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# Energy, exergy, and economic analysis of a new triple-cycle power generation configuration and selection of the optimal working fluid

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**Abstract.** The present study investigated energy, exergy and economic analyses on a new triple-cycle power generation configuration. In this configuration, the energy of the exhaust gas and the wasted energy in the condenser of the steam cycle is recovered in the heat recovery steam generator (HRSG) and the evaporator of organic Rankine cycle (ORC), respectively. A computer code was written in MATLAB to analyze the triple-cycle configuration. Validation through this program showed that the highest errors were 5.6 and 7.1%, which occurred in gas and steam cycles, respectively. The results revealed that the highest generated entropy was associated with the combustion chamber and the evaporator in the steam cycle. The first and second laws of thermodynamics efficiencies were improved by roughly 270 and 8%, respectively, through adding each of the steam and organic Rankine cycles. The entropy generated by the cycle increased by roughly 400 and 4% by adding the steam and organic Rankine cycles, respectively. The price of the produced electricity was also reduced by roughly 60 and 70%, respectively, for these two cycles.

**Keywords:** ORC / triple / cycle / energy / exergy

## 1 Introduction

After the industrial revolution, consumption of fossil fuels begin to grow. Considering that, the non-renewable energy sources are limited and have harmful environmental impacts such as global warming, efforts were made by the governments and researchers to employ new approaches in order to decrease the demand for fossil fuels. These efforts were focused on two major fields. In the first, renewable energies were considered as the solution, while the second group attempted to decrease energy consumption of industries and/or use the dissipated energy from these systems [1–2].

In different industries, loss of thermal energy occurs through different means including heat transfer from hot surfaces and/or emission of hot gases or steams from the system. In the cycle used in gas power plants, the exhaust gas from gas turbines carries a considerable amount of energy. Similarly, the exhaust steam from steam turbine releases its energy to the environment as it passes through the condenser. Therefore, recovering these wasted energies reduces environmental pollution in addition to reducing fuel consumption [1–2]. The results of a study conducted in

the United States indicated that the amount of electricity produced through recovery of thermal energy from industries with high-energy wastes was higher than that of renewable energy sources [3]. Recovery of the dissipated energy from gas turbine exhaust gases using heat recovery steam generator (HRSG) has been a matter of interest in the past decades. The first prototype of this system was implemented in 1958 in Texas, US. The aforementioned system, also referred to as dual-cycle system [4]. As a drawback of these systems, they are unable to use energy sources within a temperature range of 80 to 200 °C. However, this problem was resolved after invention of the organic Rankine cycle (ORC) in 1961. The mechanism of ORCs is similar to steam Rankine cycle, with the exception of the working fluid which is replaced with ammonia, R11, R134, etc. [5–6]. Numerous studies have been conducted on energy recovery through ORC. Schulitz (1986) conducted a thermodynamic analysis on ORC. The exhaust gases from internal combustion engines (ICEs) are considered the heat source for ORC. The temperature of the exhaust gases is within a temperature range of 200 to 500 °C. The aforementioned study managed to recover 60% of the energy from the exhaust gases of ICEs [7]. Mago et al. (2007) conducted a thermodynamic investigation on ORCs to assess recovery of wasted energy from different systems. The studied fluids were R134a, propane, R113, R245a,

R123, and isobutane, the boiling point for all of which ranges from 48 to 243 °C. The results of this study suggested that R113 achieved the highest efficiency at a temperature of 430 K [8]. Invernizzi et al. (2007) investigated a combination of gas micro-turbine and ORC. The ORC recovered heat from the micro-turbine exhaust gas within a temperature range of 250 to 300 °C. To this end, a gas 100-kWe micro-turbine combined with ORC was assessed, according to the results of which the output power and the combined cycle efficiency were increased by 1.3 and 40%, respectively, compared to the gas micro-turbine [9]. Ahmadi et al. (2011) did energy and environmental analyses on a combined ORC, micro-turbine, and absorption chiller system. The results showed that the energy efficiency of the considered system was higher than the conventional CHP systems. Additionally, this system also produced a lower amount of carbon dioxide with respect to conventional systems [10]. Tchanche et al. (2011) explored various applications of ORC in power generation using low-temperature heat sources. They assessed various types of low-temperature heat sources such as ocean thermal, biomass, and wasted heat energy from different sources. They showed that use of combined ORC, especially in recovery of heat from different industries, has grown [11]. Wang et al. (2012) investigated 5 different configurations of ORC. They concluded that the ORC with an internal heat exchanger (IHE) achieved the best performance [12]. Bahrami et al. (2013) investigated the effect of 9 working fluids on the first law efficiency in combined Stirling-ORC power cycle. Moreover, they also addressed the environmental problem associated with the different working fluids. They showed that toluene, from a thermodynamic perspective, achieves the highest energy efficiency, while HFE7100 was concluded to be the most appropriate working fluid for the environment [13]. Tanczuk and Ulbrick (2013) investigated a combined ICE, gas boiler, and ORC system in Germany and Netherlands. Their results suggested the system was economical in Germany, but not in Netherlands. The payback periods for Germany and Netherlands were 9.2 and 15 years, respectively [14]. WClemente et al. (2013) conducted the thermodynamic cycle of combined gas cycle and ORC. The gas power plant included a gas turbine with regenerative heat exchanger, producing a 100 kW nominal power. The exhaust gases from the gas turbine were recovered in an ORC. In this analysis, 6 working fluids and 4 turbine models were investigated in an ORC [15]. Cavazzini and Daltoso (2015) conducted an economic-technical analysis on the ORC in recovery of waste heat from different industries. They developed a semi-experimental model for small-sized ORC units for economic assessments. In this model, the different applications of ORC in heat recovery from the exhaust gases and cooling water of engines were studied. According to the results, the aforementioned system was not economic [16]. Carcasci and Winchler (2016) studied the ORC for regeneration of heat from compressor output air. In this analysis, two low- and high-pressure compressors were used, between the compression stages of which air was cooled. The obtained heat was then used as the heat source for ORC. The results indicated that the efficiency of the system was improved by roughly 10% through this

method [17]. Cao and Dai (2017) assessed the power cycle efficiency in combined gas turbine and ORC system under different operating conditions. Their results showed that variable-pressure ORCs deliver a better efficiency compared to constant-pressure ORCs [18]. Also many researches have been developed about energy, exergy, economic and environmental analysis and application in power generation systems both renewable and non-renewable energy resources [19–37].

A review of the literature studies indicates that so far no research has been conducted on heat recovery steam generator (HRSG) and organic rankine cycle. The present study derived energy and exergy models for a triple system combining gas turbine, HRSG, and ORC. In the first stage, the thermal energy of the turbine exhaust gas is recovered and converts water into high-pressure steam in the HRSG. In the next stage, the energy of the low-pressure steam output from the steam turbine is transferred to the working fluid of ORC through a condenser and converts the ORC working fluid to superheated steam. In this proposed cycle, electricity is produced through three equipment: (1) gas turbine in the gas cycle, (2) steam turbine in the steam cycle, and (3) expander in the ORC cycle. Energy and exergy analyses along with the concept of pinch technology were employed to model this system. The contributions and innovations of the present study are as follows:

– presenting a new thermodynamic cycle to increase the recovered energy;
– conceptual comparison between the efficiency of each of the thermal cycles and the triple-cycle power generation system;
– extending the first and second laws efficiencies to triple-cycle power generation system;
– selection of the ORC optimal working fluid in the triple-cycle power generation system.

