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Thermodynamic Analysis of a Combined Brayton and Rankine Cycle based on Wind Turbine

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Abstract

This paper presents the thermodynamic study of a heat and power system which combines an organic Rankine cycle and a gas turbine (GT) cycle. The required power of the compressor in the GT cycle and pump in the Rankine cycle is provided by the WT which according to the amount of cycle required power. For analysis of the cycle, a simulation has been performed using R123 as the working fluid in the Rankine cycle and air and combustion products in the GT cycle. In this end, effect of various parameters such as wind speed, the angular speed of WT, and the gas turbine inlet temperature as well as the compressor pressure ratio, gas turbine isentropic efficiency, condenser temperature and compressor isentropic efficiency on the total thermal efficiency and total exergy efficiency is calculated and analyzed. Thermal efficiency and exergy efficiency of 21.31% and 23.54% is obtained. Also, it is observed that the greatest exergy destruction occurs in the combustion chamber.

Keywords: Wind turbine; Combined cycle; Rankine/Brayton; Exergy; First and second laws

Introduction

Excessive use of fossil fuels has led to more environmental problems such as global worming atmospheric pollution, ozone depletion. Use of renewable energy sources such as wind energy, solar energy rather than fossil fuels can prevent of global warming. Renewable energy is abundant and its technologies are well established to provide complete security of energy supply [1]. Combined cycles are an important tools for energy production which are used in various forms. Combined cycles are flexible, reliable, economical and environmental protection. Several researchers have investigated the performance of power systems using these various heat sources and various power drivers [2-8]. Khaliq et al. [9] analyzed the reheat combined Brayton/Rankine power cycle. They used the second-law approach for the thermodynamic analysis of cycle and investigated the effect of parameters such as pressure ratio, cycle temperature ratio, number of reheats and cycle pressure-drop on the combined cycle performance. Their simulation results showed that the greatest exergy destruction occur in the combustion chamber. Wang et al. [10] proposed the new combined power and ejector–absorption refrigeration cycle that could produce both power output and refrigeration output simultaneously. Khaljani et al. [11] proposed the heat and power combined cycle combines a gas turbine (GT) and an ORC through a single-pressure heat recovery steam generator (HRSG). Their simulation results showed that the increase in pressure ratio and isentropic efficiency of air compressor and gas turbine efficiency improves thermodynamic performance of the system. Also, the most exergy destruction rate takes place in the combustion chamber, and after that in heat recovery steam generator and gas turbine. Maeero et al. [12] analyzed and optimized the second low processes of combined triple power plants. The combined triple power plants includes of the Brayton cycle (gas-based) and two Rankine cycles (steam and ammonia-based). The results of the analysis showed that the most exergy destruction occurs in the heat exchanger. Also, their study showed that the use of feed water heaters increases the efficiency and with increases in ambient temperature, the exergy efficiency decreases. Rabbani et al. [13] analyzed the combined system that coupled a Wind Turbine (WT) with a combined cycle. Required energy of cycle was provided by the wind energy which according to the amount of cycle required energy. Their analysis showed that increasing the combustion temperature reduces the critical velocity and mass flow rate. Increases in wind speed reduce both energy and exergy efficiency of the overall system. Baskut et al. [14] analyzed the exergy processes of wind turbine power plants. Exergy efficiency of the power plant found to be between 0% and 68.20% and the values of exergy efficiencies of the WTPP were different for different power factor value. Zhu et al. [15] analyzed the energy and exergy processes of a bottoming Rankine cycle for engine exhaust heat recovery. The results showed that working fluid properties, evaporating pressure and superheating temperature are the main factors influencing the system design and performances, the distributions of exergy destruction are varied with working fluid categories and system design constraints. Also, the results showed that with the increasing of the evaporating pressure, the internal exergy destruction of the evaporator decreases, and the external exergy destruction increases. Most working fluids do not have the optimal evaporating pressure due to the relatively high exhaust gas temperature. Ozgener et al. [16] analyzed the exergy and reliability of wind turbine systems. The results showed that exergy efficiency changes between 0% and 48.7% at different wind speeds, considering pressure differences between state points. The research done on the combined cycle can be divided into four categories:

- Examining the main drivers of the combined system and the feasibility of using different energy sources, especially low temperature heat sources.
- Study of types of working fluid that can be used in combined system and cooling various technologies in combined system.
- Investigation of exergy, energy laws, and mass conservation in system.

