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Analysis of electricity generation in a integrated gasification combined cycle unit

This paper presents an analysis of the electricity generation system, which, according to the presented assumptions, is part of the integrated gasification combined cycle (IGCC). In this study, fuels with characteristics different from the ones of natural gas as well as two aspects of their use in combined cycles were considered. In the first step, a gas turbine designed for natural gas combustion (but operated in the IGCC system) was analyzed. The operation of the turbine was analyzed with or without regulation ensuring the high temperature of flue gases leaving the expander. In the next step, the whole system of electricity generation was analyzed. For this step, a suitable steam cycle structure was assumed. For different cases of system operation, the influence of steam pressure produced in three levels of heat recovery steam generator on the efficiency of the steam cycle as well as the effectiveness of the entire electricity generation system was studied. The analysis was performed for four cases that covered two different gaseous fuels, use of gas turbine regulation and use of supplementary firing.

1 Introduction

In the last decade, integrated gasification combined cycle (IGCC) units have emerged as useful alternatives for supercritical coal-fired power plants [1]. These units are used in the energy sector because of their high efficiency of conversion of primary energy into useful energy, and because of their low emissions of harmful substances [2]. Currently, with the implementation of increasingly rigorous legislation for greenhouse gas emissions control the primary advantage of IGCC

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technology is its low-energy carbon capture technology (precombustion technology) [3].

This paper addresses the use of fuels with characteristics different the ones of natural gas in the classically understood combined cycle. The results are dervied from a project investigating IGCC systems integrated with membrane carbon dioxide capture installations. The models used in this analysis will be the basis for the final model of the entire IGCC unit.

2 Characteristics of the IGCC unit

The structure of the IGCC unit that was studied can be divided into two parts: the process gas generation system, which sends gas to the combustion chamber of the gas turbine, and the electricity generation system (combined cycle). The process gas generation system consists of an air separation unit (ASU), coal gasifier, raw gas cooler, gas purification installation, shift reactor and carbon capture installation. In this paper, the process gas generation system is not the subject of analysis. Therefore, the detailed characteristics of the components of this system are not shown here. Figure 1 shows the components of the electricity generation system in more detail. In this system, there are two basic parts: the gas turbine installation and the steam cycle, the latter of which consists of a three-pressure heat recovery steam generator (HRSG), steam turbine (ST), condenser (CND) and deaerator (DA).

2.1 Gas turbine

The gas turbines used in the IGCC system are designed for natural gas combustion. For low-quality gas combustion, the construction of the machine must be slightly modified; the geometry of the combustion chamber must be adapted to the requirements of the low-energy fuel, which in turn leads to an increase in the fuel/air flow ratio. For natural gas combustion, the fuel constitutes approximately 2% of the total stream of gases flowing through the machine. During low-quality combustion in the combustion chamber this quantity is between 15–20%. Due to the higher velocity of combustion of hydrogen-rich gas in the combustion chamber, it can become problematic to maintain the flame. Additionally, due to the high temperatures accompanying hydrogen-rich gas combustion, it can become challenging to limit nitrogen oxide emissions [4].

To define the performance of a machine as part of the analyzed system, we decided to use parameters corresponding to F-class gas turbines. This class of

