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Thermodynamic analysis of the advanced zero emission power plant

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Abstract The paper presents the structure and parameters of advanced zero emission power plant (AZEP). This concept is based on the replacement of the combustion chamber in a gas turbine by the membrane reactor. The reactor has three basic functions: (i) oxygen separation from the air through the membrane, (ii) combustion of the fuel, and (iii) heat transfer to heat the oxygen-depleted air. In the discussed unit hot depleted air is expanded in a turbine and further feeds a bottoming steam cycle (BSC) through the main heat recovery steam generator (HRSG). Flue gas leaving the membrane reactor feeds the second HRSG. The flue gas consist mainly of CO_2 and water vapor, thus, CO_2 separation involves only the flue gas drying. Results of the thermodynamic analysis of described power plant are presented.

Keywords: AZEP; Gas turbine; Membrane reactor; Oxy-combustion; CCS

1 Introduction

Power plants fed by natural gas are becoming increasingly important in the energy sector. Currently the high price of natural gas is the main limitation for this technology in Poland. In 2012, 3.6% of electricity was produced from natural gas, while the Polish energy policy until 2030 assumes an increase of this share to 6.6%. This results from the need to diversify the energy sources in Poland and to reduce the carbon dioxide (CO_2) emissions. Combined cycle plants are ecologically favorable and characterized

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by a low investment cost, short construction time, high reliability and flexibility of operation. At currently achieved 60% electric efficiency the CO_2 emission is around 330 kgCO_2/MWh, which is about 2.5-times less than the emission of modern coal-fired plants which exceeds 800 kgCO_2/MWh. The rapid development of natural gas technologies allow to further increase of electric efficiency and reduce harmful emissions [1,2].

The energy sector is facing the new challenges of reducing the CO_2 emission level. The carbon dioxide capture and storage (CCS) technologies are currently under development. They allow for near zero-emission production of electricity from fossil fuels. CCS technologies can be classified into three main groups: post-combustion, pre-combustion and oxy-combustion.

Oxy-combustion technology is based on the fuel combustion in an oxidant atmosphere with increased proportion of oxygen. By elimination of nitrogen from the combustion process flue gas consists mainly of carbon dioxide and water vapor. This allows for CO_2 separation with a relatively low energy cost, realized only by drying the flue gas. However, oxygen separation from air is required, involving significant energy consumption. Currently the use of cryogenic air separation units is considered due to the requirement of high performance and sufficient purity of the oxygen. In the literature different concepts of the natural gas fueled units with oxy-combustion, such as [3–5], can be found. However, in most of presented structures the application of oxy-combustion involves a significant efficiency drop at the level of 7–9 p.p. The advanced solution utilizing the oxy-combustion technology is the replacement of the combustor in gas turbine with mixed conductive membrane (MCM) reactor, in which three processes take place: (i) the oxygen separation from compressed air by a high-temperature membrane; (ii) near-to-stechiometric fuel combustion; (iii) heat exchange to heat the oxygen depleted air. This air is expanded in a turbine and further feeds the heat recovery steam generator (HRSG). The flue gas, consisting mainly of CO_2 and H_2O is mostly recirculated. The remaining part is leaving the membrane reactor and hence its thermal energy is utilized. Then, after moisture removal, CO_2 is compressed and transported to a place of storage.

In the literature units with MCM reactor are defined as advanced zero emission plants (AZEP) and two concepts can be found, differing in the way of energy utilization from flue gas leaving MCM reactor. The first concept, presented among others, in [6–8], is based on the use of increased pressure in the reactor cycle and the flue gas is expanded in the CO_2/steam turbine. In the condenser water vapor is separated from carbon dioxide. The second concept, presented for example in [9], assumes the use of flue gas in additional HRSG, which in consequence allows to increase the power of a bottom steam cycle (BSC). Both AZEP concepts, according to authors, achieve similar electric efficiency, exceeding 50%. The efficiency drop due to the application of CCS technology in relation to the units with similar parameters is assumed at about 5 p.p.

2 Structure of the AZEP power plant

The general structure of the AZEP is shown in Fig. 1. The unit consists of: (I) gas turbine with membrane reactor; (II) bottoming steam cycle with two HRSGs; (III) carbon dioxide separation installation consisting of a condenser and a CO_2 compressor. The models of all analyzed components are made with the use of commercial GateCycle software [10].