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- Optimization of combined systems.

The present study analyzes a system that couples a Wind Turbine (WT) with a Brayton/Rankine combined power cycle. The aim of the current study is to determine the performance of the combined system through examining the first and second laws of thermodynamics and utilization of wind power to provide cycle required work. The exhaust gases released from Gas Turbine are high temperatures and low pressures, these high temperatures gases drive the Rankine cycle. A working fluid in organic Rankine cycle machine plays a key role. It determines the performance and the economics of the plant [17]. Liu [18] showed that thermal efficiency for various working fluids was a minorant of the critical temperature. Hung [19] showed that wet fluids are unsuitable for ORC systems due to hydrogen bond in wet fluids. Dry and isentropic fluids do not contain liquid droplets at the turbine outlet. This is mainly due to disastrous impact of liquid droplets in turbine exhaust on the performance of turbine in the process of expansion causing the turbine blades wear off and may reduce the turbine efficiency. Rankine cycle employs an organic fluid such as refrigerant, the refrigerant used in the ORC cycle is R123. The working fluid used the Brayton cycle is air that behaves as an ideal gas in throughout the cycle.

Wind Turbine System

Wind turbines convert the kinetic energy of wind in to the mechanical power and the mechanical power transmitted in the power generation system by the shaft. Wind turbines are mounted on tall towers to receive the most possible energy. AWT with a diameter of 100 m is chosen for the present analysis (Figure 1).

The available power can be determined form the amount of air passing through the rotor of wind turbine per unit time. Mass flow rate of the air stream touching the rotor surface ($A=\pi R^2$) was estimated using Eq. (1). Taking $T_0 = 25^\circ C$ and $p_0 = 0.101$ *MPa* as reference temperature and pressure of environment and the density of air is $\rho = 1.18$ kg/m^3, its mass flow rate is

$$m = \rho \times A \times V_r \tag{1}$$

where *R* is the radius of the rotor and V_r is wind speed.

Exergy of kinetic energy is found using the Eq. (2).

$$ke = \frac{V_r^2}{2} \tag{2}$$

Available power is found using the Eq. (3).

$$W_a = m\ .ke \tag{3}$$

$$\frac{p_1}{\rho} + \frac{V_1^2}{2} = \frac{p_2}{\rho} + \frac{V_2^2}{2} \tag{4}$$

Where V_1 is the upstream wind velocity and P_1 is the upstream pressure at the entrance of the rotor blades and V_2 is the downstream wind velocity and P_2 is the downstream pressure at the exit of the rotor blades.

The exit velocity can be determined by using Eq. (5).

$$V_2 = \frac{\text{angular speed of turbine}(\omega) \times R}{V_1} \tag{5}$$

where *R* is the radius of the rotor and ω is measured in radian per second.

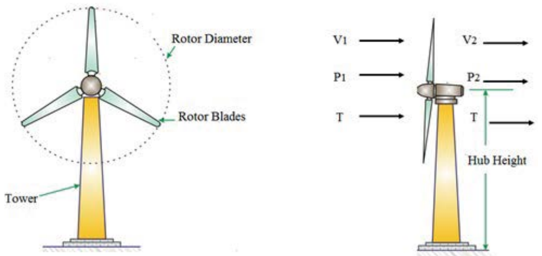

Figure 1: Schematic representation of the wind turbine.

C_p is the fraction of upstream wind power captured by the rotor blades. C_p is often called the Betz limit.

