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Research and Development of the Combined Cycle Power Plants Working on Supercritical Carbon Dioxide

Andrey Rogalev [1], Nikolay Rogalev [2], Vladimir Kindra [1,*], Ivan Komarov [1] and Olga Zlyvko [1]

[1] Department of Innovative Technologies of High-Tech Industries, National Research University "Moscow Power Engineering Institute", 111250 Moscow, Russia
[2] Department of Thermal Power Plants, National Research University "Moscow Power Engineering Institute", 111250 Moscow, Russia
* Correspondence: kindra.vladimir@yandex.ru

Abstract: Today, the use of combined cycle gas turbine (CCGT) plants allows the most efficient conversion of the chemical heat of fossil fuels for generating electric power. In turn, the combined cycle efficiency is largely dependent on the working flow temperature upstream of a gas turbine. Thus, the net electric efficiency of advanced foreign-made CCGT plants can exceed 63%, whereas the net efficiency of domestic combined-cycle power plants is still relatively low. A promising method to increase the heat performance of CCGT plants may be their conversion from a steam heat carrier to a carbon dioxide one. In this paper, we have presented the results of thermodynamic research of a promising combined plant with two carbon dioxide heat recovery circuits based on the GTE-160 gas turbine plant (GTP). We have determined the pressure values that are optimal in terms of the net efficiency upstream and downstream of Brayton cycle turbines using supercritical carbon dioxide with recompression (30 and 8.5 MPa) and base version (38 and 8.0 MPa). The percentage of recompression was 32%. Based on the results of mathematical simulation of heat circuits, we have found out that the use of the solutions suggested allows the increase of the power plant's net efficiency by 2.4% (up to 51.6%).

Keywords: thermodynamic analysis; net efficiency; turbine; heat exchanger; cooler

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1. Introduction

1.1. Promising Methods to Increase the CCGT Plant Efficiency

Natural gas is currently one of the most common types of fuels: it accounts for about 23.6% of the total structure of electricity generation [1]. The most efficient conversion of the chemical energy of natural gas into electrical power is carried out by CCGT plants, in which most components of the working flow are combustion products of the air-and-fuel mixture (Brayton cycle) and water steam (Rankine cycle). The net electric efficiency of advanced CCGT plants exceeds 63%.

High efficiency figures, in particular, are demonstrated by CCGT using SGT5-9000HL gas turbine plants from Siemens, with the working flow temperature upstream of such plants being about 1550 °C and an exhaust gas temperature of 670 °C. In Russia, GTE-160 gas turbine plants are widely used, but they feature a significantly lower initial temperature of the working flow, of about 1100 °C, and an exhaust gas temperature equal to 537 °C. The net efficiency of these GTPs in a free running mode reaches 34.4%, and when operated as part of the CCP-220T CCGT power plant, it reaches 50.4%.

The most obvious method to increase the CCGT efficiency is increasing the working flow temperature upstream of the gas turbine. However, for this method to be implemented, we need to develop high-temperature technology for electric power generation [2,3].

An alternative way to increase the CCGT plant efficiency (by 2–3%) consists of the useful application of low-potential heat sources [4–6]. In particular, it is possible to add the organic Rankine cycle to CCGT plants for a deeper utilization of the exhaust gas heat or for decreasing the cold source temperature [7–9]. However, most heat carriers in low-temperature cycles (first of all, freons) are characterized by lower availability and chemical compound stability, as well as higher toxicity as compared to water. Besides, adding one more cycle entails an increase in capital costs for the power facility. The combination of the above factors was the main reason for the lack of widespread use of this technology [10,11].

1.2. Replacing the Water Stream Circuit with Carbon Dioxide for CCGT Plants

A promising method to increase the heat efficiency and to reduce the cost of power plants is the use of carbon dioxide as the working flow that works at supercritical parameters, which allows implementation of the Brayton cycle with low costs for auxiliary needs at a moderate initial temperature and compact dimensions of the main power equipment [12–14]. Today, at least five variations of carbon dioxide cycles are known [15–17]. The reasons why researchers worldwide are interested in this technology are largely related to the advantages of carbon dioxide as compared to other heat carriers.

In particular, carbon dioxide heat carriers feature low values of critical temperature (30.98 °C) and pressure (7.38 MPa). The low critical temperature of carbon dioxide, being close to the ambient temperature, makes it possible to compress the working flow near the saturation line [18,19], which reduces the compressor workload and the temperature of heat removal from the cycle without condensation of the working flow. In addition, carbon dioxide has a relatively low aggressiveness as compared to water and shows its corrosive activity only in the presence of moisture in gas or a water film on the metal surface. The price of carbon dioxide gas is comparable to that of the water heat carrier.

Long-term thermodynamic studies of S-CO_2 power facilities resulted in the development of the five cycles presented in Figure 1. The simplest S-CO_2 cycle is a closed Brayton cycle with the heat utilization of the exhaust gases (Figure 1a). It contains a compressor (C), regenerator (RH), reactor (R), turbine (T), electricity generator (G), and pre-cooler (PC). The S-CO_2 Brayton cycle with reheating is shown in Figure 1b. Here, the turbine consists of a high-pressure turbine (HPT), and a low-pressure turbine (LPT). The S-CO_2 Brayton cycle with intermediate cooling is presented in Figure 1c. The introduction of intermediate cooling allows for an increase in the cycle efficiency due to a reduction in the compressor's energy consumption. The S-CO_2 Brayton cycle with partial cooling is presented in Figure 1d. It differs from the simplest S-CO_2 Brayton cycle (Figure 1a) by application of a cooler (CR), pump (P), recompressing compressor (RC), high-temperature regenerator (HTR), and low-temperature regenerator (LTR). The use of partial cooling together with two sections of regenerators improves the regeneration system's efficiency.

With the initial working flow temperature of 550 °C, the most effective is the Brayton cycle using supercritical CO_2 with recompression. Its net efficiency is 47.3% [20]. However, it should be noted that all the cycles under consideration are characterized by a high initial temperature of heat supply, which will certainly have a negative impact on the efficiency of the waste heat boiler. In this connection, additional measures are required to reduce the temperature of gases released into the atmosphere.