## 2 Mathematical modeling

The triple system combining gas, steam, and organic Rankine cycles is shown in Figure 1. As shown in Figure 1, air enters the compressor from Point 1 at room conditions and is then compressed (Point 2). The compressed air at Point 2 reacts with the compressed fuel in the booster compressor (Point 12) in the combustion chamber (CC) and burns. The resulting combustion gases enter the gas turbine (Point 3) where they are expanded and generate power (Point 4). The gas then enters the HRSG and converts the water at the pump outlet (Point 9) to superheated steam (Point 12). The HRSG comprises three parts, namely the economizer, the drum, and the superheater. The economizer converts the compressed water at the pump outlet to saturated liquid (Point 10). The drum then converts the saturated liquid into saturated steam (Point 11), which is then converted into superheated steam in the superheater (Point 12). Points 5, 6, and 7 denote the output gas from superheater, drum, and economizer, respectively. The superheated steam expands in the steam turbine and generates power (Point 13). The expanded steam exchanges its heat with the ORC
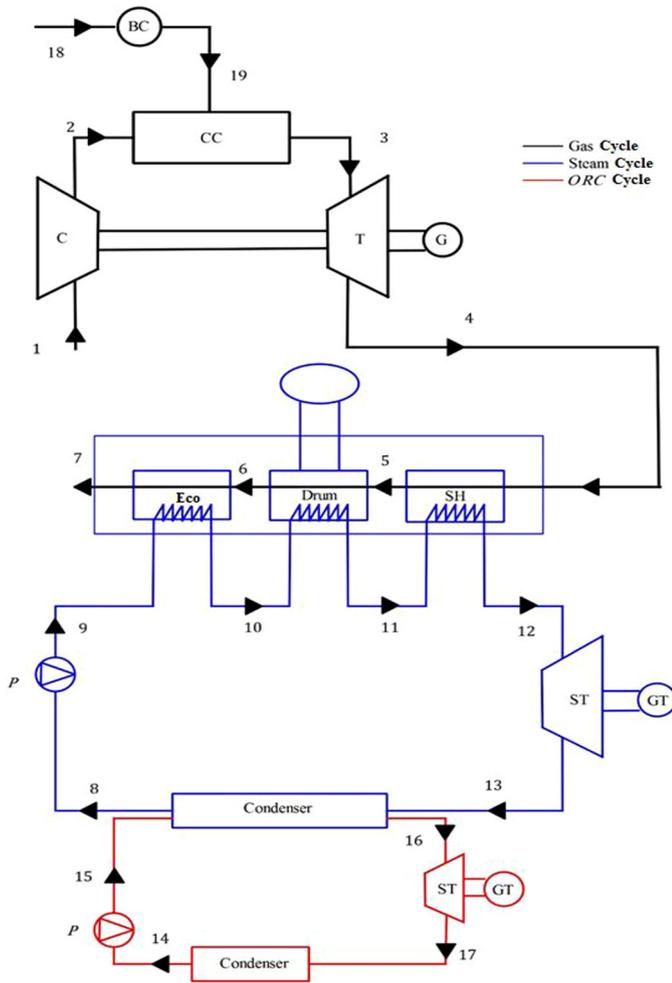
**Fig. 1.** The triple system combining gas, steam and ORC cyles.

evaporator in the steam cycle condenser (Point 8). The coolant (working fluid) in the ORC is then compressed (Point 15) and converted into superheated steam (Point 16) after exchanging heat with the condenser of the steam cycle. This superheated steam is expanded in the expander of the ORC and generates power (Point 17). The expanded steam (Point 17) is converted into liquid after exchanging its heat with the surrounding air in the condenser of ORC (Point 18).

The following assumptions were made to model the cycle shown in Figure 1:
– steady-state conditions were assumed;
– temperature and pressure of the input air were considered 15 °C and 101.3 kPa, respectively;
– air compressor pressure ratio and efficiency were considered 10 and 85%, respectively;
– temperature and pressure of the natural gas entering the booster compressor were considered 10 °C and 250 kPa, respectively;
– the pressure ratio and efficiency of the compressor booster were considered 5 and 80%, respectively;
– molar ratio of the components in the fuel mixture were in accordance to Table 1;

– gas turbine pressure ratio and efficiency were considered 1 and 80%, respectively;
– air-fuel ratio for the gas turbine was considered to be 2.5;
– pressure drop in the steam cycle and ORC were assumed 2 and 3%, respectively.
– steam mass flowrate in the steam cycle was 78.9 kg/sec;
– upper and lower pressure limits in the steam cycle were 40 Bar and 3 Bar, respectively;
– temperature of the pinch point in the steam cycle ($T_{ppst}$) was 40 °C;
– efficiency of the HRSG was considered to be 95%;
– efficiencies of the pump and steam turbine were considered to be 85%;
– coolant mass flowrate in the ORC was 47.5 kg/s;
– upper and lower pressure limits in the ORC were 3 Bar and 0.7 Bar, respectively;
– pinch temperature in the ORC ($T_{pp,ORCT}$) was 20 °C;
– pump and turbine efficiencies in the ORC were 85%;

The output temperature and pressure and the work consumption based on the compressor mass flow rate can be calculated from equations (1) and (2) [38,39]:

$$T_2 = T_1(1 + (r_c^{(k-1)/k} - 1)/\eta_c, \tag{1}$$

$$P_2 = r_c P_1, \tag{2}$$

$$w_{c,g} = (kR/k - 1)T_1\left[\left(r_c^{(k-1)/k} - 1\right)/\eta_c\right], \tag{3}$$

where $T_1$ is the input temperature at the compressor (K), $T_2$ is the output temperature at the compressor output (K), $r_c$ is the compressor pressure ratio, k is the ratio of constant-pressure specific heat to the constant volume specific heat, $\eta_{c,g}$ is the compressor efficiency, $P_1$ is the compressor input pressure (kPa), $P_2$ is the compressor output pressure (kPa), R is the gas constant (kJ/kg.K), and $w_{c,g}$ is the unit work done by the compressor mass flowrate (kJ/kg). Note that the fuel is compressed in booster compressor. Calculation of temperature, output pressure, and work based on the unit mass flowrate is similar to equations 1 to 3. The results of gas analysis, as presented in Table 1, were used to assess and design the combustion chamber [38,39].

Molecular mass and gas constant are calculated as follows:

$$M = \sum_{i=1}^{n} y_i M_i, \tag{4}$$

$$R = \frac{R_u}{M}, \tag{5}$$

where M is the molecular mass of the mixture (kg/kmole), $M_i$ is the molecular mass of each component (kg/kmole), $y_i$ is the molar percentage of each component, $R_u$ is the universal gas constant (kJ/kmole.K), and R is the gas constant (kJ/kg.K).

