Energetic and Exergetic Investigation of Diesel - Brayton and Organic Rankine Combined Cycle

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Abstract: The amount of energy lost through the exhaust gases from diesel as well as gas turbine power plant is much enough to enhance the performance of the power system as well as to operate another secondary power generation system. If this energy is not to be used, it will results in global warming and enhance the local air pollution problem. Present study proposes a new cycle to utilizes exhaust gas energy from diesel engines and gas turbine power plants to optimize the performance of the combined cycle. In the proposed system, the waste energy from the exhaust gas of the diesel power plant is to used to enhance the thermal efficiency of the gas turbine cycle (GTC), and the waste energy from the exhaust gases of the GTC is to utilized to operate an organic Rankine cycle (ORC). To investigate the overall performance of the proposed system, the pressure ratio and turbine inlet temperature (TIT) of GTC are used as variables. The result shows that the gain in thermal efficiency and work output of the proposed system is about 15% to 43% and 13% to 20.4% as compared to diesel plus simple GTC system when TIT increases from 900K to 1200K. The exhaust gases exergy loss from the diesel system and simple GTC is much high as compared to the proposed-system. The total exhaust gases exergy loss from the proposed-system is decreased by 3.5 to 1.35 times as compared to the total exhaust gases exergy loss from diesel cycle and simple GTC.

Keywords: Work output, Thermal efficiency, SFC, Exergy loss, Diesel engine, GTC, ORC

بحث نشط ودقيق للديزل - دورة برايتون ورانكين العضوية المركبة

الملخص: إن كمية الطاقة المفقودة من خلال غازات العادم من الديزل وكذلك محطة توليد الطاقة التوربينية الغازية كافية لتحسين أداء نظام الطاقة وكذلك لتشغيل نظام ثانوي آخر لتوليد الطاقة. إذا لم يتم استخدام هذه الطاقة، فسيؤدي ذلك إلى ارتفاع درجة حرارة الأرض وتعزيز مشكلة تلوث الهواء المحلي. تقترح الدراسة الحالية دورة جديدة لاستخدام طاقة غاز العادم من محركات الديزل ومحطات الطاقة التوربينية الغازية لتحسين أداء الدورة المركبة. في النظام المقترح، يتم استخدام الطاقة المهدرة من غاز العادم لمحطة توليد الطاقة بالديزل لتعزيز الكفاءة الحرارية لدورة التوربينات الغازية (GTC)، ويتم استخدام الطاقة المهدرة من غازات العادم في GTC للعمل دورة رانكين عضوية (ORC). لاستكشاف الأداء العام للنظام المقترح، يتم استخدام نسبة الضغط ودرجة حرارة مدخل التوربينات (TIT) لـ GTC كمتغيرات. تظهر النتيجة أن الكسب في الكفاءة الحرارية وإخراج العمل للنظام المقترح هو حوالي ١٥٪ إلى ٤٣٪ و١٣٪ إلى ٢٠.٤٪ مقارنة بالديزل بالإضافة إلى نظام GTC البسيط عندما يزيد TIT من ٩٠٠ ألف إلى ١٢٠٠ ألف. إن فقدان الطاقة الناتجة عن غازات العادم من نظام الديزل وGTC البسيط مرتفع للغاية مقارنة بالنظام المقترح. تم تقليل إجمالي فقدان الطاقة الناتجة عن غازات العادم من النظام المقترح بمقدار ٣.٥ إلى ١.٣٥ مرة مقارنة بإجمالي فقدان الطاقة الناتجة عن غازات العادم من دورة الديزل وGTC البسيط.

