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An approach to enhance combined cycle performance by integration with a gas pressure reduction station

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Abstract

A novel system enhancing the performance of a combined cycle gas turbine has been proposed in the paper, based on the example of the thermal-electric power station currently under construction in Włocławek. Analysis shows that annual average electrical efficiency rises by 0.6% and that power output rises by up to 4%, depending on the ambient conditions. GE's GateCycle software was used in the modeling.

Keywords: Expander, Combined cycle, Gas turbine, Inlet air cooling, Fertilizer plant

1. Introduction

Combined cycle plants are currently the leading technology for fossil fuel conversion due to their high efficiency and the large number of units installed worldwide. Attempts to exceed the threshold of 60% efficiency were made in the late 20^{th} century [1, 2]. Finally, 60.43% electric efficiency was reached in the Irsching Power 4 power unit in Ingolstadt, Germany [3]. In Poland, due to the dominant role of coal, only a minor part of the total installed power is based on combined cycle (CC) units. At the time of writing, the two largest ones are the combined heat and power (CHP) plants at Lublin-Wrotków and Zielona Góra, rated at 235 and 198 MWe, respectively. However, this is set to change by the end of 2015 as the major Polish oil refiner PKN Orlen has started construction of the Włocławek CC plant, which will be located in the direct vicinity of the Anwil fertilizer plant, which also belongs to the capital group of PKN Orlen. Two power units, each one rated 463 MWe, should totally cover the demand of the fertilizer plant, and supply electricity to the local market [4]. In December 2012 a consortium of General Electric International Inc. and SNC-LAVALIN POLSKA sp. z o.o. was selected to construct the plant.

The present work proposes a modification to one of the plant's units by adding an inlet air cooling system. The system was investigated inter alia by Pyzik et al. [5] who proved there was a significant power gain due to cooling. The system proposed in this paper involved the supply of cold from expanded natural gas (NG). The aim of the work is to evaluate the technical performance of the system. A similar solution was also proposed by Farzaneh-Gord and Deymi-Dashtebayaz, 2009 [6]. In the present paper, additional attention has been paid to the security of operation under low ambient conditions. The presented solution assumes that the required cold is generated in the process of NG expansion, carried out in a turbine or piston expander, and that NG is supplied from the high-pressure Yamal Pipeline located merely 2.5 km from the CHP construction site.

The reference plant for the system was chosen prior to the beginning of the construction works in the Włocławek plant. The proposed cycle was configured for efficiency and analyzed over a one year period of operation with variant ambient air conditions and NG temperature.

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Table 1: Nominal parameters of the considered gas turbines

Parameter	Unit	ABB GT26	GE 9FA	MHI 701F	SGT5-4000F
Power output	MWe	234.2	255.6	270.0	250.8
Electric efficiency	%	37.1	36.9	38.2	38.1
Compression ratio	–	30.0	15.3	17.0	17.0
Exhaust temperature	°C	615.0	602.0	589.3	580.3
Exhaust mass flow	kg/s	538.5	643.9	–	639.7
Air mass flow	kg/s	–	–	650.0	–

2. Methodology

The proposed solution is compared to a standard CC unit. The first part contains a description and the configuration method for the reference cycle, along with the input data for the analysis. Subsequently, the enhanced cycle is described.

2.1. Reference cycle

The reference CC plant consists of a GT and a three pressure level preheated steam cycle, operating at constant power under baseload conditions, no agent losses, with a pressure loss in the gas turbine of 1 kPa and in the heat recovery steam generator (HRSG) of 2.49 kPa.

The gas turbine output power was chosen based on the total CC output, in 2011 estimated at 400–500 MWe. Currently, it is known that the CC unit was designed for a power output 463 MWe, but before this information was published the authors had assumed desired power of 400 MWe. Given that in a typical CC plant the GT/ST power ratio is about 1.5, the desired GT output is about 266 MWe. Currently, this power output of GT in F-class is offered by four main manufacturers: Alstom (ABB GT26), General Electric (GE 9FA), Mitsubishi (MHI 701F) and Siemens (SGT5-4000F v94.3a) (see Table 1). F-class turbines were chosen due to the availability of their load performance.

The Mitsubishi MHI 701F turbine was chosen for the analysis. This choice is justified by the highest efficiency and the best fit of the power output to the requirements. The model assumes constant TIT (Turbine Inlet Temperature) at 1454°C, which is controlled by adjusting the flow rate of fuel.

