Optimization of the structure of the Combined Cycle Power Plant with Oxy-combustion – Case study

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Abstract:

This paper presents a gas turbine combined cycle plant with oxy-combustion and carbon dioxide capture. Oxy-combustion technology is based on the burning of fuel in an oxidant atmosphere with increased proportion of oxygen. The main advantage of this solution is elimination of nitrogen in the combustion process, allowing for simplified separation of carbon dioxide from flue gas.

Basic structure of the plant with triple-pressure heat recovery steam generator (HRSG) with reheating is presented. There are considered ways to improve the power plant efficiency by changing its structure in order to reduce the compressors work, to improve the expander cooling efficiency, or to efficient use of the waste heat within the unit. Different Cases of a modification of the basic plant structure with thermodynamic analysis are presented. A comparison of presented Cases was made in order to select the optimal structure of combined cycle power plant with oxy-combustion.

Keywords:

Carbon Capture, Combined Cycle Power Plant, Oxy-combustion.

1. Introduction

Combined cycle power plants (CCPP) are one of the fastest developing electricity production technologies. Modern CCPP units achieve efficiency exceeding 60% and CO_2 emission on the level of 330 kg/MWh, which is about 2.5-times lower than CO_2 emission of modern coal-fired power plants, producing about 800 kg/MWh. Moreover, a number of advantages, such as low investment cost, short construction time, high flexibility and reliability of operation, contribute to increasing popularity of natural gas power plants [1,2]. The share of natural gas in global electricity generation is growing from 15.2% in 1995 to 22.5% in 2012 [3].

Main directions for CCPP efficiency improvement are: the increase of gas turbine parameters (temperature and pressure), the improvement of turbine air-cooling or closed steam-cooling techniques and the increase of steam cycle parameters [4,5]. Most producers currently set the combustor outlet temperature (COT) to 1500°C, only one introduced a COT = 1600°C and is doing research towards the use of 1700°C [6-8]. In fact, the maximal allowable temperatures of alloys used to produce turbine blades are significantly lower, at the level of 900°C. Elements exposed to the highest temperature are covered by thermal barrier coatings (TBCs) and chilled using advanced blade cooling techniques, such as film or transpiration air-cooling [9,10].

The energy sector is facing a new challenges of growing demands for reducing the CO_2 emission into the atmosphere, in a form of, e.g., the Kyoto Protocol, energy-climatic package and the European Emission Trading System introduced by the Directive 2009/29/EC [11]. Current ecologic characteristics of CCPP are insufficient to achieve the assumed reduction of CO_2 emission. As a solution, the carbon capture and storage technologies (CCS) are being developed, allowing for near zero-emission production of electricity from fossil fuels [12,13].

One of proposed CCS technologies is oxy-combustion, which is based on the combustion of fuel in an atmosphere of pure oxygen. The nitrogen is eliminated from the combustion process and the flue gas consists primarily of carbon dioxide and water vapour. In practice, minor amounts of other gases, such as oxygen, nitrogen and argon, are present due to an excess of oxidant in the combustion chamber, the presence of other gases in oxidant (the oxygen purity is below 100%) or leakages in the installation. The main advantage of the units with oxy-combustion is the possibility of CO_2 separation with a relatively low energy consumption, often limited to the removal of water vapour from the flue gas. In the literature different concepts of the natural gas units with oxy-combustion have been analyzed, e.g. [14-17].

One of the biggest challenges for oxy-combustion units is the need to produce large amounts of oxygen. The only mature technology allowing for the oxygen production in required quantity and quality are the cryogenic air separation installations. However, they are characterized by high energy consumption, at the level of 0.2-0.24 kWh/kgO_2 [18-20]. In the literature other methods are presented, with lower energy demand, such as low-temperature and high temperature membrane installations, adsorption methods and hybrid units [21-25]. Yet, in most cases, at the present development stage, they do not allow to produce oxygen with the required parameters.

In order to identify the operating parameters of CO_2 capture installation and effectiveness of the unit, it is necessary to define guidelines for the parameters and composition of captured gas to be stored. Currently, there are no conclusive guidelines, and the requirements concerning carbon dioxide are mainly based on the experience of the food industry and the Enhanced Oil Recovery (EOR) technology [26,27]. The most commonly used parameter is purity (molar share of CO_2 in captured gas) equal or higher than 0.9. Minimal recovery rate of CO_2 in captured stream to the flue gas (molar) is also assumed at the level of 0.9. Captured gas is stored in super-critical condition, therefore, it has to be compressed. Pressure of the gas before transportation should be no lower than 13-15 MPa.