**Table 1.** Analysis of the natural gas consuled in combusion chamber [40].

| i component | $CH_4$ | $C_2H_6$ | $C_4H_{10}$ | $Co_2$ | $N_2$ |
|---|---|---|---|---|---|
| yi, % Mole fraction | 81 | 7.9 | 4.2 | 1.2 | 1 |
| Xi, % Mass fraction | 62.58 | 11.44 | 8.92 | 2.55 | 1.35 |
| Mi, kg/kmole | 16 | 30 | 44 | 44 | 28 |
| Cpi, kJ/kg K | 2.25 | 1.76 | 1.67 | 0.84 | 1.04 |

$M_f = \Sigma y_i M_i = 20.712 \,(kg/kmoles)$
$(c_p)_f = \Sigma x_i c_{pi} = 2.013 \,(kJ/kg \quad K)$
$(C_v)_f = (C_p) - R_f = 1.612 (kJ/kg \quad K)$
$k_f = (C_p)_f/(C_v)_f = 1.249$

The constant-pressure specific heat can be calculated based on the gas specific constant and the molar percentage as follows:
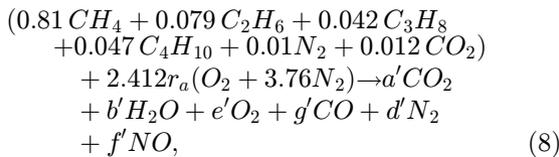
$$C = \sum_{i=1}^{n} y_i C_p, \qquad (6)$$

where $C_p$ is the constant-pressure specific heat (kJ/kg.K), and $C_{pi}$ is the constant-pressure specific heat of the mixture components (kJ/kg.K).

The coefficients a, b, c, d, e, f, and g are related to a', b', c', d', e', f', and g' as follows:

$$\begin{aligned} a &= a'/n', \ b = b'/n', \ d = d'/n', \ e = e'/n', \ f \\ &= f'/n', \ g = g'/n'....n' \\ &= a' + b' + d' + e' + f' + g'. \end{aligned} \qquad (7)$$

The following equation can be considered to calculate the temperature and combustion products. The combustion process is assumed to be balanced:

$$\begin{aligned} (0.81 \, CH_4 &+ 0.079 \, C_2H_6 + 0.042 \, C_3H_8 \\ &+ 0.047 \, C_4H_{10} + 0.01 N_2 + 0.012 \, CO_2) \\ &+ 2.412 r_a (O_2 + 3.76 N_2) \rightarrow a' CO_2 \\ &+ b' H_2O + e' O_2 + g' CO + d' N_2 \\ &+ f' NO, \end{aligned} \qquad (8)$$

where $r_a$ is the air-fuel molar ratio, and a', b', c', d', e', f', and g' are the coefficients of (Eq. (7)).

Consider the following balance equations:

$$2CO + O_2 \quad \overset{\rightarrow}{\underset{\leftarrow}{}} \quad 2CO2......K_{CO} = \frac{g^2 e}{a^2} \left(\frac{P_3}{P_2}\right)^{2+1-2}, \quad (9)$$

$$N_2 + O_2 \quad \overset{\rightarrow}{\underset{\leftarrow}{}} \quad 2NO....K_{NO} = \frac{f^2}{de} \left(\frac{P_3}{P_2}\right)^{2-1-1}, \qquad (10)$$

where $K_{co}$ is the equilibrium constant for (Eq. (9)), $K_{NO}$ is the equilibrium constant for (Eq. (10)), $P_2$ is the pressure before entering the combustion chamber (kPa), $P_3$ is the combustion pressure (kPa), and a, b, c, d, e, f, and g are the coefficients of equations (9) and (10).

The combustion process in the combustion system is expressed in the following form:

$$\sum \dot{m}_p(h_f^\circ + (h - h_o))_p = \eta_{cc} \sum \dot{m}_r(h_f^\circ + (h - h_\circ))_r, \qquad (11)$$

where $\dot{m}_p$ is the mas flowrate of the products (kg/sec), $\dot{m}_r$ is the mass flowrate of the reactants (kg/sec), $\eta_{cc}$ is the combustion efficiency, $h$ is enthalpy (kJ/kg), $h_o$ is the enthalpy at the reference temperature (kJ/kg), $h_f^\circ$ is the formation enthalpy (kJ/kg).

The combustion temperature and the produced products can be calculated by solving equations (7) to (11). The pressure of the combustion chamber is calculated from the following relation:

$$P_3 = P_2 \frac{n_3 T_3}{n_2 T_2}, \qquad (12)$$

where $n_3$ and $T_3$ are the number of moles and temperature (K) before entering the combustion chamber, respectively.

The temperature and pressure at the outlet of gas turbine can be calculated by the following (Eq. (38–41)):

$$T_4 = T_3(1 - \eta_t(1 - \frac{1}{r_{t,g}^{\frac{k-1}{k}}})), \qquad (13)$$

$$P_4 = \frac{P_3}{r_{t,g}}. \qquad (14)$$

In which $T_4$ is the output temperature of the gas turbine (K), $\eta_{t,g}$ is its efficiency, $r_{t,g}$ is gas turbine pressure ratio, and $P_4$ is its output pressure (kPa).

The output work of the turbine based on mass unit [38–41]:

$$w_{t,g} = \frac{kRT_3}{k - 1} \left[1 - (\frac{P_4}{P_3})^{\frac{k-1}{k}}\right] \eta_{t,g}. \qquad (15)$$

The network produced of the turbine is expressed as follows [38–41]:

$$\dot{W}_{net,g} = (\dot{m}_a + \dot{m}_f) \, w_{t,g} - m_a w_{c,g} - m_f w_{bc,g}, \qquad (16)$$

where $\dot{W}_{net,g}$ is the net work (kW), $W_{bcg}$ is the booster compressor work consumed by the unit fuel mass flowrate (kJ/kg), $\dot{m}_a$ is the air mass flowrate (kg/sec), and $\dot{m}_f$ is the fuel mass flowrate (kg/sec).

In a simple case, the efficiency of the first law of thermodynamic can be obtained from the following relation [38–41]:

$$\eta_{I,g} = \frac{\dot{W}_{net,g}}{\dot{m}_f LHV}, \qquad (17)$$

in which $\eta_{I,g}$ is the efficiency of the first law of thermodynamics and LHV is the fuel lower heating value (kJ/kg).

The mass flowrate of the gas can be calculated by the equation below [38–41]:

$$\dot{m}_g = \dot{m}_a + \dot{m}_f. \qquad (18)$$

The equations of energy conservation for the superheater, drum, and economizer are written as follows [38–41]:

$$\dot{m}_{st}(h_{12} - h_{11}) = \eta_{HRSG}\ m_g\, c_p(T_4 - T_5), \qquad (19)$$

$$\dot{m}_{st}(h_{11} - h_{10}) = \eta_{HRSG}\ m_g\, c_p(T_5 - T_6), \qquad (20)$$

$$\dot{m}_{st}(h_{10} - h_9) = \eta_{HRSG}\quad m_g\, c_{pg}(T_6 - T_7), \qquad (21)$$

where $h_9$, $h_{10}$, $h_{11}$, and $h_{12}$ are the enthalpies of the points marked in Figure 1 (kJ/kg), $T_4$, $T_5$, $T_6$, and $T_7$ are the temperatures of the corresponding points in Figure 1 (°C), $\dot{m}_{st}$ is the steam mass flowrate (kg/s), $\dot{m}_9$ is the combustion gas mass flow rate (kg/s), $C_{pg}$ is the constant-pressure specific heat (kJ/kg°C), and $\eta_{HRSG}$ is the efficiency of HRSG.