Other names for this quantity are the power coefficient of the rotor or rotor efficiency. The power coefficient is not a static value. It varies with tip speed ratio of the wind turbine. The real world is well below the Betz limit with values of 0.35-0.45 common even in best designed wind turbines [20]. Maximum value of C_p as 0.5926 according to Betz criterion. The power coefficient is given by Eq. (6). In this study, electrical equipment and mechanic equipment losses were assumed to be $\eta_{alternator} = 0.98$ and $\eta_{mechanic} = 0.97$, respectively.

$$C_P = \frac{W_e}{\eta_{alternator} \times \eta_{mechanic} \times .5 \times \rho \times 3.14 \times R^2 \times V_r^3} \tag{6}$$

The useful work is found using the Eq. (7).

$$W_u = (p_1 - p_2)\frac{m}{\rho} \tag{7}$$

The exergetic efficiency of a wind turbine is defined as a measure of how well the stream exergy of the fluid is converted in to useful turbine work output or inverter work output. The exergy efficiency is found by using the Eq. (8).

$$\varepsilon = \frac{W_e}{W_u} = \frac{W_e}{E_1 - E_2} \tag{8}$$

System Description and Assumptions

A schematic of the proposed cogeneration cycle is shown in Figure 2 that includes a cycle of gas turbine (GT) and a cycle of organic Rankine cycle (ORC). WT is used to supply power to the compressor in the GT cycle and pump fluid through a Rankine cycle. The power generation capacity of combined cycle is 26 MW. The Brayton cycle components consist of a combustion chamber, air compressor and GT. The Rankine cycle components consists of a pump, steam turbine, condenser and heat recovery steam generator. Valves of V_1 and V_2 are used in the system that controls the power penetration to the combined power plant. In the case of the high penetration system, WT produces more power than the required power of compressor and pump. So in this case, the wind speed is above the critical speed and the flow valve V_2 is opened and V_1 is closed up to additional power is stored in storage unit. In the case of the low penetration system, WT produces less power than the required power of compressor and pump. So in this case, the wind speed is under the critical speed and the flow valve V_1 is opened and

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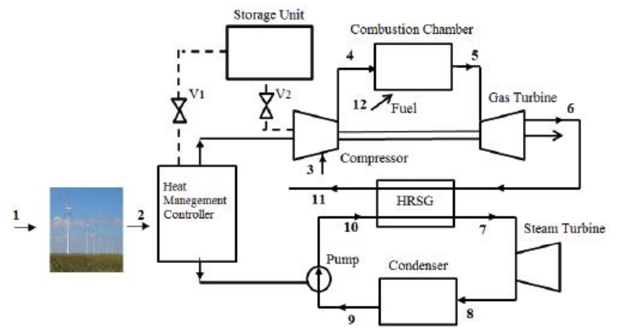

Figure 2: Scheme of combined cycle.

V_2 is closed up to additional power stored in storage unit be injected in the cycle.

According to Figure 2, the ambient air at point 3 with pressure of 0.101 *MPa* and temperature of 298.15 K is compressed in air compressor. Then the compressed air flows is entered into combustion chamber. Fuel is injected in the combustion chamber at a pressure of 1.2 *MPa*. Output stream of combustion chamber with a temperature of 1100°C is expanded in the gas turbine and produces net power of 26MW. The exhaust of the Brayton cycle at high temperature and low pressure is used to drive a Rankine cycle. Working fluid in a saturated liquid phase is pumped to high pressure. After being heated in the internal heat recovery steam generator is entered the turbine to produce power and is expanded to the condenser and is condensed to saturated liquid phase.

Thermodynamic Analysis

The assumptions made for the analysis of systems of expression are:

- The system has reached steady state.

- Pressure drop in the system's components is ignored:

$$\mathrm{p}_4 = p_5;\ \mathrm{p}_6 = p_{11};\ \mathrm{p}_{10} = p_7;\ \mathrm{p}_8 = p_9 \tag{9}$$

- Kinetic and potential energy and frictional losses are neglected.

- System components are Adiabatic.

