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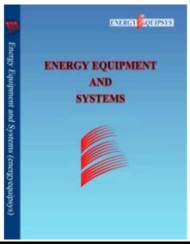

Simulation and exergy evaluation of a MED unit based on waste heat recovery from a gas turbine unit

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ABSTRACT

Integrating MED-TVC unit with gas turbine cycle (GTC) and organic Rankine cycle (ORC) can be an effective way to take advantage of the hot exhaust gas of gas turbines. In this study, a multi-product system consisting of GTC, MED-TVC, and ORC is investigated. The energy and exergy analysis is carried out and influences some design variables such as inlet air temperature of air compressor, air compressor pressure ratio, high pressure in ORC, pinch point temperature difference, the pressure of motive steam, and TVC compression ratio on the developed system are examined. Calculation shows that the developed unit can produce 39.6 MW of power and 137.3 kg/s of fresh water with a gain output ratio of 4.41 and energy efficiency of 21.5%. According to the result, precooling the air at the entrance of the air compressor and decreasing the pinch point temperature can lead to enhancement exergy and energy efficiency of GTC and the gain output ratio of MED unit, respectively. In addition, the highest exergy destruction takes place in the combustion chamber and desalination unit.

Keywords: Exergy; Desalination; Multi-Effect; Freshwater, Heat Recovery.

1. Introduction

Increasing fuel consumption, reducing fossil fuel resources, and the need to control environmental pollution show the importance of optimal heat energy recovery and prevention of its loss in various industries [1]. Today, optimal energy consumption has been proposed as one of the major indicators in assessing the development of communities. The high intensity of energy consumption in electricity generation processes increases production and application costs and reduces the efficiency of material extraction in industrial products. Increasing energy efficiency and reducing waste are achieved by designing optimal heat recovery systems.

One of the energy-consuming processes in the industry especially for potable water production is thermal desalination systems. The evaporative desalination systems consume a large amount of thermal energy. Multi-effects distillation (MED) is one of the most widely used methods. the MED system can be combined with the thermal power plant units and they receive the necessary energy for desalinating saltwater by waste heat boiler [2].

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For example, these power cycles can be a gas turbine cycle (GTC) or a gas turbine cycle and solid oxide fuel cell (SOFC) combination [3].

The waste heat from these cycles can be employed as a heat source for bottoming systems. Several studies have been performed in the field of system combination to achieve higher performance.

Singh et al. [4] investigated a hybrid system including different subsystems and examined the impacts of some variables on the system.

Musharavati et al. [5] studied a new integrated biomass system for power generation and potable water. They applied energy, exergy and exergo-economic to the system and carried out a multi-objective optimization based on the artificial neural network (ANN) and Grey Wolf Optimizer. It was reported that according to the applied optimization the exergy efficiency and cost rates was 15.61% and 206.78 $/h.

Rezvani Dastgerdi et al. [6] studied a new MED-based desalination process with low-grade energy resources. A comparison of the new system with optimized traditional MED showed that it produced 45% more fresh water. Maheswari et al. [7] examined a system to produce desalinating seawater from an internal combustion engine. They reported that the waste heat of a 5 hp engine could produce 3 L/h distilled water. Also, it was claimed that by using the heat of waste exhaust the overall efficiency improved and the thermal pollution reduced considerably.

In another research Kim et al. [8] optimized a supercritical carbon dioxide Rankine cycle for GT waste recovery. Energy and exergy examinations were carried out and the effect of upper pressure of S-CO_2 Rankine cycle for waste heat recovery was presented. In another work, Musharavati et al. [3] developed a power and heat energy system in the form of an intercooler gas turbine system integrated with a hot water system and thermoelectric generator. The result of optimization in the form of Pareto points indicated that the energy efficiency, total exergy destruction rate and electricity cost rate are 53.91%, 2392.01 kW, and 52.29 $/h.

In this study, a combined system consisting of GTC, MED-TVC, waste heat boiler, and ORC cycle is modeled in MATLAB and the energy and exergy performance of the system is investigated. The influence of some design variables such as compression ratio of thermal vapor compressor (TVC), inlet air temperature of air compressor, air compressor pressure ratio and so on are studied. The main novel aspects of current work are integrating an MED-TVC unit and thermodynamic basement of system.

Nomenclatures

γ_g	Heat ratio of gas turbine
CR	Compression ratio of ejector
ER	Ejector expansion ratio
Ex	Exergy, J kg^{-1}
h	Enthalpy, J kg^{-1}
LHV	Lower heating value, J kg^{-1}
m	Mass flow rate, kg s^{-1}
n_a	Moles of air
n_f	Moles of fuel
PP	Pinch point
R_a	Ejector entrainment ratio
R_{ac}	Air compressor pressure ratio
T	Temperature, K
W	Power, MW
γ_a	Specific heat ratio of air compressor
η_{ac}	Air compressor isentropic efficiency
η_{gt}	Gas turbine isentropic efficiency

subscripts

ch	Chemical
ph	Physical

Greek symbols

η	Efficiency
λ	Fuel to air ratio

Abbreviations

GOR	Gain output ratio
GTC	Gas turbine cycle
MED	Multi-effect desalination
TVC	Thermal vapor compressor

