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TECHNICAL NOTE

Thermodynamic Analysis and Optimization of a Novel Cogeneration System: Combination of a Gas Turbine with Supercritical CO_2 and Organic Rankine Cycles

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ABSTRACT

Thermodynamic analysis of a novel combined system which is combination of methane fired gas turbine cogeneration system (CGAM) with a supercritical CO_2 recompression Brayton cycle (SCO_2) and an Organic Rankine Cycle (ORC) is reported. Also, a comprehensive parametric study is performed to investigate the effects of some important parameters on the performance of the proposed system. Finally, a thermodynamic optimization is done to maximize energy and exergy efficiencies. The results showed that, the energy and exergy efficiencies are maximized at particular compressor pressure ratios and the values depend on the operating parameters of the system. Energy and exergy efficiencies are determined to be 85.33% and 54.18%, respectively, for the proposed system under the base condition. Moreover, the parametric study showed that in addition to the operating parameters of the system, ambient temperature has also an important effect on the system performance as energy efficiency increases and exergy efficiency decreases with the ambient temperature increment.

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ACRONYMS			
AC	Air compressor	e_i	Specific thermomechanical flow exergy at state i [kJ/kmol]
AP	Air preheater	e_{ch}	Specific chemical exergy [kJ/kmol]
CC	Combustion chamber	e_{ph}	Specific physical exergy [kJ/kmol]
COND	Condenser	h	Specific enthalpy [kJ/kmol]
C1	Compressor 1	LHV	Lower heating value [kJ/kmol]
C2	Compressor 2	$\dot{n}$	Molar rate [kmol/s]
GT1	Gas turbine 1	P_i	Pressure at state i [bar]
GT2	Gas turbine 2	r_p	Pressure ratio [-]
HRSG	Heat recovery steam generator	$\overline{R}$	Universal gas constant [kJ/kmol.K]
HTR	High temperature recuperator	s	Specific entropy [kJ/kmol.K]
HE1	Heat exchanger 1	T_i	Temperature at state i [K]
HE2	Heat exchanger 2	$\dot{W}$	Produced or consumed power by components [kW]
LTR	Low temperature recuperator	**Greek letters**	
ORCT	Organic Rankine cycle turbine	ε	Exergy efficiency [%]
ORCP	Organic Rankine cycle pump	$\eta_{is,C}$	Isentropic efficiency of compressor [%]
Nomenclature		$\eta_{is,GT}$	Isentropic efficiency of gas turbine [%]
		$\eta_{is,P}$	Isentropic efficiency of pump [%]
$\dot{E}_i$	Exergy rate [kW]	η	Energy efficiency [%]

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$\dot{E}_D$	Exergy destruction rate [kW]	**Subscripts**	
$\dot{E}_F$	Fuel exergy rate [kW]	0	Reference environment state
$\dot{E}_{in}$	Entrance Exergy rate [kW]	in	Input
$\dot{E}_P$	Product Exergy rate [kW]	out	Outlet

1. INTRODUCTION

Technology developments require design of efficient energy systems as one of the engineering challenges. Increasing energy demand by developed countries in a world with finite fuel resources, clarify the importance of attention to the most efficient energy systems [1]. In order to introduce an efficient energy system, the exergy analysis can be a powerful tool to determine the type and exact magnitude of exergy destruction (or loss) [2]. Consequently, it can play an important role in the effective energy conversion in existing power plants [3].

One of the most known proposed cogeneration systems is CGAM [4-9], which is a cogeneration system to produce power and steam as a byproduct [4]. A wide variety of thermodynamic analysis, and optimization methods have been provided by researchers for this system [5-9]. Some of the suggested approaches are: direct use of a nonlinear programming algorithm, thermos-economic functional approach, and modular simulation and optimization of the system.

Another efficient energy system is supercritical carbon dioxide (SCO_2) power cycle. As a working fluid, thermos-physical properties of carbon dioxide have sharp changes near the critical point, and the SCO_2 cycle takes advantage of these phenomena to reduce the compression work, and consequently to increase the cycle efficiency. The supercritical CO_2 recompression Brayton cycle is one of the promising configurations to use the mentioned advantage of CO_2 behavior. In recent years, it has received more attention because it is simple, compact and less expensive, and offers high efficiency [10]. Angelino [11] showed that the recompression cycle is more promising than other cycles, because of its high effectiveness, easiness, compactness, stableness and economic benefits and this cycle is very good option to generate power, where a heat source with a temperature of 700-1000 K is available [12].