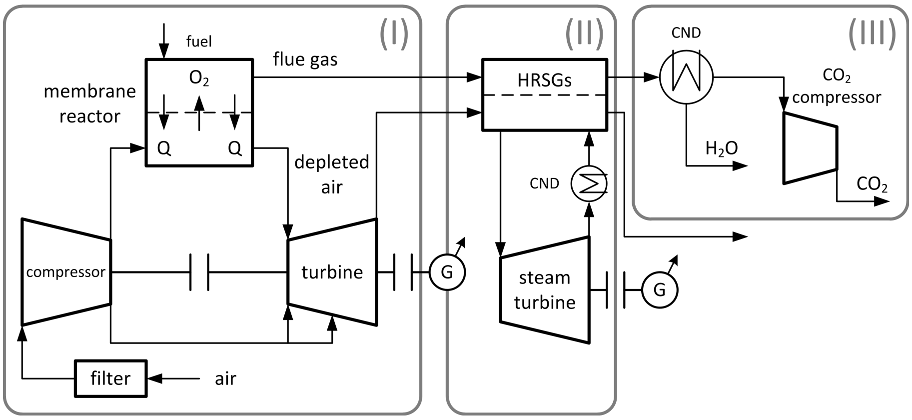

Figure 1: A structure of the advanced zero emission power plant (HRSG – heat recovery steam generator, CND – condenser, G – electric generator, Q – heat transfered).

2.1 Gas turbine with membrane reactor

A detailed scheme of the gas turbine with membrane reactor, consisting of a membrane, heat exchangers and combustion chamber, is presented in Fig. 2. A four-end type high-temperature separation membrane is used in the reactor. Such membranes are made from a material which is an ionic oxygen conductor at temperatures of 700–900 °C and it is using an oxygen partial pressure difference between factors on both sides of the membrane.

The compressed air is heated (pt 2.6a) to the membrane operating temperature, assumed at 850 °C. In the membrane part of the oxygen permeates to flue gas cycle in the reactor. The oxygen-depleted air (pt 2.7a) after additional heating is expanded in the turbine. Recirculated flue gas (pt 5g) consisting mainly of CO_2/H_2O, is enriched in oxygen in the membrane module. Resulting gas is cooled and in the combustion chamber is used as an oxidant in the natural gas combustion process. Hot flue gas is cooled down to the membrane operating temperature by heating the air. Part of the flue gas is recirculated to the membrane, the remaining part (pt 4.1g) is cooled in the regenerative heat exchanger and directed in part to the additional HRSG (pt 1c) and in other part is recirculated to the combustion chamber for cooling (pt 4.3g).

The most important assumptions for the gas turbine and the membrane reactor are presented in Tab. 1. The unit is supplied by the air with ISO conditions (15 °C; 101.325 kPa; relative humidity 60%). Combustion chamber is fed by natural gas with 100% CH_4 content, lower heating value (LHV) equal to 50.049 kJ/kg and parameters at 15 °C/3.5 MPa. The gas turbine is air-cooled with an air flow from compressor, $\dot{m}_{1.1a}$, determined by the relation:

$$\dot{m}_{1.1a} = 0.24\,\dot{m}_{3a}\,\frac{t_{3a} - 1000}{600}\,, \tag{1}$$

where $\dot{m}_{3a}$ and t_{3a} are the turbine inlet air flow and temperature, respectively.

First two stages of the turbine are cooled, with the cooling air flow divided into 70% to the first stage and 30% to the second stage. The selection of key parameters for the gas turbine installation, i.e., pressure ratio, β, and turbine inlet temperature, t_{3a}, is preceded by the thermodynamic analysis presented in Section 3.

2.2 Bottoming steam cycle

The bottoming steam cycle is powered through two triple-pressure HRSGs with reheating. Main HRSG is fed by depleted air from the gas turbine (pt 4a), and the second HRSG is fed by flue gas leaving the membrane reactor (pt 1c). In both HRSGs the steam is generated with the same parameters, then the corresponding flows are mixed to power a common steam turbine. The steam parameters are optimized for the main HRSG, since it provides the major heat flow to the BSC. The flue gas temperature is

Table 1: Assumptions for the gas turbine and the membrane reactor.

