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Exergy Analysis of a Novel Combined System Consisting of a Gas Turbine, an Organic Rankine Cycle and an Absorption Chiller to Produce Power, Heat and Cold

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ABSTRACT

The current work investigates the exergy analysis of a new system to generate power, heat, and refrigeration. In the proposed system, the heat loss of a gas turbine (GT) is first recovered by a Heat Recovery Steam Generator (HRSD), then by an Organic Rankine Cycle (ORC) to generate warm water and additional power, respectively. In the ORC, reheating is used to increase the output power, the required heat of which is provided by a geothermal resource. Moreover, there is an absorption refrigeration cycle in the system that operates with the remaining geothermal heat. The exergy efficiency of the system were 50.65%; while the coefficient of performance of the refrigeration system was calculated to be 0.5. In this regard in the entire system, the combustion chamber accounted for the major exergy destruction, making the GT/HRSG system have the highest portion of 87.71%. The greatest exergy efficiency was 96%, which was obtained for the gas turbine.

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NOMENCLATURE

$\dot{E}$	Exergy rate (kW)	Ev	Evaporator
$\dot{E}_d$	Exergy destruction rate (kW)	F	Fuel
$\dot{m}$	Mass flow rate (kg s^{-1})	*Gen*	Generator
P	Pressure (kPa)	GT	Gas turbine
$\dot{Q}$	Heat transfer rate (kW)	HPT	High pressure turbine
r_{AC}	Pressure ratio of compressor (-)	IHE	Internal heat exchanger
T	Temperature (K)	LPT	Law pressure turbine
$\dot{W}$	Power (kW)	ORC	Organic Rankine cycle
Subscripts and Abbreviations		P	Pump, Product
0	Dead state	PP	Pinch point
a	Absorption chiller	RH	Reheat
AC	Air compressor	SHE	Solution heat exchanger
APH	Air preheater	**Greek Symbols**	
CC	Combustion chamber	ϵ	IHE effectiveness
Cond	Condenser	ε	Exergy efficiency

1. INTRODUCTION

Limited fossil fuels and emissions of environmental pollutant have made many researchers to search for solutions to use the lost energies. Waste heat of gas turbines (GT) with high temperature have a good potential to be recoverd by other systems [1]. The organic Rankine cycle (ORC), with the heat exchangers are suitable candidates for this purpose. The heat exchangers used the heat of the gas turbine exhaust to produce hot water (or other fluid). The residual energy of combustion gases is utilized for recovery in the ORC [2]. Chacartegui et al. [3] thermodynamically studied a combined cycle of commercial gas turbine and different ORCs. They

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improved the total efficiency by parametric optimization. They also compared a typical combined cycle of gas turbine-steam Rankine cycle to a new combined cycle of gas turbine-organic Rankine cycle. The total efficiency of the new combination increased up to 3%. However, another interesting result of their study was that the same productivity ($\approx 60\%$) was obtained for the two cases; but with the difference that the second case required lower input temperature for the gas turbine as well as the reduced NO_x and construction and maintenance costs. Moreover, a simple configuration, and the superheated output flow of the turbine are among the advantages of ORC [4]. Beside the recovering of the wasted energies, the use of renewable energy resources rather than fossil fuels is another solution to control environmental pollution and global warming. A resource that is increasingly being used is the geothermal energy, which can be employed in ORC due to its low-medium temperature [5-7]. Among geothermal advantages are its reliability, stability, cheapness, and abundance.

The combined power plants can be improved in terms of energy saving and economic issue when a hot source is used to generate electricity and also another type of energy such as cold, simultaneously. Absorption chillers are widely used as downstream cycles of geothermal heat sources to generate refrigerant vapor. Behzadi et al. [8] proposed a system consisting of a concentrated PVT, an absorption refrigeration system, and a geothermal energy resource considering energy, exergy, and exergo-economic aspects to generate power and refrigeration. Their results indicated that the loss heat recovery of the geothermal unit increased the coefficient operation factor up to about 15%.