Compared to other bottoming cycles, Organic Rankine Cycles (ORCs) have several advantageous aspects. One of the most important characteristics of organic working fluids (compared to water used in the Rankine cycle) is their relatively low enthalpy drop through the turbine which causes the higher mass flow rate and reduces the gap losses and consequently increases the turbine adiabatic efficiency. Moreover, superheated vapor at the turbine exit of an ORC cycle, leads to avoiding droplet erosion and allowing reliable operation and fast start-up [13, 14].

Lazzaretto and Toffolo [15] optimized the CGAM cogeneration system from energy, economic as well as environmental viewpoints. They showed how a thermal system design can be optimized in terms of energy, economy and environment as distinct objectives. Khaljani et al. [16] combined CGAM cogeneration system with an ORC unit. Results of optimization in their study showed that exergy efficiency of the system grows up to 4.75 points at the optimized condition. Khanmohammadi et al. [17] combined CGAM cogeneration system with a biomass gasifier unit and optimized the combined system from the exergy and economic viewpoints through some decision variables.

In the present work, a novel cogeneration system is proposed and thermodynamically analyzed to produce power as well as saturated steam as a byproduct. The configuration is achieved by combining the CGAM cycle [4], the supercritical CO_2 recompression Brayton cycle [18] and the Organic Rankine Cycle (ORC). Thermodynamic analysis is performed for the proposed cycle, based on energy and exergy, to investigate the system performance considering a wide range of some of operating parameters, and then the results are compared with the previous studies in the literature [4, 18]. A comprehensive parametric study is also done to show the effects of some important parameters on the performance of the system. Finally, a thermodynamic optimization is performed to find maximum energy and exergy efficiencies.

2. SYSTEM DESCRIPTION AND ASSUMPTIONS

Figure 1 shows schematic of the proposed cycle. This cogeneration system is a combination of the gas turbine cycle (CGAM cogeneration system) [4], which uses the natural gas as fuel in the combustion chamber, and the supercritical CO_2 recompression Brayton cycle (SCRBS) [18], which uses the exhaust gases of the CGAM system as a heat source. The main purpose of the proposed system is producing power and saturated vapor as a byproduct. Compressed air as a mixture of ideal gases (state 2) enters the air preheater before the combustion chamber (3). Methane is fed to the combustion chamber as fuel (10). The produced hot gases, which are mixtures of ideal gases (4), leave the combustion chamber at a mean temperature of 1520 K. The fluid is expanded to ambient pressure and 904 K (5) in the gas turbine 1. The expanded fluid is supplied to the air preheater to preheat the entering air to the combustion chamber and leaves it at a temperature of 779 K (6). This temperature is sufficient to run a SCO_2 system [12]. Therefore, the cooled fluid provides the required heat to operate the SCRBS cycle by the heat exchanger 1 and then (7) flows to the heat recovery steam generator to produce saturated vapor. The combustion products leave the heat recovery steam generator at 320 K (8) which is higher than the corresponding dew point temperature. The feed water of the heat recovery steam generator (9) is converted to the saturated steam (29) at 20 bar.

As shown in Figure 1, in SCRBC, the exiting CO_2 from the heat exchanger 1 (11) enters the gas turbine 2 at 769 K and 200 bar to generate power. The expanded CO_2 then flows to the high temperature recuperator (12) to heat the stream entering the heat exchanger 1 (18), and afterward (13), it is cooled down again in the low temperature recuperator (14). Then, the exiting cold gas of the low temperature recuperator is divided into two streams that have different flow rates (15 & 19). The stream with higher flow rate (19) passes through the heat exchanger 2 before being compressed in the compressor 1 (20) to provide the required heat for operation of an Organic Rankine Cycle (ORC). The other stream (15) is also compressed in the compressor 2 (16). The compressed CO_2 exiting from the compressor 1 (21) is heated in the low temperature recuperator and then (22) joins the exiting CO_2 of the compressor 2. This stream (17) is heated in the high temperature recuperator and then enters the heat exchanger 1. In the ORC, isopentane is selected as a working fluid which enters the pump with the saturated liquid state at 1.3 bar (26). Then, it is compressed to 5 bar (23), and is converted to superheated vapor at the heat exchanger 2 (24). The working fluid, then flows to the ORC turbine to produce power (25) and is converted to saturated liquid in the condenser (26) to complete the cycle.

The following assumptions are made in this work:

- The system operates at steady state condition.

Changes in the kinetic and potential energy are not noticeable [19].

- Pressure drops in the SCO_2 system and ORC cycle are not noticeable while in the CGAM cogeneration system, some suggested values in the literature [4] are considered for pressure losses.
- The cooling water enters the condenser at ambient condition.
- The molar analysis of air at the compressor inlet is: 77.48% N_2, 20.59% O_2, 0.03% CO_2 and 1.9% H_2O (g) [4].
- The combustion of the methane in the combustion chamber is complete and the low heating value of methane is 802361 kJ/kmol [4].
- The produced gas leaves the combustion chamber at 1520 K [4].
- The air compressor pressure ratio is $r_{p,AC} = 15$.
- Isentropic efficiency is considered for GT1, GT2, ORC Turbine, AC, C1 and C2 [20, 21].
- Ambient temperature and pressure are 298.15 K and 1.013 bar, respectively.

