THERMODYNAMIC ANALYSIS OF COMBINED POWER CYCLE, COMBINING HEAT FROM A WASTE HEAT SOURCE WITH SUB-CYCLES

by

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Significantly increasing consumption and demand in conventional fossil energy sources require energy sources to be more efficient and sustainable. In this study, it is aimed to increase the efficiency of the systems by using thermodynamic cycles from waste heat sources. The designed system is aimed at increasing the efficiency of the system by adding sub-cycles of the waste heat of a gas turbine. The results analyzed with the engineering equation solver program, when all the cycles are combined, the system energy efficiency is 75% and the total exergy efficiency is 24%. Brayton cycle when the system is evaluated alone, the energy efficiency of the system is 65%, the exergy efficiency is 14%. The S-CO_2 cycle system when the system is evaluated alone, the exergy efficiency is 23% and the exergy efficiency is 11%. The ORC system when the system is evaluated alone, the exergy efficiency is 19% and the exergy efficiency is 22%. Rankine system when the system is evaluated alone, the exergy efficiency is 17% and the exergy efficiency is 88%. Turbine inlet temperatures tend to decrease as the exergy destruction in the system also affects the subcomponents.

Key words: *energy analysis, exergy analysis, Brayton cycle, n-pentane cycle, Rankine cycle, S-CO_2 cycle*

Introduction

In the global world, the increase in industrialization constantly increases the energy demand. The increase in energy consumption is rapidly depleting fossil energy resources at an alarming rate. Increasing consumption of fossil fuels (coal, oil, natural gas, *etc.*) brings climate change issues to the agenda. There is a need for sustainable and potentially economical use of energy resources, which are necessary to increase the efficiency of energy in is industry. Improvement in energy efficiency and reduction of GHG will be achieved by the technological development of energy production systems and by using innovative technologies. In this context, more compact systems emerge by combining with RES (solar, wind, biomass, *etc.*). Today, the need for the development of new technologies for renewable energy production has become more necessary than ever. Today, new combined power cycles have been designed and energy analysis has been made by utilizing the waste heat of energy facilities and combining them with sub-cycles.

Guo *et al.,* [1] performed a thermodynamic analysis of a S-CO_2 Brayton cycle combined with a compressed CO_2 energy storage system for waste heat recovery (WHR) of

