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Influence of operating variables on exergetic performance of a combined cycle powered plant (CCPP) with intercooled-compression

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Abstract
The influence of operating variables (OPV) on the exergetic performance of a CCPP with intercooled compression is presented. The objective is to study the effect of OPV such as ambient temperature (TA), isentropic efficiency, turbine inlet temperature, and pressure ratio at varying intercooler effectiveness (EF) on plant performance and also evaluate the thermo-sustainability indicators at different EF. The results indicate at EF and TA between $0.75 \leq \mathrm{EF} \leq 0.9$ and $298 \leq \mathrm{TA} \leq 305$ K, the system exergy destruction (ED) decreased by 3.2 % with a 1.2 % reduction in the CCPP components. The EF improved the system efficiency at 90 % by 2.79 % for increasing PR. The exergetic sustainability index (ESI) was 1.2 while the exergy recoverability ratio (ERR) and environmental impact factor (EIF) fluctuated between $0.486 \leq \mathrm{ERR} \leq 0.629$ and $1.099 \leq \mathrm{EIF} \leq 2.560$. Optimum system efficiency ranged between 60.05 and 60.87 % for optimum EF with ESI of 2.56 compared to 1.2 at normal scenario.

Keywords: Exergy, efficiency, thermo-sustainability, intercooler, Gas-turbine.

1. Introduction

Energy and exergy concepts are extensively applied at present to adequately comprehend the thermodynamics of energy transformation processes and effective utilization of energy sources. However, the latter will ensure an equilibrium between eco-friendly and socio-economic sustainability (Aydın et al. 2013). Additionally, exergy has been defined as a thermodynamic retreat between a system and its immediate surroundings, which is progressively acknowledged as a measure for assessing environmental impacts occasioned by waste gas emissions (Dincer and Rosen, 2005). Conversely, this proposes that exergy as a thermodynamic model may offer the opportunity to identify areas in a thermal system with high enhancement prospective (Dincer and Rosen, 1998). For this reason, most power plant researchers have performed different optimization of the various operating parameters through exergy analysis. Intended at establishing the optimal conditions which will bring the best performance and less environmental concerns. The works of (Omendra and Kaushik, 2012) considered different thermodynamic variables affecting the exergy-based performance of a thermal steam plant. Results from these studies indicate that energy and exergy efficiencies, as well as the irreversibility rates in plant components, are affected by variations in ambient and stack gas temperatures. Further applications of exergy have brought different innovations in systems and thermal processes associated with energy generation for different thermal cycles. Some are contained works of (Abam et. al. 2012; Reddy et al. 2012; Balli and Hepbasli 2013, Reddy et al. 2014, Abam et. al.2017). Added applications of exergy for sustainability evaluation exist in (Aydin, 2013; Aydın and Önder 2013; Midilli et al. 2011).

Similarly, the exergy framework has been applied with economic principles to develop cost functions for thermal processes, a term called thermoeconomic or exergoeconomic. Research in this area include the works of (Ahmadi et al. 2011) who formulated objective functions, to contain capital investment, maintenance and operation cost of a system for district heating based on thermoeconomic framework while (Balli, et. al. 2010) described an intercooled reheat gas turbine (GT) plant, without and with recuperation for cogeneration applications based on the same principles. Also, other applications based on exergo-environmental analysis are discussed in (Abam et al., 2017; Kaviri et al. 2013, Ahmadi et al. 2012; Pouria et al. 2012; Ahmadi et al. 2011).

The knowledge on how the operating parameters of a thermal system vary with system conditions is imperative, as this may affect investment decision. On this circumstance, this paper is aimed at providing a theoretical itemisation on the influence of operating variables on exergeticperformance for an adapted GT cycle applied to a CCPP with an intercooled-compression at the topping cycle. The considered operating variables are the ambient temperature (TA), isentropic efficiency (IE), turbine inlet temperature (TIT) and pressure ratio (PR). The specifics of the research will comprise (i) the influence of these operating variables on exergetic performance and thermo-sustainability indicators at specified intercooler effectiveness (EF), and (ii) the estimation of optimal conditions for best system performance. Nonetheless, the latter may influence future design and in practice assist operational decision for in-service systems. Therefore, the study contribution is considered worthwhile since data for such GT configuration and in the study, perspectives are limited in the open domain.

2. The CCPP System description

Fig. 1 presents the combined cycle thermal plant with intercooled compression at the topping section. Air from the atmosphere enters state 1 at the low power compressor (LPC) and is compressed isentropically at high temperature to state 2. The exiting air at state 2 enters the intercooler where the temperature is reduced at constant pressure to state 3 and compressed further to state 4 through the high-pressure compressor (HPC). The compressed air from the HPC goes into the regenerative heat exchanger (HE) from state 4 to 5. At state 6 (combustion chamber) fuel is added to the compressed air increasing the exit burnt gas temperature to state 7. Additionally, the hot gas expands through the high-pressure turbine (HPT) to state 8 performing mechanical work. The expanded hot gas is further repeated at state 8 through 10 by a reheater (REH). Subsequently, the exhaust gas expands in the power turbine (PT) which drives a generator thus producing electricity. Moreover, the exhaust gas at state 11 partially increased the air temperature leaving the high-power compressor to the combustion chamber whereas that at state 12 is directed to the waste heat recovery boiler (WHRB) to generate steam. The produced steam in the WHRBenters the steam turbine (ST) at state 13 expanding to condenser pressure at state 16 thus driving a load. The steam turbine exhaust in the condenser condenses to saturation liquid at state 17 and further fed by the pump (P1) in the FWH 2 at state 18 where direct mixing is obtained with the steam bled from the steam turbine at state 15. At state 19 the saturated liquid is directed by pump 2 to the FWH1 at state 20 mixing takes place with bled steam from the ST at state 14. Furthermore, at state 21, the saturated liquid through pump 3 to the WHRB generating heat and the cycle is reiterated.

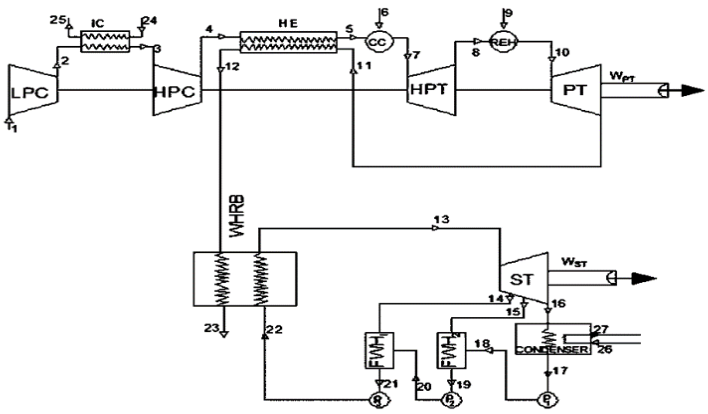
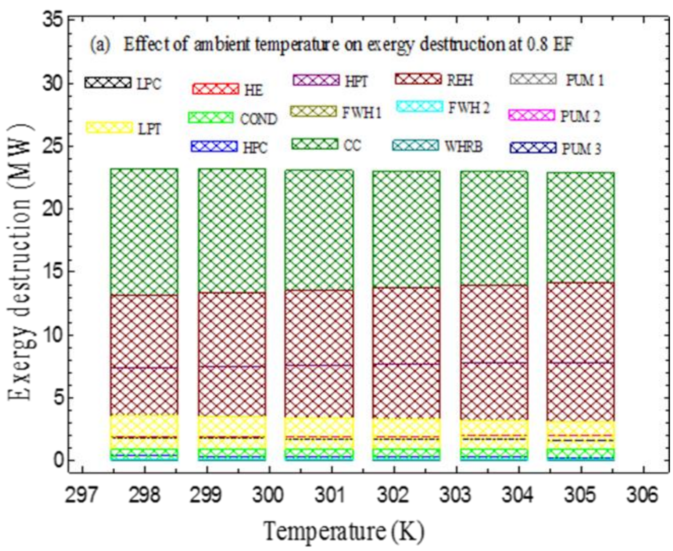


Fig. 1: Schematic of the combined cycle plant with intercooled compression at the topping cycle.