2. System description

The proposed combined system in this study contains GTC, MED-TVC, waste heat boiler, and ORC, which is illustrated in Fig. 1. At first, the air is compressed in the compressor and reacts with the fuel in the combustion chamber. Hot exhaust gas (state 5) enters the gas turbine and after generating power, the exhaust gases of the gas turbine (state 6) enter the waste heat boiler and supply the required heat for the desalination system. The saturated motive steam exits the evaporator of the waste heat boiler and enters the steam ejector (TVC) (state 17), where it mixes with low-pressure suctioned vapors from the fifth effect (state 42) and goes to the first effect. After heat exchange with saline water sprayed on the evaporator tubes in the first effect, this steam condenses and returns to the waste heat boiler (state 11). To improve the system performance, exhaust gases of the waste heat boiler are used as a heat source for the ORC cycle (evaporator 1) and the ORC turbine produces power.

In the MED unit, the generated vapor in effects feeds to the next effects and condenses after the vaporization of saline water. In addition, the brine water enters the next effect to evaporate saltwater. Condensed vapor in each evaporator in a flashing box generates more distilled water.

3. Thermodynamic analysis

To simulate the developed system in MATLAB, the thermophysical properties of medium fluids are found using CoolProp software. The first law of thermodynamics is used for each part of the present plant.

3.1. Assumptions

The following hypotheses in modeling the proposed system are considered:

- Air is considered an ideal gas comprised 79% N_2 and 21% O_2.
- Heat and pressure losses in pipes are negligible.
- Combustion chamber 4% pressure loss takes place and there is 2% heat loss in the combustion chamber.
- The environment temperature and pressure are considered 298.15 K and 101.325 kPa, respectively.
- The working pressure of ORC is considered R600a.
- In the MED-TVC unit, the temperature difference of the adjacent effects is the same.
- The working medium in ORC is R600a.
- In the MED-TVC unit, the temperature difference of the adjacent effects is the same.

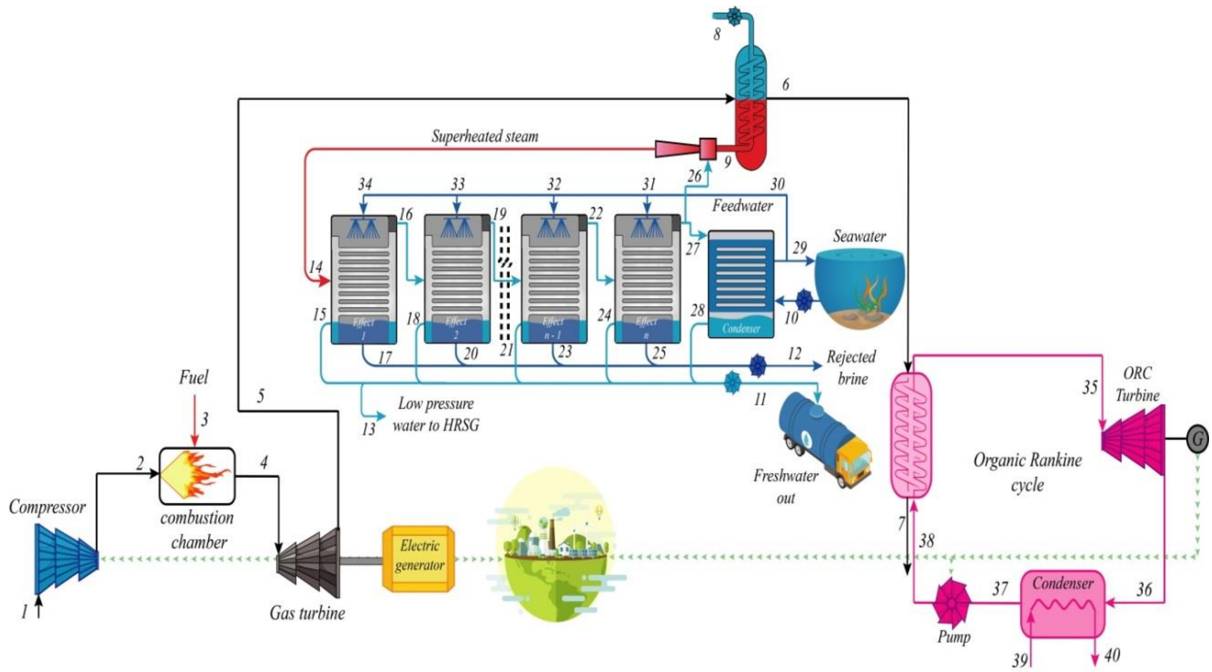


Fig. 1. The diagram of the multi-generation system.

3.2. Gas turbine cycle

The governing equation of the air compressor can be expressed as [9]:

$$T_2 = T_1\left[1+\frac{1}{\eta_{ac}}\left(R_{ac}^{\frac{\gamma_a-1}{\gamma_a}}-1\right)\right] \quad (1)$$

$$\dot{W}_{ac} = \dot{m}_{air}\left(h_2 - h_1\right) \quad (2)$$

where $\dot{W}_{ac}$ is the consuming work in the compressor unit and m_{air} is the inlet air mass flow rate of the air compressor.