Many scientific studies consider combined plants with several sequential carbon dioxide recovery cycles [20,21]. Thus, the paper [21] assesses the impact of the heat circuit in heat recovery cycles on the efficiency of a combined plant operating based on Siemens SCC5-4000F. The exhaust gas temperature of this GTP is about 580 °C. The above research has established that one of the most effective options is using circuits with the recompression Brayton cycle and the Brayton cycle with partial heating; in this case, the net efficiency of a combined plant reaches 57.9%. At the same time, this study does not include a full optimization of the basic thermodynamic parameters, whereas the analysis of initial temperatures is little relevant for the domestic power sector.

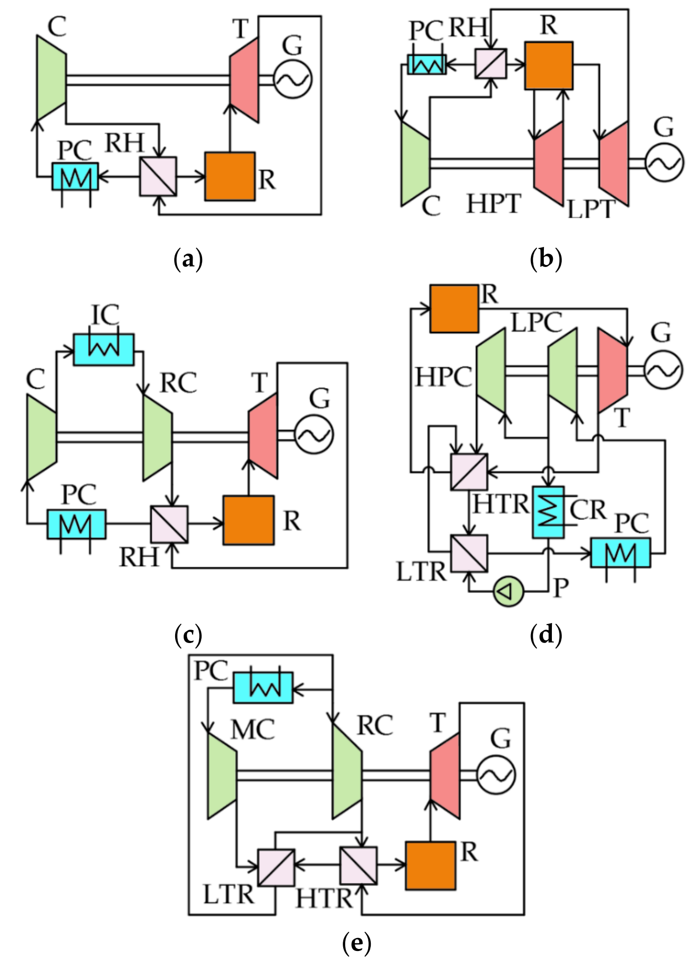

Figure 1. Supercritical CO_2 Brayton cycles: (**a**) S-CO_2 Brayton cycle with regeneration; (**b**) S-CO_2 Brayton cycle with reheating; (**c**) S-CO_2 Brayton cycle with intermediate cooling; (**d**) S-CO_2 Brayton cycle with partial cooling; (**e**) S-CO_2 Brayton cycle with recompression. C—compressor, RH—regenerator, R—reactor, T—turbine, G—electricity generator, PC—pre-cooler, HTP—high-pressure turbine, LPT—low-pressure turbine, CR—cooler, P—pump, RC—recompressing compressor, HTR—high-temperature regenerator, LTR—low-temperature regenerator, MC—main compressor, and IC—intermediate cooler.

In this paper, we have presented the results of our thermodynamic research of a promising combined plant with two carbon dioxide heat recovery circuits followed by a description of the main operation parameters of a power plant for electricity generation.

1.3. Current State of Research on Carbon Dioxide Cycles

In the modern literature, one can find many modifications of the considered basic configurations of the Brayton supercritical carbon dioxide cycle, which allow for increasing the efficiency of the power unit and expanding the applicability range [22,23].

One of the most attractive directions is the usage of the Brayton cycle with liquefied natural gas (LNG). Advantages of this technology are described in the work [24]. In particular, the use of liquefied natural gas increases the attractiveness of energy resources for maritime transport, which is relevant for plants operating in coastal areas. In addition, the use of liquefied natural gas could be used as a cooling stream. Therefore, it is advisable to use the heat generated at low potential sources, which significantly increases the efficiency of the Brayton cycle. According to the presented research results, the cycle efficiency could be equal to 52% at an inlet gas temperature of 550 °C. At the same time, the main disadvantages of this scheme are the narrow application range (it can be used only near a regasification terminal) and the expensiveness of LNG technology (it is more than twice as expensive as pipeline natural gas).

Due to the low boiling point of carbon dioxide at atmospheric pressure, the use of trigeneration cycles (plants producing heat, power, and chilled water for air conditioning or refrigeration), is becoming relevant. Particularly, the article [25] states that the advantage of trigeneration cycles is the high efficiency of the power unit up to 78% (heat cold and power energy are considered to be of equal value in calculations). The main disadvantages of this design are high complexity, poor maneuverability, and high capital costs due to additional compressor technology and heat exchangers.

The use of a pump compression process is more effective than the use of a compressor. In work [26], the Brayton cycle with condensation of heat carrier using acetone to dissolve carbon dioxide is considered. After compression in the compressor, the carbon dioxide is cooled down and dissolved in acetone. The mixture is then compressed by a pump, the carbon dioxide is separated, heated in a regenerator, and heated in a heater. Then supercritical CO_2 is expanded in the turbine and cooled in the regenerator. The cycle efficiency is 56% at the turbine inlet temperature of 1400 °C.

It is worth noting that the modifications considered earlier lead to an increase in capital costs and technological complexity of the power unit. Therefore, the use of these solutions will negatively affect the payback period, reliability, and maneuverability of the combined cycle unit. Thus, it was decided to use the simple supercritical CO_2 Brayton cycle with recompression and the basic option, allowing efficient utilization of low-potential heat.

