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First- and second-law thermodynamic analyses of a combined natural gas cycle power plant: Sankey and Grossman diagrams

Mehmet Onur KARAAĞAÇ[1,*], **Ahmet KABUL**[2], **Hasan OĞUL**[3]
[1]Department of Energy Systems Engineering, Faculty of Engineering and Architecture, Sinop University, Sinop, Turkey
[2]Department of Energy Systems Engineering, Faculty of Technology, Süleyman Demirel University, Isparta, Turkey
[3]Department of Nuclear Engineering, Faculty of Engineering and Architecture, Sinop University, Sinop, Turkey

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Abstract: The natural gas combined cycle power plant is one of the best options for generating electricity due to its use of low carbon fuels, high efficiency, and operational flexibility. These plants consist of a combination of Brayton and Rankine cycles, and investigation of these cycles is performed in this paper. Here the parameters of pressure and temperature used in the calculations are taken from a combined cycle power plant. The net power output of the system at 25 °C ambient temperature and 101.325 kPa pressure was calculated as 45 and 12 MW for Brayton and Rankine cycles, respectively. In addition, Brayton, Rankine, and combined cycle efficiencies were calculated as 37.5%, 27%, and 47.5%, while the exergy efficiencies were determined as 36%, 44%, and 46%, respectively. In the system elements, the most energy was lost in the combustion chamber and the highest exergy efficiency was achieved in the compressor at 95% level. Impacts of an increase in ambient temperature, compressor pressure ratio, and change on turbine inlet temperature were further investigated. Energy (Sankey) and exergy (Grossman) flow diagrams were further drawn based on the analyses obtained from the combined cycle power plants.

Key words: Combined cycle, Brayton cycle, Rankine cycle, energy analysis, exergy analysis, Sankey diagram, Grossman diagram

1. Introduction

The decline in fossil fuel resources and the high prices of needed energy have increased the importance of properly managing energy consumption. The combined cycle power plant is one of the best options for generating electricity due to its use of low carbon fuels, high efficiency, and operational flexibility [1]. The performance of power generation systems is based on the first law of thermodynamics (energy conservation) and second law (usability/exergy) analysis. Energy and exergy efficiencies in power generation plants are important decision parameters in designing and selecting the system as well as in determining the operating conditions of the system. The first law of thermodynamics is based on the conservation of energy and emphasizes that energy cannot be created or destroyed. Therefore, first-law analysis is performed to determine the energy inputs and outputs in the system. This law applies to all energy-consuming appliances in industrial plants and does not take into account the losses incurred by the irreversibility in the system. The second law of thermodynamics analyzes entropy generation and the ability to do work in order to understand the real energy potential of systems. Except for reversible systems, exergy is not protected like energy; some of the exergy is destroyed due

*Correspondence: okaraagac@sinop.edu.tr

to irreversibility in the system and a part of it is thrown around the system boundary [2–4]. Exergy efficiency is an indication of how close you are to an ideal system or process. In addition, exergy analysis identifies the reasons for thermodynamic inefficiencies in a system, enabling more efficient system design and accurate identification of existing conditions [4]. When the Brayton cycle, which describes the principles of real gas turbines, is solved, turbine inlet temperature (T3) and compressor pressure ratio (Prc) are found to be the design parameters affecting total work and thermal efficiency [5,6]. As the turbine inlet temperature increases, both total work and thermal efficiency increase. For this reason, it can be safely stated that increasing the turbine inlet temperature improves the performance.

Athari et al. [7] have conducted energy, exergy, and exergoeconomic analyses of the biomass (wood) integrated steam injection cycle and combined power cycle. The authors evaluated the performances of both systems under similar working conditions. Their thermodynamic analyses showed that the combined cycle at lower values of compressor pressure had higher thermodynamic efficiency but that the steam injection plant at higher pressure ratios was advantageous. Under the same conditions, more net power output is obtained in the steam injection cycle. The exergy destruction rate and cost were found to be higher for the combined cycle at all pressure ratios. In addition to Athari et al., Adibhatla and Kaushik [8] performed energy, exergy, and economy analyses of a natural-gas-fired combined cycle power plant by integrating solar energy into the steam cycle. Solar integration was achieved directly with parabolic collectors at medium temperature steam production. Some of the high-pressure feed water from Economizer-1 entered the solar collectors for preheating and evaporation. In that study, 53.79% energy and 27.39% exergy efficiencies were determined in the solar collectors. These results show a power increase of 7.84% when the plant is operated with a 50-MWth nominal solar collector design. The authors also observed that the leveled cost of electricity production fell from 7.4 to 6.7 cents/kWh. Mokhtari et al. [9] performed energy, exergy, exergoeconomic, and environmental analyses of a combined cycle power plant using the steam injection method in the combustion chamber. The injection steam was provided by the low pressure (LP) of the heat recovery steam generator (HRSG). According to the results, the steam injection increased the total combined cycle power by 2 MW, reducing the design costs to the optimum level. It was also found that exergy and thermal efficiency values at the optimum point were increased from 42% and 47.6% to 47.28% and 48.94%, respectively.

In another study, Chen et al. [10] conducted analysis of a combined cycle power plant to strengthen the cooling of the turbine inlet air during hot seasons. The power output and efficiency of the gas turbine, steam turbine, and combined cycle power plant with and without a VAIAC (turbine inlet air cooling) system under different environmental conditions were studied. A new concept for evaluating the feasibility of the VAIAC system for increasing power, taking into account ambient conditions, has been proposed: the relative humidity critical value. It is observed that the critical value of relative humidity is 79% at an ambient temperature of 30 °C, and the ambient temperature is 68% at 40 °C. Hosseini et al. [11] investigated various parameters using an additional ignition system after the gas turbine to provide optimum performance conditions in a combined cycle power generation plant. In that study, their model shows that the combined cycle power plant's efficiency increased by about 6% and CO_2 emissions fell to 5.63%. In addition, the exergy of fluid gases increased at all other points except the steam boiler. Mohtaram et al. [12] conducted a study to investigate the effect of compressor pressure on the thermodynamic performance of an ammonia–water combined cycle power plant, energy and exergy destruction, enthalpy of heat, efficiency, and flow rate. Energy/exergy analysis was carried out on the ammonia–water combined cycle and Rankine cycle. The authors used Engineering Equation Solver (EES) software to solve the detailed analyses. The results obtained from the energy/exergy analysis show that by increasing the ratio of the pressure compressor the intercooler, the gas turbine's exergy destruction, and the work produced from the gas turbine are continuously increased and the condenser's exhaust degradation is continuously reduced. When the compressor pressure ratio is not less than 7.5, the efficiencies of the gas turbine and the combined cycle are reduced since the minimum amount of fuel enters the combined cycle power plant at the beginning and then the fuel entering the combined cycle increases.