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ship gas turbines under variable load conditions. They said that the thermal efficiency reached over 40% and the exergy efficiency over 65%. Pan *et al.*, [2] carried out thermoeconomic analysis of combined natural gas cogeneration system with S-CO_2, Brayton cycle, and ORC. They said that the energy efficiency of the system is 56.47% and the exergy efficiency is 45.46%. Wang *et al.*, [3] presented an exergoeconomic analysis of a new trigeneration system for gas turbine WHR, which includes a S-CO_2 Brayton cycle, and ORC, and an absorption refrigeration cycle. They also compared the effects of ORC with different working fluids on overall system performance. Cao *et al.*, [4] performed a thermodynamic analysis of a Brayton and ORC of a S-CO_2 for solar energy use with typical geothermal as an auxiliary heat source. They found that the thermal efficiency of the combined cycle is 35.07%. They emphasized that the combined cycle has a thermodynamic advantage using solar energy and auxiliary geothermal energy. Song *et al.*, [5] performed analysis of various configurations in a combined S-CO_2 cycle and ORC system for hybrid solar and geothermal power generation. They said the S-CO_2 cycle system, a split geothermal flow system combined with ORC working fluid, using geothermal heat first and then heat from the top in series, provides 45% more power output, the maximum among all proposed configurations. Baglietto *et al.*, [6] performed a techno-economic comparison of Supercritical CO_2, steam, and ORC Cycles for their Applications to WHR. They performed techno-economic analysis and optimization by focusing on WHR applications for different dimensions and cycle parameters operating conditions. The analyzed cycles were carried out by first maximizing the net electrical power and then minimizing the specific capital cost. Comparing the results, they said that in the first case, the more complex S-CO_2 loop configuration performs competitively, while in the second case, the simpler S-CO_2 loop configuration requires lower specific cost for the same electrical power produced. Jin *et al.*, [7] have installed a recompression S-CO_2 Brayton cycle operating system that combines a gas turbine with a preheated S-CO_2 Brayton cycle for waste heat. They presented thermodynamic analysis of the effects of the cycle on thermal efficiency. Hou *et al.*, [8] performed thermodynamic and exergoeconomic analyses for a new combined S-CO_2 recompression cycle and the regenerative organic ORC using a zeotropic mixture. Their results presented that the optimum zeotropic mixture is R236fa/R227ea (0.46/0.54). The exergy efficiency was found to be 73.65%. They demonstrated the superiority of their proposed combined cycle over a single S-CO_2 cycle and a combined S-CO_2 cycle and basic ORC. Energy and analysis of an ORC operating with low-grade waste heat and a combined system of vapor compression were done by Xia *et al.* [9]. Hai *et al.*, [10] performed their AI-based analysis and optimization on a low-temperature ORC for waste heat recovery from a Kalina cycle connected to a geothermal heat source. Qin *et al.*, [11] performed thermodynamic analysis and optimization of a WHR system with a combined supercritical/transcritical CO_2 cycle. The aim is to propose a new combined cycle system consisting of a S-CO_2 recompression Brayton cycle and a transcritical CO_2 refrigeration cycle to utilize the waste heat of a marine turbine for both power generation and cooling. Gao *et al.*, [12] performed the performance analyzes of the S-CO_2 Brayton cycle and ORC combined system, taking into account the daily distribution of solar radiation. AlZahrani and Dincer [13] conducted energy and exergy analyzes of the integrated CO_2 Brayton-ORC with the solar tower facility. They used a combined cycle, an ORC and a S-CO_2 Brayton cycle as a cascading kick cycle. Sahin [14] investigated the WHR potential of an autoclave device and made an energy analysis. Khan and Mishra [15] presented the energy analysis of the solar combined precompression S-CO_2 cycle and ORC. They said that the net power output and thermal efficiency of the pre-compression loop increased by more than 4.51% and 4.52%, respectively, using ORC. Ping *et al.*, [16] ORC can effectively use the waste heat energy of the internal combustion (IC) engine. They combined an integrated system model of IC motor-ORC with a subsystem model. The dynamic response of the ORC system was evaluated under different road conditions. They stated that there are significant strong coupling and non-linear properties between the different performances of the ORC system in road conditions. Manente and Fortuna [17] They performed analysis of supercritical CO_2 power cycles for WHR. They made a comparison between traditional and new double expansion arrangements. Multi-objective optimization said that the significant increase in performance is only a limited 5.0-6.2% increase in the specific investment cost compared to traditional cycles. The development of improved S-CO_2 power cycle schemes is particularly dependent on the upper temperature range. Zhang *et al.*, [18] proposed a new S-CO_2 power cycle based on the recompression cycle configuration to efficiently recover the exhaust heat of the IC engine. The maximum cycle power corresponding to an optimal combination of system parameters is 39.49 kW and the WHR efficiency is 74.83%. Ruiz-Casanova *et al.*, [19] performed a thermodynamic analysis of S-CO_2 Brayton cycles for use with low-grade geothermal heat sources. With a 20 kg/per second geothermal brine flow at 150 °C as the heat source and a minimum allowable re-injection temperature of 70 °C, the intercooled reclaimed Brayton cycle achieved the highest electrical power output, energy and exergy efficiencies. They found that 779.99 kW, 11.51% and 52.49%, respectively. In Liu *et al.*, [20] technological enhancement has been made regarding the S-CO_2 Brayton cycles. On the basis of energy saving and emission reduction, they say that the development of power generation technology has always resulted in higher efficiency at lower cost. it is said that the use of RES or waste heat is one of the important solutions here. Papingiotis *et al.*, [21] performed thermodynamic analyzes of transcritical and supercritical organic Rankine and Brayton cycles connected to parabolic trough collectors. Subcritical ORC with n-Pentane and R245fa as working fluid, supercritical ORC with R245fa as working fluid and supercritical Brayton cycles with CO_2 and R245fa as working fluid were studied. The overall efficiency of the system (power to sun) for subcritical ORC with n-Pentane and R245fa is 14.12% and 10.31%, respectively. The efficiencies for R245fa and supercritical ORC were 15.14% and for CO_2 and R245fa the supercritical Brayton cycle was 13.23% and 8.87%, respectively. In Seyed Mahmoudi *et al.*, [22] the use of the ORC, organic flash cycle (OFC), and Kalina cycle (KC) has been proposed to increase the electricity produced by a S-CO_2 recompression Brayton (SCRB) cycle. The highest exergy efficiencies for SCRB/OFC and SCRB/ORC systems were obtained when n-nonan and R134a were used as working fluids for OFC and ORC, respectively. Ozer and Dogan, [23] used the exhaust emission results of the engine obtained in their experimental studies. They made exergy, energy and exoeconomic analysis. In Dogan, *et al.*, [24] energy and exergy analysis were performed using different engine speeds. As a result of the energy analysis, the energy distribution of the engine was determined and the thermal efficiency was calculated. In the exergy analysis, fuel exergy, exhaust exergy, entropy production and exergy efficiency were calculated separately.