1.4. Approaches to Energy Cycle Optimisation

Determination of the power cycle optimum parameters requires the definition of the target function, the variables, and the way of optimization. The main target function for optimization is efficiency. The main variable parameters usually are pressures and temperatures. In turn, the optimization approaches can be divided into three main categories.

Firstly, there is the equation-oriented approach. In this approach, the optimization problem and the simulation problem are solved simultaneously since the model equations are included as constraints in the problem. Thus, there is no distinction between independent and dependent variables or between model equations and technical-economic constraints. As a rule, the resulting optimization problem is solved by means of gradient-based algorithms using quadratic programming. It is worth noting that due to the large number of variables the algorithm may return local minima or even an unallowable solution if the first iterative point is far from the real value [27].

A second way to optimize energy cycles is to apply a black-box approach. In this approach, the optimization algorithm examines the space of independent design, and, for each sample solution, the simulation program develops a cycle diagram, outputting the power unit performance. The main advantage of the black-box approach is the higher probability of determining the global optimum, as the optimization only considers the independent variables and specification constraints that are hidden in the computational model. Another advantage of this approach is the ability to introduce corrective design variables to resolve potential inconsistencies. However, the black-box method has the disadvantage of omitting modeling constraints and transition conditions that are not obvious from the specification [28].

A final, common, way to optimize thermal schemes is to use the infeasible path approach. This model consists of a set of black boxes in which some equations are "open" and are included in the algorithm as constraints. To speed up the convergence of the target function in closed loops, the "open" equations are disabled (e.g., in the so-called "recycling equations" related to the recirculation of material and energy flows, as in the case of a recuperator), provided the desired accuracy is achieved. Thus, the main advantage is saving the computing time required for each black-box evaluation. On the other hand, the intermediate solutions chosen by the optimization algorithm are generally inadmissible since they do not satisfy the open constraints and feasibility is only guaranteed when the optimization algorithm achieves convergence. Furthermore, in addition to the independent decision variables, additional auxiliary variables (called "gap variables") need to be optimized, which increases the computation time of the optimization algorithm [29].

Due to the aforementioned advantages, such as simplicity and relative convergence accuracy, this research will use the black-box approach, which is widely used in energy cycle optimization [28]. On the other hand, due to the complex equations modeling the cycle components, the resulting performance figures returned by the solver may be noisy and unsmooth.

2. Materials and Methods

2.1. Research Object

The object of study in this paper is a combined power plant. The initial design was that of a traditional CCP-220T combined-cycle plant with a dual-circuit steam turbine cycle. As a promising design, we have considered a combined plant based on GTE-160 using carbon dioxide recovery Brayton cycles with recompression and base version (Figure 2).

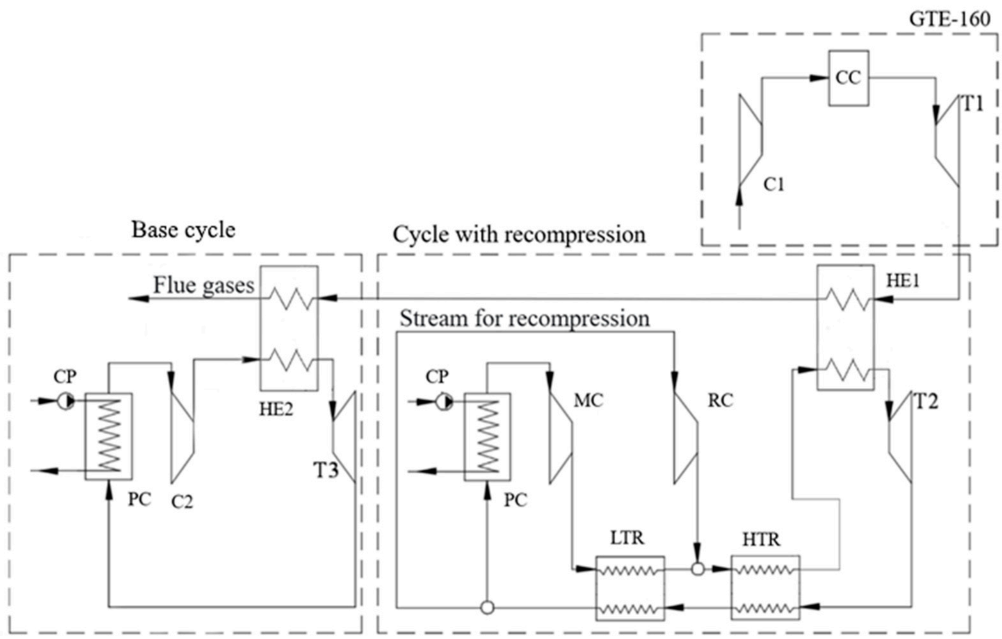

Figure 2. Combined Recompression & base version S-CO_2 Brayton cycle.

The promising power plant works as follows. Exhaust gases of GTE-160 (consisting of compressor C1, combustion chamber CC and gas turbine T1), with a temperature of 537 °C and a flow rate of 509 kg/s, are supplied to a high-temperature HE1 superheater where heat is transferred to the recompression cycle working flow, which is to carbon dioxide. After that, carbon dioxide is sent to carbon dioxide turbine T2 where the useful work is performed when the gas expands to the exhaust pressure. Then, the recompression cycle working flow is sent to a high-temperature heat regenerator HTR where heat is transferred from exhaust gases of the carbon dioxide turbine to a mixed flow downstream of the main and recompression compressors. Further on, carbon dioxide gas is supplied to a low-temperature regenerator LTR where exhaust gases, which have been previously cooled in HTR, transfer their residual heat to the flow downstream of the main compressor MC. Next, the working flow is separated: its main part is sent to the precooler PC, where the flow is cooled by circulation water (water is pumped by a circulation pump CP), and then, to the main compressor for compressing to the required pressure, whereas the other part of the flow with a higher temperature is sent directly to the recompression compressor RC. The first flow passes through low-temperature heat exchanges and is mixed with the recompression flow; after that, the full flow of CO_2 is sent to the high-temperature heat exchanger and, further on, to the high-temperature superheater 1.