Parameter	Value
Gas turbine gross electric power, $N_{el\,GT}$, MW	200
Turbine isentropic efficiency, η_{iT}, –	0.90
Compressor isentropic efficiency, η_{iC}, –	0.88
Circulation fans isentropic efficiency, η_{iF}, –	0.80
Turbine and compressor mechanical efficiency, η_{mT},η_{mC}, –	0.995
Generator efficiency, η_G, –	0.99
Gas turbine and steam cycle own needs, ΔN_{el}, %	2.0
Compressor inlet pressure loss rate, ζ_1, –	0.007
Turbine outlet pressure, $p_{4\,a}$, kPa	102.5
Membrane operating temperature, t_{MEM}, °C	850
Combustion chamber inlet oxidant temperature, t_{1g}, °C	600
Combustion chamber outlet flue gas temperature, t_{2g}, °C	1700
Regenerative heat exchanger outlet flue gas temp., $t_{4.2g}$, °C	630
Combustion chamber inlet gas oxygen content, $(O_2)_{1g}$, –	0.21
Combustion chamber outlet gas oxygen content, $(O_2)_{2g}$, –	0.02
Circulation fan outlet pressure, p_{1g}, $p_{4.4g}$, kPa	107.0
Combustion chamber heat loss, δ_{CCH}, –	0.01
Heat exchangers efficiency, η_{HX}, η_{RX}, –	0.99
Heat exchangers pressure loss ζ_{HX}, ζ_{RX}, kPa	1.0

assumed to achieve the same steam parameters in the second HRSG. Main assumptions for the BSC are presented in Tab. 2. Selected optimal steam pressure and temperature for all pressure levels are presented in Section 4.

2.3 Carbon dioxide separation

Flue gas leaving the HRSG is prepared for the storage of carbon dioxide. In the first place flue gas is directed to the condenser, in which it is cooled to the temperature of 30 °C, and phase separation of the condensed water vapor from the gaseous CO_2 takes place. The prepared gas containing over 90% CO_2 is compressed to the pressure of 13 MPa in 8-stage compressor. Every compressor stage has an identical pressure ratio and intersection cooling of CO_2 stream to a temperature of 30°C, accompanied by subsequent water vapor condensation. Assumed isentropic efficiency of the compressor

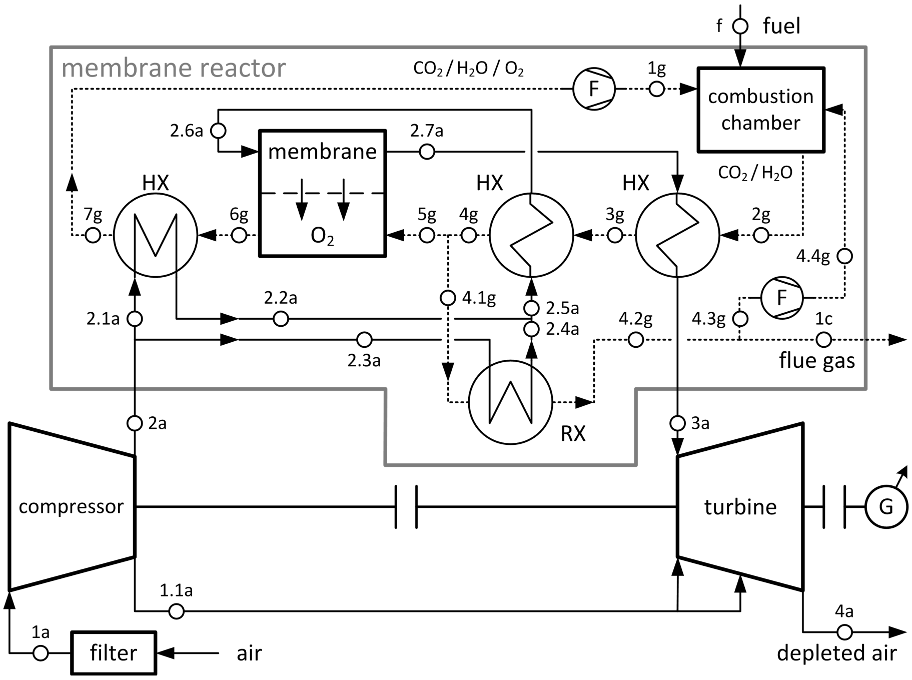

Figure 2: A scheme of the gas turbine with membrane reactor (HX – heat exchanger, RX – regenerative heat exchanger, F – fan, G – electric generator).