1. Introduction

Now a day, the domestic, as well as industrials demand for electrical energy worldwide, is mainly covered by the massive capacity power plant. Due to the continuous increase of fuel prices at the international level and continuous power demand, a simple gas turbine cycle (GTC) based power plant and diesel power plant is not sufficient to fulfill the demand for electrical power and price of power generation. Generally, the temperature of exhaust gases from the GTC plant and diesel plant is very high that results in global warming as well as increase air pollution. The energy carried out by the exhaust gases from large capacity GTC and diesel power plant is much enough that is capable to run another cycle which may be GTC, Rankine cycle (RC), and organic Rankine cycle (ORC). Whatever is the cycle, in general such cycles are known as the bottoming cycle. The results of the operation of the bottoming cycle by the exhaust of the topping cycle (GTC or Diesel power plant), helps to reduce global warming and air pollution [1–6]. Power generation from the exhaust of the topping cycle gives the concept of a combined cycle power plant (CCPP). From last decade, several researchers show that some major factors affect the overall performance of CCPP like turbine inlet temperature (*TIT*) [7–9], air compressor inlet temperature and density [10–13], and last but not the least is air compressor pressure ratio [8,14]. Hüseyin et al. [15] parametrically investigate the organic Rankine cycle (ORC) in the case of supercritical and subcritical operated by the energy of exhaust gases of power engine and biogas-fueled combined cycle. Results show that in comparison to subcritical ORC and supercritical ORC, supercritical ORC is superior to subcritical ORC in terms of energy and exergy efficiency, and net power. Chen et al. [16] thermodynamically investigated the open combined Brayton and two parallel inverse Brayton cycles. Results show that by regulating the compressor inlet pressure of the bottom cycles, the proposed model performance was improved. Khan [17] investigate the performance of air bottoming combined cycle and regenerative gas turbine cycle operated by the partial amount of exhaust gasses from the gas turbine. This study presents the unique technique to compare the performance of these two cycles and prove that for thermal efficiency and exhaust gasses exergy loss by regenerative GTC is much better as compared to the air bottoming cycle but for net power output air bottoming cycle is better than regenerative GTC. In another study, Khan and Tlili [9] investigated the importance of heat exchanger in regenerative topping gas turbine and air bottoming cycle connecting through the bypass valve. The study proves that by proper use of heat exchanger and bypass valve work output of the combined cycle increases from 13.5 to 45% and combined cycle efficiency increases from 15% to 31%. Mishra and kumar [18] proposed organic bottoming cycle operated by the energy carried out by the exhaust gases from the gas turbine cycle. The study was conducted for two different fluids that is R123 and R245fa to evaluate the overall performance of combined system. Results show that the thermal efficiency of combined cycle in case of R123 is more as compared to R245fa but the work output of combined cycle for R123 is more as compared to R245fa. Galindo et al. [19] theoretically investigated the Brayton cycle operated by the exhaust of a passenger car. In this study, energy recovery from the exhaust from a 2-L turbocharged gasoline engine was suggested. The result shows that the suggested method increased engine efficiency by up to 15%. Zhu et al. [20] theoretically investigate the energy and exergy analyses of the bottoming Rankine cycle run by the heat carried by the exhaust gases from the engine. Results show that out of five selected fluids for the engine, R113 and ethanol reflect the best thermodynamic performance over the complete exhaust gasses temperature range. Also, the main factor that influences the system performance is superheating temperature, working fluid properties, and evaporating pressure.

The above literature reveals that many factors affect the performance of the combined cycle power plant (CCPP). A several number of techniques have been suggested by several researchers to boost the performance of the combined cycle power plant (CCPP) [21–26]. In the proposed system, the exhaust of the diesel engine is used to preheat the air to the combustion chamber of GTC, which results in a decrease of fuel supply for attending the required *TIT*. Also, the exhaust of the GTC used to operate the ORG where working fluid is R134a under the present study.

The novelty of the present study is that the work output and thermal efficiency of the proposed system is much effective as related to Simple GTC and diesel cycle. The exhaust gases exergy losses from the proposed system is much less as compared to the exhaust gases exergy losses from the diesel cycle and simple GTC. This means that the proposed system is very efficient to reduce the exhaust gases exergy losses from the diesel cycle as well as from Simple GTC. The application of the present study is directly applicable where the temperature of exhaust from the diesel power plant is very high and the energy carried out by the exhaust gases from the diesel power plant is enough to preheat the compressed air of the GTC power plant. The one of the major challenge of the present system is that the proposed system is more complicated in design and operation as compared to the Diesel power plant and simple GTC power plant. The graphical presentation of energy and exergy results are helpful for the readers to understand the uniqueness of the proposed system.

2. Cycle description

Figure 1 and fig. 2 shows the schematic and T-s diagram of the proposed system. Air from the environment at atmospheric temperature and pressure T_1 and P_1 enter the diesel engine. The exhaust gases from the diesel engine at temperature T_4 and pressure P_4 leaves to enters the environment via a heat exchanger (*H.E.*) where it heats the compressed air from the air compressor of the GTC plant. This results in a temperature decrease of exhaust gases of diesel engine from T_4 to T_5 and the compressed air temperature increased from T_7 to T_8. The net work output, mass flow rate of flue gases and thermal efficiency of the diesel engine are given by the equation (4), equation (8), and equation (9) respectively.