- Condenser exit state are saturated liquid;

$$x_9 = 1\ T_8 = T_9 \tag{10}$$

- The ambient temperature and pressure are constant (T_0 = 25°C and P_0 = 100 kPa).

- Air is treated as an ideal gas with a molar composition of 21% oxygen and 79% nitrogen.

The principle of mass conservation for the various components of the cycle can be written as follows:

$$\sum m_{in} = \sum m_{out} \tag{11}$$

The calculations are carried out based on the basic assumptions, which are listed in Table 1. Using the law of environmental protection, theory of energy and exergy balance cycles, the balance equations of each component for enthalpy, energy, entropy and exergy are written as follows.

Environment temperature	25°C
Environment pressure	0.1013 MPa
Steam turbine inlet pressure	0.65 MPa
Compressor isentropic efficiency	0.8
Gas turbine isentropic efficiency	0.85
Steam turbine isentropic efficiency	0.82
Pump isentropic efficiency	0.75
Steam turbine inlet temperature	130.15°C
Pump inlet temperature	30.15°C
Compressor inlet temperature	25.15°C
Compressor inlet pressure	0.1013 MPa
Compressor outlet pressure	1 MPa
Net power output	26 MW

Table 1: Basic assumptions for the simulation of combined cycle.

- Compressor

$$W_C = m_3 c_p \left(T_4 - T_3\right) \tag{12}$$

$$\eta_C = \frac{(h_{4.is} - h_3)}{(h_4 - h_3)} \tag{13}$$

- Combustion Chamber

$$-.002\lambda LHV_{ch4} + h_a + \lambda h_f - \left(1+\lambda\right) h_p = 0 \tag{14}$$

Where ƛ is fuel-air ratio and LHV (MJ/Kg) is lower heating value.

$$m_4 h_4 + m_{12}\mathrm{h}_{12} = m_5 h_5 \tag{15}$$

- Gas Turbine

$$W_{GT} = m_5 c_p \left(T_5 - T_6\right) \tag{16}$$

$$\eta_{GT} = \frac{(h_5 - h_6)}{(h_5 - h_{6.is})} \tag{17}$$

- Heat recovery steam generator

$$m_{10} h_{10} + m_6 \mathrm{h}_8 = m_7 h_7 + m_{11} h_{11} \tag{18}$$

- Pump

$$W_\mathrm{P} = \mathrm{m}_{10}\left(\mathrm{h}_{10} - h_9\right) = \frac{m_{10} v_{10}\left(\mathrm{p}_{10} - \mathrm{p}_9\right)}{\eta_p} \tag{19}$$

- Steam Turbine

$$W_\mathrm{ST} = \mathrm{m}_7\left(\mathrm{h}_7 - h_8\right) \tag{20}$$

$$\eta_{ST} = \frac{(h_7 - h_8)}{(h_7 - h_{8.is})} \tag{21}$$

- Condenser

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$$Q_{Cond} = m_8\left(h_8 - h_9\right) \quad (22)$$

- The net power input

$$W_{net,input} = W_C + W_P \quad (23)$$

-The net power output

$$W_{net,output} = W_{GT} + W_{ST} - W_C - W_P \quad (24)$$

- The net power input

$$W_{net,input} = \eta_m \varphi \sum_{I=1}^{n} W_u \quad (25)$$

Where Betz limit φ = 0.4 and n is number of wind turbine and W_u is the output useful work of wind turbine and motor efficiency.

Thermal efficiency of the combined system is defined as the ratio of useful output (specific work output from the cycle and the heat extracted in the condenser) to the input energy (specific work input to the cycle and heat entered to the cycle in the combustion chamber). Performance of the system is shown based on the first law of thermodynamics.

$$e_i = \left(h_i - h_0\right) - t_0\left(s_i - s_0\right) \quad (26)$$

where HHV is higher heating value.