Table 2: Assumptions for the bottoming steam cycle.

Parameter		Value
Condenser operating pressure, p_{CND}, kPa		5
Steam turbine isentropic efficiency, $\eta_{i\,ST}$, -		0.90
Steam turbine mechanical efficiency, η_{mST}, –		0.99
Generator efficiency, η_G, –		0.99
HRSGs heat exchangers efficiency, –		0.99
Pressure loss factor in heat exchangers:	– economizers	0.01
	– evaporators	0.04
	– superheaters:	0.03
Steam pressure loss before steam turbine, –		0.03

equals 80%. Compressed supercritical carbon dioxide is transported to the place of storage.

3 Thermodynamic analysis of the gas turbine

3.1 Evaluation methodology

Effectiveness of the combined cycle power plants is evaluated by the efficiency of electric energy production. Gross electric efficiency, $\eta_{el.gross}$, is defined by the relation

$$\eta_{el.gross} = \frac{N_{el.gross}}{\dot{m}_f \, LHV} = \frac{N_{elGT} + N_{elST}}{\dot{m}_f \, LHV} \;, \tag{2}$$

where: $N_{el.gross}$ – gross electric power of the combined cycle unit, N_{elGT} – gas turbine electric power, N_{elST} – steam turbine electric power, $\dot{m}_f$ – fuel mass flow, LHV – lower heating value of the fuel. Electric efficiency of the gas turbine, η_{elGT}, and bottoming steam cycle, η_{elST}, are given by

$$\eta_{elGT} = \frac{N_{elGT}}{\dot{m}_f \, LHV} \;, \tag{3}$$

$$\eta_{elST} = \frac{N_{elST}}{\dot{Q}_{in}} \;, \tag{4}$$

$$\dot{Q}_{in} = \dot{Q}_{4a} + \dot{Q}_{1c} \;, \tag{5}$$

where: $\dot{Q}_{in}$ – total heat flow directed to the HRSGs, $\dot{Q}_{4a}$ – turbine outlet depleted air heat flow, $\dot{Q}_{1c}$ – flue gas heat flow leaving the membrane reactor.

The net electric efficiency of the combined cycle unit is defined with analogy to (2), taking into account the own needs of individual installations within the unit, i.e., the gas turbine and the BSC ΔN_{el}, installation of CO_2 compression, ΔN_{CC}, and circulation fans in the membrane reactor, ΔN_F:

$$\eta_{el} = \frac{N_{elGT} + N_{elST} - \Delta N_{el} - \Delta N_{CC} - \Delta N_F}{\dot{m}_f \, LHV} \;. \tag{6}$$

The total own needs rate of the unit, δ , is given by the expression

$$\delta = \sum \frac{\Delta N_i}{N_{el.gross}} = \frac{\Delta N_{el} + \Delta N_{CC} + \Delta N_F}{N_{el.gross}} \;. \tag{7}$$

3.2 Influence of chosen parameters on the gas turbine efficiency

The key parameters of the gas turbine are pressure ratio and turbine inlet temperature. Therefore, the influence of pressure ratio in the range of $\beta = 15$–30 and the temperature in the range of $t_{3a} = 1400$–$1600\,^{\circ}$C on the efficiency of gas turbine is analyzed. Adopted ranges of analysis correspond to currently available gas turbines class F, G or H. The impact of β and t_{3a} on the gas turbine efficiency, η_{elGT}, the heat flow, $\dot{Q}_{in}$, and the turbine outlet temperature, t_{4a}, respectively, are presented in Figs. 3–5. The results for η_{elGT} and $\dot{Q}_{in}$ are compared to the conventional gas turbine (conv. GT) with similar assumptions.