Reviewing previous studies shows the ORC is widely considered for electricity generation by using of both waste heat recovery and geothermal heat source. Therefore, using the methods that improve its performance and increase capacity is highly recommended. One of these technics is reheating, which is usually used in a boiler for removing turbine output moisture. Despite the much attention of scholars to the geothermal energy sources because of their proven benefits, to our knowledge, there is no system that utilize this energy source for the reheating purpose. However, due to the having enough temperature and energy, it is capable to do so instead of boilers, producing much less emissions than fossil fuels, so it seems worth to be investigated. In this project, geothermal energy, before being used in the absorption chiller to create refrigerant steam, is first used for reheating the organic fluid to improve system performance.

In general, the proposed system is a combined cycle to generate power, heat, and cold. The system consists of a gas turbine whose output gas heat is first used by a recuperator and then an ORC to generate hot water and power, respectively. At the same time, the geothermal energy is employed to reheat in the ORC. Leaving the ORC, the geothermal fluid is directed to an absorption refrigeration cycle so its remaining heat would be used in the generator to generate ammonia vapor. The proposed system is analyzed in terms of exergy and the effects of design parameters on the system performance will be investigated.

2. SYSTEM DESCRIPTION

A schematic viewof the proposed system is demonstrated in Figure 1. This system involves a gas turbine cycle, an ORC, and an absorption chiller. In the gas turbine cycle, the air enters the compressor at 298 K and 1 bar. Then, to increase in temperature, it enters the air preheater, going to the combustion chamber where the fuel is sprayed at 12 bars and combustion takes place. The combustion gases leave the chamber and pass through the gas turbine. The total power generated by the gas turbine is considered 30 MW [9]. There is a heat recovery steam generator (HRSG) in the path of the combustion gases to generate saturated water vapor at 35 bars using their heat. The remaining heat of the gases is directed to the evaporator of ORC.

In the ORC, the organic fluid passes through the pump in the form of saturated liquid. It then goes to the internal heat exchanger to be preheated before entering the evaporator. It absorbed heat from the combustion gases in the evaporator and becomes saturated vapor [2], entering the high-pressure turbine. After it expands in the turbine, it is reheated by a geothermal heat resource in a heat exchanger while temperature increased. The reheating pressure was considered to be the mean of the evaporation and condensation pressure [10]. Then, the organic fluid goes to the low-pressure turbine. Since the fluid temperature is still high in its outlet, an internal heat exchanger was placed there to make the best use of the organic fluid heat. The working fluid then enters the condenser and returns back to the pump to repeat the cycle.

After exchanging heat in the ORC, the geothermal fluid goes to the generator of the absorption refrigeration cycle. In this cycle, a solution of water and ammonia was used as the absorber and refrigerant. Once the heat is absorbed by the solution from the geothermal fluid, the refrigerant vapor is separated from the absorber and goes to the condenser. The refrigerant vapor condenses in the condenser and passes through the expansion valve to the evaporator. After evaporation, the refrigerant receives heat from the ambient and creates coldness. Then, the refrigerant enters the absorber where it is absorbed by the absorber solution that comes from the generator to the absorber by reducing the pressure. Now the refrigerant and absorber solution created in the absorber with a large amount of the refrigerant fluid is pumped to the generator and the absorption refrigeration cycle is completed.