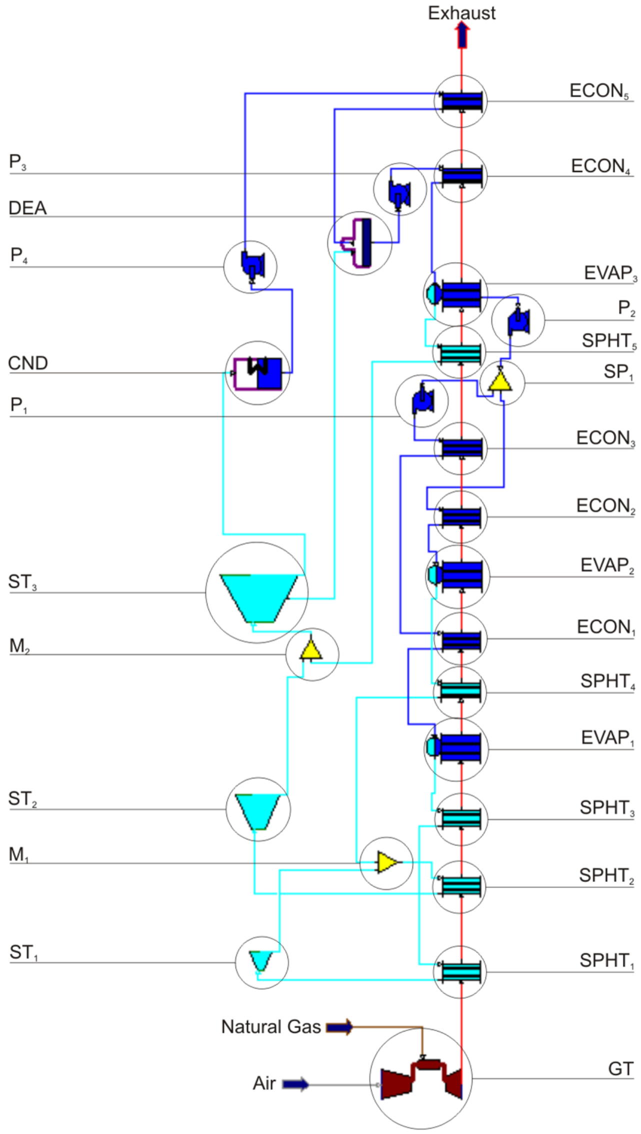

Figure 1: Configuration of the reference combined cycle

The modeled steam cycle consists of a three-staged extracting-condensing steam turbine (ST), a feedwater pump and three circulation water pumps, deaerator (DEA), condenser (CND), five sections of economizer (ECON), three sections of evaporator (EVAP) and five superheater sections (SPHT), including one reheater. The design parameters for each of the listed elements are set out in Appendix Appendix A and concern the optimized system. The reference plant is presented in Fig. 1.

Cross-counter-current heat exchangers with one tube row per pass and 10 passes were assumed for the economizers and superheaters. Inner flow is mixed while outer flow is unmixed. The heat exchange area in the superheaters is calculated based on the Hot End temperature difference. The heat exchange area in the economizers is

Table 2: Molar gas composition, %

Component	PSG	NIST-2
Methane	96.555	90.644
Ethane	1.540	4.553
Propane	0.436	0.833
Isobutane	0.069	0.100
n-Butane	0.085	0.156
Isopentane	0.022	0.030
n-Pentane	0.016	0.045
n-Hexane	0.018	0.040
Nitrogen	0.950	3.134
Carbon Dioxide	0.310	0.466
Molar mass, kmol/kg	16.18	17.61

calculated based on the Approach Point temperature difference. The calculations of heat exchangers were made using the ε-NTU method, broadly described inter alia by H. A. Navarro and L.A. Cabezas-Gómez [7].

Cycle efficiency is defined as follows:

$$\eta_{\mathrm{gp}} = \frac{N_{CC}^{net}}{\dot{m}_{\mathrm{f}} LHV} \tag{1}$$

where η_{gp} is the efficiency of power generation in CC, N_{CC}^{net} is net power output of whole cycle, $\dot{m}_f$ is fuel flow and LHV fuel's lower heating value.

Power output is a net value, as it is reduced by its own demands, mostly by the demand of the water pumps, as follows:

$$N_{gp} = N_{TG} + \eta_{g-m} \sum_{i=1}^{3} N_{STi} - \sum_{i=1}^{4} N_{Pi} \tag{2}$$

where η_{g-m} is the mechanical and power generation efficiency of the steam turbine, N_{TG} is the power output of the gas turbine, N_{STi} is the power output of the i-th part of the steam turbine, N_{Pi} is the power demand of the i-th pump.

In equation 1 LHV is 48 346 kJ/kg and was calculated based on NG composition. Table 2 represents the assumed gas composition corresponding to a sample measurement by the Polish Gas Distribution Company (PSG), and for comparison the composition of reference NG proposed by the National Institute of Standards and Technology (NIST-2) [8]. For the design conditions of CC (ambient air temperature: t = 15°C; ambient air pressure p = 101,325 kPa; ambient air relative humidity φ = 60%) the fuel demand of 14.56 kg/s (37 827 m_n^3/h) was calculated. The PSG composition was applied for the calculations.