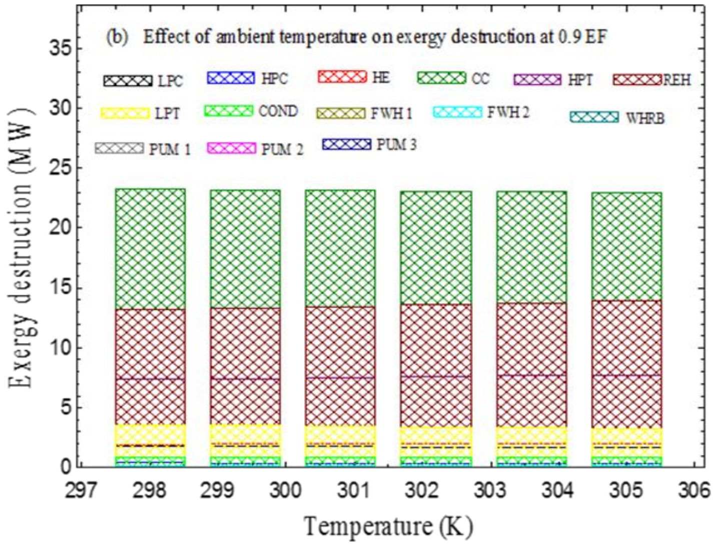


Fig. 2: Effect of ambient temperature on exergy destruction in plant components for (a) 0.8 EF and (b) 0.9 EF.

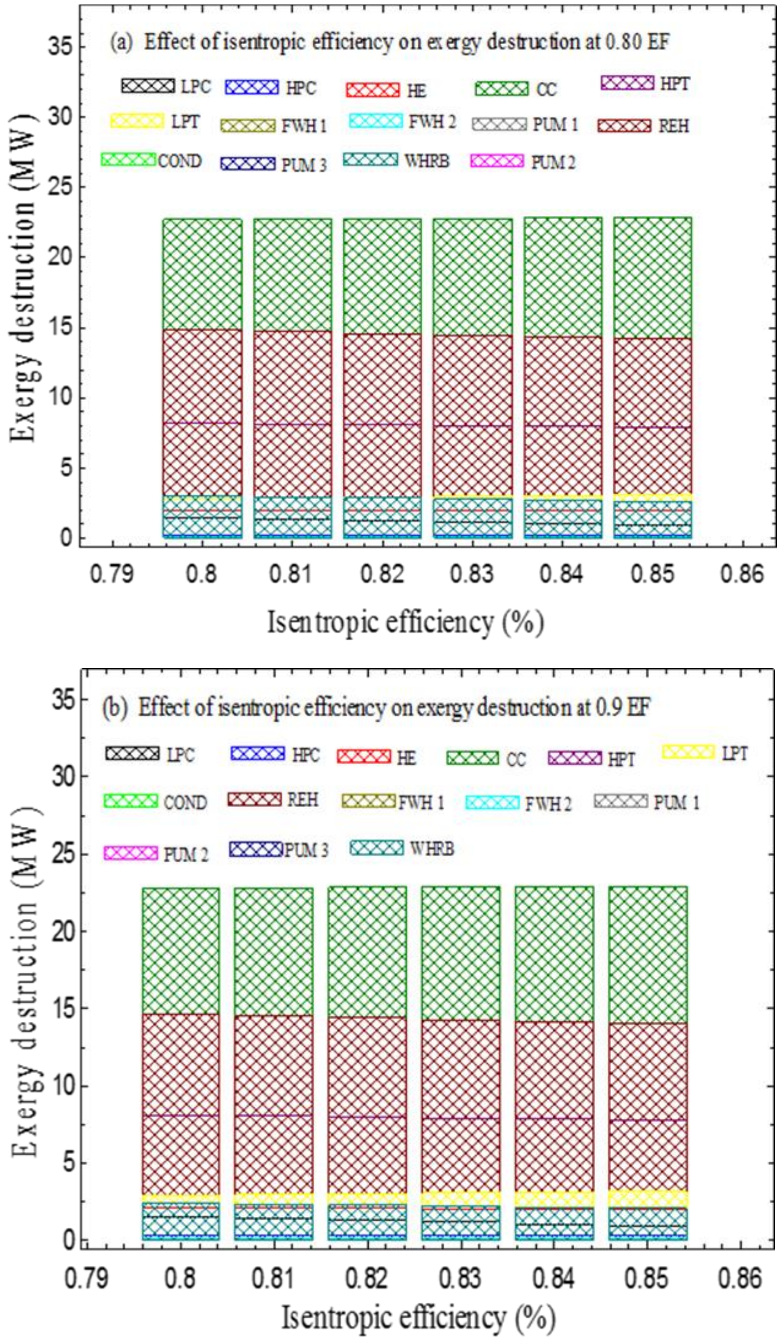


Fig. 3: Effect of isentropic efficiency on exergy destruction of plant components for (a) 0.8 EF and (b) 0.9 EF.

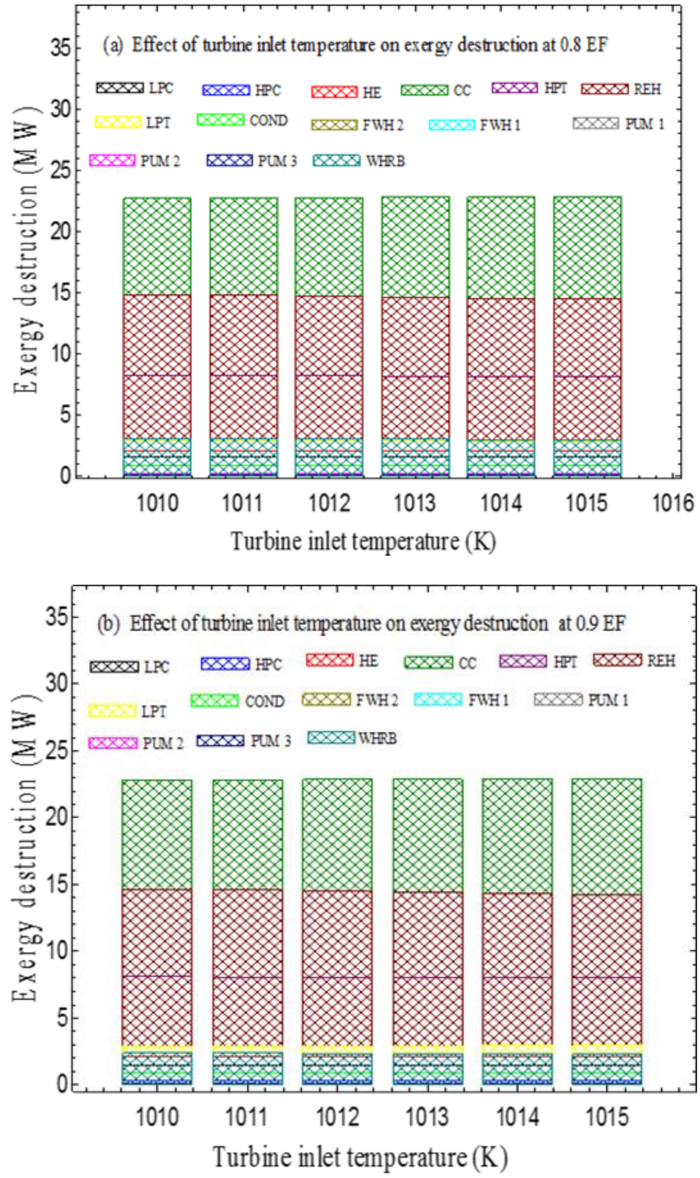
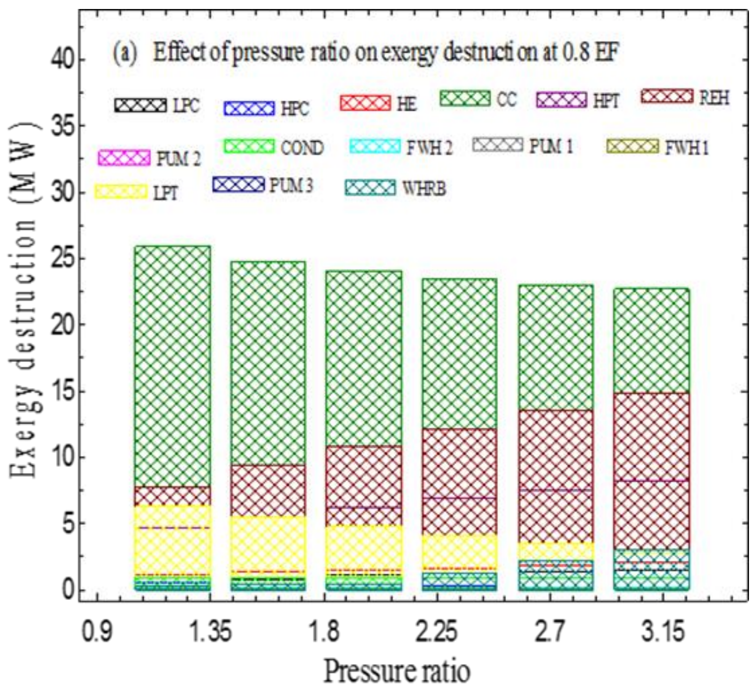


Fig. 4: Effect of turbine inlet temperature on exergy destruction of plant components for (a) 0.8 EF and (b) 0.9 EF.