In our work, it was aimed to investigate where and how losses occur in combined power plants, the net power outputs calculated for the Brayton and Rankine cycles of the system, and the effects of the ambient temperature, the compressor pressure ratio, and the turbine inlet temperature on the performance of the system in addition to exergy and energy efficiencies. Based on the losses at the plant, energy (Sankey) and exergy (Grossman) diagrams were drawn. The first (energy) and second (exergy) law analyses of the cycles were performed by finding the thermodynamic properties of each element shown in the flow diagram, such as mass, temperature, pressure, enthalpy, entropy, exergy, and energy. All analyses and calculations were conducted using EES.

2. Combined cycle power plant

The natural-gas-fired combined cycle plant shown in Figure 1 is sucked and compressed by the air compressor at ambient conditions. The compressed air is burnt in the combustion chamber with natural gas at constant pressure. At the end of the combustion, the gas at high temperature is expanded and converted to work in the gas turbine. Gas from the exhaust of the gas turbine is used to produce steam and hot water in the heat recovery steam generator before it is thrown into the atmosphere. Here the generated superheated steam is sent to the steam turbine to turn the turbine. The resulting mechanical energy drives the generator with a shaft and electrical energy.

2.1. Thermodynamic analyses

For the evaluation and development of thermal systems, the interaction between the source and system components of thermodynamic inefficiencies needs to be understood. The total energy of a closed system can only change around as a result of heat or work interaction. Accordingly, the total energy change during a state change of a closed system is equal to the net heat and work transition at system boundaries [13]. All energy conversion processes are irreversible due to the extreme temperature effects that occur during heat transfer, the mixing of substances in different compositions or phases, and sweaty effects such as uncontrolled expansion and friction. Exergy balances help to estimate the exergy destruction in system components. Thus, the thermodynamic inefficiency and reasons for the inefficiencies are defined.

Acceptances during the system analysis:

All system components are in stable regime conditions.

- Air and exhaust gases are considered ideal gases.
- Combustion reactions are completed in the combustion chamber.
- Kinetic and potential energy changes are neglected.
- Compressed air is assumed to enter the environment.
- Since more than 98% of the natural gas fuel is methane, the fuel used is methane.

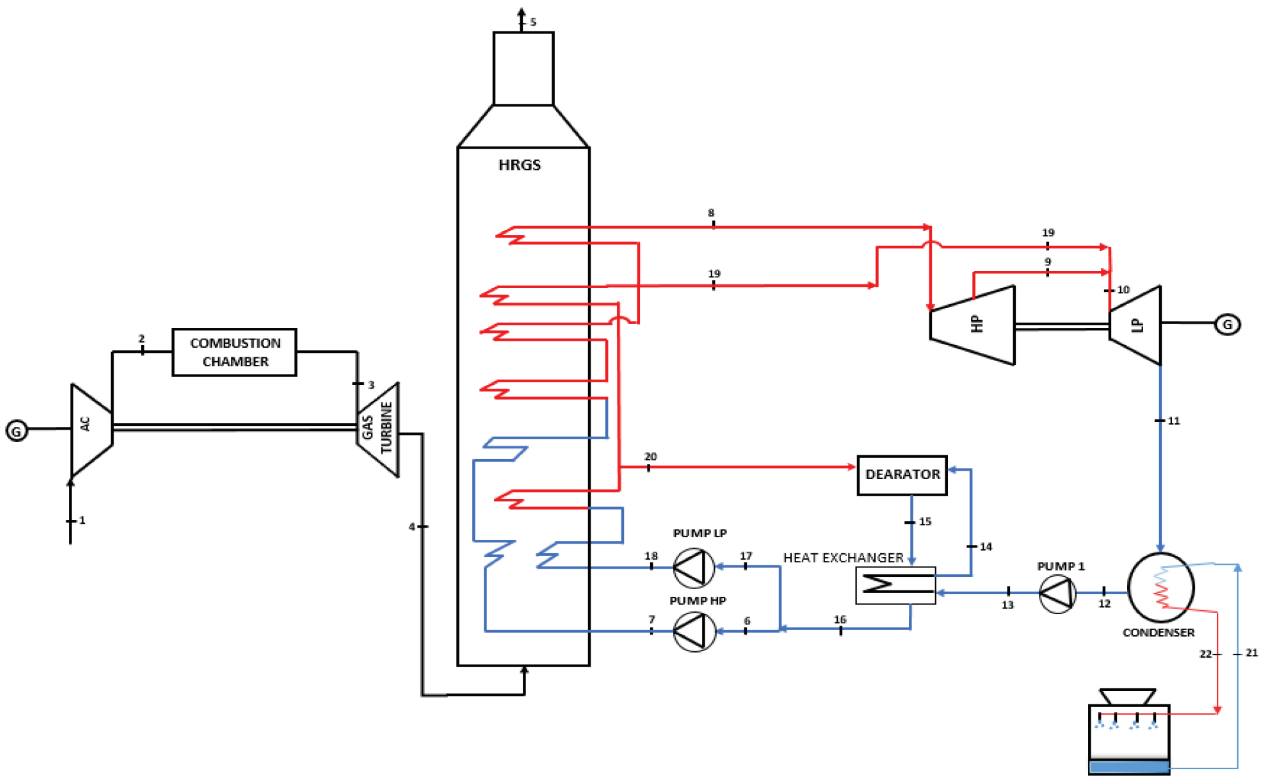

Figure 1. Combined power system with HRSG.

For any stable regime system, the mass, energy, and entropy balance based on the above assumptions can be written as below. For continuous flow open systems, the principle of conservation of mass according to the first law of thermodynamics is expressed as in Eq. (1).

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

Here $\dot{m}$ is the mass of the fluid. For energy balance, the first law of thermodynamics is applied to each element of the system and energy balance is expressed by Eq. (2).

$$\dot{Q} + \sum (\dot{m}h)_{in} = \dot{W} + \sum (\dot{m}h)_{out}, \tag{2}$$

where Q is the thermal energy, W is the work, and h is the enthalpy. For any system that undergoes any state change, the entropy balance can be expressed as

$$\sum \dot{S}_{in} - \sum \dot{S}_{out} + \dot{S}_{gen} = \Delta \dot{S}_{system}/dt, \tag{3}$$

entropy transfer amount related to heat transfer,

$$\dot{S}_{heat} = \dot{Q}/T, \tag{4}$$

and the entropy transfer amount related to mass flow,

$$\dot{S}_{mass} = \dot{m}s. \tag{5}$$

For the exergy analysis of the system, the following expression is used, neglecting the kinetic, potential, and chemical exergy:

$$\dot{E}x_{heat}+\dot{W}=\sum\dot{m}_{out}\varepsilon_{out}-\sum\dot{m}_{in}\varepsilon_{in}+\dot{E}x_{dest} \tag{6}$$