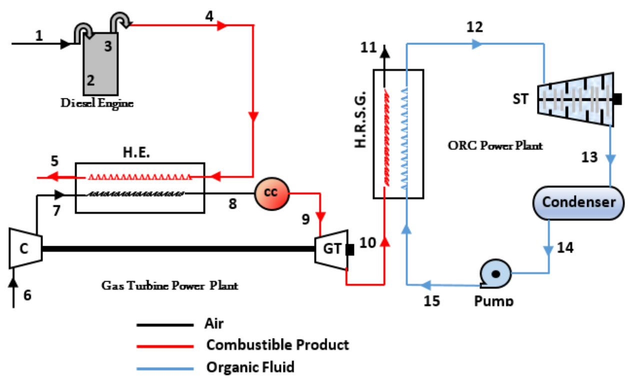

Figure 1. Schematic diagram of the proposed system

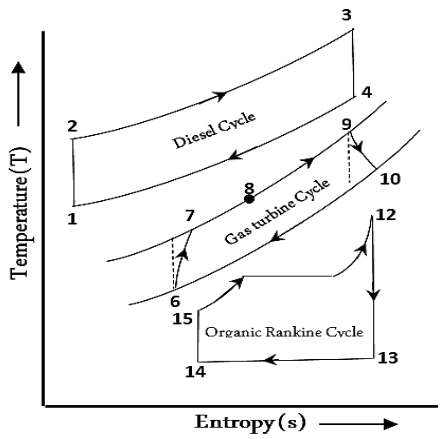

Figure 2. T-s diagram of the proposed system

In GTC plant, air from the environment at atmospheric temperature and pressure T_1 and P_1 enters the air compressor and leaves at temperature T_2 and pressure P_2. The compressed air at temperature T_7 and pressure $P_7=P_8$ enters the combustion chamber via the *H.E.* where it mixes with fuel (Natural gas) resulting in the rise of combustion products temperature to T_9 (*TIT*). The combustible product enters the heat recovery steam generator (HRSG) at temperature T_{10} and pressure P_{10} via a gas turbine. The exhaust gases from the gas turbine leave to the environment via HRSG exchange which heats to the refrigerant (R134a) of the organic Rankine cycle (ORC). The governing equations for work output, thermal efficiency, SFC, etc. for GTC are listed below from equation 10 to equation 21. In ORG, refrigerant at pressure P_{15} enters the HRSG where it gains heat from the exhaust gases of the GTC plant. The refrigerant at pressure P_{15} expands to pressure P_{13} in the turbine. The governing equations for ORG are listed below from equation 22 to equation 25. The major assumptions considered for the analysis of the proposed system are itemized in table 1.

Table 1. List of supposed constraints and variables [11,27,28].

Variables		
Turbine inlet temperature of GTC	*TIT (K)*	900 to 1200
Pressure ratio of GTC	r_p	4 to 12
Assumed parameters		
Isentropic efficiency of gas turbine	η_{gt}	0.85
Isentropic efficiency of compressor	η_c	0.85
Combustion chamber efficiency	η_{cc}	1.0
Heat exchanger effectiveness	ε	0.9
Isentropic efficiency of steam turbine	η_{st}	0.95
Pump efficiency	η_{pump}	0.95
Volume ratio of diesel engine	r	20
Cut-off to stroke ratio of diesel engine	k	5%
Compressor inlet & ambient temperature	T_1 (*K*)	300
Mass flow rate of air to GTC	m_a (kg/s)	1
Mass flow rate of air to diesel engine	m_a (kg/s)	1
Lower calorific value of fuel to GTC	LCV (kJ/kg)	42000
Lower calorific value of Diesel	LCV (kJ/kg)	45500

3. Solution Technique

Using first and second laws of thermodynamics, parametric analyses of work output, thermal efficiency, SFC, and exhaust gases exergy losses of simple GTC, and the combined cycle has been carried out in the present study with certain assumptions listed in table 1. The pressure ratio and turbine inlet temperature of GTC are the variables to analyse and compare the proposed system with simple GTC and diesel engine in the range of 4 to 12 and 900K to 1200K respectively. Engineering Equation Solver (EES) is used to solve equations listed in section 4 based on assumptions and variables listed in table 1.