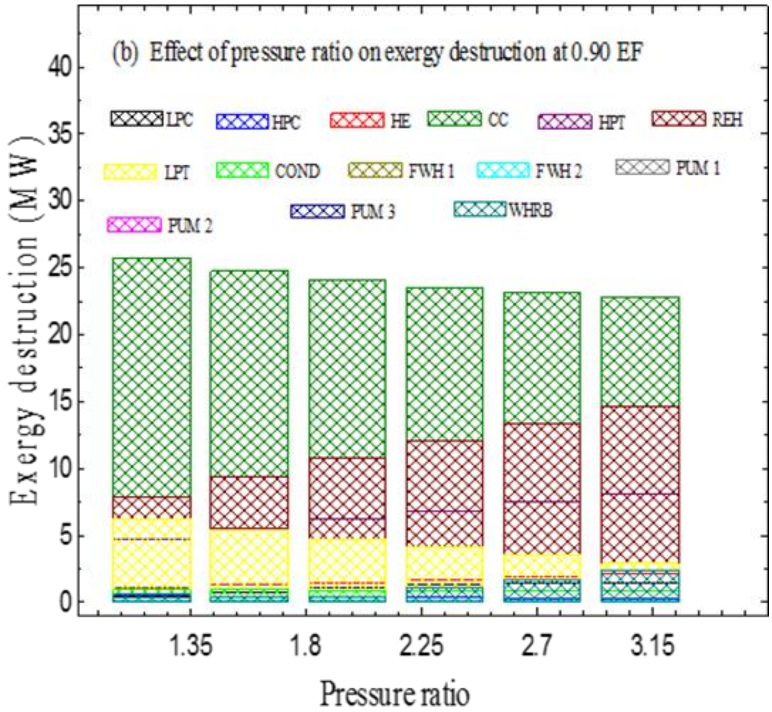
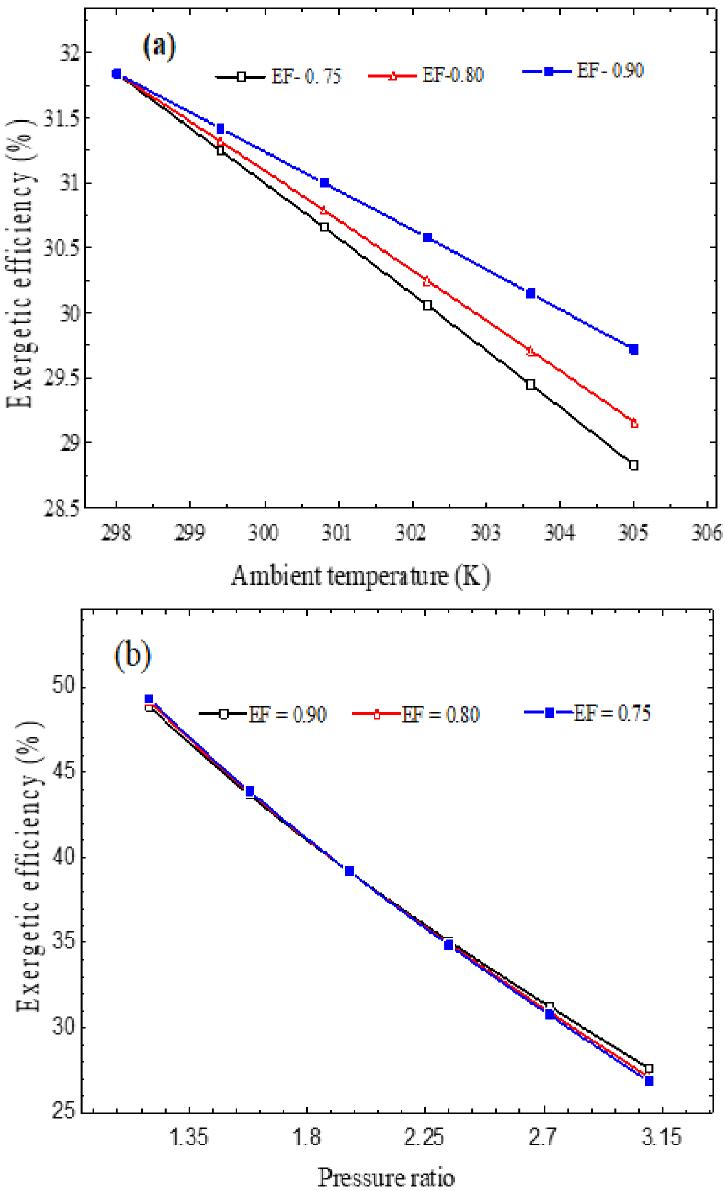


Fig 5. Effect of pressure ratio on exergy destruction of plant components for (a) 0.8 EF and (b) 0.9.



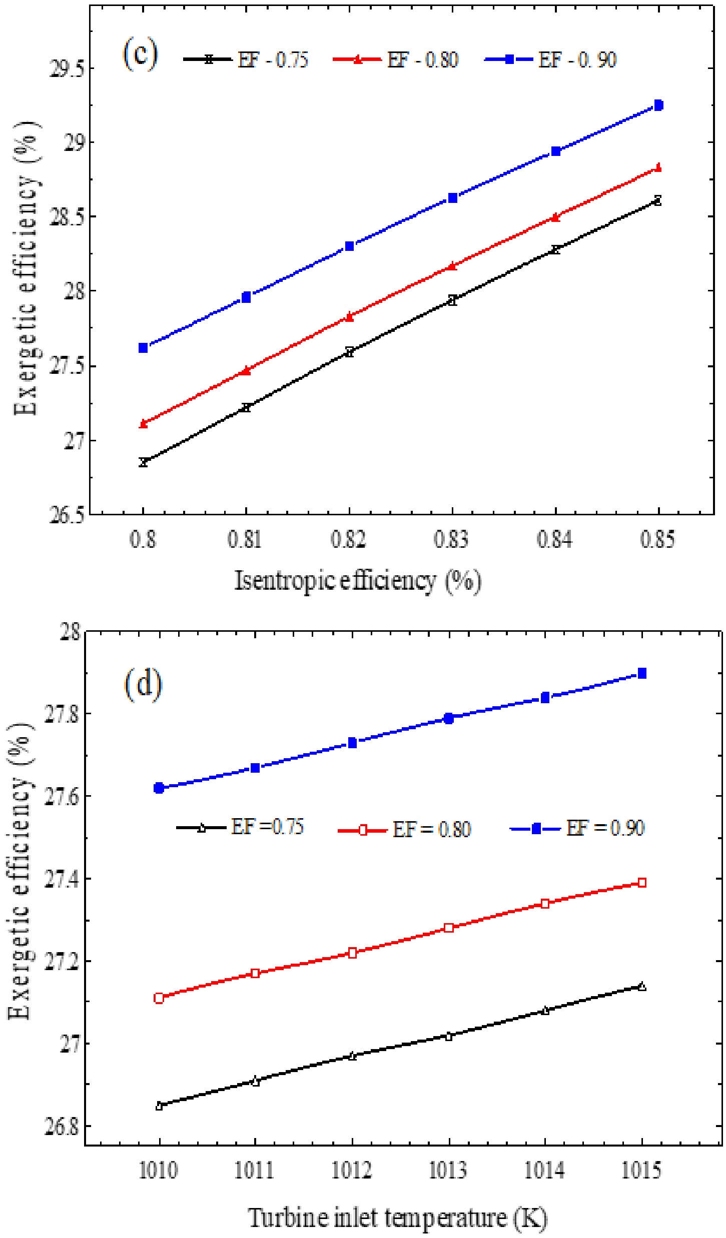
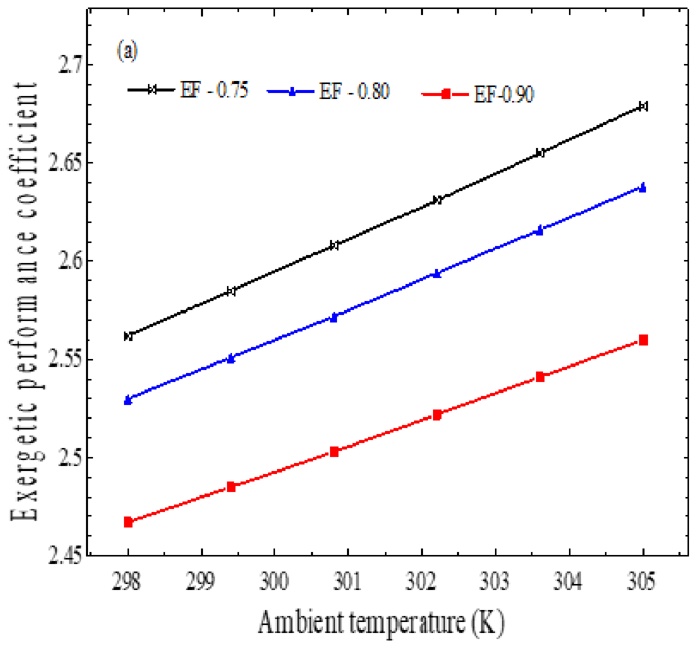


Fig. 6: Effect of operating variables (a) Ambient temperature (b) Pressure ratio (c) Isentropic efficiency (d) Turbine inlet temperature on exergetic efficiency.