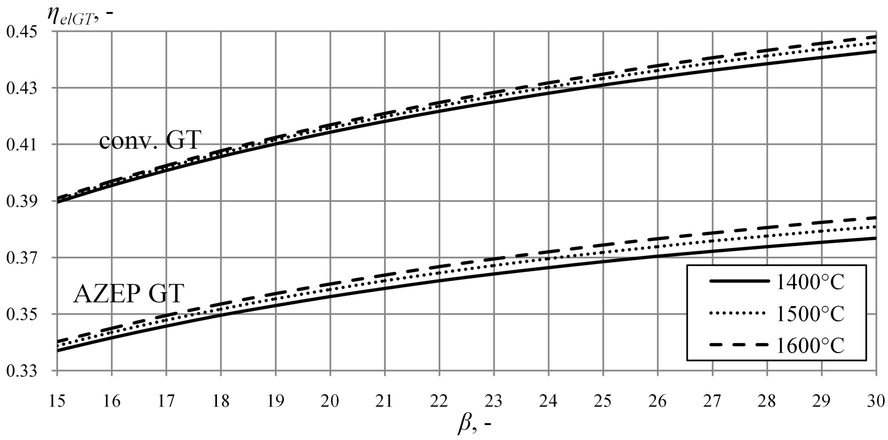

Figure 3: Gas turbine electric efficiency, η_{elGT}, in a function of β and t_{3a}.

Higher pressure ratios and turbine inlet temperatures allows to achieve better gas turbine efficiency. However, the temperature rise entails the need to increase the air flow cooling expander blades, which are the most vulnerable to high temperature, in the analyzed model according to Eq. (1). Therefore, the increase in t_{3a} results in a slight η_{elGT} improvement, but also leads to an increase in turbine outlet temperature and, consequently, enables to obtain higher efficiency of the BSC. The η_{elGT} growth with the increase of β binds to a significant decrease of the turbine outlet gas temperature and flow, which reduces the effective use of the heat in the BSC.

The resulted efficiency η_{elGT} in the range of 34–38% is lower than achieved in the conventional gas turbine by 5–6 p.p. due to the structure of installation with the membrane reactor. In the conventional gas

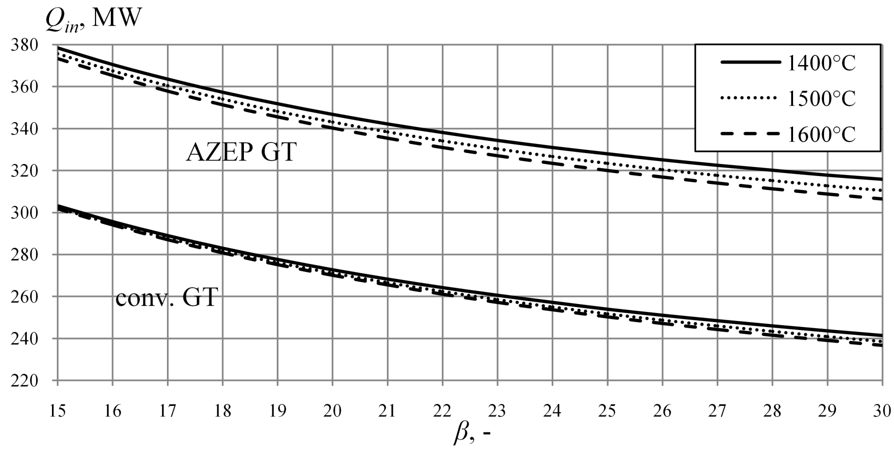

Figure 4: Total heat flow, $\dot{Q}_{in}$, in a function of β and t_{3a}.

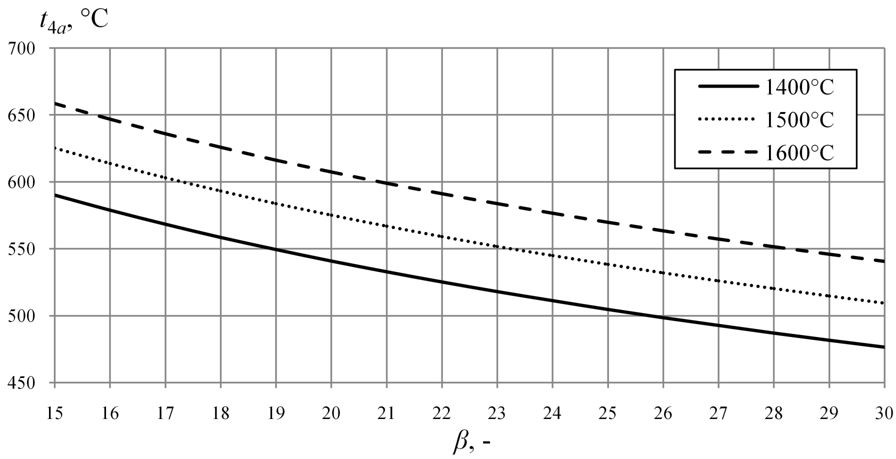

Figure 5: Turbine outlet temperature, t_{4a}, in a function of β and t_{3a}.

turbines the turbine inlet gas flow is greater than the compressed air flow by the amount of used fuel. In AZEP the turbine inlet gas flow is lower than compressed air flow by the amount of oxygen permeated in the membrane, thus, the turbine effective work is lower for the same compressor air flow. On the other hand, membrane reactor is a source of additional flue gas stream feeding the HRSG and allows to increase the BSC power. Total heat flow applied to the HRSGs is increased by 26–31% relative to the conventional gas turbine.