4. Mathematical modelling of Proposed system

For mathematical modeling of the proposed system shown in figure 1, the complete system is divide into three major sections. The major governing equations for the parameter under an analysis of the first one (diesel cycle), the second section (gas turbine cycle), and the third section (organic Rankine cycle) is listed below

Analysis of Diesel cycle

Heat supplied to Diesel cycle is given by $\left(\dot{Q}_s\right) = \dot{m}_g .c_{pg} .(T_3 - T_2)$ (1)

Heat rejected by Diesel cycle is given by $\left(\dot{Q}_R\right) = \dot{m}_g .c_{vg} .(T_4 - T_1)$ (2)

Net output from Diesel cycle $(W)_{netdiesel} = \dot{m}_g .c_{vg} .\left\{\gamma_g (T_3 - T_2) - (T_4 - T_1)\right\}$ (3)

After putting the value T_2, T_3, T_4 in term of r, ρ and T_1 in equation (3) we get

$$(W)_{net} = T_1.\dot{m}_g .c_{vg} .\left\{\gamma_g .r^{(\gamma_g - 1)} (\rho - 1) - \left(\rho^{\gamma_g} - 1\right)\right\} \quad (4)$$

Where $\rho = 1 + k\,(r - 1)$ (5)

From equation (4) & (5)

$$(W_{net})_t = T_1.\dot{m}_g .c_{vg} .\left\{k .\gamma_g .r^{(\gamma_g - 1)} (r - 1) + 1 - \left[\left(1 + k\,(r - 1)\right)^{\gamma_g} - 1\right]\right\} \quad (6)$$

Temperature of exhaust gases from the Diesel cycle is

$$(T_4) = T_1.\left\{1 + k\,(r - 1)\right\}^{\gamma_g}$$

(7)

Mass flow rate of flue in Diesel Engine is given by $(\dot{m}_f)_{diesel} = \dfrac{T_1.(\dot{m}_a)_t .c_{pg} .k .(r - 1).r^{(\gamma_g - 1)}}{cv - c_{pg} .T_1.\left[1 + k\,(r - 1)\right].r^{(\gamma_g - 1)}}$ (8)

Air Standard Thermal Efficiency of Diesel cycle $(\eta_{th})_{diesel} = 1 - \dfrac{\left[\left(1 + k\,(r - 1)\right)^{\gamma_g} - 1\right]}{k .\gamma_g .r^{(\gamma_g - 1)} (r - 1)}$ (9)

Analysis of Gas Turbine Cycle

Compressor Work is $(W_c) = \dot{m}_a .(h_7 - h_6)$ (10)

where $T_7 = T_6 \left\{1 + \left(r_p^{\alpha} - 1\right)/\eta_c\right\}$ and $\alpha = (\gamma_a - 1)/\gamma_a$ (11)

Work from gas turbine is $(W_{GT}) = \dot{m}_g .(h_9 - h_{10})$ (12)

where $T_{10} = T_9 \left\{1 - \eta_t \left(1 - r_p^{-\beta}\right)\right\}$ and $\beta = \left(\gamma_g - 1\right)/\gamma_g$ (13)

Mass flow rate of gasses is $\dot{m}_g = \dot{m}_a + \dot{m}_f$ (14)

where $\dot{m}_f = \dot{m}_a .\left\{\left(c_{pg} .T_9 - c_{pa} .T_8\right)/\left(LCV - c_{pg} .T_9\right)\right\}$ (15)

The work output from GTC is

$$(W_{net})_{GTC} = (W_{GT}) - (W_c) \quad (16)$$

The GTC thermal efficiency is $(\eta_{th})_t = (W_{net})_t / Q$ (17)

The SFC of GTC is $(SFC)_t = \left(3.6 \times 10^5\right)/\left((\eta_{th})_t .LCV\right)$ (18)

For first heat exchanger (H.E.)

$$\dot{m}_a .\left(h_8 - h_7\right) = \varepsilon_1 .\dot{m}_{gdiesel} .\left(h_4 - h_7\right) \quad (19)$$

Therefore

$$T_8 = (1-\varepsilon_1)T_7 + \varepsilon_1.\left(\dot{m}_{gdiesel}/\dot{m}_a\right).\left(c_{pg}/c_{pa}\right)T_4 \quad (20)$$

$$T_5 = T_4 - \left(\dot{m}_a/\dot{m}_{gdiesel}\right).\left(c_{pa}/c_{pg}\right).\left(T_8 - T_7\right) \quad (21)$$

Analysis of Organic Rankine Cycle

Mass flow rate of refrigerant is $\dot{m}_{ref} = \varepsilon.m_g.\left(h_{10}-h_{11}\right)/\left(h_{12}-h_{15}\right)$ (22)