$$\dot{E}x_{loss}=T_0\dot{S}_{gen} \tag{7}$$

Ex_{des} is exergy destruction rate, and Ex_{heat} is the net exergy transfer at temperature T through heat as follows:

$$\dot{E}x_{heat}=\sum\left(1-\frac{T_0}{T}\right)\dot{Q} \tag{8}$$

Here T is the temperature involved in the actual heat transfer. The partial flow exergy rate and total exergy rate are given as follows:

$$\varepsilon=(h-h_0)-T_0\,(s-s_0)\,, \tag{9}$$

$$\dot{E}x=\dot{m}\varepsilon, \tag{10}$$

where 0 represents the reference conditions and T_0 is the reference state temperature energy, and exergy efficiency can be specified as follows:

$$\eta=\frac{E_{in}}{TotalE_{in}} \tag{11}$$

$$\eta_{II}=\frac{Exergyout}{TotalExergy} \tag{12}$$

2.2. Chemical exergy

It is not a matter of protecting the molar amount during the chemical reaction. The fuel–air ratio (λ), which is a parameter used in combustion processes to determine the amount of air and fuel, is the ratio of the molar amount of fuel to the molar amount of air.

$$\lambda=\frac{n_f}{n_A}=\frac{fuelmolefraction}{airmolefraction} \tag{13}$$

In the combustion chamber, it is assumed that the combustion is complete. Fuel (CH_4) theoretical combustion:

$$\lambda CH_4+[0.7748N_2+0.2059O_2+0.003CO_2+0.019H_2O]$$

$$\rightarrow|1+\lambda|\,[x_{N_2}N_2+x_{O_2}O_2+x_{CO_2}CO_2+x_{H_2O}H_2O]$$

Expressions giving molar ratios of combustion products (carbon, hydrogen, oxygen, and nitrogen) [14]:

$$\begin{gathered} x_{N_2}=\frac{0.7748}{1+\lambda}\quad x_{O_2}=\frac{0.2059-2\lambda}{1+\lambda}\\ x_{CO_2}=\frac{0.003+\lambda}{1+\lambda}\quad x_{H_2O}=\frac{0.019+2\lambda}{1+\lambda}\end{gathered} \tag{14}$$

$$0=\dot{Q}_{KH}+n_fh_f+n_Ah_A-n_{cp}h_{cp} \tag{15}$$

The ideal gas-mixture principles and the enthalpies of air and combustion products can be calculated using the equations described below:

$$h_A = [0.7748 h_{N2} + 0.2059 h_{O2} + 0.003 h_{CO2} + 0.019 h_{H_2O}]\,(T_2) \tag{16}$$

$$(1+\lambda)\,h_f = [0.7748 h_{N2} + (0.2059 - 2\lambda)\,h_{O_2} + (0.003)\,h_{CO_2} + 0.019 h_{H_2O}]\,(T_3) \tag{17}$$

To calculate the mass flows of the fuel and air, the following equation is used:

$$m_f = \lambda \left(\frac{M_f}{M_A}\right) m_A \tag{18}$$

Here m_f, m_a, and h_f stand for fuel molecular weight, air molecular weight, and enthalpy of the fuel at 298.15 K (25 °C) and 1 bar pressure, respectively.

Or it can be stated as follows:

$$Q_G = m \cdot {}_3 h_3 - (m \cdot {}_2 h_2 + m \cdot {}_f LHV) \tag{19}$$

Here *LHV* represents the fuel subthermal value. For the determination of chemical exergy and energy, an approach was developed by Coskun et al. [15]. To use this approach, we need to know the elemental analysis of natural gas. In the literature [14,16], it is stated that a large part of the natural gas is CH_4; therefore, the combustion can be regarded as CH_4. The standard chemical exergy of CH_4 at 298.15 K and 1.0019 atm pressure can be taken as 830,348 (kJ/kmol).

$$Ex_f = m_f \frac{e_{CH_4}}{M_{CH_4}} \tag{20}$$

e_{CH_4} is the standard chemical exergy of CH_4 and M_{CH_4} is the molar mass of CH_4.

$$e_f = \xi x LHV \tag{21}$$

The chemical exergy of the fuel can be calculated by taking $\xi = 1.06$ [14,17].

Thermodynamic analysis of a thermal system usually involves evaluating the performance of system components separately. Table 1 summarizes all of the balance equations we used in the calculations.

3. Results and discussion

In this study, air-fired fuel enters the turbine at 1200 °C and exits at 460 °C in the combustion chamber of the natural gas combined cycle plant designed at an ambient temperature of 15 °C and pressure of 101.325 kPa. The exhaust gas from the gas turbine exits from the shaft at 101 °C by transferring heat to the Rankine cycle in the heat recovery steam generator. As can be seen from the general system data given in Table 2, the Rankine cycle operates at two different pressures: high and low. The hot steam in the high pressure line begins to rotate the turbine blades by entering the turbine at the first stage. The low pressure line enters the turbine at the fourth stage. The high pressure and low pressure lines join here to transfer the energy to the turbine blades. The resulting mechanical energy drives the generator shaft to generate electrical energy.

Table 3 shows the mass, temperature, pressure, enthalpy, and energy values of the combined cycle power plant using the real work data and analyzed by EES for each point. The thermodynamic analysis of the combined

Table 1. Balance equations of system components of the combined cycle power plant.