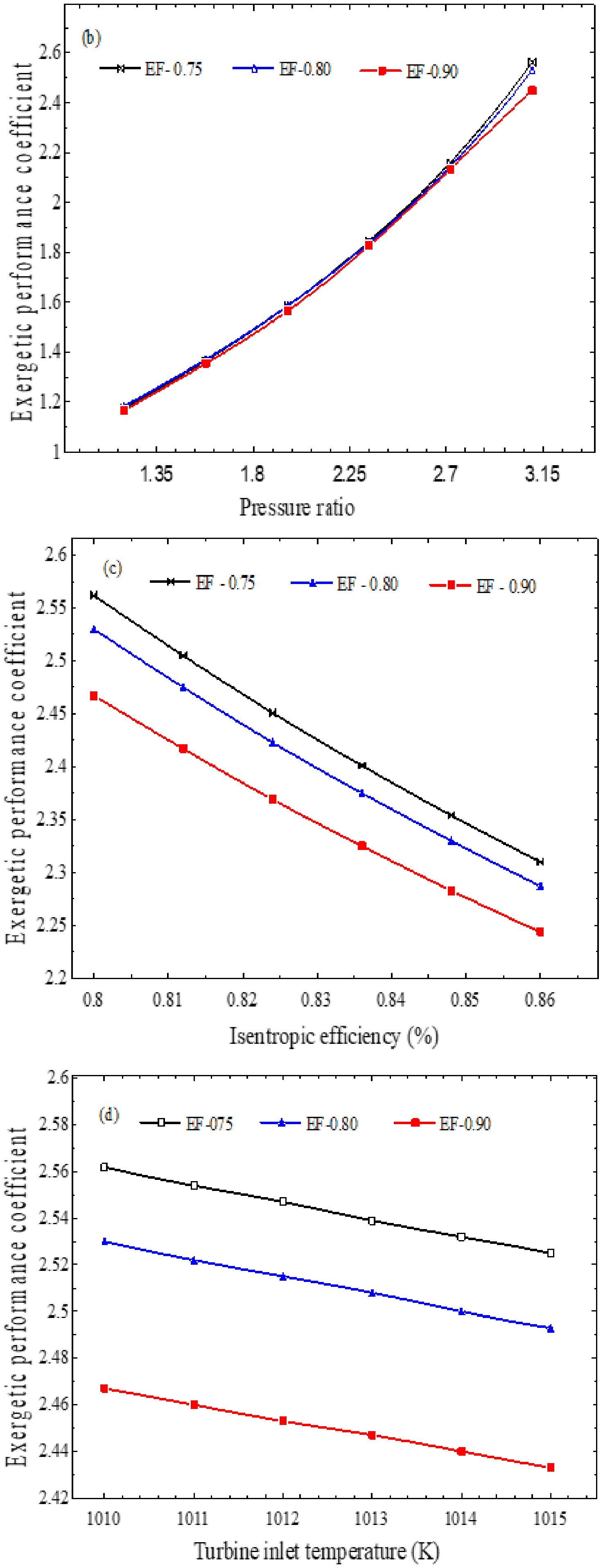


Fig. 7: Effect of operating variables (a) Ambient temperature (b) Pressure ratio (c) Isentropic efficiency (d) Turbine inlet temperature on exergetic performance coefficient.

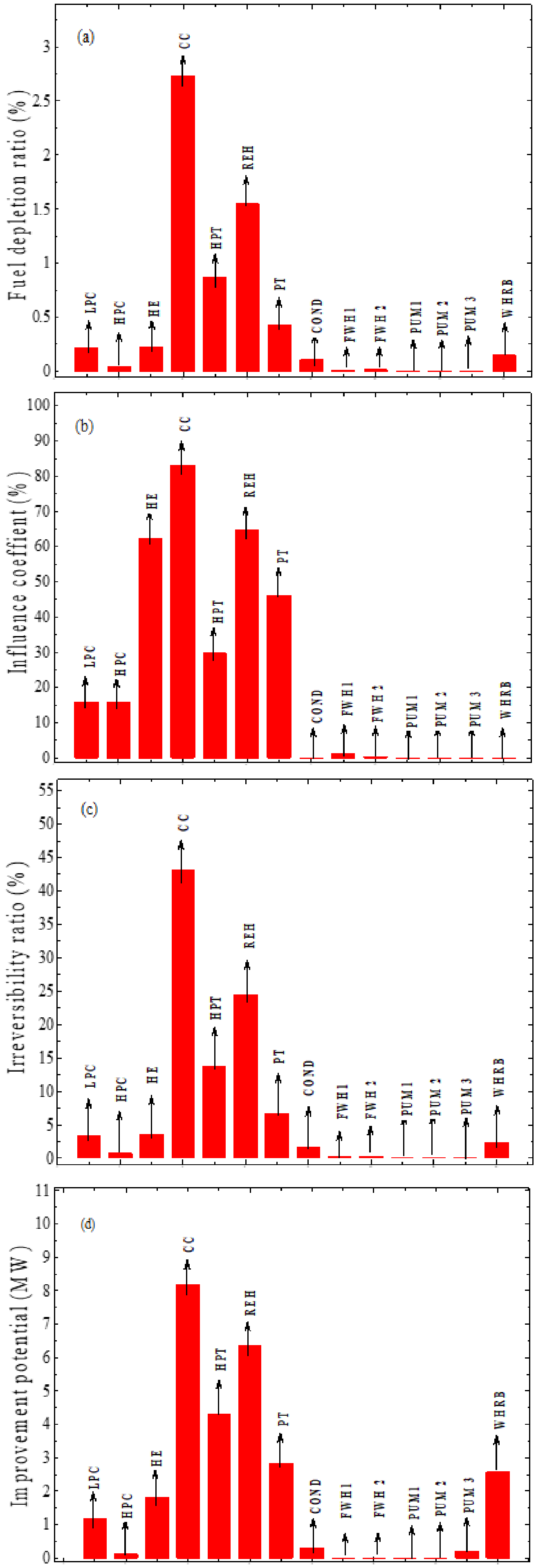


Fig. 8: Analysis of components performance at T = 298K and 0.75 EF for (a) fuel depletion ratio (b) influence coefficient (d) irreversibility ratio and (d) improvement potential.