Thermodynamic optimization of CC due to power output and cycle efficiency requires multiple parameters to be

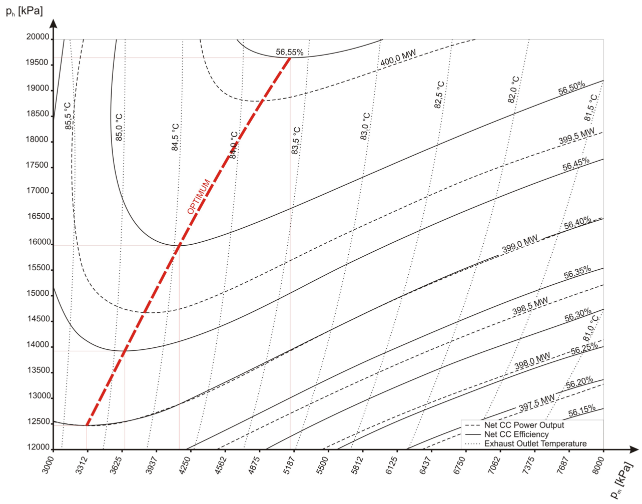

Figure 2: Map of: electrical efficiency, power output, exhaust temperature of combined cycle due to medium and high pressure. Low pressure fixed at 300 kPa

controlled, such as maximum steam temperature, deaerator's operating pressure, pressure levels in the steam cycle, pinch point temperature difference for evaporators, exhaust gas outlet temperature, pressure level for steam extraction from the turbine while each parameter affects the optimal level of the remaining ones. This problem was investigated inter alia by Bassily [9] using the example of a two-pressure-level cycle. In the present paper, the optimization is limited to finding the optimum value of efficiency in terms of the three pressure levels in the steam cycle.

The optimization begins with the determination of the parametric map showing efficiency in terms of medium and high pressure levels, at a fixed value of low pressure. High pressure varies between 12000 kPa and 20000 kPa with a discretization step of 40 kPa, while medium pressure varies between 3000 kPa and 8000 kPa with a step of 25 kPa. Low pressure was assumed at 300 kPa, as a result of preliminary calculations.

An original, gradient-based approach was used. The high, medium and low pressure levels are further referred to as PH, PM, PL. The optimization was organized into the following steps:

Step 1: Finding the minimum values of PH and PM

The objective of the step is to analyze the efficiency map in terms of PH and PM (Fig. 2) and to find points corresponding to minimum values of PH and PM allowing one to achieve a given level of efficiency:

$$\underset{\eta = const}{\wedge} \quad p_H \to min \quad \wedge \quad p_M \to min \tag{3}$$

The pairs of PH and PM which are optimal for each level of efficiency are depicted with the red line crossing the isolines of efficiency on PH vs. PM map on Fig. 2. Minimum values of PH and PM are considered optimal, as it can be supposed that this solution would attract the lowest investment cost.

Step 2: Searching for optimal PL

Once Step 1 is carried out, pairs of PH and PM are tabularized along with the corresponding efficiencies and subsequently for each table row the impact of PL on efficiency is monitored (PL is varied from 300 kPa to 1000 kPa). The PL giving the highest efficiency is chosen for further optimization, as illustrated in Table 3.

Step 3: Searching for optimal PM

In the third step, the optimum value of PL determined in Step 2 for each row is treated as a constant, and PM is varied in the same way (but for an unchanged range) as PL was in Step 2. As a result, new optimum values of PM are found for each level of efficiency (according to the adjusted values of PL). PH remains constant, as in Step 2.

Subsequent steps:

Efficiency improvement

The subsequent steps are similar to the previous one, where two pressure levels are considered constant, and the third one is varied to find maximum efficiency. For each step a different pressure level (PH or PM or PL) is varied. The iterative procedure of optimization is repeated until the increment of efficiency is below the display resolution of 0.001%.

As a result of the analysis it was observed that efficiency is a monotonous function of PH. Hence, in this case the optimum value of PH cannot be found using thermodynamic criteria alone. Therefore, PH was set at 20 000 kPa and the optimum values of PM and PL were found using a parametric grid search, depicted on Fig. 3, to find the optimum:

$$\eta\,(P_M, P_L) \to \max \; for\, P_H = const. \tag{4}$$

The configured cycle has rated nominal power output of 396.70 MWe, electric efficiency of 56.36% and exhaust outlet temperature of 92.3°C. The chosen high,