2. 1. Energy Analysis In this study, assuming the pressure losses across the heat exchangers and the components isentropic efficiencies, the work quantities associated with compressor, pumps and turbines are calculated as [20]:

$$\eta_{C,isen} = \frac{W_{C,isen}}{W_C} \tag{1}$$

$$\eta_{T,isen} = \frac{W_T}{W_{T,isen}} \tag{2}$$

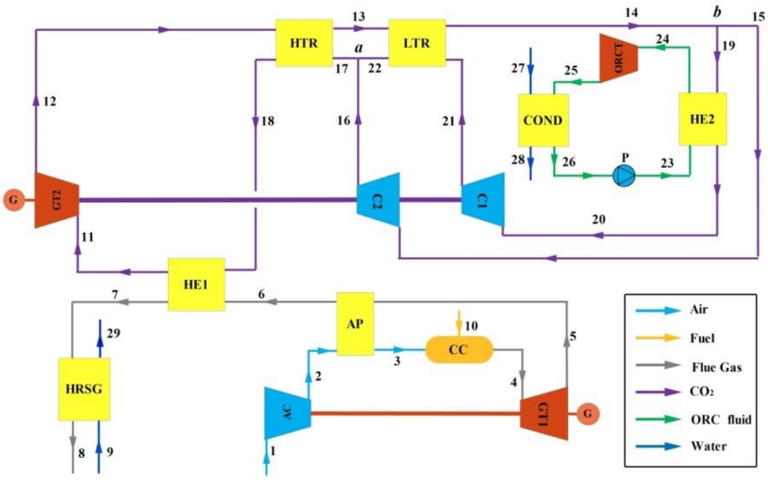

Figure 1. Schematic of the proposed cycle

$$\eta_{P,isen} = \frac{w_{P,isen}}{w_P} \quad (3)$$

The thermal or energy efficiency of the proposed system can be expressed as [21, 22]:

$$\eta_{\text{thermal}} = \frac{output_{energy}}{\dot{n}_{fuel}\overline{LHV}_{fuel}} \quad (4)$$

where, $\overline{LHV}_{fuel}$ is the lower heating value of methane as fuel and $output_{energy}$ is the net produced power of the system plus the transferred heat in the HRSG.

The entering air to the combustion chamber and the produced hot gases are considered to be ideal gas mixtures. Also, it is assumed that the combustion process in the combustion chamber is complete [4]. Thermodynamic simulation equations for each component can be found in the literature.

2. 2. Exergy Analysis The exergy balance for an energy system can be expressed as [23]:

$$\sum_{in} \dot{E}_i = \sum_{out} \dot{E}_j + \dot{E}_D + \dot{E}_L \quad (5)$$

where $\sum_{in} \dot{E}_i$ and $\sum_{out} \dot{E}_j$ are inlet and outlet exergy rates of the system, $\dot{E}_D$ and $\dot{E}_L$ represent the rate of exergy destruction and exergy loss, respectively.

Neglecting the kinetic and potential exergy changes, the specific exergy of a stream is the sum of the specific physical exergy (e_{ph}) and specific chemical exergy (e_{ch}):

$$e_i = e_{ph,i} + e_{ch,i} \quad (6)$$

Accordingly, the exergy rate of each stream will be:

$\dot{E}_i = \dot{n}_i e_i$

The specific physical exergy of a stream depends on its temperature and pressure as well as the reference ambient condition [24-27]:

$$e_i^{ph} = h_i - h_0 - T_0(s_i - s_0) \quad (7)$$

here *0* shows the reference dead state.

For a mixture of ideal gases the specific chemical exergy is expressed as [23]:

$$e_{mixture}^{ch} = \sum_i x_i e_{0,i}^{ch} + \overline{R}T_0 \sum x_i \ln x_i \quad (8)$$

here, $e_{0,i}^{ch}$ and x_i stand for the standard chemical exergy and molar fraction of the *i*th mixture component.