The input power for the pump in the steam cycle [38–41]:

$$\dot{W}_P = \dot{m}_{st}(h_9 - h_8)/\eta_P, \qquad (22)$$

in which $h_8$ and $h_9$ denote, respectively, the enthalpies of Points 8 and 9 in Figure 1 (kJ/kg), $\dot{W}_P$ is the input power consumed by the pump (kW), and $\eta_P$ is the pump efficiency in the steam cycle.

The output power of the steam turbine [38–41]:

$$\dot{W}_{st} = \dot{m}_{st}\eta_{st}(h_9 - h_8), \qquad (23)$$

where $h_{12}$ and $h_{11}$ correspond to the enthalpies of Points 12 and 11 in Figure 1 (kJ/kg), respectively, $\dot{W}_{st}$ is the output power of the steam turbine (kW), and $\eta_{st}$ is the efficiency of the steam turbine in the steam cycle.

The energy exchanged between the steam cycle condenser and the ORC evaporator can be obtained as follows [38–41]:

$$\dot{m}_{ORC}(h_{16} - h_{15}) = \eta_{Co}\dot{m}_{st}(h_{13} - h_8), \qquad (24)$$

in which $h_{16}$ and $h_{15}$ correspond to the enthalpies of Points 15 and 16 (kJ/kg) in the steam cycle, respectively, $h_8$ and $h_{13}$ are the enthalpy values at Points 8 and 3 (kJ/kg) in the ORC cycle, respectively, $\dot{m}_{ORC}$ denotes the mass flowrate of the working fluid in the ORC cycle (kg/sec), and $\eta_{Co}$ is the efficiency of the condenser.

The power produced by the ORC expander turbine in the ORC is calculated as follows [38–41]:

$$\dot{W}_{st,ORC} = \dot{m}_{ORC}(h_{16} - h_{17}), \qquad (25)$$

where $h_{16}$ and $h_{17}$ are, respectively, the enthalpy of the working fluid at Points 16 and 17 (kJ/kg), and $\dot{W}_{st,ORC}$ represents the power generated by the ORC steam turbine (kW).

The condenser heat transfer rate of the ORC is determined by the following relation [38–41]:

$$\dot{Q}_{co,ORC} = \dot{m}_{ORC}(h_{17} - h_{14}), \qquad (26)$$

in which $h_{17}$ and $h_{14}$ are, respectively, the enthalpy of the working fluid at Points 17 and 14 (kJ/kg), and $\dot{Q}_{co,ORC}$ indicates the heat transfer rate in the ORC condenser (kW).

The power required by the pump in the ORC can be obtained through the following relation [38–41]:

$$\frac{\dot{W}_{P,ORC} = \dot{m}_{ORC}(h_{15} - h_{14})}{\eta_{P,ORC},} \qquad (27)$$

in which $h_{15}$ and $h_{14}$ are, respectively, the enthalpy of the coolant fluid in the ORC at Points 15 and 14 (kJ/kg), $\eta_{P,ORC}$ is the efficiency of the pump in the ORC cycle, and $\dot{W}_{P,ORC}$ is the power required by the pump (kW).

The net power produced by the steam cycle and the ORC can be computed from the following relations [38–41]:

$$\dot{W}_{net,st} = \dot{W}_{st} - \dot{W}_p, \qquad (28)$$

$$\dot{W}_{net,ORC} = \dot{W}_{st,ORC} - \dot{W}_{p,ORC}. \qquad (29)$$

The efficiency of the first law of thermodynamics for the steam cycle and the ORC is determined as:

$$\frac{\eta_{I,steam} = \dot{W}_{net,st}}{m_g c_{pg}(T_4 - T_7),} \qquad (30)$$

$$\frac{\eta_{I,ORC} = \dot{W}_{net,ORC}}{\dot{m}_{st}(h_{13} - h_8).} \qquad (31)$$

In general, the efficiency of the entire cycle is defined by the following equation:

$$\frac{\eta_I = \dot{W}_{net,st} + \dot{W}_{net,ORC} + \dot{W}_{net,g}}{\dot{m}_f LHV.} \qquad (32)$$

The exergy of Points 1 to 7 and Points 18 and 19 (gas turbine cycle) can be computed from the following relation [41]:

$$ex = e_{ph} + e_{ch}, \qquad (33)$$

where $e_{ph}$ and $e_{ch}$ are, respectively, physical and chemical exergies (kJ/kg).

Assuming the relations for ideal gases hold true, the physical and chemical exergies in the gas turbine can be calculated [41]:

$$e_{ph} = c_p T_0 \left[ \left(\frac{T}{T_0}\right) - 1 - ln\left(\frac{T}{T_0}\right) \right] + RT_0 ln\frac{P}{P_0} , \qquad (34)$$

$$e_{ch} = \sum_{i=1}^{n} x_i e_{ch,i} + RT_0 \sum_{i=1}^{n} x_i \, ln(x_i) , \qquad (35)$$

in which $e_{ph}$ and $e_{ch}$ denote the physical and chemical exergies (kJ/kg) in the gas cycle, respectively, $e_{ch,i}$ is the each component chemical exergy (kJ/kg), and $x_i$ denotes the mass fraction. $P_0$ and $T_0$ represent the reference temperature and pressure, respectively which are equal to 1 atm and 298 (K). In order to calculate the exergy of Points 1 to 7 and Points 18 and 19, we can simply substitute the thermodynamic properties (temperature, pressure, mass fraction, and mixture combination) in equations (33) to (35) and calculate the exergy at these points.

The exergy of the steam cycle and ORC can be obtained as follows [38–41]:

$$ex = (h - h_o) - T_0(s - s_o). \qquad (36)$$

The entropy produced in the gas cycle and that produced in the compressor, booster compressor, combustion chamber, and gas turbine can be obtained from the following equations:

$$\dot{S}_{gen,c,g} = \frac{1}{T_0} \left[ \dot{m}_1 ex_1 - \dot{m}_2 ex_2 - \dot{W}_{c,g} \right], \qquad (37)$$

$$\dot{S}_{gen,bc,g} = \frac{1}{T_0} \left[ \dot{m}_{18} ex_{18} - \dot{m}_{19} ex_{19} - \dot{W}_{bc,g} \right], \qquad (38)$$

$$\dot{S}_{gen,cc,g} = \frac{1}{T_0} \left[ \dot{m}_2 ex_2 + \dot{m}_{19} ex_{19} - \dot{m}_3 ex_3 \right], \qquad (39)$$

$$\dot{S}_{gen,t,g} = \frac{1}{T_0} \left[ \dot{m}_3 ex_3 - \dot{m}_4 ex_4 - \dot{W}_{t,g} \right]. \qquad (40)$$