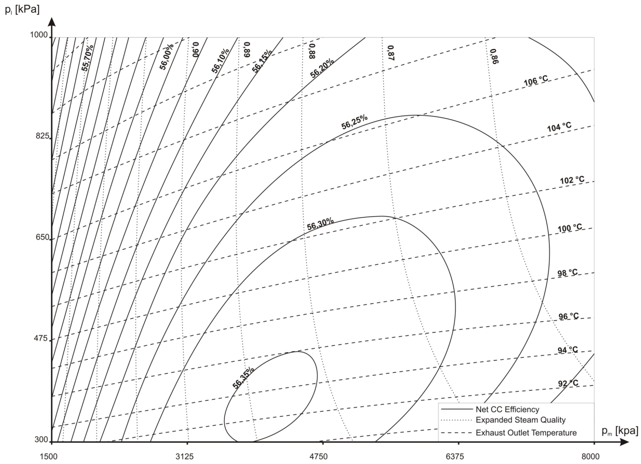

Figure 3: Map of: electric efficiency, exhaust outlet temperature, steam quality at the outlet of steam turbine due to changes of medium and low pressure level. High pressure fixed at 20000 kPa

medium and low pressure levels are 20000 kPa, 4100 kPa and 360 kPa, respectively. The cycle is also presented in Fig. 4 in a temperature-entropy (T-s) diagram.

In order to evaluate transient output parameters of the combined cycle over a one year period, ambient air temperature and relative humidity were adapted for the purpose of calculations. The data is available in the Polish Ministry of Transport, Construction and Maritime Economy [9]. Data on variable NG temperature were acquired through the courtesy of the Polish Gas Distribution Company. The input data are presented altogether in Fig. 5.

It should be noted that due to the lack of direct data, the ambient air parameters for the city of Włocławek were assumed to be the arithmetic average between the nearest available stations (Płock and Toruń, both approx. 50 km away), since Włocławek is located approximately in the middle of the line connecting both cities, and at a comparable height of about 90 m above sea level. The data covers the variation in average ambient air parameters over the year with one hour step accuracy.

The range of the data varies between 35% and 100% for air relative humidity, -15.5°C to 32.2°C for ambient air temperature and 0.5°C to 24.0°C for NG temperature.

2.2. Enhanced cycle

NG is supplied to the cycle under high pressure (before reduction) and its pressure reduced on-site (within the area of the plant). The reduction is divided into two steps. The first one results in a pressure drop to the level required for the gas turbine fuel inlet. The flux of gas is divided into one part feeding the gas turbine and the other feeding the fertilizer plant. The second reduction step is

Table 3: Steam cycle optimization

Constants determined in previous step			Varied	Objective function
$\eta_1(P_{H,1}, P_{M,1})$	$P_{H,1}$	$P_{M,1}$	$P_{L,1}$	$\eta_1 \rightarrow max$
$\eta_2(P_{H,2}, P_{M,2})$	$P_{H,2}$	$P_{M,2}$	$P_{L,2}$	$\eta_2 \rightarrow max$
...	...	...	...	...
$\eta_n(P_{H,n}, P_{M,n})$	$P_{H,n}$	$P_{M,n}$	$P_{L,n}$	$\eta_n \rightarrow max$