To define the exergy efficiency and exergy destruction of a component, it is essential to specify *product* and *fuel* rates for the component which are defined in exergy terms. The product is what we desire from a component, and the fuel is the required exergy to generate the product. Exergy efficiency indicates the percentage of providing fuel exergy to a system that is found in the product exergy. Table 1 presents the fuel and product definitions for each component of the proposed cogeneration system. Exergy efficiency and exergy destruction of each component are as follows [4, 28, 29]:

$$\dot{E}_D = \dot{E}_F - \dot{E}_P \quad (9)$$

$$\varepsilon = \frac{\dot{E}_P}{\dot{E}_F} \quad (10)$$

The total exergy efficiency of the proposed system is defined as the ratio of net produced power plus exergy of the produced steam to the input exergy, as follows [4]:

$$\varepsilon_{total} = \frac{\dot{W}_{net} + \dot{E}_{29} - \dot{E}_9}{\dot{E}_{in}} \quad (11)$$

where, $\dot{W}_{net} = \dot{W}_{GT1} + \dot{W}_{GT2} + \dot{W}_{ORC} - \dot{W}_{C1} - \dot{W}_{C2} - \dot{W}_{AC}$
here, $\dot{W}_{GT1}$,$\dot{W}_{GT2}$ and $\dot{W}_{ORC}$ are the produced power by GT1, GT2 and ORC cycle, respectively.

TABLE 1. Definitions of fuel and product for the proposed system components

Component	Exergy rate of fuel	Exergy rate of product
AC	$\dot{W}_{AC}$	$\dot{E}_2 - \dot{E}_1$
CC	$\dot{E}_3 + \dot{E}_{10}$	$\dot{E}_4$
GT1	$\dot{E}_4 - \dot{E}_5$	$\dot{W}_{GT1}$
AP	$\dot{E}_5 - \dot{E}_6$	$\dot{E}_3 - \dot{E}_2$
HE1	$\dot{E}_6 - \dot{E}_7$	$\dot{E}_{11} - \dot{E}_{18}$(7)
HRSG	$\dot{E}_7 - \dot{E}_8$	$\dot{E}_{29} - \dot{E}_9$
GT2	$\dot{E}_{11} - \dot{E}_{12}$	$\dot{W}_{GT2}$
HTR	$\dot{E}_{12} - \dot{E}_{13}$	$\dot{E}_{18} - \dot{E}_{17}$
LTR	$\dot{E}_{13} - \dot{E}_{14}$	$\dot{E}_{22} - \dot{E}_{21}$
C1	$\dot{W}_{C1}$	$\dot{E}_{21} - \dot{E}_{20}$
C2	$\dot{W}_{C2}$	$\dot{E}_{16} - \dot{E}_{15}$
COND	$\dot{E}_{25} - \dot{E}_{26}$	$\dot{E}_{28} - \dot{E}_{27}$
ORCT	$\dot{E}_{24} - \dot{E}_{25}$	$\dot{W}_{ORCT}$
ORCP	$\dot{W}_{ORCP}$	$\dot{E}_{23} - \dot{E}_{26}$
HE2	$\dot{E}_{19} - \dot{E}_{20}$	$\dot{E}_{24} - \dot{E}_{23}$

$\dot{W}_{AC}$, $\dot{W}_{C1}$ and $\dot{W}_{C2}$ are the consumed power by AC, C1 and C2. For the case of ORC, $\dot{W}_{ORC}$ means produced power by turbine minus consumed power by pump.

$$\dot{E}_{in} = \dot{E}_1 + \dot{E}_{10} \tag{12}$$

Based on the Equation (11), the term of $\dot{E}_{29} - \dot{E}_9$ refers to the exergy of the produced steam in HRSG as a byproduct.

3. MODEL VALIDATION

In order to validate the developed simulation model of the proposed cogeneration system, the reported data in the literature are used. The validation is performed for the CGAM cogeneration system. Table 2 indicates a comparison between the results of the present model for the CGAM system and the results reported by Bejan et al. [4].

4. RESULTS AND DISCUSSION

4. 1. Exergy Analysis Results

Exergy analysis of the proposed system is performed at the air compressor pressure ratio of 15 ($r_{p,AC}$=15), the compressor isentropic efficiency of 0.86 ($\eta_{is,AC}$=0.86), the maximum pressure for SCO_2 cycle of 200 bar (P_{11}=200 bar), the air preheater effectiveness of 0.86 (eff_{AP}=0.6) and the ambient temperature of 298 K (T_0=298 K).

The exergy efficiencies of the system components are calculated and compared in Figure 2, which shows that, the exergy efficiencies of the AC, GT1, GT2, HE1, HRSG and HTR, are above 90%, while the COND has the lowest exergy efficiency. This is mainly because of the temperature difference between the streams of the COND. HE2 has the similar condition.