Work output from the turbine is $W_{ST} = \dot{m}_{ref}.\eta_{ST}.\left(h_{12}-h_{13}\right)$ (23)

Work required to run the pump $W_{pump} = \dot{m}_{ref}\,\nu_f.\left(p_{12}-P_{14}\right)$ (24)

Work output is $\left(W_{net}\right)_{ORC} = \left(W_{ST}\right) - \left(W_{pump}\right)$ (25)

Analysis of combined Cycle

The combined cycle work output is $\left(W_{net}\right)_{comb} = \left(W\right)_{netdiesel} + \left(W_{net}\right)_{GTC} + \left(W_{net}\right)_{ORC}$ (26)

Combined cycle thermal efficiency is $\left(\eta_{th}\right)_{comb} = \left[\left(W_{net}\right)_{comb} \Big/ \left(\left(\dot{m}_f\right)_{diesel} \times LCV_{diesel} + \left(\dot{m}_f\right)_{GTC} \times LCV_{GTC}\right)\right] \times 100$ (27)

combined cycle SFC is $\left(SFC\right)_{comb} = 3600 \times \left(\left(\dot{m}_f\right)_{diesel} + \left(\dot{m}_f\right)_{GTC}\right) \Big/ \left(W_{net}\right)_{comb}$ (28)

5. **Result and discussion**

The pressure ratio and turbine inlet temperature (TIT) is used to analyse the energetic and exergetic performance of the proposed system and comparative analysis of the proposed system with simple GTC and diesel engine power plant. In the proposed system the exhaust gases are used to heat the compressed air of the GTC cycle and the exhaust gases of the GTC cycle are used to heat the refrigerant in the ORC cycle. In this way, the heat gains by the compressed air of the GTC cycle from the exhaust gases of the diesel cycle helps to improve the GTC cycle efficiency by increasing the combustion chamber air inlet temperature. On the other hand, the heat grains by the refrigerant of the ORG cycle from the exhaust gasses of the GTC cycle helps to enhance the work output of combined GTC-ORC. The effect of pressure ratio (r_p) and turbine inlet temperature (*TIT*) on the thermal efficiency, work output, SFC, and exhaust gas exergy losses of the combined cycle are examined parametrically using the first and second law of thermodynamic.

The variation of work output of diesel cycle, simple GTC plus diesel cycle, and combined cycle (proposed system) with respect to a pressure ratio (r_p) and turbine inlet temperature (*TIT*) of GTC cycle in figure 3 (a) and 3(b). It is clear from these figures that the work output of the combined cycle as well as simple GTC is significantly affected by *TIT* and r_p. It is also observed that the work output of the combined cycle as well as simple GTC decreases with r_p and increases with an increase in *TIT*. The work output of the diesel cycle is independent of *TIT* and r_p as clear from equation (3) and due to this, it is not affected by *TIT* and r_p. Figure 3 (a) and (b) also indicates that the work output of the diesel cycle as well as summation of work output from diesel and GTC cycle is much less as compared to the proposed system. The work output of the proposed system at r_p=4, is 35.6% and 69.6% more as compared to the work output of diesel cycle when *TIT* = 900K and *TIT* = 1200K respectively. At r_p=12, it is 17.6% and 60.9% when *TIT* = 900K and *TIT* = 1200K respectively more as compared to work output of diesel cycle.