Exergy analysis determines the system performance based on exergy, which is defined as the maximum possible reversible work obtainable in bringing the state of the system to equilibrium with that of the environment, and the evaluation is based on the second law of thermodynamics, because the second law considers not only quantity but also the quality of energy. Taking T_0 and p_0 as reference temperature and pressure of environment, thermal losses in each of the system components were assumed negligible. Exergy at each point of the cycle is calculated as follows by considering the following assumptions:

$$e_i = \left(h_i - h_0\right) - t_0\left(s_i - s_0\right) \quad (27)$$

$$E_i = m_i . e_i \quad (28)$$

By forgoing the kinetic and potential exergies, the total exergy of fuel (The mixture of N_2, H_2O, CO_2, O_2 gases) can be expressed as [21]:

$$ex_{12} = h(T,P) - h_0 - T_0\left(s(T,P) - s_0\right) + \sum x_k e_k^{CH} + RT_0 \sum x_K ln x_k \quad (29)$$

Where e^{CH} is chemical exergy per mole of gas k, x_K is the mole fraction of gas k in the environmental gas phase and R is universal gas constant. Exergy efficiency is defined as the output exergy (net exergy work output from the cycle) to the input exergy (net exergy work input to the cycle and exergy entered in the fuel injection):

$$\eta_{ex} = \frac{w_{net,output}}{W_{net,input} + E_{12}} \quad (30)$$

Exergy destruction in each component of the combined cycle is calculated as follows,

$$I = \sum m_{in} e_{in} - \sum m_{out} e_{out} \pm W \quad (31)$$

The total exergy destruction is equal to the summation of the exergy destruction by each of its components.

State	T (˚C)	P (kPa)	h (kj/kg)	s (kg/kg.k)	m (kg/s)	E (KW)
1	25	101.3	298.4	5.7	-	-
2	25	157.7	298.4	5.7	-	-
3	25	101.3	298.4	5.7	115.3	0
4	371	1000	647.5	7.377	115.3	22399
5	1100	1000	1378	8.065	117.3	84493
6	537	101.3	813.7	8.229	117.3	12554
7	130	650	471.6	1.773	18.34	30971
8	30	260	456.8	1.831	18.34	30402
9	30	109.7	231.4	1.109	18.34	30221
10	30.31	650	231.9	1.109	18.34	30228
11	514	101.3	779.4	7.909	117.3	12047
12	298.15	1200	298.4	5.7	2	85000

Table 2: Results of simulation for the combined cycle.

Pump work (KW)	9.105
Gas turbine work (KW)	65869
Net work (KW)	26000
Steam turbine work (KW)	273.1
Compressor work (KW)	40133
Thermal efficiency (%)	21.31
Exergy efficiency (%)	23.54

Table 3: Performance of the combined cycle.

Results and Discussion

Parametric analysis is carried out to evaluate the effects of various design parameters such as wind speed, angular speed of WT , compressor pressure ratio, compressor isentropic efficiency, gas turbine inlet temperature, gas turbine isentropic efficiency and condenser temperature on the performance of cycle. When one specific parameter is studied, other parameters are kept constant. Table 2 shows the thermodynamic properties such as enthalpy and entropy as well as the mass flow rate and exergy rate at each point of the combined cycle at typical working conditions. The mass flow rate and exergy rate in the wind turbine is variable and depends on the wind speed changes. Table 3 shows the performance of the Brayton/Rankine combined cycle at typical working conditions. Thermal efficiency and exergy efficiency have been obtained respectively 21.3% and 23.5% with existence of wind turbines as the supplier of power of combined system whereas thermal efficiency and exergy efficiency have been obtained respectively 36% and 48% without the wind turbines. Table 4 shows exergy destruction in each component of the combined cycle. The largest exergy destruction occurs in the Combustion Chamber.