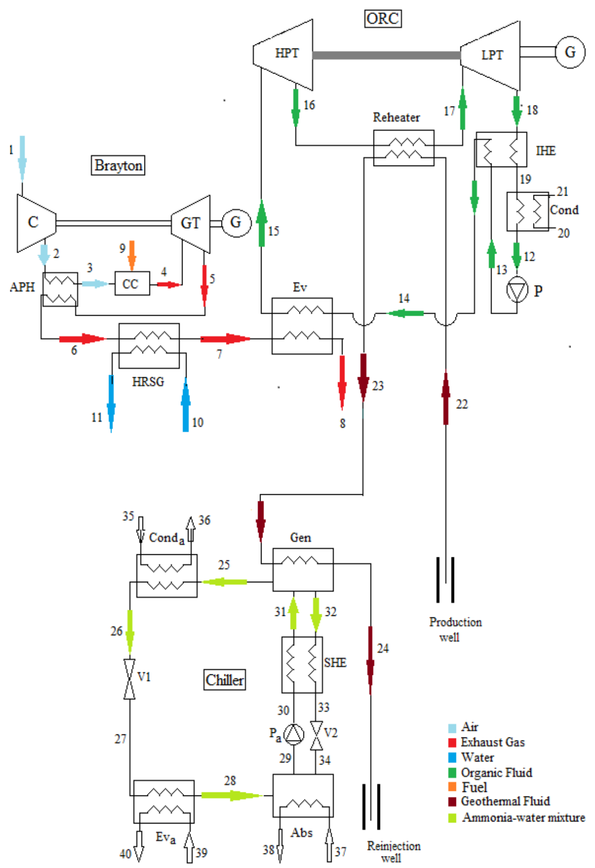


Figure 1. A schematic view of proposed triple generation system

Input data to analyze the gas turbine, organic Rankine cycle, and absorption chiller are listed in Tables 1 and 2, respectively.

TABLE 1. Parameters related to GT/ HRSG and ORC [2, 7]

GT/ORC Parameters	Value	GT/ORC Parameters	value
T_0 [K]	298.15	T_{Ev} [K]	75
P_0 [kPa]	101.3	T_{Cond} [K]	303.15
$\dot{W}_{net,GT}$[MW]	30	$\Delta T_{PP,HRSG}$ [K]	28
r_{AC} [-]	10	$\Delta T_{PP,E}$ [K]	8
η_{GT} [%]	86	$\Delta T_{PP,RH}$ [K]	6
η_{AC} [%]	86	ϵ_{IHE} [-]	0.9
η_P [%]	85	$\dot{m}_{22}$ [kg/s]	83
T_3 [K]	850	T_{25} [°C]	175
T_4 [K]	1520	P_{25} [MPa]	7

TABLE 2. Parameters related to absorption chiller cycle [11]

Chiller Parameters	Value
η_P [%]	80
P_{max} [bar]	20.33
P_{min} [bar]	4.7
ϵ_{SHE} [-]	0.8
Concentration of strong solution [%]	43
Concentration of weak solution [%]	26
$\Delta T_{PP,Ev_a}$ [K]	2

3. THERMODYNAMIC MODELING

3. 1. Presumptions

- All the processes are steady [9];
- Methane is a fuel with a lower heating value of 800661 kJ/kmol [9];
- Air and combustion gases are ideal gases [9];
- Air and gas flow in the gas turbine cycle experience 5 and 3% of pressure drop passing through the main devices, respectively [9];
- 2% of the lower heating value of the fuel is lost in the combustion chamber [9];
- The HRSG input water is at 298 K and 35 bars. The output water is saturated vapor [2].
- In the ORC, the input flow of the high-pressure turbine is saturated vapor and that of the pump is considered saturated liquid [2];
- The reheat pressure is considered to be the average of the condensation and evaporation pressures [10];
- In the absorption chiller, the generator output is considered saturated ammoniac vapor, while the condenser output is considered saturated liquid [11];

3. 1. Exergy Analysis

In the steady state, the exergy equilibrium of a control volume is given [7]:

$$\dot{E}_W - \dot{E}_Q + \sum_e \dot{m}_e ex_e - \sum_i \dot{m}_i ex_i + \dot{E}_D = 0 \qquad (1)$$

where *i* and *e* are the input and output flows, respectively, *ex* is the specific exergy, $\dot{E}_D$ is the exergy destruction rate, $\dot{E}_W$ is the work exergy, and $\dot{E}_Q$ is the exergy of the heat exchanged between the control volume and ambient stated as follows:

$$\dot{E}_Q = \left(1 - \frac{T_0}{T_j}\right)\dot{Q}_j \qquad (2)$$

$$\dot{E}_W = \dot{W} \qquad (3)$$

Specific physical exergy in a given statement, defined as follows [2]:

$$ex^{ph} = (h - h_0) - T_0(s - s_0) \qquad (4)$$

Index *0* denotes the amount of a variable in the ambient condition. Chemical exergy of an ideal gas mixture is defined as stated as follows [2]:

$$\bar{e}^{ch} = \sum_{k=1}^{n} x_k.\overline{ex}_k{}^{ch} + \bar{R}.T_0.\sum_{k=1}^{n} x_k.Lnx_k \quad (5)$$