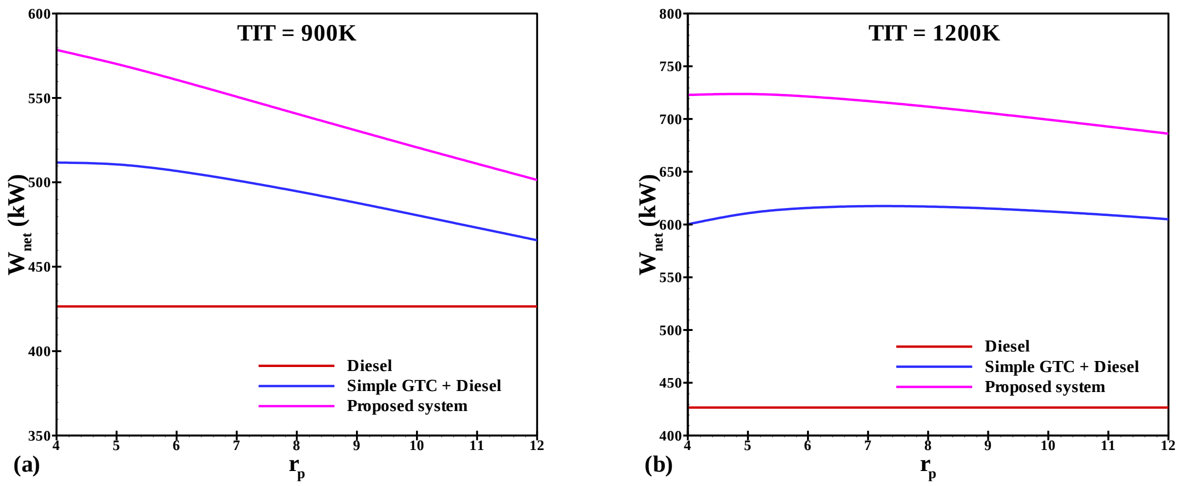

Figure 3. Variation of work output of diesel cycle, simple GTC plus diesel cycle and combined proposed system with respect to r_p at (a) *TIT* = 900K and (b) *TIT* = 1200K.

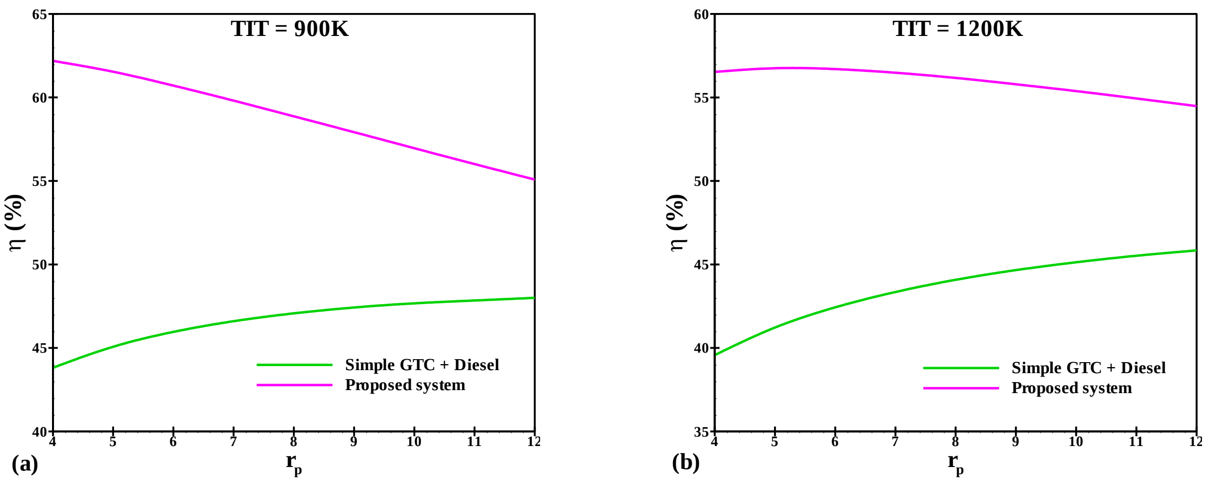

Figure 4. Variation of thermal efficiency of simple GTC plus diesel cycle and combined proposed system with respect to r_p at (a) TIT = 900K and (b) TIT = 1200K

From figure 3(a) and 3(b), at r_p=4, the work output of the proposed system is 13.1% and 20.5% more as compared to the work output of simple GTC plus diesel cycle when TIT = 900K and TIT = 1200K respectively. Whereas at r_p=12, it is 7.7% and 13.5% when TIT = 900K and TIT = 1200K respectively more as compared to work output of simple GTC plus diesel cycle. Figure 4(a) and 4(b) shows the variation of thermal efficiency of simple GTC plus diesel cycle and combined cycle (proposed system) with respect to a pressure ratio (r_p) and turbine inlet temperature (TIT) of GTC cycle. It noted from these figures that the thermal efficiency of the proposed system decreases with pressure ratio and increases with turbine inlet temperature (TIT) of the GTC cycle whereas, the thermal efficiency of simple GTC increases with pressure ratio and turbine inlet temperature (TIT) of GTC cycle. From figure 3(a) and 3(b), the work output of the diesel cycle is independent of the pressure ratio (r_p) and turbine inlet temperature (TIT) of the GTC cycle, this means that the thermal efficiency of the diesel cycle is also independent of pressure ratio (r_p) and turbine inlet temperature (TIT) of GTC cycle. From figure 4(a) and 4(b), at r_p=4, the thermal efficiency of the proposed system is 42% and 43% more as compared to work thermal efficiency of simple GTC plus diesel cycle when TIT = 900K and TIT = 1200K respectively. Whereas at r_p=12, it is 14.8% and 18.9% when TIT = 900K and TIT = 1200K respectively more as compared to the thermal efficiency of simple GTC plus diesel cycle.