2. Description of the oxy-combustion combined cycle plant model

The CCPP unit consist of two separate heat cycles, the gas turbine and the steam cycle, connected by heat regenerative steam generator (HRSG). The oxy-combustion technology requires additional installations. The oxygen is provided by hybrid membrane-cryogenic air separation unit (ASU). The flue gas leaving HRSG is partially recirculated to the compressor as a working gas replacing nitrogen. The remaining part of flue gas leaving the gas turbine cycle is sent to the carbon dioxide conditioning installation (CC), where captured gas is prepared for transportation to the place of storage. A scheme of this unit is presented in Fig. 1. The models of analyzed components were built using the GateCycle™ software [28].

2.1. Gas turbine

The gas turbine includes a compressor, separate oxidant compressor, combustion chamber and expander. In the gas turbine assumed electric power equal to 200 MW and combustor outlet temperature COT = 1500°C. In the oxy-combustion unit the recirculated gas composition is completely different than the air, therefore, it has other thermodynamic properties. Thermodynamic analysis presented in [29] revealed that the advisable compression ratios in the oxy-combustion units should be higher than in conventional air-combustion gas turbines. In this unit the compression ratio $\beta = 50$ is selected. Such high value is applied in aviation turbine Rolls Royce Trent 1000.

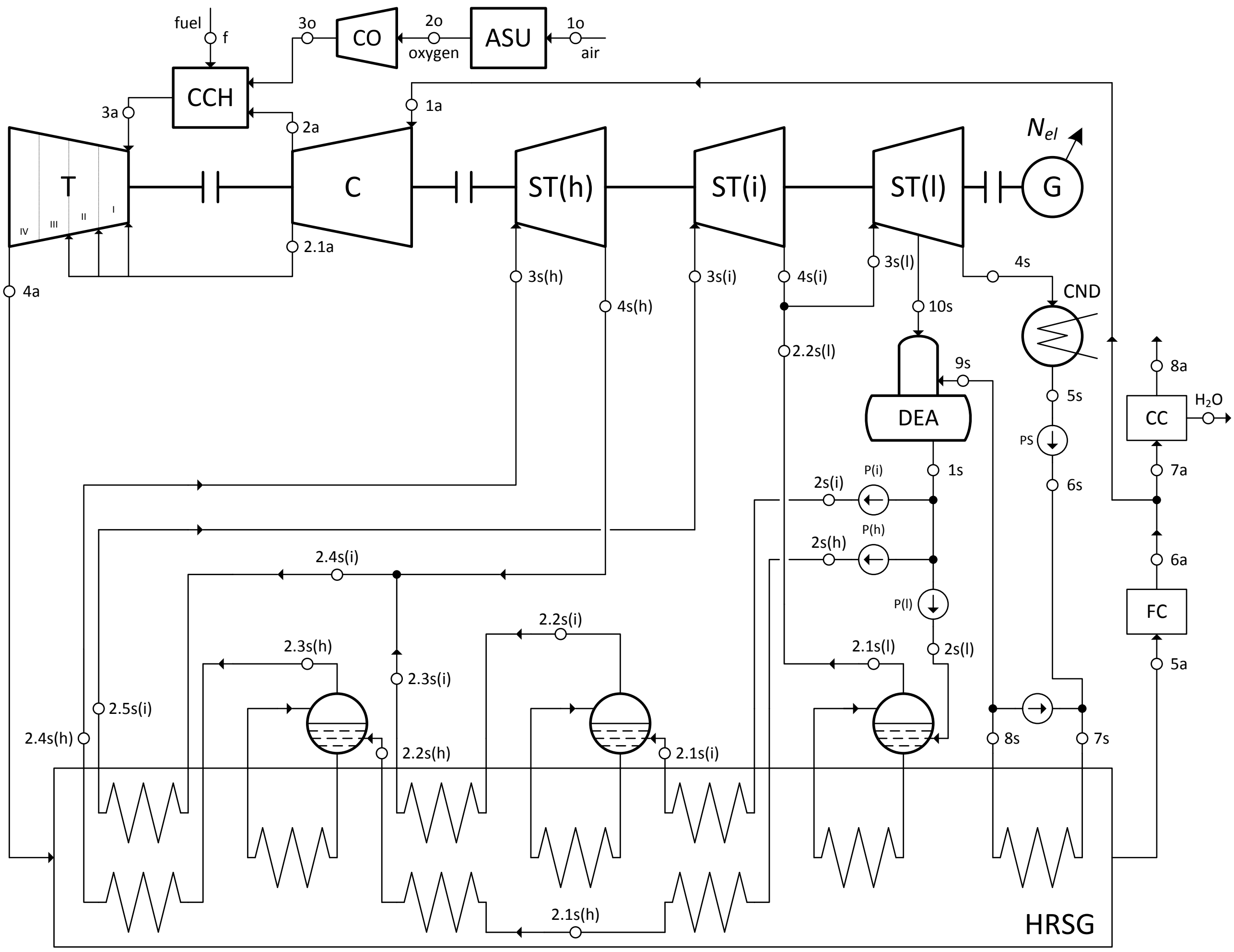

Fig. 1. Scheme of the oxy-combustion combined cycle power plant (ASU – air separation unit, C – compressor, CC – carbon dioxide conditioning installation, CCH – combustion chamber, CND – condenser, CO – oxygen compressor, DEA – deaerator, FC – flue gas cooler, G – generator, HRSG – heat regenerative steam generator, P – pump, ST – steam turbine, T – turbine; (h) – refers to high, (i) – intermediate, (l) – low pressure level)

The working factor in the gas turbine is flue gas leaving the cooler (FC), which maintain the constant temperature of recirculated gas. This eliminates the possible influence of changes in the temperature on the gas turbine parameters. The oxidant from ASU is compressed in separate oxidant compressor and mixed with flue gases in combustion chamber. The flue gas recirculation rate is determined by an assumption of constant oxygen composition equal 2% in combustor outlet flue gas.