In which $\overline{ex}_k{}^{ch}$ is the chemical exergy per mole, x_k is the mole fraction of component k of the gas mixture, and $\overline{R}$ is the universal gas constant. Exergy destruction of each system component is calculated by deducting the product exergy from the fuel exergy, while exergy efficiency is calculated by dividing the exergy of fuel by the exergy of product. The total exergy efficiency (i.e. the second law efficiency) is defined as follows:

$$\varepsilon = \frac{\dot{W}_{Net,system}+(\dot{E}_{11}-\dot{E}_{10})+(\dot{E}_{40}-\dot{E}_{39})}{\dot{E}_9+(\dot{E}_{22}-\dot{E}_{24})} \quad (6)$$

$$\dot{W}_{net,system} = \dot{W}_{GT} + \dot{W}_{ORC} - \dot{W}_{P_a} \quad (7)$$

$$\dot{W}_{ORC} = \dot{W}_{HPT} + \dot{W}_{LPT} - \dot{W}_{AC} - \dot{W}_{P} \quad (8)$$

TABLE 4. Temperature of different points of present work compared to literature [11] for absorption refrigeration cycle

Point	Fluid	T (ºC) [Present work]	T(ºC) [11]
25	Ammonia	49.95	49.98
26	Ammonia	49.95	49.98
27	Ammonia	2.45	2.44
28	Ammonia	2.45	2.44
29	Ammonia-water	49.65	47.61
30	Ammonia-water	49.95	47.93
31	Ammonia-water	107.8	92.77
32	Ammonia-water	146.75	144.34
33	Ammonia-water	69.05	67.21
34	Ammonia-water	69.35	67.47

4. RESULTS and DISCUSSION

First, the results are compared to those of the previous studies to validate them. Bejan et al. [9], Mohammadi et al. [11], Yari and Mahmoudi [12] have employed to validate the results obtained for the gas turbine, ORC, and absorption chiller, respectively. The results are indicated in Tables 3 and 4, showing good validity.

Table 5 represents the exergy results. For each device, the fuel exergy rate, product exergy rate, and exergy destruction rate were determined. The lowest and highest exergy destruction was obtained for the pump and the combustion chamber. Thus, a reduction of this variable in CC can considerably reduce it for the entire system. A method is to preheat the air before it enters the combustion chamber [9]. When the air is preheated in the APH, the required mass flow rate of fuel diminishes and thus the exergy efficiency rises.

TABLE 3. Exergy rate of different points of present work compared to literature [9] and [12] for gas turbine and ORC, respectively.

Point	Fluid	$\dot{E}$(MW) [Present work]	$\dot{E}$(MW) [9]	$\dot{E}$(MW) [12]
1	Air	0	0	-
2	Air	27.149	27.5382	-
3	Air	40.964	41.9384	-
4	Comb.gases	100.706	101.4538	-
5	Comb.gases	37.990	38.7823	-
6	Comb.gases	21.814	21.3851	-
9	Fuel	84.994	84.9939	-
12	R123	0.08753	-	0.08868
13	R123	1.514	-	1.515
14	R123	2.827	-	2.831
15	R123	66.992	-	66.969

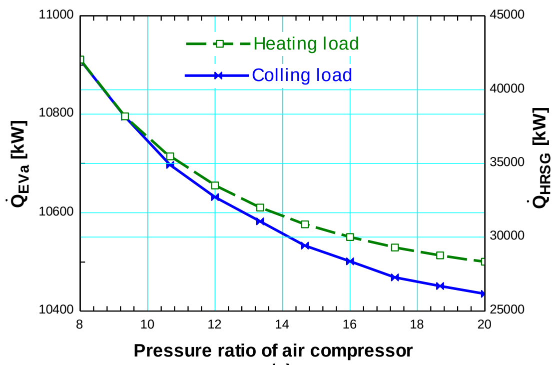
TABLE 5. The numerical results of exergy for every part of the system