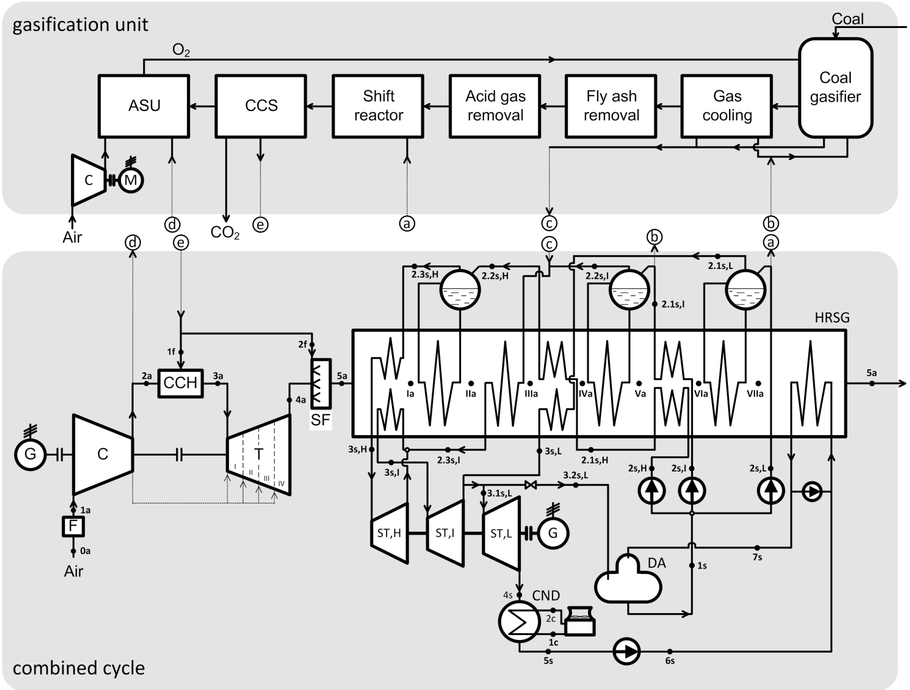

Figure 1. Diagram of the IGCC unit: ASU - air separation unit, CCS – carbon capture installation, F – filter, C – compressor, CCH – combustion chamber, T – turbine, SF – supplementary firing installation, HRSG – heat recovery steam generator, ST – steam turbine, (H – high, I – intermediate, L – low pressure) parts, CND – condenser, DA – deaerator, G – generator.

machines is widely used in power systems and most frequently offered by producers for use in IGCC systems.

The transition from the nominal fuel (natural gas) to the process gas leads to significant changes in work characteristics [5–7]. In the model of the electricity generation system, the in-house code was used for gas turbine calculations. A description of this code can be found in [8]. In Tab. 1, the basic assumptions implemented in the turbine model are presented. Assumptions were made concerning the designed operating point.

2.2 Steam cycle

Generally, steam cycles operating within IGCC systems are structured similarly to classical combined cycles. Some of the differences result from the integration

Table 1. Design-point data of the gas turbine.

Characteristic quantity	Value
Pressure ratio, –	17
Combustor exit temperature, °C	1450
Compressed air flow, kg/s	525
Isentropic efficiency of compressor, –	0.89
Isentropic efficiency of respective stages of expander, –	0.91
Mechanical efficiency of compressor, –	0.97
Mechanical efficiency of expander, –	0.97
Generator efficiency, –	0.985
Cooling air ratio, –	0.2
Cooling air ratio of the first stage blades, –	0.50
Cooling air ratio of the second stage blades, –	0.35
Cooling air ratio of the third stage blades, –	0.15
Cooling air ratio of the fourth stage blades, –	0.00
Relative inlet pressure drop, –	0.007
Relative combustor chamber pressure drop, –	0.03
Exhaust flue gas pressure, –	105

of the cycle with a process gas generation system (in Fig. 1 links: a-a, b-b, c-c). This integration involves the use of the working medium for cooling the gas generator and the gas flowing through the cooler as well as the use of part of this medium in a shift process. Additionally, in IGCC units, a supplementary firing installation is more and more often used. Such an installation is used to increase both work flexibility and flue gas temperature at the inlet of the HRSG. Combustion of gaseous fuel in the supplementary firing installation proceeds only in the atmosphere of exhaust gases and is possible due to a significant amount of oxygen in their composition.

The steam cycle model was created in GateCycle [9]. For nominal conditions, it was assumed that the live steam had three pressure profiles: 18 MPa, 5 MPa and 0.3 MPa. The temperatures of live steam produced in the high-pressure and intermediate-pressure profiles are determined in the model based on the assumed differences in temperature at the so-called hot ends of the profiles. In these cases, the differences are 30 K and 25 K, respectively. The temperature difference at the hot end of the reheater is 30 K. For the low-pressure superheater, an efficiency of 70% was assumed. Pinch-points for each of the profiles are as follows: high-pressure and intermediate-pressure, 8 K; low-pressure, 15 K. The approach point for the high-pressure profile is 10 K and 15 K for the intermediate-pressure one. The temperature difference at the hot end of the first section of the high-pressure economizer is 20 K. In the model, pressure losses were assumed for the combustion gas flows and for the working medium flow through the heat exchangers: the relative pressure loss on the combustion gas side is 0.5%, whereas on the working medium side, it is 1%. The relative heat loss to the environment was assumed to be 1%. For each part of the steam turbine, the isentropic efficiency was assumed to be 92% for the high-pressure part, 90% for the intermediate-pressure part and 88% for the low-pressure part. The pressure in the condenser was assumed to be 4 kPa, and the deaerator operating pressure was assumed to be 190 kPa. The generator efficiency and the isentropic efficiency of the pumps were assumed to be 98% and 85%, respectively.