The gas turbine is fed by natural gas with a purity of 100% CH_4, with the temperature and pressure equal to 15°C and 3.5 MPa, respectively. The fuel is compressed to the required pressure inside the combustion chamber. The lower heating value is LHV = 50.049 MJ/kg. Gas film cooling is applied to the turbine blades, with compressed flue gas used for cooling. The implemented turbine cooling model results from the heat flow balance in the turbine blading system and is presented, e.g., in [10,30]. The expander is composed of four stages, wherein first three have cooling system. The required cooling gas mass flow is separately determined for each turbine stage using the relationship:

$$\dot{m}_c = \dot{m}_g \cdot \frac{k \cdot St}{\eta_c} \cdot \left(\frac{t_{g.i} - t_b}{t_b - t_{c.i}} \right) \cdot \frac{c_{p.g}}{c_{p.c}} \cdot \left(1 - \eta_{film}\right), \quad (1)$$

where:

m_c, m_g – mass flows of the cooling gas and the hot gas at the turbine stage inlet, respectively,

k – ratio between the heat transfer area and the hot gas cross-section area (k is about 6-10, here $k = 8$),

St – Stanton number (approximately 0.005),

η_c – cooling efficiency (assumed $\eta_c = 0.5$),

t_b – blade temperature (here $t_b = 900°C$),

$t_{c.i}$, $t_{g.i}$ – the cooling gas and the hot gas temperature at the turbine stage inlet, respectively,

$c_{p.c}$, $c_{p.g}$ – the average specific heat capacity between inlet and blade temperatures of the cooling gas and the hot gas, respectively,

η_{film} – film cooling isothermal effectiveness (here $\eta_{film} = 0.4$).

The cooling gas flow ratio γ_c is based on the total compressed gases flows, including oxidant:

$$\gamma_c = \frac{\dot{m}_{2.1a}}{\dot{m}_{1a} + \dot{m}_{2o}}. \quad (2)$$

The isentropic efficiencies of the compressors and the turbine are determined on the basis of the compressor polytropic efficiency as a function of β and the turbine polytropic efficiency as a function of turbine inlet temperature (TIT) and β. Efficiency characteristics are taken from [31] for conservative scenarios. Detailed calculation algorithms are described in [32]. The other important assumptions for the gas turbine are presented in Table 1.