Next, the exhaust gas heat is recovered in a cycle with the base set, and the heat is transferred to the carbon dioxide gas in the high-temperature HE2 superheater. The heated CO_2 is sent to the T3 carbon dioxide turbine, where the expanding working flow performs useful work. After that, the carbon dioxide gas is sent to the heat exchanger and, further on, to the precooler PC. The cooled carbon dioxide gas is sent to the C2 compressor for compression to the required pressure, after which the working flow is sent to the heat exchanger for utilization of the exhaust gas heat of the T3 carbon dioxide turbine [30].

The power equipment characteristics and invariable parameters in heat circuit plants, as used to calculate design options, are shown in Table 1.

Table 1. Main characteristics and parameters for the calculation of thermal schemes.

Parameter	Value
GT flue gas massflow, kg/s	509
GT inlet temperature, °C	1200
GT inlet pressure, MPa	12
GT pressure ratio	10.9
Heater the first cycle hot end temperature difference, °C	20
Heater the second cycle hot end temperature difference, °C	10
Regenerator temperature difference, °C	10
Turbine outlet temperature (GTE-160), °C	537
Exhaust gas mass flow (GTE-160), kg/s	509
Electrical net efficiency (GTE-160), %	34.4
The cycle minimal temperature, °C	30
Cooler circulation water pressure, bar	1.3
Turbine specific internal efficiency, %	90
Compressor specific internal efficiency, %	85
Mechanical efficiency, %	99
Power generator efficiency, %	99
Power motor efficiency, %	99
Heat transportation efficiency, %	99

2.2. Modeling Method

Aspen Plus was used for the computer simulation of the power production facilities. The working fluid thermophysical parameters were taken from the NIST Reference Fluid Thermodynamic and Transport Properties Database (REFPROP), a high-accuracy database [6]. The method for calculating thermodynamic processes is briefly explained below [21].

We have developed mathematical simulations for the recompression cycle and the cycle with base set. The exhaust gas parameters of the CCGT plant have been taken from open sources. An example of the mathematical simulation of a combined plant for electricity generation with a combined gas cycle and two carbon dioxide cycles is shown in Figure 3.

Heat exchangers have been calculated using heat and material balance equations. High-temperature and low-temperature heat exchangers in heat circuits have been simulated, taking into account the minimum temperature difference equal to 5 °C in high-temperature heaters, where the GT exhaust gas heat is recovered. The minimum temperature difference was assumed equal to 20 °C.

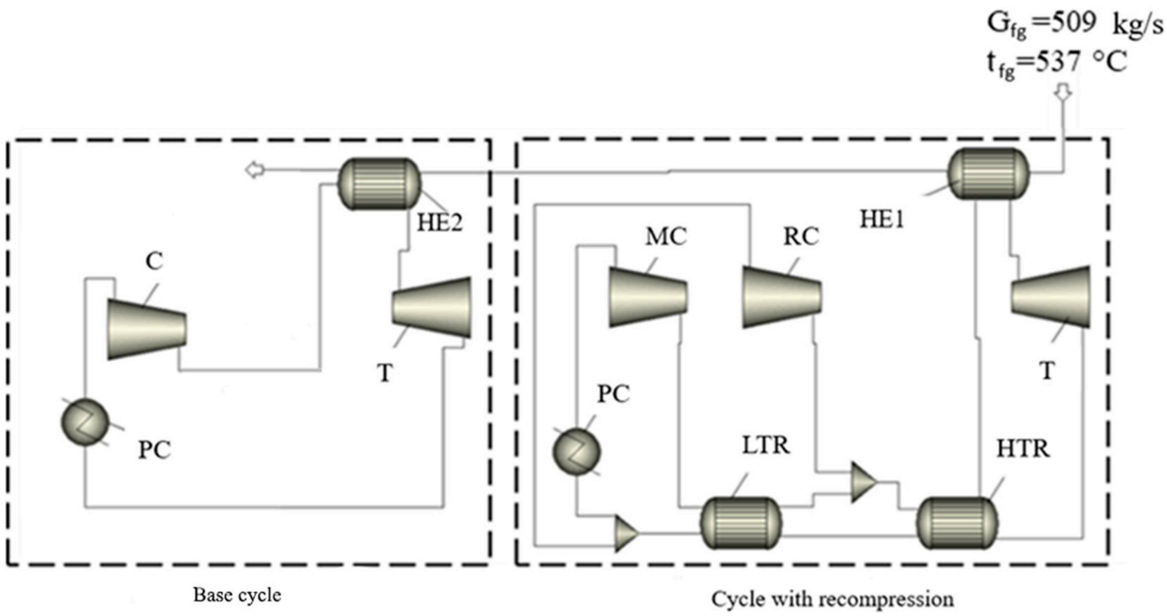

Figure 3. Combined Recompression and base set S-CO_2 Brayton cycle in Aspen Plus.

A parametric optimization of the carbon dioxide recovery cycles was carried out in a sequential direct search method. First, we varied the turbine exhaust gas pressure between 7 and 9.5 MPa. In the next step, we studied the effect on cycle performance of the degree of recompression ratio in the range 0 to 55%, as well as the values of turbine inlet pressure in the range 18 to 30 MPa. The net cycle efficiency was used as the optimization criterion. The mathematical model is a black-box.

The enthalpy at the end of the working flow expansion process in the turbine has been calculated using the following formula:

$$h_{outlet.CO_2-T} = h_{inlet.CO_2-T} - \left(h_{inlet.CO_2-T} - h_{outlet.is.CO_2-T}\right) \cdot \eta_{CO_2-T}, \quad (1)$$

where $h_{outlet.CO_2-T}$—working fluid enthalpy at the turbine outlet, kJ/kg;

$h_{inlet.CO_2-T}$—working fluid enthalpy at the turbine inlet, kJ/kg;

$h_{outlet.is.CO_2-T}$—working fluid enthalpy at the turbine outlet at isentropic expansion, kJ/kg;

η_{CO_2-T}—turbine isentropic efficiency, %.

The enthalpy at the end of the process of compression of the working fluid in the compressor was calculated using the following formula:

$$h_{outlet.C} = h_{inlet.C} + (h_{outlet.is.C} - h_{inlet.C})/\eta_C, \quad (2)$$

where $h_{outlet.C}$—working fluid enthalpy at the compressor outlet, kJ/kg;

$h_{inlet.C}$—working fluid enthalpy at the compressor inlet, kJ/kg;

$h_{outlet.is.C}$—working fluid enthalpy at the compressor outlet at isentropic expansion, kJ/kg;

η_C—compressor isentropic efficiency, %.