Figure 3 shows the exergy destruction rate of each system component as a percentage of the total exergy destruction of the proposed system. Also, Table 3 presents the value of the exergy destruction rate of each component. Referring to Figure 3 and Table 3, the major part of total exergy destruction occurs at the CC (70.18%), GT1, (11.81%), and AC (6.781%). Higher exergy destruction of the CC is due to irreversibility sources, i.e. combustion, mixing and temperature difference [30]. The exergy destructions in the HRSG, GT2, HTR and HE2, are not much different, being about 1.001% to 2.08 % of the total exergy destruction. The contributions of other components in the total exergy destruction are comparatively small being about 4.853% of the total exergy destruction.

TABLE 2. Comparison between the Bejan et al.'s results and the present work for Bejan et al.'s configuration

Performance parameters		Bejan et al. [4]	Presented work for Bejan et al.'s configuration
Input values	Pressure ratio	10	10
	Combustion chamber entering temperature [K]	850	850
	Air mass flow rate [kg/s]	91.28	91.28
	HRSG inlet gas temperature	780	780
	Gas turbine inlet temperature [K]	1520	1520
Output values	Fuel-air ratio [kmol/kmol]	0.0321	0.03226
	Exergy efficiency [%]	50.3	50.28

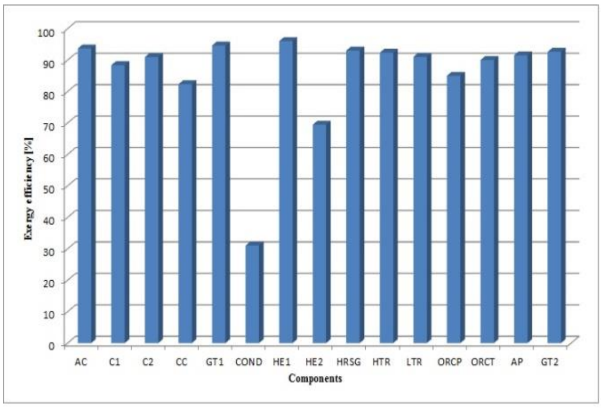

Figure 2. Exergy efficiency of the system components

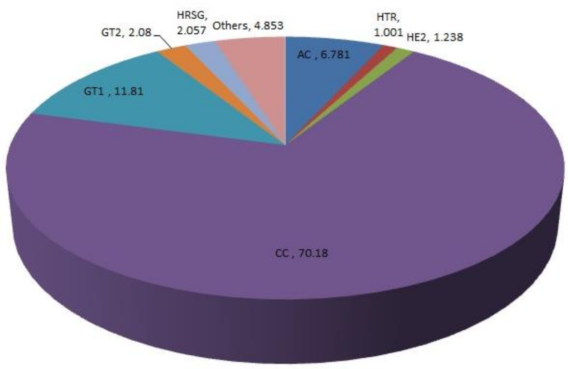

Figure 3. Percentage of total exergy destruction rate in the major components of the proposed system

TABLE 3. Exergy destruction rate of the components of the system

Component	Exergy destruction rate [kW]	Component	Exergy destruction rate [kW]
AC	2357	HTR	348
CC	24398	LTR	331
GT1	4105	C1	292
AP	751.9	C2	153.2
HE1	86.4	COND	149.4
HRSG	715.2	ORCT	70.7
GT2	722.9	ORCP	1.6
HE2	430.4	Total	34913

Figure 4 illustrates the total exergy balance of the proposed cogeneration system. As shown in this figure, 42% and 12% of the input exergy is converted to the net power and saturated steam in the HRSG, respectively. Referring to Figure 4, 37% of the input exergy is destroyed by components of the system and 9% of entering exergy is the exergy loss from the system (state 8). As previously mentioned in Equation (13), the useful part of the input exergy is the sum of the net produced power in the system and produced saturated steam in the HRSG. According to Figure 4, 54% of the input exergy will be useful and available.

4. 2. Parametric Study Parametric study helps to find the effective parameters and their effects on the system performance. This can be also done via "sensitivity analysis" [31]. However, a comprehensive parametric study is carried out in this work to investigate the effects on the system energy and exergy efficiencies of five important parameters of the system as the air compressor pressure ratio ($r_{p,AC}$), the air compressor isentropic efficiency ($\eta_{is,AC}$), the maximum pressure of SCO_2 cycle (P_{11}), the air preheater effectiveness (eff_{AP}) and the ambient temperature (T_0).

