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Performance of a combined cycle power plant due to auxiliary heating from the combustion chamber of the gas turbine topping cycle

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Abstract Energy demand is increasing exponentially in the last decade. To meet such demand there is an urgent need to enhance the power generation capacity of the electrical power generation system worldwide. A combined-cycle gas turbines power plant is an alternative to replace the existing steam/gas electric power plants. The present study is an attempt to investigate the effect of different parameters to optimize the performance of the combined cycle power plant. The input physical parameters such as pressure ratio, air fuel ratio and a fraction of combustible product to heat recovery heat exchanger *via* gas turbine were varied to determine the work output, thermal efficiency, and exergy destruction. The result of the present study shows that for maximum work output, thermal efficiency as well as total exergy destruction, extraction of combustible gases from the passage of the combustion chamber and gas turbine for heat recovery steam generator is not favorable. Work output and thermal efficiency increase with an increase in pressure ratio and decrease in air fuel ratio but for minimum total exergy destruction, the pressure ratio should be minimum and air fuel ratio should be maximum.

Keywords: Pressure ratio; Air-Fuel ratio; Supplement heating; Exergy analysis; Energy analysis

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Nomenclature

A/F	–	air-fuel ratio
c_{pa}	–	specific heat, J/kgK
h	–	specific enthalpy, kJ/kg
I	–	exergy destruction, kW
k	–	specific heat ratio
LCV	–	lower calorific value of fuel, kJ/kg
$\dot{m}$	–	mass flow rate, kg/s
P	–	pressure, Pa
r_p	–	pressure ratio
R	–	gas constant, kJ/kgK
s	–	specific entropy, kJ/kgK
T	–	thermodynamic temperature, K
W	–	work rate, kW
z	–	partial amount of the combustible gases
η	–	efficiency

Subscripts and superscripts

a	–	air
c	–	air compressor
cc	–	combustion chamber
$comb.$	–	combined cycle
f	–	fuel
g	–	gasses
net	–	net
pc	–	primary cycle
sc	–	secondary cycle
t	–	gas turbine
th	–	thermal

Abbreviations

ABC	–	air bottoming cycle
EGT	–	evaporating gas turbine
EES	–	engineering equation solver
HRSG	–	heat recovery steam generator
STIG	–	steam injection gas turbine
TET	–	turbine exhaust temperature
TIT	–	turbine inlet temperature

1 Introduction

Energy is the elementary demand of human needs, and driving force of civilization, so it's efficient utilization at any level is the mandatory condition. Energy conversion systems are required to modify towards high-efficiency energy conversion systems because of the continuous increase in fuel prices and depletion of fossil fuel resources. Modified energy conversion systems can recover energy from the exhaust gases to the maximum extent. Wastage of a large amount of heat from the exhaust gases from simple gas turbine plants results in not only a decrease in the plant efficiency and work power output but also increases global warming and air pollution [1,2]. By utilizing the waste energy of exhaust gases from the simple gas turbine to operate another cycle, which, maybe Rankine or Brayton cycle, the plant thermal efficiency can be improved. Thus the combined cycle concept came into existence. The significance of the combined power plant is that it utilizes the waste energy for the generation of power through the bottoming cycle. The performance of the combined cycle depends upon the number of parameters like turbine inlet temperature (TIT), component efficiency, turbine exhaust temperature (TET), degree of supplementary heating, and condition of steam generation. Power generating by utilizing the heat of exhaust gases of the gas turbine is the basic principle of the combined-cycle power plant. Major factors available in the literature that affect the performance of the combined-cycle power plant are the ratio of cycle peak pressure to minimum pressure [3–6], inlet temperature of the gas turbine [7–12], and ambient conditions [13–16]. These factors not only affect the energy performance but also play a vital role in the exergy destruction of the cycle main components as well as a complete cycle. Apart from these, many other factors affect the performance of a combined cycle performance. Khan *et al.* [17] investigated the importance of the bypass valve that controls the path of exhaust gases through the heat exchangers in a combined cycle on its work output, thermal efficiency, and exergy destruction. Authors prove that the thermal efficiency of the combined cycle plant possibly improved to 4–15% at TIT =1000 K and 15–31% at TIT = 1400 K by proper use of the bypass valve. Ghazikhani *et al.* [18] investigated the energy and exergy analysis of the air bottoming cycle (ABC). Also, in the air bottoming cycle, at lower specific air consumption, higher specific work can be achieved at a small value of pressure ratio as compared to a simple gas turbine cycle. Results also show that due to energy recovery from the exhaust gases in ABC, fuel exergy increases, and because of more components in ABC as compared to the simple gas turbine cycle, the total exergy destruction increases. Costea *et al.* [19] investigated gas turbine cogeneration systems of three different configurations. The result shows that the optimum performance of each cycle was obtained at the same corresponding optimal compression ratio under the same rate of fuel supply to the combustion chamber. The conditions for maximum exergetic efficiency were not the same as the conditions for the optimum performance of the cycles. Khan *et al.* [20] proposed and investigated the energy and exergy analysis of five different configurations of the combined cycle plant. Result proves that the performance of the combined cycle is significantly affected by the configuration achieved by combination of plant components. Bataineh and Khaleel [21] performed energetic and exergetic analyses of real combined cycle power plant situated in Jordan according to the climatic conditions. The results show that the maximum exergy destruction occurs in the combustion chamber equal to 73% at 340°C followed by the destruction in heat exchanger [21]. Two cycles were proposed by Ghazikhani *et al.* [22] for improving the performance of the air bottoming cycle. The steam injection gas turbine (STIG-ABC), and the other one was the evaporating gas turbine (EGT-ABC). It was observed that EGT-ABC shows better results as compared to STIG-ABC in terms of output and irreversibility losses. The performance of combined cycle gas turbine power plants has been improved significantly with supplementary firing [23–26]. Arora *et al.* [27] investigated the effect of single pressure combined cycle with supplementary heating. The performance of the combined cycle was analyzed by varying the pressure ratio of the topping cycle from 4 to 20, air-fuel ratio of 50, 55, 60, and 65, and supplementary heating from 0.1 to 0.5. The result shows that the combined cycle performance is optimized at lower air fuel ratio and lower supplementary heating and at a higher value of pressure ratio. Khan investigated the performance of air bottoming combined cycle and regenerative gas turbine cycle operated by the partial amount of exhaust gasses from gas turbine. This study presented the unique technique to compare the performance of these two cycles and proved that for thermal efficiency and exhaust gasses exergy loss by regenerative gas turbine cycle is much better as compared to air bottoming cycle but for net power output air bottoming cycle is better than regenerative gas turbine cycle. The present study emphasizes the possible methods of additional heating to enhance the performance of the combined cycle power plant.