Heat exchangers have been calculated using heat and material balance equations. High-temperature and low-temperature heat exchangers in heat circuits have been simulated, taking into account the minimum temperature difference equal to 5 °C in high-temperature heaters, where the GT exhaust gas heat is recovered, the minimum temperature difference being taken as equal to 20 °C.

2.3. Validation of the Modelling

Calculations for the gas turbine power plant as well as for the Brayton cycle with recompression are performed by the same iterative process. In this work, we have compared the energy performance of the recompression cycle with the literature data presented in [31]. Table 2 contains the main modeling results comparison.

Table 2. Validation of Brayton supercritical cycle with recompression.

*T*min, °C	*T*max, °C	*P*max, Bar	*r*copt	Xopt	ηth, % [31]	ηth, % (This Work)	*Delt*
50	550	200	2.39	0.1837	36.71	36.93	0.599
50	550	300	2.8	0.254	38.93	39.021	0.234
32	550	200	2.64	0.3337	41.18	41.5	0.777
32	550	300	3.86	0.3549	43.32	43.17	0.346

Table 2 shows that the maximum calculation error of the mathematical model is 0.78%. This deviation is mostly due to the use of different databases of thermodynamic properties of carbon dioxide.

3. Results and Discussion

3.1. Optimisation of the CO_2 Brayton Cycle with Recompression

The influence of turbine outlet pressure on the thermal efficiency of heat recovery cycles was estimated (Figure 4). Analysis of optimization results showed that the optimal final pressure at the turbine outlet in terms of useful efficiency for the Brayton cycle with recompression is 8.5 MPa (43.41%). Deviation from the above-mentioned values of pressure behind the turbine by 0.5 MPa will lead to a decrease in cycle efficiency, on average by 1.10%.

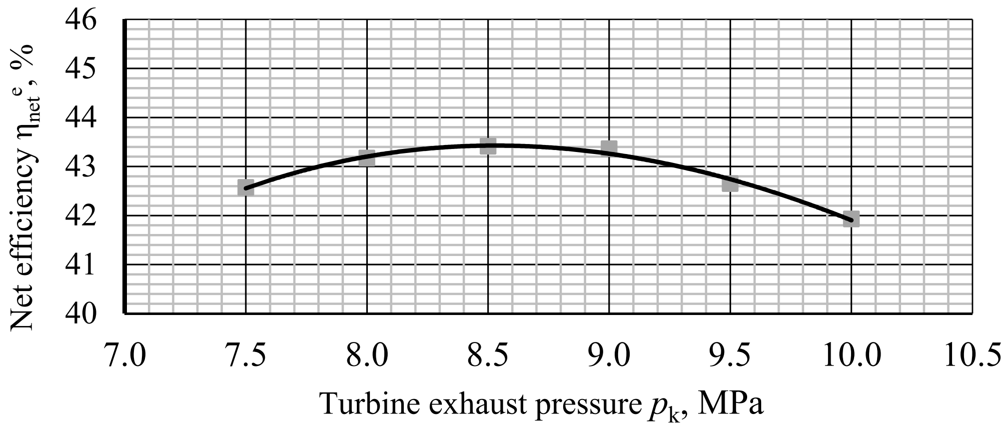

Figure 4. Dependence of the cycle net efficiency on carbon dioxide turbine exhaust pressure.

The nature of the resulting dependencies is due to the various intensity of changing the cost for compressing the working flow and the useful power generated by the turbine (Figure 5).

Optimization of Brayton's cycle with recompression showed that the maximum value of cycle productivity can be achieved with inlet pressure equal to 24 MPa and at a recompression ratio equal to 37.5%: the cycle net efficiency reaches 43.7%. Thus, a deviation from optimum values for 1 MPa is accompanied by a mean 1.16% decrease in useful cycle efficiency, and a deviation of recompression fraction from optimum values for 5% is accompanied by a 0.90% decrease in net efficiency.

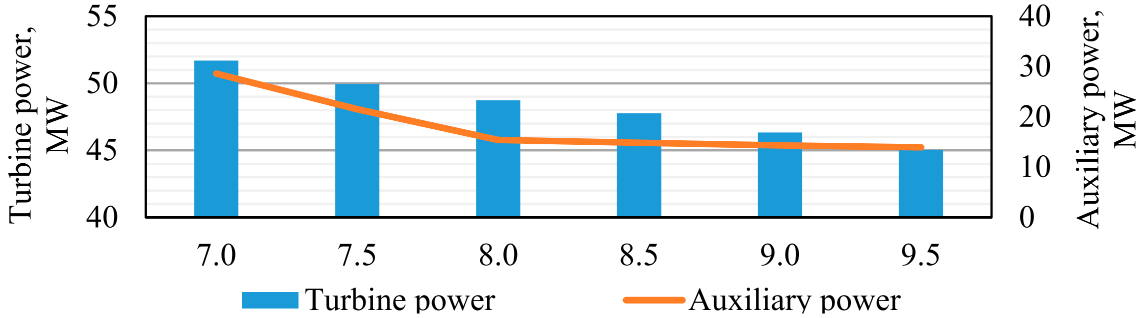

Figure 5. Dependence of the useful efficiency of the cycle on the power of the turbine and the power of auxiliary needs at the pressure at the change outlet of the carbon dioxide turbine in the cycle with the base set.

The main results of the optimization are shown in Figure 6. Decreasing the cycle efficiency with an increase in the recompression ratio is due to the rise in the total power used for the compressor drive.