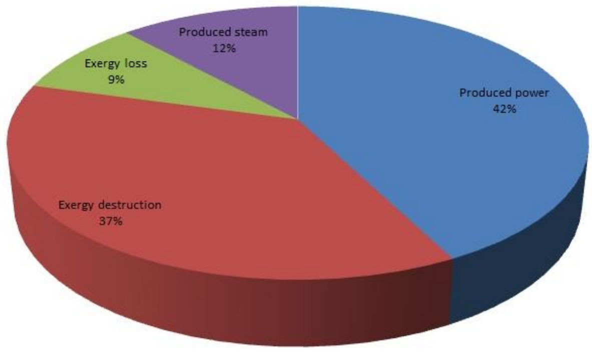

Figure 4. Total exergy balance of the proposed cogeneration system

The variation of energy and exergy efficiency with the air compressor isentropic efficiency is shown in Figure 5, indicating that the efficiencies are increased with the isentropic efficiency. In fact, increasing the isentropic efficiency of air compressor reduces the consumed power by the AC which leads to efficiency increasing.

Figure 6 shows the effects of variation of the maximum pressure of SCO_2 cycle on the energy and exergy efficiencies of the proposed cogeneration system. As shown in this figure, opposite to the energy efficiency, the exergy efficiency of the system has an optimum value of 54.32% at P_{11}=201.1 bar.

Figure 7 presents the variation of energy and exergy efficiency of the proposed system with the air preheater effectiveness (eff_{AP}). As shown in Figure 7, variation of eff_{AP} has reverse effects on the energy and exergy efficiencies. The energy efficiency of the system decreases with increasing the eff_{AP} while the exergy efficiency of the system increases. System simulation shows that in addition to the design parameters of the system, the ambient temperature has a noticeable effect on the performance of the proposed system.

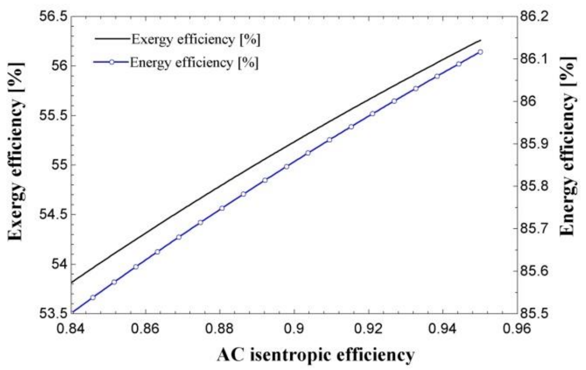

Figure 5. Variation of energy and exergy efficiency of proposed system by isentropic efficiency of the AC ($r_{p,AC}$=15, T_0=298 K, eff_{AP}=0.6 and P_{11}=200 bar)

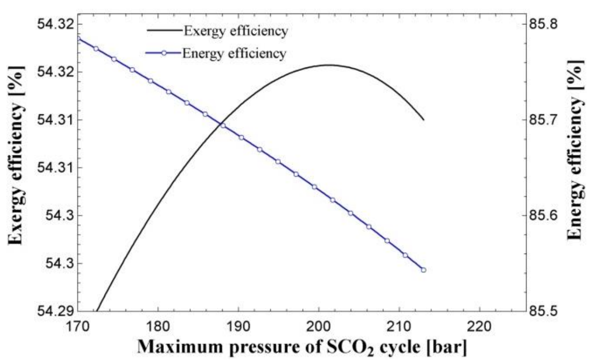

Figure 6. Variation of efficiency of the proposed system with maximum pressure of the SCO_2 ($r_{p,AC}$=15, eff_{AP}=0.6, T_0=298 K and $\eta_{is,AC}$=0.86)

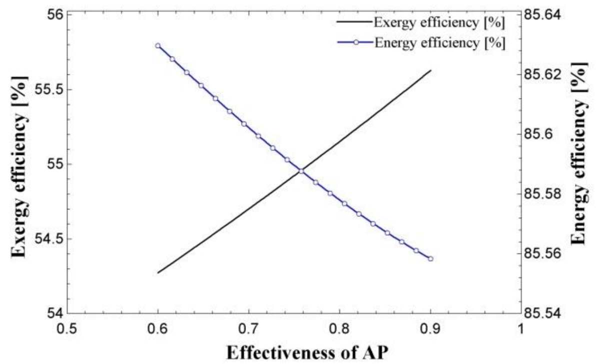

Figure 7. Effects of eff_{AP} on energy and exergy efficiency of the proposed system (T_0=298 K, $\eta_{is,AC}$=0.86, $r_{p,AC}$=15 and P_{11}=200 bar)

Thus, along with the air compressor pressure ratio ($r_{p,AC}$), the effects of ambient temperature on the performance of the system have also been studied in the following section.

The effects on the energy and exergy efficiency of the ambient temperature and air compressor pressure ratio are shown in Figures 8 and 9, respectively. Referring to these figures, an increase in the ambient temperature increases energy efficiency and decreases exergy efficiency.

