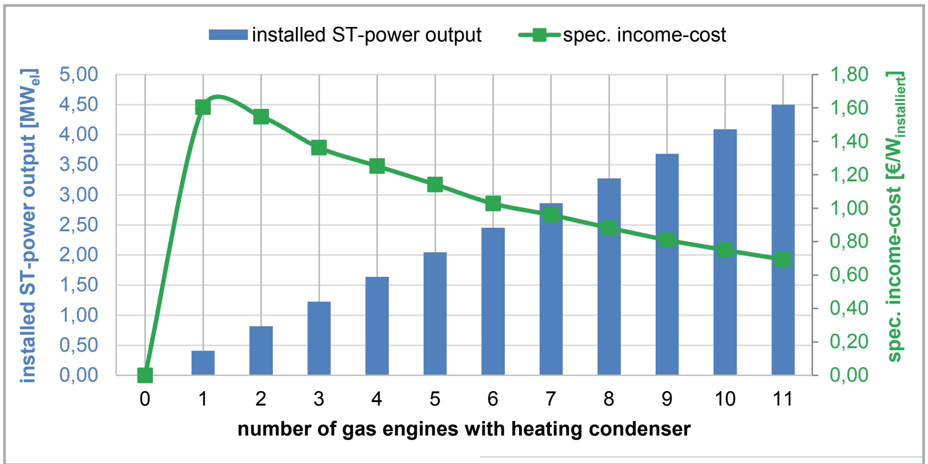


Image 7. Profile of installed ST power and specific revenue costs with variable number of engine (CHP) and engine (GCC)+heating condenser module in a combined power plant consisting of 11 engines in the reference year.

The specific revenue costs form a curve with a maximum, which represents the economic optimum in this respect and lies at one Engine (GCC)+HeaCo module. Due to the low installed power and the likely high specific investment costs, we must however assume that the economic optimum is to be found at a higher number of Engine (GCC)+HeaCo modules using a common steam turbine. For this reason, it is worthful to carry out a final comparison of ST investment costs with specific revenue costs.

Outlook – flexibility and participation in the balancing energy market

As well as ensuring the supply of an own municipal power network, engine combined power plants can be used for grid supportdue to their high flexibility and modular construction, e.g. in order to supply residual load. Additional revenues can be generated from this, which substantially increase the economic efficiency of an engine combined power plant. Due to the good planning capability of the district

Summary

In this article, using a representative example application in public district heat supply, an alternative application of engine combined power plants instead of a typical CCPP plant was investigated in order to illustrate the supply during heat-operation. As well as a classic CHP engine application, an engine-module configuration was developed with a downstream located water-/ steam cycle, including back-pressure turbine and heating condenser. For all three analysed systems, the heat supply was secured and all investigated systems fulfil the requirements for support under the CHP regulations. With regard to energy efficiency and economic efficiency, engine combined power plants offer advantages compared to the CCPP reference plant. When configuring such an engine combined power plant, a mix of engines in simple CHP operation together with modules with downstream coupled water-/ steam cycle appear to be economically attractive. When considering the future changes in the German energy supply system, it must also be pointed out that, due to their modular operation in CHP applications, these highly efficient gas engines will be due to their high specific load gradients in a position to participate also in the balancing energy market.

Acknowledgements

The investigations were carried out under the auspices of the joint project TURIKON, which was financially supported by the federal state of North Rhine-Westphalia as well as the European Fund For Regional Development within the progress. NRW and the Ziel 2 programme 2007-2013, Phase VI (funding code: 64.65.69-EN-2019).

This text was first published in German language in VGB PowerTech Journal, edition 03/2016.

List of abbreviations

Amb	Ambient
CCPP	Combined Cycle Gas Turbine Power Plant
CHP	Combined Heat and Power
CW	Cooling Water
DH	District Heating
el	electrical
FL	Flow Line
Eq.	Equation
f	Fuel
DEA	Deareator
GCC	Gas Combined Cycle
HE	Heat Exchanger
HeaCo	Heating condenser
HPS	High Price Scenario
HT	High Temperature
inst	installed
LHV	Lower Heating Value
LP	Low Pressure
LPS	Low Price Scenario
LT	Low Temperature
IP	Intermediate Pressure
PES	Primary Energy Savings
ref	Reference
RL	Return Line
ST	Steam Turbine
th	thermal
HRSG	Heat recovery steam generator

Equation symbols

P	electrical power
Q	thermal power
Q	thermal energy

T	temperature
W	electrical energy
ζ	exergetic efficiency
η	efficiency
σ	CHP coefficient
ω	fuel utilisation factor

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