3 Assumptions and results of calculations

This section presents the assumptions and the results of two analyses. The first analysis involved the determination of the gas turbine work characteristics in three operation cases that were described in Section 3.1. The second analysis concerned the effect of changes in live steam pressures on the efficiency characteristics of the electricity generation system. In this case, calculations were performed for four different cases.

3.1 Influence of changes in fuel characteristics on the gas turbine

Using the gas turbine model described in [8] with assumptions implemented according to Section 2.1, calculations were performed for the following three cases:

I. Gas turbine powered by natural gas with the following composition: (CH_4)= 0.9733, (C_2H_6)=0.0081, (C_3H_8)=0.0046, (C_4H_{10})=0.0026, (CO_2)=0.0028, (N_2)=0.0086. Lower heating value of 48,787 kJ/kg (nominal point of operation).

II. Gas turbine powered by process gas with the following composition: (H_2) =0.5, (N_2)=0.5. Lower heating value of 8,015 kJ/kg, without regulation in the gas turbine area.

III. Gas turbine powered by process gas with the following composition: (H_2) =0.5, (N_2)=0.5. Lower heating value of 8,015 kJ/kg, with regulation in the gas turbine area accomplished by changing the angle of the guide vanes of the compressor.

The gas used for analysis may be obtained trough: oxygen gasification of coal, cleaning of the primary gas, carbon dioxide sequestration and mixing of gas with nitrogen, obtained as a by-product of oxygen production in an ASU. In case III, it was assumed that regulation makes it possible to increase the temperature of flue gases leaving the expander to the level characteristic of the nominal operation point (nominal operation point as defined in case I). Table 2 shows the results of calculations of basic quantities that characterize turbine operation for each case.

Table 2. Basic gas turbine characteristics for three cases of operation.

Characteristic quantity	Case I	Case II	Case III
Power rate, MW	215.8	278.0	249.6
Efficiency, –	0.3728	0.4137	0.4107
Compressor internal power, MW	214.3	229.2	173.3
Expander internal power, MW	453.7	534.6	445.5
Compressed air flow, kg/s	525.0	509.4	433.1
Fuel flow, kg/s	11.87	83.86	75.83
Cooling air ratio, –	0.200	0.255	0.240
Pressure ratio, –	17.00	18.95	16.46
Compressor isentropic efficiency, –	0.8900	0.8537	0.8927
Temperature of flue gases leaving the expander, °C	588.69	557.08	588.69

The calculation results show that the fuel change results in an increase in power and efficiency of the gas turbine, especially in case II. This results mainly from a change in the relationship of the expander power and compressor power, which in turn results from a decrease in the rate of the compressed air flow and a concomitant small change in flue gas flow. In relation to the nominal operation point, the combustion of low-quality gas causes the expander blades to require a relatively large cooling air flow. In case II, an increase in the pressure ratio and a decrease in compressor isentropic efficiency are observed. In case III, the reverse relationships are observed.

To understand the interplay of the gas turbine with the steam cycle, it is important to determine the changes in gas turbine characteristics that result from a change in fuel type. As we can see in Tab. 2, many important parameters of gas turbine operation are altered, but an extremely important change is the change in the temperature of flue gases leaving the expander. For the assumptions specified in Section 2, the fuel change leads to a decrease in the flue gas temperature, from 588.69 °C to 557.08 °C (a drop of 31.61 K). The decrease in heat supplied to the steam cycle leads to a decrease in its efficiency, the importance of which is considered in Section 3.2.