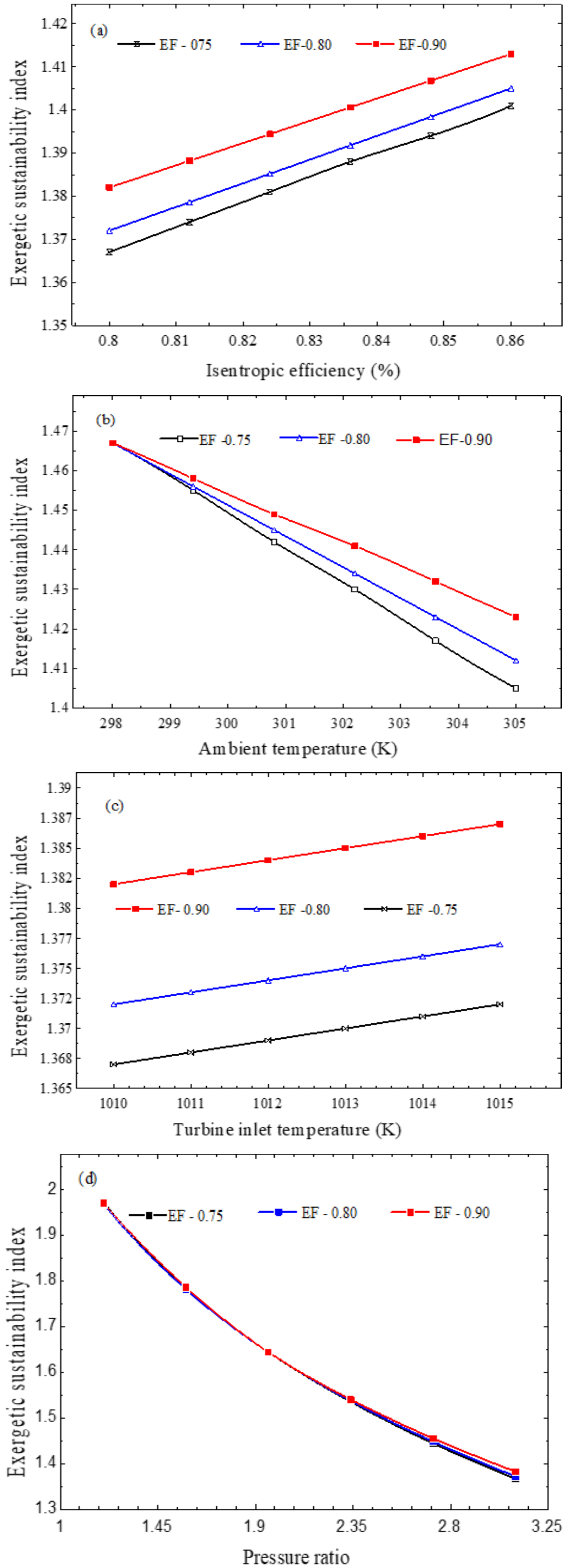


Fig. 9: Effect of operating variables (a) Ambient temperature (b) Pressure ratio (c) Isentropic efficiency (d) Turbine inlet temperature on ESI.

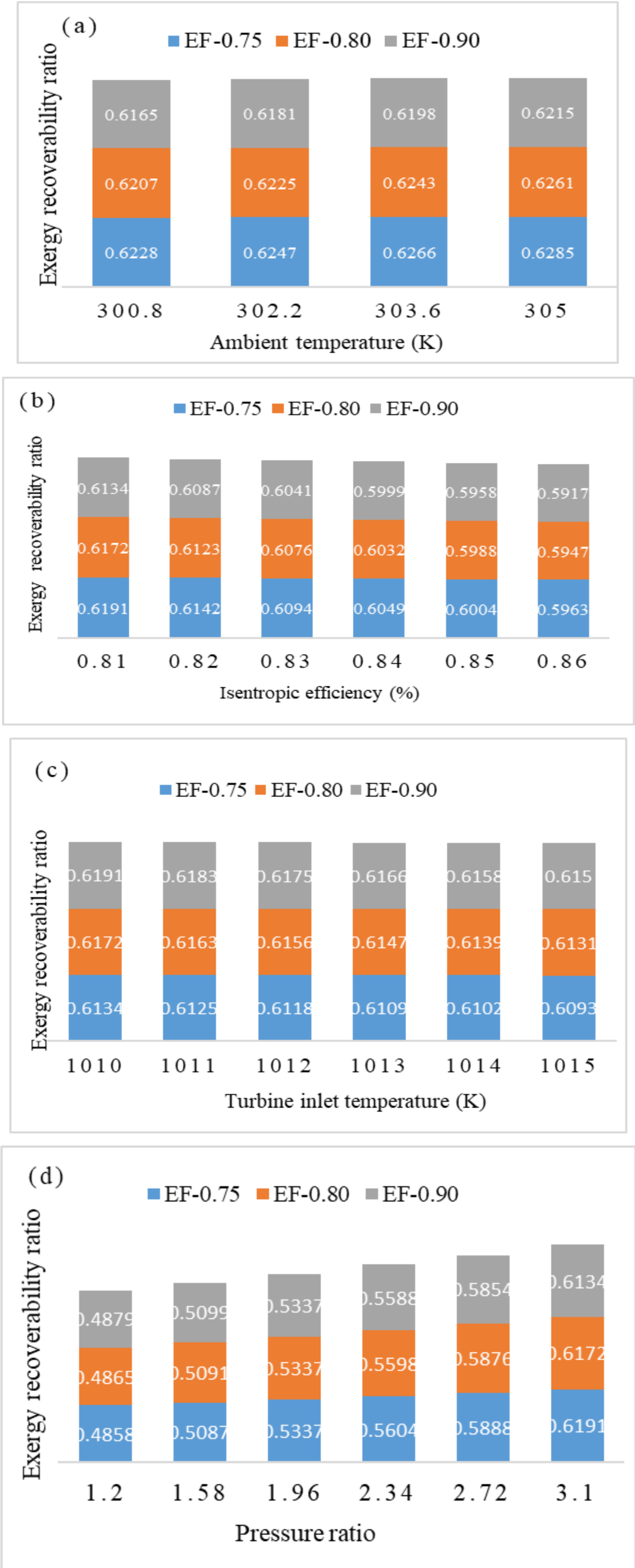


Fig 10: Effect of operating variables (a) Ambient temperature (b) Isentropic efficiency (c) turbine inlet temperature (d) pressure ratio.

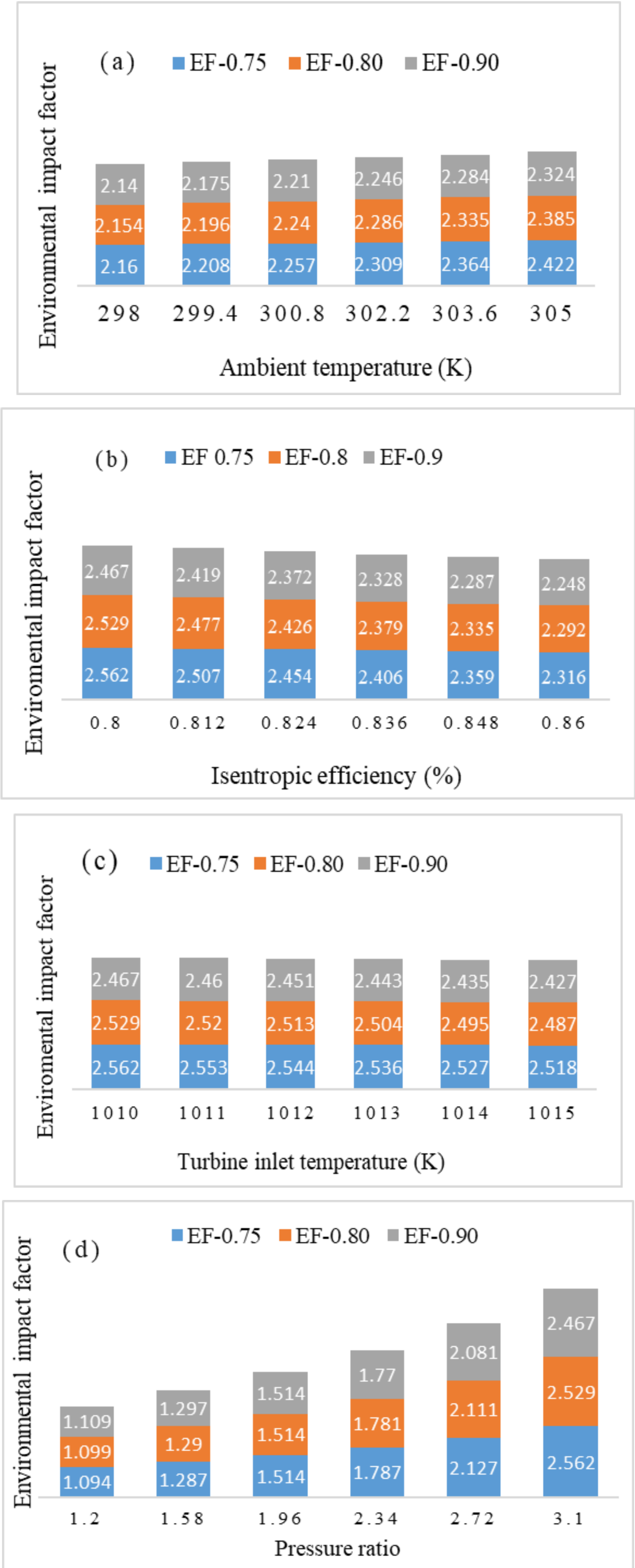
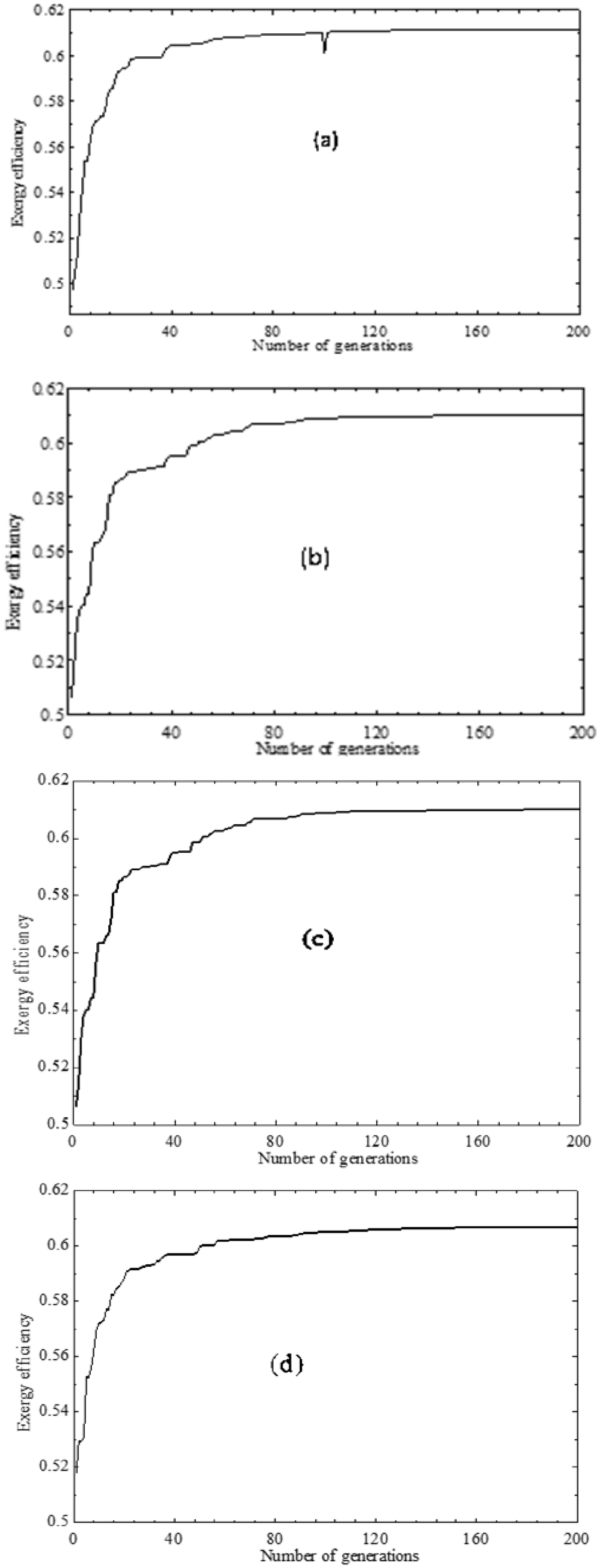