To clarify the effects of the ambient temperature (T_0) on the efficiency of the proposed system, it is necessary to analyze the effects of ambient temperature on some state points of the system. Due to variation of the specific volume of air at the compressor inlet (air considered as a mixture of ideal gases), the consumed power of air compressor increases with increasing of ambient temperature and as a result, the net produced power of the system decreases.

Increasing the ambient temperature makes a growth on the AC and AP outlet temperatures (state 2 and 6) and also increases temperature of the HRSG inlet (state 7), rate of produced steam and heat rate of the HRSG.

Figure 9 provides the variation of exergy efficiency with T_0 and $r_{p,AC}$. There are three parameters which affect the exergy efficiency of the system directly; net produced power, exergy rate of produced steam in the HRSG and the exergy input.

Net produced power of the system decreases with T_0 due to AC consumed power rising. Also exergy input rate of the system decreases due to fuel rate reduction. Despite growth in the molar rate of produced steam in the HRSG, the exergy rate of produced steam decreases with the ambient temperature. Reduction in the exergy rate of produced steam is because of the effect of ambient temperature on specific physical exergy (e_{ph}) of the steam (see Equation (8)).

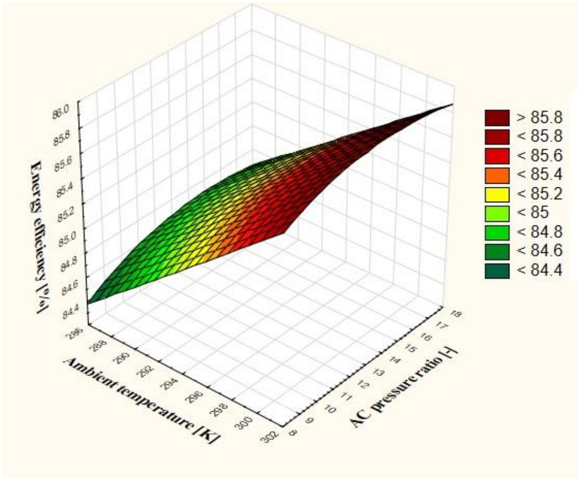

Figure 8. Effects of T_0 and $r_{p,AC}$ on the energy efficiency of the proposed system (when $\eta_{is,AC}$=0.86, eff_{AP}=0.6 and P_{11}=200 bar)

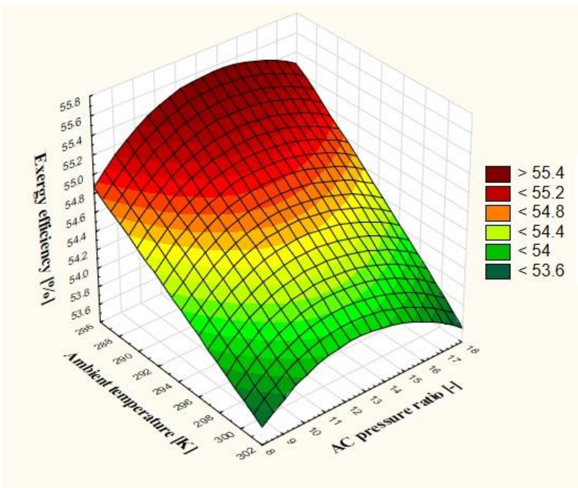

Figure 9. Effects of T_0 and $r_{p,AC}$ on the exergy efficiency of the proposed system

Referring to Figure 9, the net effect of reduction of the net produced power, exergy rate of produced steam and exergy input, is reducing the exergy efficiency of the system with ambient temperature.

4. 3. Thermodynamic Optimization As discussed above, the energy efficiency of the system is maximized at particular air compressor pressure ratios. Also, Figure 8 shows that, the optimum compressor pressure ratio changes with increasing the ambient temperature. For this reason, a thermodynamic optimization is performed to find the optimum compressor pressure ratio at different ambient temperatures using direct search method by EES software.

Table 4 outlines the optimization results. It can be concluded from this table that, with increasing the ambient temperature, the optimum air compressor pressure ratio decreases while the maximum energy efficiency increases. The values of other performance parameters of the system under the maximized energy efficiency are also given in Table 4.

Similarly, the ambient temperature changes the air compressor pressure ratio in which, the exergy efficiency is maximized. Table 5 presents the results of the optimization for maximizing the exergy efficiency of the proposed cogeneration system. As this table indicates, the values of the optimum air compressor pressure ratio and corresponding exergy efficiencies decrease with increasing the ambient temperature.