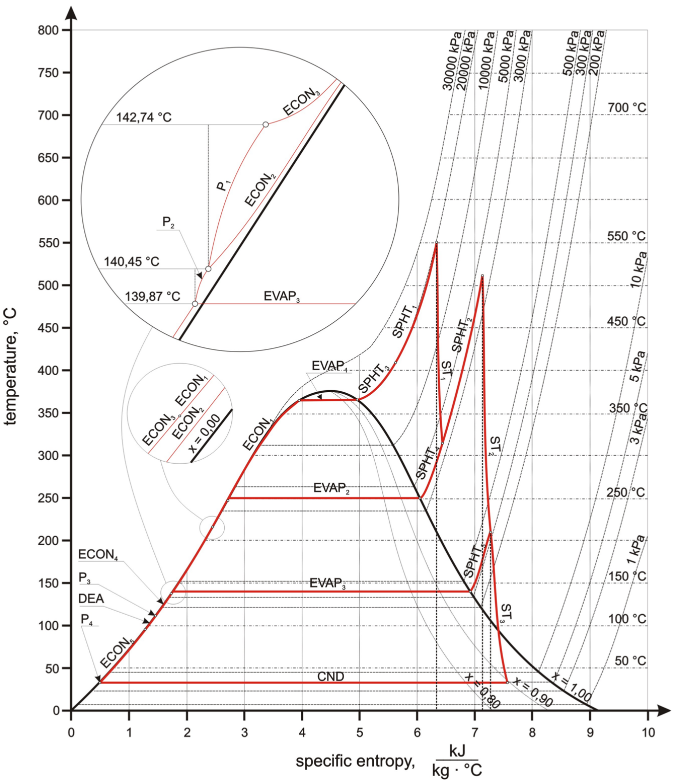

Figure 4: Steam cycle on a temperature-specific entropy chart

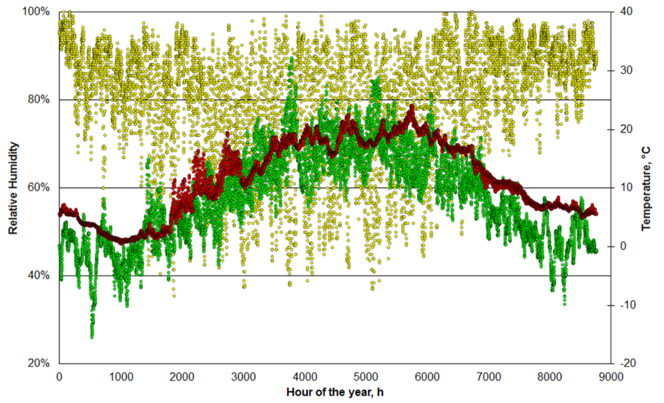

Figure 5: Input parameters: green—ambient air temperature, red—NG temperature, yellow—air relative humidity [10]

only applied to the part of the flux supplied to the fertilizer plant.

In the present study, the high pressure of NG has been assumed at the level of 5000 kPa, which corresponds to the operating pressure in the Yamal gas pipeline. Gas pressure after the reduction results from the GT compression ratio of 17, which means that air is compressed to 1700 kPa. In order to enable fuel injection to the gas turbine, NG pressure should be higher, for the analysis the value of 2000 kPa was assumed. NG pressure after the second step of reduction is 800 kPa, which is typical for large scale NG consumers.

The cycle is equipped additionally with NG preheaters installed upstream of the expanders, working under variable load and designed to maintain constant temperature at the NG expansion outlet. Preheaters are fed with exhaust heat. During the analysis exhaust temperature is monitored, as it should not drop below the exhaust gas dew point. The enhanced cycle is presented in Fig. 6.

NG expansion with limited preheating leads on the one hand to a reduction in the power extracted, but on the other hand it provides a source of cold. The process is technologically challenging and most of the expanders available on the market cannot handle it (the difficulties involved are described later on in the paper). Nonetheless, cryogenics companies like Cryostar [11] offer expanders adapted for such conditions, which makes practical implementation possible.

The cold generated is used to decrease the temperature of GT inlet air. This solution reduces the power demand of the GT compressor due to the reduced specific volume of air. As the ratio of natural-gas-to-air flow is low (~1:50), the decrement in air temperature is small. However due to the high sensitivity of GT to ambient conditions, this slight air temperature decrement generates a noticeable improvement in performance.

Next, integration with a large NG consumer—here a fertilizer plant—is considered. The aim of integration with the fertilizer plant is to increase the mass flow of NG flowing through the expander in order to boost the chilling effect of air supplying the gas turbine (improving the poor gas-to-air mass flow ratio). Based on annual production levels in the Anwil fertilizer plant [12], an estimate of NG

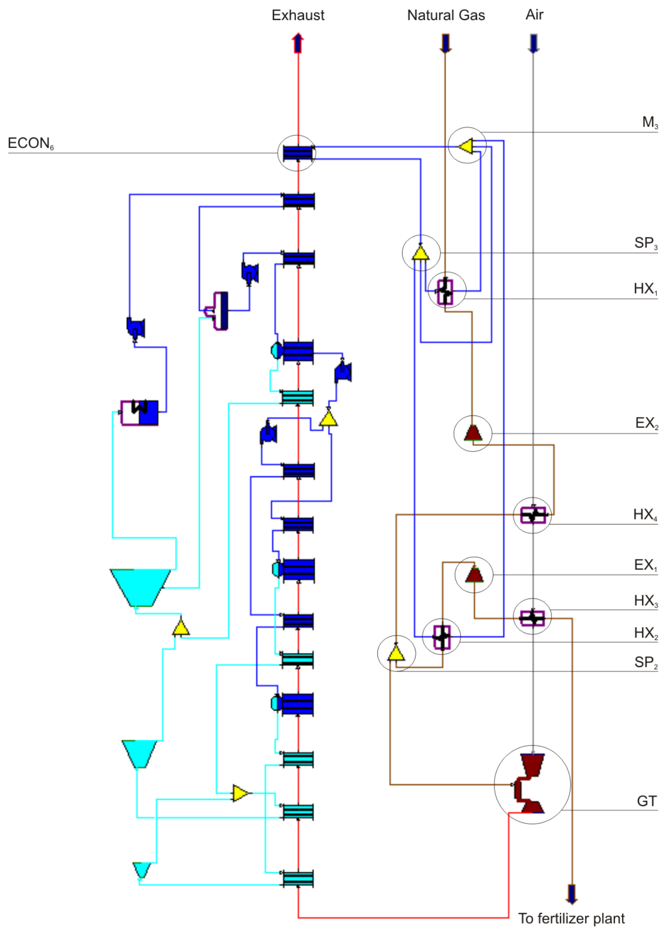

Figure 6: Enhanced combined cycle configuration

demand was made, assuming consumption as for a typical production process for several chosen products [13]. The results are converted into momentary flow over a one year period and presented in Table 4.