3.2 Influence of change in steam pressure levels on characteristics of the electricity generation system

The decrease in temperature of flue gases leaving the expander of the gas turbine while maintaining the assumed temperature difference at the hot ends of the superheaters (Section 2.2) leads to a temperature decrease for steam produced in the HRSG. As shown in Section 3.1, the way to maintain the nominal level for the temperature of flue gases during the combustion of process gas may be to regulate the compressed air flow. The second option is the use of a supplementary firing installation. The aim of the analysis performed in this section is to determine the effects of pressure changes in the live steam generated in three HRSG profiles on the effectiveness of the entire electricity generation system and on the steam cycle. For this purpose, two indicators are defined:

- combined cycle efficiency:

$$\eta_{CC} = \frac{N_{GT} + N_{ST}}{(\dot{m}_{1f} + \dot{m}_{2f})\, LHV} \,, \tag{1}$$

  where N_{GT}, N_{ST} are respectively the power of the gas turbine and the power of the steam cycle, $\dot{m}_{1f}$, is the flow of fuel combusted in combustion chamber of gas turbine, and $\dot{m}_{2f}$ is the flow of fuel combusted during supplementary firing installation;

- steam cycle efficiency:

$$\eta_{SC} = \frac{N_{ST}}{\dot{Q}_{5a}} \,, \tag{2}$$

  where $\dot{Q}_{5a}$ is the thermal power supplied to the steam cycle.

During the analysis, the pressure levels for three profiles of HRSG, i.e., to stay consistent high-, intermediate- and low-pressure profiles, were changed. The values were varied within the following ranges: high pressure, from 10 to 20 MPa, intermediate pressure from 2 to 7 MPa and low pressure, from 0.2 to 7 MPa.

In the analysis, calculations were performed for four system operation cases:

A. Gas turbine with natural gas combustion (continuous line).

B. Gas turbine with process gas combustion; supplementary firing installation is not working; no regulation in the gas turbine – flue gas temperatures at the inlet to the HRSG are lower than the nominal temperature (dashed line).

C. Gas turbine with process gas combustion; supplementary firing maintains the nominal temperature of flue gases at the inlet to the HRSG; no regulation in the gas turbine (dash-dotted line).

D. Gas turbine with process gas combustion; supplementary firing installation is not working; regulation of the gas turbine maintains the nominal temperature of flue gases at the inlet to the HRSG (dotted line).

It should be noted that each case was analyzed as a greenfield system.

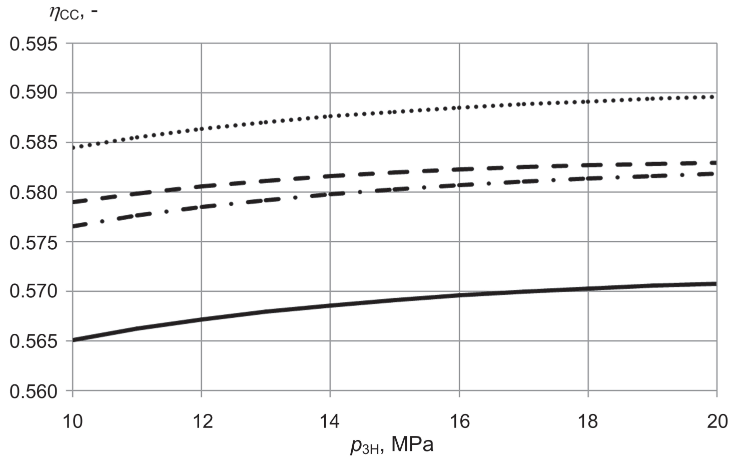

Figure 2. Combined cycle efficiency as a function of the pressure of high-pressure steam.

Combined cycle efficiencies obtained for the studied cases differed from each other regardless of the pressure level considered. The results of calculations are shown in Figs. 2, 4 and 6. The highest efficiencies were obtained for case D (with nominal pressure 0.5891 and the use of gas turbine regulation). The highest efficiency values were obtained for case B (nominally 0.5827) and case C (nominally 0.5813). The lowest values were obtained for the case involving natural gas combustion, i.e., in case A (nominally 0.5703). Therefore, regardless of the process gas combustion case considered, the combined cycle efficiencies obtained for such combustion are always higher than the efficiencies obtained for case A. As shown in Figs. 3, 5, and 7, in the cases that involve maintaining the temperature of the flue gases at the inlet to the HRSG (cases A, C and D), almost the same values of steam cycle efficiencies are obtained (overlapping lines in the figures). For nominal pressure levels, the steam cycle efficiencies for cases A, C and D are approximately 0.3352, and for case B the efficiency is 0.3201.