Recovery of waste heat of developing and advancing industrial processes is inevitable in order to obtain high efficiency in the use of primary energy resources. When waste heat is present at high temperatures, it is our indispensable option where cascading the evaporation process is required. In this study, it is aimed to increase the total energy efficiency of the system with the help of sub-cycles by using the waste heat of a gas turbine in a gradual way.

System description

The system designed for combined power generation is given in fig. 1.

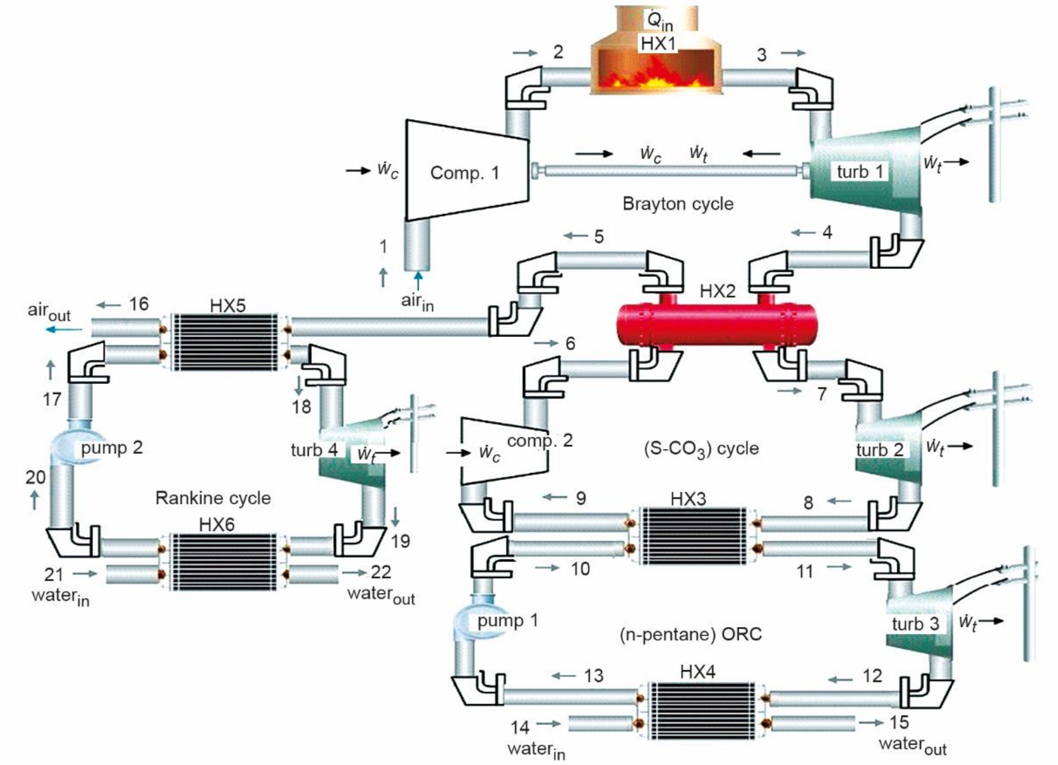

Figure 1. The designed combined power system

The air entering the 1st state leaves the compressor with its temperature and pressure increases in the 2nd state. Then, with the heat it receives from the heat source (Sun, biomass), the hot air exiting in the third state, in the case of increased temperature, passes through the turbine and provides electricity production. In the 4th case, the hot air coming out of the 1st turbine is transferred gradually to the S-CO_2 cycle system with the 2nd heat exchanger (HX), and then to the Rankine cycle system with the 5th heat exchanger, and in the 16th state, the air is discharged.