The fuel is methane and the combustion process takes place in the combustion chamber with extra air. The combustion equation can be expressed as follows:

$$\begin{aligned}&\lambda CH_4 + 1[0.7748N_2 + 0.2059O_2 + \\ &0.0003CO_2 + 0.019H_2O] \rightarrow \\ &(1+\lambda)\left[y_{N2}N_2 + y_{O_2}O_2 + \right. \\ &\left. yCO_2CO_2 + y_{H_2O}H_2O\right]\end{aligned} \quad (3)$$

where λ is the fuel-to-air ratio as follows:

$$\lambda = \frac{n_f}{n_a} \quad (4)$$

The molar fraction of combustion products element, y, can be expressed as:

$$\lambda = \frac{n_f}{n_a} \quad (5)$$

$$y_{O2} = \frac{0.2059 - 2\lambda}{1+\lambda} \quad (6)$$

$$y_{CO2} = \frac{0.0003 + \lambda}{1+\lambda} \quad (7)$$

$$y_{H_2O} = \frac{0.019 + 2\lambda}{1+\lambda} \quad (8)$$

The governing equations of the GT unit are as follows [9]:

$$T_6 = T_5\left[1-\eta_{gt}\left(1-\left(\frac{P_5}{P_6}\right)^{\frac{1-\gamma_g}{\gamma_g}}\right)\right] \quad (9)$$

$$\dot{W}_{gt} = \dot{m}_5\left(h_5 - h_6\right) \quad (10)$$

where $\dot{W}_{gt}$ is the power generated by the gas turbine. $\dot{m}_5$ is the mass flow rate of outlet gases of the combustion chamber and it is obtained according to mass conservation:

$$\dot{m}_5 = \dot{m}_{air} + \dot{m}_{fuel} \quad (11)$$

3.3. Desalination unit

The energy balance of the waste heat boiler in the evaporator and economizer can be written as shown in Eqs (12) and (13), respectively:

$$\dot{m}_7\left(h_7 - h_{ge}\right) = \dot{m}_{14}\left(h_{15} - h_{14}\right) \quad (12)$$

$$\dot{m}_7\left(h_{ge} - h_8\right) = \dot{m}_{12}\left(h_{13} - h_{12}\right) \quad (13)$$

h_{ge} is the enthalpy of the products that leave the evaporator and enters the economizer. $T_{pinchpoint}$ is one of the important parameters for waste heat boiler and can be expressed as:

$$T_{pinchpoint} = T_{ge} - T_{14} \quad (14)$$

$$\dot{W}_{pump1} = \dot{m}_{11}\left(h_{12} - h_{11}\right) \quad (15)$$

where $\dot{W}_{pump1}$ is the power consumption of pump 1. The TVC mass balance can be written as follow:

$$\dot{m}_{28} = \dot{m}_{15} + \dot{m}_{42} \quad (16)$$

$$CR = \frac{P_{28}}{P_{42}} \quad (17)$$

$$ER = \frac{P_{15}}{P_{42}} \quad (18)$$

where cr is the ejector compression ratio and er is the expansion ratio. for calculating the entrainment ratio (R_a) the semi-empirical model is used by ref. [10]:

$$R_a = 0.235\frac{\left(P_{28}\right)^{1.19}}{\left(P_{42}\right)^{1.04}}\left(ER\right)^{1.05} \quad (20)$$

The entrained vapor (m_{42}) can be calculated by:

$$\dot{m}_{42} = \frac{\dot{m}_{15}}{R_a} \quad (21)$$

MED unit mathematical model proposed by Ref. [10], is used in this study. Finally, the performance of MED-TVC can be shown by the ratio of the produced fresh water to the motive steam in the MED-TVC plant.

$$GOR = \frac{\dot{m}_{19}}{\dot{m}_{15}} \tag{22}$$

3.4. ORC cycle

The evaporator 1 governing equations can be defined as:

$$\dot{m}_8 \left(h_8 - h_9 \right) = \dot{m}_{22} \left(h_{25} - h_{22} \right) \tag{23}$$

where m_8 is the mass flow rate of flue gas at the outlet of the waste heat boiler.

The power generated in the ORC turbine and the consuming work in the ORC pump can be obtained as follows:

$$\dot{W}_{ORC,Turbine} = \dot{m}_{25} \left(h_{25} - h_{24} \right) \tag{24}$$

$$\dot{W}_{ORC,pump} = \dot{m}_{23} \left(h_{22} - h_{23} \right) \tag{25}$$

3.5. Exergy investigation

The equation of the exergy balance for each element is written as below according to thermodynamic laws [11]:

$$\sum \dot{E}x_{in,i} = \sum \dot{E}x_{out,i} + \dot{E}x_{destruction,i} \tag{26}$$

$$\dot{E}x_{destruction,total} = \sum \dot{E}x_{destruction,i} \tag{27}$$

where $\sum \dot{E}x_{in,i}$ is the sum of exergy of work, exergy of heat transfer and exergy of flow rates at inlet and $\sum \dot{E}x_{in,out}$ is the sum of these exergies at the outlet of component i. $\dot{E}x_{destruction,i}$ is the exergy destruction in component i and $\dot{E}x_{destruction,total}$ is the exergy destruction rate for the entire system.