Table 1. Assumptions for gas turbine

Parameter		Unit	Value
Mechanical efficiency	η_m	-	0.995
Generator efficiency	η_G	-	0.985
Combustion efficiency	η_{CCH}	-	0.99
Combustion chamber pressure loss	ζ_{CCH}	-	0.045
HRSG pressure loss (including FC)	ζ_{HRSG}	-	0.04
Turbine outlet pressure	p_{4a}	kPa	105.5
Gas turbine and steam cycle own needs ratio	δ_{el}	-	0.02

2.2. Steam cycle

The steam cycle is fed by flue gas leaving gas turbine through a HRSG. In the analyzed unit a triple-pressure HRSG with intermediate steam reheating is applied. The produced steam powers a three-section steam turbine. A deaerator is fed by extraction steam from the low-pressure steam turbine section. In the HRSG the deaeration economizer replaces the low-pressure economizer and the high-pressure economizer is divided into two sections to optimize the flue gas temperature distribution. The main assumptions for steam cycle are summarized in Table 2. Due to the high water vapour content in flue gas, the temperature of gas leaving HRSG may not be lower than 90°C, to avoid condensation. The underheating of water in deaeration economizer $\Delta t_{ap(D)}$ is controlled not to exceed that limit.

Table 2. Assumptions for steam cycle

Parameter		Unit	Value
Live steam temperature at the turbine inlet	$t_{3s(\mathrm{h})}$	°C	600.0
Live steam pressure at the turbine inlet	$p_{3s(\mathrm{h})}$	MPa	18.0
Reheated steam temperature at the turbine inlet	$t_{3s(\mathrm{i})}$	°C	600.0
Reheated steam pressure at the turbine inlet	$p_{3s(\mathrm{i})}$	MPa	3.6
Low-pressure level steam pressure at the turbine inlet	$p_{3s(\mathrm{l})}$	MPa	0.3
Deaerator pressure	p_{DEA}	MPa	0.2
Condenser pressure	p_{CND}	kPa	5.0
Steam turbine isentropic efficiency	η_{iST}	-	0.90
Steam turbine mechanical efficiency	η_{mST}	-	0.99
HRSG heat exchangers efficiency	η_{HE}	-	0.99
Pinch point temperature differences in evaporators	Δt_{pp}	°C	5.0
Underheating of water at the economizer outlet – approach point	Δt_{ap}	°C	5.0

2.3. Air separation unit

The oxidant supplied to the combustion chamber is produced in hybrid membrane-cryogenic ASU, composed of two parts. The first part is a low-temperature selective membrane, separating the supplied air into the permeate with increased oxygen content and to the retentate with reduced oxygen content, which is discharged to the atmosphere. The permeate is sent to the second, cryogenic part, in which the high purity oxygen is obtained through phase separation at a very low temperatures. The use of membrane part leads to a significant reduction of air flow in the cryogenic part, and also its dimensions and power demand. Oxygen purity is assumed to be 99.5% and remaining 0.5% is nitrogen. The hybrid ASU is over 10% less energy consuming than cryogenic units, which confirms analyzes in [20,21]. In described model the structure and operating parameters of ASU installation are not analyzed, the unit energy consumption rate is used to determine total power demand. This rate for cryogenic ASU is about 0.20-0.24 kWh/kgO_2 [19]. In calculations for hybrid ASU assumed unit energy consumption equal to $E_{N(ASU)}$ = 0.18 kWh/kgO_2. Power demand of ASU installation is calculated, using oxidant mass flow m_{2o}, from relation:

$$\Delta N_{\mathrm{ASU}} = \dot{m}_{2\mathrm{o}} \cdot E_{\mathrm{N(ASU)}} . \tag{2}$$

2.4. Carbon dioxide conditioning

The not recycled part of flue gas is leaving gas turbine cycle and sent to the carbon dioxide conditioning installation. At first, the flue gas is cooled to the temperature of 20°C followed by phase separation of condensed moisture and gaseous carbon dioxide. The prepared CO_2 has over 90% purity, thus, there is no need to introduce any additional purification process. The captured stream is compressed to the pressure of 13 MPa in eight-section compressor with identical compressor ratios and intercooling to the temperature of 30°C in each section. The CO_2 compressors isentropic efficiency is equal to 80%. The compressed carbon dioxide as a supercritical fluid is prepared for transport to the place of storage. The carbon capture effectiveness in analyzed installation may be assumed equal to 100%, without taking into account unexpected leaks.

3. Description of the analyzed cases

The application of carbon capture in oxy-combustion technology in CCPP is bound with efficiency drop, caused mainly by high energy consumption of ASU and CO_2 compression installations. In the analyzed unit these installations have set up fixed operating parameters. Thus, their power demands depends only on the oxidant and captured carbon dioxide mass flows, and therefore, on the gas turbine efficiency. Therefore, there are considered ways to improve the power plant efficiency by reducing power of the compressor, improving the expander cooling efficiency, or by the efficient use of the waste heat within the unit.