Components	$\dot{E}_F$ [MW]	$\dot{E}_P$ [MW]	$\dot{E}_d$ [MW]	ε [%]
AC	27.049	25.701	1.347	95.02
APH	15.313	13.079	2.235	85.41
CC	119.608	95.343	24.265	79.71
GT	59.376	57.049	2.327	96.08
HRSG	16.930	13.324	3.606	78.7
Ev	0.879	0.497	0.382	56.49
HPT	0.163	0.135	0.028	82.82
RH	0.418	0.327	0.091	72.28
LPT	0.541	0.479	0.062	88.56
IHE	0.219	0.145	0.074	66.08
Cond	0.054	0.021	0.033	38.2
Pump	0.010	0.0085	0.0015	85.42
Gen	5.270	2.959	2.311	56.14
$Cond_a$	0.872	0.094	0.778	10.74
V1	3.456	3.228	0.228	93.39
Ev_a	0.85	0.4	0.45	46.99
Abs	2.286	0.733	1.553	32.06
P_a	0.109	0.089	0.020	81.52
SHE	3.376	1.883	1.494	55.76
V2	0.535	0.479	0.056	89.53

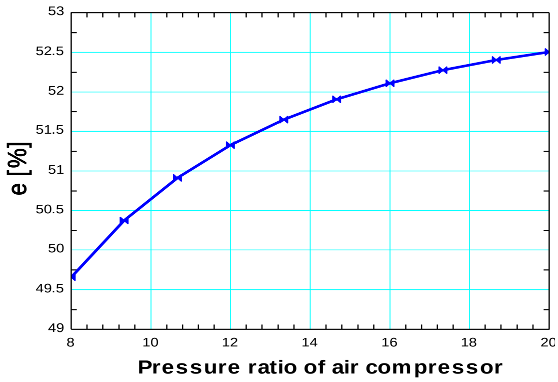
The greatest portion in the exergy destruction is 58.7% and belongs to the combustion chamber. As a result, the gas turbine cycle has the highest portion of 81.71%, followed by the ORC and absorption refrigeration cycle with 1.63 and 16.67% of this parameter, respectively. Thus, any change in the thermodynamic parameters of the gas turbine cycle can significantly affect system performance. Furthermore, Table 5 indicates that the highest exergy efficiencies belonged to the gas turbine and air compressor, while the lowest exergy efficiency belonged to the condenser of the absorption refrigeration cycle. Overall results obtained from the proposed system's modeling, are: 30.498 MW of total output power, 36.715 MW of generated heat in HRSG, and 10.532 MW of cooling load. Moreover, the overall exergy efficiency and the coefficient of performance (COP) of chiller were obtained to be 50.65% and 0.5, respectively. The output power of ORC improved by 29.4% with respect to non-reheating mode. Now let's explore the effects of a number of key parameters on the proposed cycle's behavior.

Figure 2a demonstrates the compressor pressure ratio effect on the heating and cooling loads in the HRSG and the evaporator of chiller. While Fig. 2b shows its effects on the exergy efficiency. As it is seen in Fig. 4a, the pressure ratio inversely affects both parameters, that is, an increase in the compressor ratio rises its output temperature (the combustion chamber input temperature in other words). This process reduces the mass flow rate of fuel and combustion gas, and heat generation in the vapor generator. Considering the energy balance in the ORC evaporator, the mass flow rate and the output power of ORC increased. Increased ORC output power and reduced fuel consumption in the combustion chamber are the factors that raise the exergy efficiency. Moreover, a reduction in the geothermal fluid temperature going to the generator reduces the refrigeration capacity.