TABLE 4. Summary of the optimization results for maximum energy efficiency of the proposed system at different ambient temperatures

Optimum design/operating parameters	**T0=283 K**	**T0=288 K**	**T0=293 K**	**T0=298 K**
rp,AC	16.24	15.49	14.96	14.27
GT1 produced power [kW]	73454	72560	71879	70963
GT2 produced power [kW]	8985	9046	9111	9169
AC consumed power [kW]	36925	36660	36600	36292
C1 consumed power [kW]	3148	3159	3171	3181
C2 consumed power [kW]	1190	1169	1148	1128
Net produced power [kW]	41864	41306	40760	40219
Molar rate of produced steam [kmol/s]	0.6648	0.6719	0.6795	0.6864
Heat rate of produced steam [kW]	32241	32586	32956	33291
Exergy rate of produced steam [kW]	11986	11753	11521	11268
Entrance exergy [kW]	96630	95977	95366	94719
Fuel molar rate [kmol/s]	0.1093	0.1085	0.1078	0.107
Energy efficiency [%]	84.48	84.86	85.24	85.63
Exergy efficiency [%]	55.73	55.28	54.82	54.36

TABLE 5. Summary of the optimization results for maximum exergy efficiency of the proposed system at different ambient temperatures

Optimum design/operating parameters	**T0=283 K**	**T0=288 K**	**T0=293 K**	**T0=298 K**
rp,AC	13.85	13.44	12.92	12.51
GT1 produced power [kW]	70447	69847	69060	68412
GT2 produced power [kW]	8920	8988	9051	9116
AC consumed power [kW]	33972	34003	33847	33812
C1 consumed power [kW]	3135	3148	3159	3171
C2 consumed power [kW]	1208	1186	1165	1143
Net produced power [kW]	41739	41186	40628	40091
Molar rate of produced steam [kmol/s]	0.6575	0.6654	0.6729	0.6805
Heat rate of produced steam [kW]	31890	32273	32633	33003
Exergy rate of produced steam [kW]	11855	11640	11408	11171
Entrance exergy [kW]	96061	95460	94825	94224
Fuel molar rate [kmol/s]	0.1086	0.1079	0.1071	0.1064
Energy efficiency [%]	84.47	84.84	85.23	85.62
Exergy efficiency [%]	55.79	55.34	54.88	54.4

5. CONCLUSIONS

A novel cogeneration system based on the CGAM and SCO_2 cycles is proposed. A thermodynamic model has been developed by applying the first and second laws of thermodynamics for each system component to examine components in terms of exergy efficiency and destruction. Also a comprehensive parametric study is done to determine the effects of some key parameters on the performance of the system. Moreover, an optimization is performed to maximize the energy and exergy efficiencies. Energy and exergy efficiencies of 85.33% and 54.18%, respectively, are obtained for the proposed system, assuming the values of 15 for air compressor pressure ratio, 86% for the air compressor isentropic efficiency, 298.15 K for the ambient temperature, 0.6 for air preheater effectiveness and 200 bar for the maximum pressure of the SCO_2 cycle.

The proposed cogeneration system has a higher exergy efficiency than the CGAM system. Beside the high exergy efficiency, using the waste heat of the system to run an ORC unit, low exergy loss (state 8) and producing saturated vapor as a byproduct are the other advantages of the proposed system. Referring to exergy efficiency and exergy destruction of the components, CC, GT1 and AC are the main sources of irreversibility in the proposed cogeneration system, respectively.

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Thermodynamic Analysis and Optimization of a Novel Cogeneration System: Combination of a Gas Turbine with Supercritical CO_2 and Organic Rankine Cycles

TECHNICAL NOTE

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چکیده

تحلیل ترمودینامیکی یک سیستم ترکیبی جدید بر مبنای ترکیب سیستم تولید همزمان توربین گاز، سیستم بازتراکمی فوق بحرانی دی‌اکسید کربن و سیکل رانکین آلی ارایه شده است. یک مطالعه‌ی پارامتری جامع برای بررسی اثر پارامترهای مهم روی عملکرد سیستم پیشنهادی انجام گردیده و نهایتا یک بهینه سازی ترمودینامیکی جهت دست‌یابی به راندمان انرژی و اگزرژی بیشینه آورده شده است. نتایج نشان می‌دهد که راندمان انرژی و اگزرژی سیستم در یک مقدار مشخصی از نسبت فشار بیشینه می‌شود که این مقدار مشخص به پارامترهای دیگری نیز بستگی دارد. راندمان انرژی و اگزرژی برای سیستم پیشنهادی به ترتیب ۸۵/۳۳ و ۵۴/۱۸ درصد در شرایط پایه می‌باشد. همچنین مطالعه‌ی پارامتری نشان داد که علاوه بر پارامترهای طراحی، دمای محیط نیز روی عملکرد سیستم اثر می‌گذارد به ترتیبی که افزایش دمای محیط، راندمان انرژی را افزایش داده و راندمان اگزرژی را کاهش می‌دهد.

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