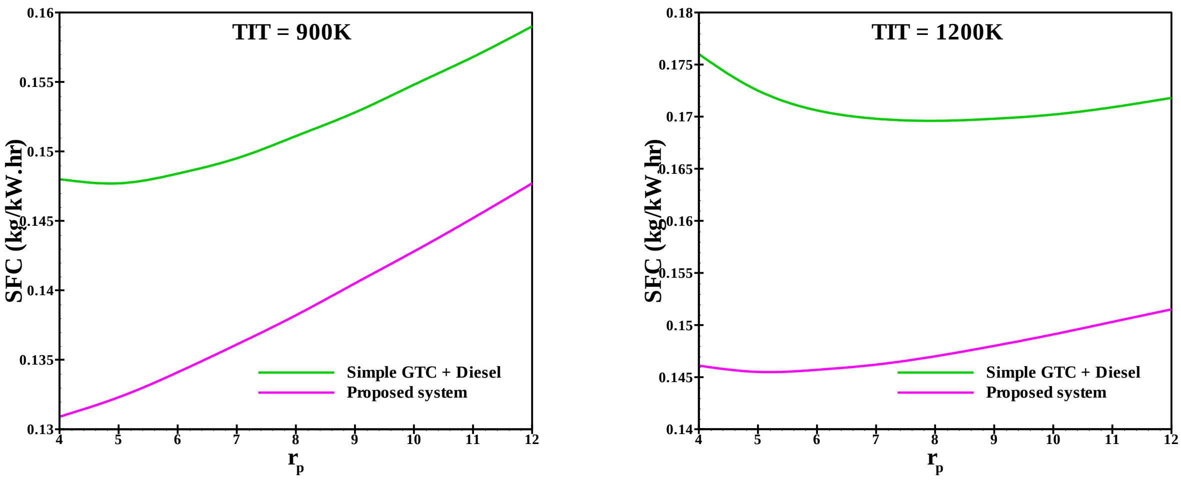

Figure 5. Variation of SFC of simple GTC plus diesel cycle and combined proposed system with respect to r_p at (a) TIT = 900K and (b) TIT = 1200K

The SFC of any cycle defines, as the ratio of the mass flow rate of fuel to work output of the cycle, this means that if the work output of the cycle increases, SFC decreases and vice versa. Figure 5 (a) and 5(b) shows the variation of SFC of simple GTC plus diesel cycle and combined cycle (proposed system) with respect to r_p and *TIT* of GTC cycle. It observed that the SFC of the proposed system increase with pressure ratio because the work output of the proposed system decreases with r_p as shown in figure 3(a) and 3(b). Also, the SFC of the proposed system decreases with the turbine inlet temperature (*TIT*) of the GTC cycle. The SFC of simple GTC increases with pressure ratio and turbine inlet temperature (*TIT*) of the GTC cycle. From figure 5(a) and 5(b), at r_p=4, the SFC of the proposed system is 11.5% and 17% less as compared to work thermal efficiency of simple GTC plus diesel cycle when *TIT* = 900K and *TIT* = 1200K respectively. Whereas at r_p=12, it is 7.1% and 11.8% when *TIT* = 900K and *TIT* = 1200K respectively more as compared to the thermal efficiency of simple GTC plus diesel cycle.

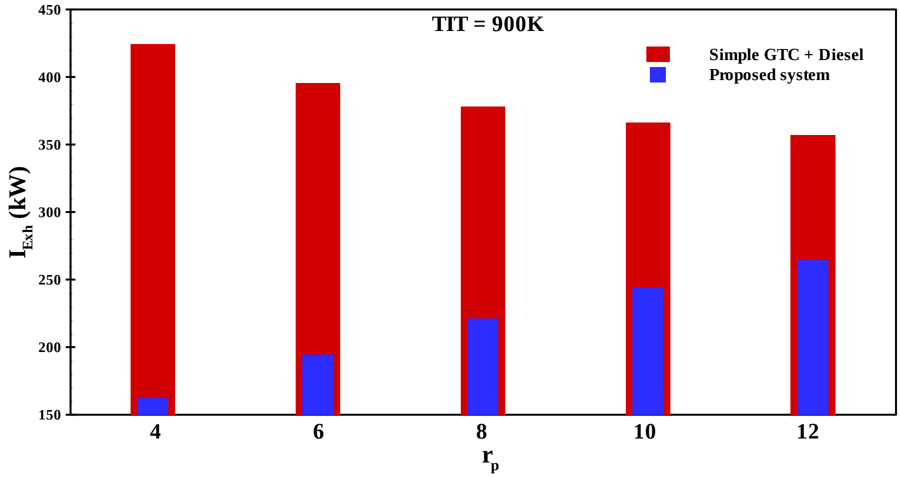

Figure 6 (a). Variation of exhaust gases exergy loss of simple GTC plus diesel cycle and combined proposed system with respect to r_p at *TIT* = 900K