Total momentary gas demand is rated at 20.19 kg/s. For calculation purposes, demand was assumed at 10.00 kg/s as the analysis is applied to one of two identical power trains.

One must bear in mind that pressure reduction of NG at low temperatures is technically challenging, inter alia due to possible liquefaction of heavier hydrocarbons. The risk is determined additionally by gas composition and pressure. Since the content of heavier hydrocarbons in the analyzed NG is very low, Dalton's law was applied to find the liquefaction temperatures of particular components at partial pressure. The saturation temperature was found using a freely available database of chemical components [14]. The results are set out in Tables 5 and 6. It was assumed that the highest obtained temperature (here: hexane) approximately corresponds to the actual hydro-

Table 4: Estimated NG consumption by a fertilizer plant

Product	Production Mg/year [12]	NG demand kg/Mg product [13]	NG demand kg/s
Ammonia	475 000	785.6	11.83
Ammonium nitrate	803 000	204.4	5.20
Caustic soda	150 000	327.0	1.56
Vinyl chloride	205 000	245.2	1.59

Table 5: Saturation temperatures of NG components, gas pressure: 800 kPa

Component	Partial pressure, kPa	Saturation temperature, °C
Propane	3.488	-97.7
Butane	1.232	-78.7
Pentane	0.304	-67.8
Hexane	0.144	-52.7

carbon dew point temperature.

Another issue is hydrate formation. Gas hydrate is a solid resembling snow, which can form above the freezing point of water. The essential conditions for gas hydrate formation are: the specific combination of temperature and pressure, sufficient moisture (not necessarily in a liquid phase) and the presence of hydrate former, inter alia methane, ethane [15]. Dry NG is assumed for analysis, and its lowest temperature after each expansion level is -20°C.

Heat exchangers, where the cold expanded NG chills the air supplying the GT is at risk of ice accretion from the air side. Due to the large flow of air compared to the flow of gas, the risk arises when the ambient air temperature approaches 0°C. In the model it is assumed that if the air temperature is lower than or equal to 4°C, NG does not chill the air and bypasses the heat exchangers. In order to avoid supplying the GT and the fertilizer plant with NG at excessively low temperature, NG is additionally pre-

Table 6: Saturation temperatures of NG components, gas pressure: 5000 kPa

Component	Partial pressure, kPa	Saturation temperature, °C
Propane	21.800	-71.8
Butane	7.700	-53.4
Pentane	1.900	-45.0
Hexane	0.900	-30.5

Table 7: Cycle configuration due to ambient air temperature

Ambient temp.	Gas temp. after reduction	Heating before 1° reduction	Heating before 2° reduction	Air chilling
<4°C	5°C	Active	Active	Inactive
4°C ÷ 27°C	-20°C	Active	Active	Active
27°C <	-20°C	Active	Inactive	Active

heated before expansion in heat exchangers designed to control the NG temperature at the outlet of expanders. In such conditions it is assumed that NG temperature after expansion must not drop below 5°C.

This mode of operation means that air is chilled only for part of the year, mainly in summer. Table 7 presents the operating schedule of the heat exchangers in terms of the ambient air temperature.

The combined cycle power output including expanders is defined as follows:

$$N_{gp} = N_{TG} + \eta_{g-m} \cdot \sum_{i=1}^{3} N_{STi} + \sum_{j=1}^{2} N_{EXj} - \sum_{i=1}^{4} N_{Pi} \quad (5)$$

where N_{EXj} is power output of j-th NG expander.

The behavior of the cycle is evaluated assuming the same boundary conditions (ambient temperature, humidity, NG temperature) as it was for the reference cycle. The observed parameters are: net power output, momentary electrical efficiency, exhaust gas outlet temperature, air temperature before and after chilling, momentary fuel demand.

3. Results

3.1. Reference cycle

Basing on the input data, the operation of the cycle in a one year period was simulated using the created model. The monitored parameters were: net power output, NG consumption, electrical cycle efficiency and exhaust gas outlet temperature. Power output and fuel consumption are presented in Fig. 7. It can be seen that in winter power output and fuel consumption increase due to the increased mass flow rate of air. The presented analysis serves to evaluate the chilling effect of the enhanced cycle.