The physical ($\dot{E}x_{ph,j}$) and chemical exergy ($\dot{E}x_{ch,j}$) can be calculated by Ref. [12]. The physical and chemical exergy of saline flows can be obtained by Ref. [13]. Then the total exergy at point j ($\dot{E}x_j$) can be obtained as below:

$$\dot{E}x_j = \dot{E}x_{ph,j} + \dot{E}x_{ch,j} \tag{28}$$

The exergy destruction ratio for each component (γ_i) can be calculated by:

$$\gamma_i = \frac{Ex_{destruction,i}}{Ex_{destruction,total}} \times 100\% \tag{29}$$

3.6. System performance

The system performance can be assessed by calculating the power generation, energy and exergy efficiency.

$$\dot{W}_{net,system} = \dot{W}_{net,ORC} + \dot{W}_{net,GTC} - \dot{W}_{pump1} \tag{30}$$

$$\dot{W}_{net,ORC} = \dot{W}_{ORC,turbine} - \dot{W}_{ORC,pump} \tag{31}$$

$$\dot{W}_{net,GTC} = \dot{W}_{gt} - \dot{W}_{ac} \tag{33}$$

$$\eta_{energy,system} = \frac{\dot{W}_{net,system}}{\dot{m}_{fuel}.LHV_{fuel}} \tag{33}$$

$$\eta_{exergy,system} = \frac{\dot{E}x_{in,system} - \dot{E}x_{destruction,total}}{\dot{E}x_{in,system}} \tag{34}$$

where $\dot{E}x_{in,system}$ is input fuel exergy.

4. Model Validation

MED-TVC proposed model in this study is compared to results of other plant modeling Ref. [14] and experimental data of commercial plant reported by Ref. [15]. According to the results reported in Table 1, there is a good agreement between the presented simulation and other literature.

Table 1. Comparison of developed simulation with data in Ref. [14] and Ref. [15].

Plant conditions	Proposed model	Ref. [14]	Ref. [15]
Effects number	5	5	5
Pressure of motive steam (kPa)	1500	1500	1500
Fow rate of motive steam (kg/s)	101.7	101.70	101.7
Temperature of feed seawater (K)	316.15	316.15	316.15
Temperature of cooling water (K)	303.15	303.15	303.15
Top brine temperature, K	336.15	336.15	336.15
Minimum brine temperature, K	318.15	318.15	318.15
Entrainment ratio of ejector	0.82	0.82	0.82
Expansion ratio	100.40	100.40	100.4
Compression ratio	1.910	1.91	1.91
Fresh water flow rate (kg/s)	867.30	791.00	793.1
Gain output ratio	8.53	7.778	7.84

5. Result and discussion

The system performance is investigated by applying input parameters that have been presented in Table 2 and output parameters are presented in Table 3.

It is seen that compared to the use of gas turbine cycle alone, not only the overall efficiency has not been reduced, but with the use of hot turbine exhaust gases, a MED-TVC unit with gain output ratio of 4.41 and ORC cycle with 14.41% energy efficiency and 50.4% exergy efficiency can also be set up.

Figure 2 illustrates the exergy destruction ratio of all elements of the plant. It is seen that the maximum of exergy destruction happens in the combustion chamber because of the combustion reaction's irreversibility and equals 69.3% of total exergy destruction. The second great exergy destruction takes place in the MED-TVC unit and is equivalent to 19.5% of total exergy destruction because there are irreversible processes such as mixing and frictions in the TVC ejector. The waste heat boiler has 6.34% of total exergy destruction because of the large temperature difference in heat transfer process between the motive steam line and the exhaust gas of GT. In the following, the effect of several parameters on system performance is examined.

5.1. Pressure ratio of the compressor

Figure 3. Depicts the influences of the pressure ratio of the compressor on the plant performance. It can be found that with increasing the compressor pressure ratio due to a decrease in the heat transfer in the boiler (approximately 30 MW) (Fig 2(a)), the GOR decreases and heat transfer in evaporator 1 increases, so the power output of ORC increases about 34.6%. The mass flow rate of fuel in the combustion chamber decreases with the compressor pressure ratio. Therefore, the energy efficiency of the plant increases by about 5%.

Table 2. Input variables for the thermodynamic simulation

GT		MED-TVC		ORC	
Variable	Value	Variable	Value	Variable	Value
Ambient temperature (K)	298.15	Pressure of motive steam (kPa)	2000	Turbine efficiency of ORC	88
Ambient pressure (kPa)	101.3	Feed water temperature (°C)	40	High pressure of ORC (kPa)	1600
Air mass flow rate (kg/s)	100.93	Number of effects	5	Low pressure of ORC (kPa)	404.2
Compressor pressure ratio	8.5234	Compression ratio	2.1	Pumps efficiency	0.85
Compressor efficiency, %	85	First brine temperature (°C)	60		
GT input temp (K)	1492	Fifth brine temperature (°C)	45		
GT efficiency (%)	88	Sea water salinity (g/kg)	36		
PP temp of boiler (K)	40	Maximum brine salinity (g/kg)	70		
LHV (kJ/kg)	50147.5	Seawater initial temperature (°C)	30		