In equations (37) to (40), $\dot{S}_{gen}$ represents the generated entropy (kW/K). The indices cc,g, bc,g, c,g, and t,g denote the compressor, booster compressor, combustion chamber and gas turbine, respectively. The entropies generated in the pump, economizer, drum, superheater, steam turbine, and condenser are obtained through the following relation:

$$\dot{S}_{gen,P} = \frac{1}{T_0} \left[ \dot{m}_8 ex_8 - \dot{m}_9 ex_9 - \dot{W}_P \right], \qquad (41)$$

$$\dot{S}_{gen,eco} = \frac{1}{T_0} \left[ \dot{m}_9 ex_9 - \dot{m}_{10} ex_{10} + \dot{Q}_{eco}(1 - \frac{T_0}{T_{eco}}) \right], \quad (42)$$

$$\dot{S}_{gen,drum} = \frac{1}{T_0} \left[ \dot{m}_{10} ex_{10} - \dot{m}_{11} ex_{11} + \dot{Q}_{drum}(1 - \frac{T_0}{T_{drum}}) \right], \qquad (43)$$

$$\dot{S}_{gen,SH} = \frac{1}{T_0} \left[ \dot{m}_{11} ex_{11} - \dot{m}_{12} ex_{12} + \dot{Q}_{SH}(1 - \frac{T_0}{T_{SH}}) \right], \qquad (44)$$

$$\dot{S}_{gen,st} = \frac{1}{T_0} \left[ \dot{m}_{12} ex_{12} - \dot{m}_{13} ex_{13} - \dot{W}_{st} \right], \qquad (45)$$

$$\dot{S}_{gen,co} = \frac{1}{T_0} \left[ \dot{m}_{13} ex_{13} - \dot{m}_8 ex_8 + \dot{Q}_{co}\left(1 - \frac{T_0}{T_{co}}\right) \right], \quad (46)$$

where $ex_8$ and $ex_{13}$ are the exergy (kJ/kg) of the specified points in Figure 1, and $T_{eco}$, $T_{drum}$, $T_{SH}$, and $T_{co}$ are, respectively, the temperatures of the economizer, drum, superheater, and condenser in the steam cycle (K). Moreover, $\dot{Q}_{eco}, \dot{Q}_{drum}, \dot{Q}_{SH},$ and $Q_{co}$ are the heat transfer rate (kW) in the economizer, drum, superheater, and condenser, respectively.

The values of $\dot{Q}_{eco}, \dot{Q}_{drum}, \dot{Q}_{SH},$ and $Q_{co}$ can be obtained from the following relations [38–41]:

$$\dot{Q}_{eco} = \dot{m}_{st}(h_{10} - h_9), \qquad (47)$$

$$\dot{Q}_{drum} = \dot{m}_{st}(h_{11} - h_{10}), \qquad (48)$$

$$\dot{Q}_{SH} = \dot{m}_{st}(h_{12} - h_{11}), \qquad (49)$$

$$\dot{Q}_{co} = \dot{m}_{st}(h_{13} - h_8). \qquad (50)$$

The entropy relations in the ORC for the pump, steam turbine, and condenser are similar to equations (41), (45) and (46). The efficiency of the second law of thermodynamics for the combined gas and steam cycles and each of the gas, steam, and organic Rankine cycles individually can be calculated as follows:

$$\eta_{II,g-st} = \frac{\dot{W}_{net,g} + \dot{W}_{net,st}}{\dot{m}_{18}} ex_{18} , \qquad (51)$$

$$\eta_{II,g-st-ORC} = \frac{\dot{W}_{net,g} + \dot{W}_{net,st} + \dot{W}_{net,ORC}}{\dot{m}_{18} ex_{18}} . \qquad (52)$$

In order to calculate the cost of the electricity produced by this system, the objective function is defined as follows:

$$C_E = C_1 + C_O + C_F, \qquad (53)$$

where $C_E$ is the cost of the produced electricity ($/kWh), $C_1$ is the cost of installation ($/kWh), $C_o$ is the maintenance cost ($/kWh), and $C_F$ is the fuel cost ($/kWh).

**Table 2.** Thermodynamics specifications of the ORC working fluids [48].

| No | Fluid | $T_c$ (°C) | $P_c$ (kPa) | $M\left(\frac{k_g}{kmol}\right)$ |
|---|---|---|---|---|
| 1 | R11 | 198 | 440.8 | 137.4 |
| 2 | R123 | 183.7 | 360 | 152.9 |
| 3 | C5F12 | 147.4 | 204.5 | 288 |
| 4 | Butane | 152 | 380 | 58.1 |

As the advantage of this method, the costs are calculated per kWh of produced power, such that the overall cost of the produced power can be calculated by changing the parameters and conditions.

The electricity cost for the initial setup can be calculated from the following relation:

$$C_I = \frac{CI}{8760\dot{W}_{net}}, \tag{54}$$

where $C$ is the installation cost and $I$ is the interest rate.

The interest of the initial cost [42]:

$$I = \frac{i(1+i)^L}{(1+i)^L - 1}, \tag{55}$$

where L and I are the equipment lifetime (year) and interest rate which can be assumed 20 years and 4%, respectively. The maintenance cost is assumed to be 4% of the installation cost.

The fuel cost can be obtained as follows [42]:

$$C_F = \frac{Fuel\cos t\left(\frac{\$}{kWh}\right)}{\eta_I}. \tag{56}$$

The fuel cost was considered 0.0296 (US$/kWh) [42].

The following relations can be used to calculate the cost of equipment including the compressor, combustion chamber, and gas turbine [43]:

$$C_{c,g} = \frac{c_{11}\dot{m}_a}{c_{12} - \eta_c}\left(\frac{P_2}{P_1}\right)\ln\left(\frac{P_2}{P_1}\right), \tag{57}$$

$$C_{cc,g} = \frac{c_{21}\dot{m}_a}{c_{22} - \frac{P_2}{P_3}}\left(1 + \exp(c_{23}T_3 - c_{24})\right), \tag{58}$$

$$C_{t,g} = \left(\frac{c_{31}\dot{m}_g}{c_{32} - \eta_{t,g}}\right)\ln\left(\frac{P_3}{P_4}\right)\left(1 + \exp(c_{33}T_3 - c_{34})\right). \tag{59}$$

The coefficients $c_{11}$, $c_{12}$, $c_{22}$, $c_{21}$, $c_{23}$, $c_{24}$, $c_{31}$, $c_{32}$, $c_{34}$ are given in [43].

The following equations can be used to compute the costs of pump, HRSG, steam turbine, and condenser in the steam cycle [44]:

$$C_{p,st} = 3540(\dot{W}_p)^{0.71}, \tag{60}$$

$$C_{HRSG,st} = 4745\frac{\dot{m}_{st}(h_{12} - h_9)}{log(T_4 - T_7)}^{0.8} + 11820\dot{m}_{st} + 658\dot{m}_g, \tag{61}$$

$$C_{St,s} = 6000(\dot{W}_{net,st})^{0.7}, \tag{62}$$

$$C_{co,st} = 1773\dot{m}_{st}. \tag{63}$$

The costs of the main equipment are calculated through the following relations as provided by [45–47]. The cost of turbine, pump, condenser, evaporator, and generator in the ORC can be calculated from the following relations:

$$C_{ST,ORC} = 2237(\dot{W}_{ST,ORC})^{0.41}, \tag{64}$$

$$C_{P,ORC} = 16800\left(\frac{\dot{W}_{P,ORC}}{200}\right)^{0.67}, \tag{65}$$

$$C_{Cond,ORC} = 43(\dot{Q}_{Cond,ORC})^{0.68}, \tag{66}$$

$$C_{HRVG,ORC} = 11.6779 \times \dot{Q}_{HRVG,ORC} + 4416.105, \tag{67}$$

$$C_{Gen,ORC} = 2447(\dot{W}_{Gen,ORC})^{0.49}. \tag{68}$$

## 3 Results and discussion

An M-File script was written in MATLAB so as to model the triple cycle (GT + HRSG + ORC). The Xstream and Refprob software along with M-Files were used for the thermodynamic properties of the steam and working fluids of the cooling cycle [48]. Considering that the proposed cycle has not been presented in any other studies before, to validate the model, the cycle of the combined gas turbine and single-pressure HRSG was investigated. The specifications of the proposed combined cycle given in [49] were used in the programmed code, the results of which were then compared to those of the same reference. The evaluation results revealed that the highest errors in the gas and steam cycles were roughly 5.6% and 7.1%, respectively. The specifications of the coolant fluids employed in this study are shown in Table 2 [48].