The reference plant described in Section 2 is indicated as a case A. There are five different cases B-F implementing such modifications in the reference structure as:

- cooling and drying of the recirculated gas,
- cooling of the gas used for turbine blade cooling to reduce its flow and using the recovered heat in the steam cycle,
- high pressure water injection in compressor, which creates fog and evaporation of fog droplets lower the compressed gas temperature,
- the use of extractions in compressor for the cooling of second and third turbine sections.

The structures of all cases are presented in Fig. 2. The descriptions of all the main features and differences of the analyzed cases are summarized in Table 3.

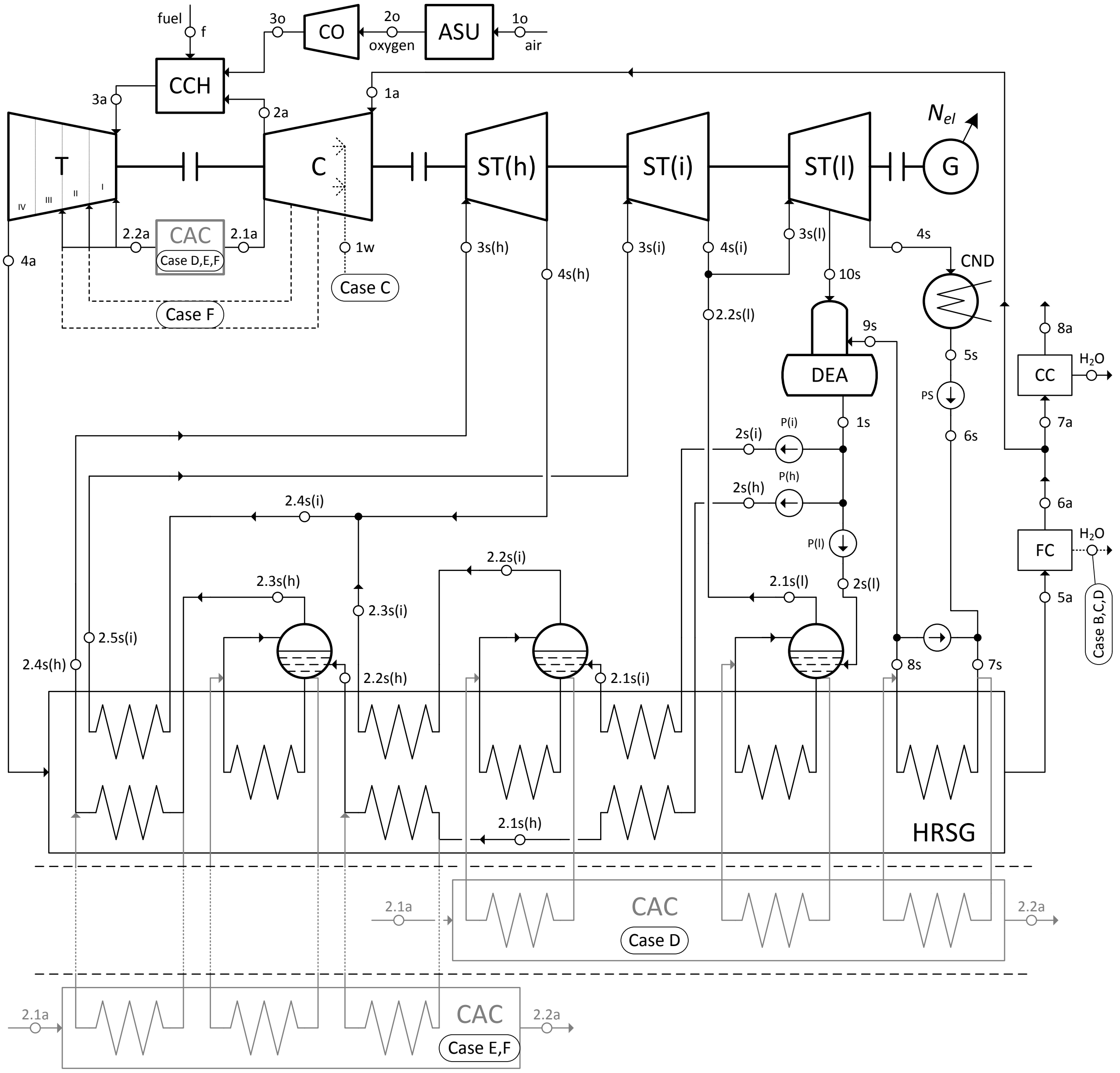

Fig. 2. Scheme of the analyzed cases of the oxy-combustion combined cycle plant (CAC – cooling air cooler)