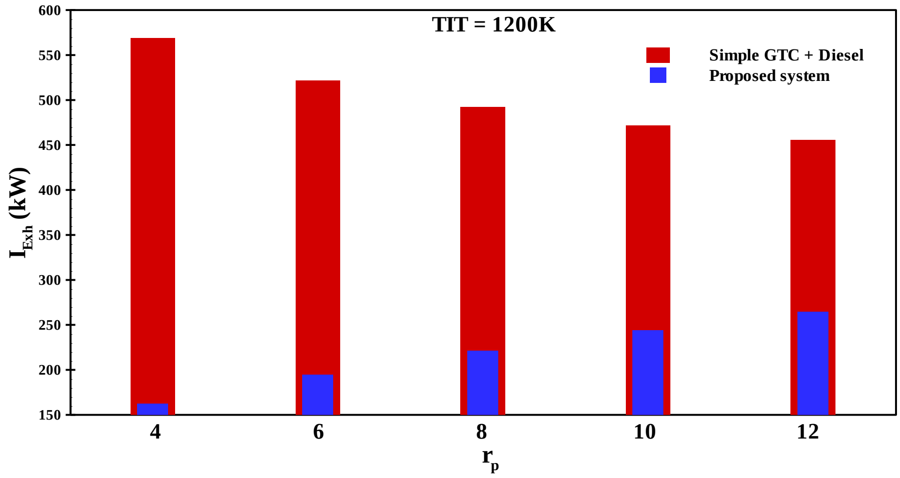

Figure 6 (b). Variation of exhaust gases exergy loss of simple GTC plus diesel cycle and combined proposed system with respect to r_p at *TIT* = 1200K

The exhaust gas exergy loss is significantly increased or decreased with an increase or decrease in the temperature of exhaust gases. Figure 6(a) and 6(b) shows the variation of exhaust gas exergy losses of simple GTC plus diesel cycle and combined cycle (proposed system) with respect to a pressure ratio (r_p) and *TIT* of GTC cycle. The exhaust gas exergy losses of the proposed system increase with pressure ratio because the temperature of exhaust gasses leaving to the environment from diesel engine increases with pressure ratio as the difference of T_8 and T_7 decreases with pressure ratio. Also, the exhaust gases exergy loss of the proposed system is not significantly affected by the turbine inlet temperature (*TIT*) of the GTC cycle. The SFC of simple GTC increases with pressure ratio and *TIT* of the GTC cycle. From figure 6(a), the exhaust gas exergy losses of the proposed system is 2.615 times at r_p=4 and 1.35 times at r_p=12 less as compared to simple GTC plus diesel cycle whereas in figure 6(b) the exhaust gases exergy loss of the proposed system is 3.5 times at r_p=4 and 1.72 times at r_p=12 less as compared to simple GTC plus diesel cycle.

6. **Conclusion**

In the present study, a thermodynamic analysis of the proposed system has been conducted. Parametric analyses have been done by varying the pressure ratio of GTC cycle from 4 to 14, and turbine inlet temperature 900*K* to 1200*K*. Based on the analyses, the following are the conclusions are finalized:

1. The work output of the proposed system plant increases by 13% to 20.4%, when TIT increases from 900K to 1200K.
2. The thermal efficiency of the proposed system plant increases by 15% to 43%, with an increase of TIT from 900K to 1200K.
3. The SFC of the proposed system is much less as compared to the sum of SFC of diesel and GTC system. The SFC of proposed system is 7.1% to 11.8% is less to the sum of SFC of diesel and GTC system.
4. The total exhaust gases exergy loss from the proposed system is decreased by 3.5 to 1.35 times as compared to the overall exhaust gases exergy loss from diesel cycle and simple GTC.
5. Overall the proposed system is much better in terms of energy as well as exergy performance as compared to simple diesel cycle and simple GTC cycle.

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