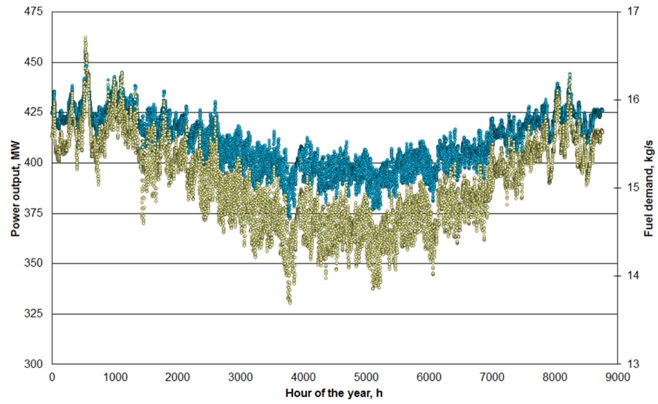

Figure 7: Plant operation with respect to the input data. Blue: net power output, yellow: fuel consumption rate

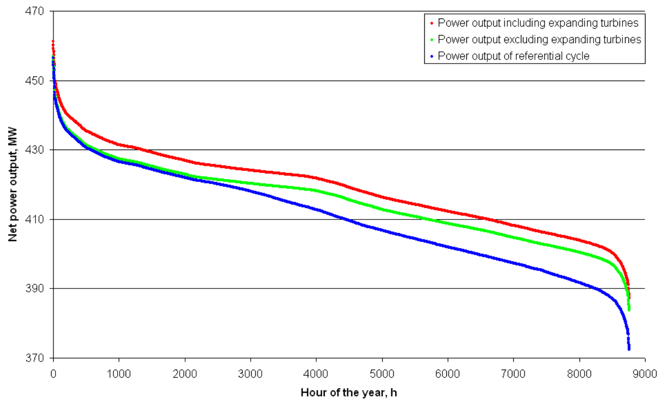

Figure 8: Load curve of the extended CC net power output including expanding turbines (red line), excluding expanding turbines (green line) and the reference CC net power output (blue line) in respect of the operational period

Additionally, a sensitivity analysis was performed which demonstrated that NG inlet temperature has negligible impact on the monitored parameters. However, it is predicted that this temperature influences the proposed chilling system. In order to maintain input data consistency, NG temperature is also included in the input data of the reference cycle.

3.2. Enhanced cycle

The same input data as for the reference cycle analysis were applied. The results are rearranged in a structured diagram. The diagram allows one to read the number of hours over the year for which a corresponding parameter takes an equal or greater value. Fig. 8 shows the net power output curves during the year while Fig. 9 depicts the effectiveness of air cooling during the year. Additionally, Table 8 shows a clear comparison of key parameters between the reference and the enhanced cycle for three sets

Table 8: Comparison of the reference and the enhanced cycle

	Parameter	Ambient air temperature Lowest	Design	Top
Ambient conditions (Fig. 5)	Hour of the year	532	6 229	3 784
	Ambient air temperature, °C	-15.5	15.0	32.2
	Air relative humidity, %	81.5	61.0	44.5
	NG temperature, °C	3.7	17.8	18.1
Reference cycle (Fig. 6)	Net power output, MW	456.4	397.9	372.4
	Momentary cycle eff., %	Average: 56.352		
	NG consumption, kg/s	16.712	14.610	13.696
	Exhaust outlet temperature, °C	95.8	93.6	92.7
	Air temp. for GT, °C	-15.5	15.0	32.2
Extended cycle (Fig. 8, Fig. 9)	Net power output (incl. exp.), MW	461.2	410.0	387.0
	Net power output (excl. exp.), MW	456.8	406.5	383.7
	Momentary cycle eff. (incl. exp.), %	Average 56.982		
	Momentary cycle eff (excl. exp.), %	Average 56.490		
	NG consumption, kg/s	16.712	14.879	14.042
	Exhaust outlet temperature, °C	88.1	92.1	92.1
	Air temp. for GT, °C	-15.5	10.9	25.9

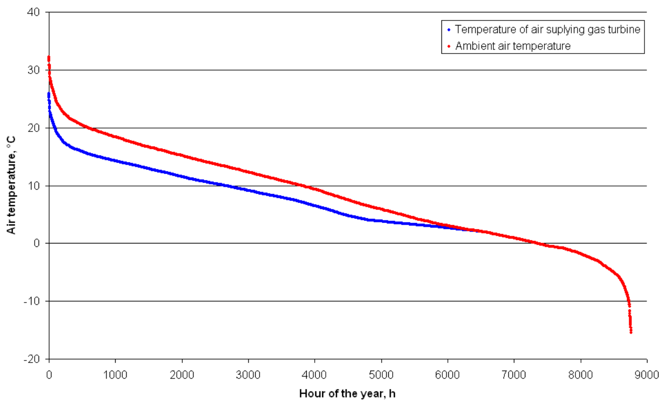

Figure 9: Load curve of ambient air temperature (red line) and the GT inlet air temperature (blue line)

of conditions: minimum, design and maximum ambient temperature.