System units	Mass balance equations	Energy balance equations	Exergy balance equations	Entropy balance equations	Exergy efficiency
Compressor	$\dot{m}_1 = \dot{m}_2$	$\dot{m}_1.h_1 + W_C = \dot{m}_2 h_2$	$E_{x1} + W_C = E_{x2} + E_{Xdest}$	$\dot{m}_1 s_1 + S_{gen} = \dot{m}_2 s_2$	$\eta_{II} = \frac{E_{x2} - E_{x1}}{W_C}$
Combustion chamber	$\dot{m}_2 + \dot{m}_f = \dot{m}_3$	$\dot{m}_2.h_2 + \dot{m}_f.h_f = \dot{m}_3 h_3$	$E_{x2} + E_{Yfuel} = E_{x3} + E_{Xdest}$	$\dot{m}_2 s_2 + S_{production} + \frac{Q_G}{T_K} = \dot{m}_3 s_3$	$\eta_{II} = \frac{E_{x3}}{E_{x2} + E_{fuel}}$
Gas turbine	$\dot{m}_3 = \dot{m}_4$	$\dot{m}_3.h_3 = \dot{m}_4 h_4 + W_T$	$E_{x3} = E_{x4} + W_T + E_{Xdest}$	$\dot{m}_3 s_3 + S_{production} = \dot{m}_4 s_4$	$\eta_{II} = \frac{W_T}{E_{x3} - E_{x4}}$
HRSG	$\dot{m}_4 = \dot{m}_5$ $\dot{m}_7 = \dot{m}_8$ $\dot{m}_{18} = \dot{m}_{19} + \dot{m}_{20}$	$\dot{m}_4 h_4 + \dot{m}_7 h_7 +$ $\dot{m}_{18} h_{18} = \dot{m}_5 h_5 + \dot{m}_8 h_8 +$ $\dot{m}_{19} h_{19} + \dot{m}_{20} h_{20} + Q_L$	$E_{x4} + E_{x7} + E_{x18} = E_{x5} +$ $E_{x8} + E_{x19} + E_{x20} +$ $Q_L\left(1 - \frac{T_0}{T_K}\right) + E_{xdest}$	$\dot{m}_4 s_4 + \dot{m}_7 s_7 + \dot{m}_{18} s_{18} + S_{gen} =$ $\dot{m}_5 s_5 + \dot{m}_8 s_8 + \dot{m}_{19} s_{19} + \dot{m}_{20} s_{20} + \frac{Q_G}{T_K}$	$\eta_{II} = \frac{(E_{x8} - E_{x7}) + (E_{x19} - E_{x18})}{E_{x4} - E_{x5}}$
Pump LP	$\dot{m}_{17} = \dot{m}_{18}$	$\dot{m}_{17}.h_{17} + W_{PLP} = \dot{m}_{18} h_{18}$	$E_{x17} + W_{PLP} = E_{x18} + E_{Xdest}$	$\dot{m}_{17} s_{17} + S_{gen} = \dot{m}_{18} s_{18}$	$\eta_{II} = \frac{E_{x18} - E_{x17}}{W_P}$
Pump HP	$\dot{m}_6 = \dot{m}_7$	$\dot{m}_6.h_6 + W_{P2} = \dot{m}_7 h_7$	$E_{x6} + W_{PHP} = E_{x7} + E_{Xdest}$	$\dot{m}_6 s_6 + S_{gen} = \dot{m}_7 s_7$	$\eta_{II} = \frac{E_{x7} - E_{x6}}{W_P}$
Pump 1	$\dot{m}_{12} = \dot{m}_{13}$	$\dot{m}_{12}.h_{12} + W_{P1} = \dot{m}_{13} h_{13}$	$E_{x12} + W_{P1} = E_{x13} + E_{Xdest}$	$\dot{m}_{12} s_{12} + S_{gen} = \dot{m}_{13} s_{13}$	$\eta_{II} = \frac{E_{x13} - E_{x12}}{W_P}$
HP	$\dot{m}_8 = \dot{m}_9$	$\dot{m}_8.h_8 = \dot{m}_9 h_9 + W_{HP}$	$E_{x8} = E_{x9} + W_{HP} + E_{Xdest}$	$\dot{m}_8 s_8 + S_{gen} = \dot{m}_9 s_9$	$\eta_{II} = \frac{W_T}{E_{x8} - E_{x9}}$
LP	$\dot{m}_{10} = \dot{m}_{11}$	$\dot{m}_{10}.h_{10} = \dot{m}_{11} h_{11} + W_{LP}$	$E_{x10} = E_{x11} + W_{LP} + E_{Xdest}$	$\dot{m}_{10} s_{10} + S_{gen} = \dot{m}_{11} s_{11}$	$\eta_{II} = \frac{W_T}{E_{x10} - E_{x11}}$
Condenser	$\dot{m}_{11} = \dot{m}_{12}$ $\dot{m}_{21} = \dot{m}_{22}$	$\dot{m}_{11} h_{11} + \dot{m}_{21} h_{21} = \dot{m}_{12} h_{12} +$ $\dot{m}_{22} h_{22} + Q_L$	$E_{x11} + E_{x21} = E_{x12} + E_{x22} +$ $Q_{in}\left(1 - \frac{T_0}{T_K}\right) + E_{xdest}$	$\dot{m}_{11} s_{11} + \dot{m}_{21} s_{21} + S_{gen} = \dot{m}_{12} s_{12} +$ $\dot{m}_{22} s_{22} + \frac{Q_{in}}{T_K}$	$\eta_{II} = \frac{E_{x22} - E_{x21}}{E_{x11} - E_{x12}}$
Deaerator	$\dot{m}_{14} + \dot{m}_{20} = \dot{m}_{15}$	$\dot{m}_{14} h_{14} + \dot{m}_{20} h_{20} = \dot{m}_{15} h_{15} +$ Q_L	$E_{x14} + E_{x20} = E_{x15} +$ $Q_{in}\left(1 - \frac{T_0}{T_K}\right) + E_{xdest}$	$\dot{m}_{14} s_{14} + \dot{m}_{20} s_{20} + S_{gen} = \dot{m}_{15} s_{15}$	$\eta_{II} = \frac{E_{x15}}{E_{x14} + E_{x20}}$
Heat exchanger	$\dot{m}_{13} = \dot{m}_{14}$ $\dot{m}_{15} = \dot{m}_{16}$	$\dot{m}_{13} h_{13} + \dot{m}_{15} h_{15} = \dot{m}_{14} h_{14} +$ $\dot{m}_{16} h_{16} + Q_L$	$E_{x13} + E_{x15} = E_{x14} + E_{x16} +$ $Q_{in}\left(1 - \frac{T_0}{T_K}\right) + E_{xdest}$	$\dot{m}_{13} s_{13} + \dot{m}_{15} s_{15} + S_{gen} = \dot{m}_{14} s_{14} +$ $\dot{m}_{16} s_{16} + \frac{Q_{in}}{T_K}$	$\eta_{II} = \frac{E_{x15} - E_{x16}}{E_{x14} - E_{x13}}$

Table 2. General data of the combined power plant system.

Parameter	Unit	Value
Gas turbine model	-	LM 6000
Fuel type	-	Natural gas
Turbine inlet temperature	°C	1200
Turbine outlet temperature	°C	460
Pressure ratio	-	1/29
HP, AIK outlet pressure	bar	57
LP, AIK outlet pressure	bar	4.2
Exhaust gas AIK input pressure	bar	1.013

Table 3. Point values of the combined cycle plant.