Table 3. The Output parameter of the suggested system

Item	Value	Item	Value
Fuel flow rate (kg/s)	3.67	Energy efficiency of ORC (%)	14.41
GT output (MW)	38.64	Overall energy efficiency of system (%)	21.5
ORC output (MW)	1	Exergy efficiency of GTC (%)	19.7
System output (MW)	39.6	Heat transferred in waste heat boiler (MW)	70.62
Energy efficiency of ORC (%)	11.5	Desalinated water (kg/s)	137.27
Energy efficiency of GTC (%)	21	GOR	4.41
exergy efficiency of ORC %	50.4	Motive steam mass flow rate, Kg/s	31.16
		ORC medium mass flow rate, Kg/s	22.1

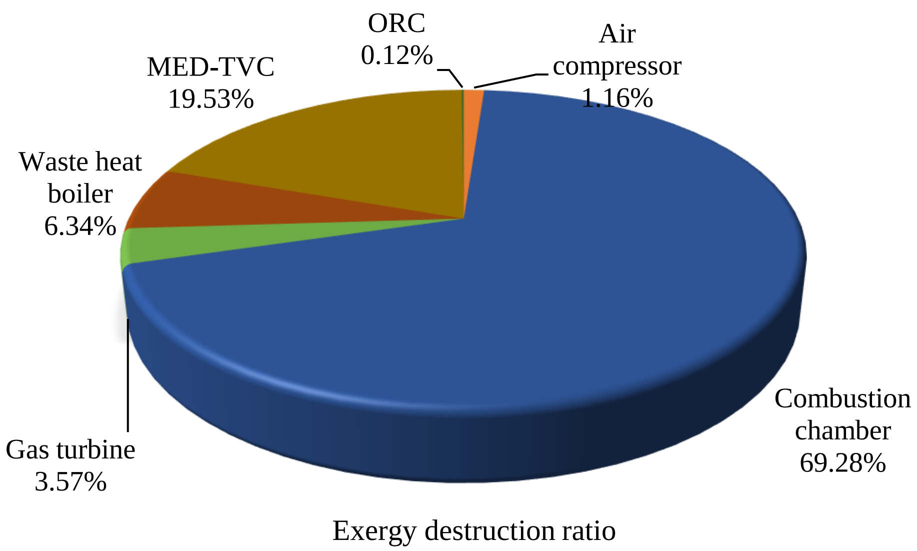


Fig. 2. The exergy destruction ratio of the main elements in the developed plant

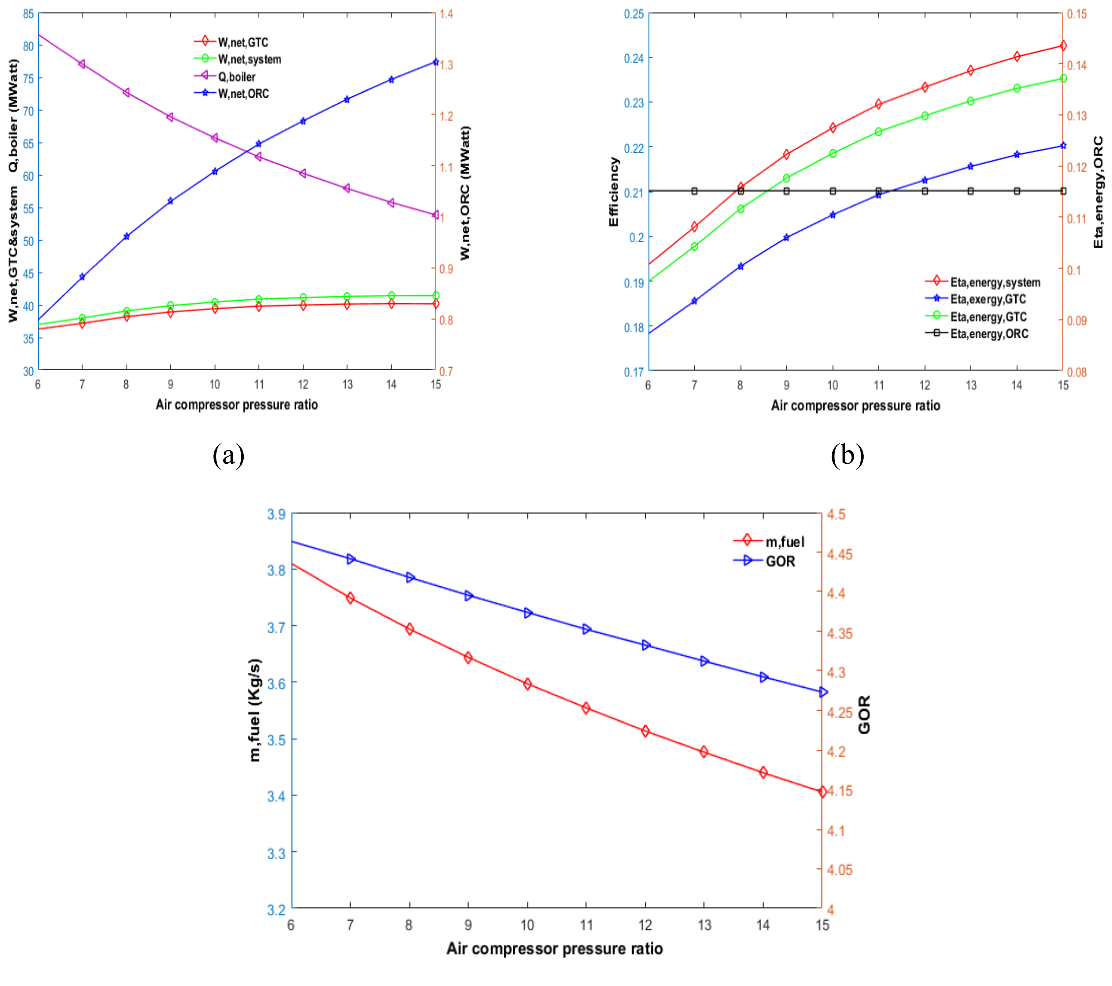


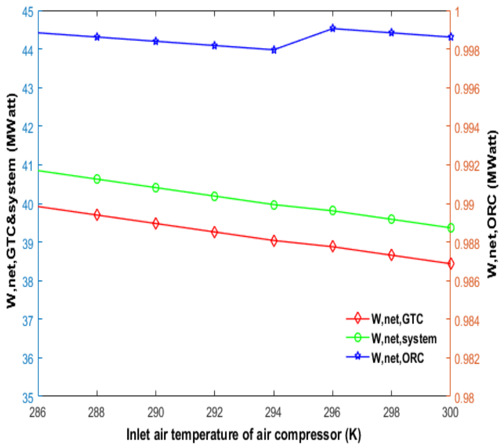
Fig. 3. Variations of suggested plant performance with changing compressor pressure ratio

5.2. Inlet temperature to the compressor

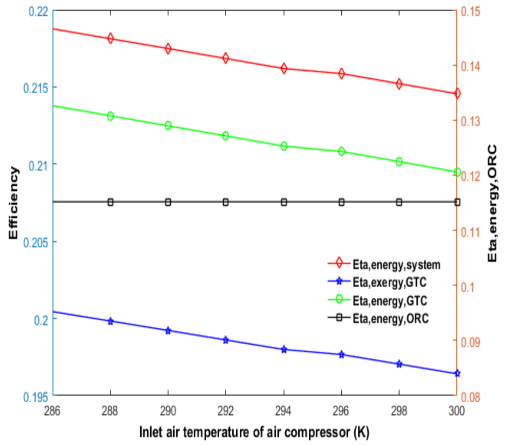
Figure 4 shows the impacts of the inlet temperature of the compressor on the suggested plant performance. It can be observed that with increasing inlet temperature of the compressor, the power generation of ORC, GTC and plant overall efficiency decrease. So that the net produced power of the system and GT decreases about 0.1 MW per K increase in inlet temperature (Fig. 4(a)). The reason for this is that the airflow rate rises with the precooling of the inlet air, so the power generation of GT would increase. In addition, the GOR remains to fix and the fuel flow rate declines with the inlet temperature of the compressor (Fig. 4(c)).

5.3. PP temperature

Figure 5 represents the influences of PP temperature of the boiler on the present plant performance. It is observed that the heat transfer rate in the boiler decreases by about 0.1 MW per K enhances in PP temperature consequently the motive steam mass flow rate decreases. Therefore fresh water generation reduces and the heat transfer rate in evaporator 1 increases. Therefore, the power generation of ORC increases by about 1 MW per K increase in PP temperature.



(a)