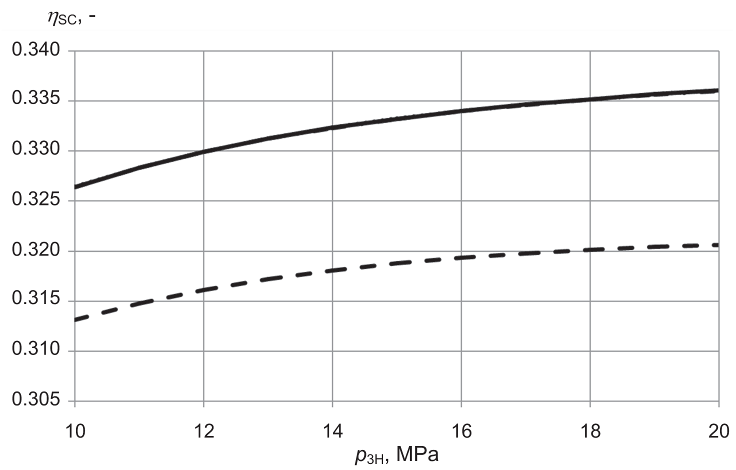

Figure 3. Steam cycle efficiency as a function of the pressure of high-pressure steam.

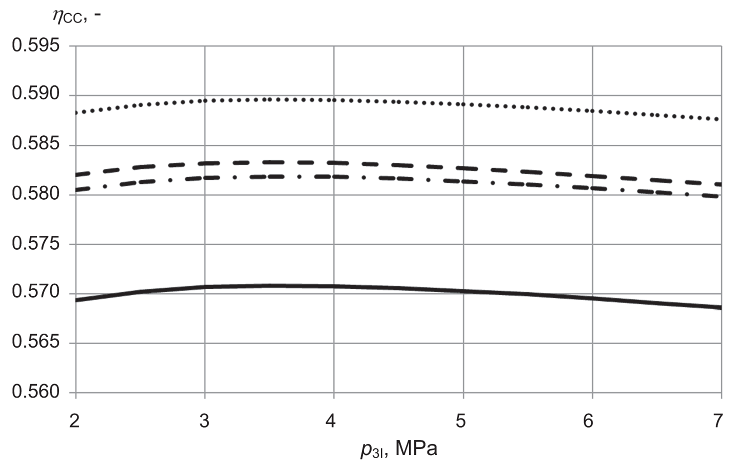

Figure 4. Combined cycle efficiency as a function of the pressure of intermediate-pressure steam.

Regardless of the case considered, an increase in the pressure of the high-pressure steam leads to increases in both the combined cycle efficiency and the steam cycle efficiency. Local extremes of efficiency were observed when examining the sensitivity of the system to changes in the pressure of intermediate-pressure steam. The extreme values for two analyzed efficiencies were obtained at a pressure of 3.6 MPa, regardless of which case was considered. In the case of a pressure increase for low-pressure steam, decreases in combined cycle efficiencies and steam cycle efficiencies were observed.

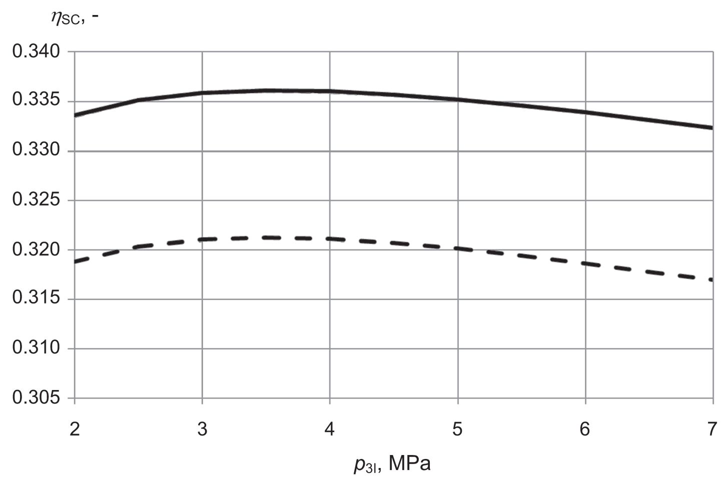

Figure 5. Steam cycle efficiency as a function of the pressure of intermediate-pressure steam.