Table 3. Main characteristic features of the analyzed cases

Name	Description
Case A	Reference CCPP system with oxy-combustion, consisting of all components described in Section 2.
Case B	A system with flue gas cooling in FC to the temperature of t_{6a} = 30°C, resulting in condensation of the majority of water vapour in the recirculated flue gas.
Case C	A system with flue gas cooling to the temperature of t_{6a} = 30°C and high pressure water injection in compressor to lower the compressed gas temperature. The water injection takes place in compressor at the pressure of $p_{1w} = p_{1a}\cdot\beta^{0.33}$, with the water mass flow $m_{1w} = 0.02\cdot m_{1a}$ at the temperature of 15°C.
Case D	A system with flue gas cooling to the temperature of t_{6a} = 30°C. The turbine blade cooling gas is chilled in cooling air cooler (CAC) to the temperature of $t_{2.2a}$ = 100°C. The heat recovered in CAC is used for generation of additional steam flow to the steam turbine. The temperature level of cooled gas allowed for the use of heat exchangers with the parameters corresponding to the following elements in HRSG: intermediate-pressure and low-pressure evaporators and deaeration economizer.
Case E	A system with cooling of the turbine blade cooling gas in CAC. The gas is cooled to the temperature of $t_{2.2a}$ = 300°C due to the high water vapour content and the associated risk of condensation in lower temperature. The heat recovered in CAC is used to generate additional steam flow to the steam turbine. The high temperature level of the gas enabled the use of high-pressure heat exchangers: economizer, evaporator and superheater (with the parameters corresponding to the HRSG components).
Case F	A system with extractions in compressor for the cooling of second and third turbine sections. The extractions have 10% higher pressure than corresponding turbine stages. The gas cooling the first turbine stage is chilled in CAC to the temperature of $t_{2.2a}$ = 300°C. The heat recovered in CAC is used identical to case E.

4. Thermodynamic analysis

4.1. Evaluation methodology

The gas turbine electric power N_{elGT} is determined taking into account internal power of the turbine N_{iT} and both compressors N_{iC}, N_{iCO}:

$$N_{elGT} = \left(N_{iT} \cdot \eta_{mT} - \frac{N_{iC} + N_{iCO}}{\eta_{mC}} \right) \cdot \eta_G \, . \qquad (3)$$

The effectiveness of the combined cycle power plants is evaluated by the efficiency of electricity generation. The gross electric efficiency $\eta_{el.gross}$ is defined by the following equation:

$$\eta_{el.gross} = \frac{N_{el.gross}}{\dot{m}_f LHV} = \frac{N_{elGT} + N_{elST} + N_{elCAC}}{\dot{m}_f LHV} \, , \qquad (4)$$

where $N_{el.gross}$ is the gross electric power of the whole unit, N_{elST} is the steam turbine electric power, N_{elCAC} is the steam turbine additional power resulting from the use of heat recovered from turbine cooling gas, m_f is the fuel mass flow, and LHV is the lower heating value of the fuel.

The electric efficiency of the gas turbine η_{elGT} and the steam turbine η_{elST} are defined by (5) and (6), respectively:

$$\eta_{elGT} = \frac{N_{elGT}}{\dot{m}_f LHV} \, , \qquad (5)$$

$$\eta_{elST} = \frac{N_{elST} + N_{elCAC}}{\dot{Q}_{4a}}, \qquad (6)$$

where Q_{4a} is the heat flow leaving the gas turbine. The heat flow recovered in CAC and used in steam cycle is treated as a waste heat and is ignored in (6).

Including (5-6) and introducing the α rate defined by (7), the $\eta_{el.gross}$ can be written as (8):

$$\alpha = \frac{\dot{Q}_{4a}}{N_{elGT}}, \qquad (7)$$

$$\eta_{el} = \eta_{elGT} \cdot (1 + \alpha \cdot \eta_{elST}). \qquad (8)$$

The net electric efficiency η_{el} includes own needs of all installations within the unit, i.e. machines and devices in gas turbine and steam turbine installations ΔN_{el}, air separation unit ΔN_{ASU} and carbon dioxide conditioning installation ΔN_{CC}:

$$\eta_{el} = \frac{N_{el}}{\dot{m}_f LHV} = \frac{N_{elGT} + N_{elST} + N_{elCAC} - \sum \Delta N_i}{\dot{m}_f LHV}, \qquad (9)$$

$$\sum \Delta N_i = \Delta N_{el} + \Delta N_{ASU} + \Delta N_{CC}. \qquad (10)$$