4. Discussion

The greatest increment in power output occurs at the highest ambient air temperatures, i.e. the lowest reference power output. The power output nearly 15 MW higher than the reference cycle under the same ambient conditions. Only 3.3 MW of the total 15 MW was generated in NG expanders. Moreover, electric efficiency rises significantly, by nearly 0.6%, if the power generated at the turboexpanders is included. In accordance with the assumptions, the air is not cooled during winter (about 2500 h). For the rest of the year, the decrease in air temperature is between 4°C and 7°C.

5. Summary

Although inlet air cooling for gas turbines is a common solution in hot climates, the presented analysis shows that it can also be technically applicable in colder climates. Despite the low air temperature drop (around 4–7 K) the power output increases by up to 4%. The average annual combined cycle efficiency rises by more than 0.6%.

Further research related to NG expanders should be oriented towards a more precise determination of the lowest allowable temperature at the expander outlet, accounting for the conditions of water and hydrocarbon dew points and the hydrate formation point.

The applicability of the proposed solution is limited by the specific technical structure, which is in the vicinity of a large, external NG consumer and a high pressure pipeline. On the other hand, a general conclusion is that integrating various industrial facilities is a good way to: achieve energy and economic savings, boost local cooperation and know-how, and contribute to a more sustainable use of natural resources.

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Appendix A. Design parameters of the optimized CC plants

Table A.9: Evaporators

Parameter	$EVAP_1$	$EVAP_2$	$EVAP_3$
Operating pressure, kPa	20,000	4,100	360
Pinch point, K	5.0	5.0	5.0
Inlet steam temperature, °C	345.8	248.9	129.9
Outlet steam temperature, °C	365.8	251.9	139.9
Inlet exhaust temperature, °C	446.3	297.7	210.4
Outlet exhaust temperature, °C	370.8	256.9	144.9
Outlet steam mass flow, kg/s	69.10	16.76	19.06
Water blowdown, kg/s	–	–	85.86
Heat exchange area, m^2	45,654	35,227	41,229
Calculated duty, kW	55,064	28844	45 390

Table A.10: Economizers

Parameter	$ECON_1$	$ECON_2$	$ECON_3$	$ECON_4$	$ECON_5$
Operating pressure, kPa	20 000	4 100	20 000	360	150
Approach point, K	20.0	3.0	150.5	10.0	15.0
Inlet water temperature, °C	215.3	140.5	142.7	111.6	32.9
Outlet water temperature, °C	345.8	248.9	215.3	129.9	96.5
Inlet exhaust temperature, °C	363.7	256.9	245.3	144.9	133.0
Outlet exhaust temperature, °C	297.7	245.3	214.2	133.0	93.4
Water mass flow, kg/s	69.10	16.76	69.10	104.92	102.06
Heat exchange area, m^2	18 079	4 967	10 011	10 164	12 712
Calculated duty, kW	47 318	8 152	21 731	8 161	27 144

Table A.11: Sections of the steam turbine

Parameter	ST_1	ST_2	ST_3	ST_3 - extraction
Inlet pressure, kPa	20 000	4 100	360	-
Outlet pressure, kPa	4 100	360	5	155
Power output, MW	27.478	50.113	58.791	-
Isentropic efficiency, %	91.0	89.0	86.0	-
Inlet steam temperature, °C	549.3	512.2	210.9	-
Outlet steam temperature, °C	313.1	212.5	32.9	138.6
Outlet steam mass flow, kg/s	69.10	85.86	102.06	2.86
Steam quality	1.000	1.000	0.898	1.000

Table A.12: Pump parameters

Parameter	P_1	P_2	P_3	P_4
Design inlet pressure, kPa	4 100	360	150	5
Design outlet pressure, kPa	20 000	4 100	360	150
Max. control valve pressure, kPa	20 053	4 323	390	-
Isentropic efficiency, %	85.0	85.0	85.0	85.0
Water mass flow, kg/s	69.10	85.86	104.92	102.06
Power demand, MW	1.393	0.408	0.027	0.018

Table A.13: Superheaters

Parameter	$SPHT_1$	$SPHT_2$	$SPHT_3$	$SPHT_4$	$SPHT_5$
Operating pressure, kPa	20 000	4 100	20 000	4 100	360
Hot end, K	40.0	30.0	80.0	15.0	10.0
Inlet steam temperature, °C	409.8	321.1	365.8	251.9	139.9
Outlet steam temperature, °C	549.3	512.2	409.8	355.8	204.2
Inlet exhaust temperature, °C	589.3	542.2	489.8	370.8	214.2
Outlet exhaust temperature, °C	542.2	489.8	446.3	363.7	210.4
Steam mass flow, kg/s	69.10	85.86	69.10	16.76	19.06
Heat exchange area, m^2	12 105	11 483	23 362	2 592	1 954
Calculated duty, kW	35 638	39 212	32 196	5 091	2 622