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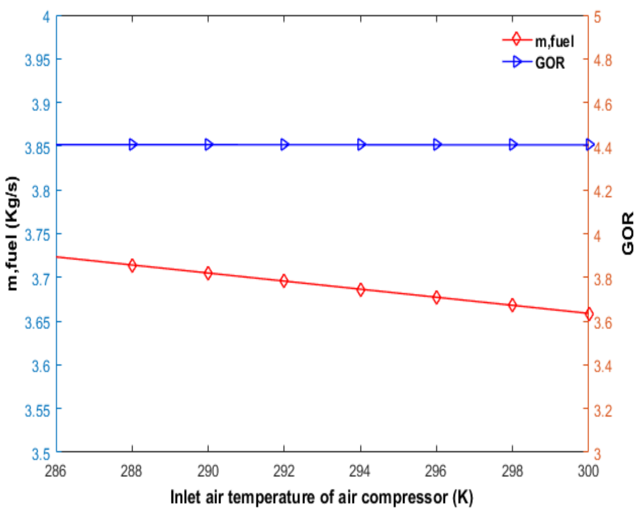
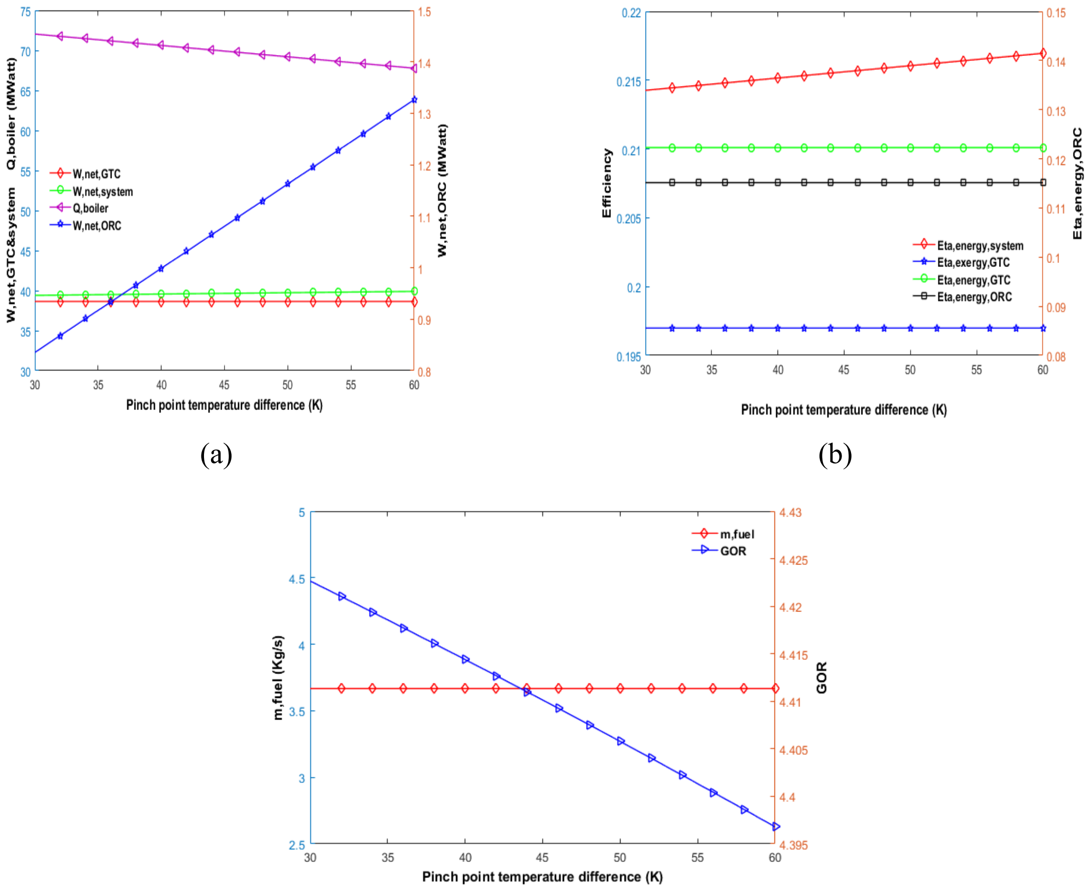


(c)

Fig. 4. Variations of suggested plant performance with changing compressor inlet temperature



(a) (b)

(c)

Fig. 5. Variations of suggested plant performance with changing PP temperature

5.4. Motive steam pressure

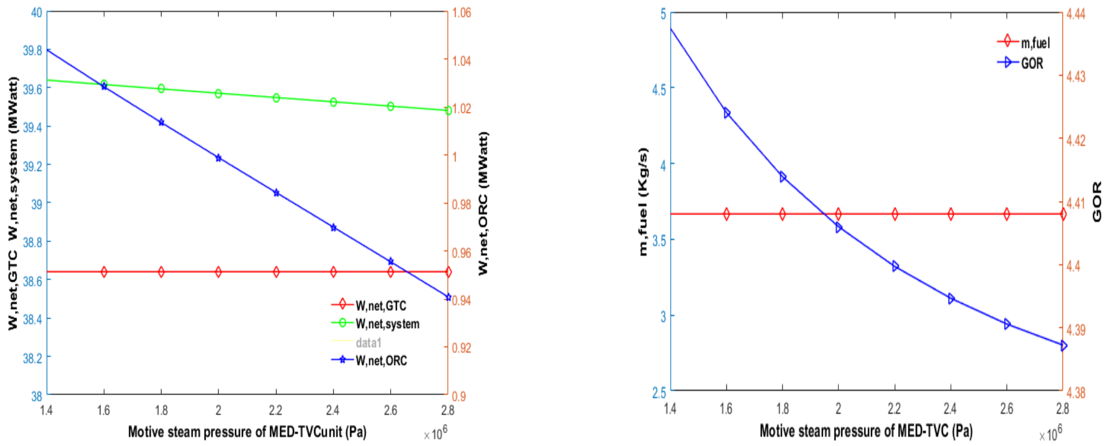
Figure 6. shows the impacts of motive steam pressure on the developed plant performance. With increasing motive steam pressure the evaporation temperature of motive steam enhances while PP temperature remains fixed. As a result, the temperature of the exhaust gas of the economizer enhances. Therefore, the heat transfer rate in the boiler decreases and this in turn, reduces the motive steam flow rate and potable water production as shown in Fig. (6b).

5.5. TVC compression ratio

Figure 7. exhibits the influences of the TVC compression ratio on the plant performance. It is observed that due to the reduction of the mass flow rate of steam feeding the TVC, the mass flow rate of steam employed as the heat source in the first and other effects decreases. Finally, the freshwater generation decreases rapidly with an increasing TVC compression ratio (Fig. 7(b)). According to Fig. 7(a), a very little reduction is seen in the heat transfer rate of the boiler because of increasing the pressure of mixed steam at the ejector outlet and the further increase in condensate temperature of the water.