Using the total own needs rate δ defined in (11), the η_{el} is expressed in a form of (12):

$$\delta = \sum \frac{\Delta N_i}{N_{el.gross}} = \frac{\Delta N_{el} + \Delta N_{ASU} + \Delta N_{CC}}{N_{el.gross}}, \qquad (11)$$

$$\eta_{el} = \eta_{elGT} \cdot (1 + \alpha \cdot \eta_{elST}) \cdot (1 - \delta). \qquad (12)$$

4.2. Results

The thermodynamic analysis of different cases of the unit is performed to determine the main quantities, thermodynamic parameters and energy consumption of all installations within the unit. The most important results of the thermodynamic analysis of all the cases are shown in Table 4. The temperature and composition of gases in the most important characteristic points in the units are presented in Table 5. The captured and compressed carbon dioxide has 93.3-96.5% purity, depending on the case, thus it meets the quality requirements (purity of >90%).

Table 4. Main results of the thermodynamic analysis

Parameter		Unit	Case A	Case B	Case C	Case D	Case E	Case F
Gas turbine electric power	N_{elGT}	MW	200.00	200.00	200.00	200.00	200.00	200.00
Turbine internal power	N_{iT}	MW	496.18	376.03	350.17	369.70	469.20	461.69
Compressor internal power	N_{iC}	MW	261.84	143.46	118.48	136.96	235.18	228.36
Oxygen compressor internal power	N_{iCO}	MW	27.36	26.79	26.17	27.02	27.30	26.70
Turbine isentropic efficiency	η_{iT}	-	0.8943	0.8885	0.8883	0.8869	0.8912	0.8919
Compressor isentropic efficiency	η_{iC}	-	0.8710	0.8744	0.8734	0.8744	0.8710	0.8710
Oxygen compressor isentropic efficiency	η_{iCO}	-	0.8604	0.8604	0.8604	0.8604	0.8604	0.8604
Chemical energy of fuel	$m_f LHV$	MW	543.17	541.43	528.51	546.03	542.05	530.03
Gas turbine electric efficiency	η_{elGT}	-	0.3682	0.3694	0.3784	0.3663	0.3690	0.3773
Gas turbine outlet heat flow	Q_{4a}	MW	362.70	334.59	306.60	327.46	341.88	337.13
Gas turbine outlet heat flow rate	α	-	1.8135	1.6730	1.5330	1.6373	1.7094	1.6857
Steam turbine electric power	N_{elST}	MW	132.91	131.32	120.10	129.23	127.04	124.98
Steam turbine additional power	N_{elCAC}	MW	0.00	0.00	0.00	3.12	8.33	4.73
Steam part electric efficiency	η_{elST}	-	0.3664	0.3925	0.3917	0.4042	0.3960	0.3847
Gross electric power	$N_{el.gross}$	MW	332.91	331.32	320.10	332.35	335.37	329.71
Gross electric efficiency	$\eta_{el.gross}$	-	0.6129	0.6119	0.6057	0.6087	0.6187	0.6221
ASU own needs power	ΔN_{ASU}	MW	29.06	28.46	27.80	28.70	29.00	28.36
CC installation own needs power	ΔN_{CC}	MW	11.09	10.58	10.35	10.67	11.07	10.82
GT and ST own needs power	ΔN_{el}	MW	6.66	6.63	6.40	6.65	6.71	6.59
Own needs rate	δ	-	0.1406	0.1378	0.1392	0.1385	0.1395	0.1388
Net electric power	N_{el}	MW	286.09	285.65	275.55	286.33	288.59	283.93
Net electric efficiency	η_{el}	-	0.5267	0.5276	0.5214	0.5244	0.5324	0.5357
Turbine cooling gas flow ratio	γ_c	-	0.1714	0.1214	0.1106	0.0777	0.0914	0.1099