Fig 11: Effect of operating variables (a) Ambient temperature (b) Pressure ratio (c) Isentropic efficiency (d) Turbine inlet temperature on exergetic efficiency.



(a)
Exergy efficiency
Number of generations
(b)
Exergy efficiency
Number of generations
(c)
Exergy efficiency
Number of generations
(d)
Exergy efficiency
Number of generations

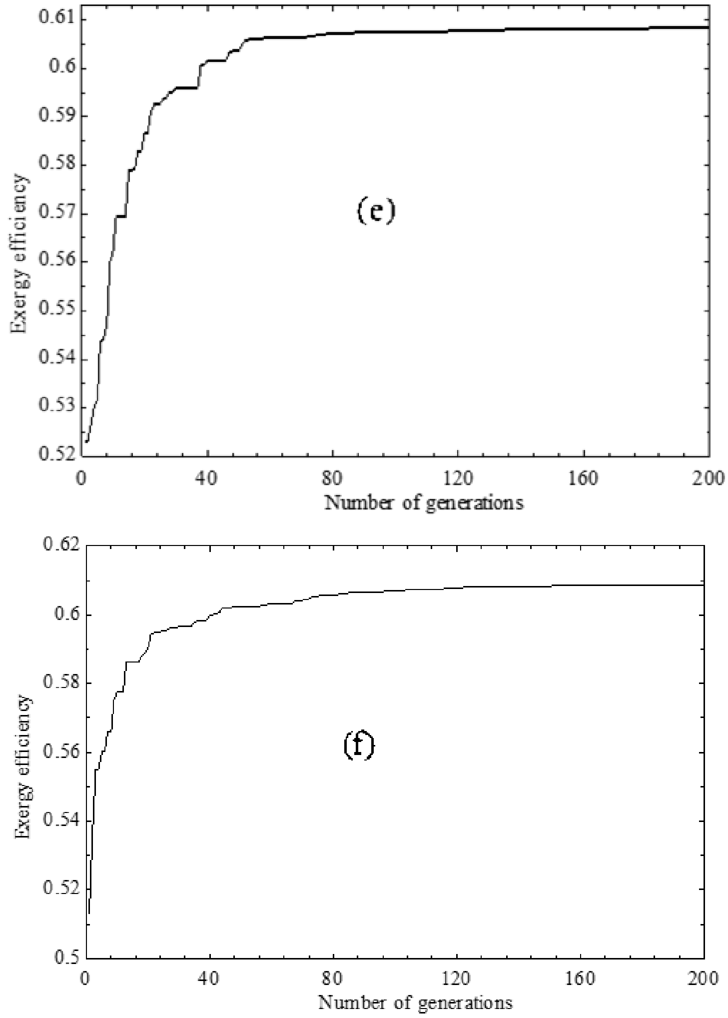


Fig. 12: Optimum values of overall exergy efficiency at (a) 0.8 EF (b) 0.82 EF (c) 0.85 EF (d) 0.87 EF (e) 0.92 EF (f) 0.95 EF.

3. Thermodynamic assumptions

All processes within the system are in steady state condition. The ambient temperature and pressure conditions are at 25^{O}C and 1.013 bars respectively. The compressor and turnine isentropic efficiencies are considered at 0.85 and 0.80 respectively. Constant pressure heat addition and 5 % pressure drop is assumed in the combustion chamber. Additionally, 95 % of the inlet temperature to the WHRB The inlet air temperature to the WHRB is approximated at 95 % of the entering steam temperature to the ST. The inlet pressures to the turbine and condenser are kept at 35 and 0.08 bars in that order whereas bled steam pressure is maintained at 5 and 1 bar for FWH_1 and FWH_2 respectively.

4. Methods and exergy modeling of the CCPP

The exergy components existing in a thermodynamic process can be described as in Equation (1) (Dincer and Cengel, 2001).

$$\dot{E}_x = \dot{E}_{xk} + \dot{E}_{xp} + \dot{E}_{xph} + \dot{E}_{xch} \quad (1)$$

Where $\dot{E}_{xk}$ is the kinetic energy, $\dot{E}_{xp}$ is potential energy, $\dot{E}_{xph}$ and $\dot{E}_{xch}$ are physical and chemical exergies respectively. Furthermore, $\dot{E}_{xk}$ and $\dot{E}_{xp}$ are neglected in this study since the changes due to elevation and speed are inconsiderable. For in-depth exergy analysis of any control surface (CS), mass and energy balances are determined to support in the calculation of the energy transfer rates in the CS. However, with the application of conservation principles and the second law of thermodynamics, expressed in Equations (2) and (3), the general exergy balance for a CS can be described as in Equation (4) (Ameri et al, 2016).

$$\sum \dot{m}_i = \sum \dot{m}_e \quad (2)$$

$$Q - W = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i \quad (3)$$

$$\dot{Ex}_Q + \sum_I \dot{m}_i ex_i = \sum_e \dot{m}_e ex_e + \dot{Ex}_W + \dot{Ex}_D \quad (4)$$

Where $e,\ i$ are subscripts describing the outlet and inlet flow of exergy streams while $\dot{Ex}_D$ represents the destroyed exergy. For gas mixture the chemical exergy is described by Equation (5) (Dincer and Cengel, 2001).

$$ex_{mix}^{ch} = [\sum_i^n X_i ex^{ch} + RT_0 \sum_i^n X_i lnX_i + G^E] \quad (5)$$

The G^E term in Equation (5) defines the Gibbs free energy which is negligible for gas mixture (Dincer and Cengel, 2001; Kanoglua et al. 2007). Nonetheless, for combustion gases, the molar component's fractions are obtained as in (Lazzaretto, 1997). Equation (5) applies accurately when calculating specific chemical exergy of fuel while, also Equation (6) may apply more easily in calculating same (Lazzaretto, 1997, Ersayin, 2015).

$$\zeta = \frac{e_{xfuel}}{LHV_{fuel}}$$

Where ζ is the ratio of chemical exergy approximated to be

1.06 (Moran, 1994). For fuel in gaseous form (C_xH_y), the succeeding experimental expression holds for calculating ζ (Ameri et al,.. 2016).