The heat exchanger No. 2 enters the 2nd turbine with the S-CO_2 in the 7th state and power generation is provided. Then, heat transfer is made to the ORC system using the lower cycle refrigerant n-pentane with the heat exchanger No. 3. The reduced temperature CO_2 enters the compressor in the 9th state, and by increasing both its temperature and pressure again, it completes the cycle by switching to heat from the upper cycle in the 6th state.

With the 3rd heat exchanger, as the sub-cycle of the S-CO_2 cycle, power generation is achieved with the 3rd turbine with n-pentane refrigerant, which becomes saturated vapor in the 11th state. In the 12th state, the fluid leaving the 3rd turbine loses its heat to pass into the saturated liquid state. In the 13th state, in the saturated liquid state, the fluid enters the pump and enters the 10th state by increasing its pressure and completes the cycle by taking heat from the 3rd heat exchanger.

In the 18th state, the fluid superheated water vapor, the temperature of which increases with the heat it receives from the heat exchanger No. 5, exits in the 19th state after power generation with the 4th turbine and its temperature is reduced by the heat exchanger No. 6 to bring it to the saturated liquid state. In the 20th state, the fluid with saturated heat is increased by the pump and sent to the 17th state to increase its temperature, and the cycle is completed by increasing the temperature with the 5th heat exchanger.

The following assumptions were taken into account while making the thermodynamic analysis of the designed system:

– System performance is assumed to be stable and regular.
– Pure substances are used in the system.
– Compression in compressors is adiabatic.
– Pressure drops and heat transfer in system components and pipeline are also neglected.
– Counter flow heat exchangers are used in the heat source heat exchangers and heat losses are neglected.
– The dead state of the fluids (air, CO_2, water, n-pentane) circulating in the system is taken as 20 ^{o}C.
– System performance is assumed to be stable and regular.
– Gravitational potential energy and kinetic energy are not taken into account.
– The superheat was increased to 60 ^{o}C in the steam Rankine cycle.
– The temperature of the 1st heat exchanger is taken as 1.24 times the heat exchanger No. 1 (HX1) outlet temperature.
– In the gas turbine, the inlet and outlet pressure ratio of the compressor 21 and the temperature of the heat taken from the heat source HX1, the outlet temperature of 1493 K and the turbine 1 outlet temperature of 738.2 K are taken from the UGT-25000 gas turbine specifications [25].

The positions of the process in the design are:

1. 1→2: Adiabatic compressor-I, isentopic efficiency 75%.
2. 2→3: Heat input from the heat source to the system with HX1, increasing the isobaric temperature.
3. 3→4: Work output of adiabatic turbine-I, isentropic efficiency 95%.
4. 4→5: Adiabatic heat transfer with counterflow HX2 (air - S-CO_2).
5. 5→16: Adiabatic heat transfer (air-water) with counterflow HX5.
6. 7→8: Work output of adiabatic turbine-2, isentropic efficiency 90%.
7. 8→9: Adiabatic heat transfer with counterflow HX3 (S-CO_2 – n-pentane).
8. 9→6: Adiabatic compressor-I, isentopic efficiency 85%.
9. 10→11: ORC heat transfer of heat of CO_2 fluid with adiabatic and counter flow heat exchanger.
10. 11→12: Work output of adiabatic turbine-III, isentropic efficiency 90%.
11. 12→13: Heat dissipation by heat exchanger as isobar.
12. 12→13: Increasing the pressure of saturated liquid with adiabatic pump, isentropic efficiency 85%.
13. 18→20: Work output of adiabatic turbine-4, isentropic efficiency 90%.
14. 19→20: Heat dissipation by heat exchanger as isobar.
15. 20→17: Increasing the pressure of saturated liquid with adiabatic pump, isentropic efficiency 85%.
16. 17→18: Increasing the superheating degree by heat transfer of air to water with adiabatic and counter flow heat exchanger.