Figure 3 shows the effect of wind speed on the WT exergy efficiency and the useful work of WT. According to Eq. (8), exergy efficiency of WT is equal to the ratio of power at inverter output to useful power from WT. With an increase in the wind speed, useful power from WT increases and exergy efficiency of WT decreases. Figure 4 shows the effect of wind speed on the total exergy and thermal efficiencies. With an increase in the wind speed, the mass flow rate of the wind turbine increases. Thus the output wok of the WT and input work to the combined cycle increase. According to Eqs. (26), (30), with the increase of input work to the combined cycle, the total exergy and thermal efficiencies reduce. Figure 5 shows the effect of the angular speed of WT on the total exergy and thermal efficiencies. With an increase in the angular speed of WT, both efficiencies increase.

Figure 6 shows the effect of the compressor pressure ratio on the total exergy and thermal efficiencies and compressor work. With the

Compressor (KW)	5082
Gas turbine (KW)	5684
Steam turbine (KW)	295.4
Condenser (KW)	108.8
Combustion chamber (KW)	22905
Pump (KW)	2.219
HRSG (KW)	21933

Table 4: Exergy destruction in each component of the combined cycle.

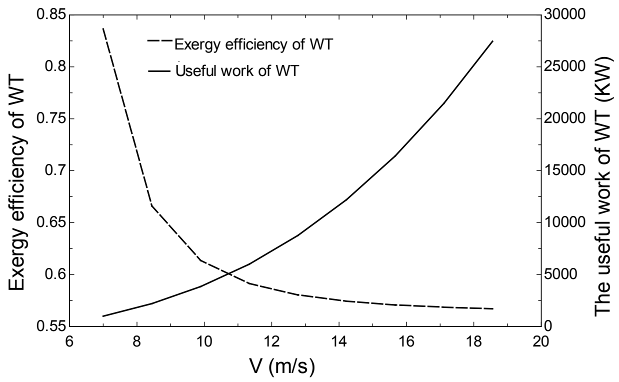

Figure 3: Effect of the wind speed on the exergy efficiency of WT and the useful work of WT.

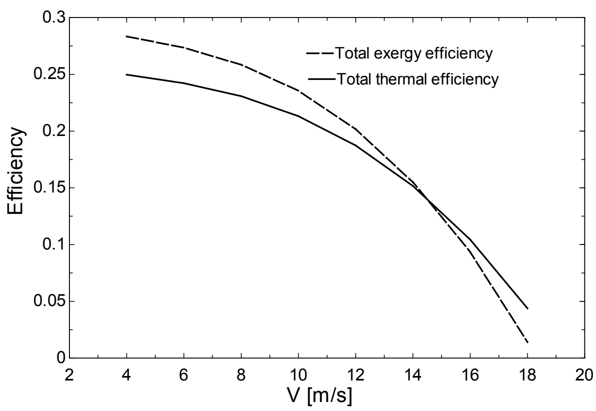

Figure 4: Effect of the wind speed on the total exergy and thermal efficiencies.

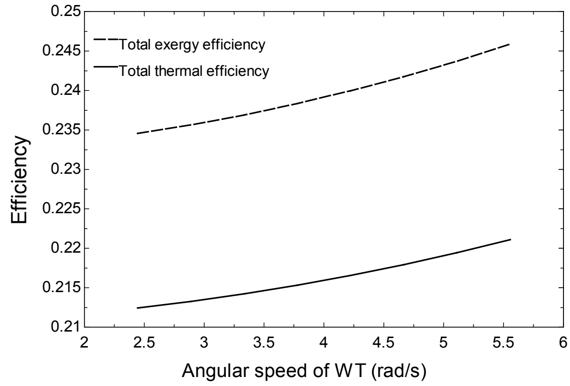

Figure 5: Effect of the angular speed of WT on the total exergy and thermal efficiencies.