Points	m (kg/s)	T (°C)	P (kPa)	h (kJ/kg)	S kJ/(kg K)	ε kJ/(kg K)	E (kW)
1	133	25	99.28	293.4	6.849	0	39,013
2	133	532.2	2920	829.4	6.924	512.2	110,282
3	135.4	1200	2820	1607	7.634	1081	217,430
4	135.4	458.1	101.3	747.5	7.785	177.9	101,169
5	135.4	100	101.3	374.1	7.087	9.348	50,630
6	11.92	63.62	120	266.4	0.8763	12.38	3175
7	11.92	65	5561	276.7	0.8904	18.54	3298
8	11.92	421.2	5561	3238	6.662	1288	38,597
9	11.92	223.8	423	2909	7.244	787.7	34,670
10	15.25	223.8	423	2909	7.244	787.7	44,355
11	15.25	44.48	7.7	2334	7.474	146.2	35,599
12	15.25	44.48	7.7	170.8	0.5829	2.853	2605
13	15.25	46	420	193	0.6515	4.873	2943
14	15.25	95.1	320	398.6	1.251	34.74	6078
15	19.5	107	140	448.6	1.385	45.44	8749
16	19.5	63.62	120	266.4	0.8763	12.38	5194
17	7.58	63.62	120	266.4	0.8763	12.38	2019
18	7.58	65	836	272.7	0.893	13.85	2067
19	3.33	223.8	423	2909	7.244	787.7	9685
20	4.25	155	600	654	1.893	102	2779
21	1053	27.6	400	116	0.4032	0.7042	122,153
22	1053	35.3	300	148.1	0.5089	1.813	155,972

cycle system was carried out by using these equations and the thermodynamic balance equations derived for each element. As a result of the analyses performed, 25 °C ambient temperature and 101.325 kPa pressure gas turbine power, steam turbine power, gas turbine efficiency, and net electrical efficiency are given in Table 4.

The entropy (3) and exergy (6) balance equations are applied for the system considering the point properties given in Table 3 and the exergy destruction and entropy generation results obtained for the system

Table 4. General output data of the combined cycle plant.

System component	Value	Unit
GT	45	MW
ST	12	MW
Total power	57	MW
Combined cycle efficiency	47.5	%
GT efficiency	37.5	%
ST efficiency	27	%
HRSG efficiency	0.9	%

components are given in Table 5. As seen in Table 5, the system element in which exergy destruction is found most frequently was the combustion chamber. After the combustion chamber, the gas turbine, steam turbine, compressor, heat recovery steam generator, condenser, pump 1, heat exchanger, deaerator, pump LP, and pump HP are seen, in that order. Due to exergy destruction, the highest entropy generation of the system components occurred in the combustion chamber.

Table 5. Exergy destructions and entropy generations of the system elements.

System component	Ex_{dest} (kJ)	S_{gen} (kJ/K)
Compressor	2978	9.99
Combustion chamber	49,850	180
Gas turbine	6119	20.52
HRSG	1857	6.229
Pump LP	49.95	3.506
Pump HP	37.78	0.1675
Pump 1	312	1.046
HP steam turbine	2070	6.943
LP steam turbine	1045	0.1266
Condenser	1049	6.334
Deaerator	53.42	0.1792
Heat exchanger	120	0.4026

The efficiency factors of the system components of the combined cycle power plant are calculated using Eq. (12) and are given in Table 6. As a result of the second-law analysis of the system elements, the most exergy efficient gas turbine compressor was then used: combustion chamber, heat exchanger, condenser, steam turbine, combined cycle, and gas turbine.

3.1. Effect of ambient temperature

In this section, it is assumed that the ambient temperature changes from 0 °C to 45 °C, while all other parameters remain constant (Table 2). The change in gas turbine, steam turbine, and combined cycle power plant with increasing ambient temperature from 0 °C to 45 °C is shown in Figure 2. As can be seen, the increase in ambient temperature of 45 °C caused a 21.2% decrease in gas turbine power and a 17% decrease in combined cycle power

Table 6. Exergy efficiencies of the combined cycle plant system components.

System component	η_{II}
Compressor	0.95
Combustion chamber	0.74
Heat exchanger	0.7
Steam turbine	0.44
Gas turbine	0.367
Condenser	0.53
Combined cycle	0.46

plant power, almost without affecting the generated power in the steam turbine. The power reduction in the gas turbine is due to the increase in the specific volume with the temperature of the air entering the compressor, the increase in the energy consumed in the resultant compressor, and the decrease in the generated power with the decreasing flow of heated air ($\dot{m} = \rho v$). In this case, the mass flow of the air that circulates the gas turbine with increasing temperature is reduced and so the amount of power that can be taken from the turbine is also reduced. As the ambient temperature increases, the temperature of the air entering the combustion chamber increases, resulting in an increase in the temperature of the exhaust gases, resulting in a 0.06% increase in power taken from the steam turbine as more steam is supplied to the steam boiler.

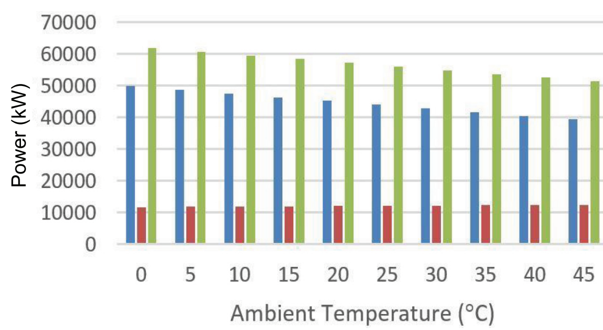

Figure 2. Change in the generated power with ambient temperature.

The effect of change in ambient temperature on the gas turbine, steam turbine, and combined cycle power plant efficiency is given in the left plot in Figure 3. As the ambient temperature increases, the gas turbine and combined cycle power plant efficiency are found to be decreased. The reason for the decrease in yield with ambient temperature is the increase in the specific volume of air entering the system by increasing the ambient temperature from 0 °C to 45 °C and consequently the increase in power consumption in the compressor.

This decrease affects the efficiency of the gas turbine as well as the generated power from the turbine directly. As can be seen from the graph, the changing ambient temperature from 0 °C to 45 °C caused a 21% decrease in gas turbine efficiency. The right plot in Figure 3 shows the effect of ambient temperature on the efficiency of the combined cycle power plant and gas turbine exergy. As can be seen from the graph, the increase in ambient temperature causes a decrease in exergy efficiency. The 45 °C increase in ambient temperature resulted in a 17% reduction in the efficiency of the combined cycle plant, which had almost no effect on the steam turbine.

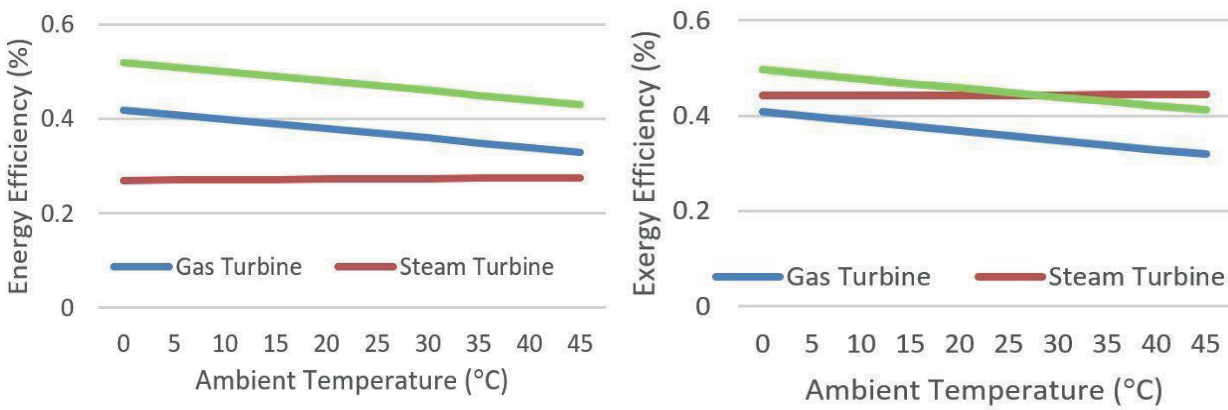

Figure 3. Impacts of ambient temperature on energy efficiency (left) and exergy efficiency (right).