Table 5. Temperature and gas composition in characteristic points of the units

Parameter		Unit	Case A	Case B	Case C	Case D	Case E	Case F
Compressor inlet temperature	t_{1a}	°C	90.0	30.0	30.0	30.0	90.0	90.0
Compressor inlet gas composition		-						
- CO_2			0.326	0.925	0.924	0.925	0.326	0.326
- H_2O			0.651	0.042	0.042	0.042	0.651	0.651
- O_2			0.020	0.023	0.025	0.024	0.020	0.020
- N_2			0.003	0.009	0.009	0.009	0.003	0.003
Compressor outlet temperature	t_{2a}	°C	628.8	431.7	360.4	431.7	628.8	628.8
Turbine cooling gas temperature	$t_{2.2a}$	°C	628.8	431.7	360.4	100.0	300.0	300.0
Turbine outlet temperature	t_{4a}	°C	648.6	725.0	723.1	734.4	667.2	664.0
Turbine outlet gas composition		-						
- CO_2			0.326	0.808	0.768	0.803	0.326	0.326
- H_2O			0.651	0.164	0.204	0.169	0.651	0.651
- O_2			0.020	0.020	0.020	0.020	0.020	0.020
- N_2			0.003	0.008	0.008	0.008	0.003	0.003
HRSG outlet temperature	t_{5a}	°C	90.0	76.3	76.6	77.3	90.0	90.0

5. Summary and conclusions

This paper presents the thermodynamic analysis of combined cycle power plant with oxy-combustion in different cases. The reference plant is presented in case A. Various modifications in the reference structure in order to improve thermodynamic parameters and electric efficiency are implemented in cases B-F.

Gas turbines with oxy-combustion are characterized by lower electric efficiency than classic gas turbines with air combustion. However, this results in large gas turbine outlet heat flow, which is feeding the heat recovery steam generator and provides high steam turbine power. Moreover, the exhaust losses are reduced by the flue gas recirculation. This gives a high gross electric efficiency of the oxy-combustion unit, in reference case A almost equal to 61.3%. Due to the use of additional installations associated with carbon capture technology, the own needs ratio is about 14%. Therefore, the net efficiency is significantly lower than in a classic system without carbon capture technology, and in Case A is equal to 52.67%.

One of the modifications, applied in cases B-D, is the cooling and drying of the recirculated gas. This process causes a significant change in the flue gas composition (Table 5), and therefore, in the gas turbine operating parameters. The compressor internal power is reduced, from N_{iC} = 261.84 MW in case A to N_{iC} = 143.46 MW in case B. The advantages of the recirculated gas cooling are: a significant reduction of the gas turbine dimensions, decreased flow of the turbine blade cooling gas, and a minor reduction of the additional installations power demand. The net electric efficiency gain in case B (related to case A) is insignificant, equal to 0.09 pp.

Implemented in case C high pressure water injection in compressor lowers the compressed gas temperature, leading to the further reduction in the compressor internal power, to the level of N_{iC} = 118.48 MW. Comparison with Case B indicates that this method allows to improve the gas turbine efficiency, accompanied by a decrease in the steam turbine power. In conclusion, the net electric efficiency of the unit in case C is lower by 0.62 pp. than in case B. Thus, the water injection in compressor is not a favorable option in the analyzed case.

Case D is distinguished from case B by cooling of the turbine blade cooling gas, which brings the reduction of cooling gas rate to the level of γ_c = 0.0777, i.e. over 2-times lower than in case A. The recovered heat from cooled gas is used in steam part, generating the additional steam turbine power equal to 3.12 MW. Relatively low temperature of cooling gas (t_{2a} = 431.7°C) is the limitation for the more effective use of the recovered heat. However, due to the lower gas turbine efficiency, the net electric efficiency dropped by 0.32 pp., relative to case B.

Case E is a modification of case A with applied cooling of the turbine blade cooling gas, so there is no cooling of the recirculated gas. The high temperature of compressed gas (t_{2a} = 628.8°C) allows to use the recovered heat more effectively than in case D, which resulted in the additional steam turbine power equal to 8.33 MW. Consequently, the net electric efficiency has improved by 0.57 pp. in relation to case A. Therefore, the analysis of cases D and E shows, that the cooling of the turbine blade cooling gas only results in improving efficiency in case of highly-efficient use of the recovered heat.

The efficiency improvement in case F is achieved by the cooling of second and third turbine sections with gas from extractions in compressor. Only the gas cooling the first turbine section is chilled, which gave the additional 4.73 MW to the steam turbine power from the use of recovered heat. The modifications incorporated in case F lead to a reduction of the compressor internal power and an increase in the gas turbine efficiency. Hence, improvement of the net electric efficiency has reached 0.9 pp., with respect to the reference case.

The oxy-combustion combined cycle units may obtain high gross electric efficiencies, exceeding 62%. However, they are characterized by large power demand of the additional installations, in particular the air separation unit. The best currently existing CCPP without carbon capture installation achieve net efficiency of about 60%. Thus, the efficiency drop due to the use of carbon capture installation in oxy-combustion technology for case F is approximately equal to 6-7 pp.

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