The entropy of each of the equipment in the cycle with R11 as the coolant is shown in Figure 2. According to the generated entropies in the table, it can be concluded that the highest entropy production rate was associated with the evaporator of the steam cycle. The evaporator of HRSG is composed of an economizer, a drum, and a superheater. The second highest amount of generated entropy is associated with the combustion chamber in the gas cycle. The reasons for these phenomena can be attributed to the conversion of the compressed water in the evaporator to superheated steam, due to which a large amount of entropy is generated. The high amount of generated entropy in the
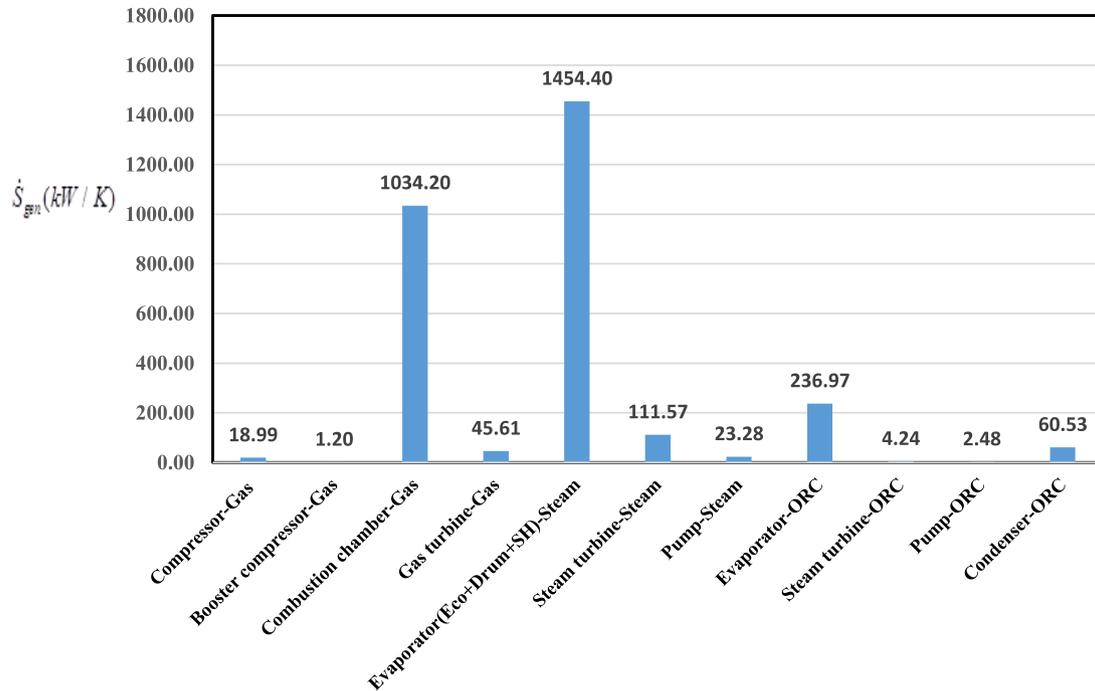
**Fig. 2.** Amount of entropy generated in the equipment of the tripe cycle with R11 as the coolany fluid.
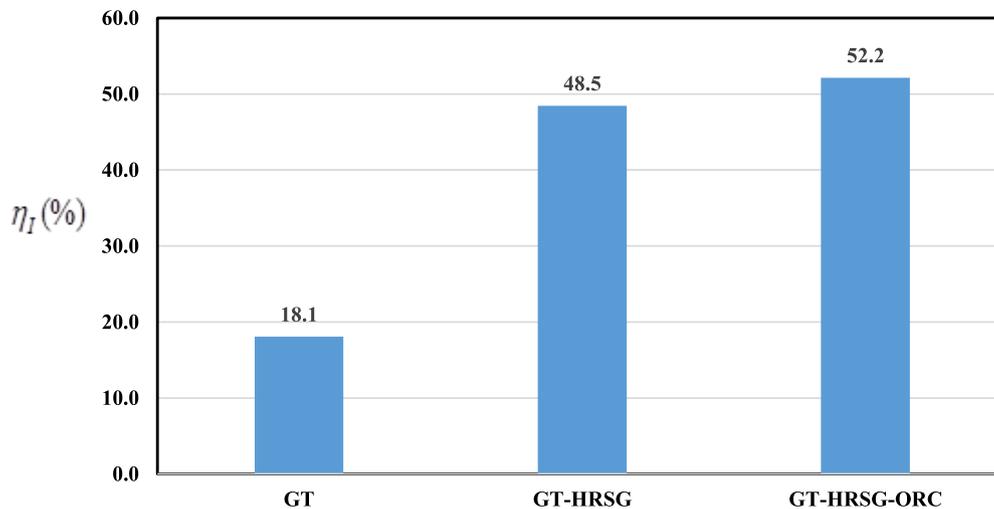


**Fig. 3.** Efficiency of the first law of the thermodynamics for coolant R11 in the three cases, namey GT, GT+HRSG, and GT+HRSG+ORC.

combustion chamber is caused by the chemical reactions occurring in this chamber. The lowest amount of generated entropy is associated with the compressor booster, the pump, and the steam turbine in the ORC. Generation of a relatively low amount of entropy in the gas cycle is reasonable given that the fuel flowrate and the increase in pressure is lower in the booster compressor compared to the air compressor. This also applies to the ORC in which the mass flowrate of the working fluid is low and the specific volume of the coolant is small.