3.2. Effect of compressor pressure ratio (Pr)

In this section, the impact of the change in the compressor pressure ratio on system performance is examined considering all the parameters remain constant (Table 2). Figure 4 shows the effect of the compressor pressure ratio (Pr) on the gas turbine, steam turbine and combined cycle power plant. In the gas turbine, the compressor produces a net power of 49 MW when the pressure ratio is 12, while the compressor produces 44 MW with a 10% decrease when the compressor pressure ratio is 32. When the compressor pressure ratio is increased, the gas turbine and combined cycle power plants are decreasing due to the increase in work spent by the compressor. When the compressor pressure ratio is increased, the output of the gas turbine decreases and the steam turbine power decreases by 17% as the exhaust gas enters the heat recovery steam generator at lower temperatures. The operation of the gas turbine compressor at high pressures causes an increase in the power consumption of the compressor, but the pressure of the compressor and the temperature of the compressed air also increase. The air entering the combustion chamber at high temperatures results in less fuel consumption to reach the required temperature for the turbine, thereby reducing the cost of generated power from turbine and increasing efficiency by 12.4%. The left plot in Figure 5 shows the variation in the compressor pressure ratio with respect to the gas turbine, steam turbine, and combined cycle power plant energy efficiencies. As can be seen, when the compressor pressure ratio increases from 12 to 32, the gas turbine and the combined cycle cause an increase in the plant efficiency, while the steam turbine has reduced energy efficiency. The exergy efficiencies of combined cycle plant, gas turbine, and steam turbine are shown in the right plot in Figure 5 in relation to the compressor pressure ratio. With increasing compressor pressure ratio, gas turbine exergy efficiency is decreased by 28%.

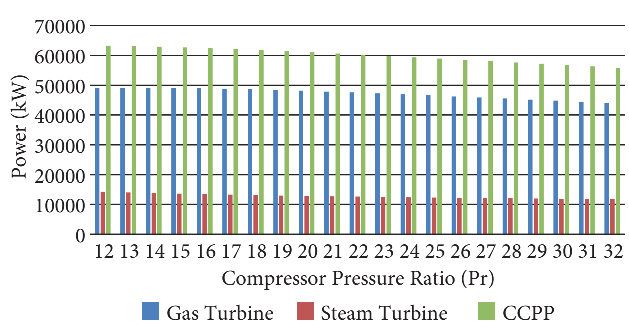

Figure 4. Impact of compressor pressure ratio on the generated power.

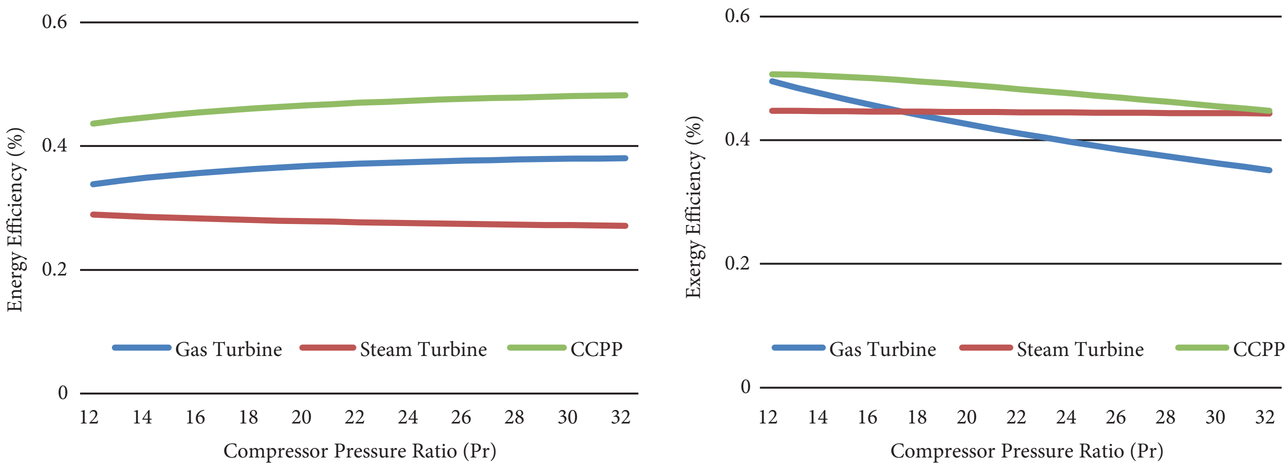

Figure 5. Impact of compressor pressure ratio on the efficiencies of gas turbine and combined cycle power plant (left) and impact of compressor pressure ratio on the exergy efficiencies of gas turbine and combined cycle power plant (right).

In Figure 6, the impact of compressor pressure ratio on the gas turbine combustion chamber, turbine, and compressor exergy destruction is presented. Along with the increase in compressor pressure ratio, exergy destruction of gas turbine components has increased. The exergy destruction is ranked as the combustion chamber, the gas turbine, and the compressor, in that order. When the compressor compression ratio was increased from 28 to 37, it was observed that combustion chamber exergy destruction increased by 14.46%, gas turbine exergy destruction by 52.2%, and compressor exergy destruction by 5.1%.

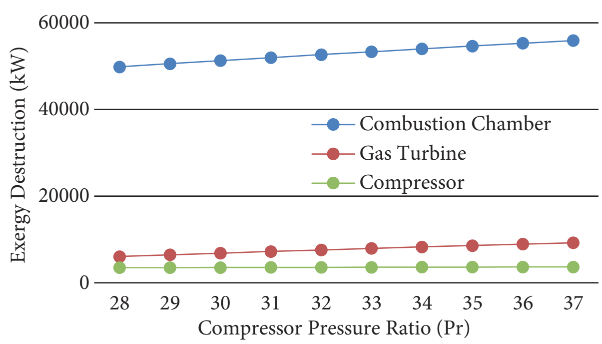

Figure 6. Change in compressor pressure ratio with gas turbine exergy destruction.