Figure 3a demonstrates variations in the cooling load of the evaporator and the COP of the absorption refrigeration cycle, while Figure 3b illustrates the exergy efficiency changes for the proposed system based on the reheating pressure. An increase in the reheating pressure raises the output power of the ORC and declines the heat absorption for the reheating. The input geothermal fluid temperature of the generator and its heat transfer rate increase, leading to the enhanced refrigeration of the evaporator of the chiller. The mentioned changes rise the system exergy efficiency and decrease the COP of the chiller. These changes are slight because the system performance is more affected by the gas turbine cycle parameters than by the ORC.

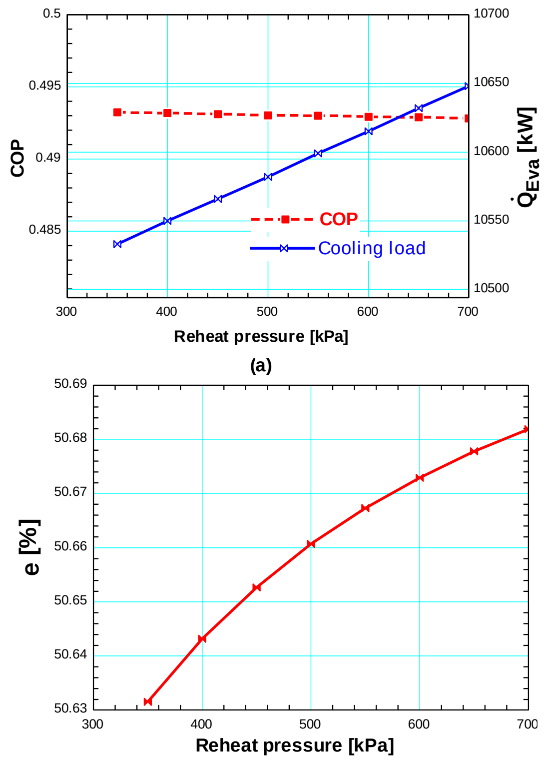


(a)



(b)

Figure 2. Compressor's pressure ratio effect on a) heating capacity of steam generator and refrigeration capacity of absorption chiller and b) exergy efficienciy



(b)

Figure 3. Reheating pressure effect on a) evaporator's refrigeration capacity and COP of absorption chiller cycle and b) exergy efficiency of system

5. CONCLUSION

This study investigated the exergy analysis of a triple generation system of power, heat, and cold. The system consisted of three cycles: gas turbine cycle, ORC, and absorption refrigeration cycle. Beside the heat recovering from the output gases of the gas turbine, the ORC used a geothermal resource for reheating, the result of which was the increased output power and system efficiency. The major results are as follows:

- Exergy efficiency was obtained 50.65%;
- The output power of ORC improved by 29.4% with respect to non-reheating mode.
- The COP of the refrigeration system was obtained to be 0.5;
- The highest portion of the exergy destruction was calculated to be 81.71%, which belonged to the GT/HTSG among the three cycles;
- The highest and lowest exergy efficiencies were obtained 96%, and 11%, which belonged to the gas turbine and the condenser of the chiller, respectively;
- GT/HRSG parameters played greater roles in the efficiency of the system than those of other parameters.

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چکیده

در کار حاضر، به مطالعه اگزرژی یک سیستم جدید برای تولید توان، گرما و سرما پرداخته شده است. در سیستم پیشنهادی، حرارت اتلافی یک سیکل توربین گاز ابتدا توسط یک مولد بخار برای تولید آب گرم و سپس یک سیکل رانکین آلی برای تولید توان بیشتر بازیافت می گردد. در ORC، برای افزایش توان خروجی از گرمایش مجدد استفاده می شود که گرمای مورد نیازآن از یک منبع ژئوترمال تامین می شود. یک سیکل تبرید جذبی نیز در سیستم وجود دارد که با استفاده از مابقی گرمای ژئوترمال کار می کند. راندمان اگزرژی سیستم ۵۰.٦۵٪ و ضریب کارآیی سیکل تبرید ۰.۵ بدست می آیند. سیکل GT / HRSG با ۸۱.۷۱٪ دارای بیشترین سهم در تخریب اگزرژی در کل سیستم می باشد. بالاترین بازده اگزرژی با مقدار ۹٦٪ به توربین گازی اختصاص دارد.

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