$$\zeta = 1.033 + 0.0169\frac{y}{x} - \frac{0.0698}{x} \quad (7)$$

The overall exergy efficiency for the cycle in (Fig. 1) is presented as:

$$\psi = \frac{\dot{W}_{net,GT} + \dot{W}_{net,ST}}{\dot{E}_{xf}} \quad (8)$$

Where $\dot{W}_{net,GT}$, $\dot{W}_{net,ST}$ and $\dot{E}_{xf}$ denotes the system network output for the GT topping cycle, bottoming cycles, and exergy of fuel respectively. The components exergy expressions and balancing for the system (Fig.1) are depicted in Table 1

5. Exergetic performance indicators

5.1 Influence coefficient (IFC)

The IFC (β) of a component i is described as the ratio of the actual available exergy for the component i to the overall available system exergy. Additionally, β identifies the component of the system that has influence or impact on the efficiency of the system (Safarian and Aramoun, 2015).

$$\beta_i = \frac{E_i^a}{E_{total}^a} \quad (9)$$

5.2 Exergetic performance coefficient.

The exergetic performance coefficient is also a performance criterion, which is defined as the rate of exergy per unit output power output expressed as (Abam et. al. 2012; Aljundi, 2009).

$$\gamma = \frac{\dot{E}_{D,total}}{\dot{W}_{net}} \quad (10)$$

5.3 Fuel depletion rate (FDR)

The fuel depletion ratio is the ratio of exergy consumption for a particular component to the rate of the exergy input (Aydın et al. 2013).

$$\delta_i = \frac{\dot{E}_{Di}}{\dot{E}_{fuel}} \quad (11)$$

5.4 Improvement potential

The improvement potential (IP) is an exergy performance measuring tool calculated as follows (Aydın et al. 2013).

$$IP = (1-\psi)(\dot{E}_{in} - \dot{E}_{out}) \quad (12)$$

Where ψ is exergy efficiency, $\dot{E}_{in}$ and $\dot{E}_{out}$ are the inlet and exit exergy flow rates for a component.

6. Thermo-sustainability indicators

Sustainability is the rate of energy resource supply and consumption in a way that is available and sustainable at a reasonable cost with insignificant adverse effects to the environment. Exergy investigation can define the sustainability level of real energy systems. The thermo-sustainability indicators to be considered for the adapted Brayton cycles comprise exergy efficiency, environmental impact factor, exergy recoverable ratio and exergetic sustainability index.

6.1 Environmental impact factor (EIF)

The environmental impact factor is a significant sustainability indicator calculated as the ratio of exergy waste ratio to the exergetic efficiency. The environmental impact factor specifies whether there be from all damage or not to the environment resulting from the unused waste exergy and flow of destroyed to the environment (Aydın et al. 2013; Ozgur and Hepbasli, 2013).

$$\mathrm{EIF} = \text{Waste exergy ratio/ Exergy efficiency} \quad (13)$$

5.1.2 Exergetic Sustainability Index (ESI)

Exergetic sustainability index (ESI) is an important indicator, estimated as a reciprocal of EIF. The value of SI range between $0 \leq ESI \leq 1$ (Aydın et al. 2013; Ozgur and Hepbasli, 2013; Ndukwu et al. 2015). Improved efficiency implies the exergy waste ratio, and environmental impact factor will be reduced resulting to high ESI.

5.1.3 Exergy recoverability ratio (ERR)

The ERR shows probable exergy recoverable from of a thermal system (Aydın et al. 2013). The exergy destroyed in the major components of the CCPP (Figure 1) cannot be recovered since they depend wholly on the operational characteristic and the design. The exergy destruction in the components can be minimized or reduced through improved design. However, the exergy loss to the environment can be recovered. The loss heat or exhaust waste can be utilized for heating purposes. Additionally, this process of recovery waste heat is capital intensive as it involves investment. Thus, ERR is the $\text{recoverable energy/total exergy input}$ expressed further in Equation (14). It is assumed that 90 % of the loss exergy is converted to heat from the study system (Aydın et al. 2013).

$$\mathrm{ERR} = \frac{0.9\mathrm{x}\dot{\mathrm{E}}\mathrm{x}_{\mathrm{loss,out}}}{\dot{\mathrm{E}}\mathrm{x}_{\mathrm{in,total}}}$$

7. Results and discussion

The results of the influence of thermodynamic operating variables on the exergetic performance of a combined cycle power plant with intercooled-compression at the topping cycle are presented. Table 1 presents the thermodynamic flow parameter calculated for each state point, used for further analysis. The effect of ambient temperature on the component exergy destruction (ED) for intercooler effectiveness (EF) of 0.8 and 0.9, and for ambient temperature (TA) range between $298 \leq \mathrm{TA} \leq 305\ \mathrm{K}$ is shown in Figure 2. The ED ranged between $1.57\mathrm{E}-08 \leq \mathrm{ED} \leq 23.13\ \mathrm{kW}$ for EF of 0.8 and $1.56\ \mathrm{E}-08 \leq \mathrm{ED} \leq 22.97\ \mathrm{kW}$ for EF of 0.9. Furthermore, between the TA range of $298 \leq \mathrm{TA} \leq 305\ \mathrm{K}$ and EF of $0.8 \leq \mathrm{EF} \leq 0.9$ the overall ED was noticed to have decreased by 3.2 %. Similarly, a 1.2 % decrease on the average was observed for the component system with the combustion chamber, reheat chamber and turbine dominating in ED. Figure 3 depicts the ED for varying isentropic efficiency (IE). The ED was observed to range between $1.54\mathrm{E}-08 \leq \mathrm{ED} \leq 27.71\ \mathrm{kW}$ and $1.55\mathrm{E}-08 \leq \mathrm{ED} \leq 22.94\ \mathrm{kW}$ at EF and TA ranges between $0.8 \leq \mathrm{EF} \leq 0.9$ and $298 \leq \mathrm{T} \leq 305\ \mathrm{K}$ respectively. Other thermodynamic variables for same EF include turbine inlet temperature (TIT) Figure 4 and pressure ratio, Figure 5. The results show that for the same temperature range and intercooler effectiveness, the ED dominates in the combustion chamber, reheat chamber, and the turbine. However, for TA, IE, TIT, and PR range between, $298 \leq \mathrm{TA} \leq 305\ \mathrm{K}, 0.8 \leq \mathrm{IE} \leq 0.86\ \%, 1010 \leq \mathrm{TIT} \leq 1015\ \mathrm{K},$ and $1.2 \leq \mathrm{PR} \leq 3.1$ the average overall cycle ED exist at 58.59 MW, 57. 10 MW, 57. 99 MW and 51. 79 MW respectively. It can be inferred that the variants of these thermodynamic parameters affect the cycle differently evident in the overall ED. The effect of TA affects the cycle more irrespective of the EF while PR variations resulted to less ED. This suggests that integrating a pre-cooler to theexisting system may improve air condition to ISO before compression. This may further improve the air condition through state 3 and via state 4 before entering the combustion chamber at state 5. Additionally, optimization at each value of EF can equally define the optimum operating conditions.

Figure. 6a-6d presents the effect of the operating thermodynamic variables on the exergetic efficiency of the considered CCPP. Figure 6a and 6b shows the effect of TA and PR for EF range between $0.75 \leq \mathrm{EF} \leq 0.9$. The results indicate a decreasing trend on the overall exergetic efficiency for all increasing TA and PR. Improved system efficiency is only observed at 0.9 EF by 1.42 and 2.79 % respectively. The previous corresponds to an average power increase not greater than 0.44 MW and 0.33 MW for TA and PR ranges between $298 \leq \mathrm{TA} \leq 305$ K and $1.2 \leq \mathrm{PR} \leq 3.1$ respectively. Additionally, since the system was kept at a fixed TIT, increasing the PR will customarily increase the compressor work leading to a decrease in network output. Consequently, this was responsible for the low efficiency and power output. Figure.6c and 6d represent the effect of IE and TIT on exergetic efficiency. For increase, IE and fixed PR (Figure 6c) implies a reduction in the compressor losses, which results in high network output in the CCPP. Since the network output increases for increasing IE and for fixed exergy input (fuel input), the overall exergetic efficiency will invariably increase. For varying TIT (Figure 6d) and at PR = 3.0, a reduction in exergy destruction rate of the (REH) is observed, leading to increasing power output and thus enhanced cycle efficiency.