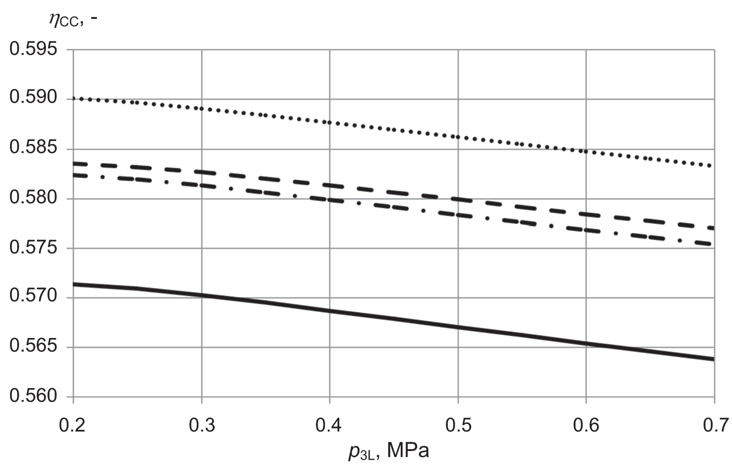

Figure 6. Combined cycle efficiency as a function of the pressure of low-pressure steam.

4 Summary

The change of fuel combusted in the gas turbine of the IGCC unit leads to changes in both gas turbine efficiency and steam cycle efficiency. These two quantities are the basis for determining the paper efficiency of the combined cycle, i.e. the efficiency of the electricity generation system of the IGCC unit. The important point is that the change of fuel (from natural gas to process gas) leads to an increase in power and efficiency of the gas turbine. From an

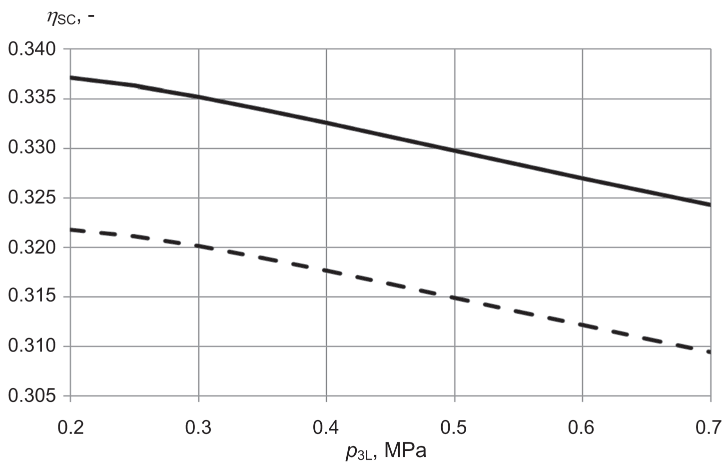

Figure 7. Steam cycle efficiency as a function of the pressure of low-pressure steam.

operational characteristics point of view, although we do not analyze it in this paper, the integration of a gas turbine with an ASU installation is also very important. Such integration leads to reduced energy consumption in the ASU installation. The analyzed models are currently used for the analysis of the entire IGCC unit. The authors are particularly interested in the area of IGCC units focused on the use of membranes for capture of carbon dioxide.

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Analiza pracy układu wytwarzania energii elektrycznej w bloku IGCC

S t r e s z c z e n i e

W artykule zaprezentowano wyniki analizy układu wytwarzania energii elektrycznej, który zgodnie z prezentowanymi założeniami jest częścią układu gazowo-parowego zintegrowanego ze zgazowaniem węgla (IGCC). W pracy omówiono dwa aspekty związane z wykorzystywaniem w obiegu kombinowanym paliwa o odmiennej charakterystyce w stosunku do gazu ziemnego. W pierwszej kolejności analizie poddano charakter pracy turbiny gazowej projektowanej dla spalania gazu ziemnego, ale wykorzystywanej w układzie IGCC. Analizowano tutaj pracę turbiny bez regulacji, jak również z regulacją mającą na celu utrzymanie na wysokim poziomie temperatury spalin opuszczających ekspander maszyny. W kolejnym etapie przeprowadzono analizę obejmującą cały układ wytwarzania energii elektrycznej. W tym celu przyjęto odpowiednią strukturę obiegu parowego. Dla różnych wariantów pracy układu badano wpływ poziomów ciśnień pary produkowanej w poszczególnych profilach kotła odzyskowego na wskaźniki efektywności pracy samego obiegu parowego, jak również całego układu wytwarzania energii elektrycznej. Analizę prowadzono dla czterech wariantów obejmujących stosowanie dwóch różnych paliw, stosowanie regulacji turbiny gazowej oraz stosowanie instalacji dopalania.