Thermodynamic analysis

For steady-state in thermodynamic analysis, the basic mass balance equation can be given [26-28]:

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

where $\dot{m}$ is the mass-flow rate, and the in and ex indices represent the inlet and outlet states, respectively. The energy balance is:

$$\dot{Q}_{\text{in}} + \dot{W}_{\text{in}} + \sum_{\text{in}} \dot{m}\left(h + \frac{V^2}{2} + \text{g}z\right) = \dot{Q}_{\text{ex}} + \dot{W}_{\text{ex}} + \sum_{\text{ex}} \dot{m}\left(h + \frac{V^2}{2} + \text{g}z\right) \tag{2}$$

where $\dot{Q}$ is the heat transfer rate, $\dot{W}$ – the power, h – the specific enthalpy, v – the velocity, z – the height, and g – the gravitational acceleration. The entropy balance equation for steady-state conditions is:

$$\sum_{\text{in}} \dot{m}_{\text{in}} s_{\text{in}} + \sum_{k} \frac{\dot{Q}}{T_k} + \dot{S}_{\text{gen}} = \sum_{\text{ex}} \dot{m}_{\text{ex}} s_{\text{ex}} \tag{3}$$

where s is the specific entropy and $\dot{S}_{\text{gen}}$ – the entropy generation rate. The exergy balance equation can be written:

$$\sum \dot{m}_{\text{in}} ex_{\text{in}} + \sum \dot{E}x_{Q,\text{in}} + \sum \dot{E}x_{\text{W,in}} = \sum \dot{m}_{\text{ex}} ex_{\text{ex}} + \sum \dot{E}x_{Q,\text{ex}} + \sum \dot{E}x_{\text{W,ex}} + \dot{E}x_{\text{D}} \tag{4}$$

The specific flow exergy can be written:

$$ex = x_{\text{ph}} + ex_{\text{ch}} + ex_{\text{pt}} + ex_{\text{kn}} \tag{5}$$

The kinetic and potential parts of the exergy are assumed to be negligible. Also, the chemical exergy is assumed to be negligible. The physical or flow exergy, ex_{ph}, is:

$$ex_{\text{ph}} = (h - h_0) - T_0(s - s_0) \tag{6}$$

where h and s represent specific enthalpy and entropy, respectively, in the real case. The h_o and s_o are enthalpy and entropy at reference medium states, respectively.

Exergy destruction is equal to specific exergy times mass:

$$\dot{E}x_{\text{D}} = ex\ m \tag{7}$$

The $\dot{E}x_{\text{D}}$ are work-related exergy ratios and given as:

$$\dot{E}x_{\text{D}} = T_0 \dot{S}_{\text{gen}} \tag{8}$$

The $\dot{E}x_{\text{W}}$ are work-related exergy ratios and are given:

$$\dot{E}x_{\text{W}} = \dot{W} \tag{9}$$

$\dot{E}x_Q$, are the exergy rates related to heat transfer and are given:

$$\dot{E}x_Q = \left(1 - \frac{T_0}{T}\right)\dot{Q} \tag{10}$$

What work comes out of the system:

$$\dot{W}\text{net}_{\text{out}} = \dot{Q}_{\text{in}} - \dot{Q}_{\text{out}} \tag{11}$$

system thermal efficiency (η):

$$\eta = \frac{\dot{W}\mathrm{net}_{\mathrm{out}}}{\dot{Q}_{\mathrm{in}}} \tag{12}$$

The exergy efficiency, φ, can be defined:

$$\varphi = \frac{\sum \text{useful output exergy}}{\sum \text{input exergy}} = 1 - \frac{\sum \text{exergy loss}}{\sum \text{input exergy}} \tag{13}$$

Results and discussion

The temperature entropy *T-s* diagrams of the integrated cycles are shown in fig. 2 (Brayton cycle), fig. 3 (ORC), fig. 4 (Rankine cycle), and fig. 5 (S-CO_2 cycle).