The effect of operating variables on exergetic performance coefficient (EPC) at different intercooler EF is presented in Figure 7. The EPC is a useful exergetic performance measure defined as the exergy loss rate per unit output power. However, the system operating variables affects the EPC at different degrees. Figure 7a to 7c shows the effect of TA, PR, and IE at TIT of 1010 K and EF range between $0.75 \leq \mathrm{EF} \leq 0.9$. Low values of EPC exist at EF of 90 %. Also, at PR = 3 and IE = 0.86, Figure.7d, the EPC was found to vary from 2.31 to 2.58 for EF range between $0.75 \leq \mathrm{EF} \leq 0.9$. The lowest EPC occurred at 90 % EF (Figure 7d) while maximum values of EPC were calculated at 75 % EF. Furthermore, good performance of a thermal system is a function of high derived cycle exergetic efficiency and low EPC (Erdem et al. 2009). From the results the influence of the operating parameters on EPC at varying intercooler EF is marginal. Other measured performance criteria for the components system include the fuel depletion ratio (Figure 8a), influence coefficient (Figure 8b), irreversibility ratio (Figure.8c), and (Figure 8d) the improvement. These performance indices were estimated at 298 K and 75 % EF. The results indicate the combustion chamber in all performance considerations dominates and has a high potential for improvement followed by the REH system, HPT, and the WHRB.

The exergetic sustainability index (ESI) at different intercooler EF of the various operating variables is presented in Fig.9. The effect of isentropic efficiency (IE) (Fig. 9a), shows a steady increase in ESI at 90 % EF for all variants of IE. Nonetheless, the increase in IE for PR and TIT fixed at 3, and 1010 K (Fig.9a) indicates a continuous drop in the compressor losses consequently, leading to high network output. The latter increases the exergetic efficiency thus enhancing ESI. The results also indicate that the system is more sustainable at 90 % EF with values of ESI varying from approximately 1.38 to 1.41 at IE range between $0.80 \leq \mathrm{IE} \leq 0.86$. The converse was attained with increasing TA at fixed TIT and PR (Fig.9b). The ESI increases for increasing TIT (Fig.9c) and decreases for increasing PR. It can be inferred that since ESI is dependent on exergetic efficiency, any improvement in performance by any operating parameter will enhance ESI. Additionally, in (Fig.9d) the ESI decreases for all increasing PR and EF between $1.2 \leq \mathrm{PR} \leq 3.1$ and EF $0.75 \leq \mathrm{PR} \leq 0.90$, respectively at fixed operating conditions of TIT, TA, and IE. The reduction in ESI is attributed to the fact that for increasing PR the compressor work is increased which may result to a decreased in network output. However, this reduction in the cycle network affects the overall cycle efficiency and thus the ESI. The effect of operating variables on exergy recoverable ratio (ERR) and the environmental impact factor (EIF) are presented in Figures.10 and 11. The thermo-sustainability indicators are considered at same EF ranged between 75 and 90 %. The result shows the ERR is low at 90 % EF and high at 75 and 80 % EF for all conditions (Figure10a and 10b). However, low ERR values indicate less waste heat liberation to the environment. Similarly, for constant PR and IE (Figure 10c), the ERR was between $0.609 \leq \mathrm{ERR} \leq 0.61$, $0.613 \leq \mathrm{ERR} \leq 0.62$, $0.615 \leq \mathrm{ERR} \leq 0.619$ for EF of 75, 80 and 90 % respectively. Furthermore, in (Figure 10d) the system was maintained at constant TIT and IE, for PR between $1.2 \leq \mathrm{PR} \leq 3.1$. The ERR got from these conditions varies from 0.486 to 0.619, 0.487 to 0.617 and 0.488 to 0.613 for 75, 80 and 90 % EF in that order. The results show at increasing PR; the ERR also increases irrespective of the EF. The reason is ascribed to increase compressor work which has led to a reduction in exergetic efficiency. The environmental impact factor (EIF) for all the scenarios is presented in (Figure 11a to 11d). Additionally, the EIF follows the same trend as ERR with marked improvements at increasing EF. Similarly, they exist variations in EIF even at same EF for varying operating conditions. This condition suggest that optimum operating conditions are possible for each EF.

7.1 Optimum parameters

Results for optimum operating parameters at best exergetic efficiency are shown for different intercooler effectiveness in the CCPP system. The optimum values were obtained using genetic algorithm (GA). The work output of the LPC/HPC, HPT/PT, ST and the total exergy input (chemical exergy) were preferred as the objective function presented in Equation (14). Additionally, the isentropic efficiencies, turbine inlet temperature, pressure ratio are the decision variables (DV), expressed in Equations (18) to (20). The DVs are selected based on commercial availability and metallurgical temperature limits. However, the optimum values of exergy efficiency obtained at 0.8 EF, 0.82 EF, 0.85 EF, 0.87 EF, and 0.92 EF and 0.95 EF are ranged between 60.05 and 60.87 % as depicted in Figure 12a to 12f. The result of ESI at optimum was not greater than 2.56 compared with an average value of 1.2 for normal scenario. The optimum operating decision variables obtained at the optimum EFs are depicted in Table 3.

$$\psi = \frac{m_{f,1}c_{p,g}(T_7-T_8)+m_{f,2}c_{p,g}(T_{10}-T_{11})+W_{ST}-m_{air}c_{p,air}(T_4-T_3)-m_{air}c_{p,air}(T_2-T_1)}{E^{ch}_{xtotal}} \quad (14)$$

Where:

E^{ch}_{xtotal} = total exergy (chemical exergy)

$$T_2 = T_1\left(1 + 1/\eta_{LPC}\left[r_{p,LPC}^{\frac{\gamma-1}{\gamma}} - 1\right]\right) \quad (15)$$

$$T_4 = T_3(1 + 1/\eta_{HPC}(r_{p,HPC}^{\frac{\gamma-1}{\gamma}} - 1)) \quad (16)$$

$$T_8 = T_7\left[1 + \eta_{HPT}\left(\frac{1}{r_{p,HPT}^{\frac{\gamma-1}{\gamma}}} - 1\right)\right];\ T_{11} = T_{10}\left[1 + \eta_{LPT}\left(\frac{1}{r_{p,LPT}^{\frac{\gamma-1}{\gamma}}} - 1\right)\right] \quad (17)$$

Subject to the following constraints:

$0.79 \le \eta_{HPT} \le 0.89; 0.79 \le \eta_{HPC} \le 0.89, 0.79 \le \eta_{LPT} \le 0.89; 0.79 \le \eta_{LPC} \le 0.89$ (18)

$1.6 \le r_{pLPT} \le 1.9; 5.7 \le r_{pHPT} \le 5.9; 2.8 \le r_{pLPC} \le 3.4; 2.8 \le r_{pHPC} \le 3.4$ (19)

$288 \le T_1 \le 310; 350 \le T_3 \le 410; 1000 \le T_7 \le 1250; 1000 \le T_{10} \le 1250$ (20)

Also, $T_1 = T_A$ = Ambient temperature, T_7, T_{10} = TIT for LPT and PT, PR = r_p = compressor pressure ratio and IE = η = isentropic efficiency.

8. Conclusions

The results of the influence of operating variables on the exergetic performance of a combined cycle natural gas-powered plant with intercooled compression at the topping cycle are presented. The findings of the study are:

- The effect of TA on the component exergy destruction (ED) for intercooler effectiveness (EF) of 0.8 and 0.9, and for TA range between $298 \le TA \le 305$ K varied between $1.57E-08 \le ED \le 23.13$ kW and $1.56\ E-08 \le ED \le 22.97$ kW respectively.
- Furthermore, for TA and EF range between $298 \le TA \le 305$ K and $0.8 \le EF \le 0.9$ a 1.2 % averaged decrease in components ED was observed while the combustion chamber, reheat chamber, and turbine dominates in ED for all operating variables.
- For TA, IE, TIT, and PR range between, $298 \le TA \le 305$ K, $0.8 \le IE \le 0.86$ %, $1010 \le TIT \le 1015$ K, and $1.2 \le PR \le 3.1$ the average overall cycle ED occur at 58.59 MW, 57. 10 MW, 57. 99 MW and 51. 79 MW between 0.75 and 0.9 EF respectively.
- They exist a decreasing trend on the overall exergetic efficiency for all increasing TA and PR with improved system efficiency at 90 % EF by 1.42 and 2.79 % in that order. The latter corresponds to a power increase not greater than 0.44 MW and 0.33 MW for TA and PR ranges between $298 \le TA \le 305$ K and $1.2 \le PR \le 3.1$ respectively.
- The exergetic sustainability index was not less than 1.2 while the lowest exergy recoverability ratio (ERR) and environmental impact factor (EIF) fluctuated between $0.486 \le ERR \le 0.629$ and $1.099 \le EIF \le 2.560$ respectively for same EF range and operating variables. The study shows that all performance indicators improved slightly with increasing EF.
- Optimum overall exergetic efficiencies obtained were ranged between 60.05 and 60.87 % for intercooler values of 0.82 EF, 0.85 EF, 0.87 EF, 0.92 EF and 0.95 EF. The average value of ESI at optimum was not greater than 2.56 compared with an average value of 1.2 for normal scenario.

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