The efficiency of the first law of thermodynamics for coolant R11 in three cases, namely the GT cycle, the GT + HRSG cycle, and the GT + HRSG + ORC cycle is demonstrated in Figure 3. The efficiency was increased by 2.7 times through addition of the HRSG cycle to the GT so as to recover heat from the turbine exhaust gases. However, addition of the ORC to recover from the HRSG cycle increased the efficiency roughly by 8%. The physical reason for this phenomenon can be explained through the higher temperature of the turbine exhaust gases compared to the water temperature at the condenser inlet, as a results of which a lower amount of energy is recovered in the ORC. The efficiency of the second law of thermodynamics for Coolant R11 in three cases is shown in Figure 4 similar to Figure 3. The trend of the efficiency diagram is similar in both.
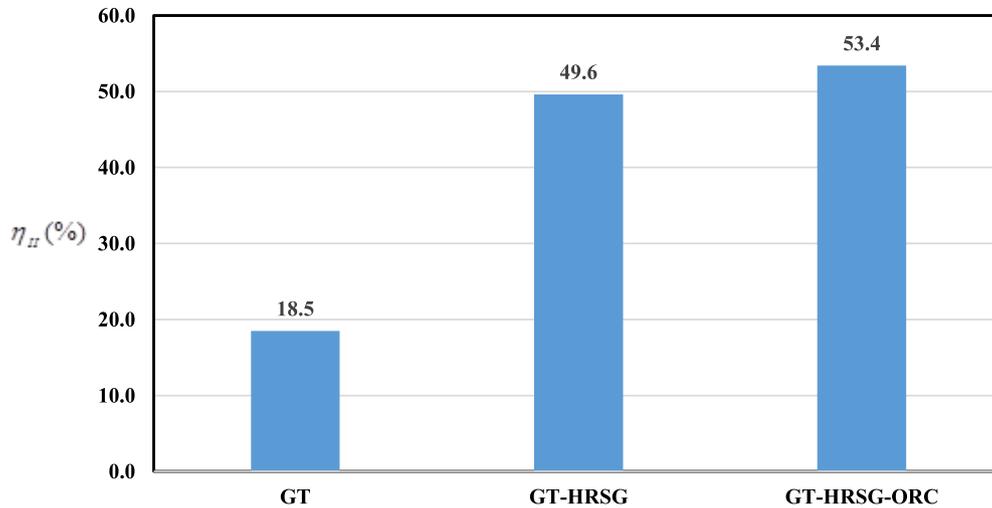
**Fig. 4.** Efficiency of the second law of thermodynamics for coolant R11 in the three cases namey GT, GT+HRSG, and GT+HRSG+ORC.
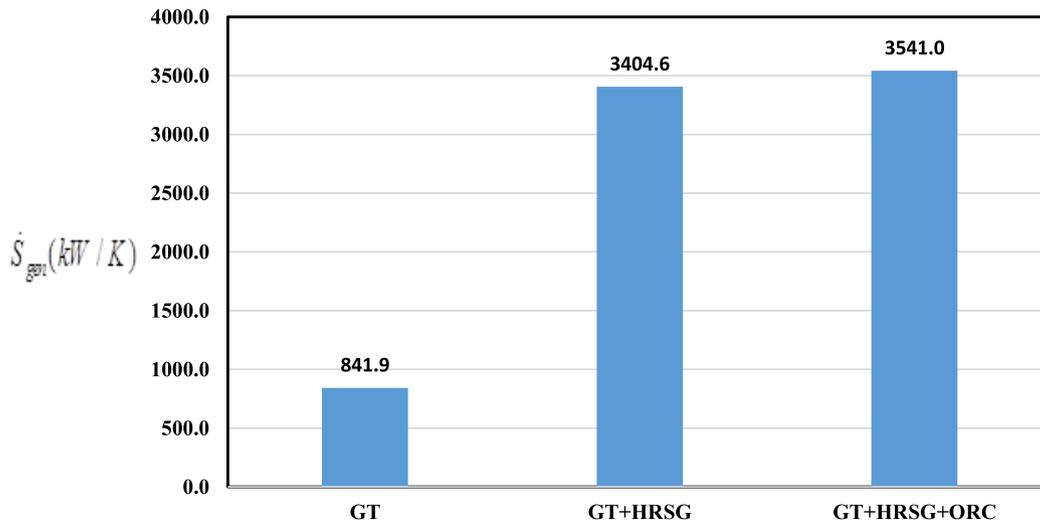


**Fig. 5.** Entropy of the cycle for coolant R11 in the three cases namely, GT+ GT+HRSG, and GT+HRSG+ORC.

Figure 5 shows the entropy generated in the cycle for Coolant R11 in three cases, namely GT, GT + HRSG, and GT + HRSG + ORC. Addition of the HRSG cycle to the GT in order to recover the energy of the exhaust gases increases the generated entropy by roughly fourfold. Addition of the ORC to the dual HRSG + GT cycle increased the generated entropy by 4%. Comparison in Figure 5 with Figures 4 and 3 suggests that addition of HRSG cycle and ORC results in increased efficiency of the first and second laws of thermodynamics in the triple cycle. However, on the other hand, the generated entropy is increased in proportion.

The efficiency of the first law of thermodynamics for the triple cycle system is shown in Figure 6 for four working fluids, namely C5F12, R123, R11, and butane. The changes in the efficiency of the first law of thermodynamics in the triple cycle system were in the range of 49.8% to 55.2%. The

different fluids in the ORC affected the efficiency of the first law of thermodynamics by roughly 10.8%. Figure 7 shows the efficiency of the second law of thermodynamics in the triple cycle system for four working fluids, namely C5F12, R123, R11, and butane. The trend of the diagram is similar to that of Figure 6.

Figure 8 shows the entropy generated in the triple cycle system for four working fluids, namely C5F12, R123, R11, and butane. It can be concluded from the figure that although butane achieves the highest efficiency in the first and second laws of thermodynamics, it also generates the highest amount of entropy.

Figure 9 demonstrates the price of the produced electricity in three cycles, namely GT, GT + HRSG, and GT + HRSG + ORC. As shown, despite the increased number of equipment in HRSG cycle and ORC, which also increases the initial price, the price of the produced
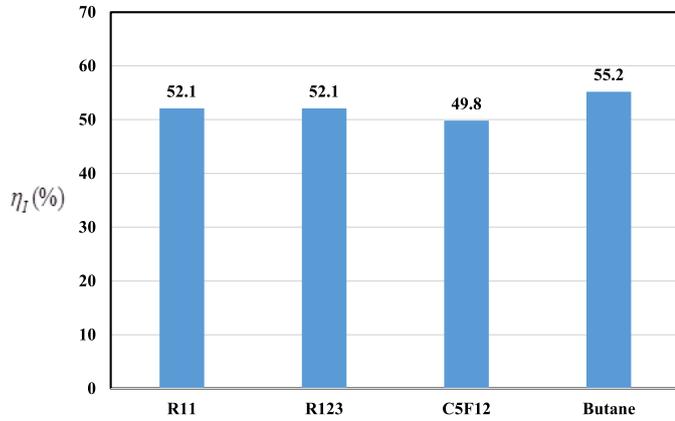
**Fig. 6.** Efficiency of the first law of thermodynamics in the triple cycle system of four working fluids, namely C5F12, R123, R11, and butane.
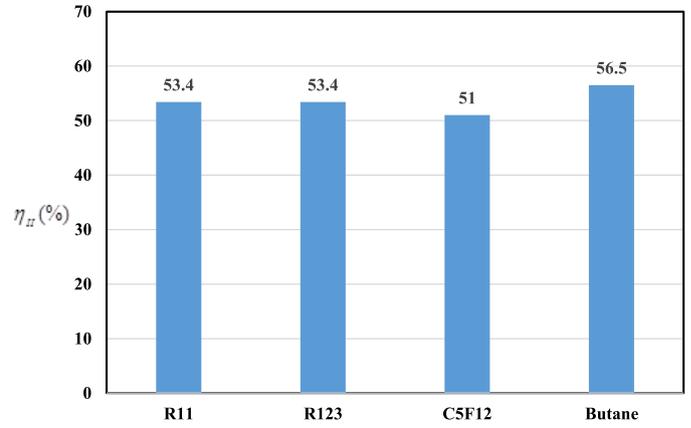