4 Results

For the analysis it were assumed the following parameters: gas turbine temperaturet_{3a} = 1600 °C and pressure ratio β = 20. Main HRSG is fed by depleted air flow $\dot{m}_{4a}$= 453.9 kg/s with temperature t_{4a} = 607.4 °C. The BSC optimization for given depleted air resulted in the following steam parameters: live steam at steam turbine inlet at 580°C/ 18.0 MPa; reheated steam at 580°C/ 4.0 MPa; low pressure saturated steam at 0.3 MPa. The additional HRSG is fed by the flue gas flow $\dot{m}_{1c}$ = 56.7 kg/s consisting of 65.3% H_2O, 32.7% CO_2, and 2.0% O_2.

The flue gas dried in condenser with mass flow $\dot{m}_{3c}$ = 32.3 kg/s and composition: 90.4% CO_2, 4.1% H_2O and 5.5% O_2, is directed to the CO_2 compressor. Almost all water vapor is condensed during intercooling, so final CO_2 purity is 94.2% with less than 0.1% H_2O. Unit energy consumption of CO_2 compression installation reached 0.097 kWh/kgCO_2.

Obtained characteristic parameters for the gas turbine and the bottoming steam cycle of the AZEP are presented in Tab. 3.

5 Conclusions

The use of membrane reactor isolates the combustion process, which is realized in the reactor internal cycle, from air in the gas turbine. Therefore, the flue gas contains no nitrogen and separation process of carbon dioxide is reduced only to drying the flue gas.

A turbine with a membrane reactor achieves a lower electric efficiency (about 36%) than conventional gas turbines with similar operating parameters, reaching over 41%. This results from lower air stream feeding the turbine in the AZEP.

Lower gas turbine efficiency is partially offset by flue gas leaving the membrane reactor, which supply the additional HRSG and increases the BSC power.

The AZEP achieved net electric efficiency at almost 54%, relative to power plants without carbon capture installation it is efficiency drop by 5–6 p.p. This result is more favorable than in a combined cycle plant with oxy-combustion or post-combustion CCS installation, reaching net efficiency at the level of 50–52%.

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Table 3: Chosen characteristic parameters of the AZEP.

Parameter	Value
Turbine internal power, N_{iT}, MW	428.0
Compressor internal power, N_{iC}, MW	222.7
Gas turbine electric power, N_{elGT}, MW	200.0
Fuel chemical energy flow, $\dot{m}_f$ LHV, MW	554.6
Gas turbine electric efficiency, η_{elGT}, –	0.3607
Turbine outlet depleted air heat flow, $\dot{Q}_{4a}$, MW	288.7
Flue gas heat flow leaving the membrane reactor, $\dot{Q}_{1c}$, MW	51.5
Total heat flow feeding the HRSGs, $\dot{Q}_{in}$, MW	340.2
Steam turbine electric power, $N_{el\,ST}$, MW	118.6
Bottoming steam cycle electric efficiency, $\eta_{el\,ST}$, –	0.3487
AZEP gross electric power, $N_{el.gross}$, MW	318.6
Gross electric efficiency, $\eta_{el.gross}$, –	0.5746
Gas turbine and BSC own needs, $\Delta\, N_{el}$, MW	6.4
CO_2 compression installation own needs, $\Delta\, N_{CC}$, MW	10.7
Circulation fans in membrane reactor own needs, $\Delta\, N_F$, MW	2.2
Total own needs rate, δ, –	0.0605
Net electric power, N_{el}, MW	299.3
Net electric efficiency, η_{el}, –	0.5398
CO_2 unit production, u_{CO_2}, kg/MWh	365.6
CO_2 unit emission, e_{CO_2}, kg/MWh	0.0

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