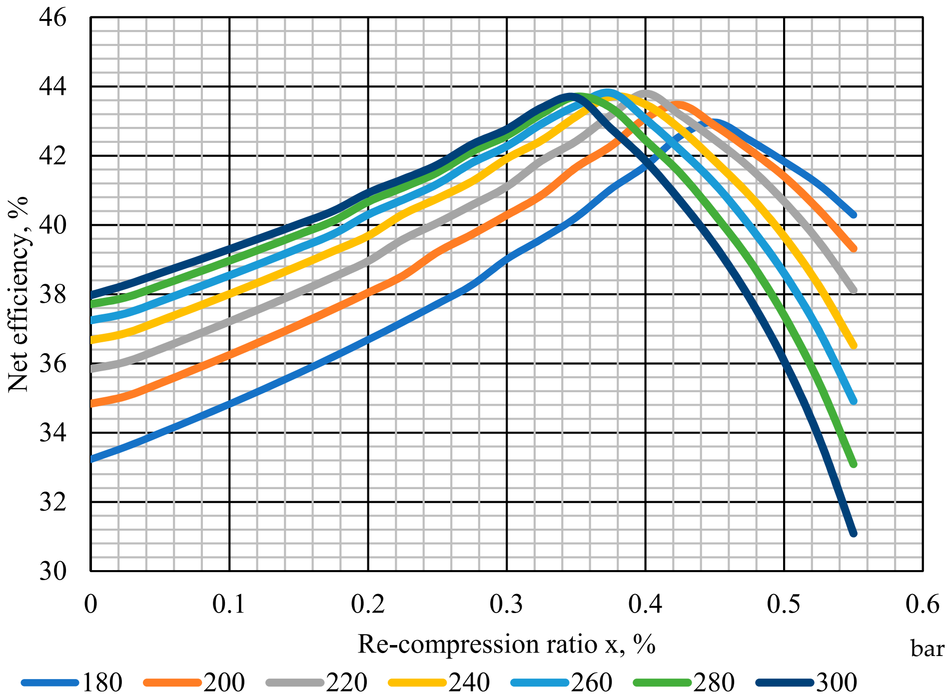

Figure 6. The effect of inlet pressure and recompression ratio on the net efficiency.

Figure 7 shows a chart of electric power consumed by the main equipment.

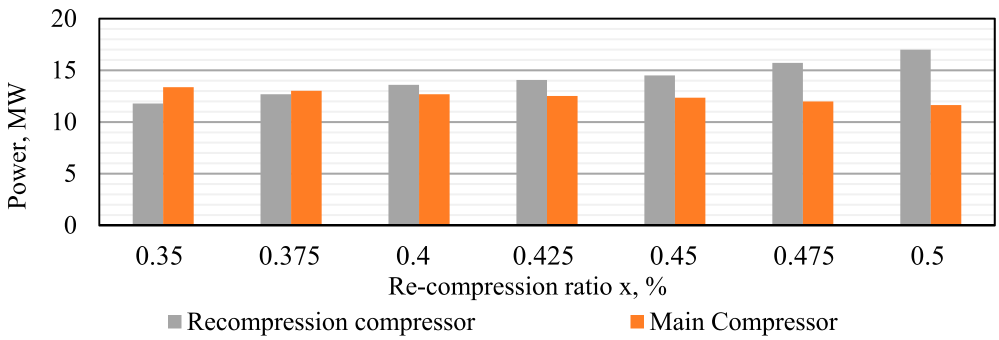

Figure 7. Technical performance comparison of recompression Brayton cycle.

The maximum power increase in the cycle is observed in the absence of recompression (Figure 8). This is due to the fact that in the absence of regeneration, the temperature of carbon dioxide flow decreases, which increases the amount of heat supplied to the cycle, reducing the temperature at the outlet of the heat recovery exchanger and increasing the amount of generated electricity. The change in waste gas temperature at the heat exchanger outlet with an increase in the recompression ratio is shown in Figure 9.

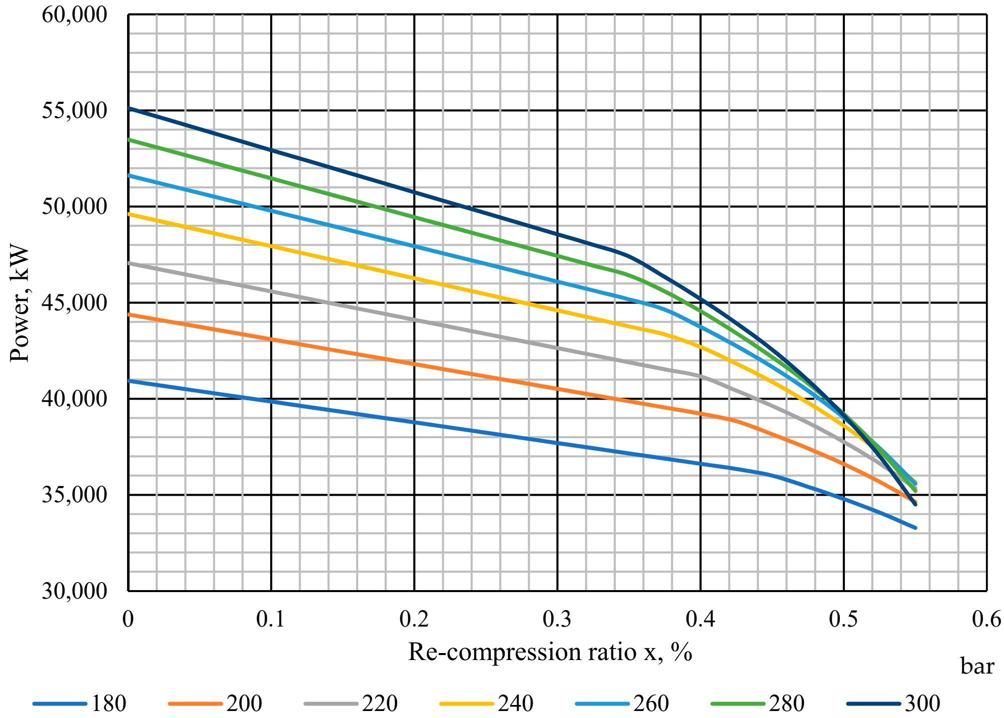

Figure 8. Dependence of cycle power on recompression ratio and turbine inlet pressure.