**Fig. 7.** Efficiency of the second law of the thermodynamics in the triple cycle system for four working fluids, namely C5F12, R123, R11, and butane.
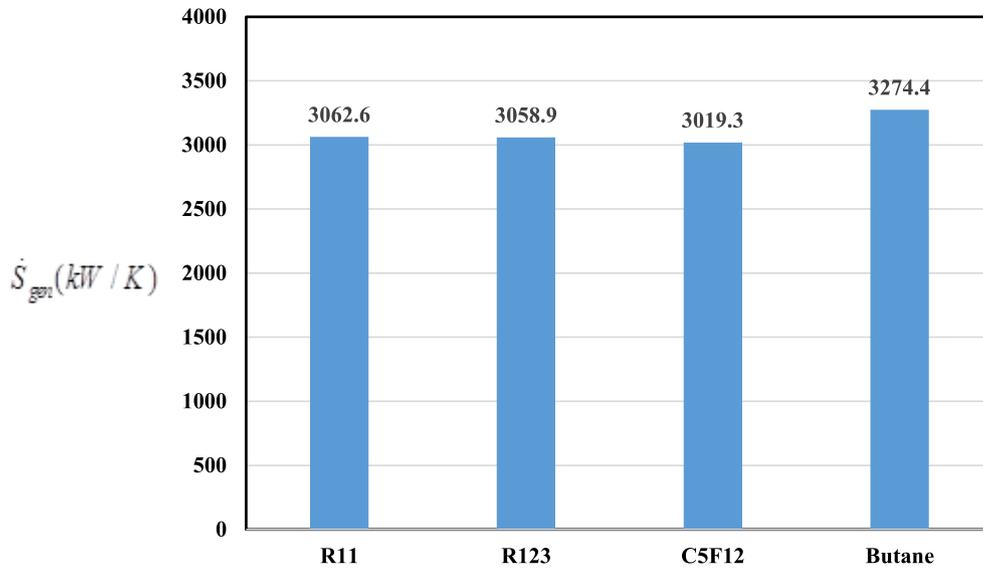


**Fig. 8.** Generated entropy in the triple cycle system for four working fluids, namey C5F12, R123, R11, and butane.

electricity is reduced due to the increase in cycle efficiency. This reduction in price is significant taking into account the addition of HRSG and GT cycles, however, in the case of ORC, the reduction is insignificant.

## 4 Conclusion

A new triple cycle thermodynamic system was presented in this study. In this configuration, the energy of the output from gas turbine and the wasted energy in the condenser of the steam cycle are recovered in the HRSG and evaporator of ORC, respectively, so as to be used as a heat source. A computer program was coded in MATLAB in order to analyze the aforementioned triple cycle. The Xstream and Refprob software along with M-Files were used to calculate

the thermodynamic properties of the steam and working fluid of the ORC. The proposed steam cycle in [30] was used to validate the results of the computer model. The highest error in the computer model was roughly 7.1% for the steam cycle. The results of this research are summarized as follows:

– the highest amount of entropy was associated with the combustion chamber of the gas cycle and the evaporator of the steam cycle;
– the efficiency of the first law of thermodynamics was improved by roughly 8% through addition of the ORC so as to recover heat from the steam cycle;
– the efficiency of the second law of thermodynamics was improved by roughly 8% through addition of the ORC so as to recover heat from the steam cycle;
– the entropy generated in the steam cycle was increased by roughly 4% through addition of the ORC cycle;
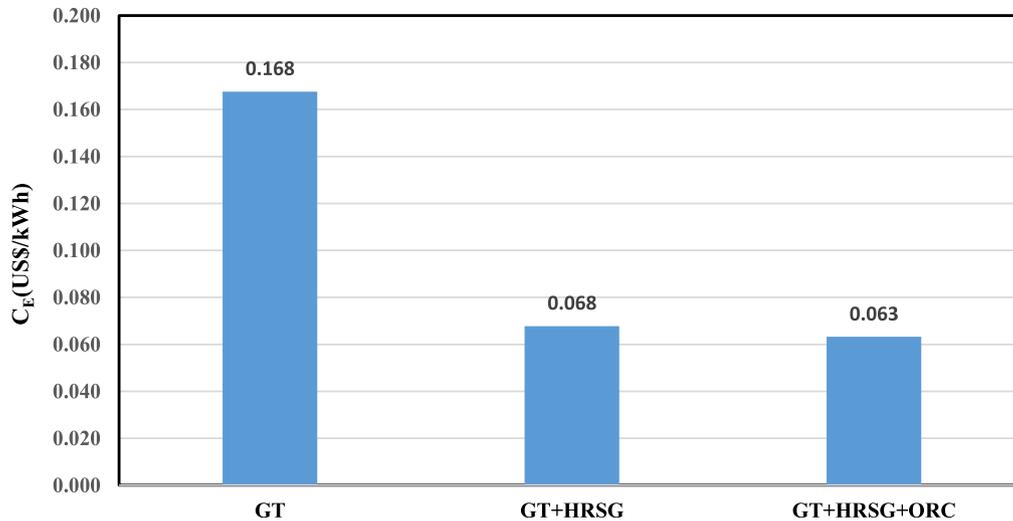
**Fig. 9.** Price of produced electricity in the three ctcles, namely GT, GT+HRSG, and GT+HRSG+ORC.

– altering the gas cycle to dual and triple cycle systems decreased the price of produced electricity from 0.168 (USD/kWh) to 0.068 (USD/kWh) and 0.063 (USD/kWh), respectively.

## Nomenclature

| | |
|---|---|
| $c_p(\frac{kJ}{kgK})$ | Constant pressure specific heat |
| $e(\frac{kJ}{kg})$ | Exergy |
| $h(\frac{kJ}{kg})$ | Enthalpy |
| $h_f^0(\frac{kJ}{kg})$ | Formation enthalpy |
| $h_0(\frac{kJ}{kg})$ | Reference point enthalpy |
| K | Equilibrium constant equations (9) and (10) |
| K | Ratio of specific heat coefficient |
| $LHV(\frac{kJ}{kg})$ | Lower heating value |
| $M(\frac{kJ}{kmole})$ | Molar mass |
| $\dot{m}(\frac{kg}{s})$ | Mass flow rate |
| N | Number of mole |
| $R(\frac{kJ}{kgK})$ | Gas constant |
| R | Pressure ratio |
| $R_u$(kJ/k mole K) | Universal gas constant |
| $T(C^0)$ | Temperature |
| $w(\frac{kJ}{kg})$ | Work per mass flow rate |
| $\dot{W}(kW)$ | Power |
| X | Mass percent |
| Y | Moral perce |

### Greek symbols

| | |
|---|---|
| $\eta$ | Efficiency |

### Subscript

| | |
|---|---|
| 5 to 19 | Points shown in Figure 1 |
| bc, g | Booster compressor |
| Cc | Combustion chamber |
| Ch | Chemical |
| c,g | Gas compressor |
| Co | Condenser |
| Drum | Drum |
| Eco | Economizer |
| F | Fuel |
| Gas | cycle |
| HRSG | Hear recovery team generation |
| ORC | Organic ranking cycle |
| P | Product (Eq. (11)), pump |
| Ph | Physical |
| R | Reactant |
| SH | Super heater |
| St | Steam, steam turbine (Eq. (23)) |
| Tg | Gas turbine |
| T | Total |

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