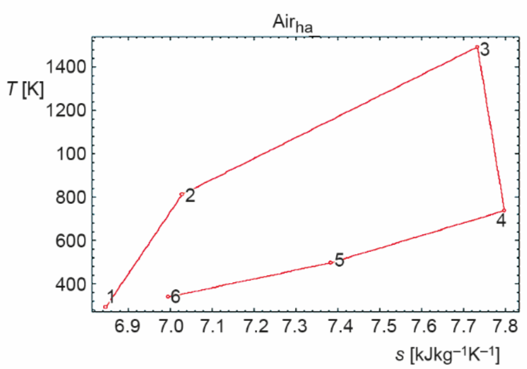

Figure 2. The *T-s* diagram ideal Brayton cycle

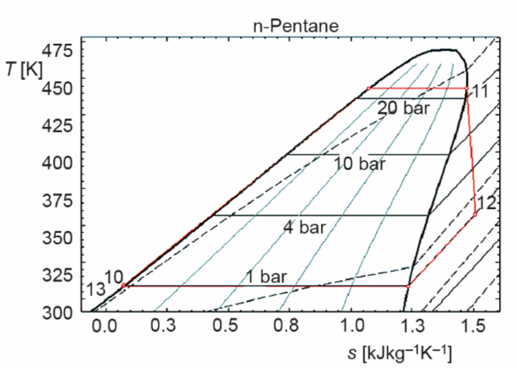

Figure 3. The *T-s* diagram ORC system

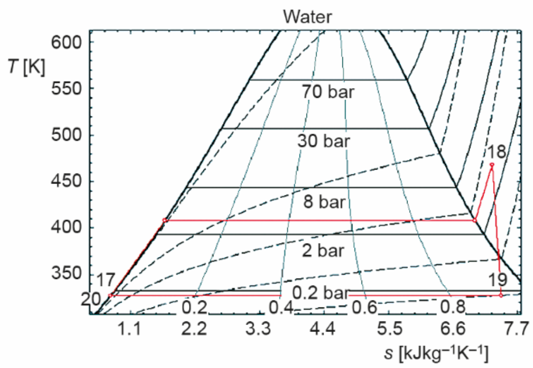

Figure 4. The *T-s* Rankine system

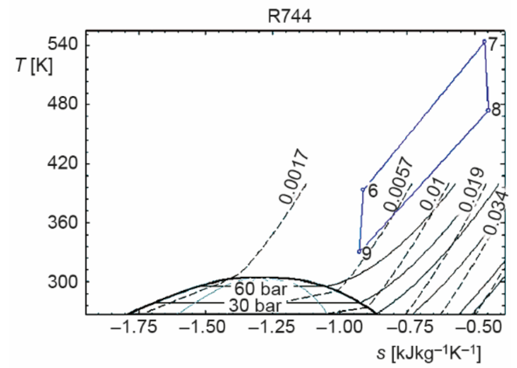

Figure 5. The *T-s* diagram S-CO_2 system

In tab. 1, thermodynamic values of the combined power system created with different thermodynamic cycles for the positions in fig. 1.

Table 1. Thermodynamic values for the combined system

Location	T [K]	$\dot{s}$ [KJkg^{-1}K^{-1}]	P [bar]	h [kJkg^{-1}]	ex [kJkg^{-1}]	$\dot{m}$ [kgs^{-1}]	Fluid
1.	293.2	6.846	1	293.4	0	88	Air
2.	813.1	7.028	21	837.5	490.6	88	Air
3.	1493	7.733	21	1630	1076	88	Air
4.	738.2	7.798	1	754.8	182.3	88	Air
5.	498.1	7.383	1	501.4	50.44	88	Air
6.	393.6	–0.9117	165	–15.85	252.5	107.2	CO_2
7.	544.1	–0.4597	165	192.2	328.1	107.2	CO_2
8.	474.2	–0.446	80	134.1	265.9	107.2	CO_2
9.	330.3	–0.9264	80	–54.28	218.4	107.2	CO_2
10.	309.2	0.07307	22.37	25.42	4.386	56.97	n-pentane
11.	443.2	1.473	22.37	579.5	147.9	56.97	n-pentane
12.	357.3	1.507	0.9835	470.8	29.22	56.97	n-pentane
13.	308.2	0.07108	0.9835	21.31	0.8598	56.97	n-pentane
14.	293.2	0.2965	1	84.01	0	245.1	Water
15.	318.2	0.6386	1	188.5	4.2	245.1	Water
16.	340.2	6.995	1	340.7	3.426	88	Air
17.	327.5	0.7598	3.132	227.9	8.005	5.385	Water (x = 1)
18.	468.2	7.27	3.132	2855	726.2	5.385	Water (x = 100)
19.	327.5	7.421	0.1527	2409	236.1	5.385	Water (x = 0.92)
20.	327.5	0.7596	0.1527	227.5	7.697	5.385	Water (x = 0)
21.	293.2	0.2965	1	84.01	0	112.4	Water
22.	318.2	0.6386	1	188.5	4.2	112.4	Water
T[0].	293.2	6.846	1	293.4			Air
T[0].	293.2	–0.01389	1	–5.125			CO_2
T[0].	293.2	–0.04468	1	–13.49			n-pentane
T[0].	293.2	0.2972	1	84.22			Water

The thermodynamic analysis of the combined power system created with different thermodynamic cycles is presented in tab. 2 [(+) entering the system (–) exiting the system].