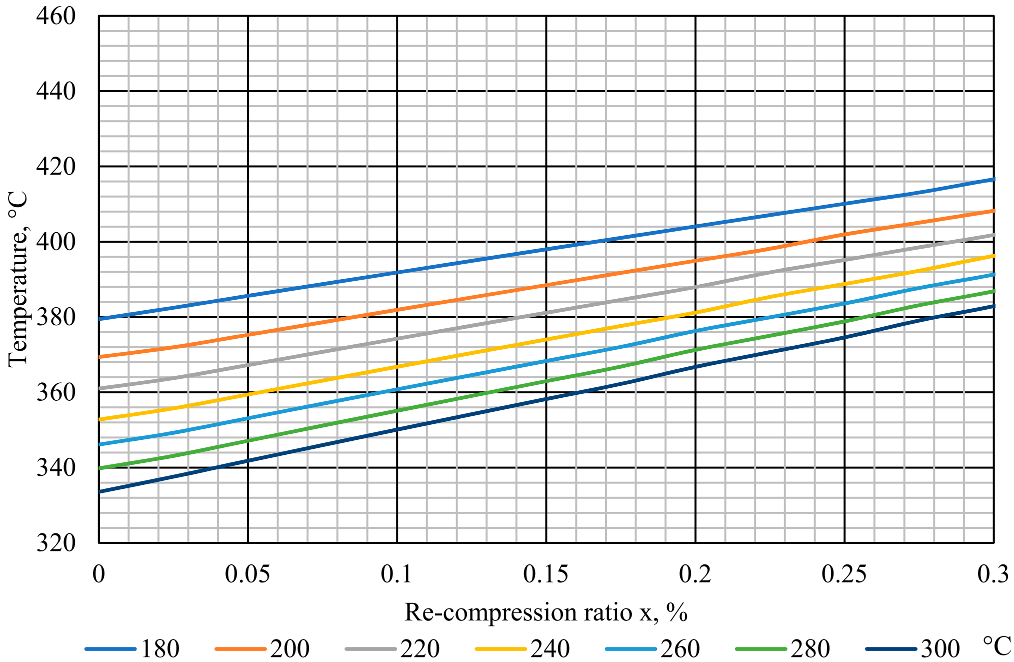

Figure 9. Dependence of exhaust gas temperature on recompression ratio and turbine inlet pressure.

Based on the graph obtained, we can conclude that it is best to use the Brayton cycle with zero recompression ratio in the combined cycle, which will allow the generation of

55 MW of electrical energy with a net efficiency of 38%. The flue gas energy from the second cycle is directed to the basic Brayton CO_2 cycle, which can reach a maximum net efficiency of 25% [21].

3.2. Optimisation of the Basic Version of the CO_2 Brayton Cycle

While modeling the basic version of the Brayton cycle, the input data is taken from the previous calculation step. When optimizing the second circuit of the combined cycle, the pressure was varied based on the gas temperature (Figure 9), which varies from 398 °C to 311 °C.

The optimization results showed that with a pressure increase from 18 MPa to 38 MPa, there was a net efficiency improvement from 14.48% to 21.95%, while the net power increased from 37.11 MW to 53.37 MW. These dependencies are illustrated in Figure 10.

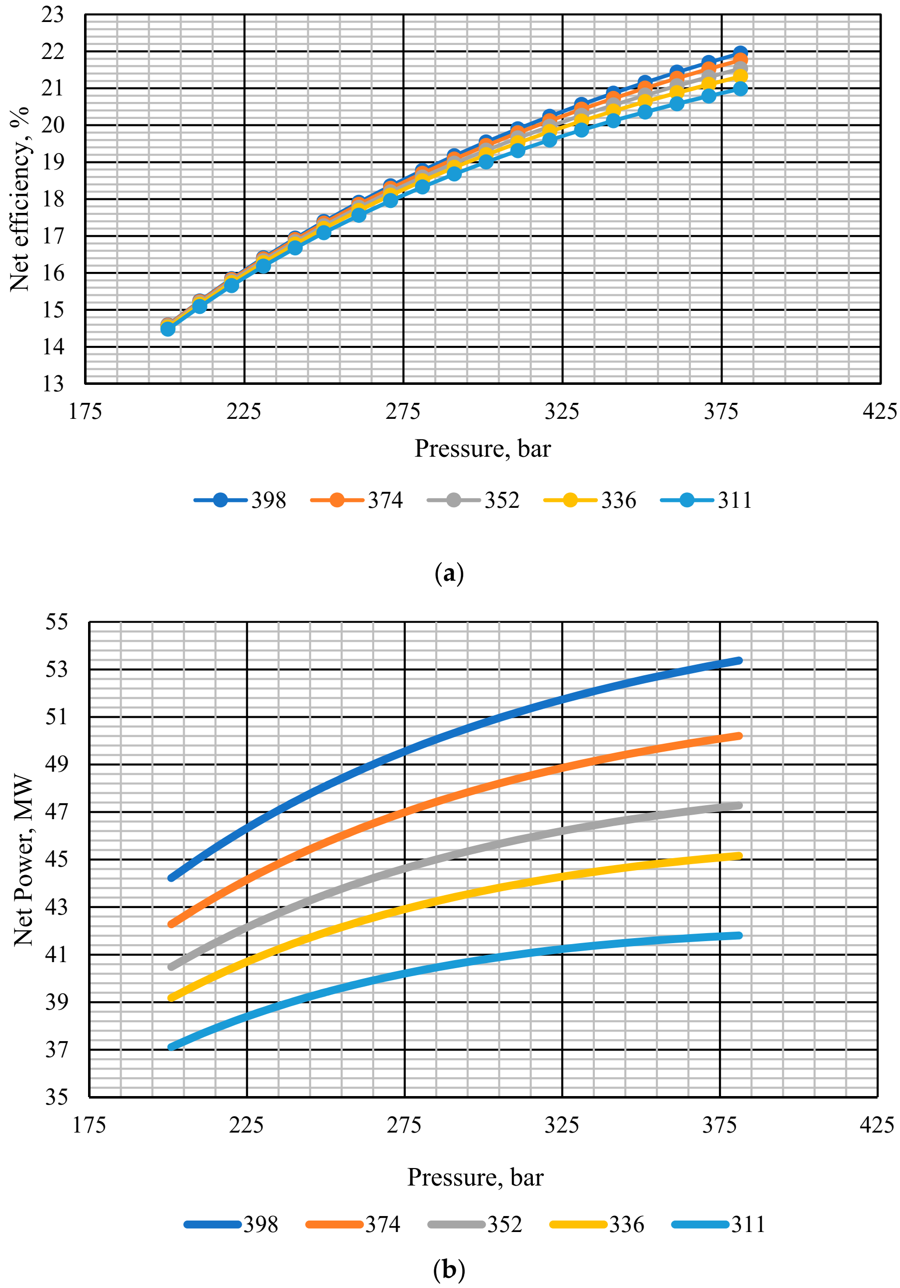

Figure 10. Investigation of the influence of pressure changes at the inlet to a carbon dioxide turbine. (**a**) Net efficiency. (**b**) Net power.