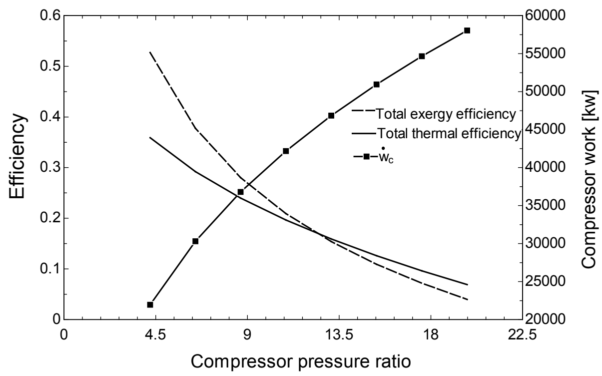

Figure 6: Effect of the compressor pressure ratio on the total efficiencies and compressor work.

increase in compressor pressure ratio, the enthalpy of the outlet of the compressor increases and the air flow rate decreases and the flow of fuel consumption also increases. Net power of gas turbine cycle is constant. Thus, work of air compressor increases and by increasing the fuel consumption, input exergy to the combined cycle increase. Reduction of the air flow rate makes decreasing in the heat transferred to ORC. Thus, flow rate of working fluid and work of ORC reduce. By impact of the above factors, the total exergy and thermal efficiencies decrease. Figure 7 shows the effect of the isentropic air compressor efficiency on the total thermal and exergy efficiencies and compressor work. By increasing the isentropic air compressor efficiency, the enthalpy of the outlet of the compressor decreases and the enthalpy of the input remains constant. Since the net power of gas turbine cycle is constant. Thus, the air flow rate increases and the flow of fuel consumption decreases and input exergy to the combined cycle also decreases. These changes makes the flow rate increasing of working fluid and Rankine cycle work increasing and increasing in the heat transferred to ORC. According to these parameters, the total exergy and thermal efficiencies increase with increasing in the isentropic air compressor efficiency.

Figure 8 shows the effect of the condenser temperature on the total exergy and thermal efficiencies and heat transfer in condenser by increasing condenser temperature, exergy efficiency does not change much but thermal efficiency reduces. With increasing condenser temperature, outlet enthalpy of the condenser increases and input enthalpy of the condenser and flow rate of working fluid of ORC cycle remain constant. Therefore, heat transfer in condenser reduce. According to Eq. (26), thermal efficiency reduce by raising the temperature of the condenser. Figure 9 shows the effect of GT isentropic efficiency on the total exergy and thermal efficiencies. Increasing the isentropic efficiency of gas turbine leads to increase both thermal and exergy efficiencies of the combined cycle. By increasing isentropic efficiency of the gas turbine, the enthalpy of the outlet of the gas turbine decreases and since net power of gas turbine cycle is constant, the air flow rate increases and fuel flow rate reduces and input exergy to the system decreases. The energy balance at the HRSG makes flow rate increasing of working fluid and increasing in the network of ORC. With these changes, the total exergy and thermal efficiencies increase. Figure 10 shows the effect of GT inlet temperature on the total exergy and thermal efficiencies and gas turbine work. Inlet temperature of gas turbine has a significant impact on the total exergy and thermal efficiencies. By increasing inlet

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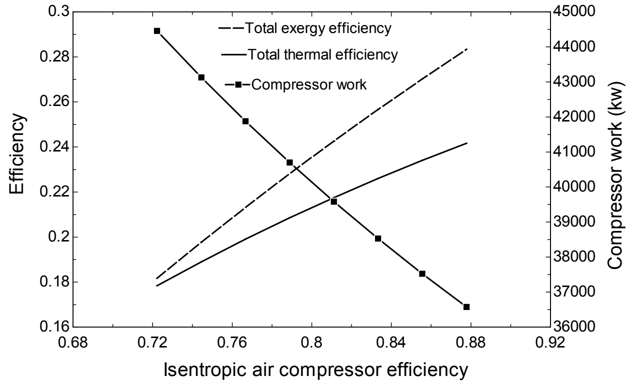

Figure 7: Effect of isentropic air compressor efficiency on the total efficiencies and compressor work.