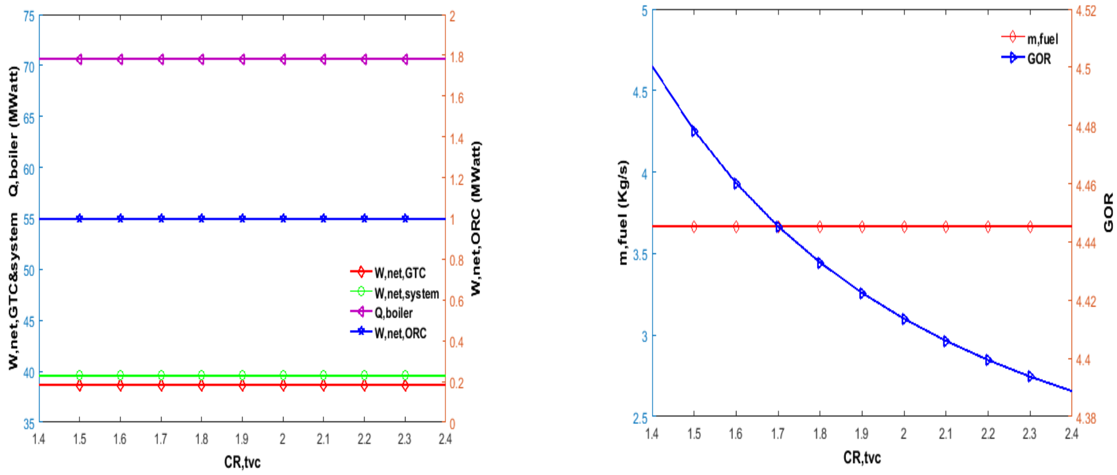
5.6. Effects of high pressure in ORC

Figure 8 presents the impacts of the high pressure of the ORC on the system performance. It is seen that the ORC medium mass flow rate decreases sharply with increasing high pressure in the ORC. Due to the increase in high pressure in the ORC, the increase in evaporation temperature of the ORC medium occurs. Therefore, the ORC turbine can produce more power. On the other hand, the heat transfer rate in evaporator 1 is kept fixed, so with increasing in evaporation temperature of ORC medium, the medium mass flow rate decreases. Therefore, the energy efficiency of ORC increases by about 5.7%.



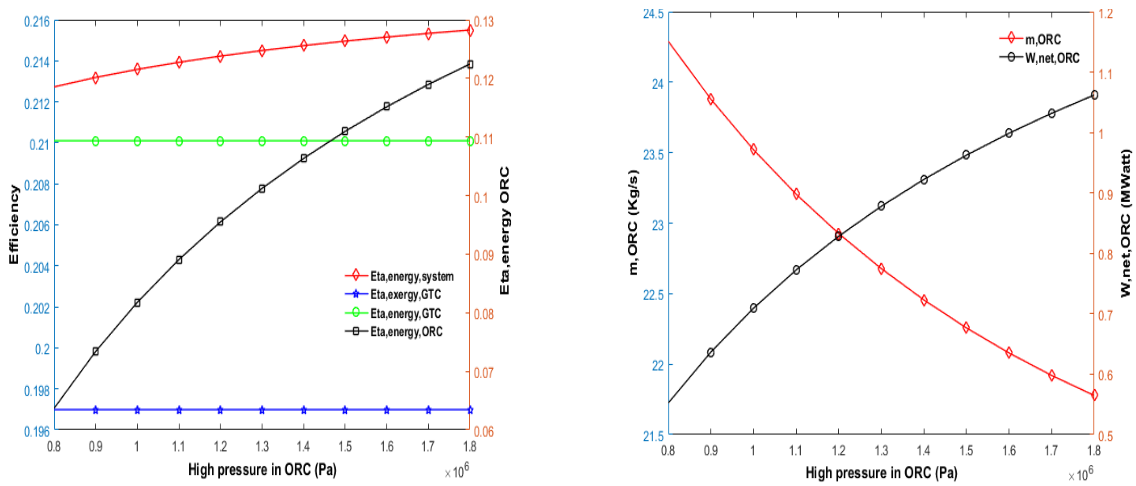
(a) (b)

Fig. 6. Variations of suggested plant performance with changing the motive steam pressure



(a) (b)

Fig. 7. The effects of TVC compression ratio on the developed plant performance



(a) (b)

Fig. 8. The effects of high pressure in ORC on the presented plant performance.

6. Conclusions

The system presented in this study was developed to supply potable water and power . A parametric examination was carried out to identify the influences of important design variables. The results showed that this system could provide 39.6 MW power and 137.3 kg/s freshwater with a gain output ratio of 4.41. According to the results, the highest irreversibility occurs in the combustion chamber and MED-TVC unit which is equal to 69.3% and 19.53% of the total exergy destruction. It was observed that the net power generation of the system and GT increased about 0.1 MW per K precooling the inlet air temperature. The heat transfer rate of the waste heat boiler increased by 0.1 MW per K decrease in the pinch point temperature.

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