When optimizing the carbon dioxide combined cycle, the net pressure dependence of the Brayton carbon dioxide cycle with an advanced regeneration system was obtained, as shown in Figure 11.

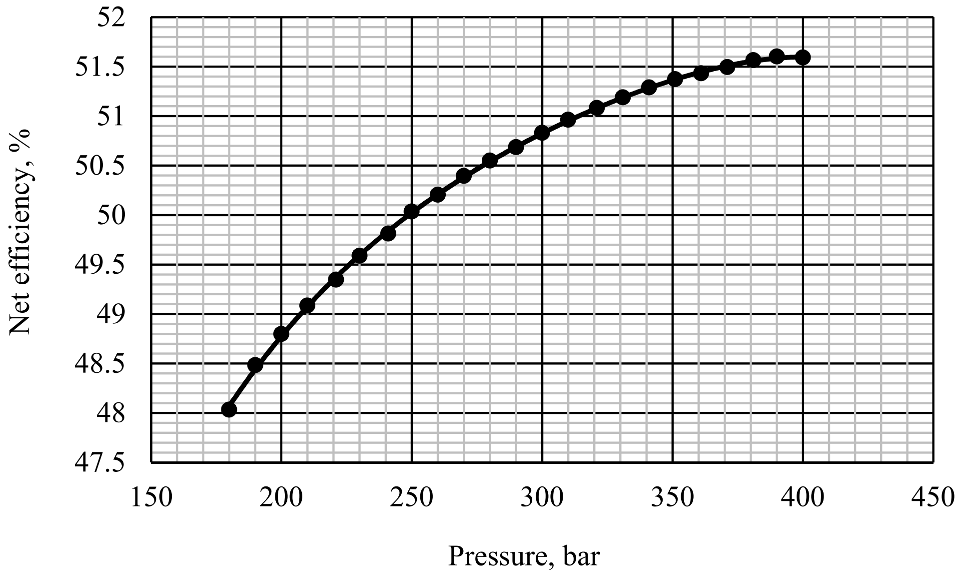

Figure 11. Dependence of the efficiency of the combined cycle on the pressure in the carbon dioxide turbine of the second cycle.

Therefore, the heat effectiveness of the promising combined plant, which includes GTE-160 gas turbine plant (34.4%), carbon dioxide Brayton cycles with recompression (37.91%), and with base version (21.98%) used for heat recovery from GTE-160 exhaust gas, is of 51.6% with the optimal parameters presented in Table 3, which is 1.2% higher than the net efficiency of PGU-220T.

Table 3. Main characteristics and parameters for the calculation of thermal schemes.

Optimized Cycle	Initial Temperature, °C	Initial Pressure, MPa	Share of Recompression, %	Final Pressure, MPa	Net Efficiency, %
Recompression cycle	517	30	0	8.5	37.91
Basic cycle version	311	32	-	8.0	21.98

An increase in energy efficiency by 2.4% is due to the fact that the electricity generated by carbon dioxide turbines is 87.7 MW, while in the combined cycle part of the PGU-220T, the electric power is 75.5 MW. This phenomenon can be explained by the high average integral temperature of heat supply to the Brayton cycle with recompression. With the same working medium temperature upstream of the turbine, the carbon dioxide temperature upstream of the high-temperature superheater of the recompression cycle is 343 °C, whereas in two-circuit CCPs, the water is supplied to the economizer after an atmospheric deaerator with a temperature of about 100 °C. An additional positive effect is that carbon dioxide remains in the supercritical region and does not change its state of aggregation when heated compared to water in a CCGT. This provides a more efficient use of waste gas heat in the waste heat boiler to heat the working flow. However, when switching to a combined energy system with carbon dioxide cycles, own needs also increase (by 41 times), which is associated with the power used to compress carbon dioxide, while the released power of the combined Brayton cycle turns out to be 6.23% higher than that of PGU-220T.

4. Conclusions

1. The thermal scheme and mathematical model for the gas turbine combined cycle working on CO_2 have been developed. The optimal values of the key thermodynamic parameters have been identified for the case of gas turbine unit GTE-160. It has been established, that at a temperature of 517 °C, the efficiency of the carbon dioxide Brayton cycle with recompression could be equal to 43.41% and achieve inlet and outlet turbine pressures of 24.0 and 8.5 MPa, respectively, and at the recompression percentage of 37.5%.
2. The maximum net power generation of 55.1 MW is achieved with a zero-recompression ratio with a net efficiency of 37.98% at a pressure of 30 MPa. This happens due to the fact that the temperature of carbon dioxide at the inlet to the waste heat boiler decreases, therefore, the heat supply to the highly efficient cycle increases.
3. When optimizing the secondary circuit, it was found that during the operation of the main Brayton cycle, the highest efficiency of the combined cycle is observed at pressures at the inlet and outlet of the turbine equal to 32 MPa and 8 MPa, at which the efficiency reaches 19.6%.
4. Based on the mathematical simulation, it was found that replacing the conventional steam power plant, which operates in combination with GTE-160 gas turbine plant and uses carbon dioxide Brayton cycles, with recompression and base version provides a 1.2% increase in the net efficiency of the combined power plant. Such an increase in efficiency can be explained by a high average integral temperature of the heat supply in the Brayton cycle, the carbon dioxide temperature upstream of HE1 being about 311 °C, whereas, in the conventional design, this value is about 100 °C (the temperature downstream of the atmospheric deaerator).

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