3.3. Effect of turbine inlet temperature

Considering that all other parameters remain constant, the changes in the gas turbine at the inlet temperature from 1050 °C to 1300 °C are examined in this section. Figure 7 shows the effect of turbine inlet temperature on the generated power in gas turbine, steam turbine, and combined cycle plants. The increase in turbine inlet temperature increases the gas turbine, steam turbine, and combined cycle power by 63%, 21%, and 53.5%, respectively. The effect of turbine inlet temperature on gas turbine, steam turbine, and combined cycle power plant energy efficiency is shown in the left plot in Figure 8. As can be seen clearly from the graph, the increase in turbine inlet temperature increases the efficiency of the gas turbine and therefore the combined cycle power plant. This increase in efficiency depends on the increase in the energy entering the combustion chamber. The effect of turbine inlet temperature on the exergy efficiency of the combined cycle plant, steam turbine, and gas turbine is given in the right plot in Figure 8. As the turbine inlet temperature increases, the gas turbine and combined cycle power plant exergy efficiency increases. When the turbine inlet temperature was increased from 1050 °C to 1300 °C, the combined cycle, gas turbine, and steam turbine exergy efficiencies increased by 53.5%, 37.8%, and 3.6%, respectively. Since gas turbines cannot withstand very high temperatures due to their design and used materials, it is not possible to go out at high efficiencies.

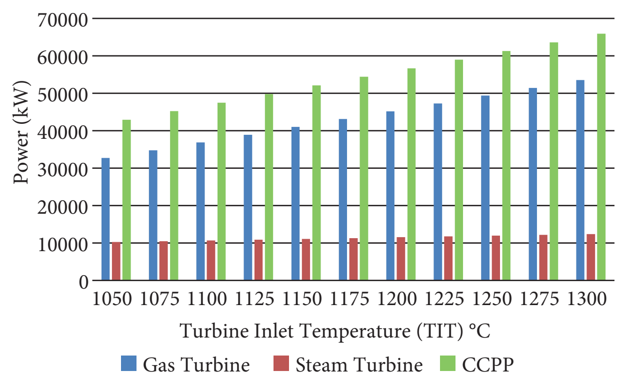

Figure 7. Variation in turbine inlet temperature according to gas turbine and combined cycle power.

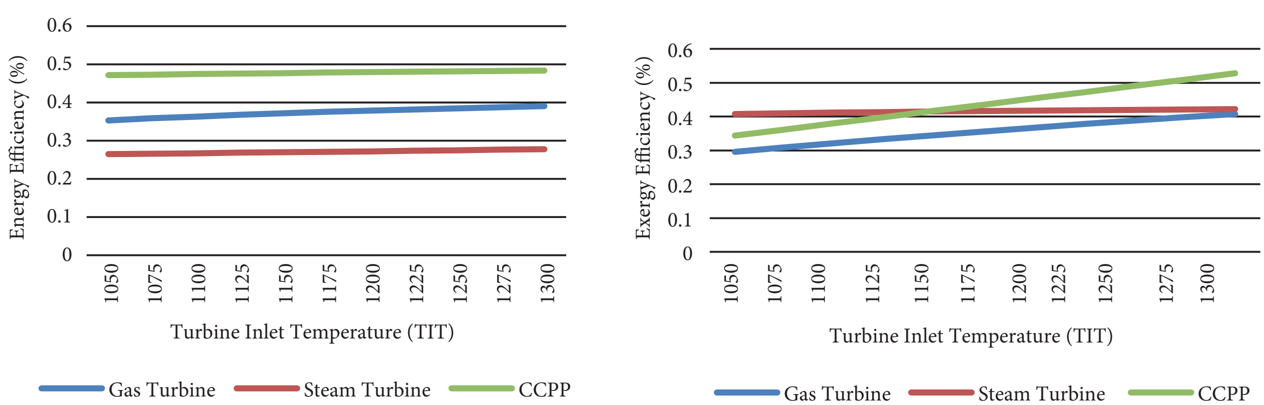

Figure 8. Effect of gas turbine inlet temperature on gas turbine and combined cycle power plant efficiency (left) and effect of turbine inlet temperature on the exergy efficiencies in gas turbine and combined cycle plant (right).

Energy analysis is calculated by taking into account the amount of energy entering and exiting the specified control volume. Various measurements are needed to make these calculations. These are the values of temperature, pressure, density, and flow rate related to the inlet and outlet points. With energy analysis studies, energy and mass equations need to be established. As a result of the energy analysis studies, diagrams are prepared that schematically show energy input and output and energy amounts [18]. The energy flow diagram (Sankey) obtained from the combined cycle plants is shown in Figure 9.

When the Sankey diagram is examined, the combined natural gas entering the combustor mixes with the pressurized air in the combustion chamber, resulting in 100 units of power, and 24.7 units of input energy is produced as net electricity while the remaining is used or lost in these production stages. Moreover, 30.8% of the energy is used to pressurize the air in the compressor, which reduces the cost of electricity production by reducing the fuel required. When we look at the other energy amounts used or lost for production, we see 4.3 unit loss of combustion, gas turbine mechanical losses of 1.7 units, 21.8 unit steam boiler exhaust loss, 4

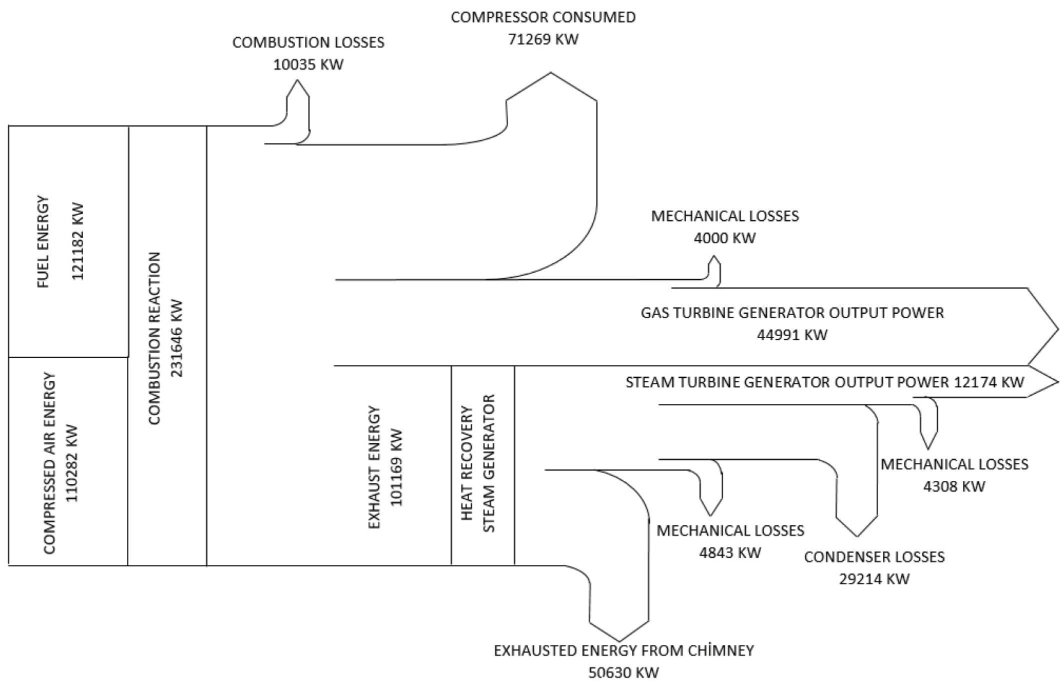

Figure 9. Combined cycle power plant Sankey diagram.

unit steam turbine mechanical losses, and 12.6 unit condenser losses. In addition, in the combined cycle power plant, 21.8 units of energy transferred from the gas turbine exhaust to the steam boiler are gainable.