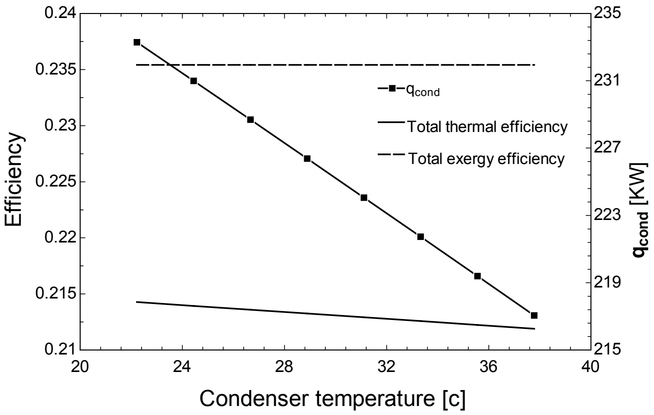

Figure 8: Effect of condenser temperature on the total efficiencies and heat transfer in condenser.

temperature of the gas turbine, the enthalpy of the inlet and outlet points of gas turbine and the air flow rate increase and input fuel to the cycle reduces. Thus, work of gas turbine increases and input exergy to the cycle decreases. With these changes, the total exergy and thermal efficiencies increase.

Conclusion

In this paper, a comprehensive study on a system that couples a Wind Turbine (WT) with a combined heat and power cycle from thermodynamic point of view was investigated with considering two objective functions of first and second law efficiency of the system. The proposed heat and power combined system in this study includes Wind Turbines to supply the power of combined cycle, a gas turbine cycle of 26 MW power and an ORC to produce more power. Adding ORC to the system can produce about 273.1 kW additional power from waste heat recovery of exhaust gases of the GT cycle for the considered base operating conditions. The wind power is used to drive the pump and compressor and if required, additional power is stored by the storage unit that enters to the system in the low wind speed.

- The parametric analysis results of the base case show that the increase in isentropic efficiencies of air compressor and gas turbine and gas turbine inlet temperature improves thermodynamic performance

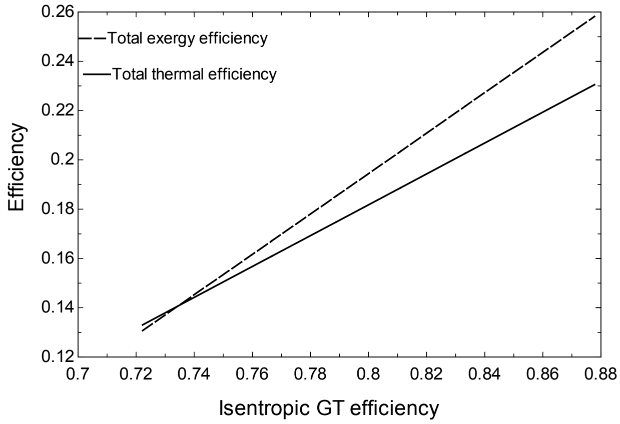

Figure 9: Effect of GT isentropic efficiency on the total exergy and thermal efficiencies.

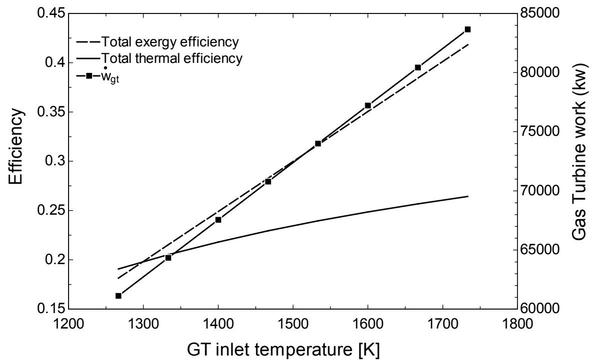

Figure 10: Effect of GT inlet temperature on the total exergy and thermal efficiencies and GT work.

of the system.

- An increasing wind speed and the compressor pressure ratio reduce both energy and exergy efficiencies of the overall system.

- Exergy analysis showed that the highest exergy destruction occurs in the combustion chamber and exergy destruction is significant in the compressor and gas turbine and the pump of the organic Rankine cycle has the least exergy destruction.

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