As seen in tab. 2, 69719 kW of heat from a heat source from HX1 and compressor power of 47883 kW in the Brayton cycle, 4120 kW of compressor power in the S-CO_2 cycle, 243.3 kW of pump power in the n-pentane cycle and 1.914 kW pump in the Rankine cycle. The total energy input of the integrated system working with power is 121.967 kW. With the power obtained in the integrated system, a total power of 91819 kW was obtained with the turbine power of 76995 kW in the Brayton cycle, 6231 kW in the S-CO_2 cycle, 6193 kW in the ORC cycle, and 2400 kW in the turbine in the Rankine cycle. The overall exergy efficiency of the system was found to be 75%.

Table 2. Thermodynamic analysis of the combined power system

Component	*W* [kW]		Ex_D [kW]	S_{gen} [kW]	φ [%]	*Q* heat		δ_{iz}
	(+) in	(–) out				(+) in	(–) out	
Compressor 1 (1-2)	+47883		4714	16.08	0.9	–		0.75
HX1 (2-3)	–		7144	24.36	0.88	± 69719		
Turbine 1 (3-4)	–76995		1665	5.679	0.98	–		0.95
HX2 (4-5)(6-7)	–		3509	11.97	0.7	± 22300		
Turbine 2 (7-8)	–6231		430.6	6.917	0.94	–		0.90
Compressor 2 (6-9)	+4120		462	1.576	0.89	–		0.85
HX3 (8-9)(10-11)	–		2504	8.541	0.77	± 31568		
Turbine 3 (11-12)	-6193		569.6	1.943	0.92	–		0.90
HX4 (12-13)(14-15)	–		373.4	1.274	0.77	± 25610		
Pump 1 (13-10)	+243.3		33.34	0.1137	0.86	–		0.85
HX5 (16-5)(17-18)	–		270.2	0.9216	0.93	± 14146		
Turbine 4 (18-19)	–2400		238.7	0.8142	0.91	–		0.90
HX6 (19-20)(21-22)	–		757.9	2.585	0.38	± 11748		
Pump 2 (15-16)	+1.914		0.257	0.0008765	0.87	–		0.85

As seen in tab. 2, in terms of exergy efficiency, the exergy of a heat source HX1 is 7144 kW and the compressor exergy of 4714 kW in the Brayton cycle, the exergy of the 462 kW compressor in the S-CO_2 cycle, the exergy of the 33.34 kW pump power in the n-pentane cycle and 0 in the Rankine cycle. The total exergy input of the integrated system, which works with [kW] pump power, is 12.353 kW. With the power obtained in the integrated system established, the turbine in the Brayton cycle has 1665 kW exergy, the turbine 430.6 kW exergy in the S-CO_2 cycle, the turbine 569.6 kW exergy in the ORC cycle, and the turbine 238.7 kW exergy in the Rankine cycle, a total of 2.934 kW. Exergy output was obtained. The overall exergy efficiency of the system was found to be 24%.

Conclusions

Sustainability and innovative technologies in the increasing energy demand have led us to seek new ways to meet the demands. In this study, new integrated systems are created by combining waste heat with different cycles with an innovative approach.

When all the cycles are combined, the system energy efficiency is 75% and the total exergy efficiency is 24%. Brayton cycle system energy efficiency 65%, exergy efficiency 14% when the system is evaluated alone. The S-CO_2 cycle system is evaluated alone, the exergy efficiency is 23% and the exergy efficiency is 11%. The ORC system is evaluated alone, the exergy efficiency is 19% and the exergy efficiency is 22%. When Rankine system is evaluated alone, the exergy efficiency is 17% and the exergy efficiency is 88%. Turbine inlet temperatures tend to decrease, affecting the subcomponents of exergy destruction in the system.

Integrated systems are progressing towards being more developable with innovative environmentally friendly fluids and suitable thermodynamic cycles. This study, it shows how valuable waste heat is and how it will be evaluated.

Nomenclature

$\dot{E}x$ – exergy
ex – specific exergy
h – enthalpy
s – entropy

Subscripts

ref. – refrigerant
0 – ambient temperature
D – destruction
ex – exit
in – inlet
Q – heat

Greek symbol

φ – exergy efficiency

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