The energy analysis conducted according to the first law of thermodynamics is insufficient to see the maximum amount of energy available. Therefore, in order to see the maximum energy available for any system, it is necessary to perform exergy analysis based on the second law of thermodynamics. Thus, it can be seen how much more efficient the system can be, as well as how efficient it is to the energy supplied to the system. Based on the results, it may be possible to improve the system [18]. Figure 10 shows the Grossman diagram plotted as a result of the calculations made for the combined cycle power plant.

The combined natural gas entering the combustion plant mixes with the pressurized air in the combustion chamber, resulting in 100 units of exergy coming into the power plant. While 29.2 units of the incoming exergy are produced as net electric energy, the remaining amount is consumed or lost in these production stages. While 36.2 parts of the energy are used to pressurize the air in the compressor, 1.5 units of exergy disappear in the compressor. Furthermore, 22.9 units of 37.7 units of exergy entering the gas turbine are converted to electrical energy while 11.6 units of exergy are extinguished and 3.1 units of exergy disappears in the gas turbine. The exergy of 8.1 units of exhaust gases from the gas turbine is recovered in the heat recovery steam generator. While 0.6 units of the remaining 3.5 units of exergy are discarded, 2.9 units of exergy are lost as heat transfer and mechanical loss. While 1, 0.5, and 0.3 units in a unit of 8.1 of exergy entering the steam turbine are lost in the condenser, the deaerator, and heat exchanger and pumps, respectively, 6.3 units are converted to electricity and 0.9 units of exergy return to the heat recovery steam generator.

3.4. Summary and conclusions

In this study system parameters from a natural gas combined cycle power plant with installed power of 60 MW are taken into consideration. The flow chart showing the input and output of each element of the system

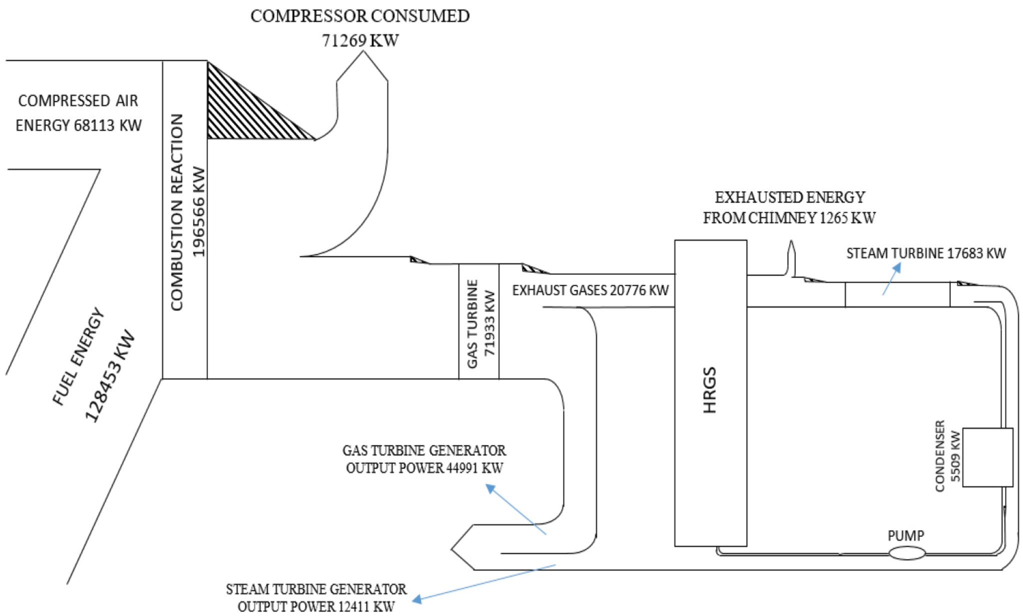

Figure 10. Grossman diagram of the combined cycle power plant.

is drawn based on the obtained data. The first (energy) and second (exergy) law analysis of the system was performed by finding the thermodynamic properties such as mass, temperature, pressure, enthalpy, entropy, exergy, and energy in the conditions taken from the real system. In the direction of the calculations made, the gas turbine power, the steam turbine power, the gas turbine generation, the system efficiency, the extinction exergy, and the exergy of each piece of equipment were found. Using system point data and thermodynamic balance equations, the Brayton cycle power, Rankine cycle power, and combined cycle power were found to be 45, 12, and 57 MW, respectively.

1. Brayton cycle, Rankine cycle, and combined cycle efficiencies were 37.5%, 27%, and 47.5%, respectively.
2. Brayton exergy, Rankine exergy, and combined cycle exergy efficiencies were 36.7%, 44%, and 46%, respectively.
3. The losses in the combined cycle plant were determined, and it was found that the most exergy lost (49 MJ) happened in the combustion chamber.
4. The highest 2nd law efficiency, which is 95%, was achieved in the compressor.

The environmental conditions, the compressor pressure ratio, and the change in turbine inlet temperature have been shown to affect the performance of the system. For these, the equations used were created with the help of a software package (EES Software) [19], and the thermodynamic properties of air and water were added to the processes using the software's own library. The findings are presented in tables and graphs, showing how efficiencies and power change.

1. As the ambient temperature increases, the power efficiency and power generated from the turbine are reduced. The reduction in the mass flow of air causes a reduction in the mass flow of the exhaust gases, which results in less power being supplied to the steam boiler, thus reducing the power taken from the

steam turbine. A 45 °C increase in ambient temperature resulted in a 21.2% reduction in gas turbine power and a 17% reduction in combined cycle power plant power.

2. When compressor pressure ratio increased from 12 to 32, gas turbine and combined cycle power plant energy efficiencies increased by 12.4% and 10.5%, respectively, while steam turbine energy efficiency decreased by 6.2%. Exergy efficiencies of the gas turbine, steam turbine, and combined cycle power plant decreased by 28%, 1%, and 11.7%, respectively.

3. The net power and efficiency increased when the turbine inlet temperature was increased. As the turbine inlet temperature increased from 1050 °C to 1300 °C, for the gas turbine, steam turbine, and combined cycle power plant, the powers increased by 63%, 21%, and 53.5%; energy efficiencies increased by 10.4%, 4.8%, and 2.5%; and exergy efficiencies increased by 53.5%, 37.8%, and 3.6%, respectively.

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