According to the available literature, to boost the performance of the combined cycle power plant, various systems have been used. However, there is no direct study on the simultaneous effect of varying the mass fraction of combustible gases from the combustion chamber of the topping cycle to the heat exchanger of the bottoming cycle on the performance of topping, bottoming, and combined cycle. The present study is the energetic and exergetic parametric analysis of the mass fraction of combustible gases from the combustion chamber of the topping cycle to the heat exchanger of the bottoming cycle on the performance of topping, bottoming, and combined cycle. The energetic and exergetic parametric analysis results are present graphically.

2 Thermodynamic analysis of combined cycle

The performance of the combined cycle measured in terms of thermal efficiency and specific work output depends upon the number of parameters. If all the parameters were taken into account in calculating specific work output and thermal efficiency, it would be a very tedious job.

The present study is based on the assumptions such that the change of kinetic energy and potential energy of the working fluid between the inlet and outlet of each component is negligible. There is no pressure loss in inlet ducting, combustion chamber, exhaust ducting, and duct connecting the components,. Ambient conditions are pressure at 100 kPa and temperature at 300 K. Isentropic efficiency of compressor and turbine have been fixed at 85%, maximum temperature permissible in the gas turbine is 1500 K, the fuel is having the lower calorific value of LCV= 43963.5 kJ/kg. Heat recovery steam generated pressure and condenser pressure is 2000 kPa and 5 kPa, respectively. The temperature of steam leaving the heat recovery steam generator (HRSG) is 813 K. The variables are pressure ratio (r_p) which has been taken to vary in the range from 4 to 30, and air fuel ratio (A/F, mass of air / mass of fuel) from 50 to 100. The approach uses to investigate the energy and exergy analysis of present proposed system described in the different textbooks and research papers [28–32].

3 Analysis of primary cycle

Air compressor work is as follows:

$$\left(\dot{W}_c\right)_{pc} = \dot{m}_a c_{pa} T_1 \left(\frac{r_p^{k_a} - 1}{\eta_c}\right) , \tag{1}$$

where specific heat of air as the function of temperature is given by [33]

$$c_{pa} = 1.04841 - \left(\frac{3.8371}{10^4}\right) T + \left(\frac{9.4517}{10^7}\right) T^2 - \left(\frac{5.4903}{10^{10}}\right) T^3 + \left(\frac{7.9298}{10^{14}}\right) T^4 \tag{2}$$

and

$$k_a = \frac{\gamma_a - 1}{\gamma_a}\,. \tag{3}$$

Air compressor exergy destruction due to irreversibility is given by

$$(I_c)_{pc} = \dot{m}_a T_1 \left(s_2 - s_1\right) = \dot{m}_a T_{1pa} \ln \frac{T_2}{T_1} - R_a \ln r_p\,. \tag{4}$$

Heat balance of combustion chamber is given by

$$\dot{m}_a h_2 + \eta_{cc} \dot{m}_f \mathrm{LCV} = \dot{m}_g h_3\,, \tag{5}$$

and

$$T_3 = \frac{\mathrm{LCV} - \left(\dfrac{A}{F}\right) c_{pa} T_2}{\left(1 + \dfrac{A}{F}\right) c_{pg}}\,, \tag{6}$$

hence combustion chamber exergy destruction due to irreversibility is given by

$$\begin{aligned}(I_{cc})_{pc} = T_1 &\left[\dot{m}_g \left(c_{pg} \ln \frac{T_3}{T_1} - R_g \ln \frac{P_3}{P_1}\right) - \dot{m}_a \left(c_{pa} \ln \frac{T_2}{T_1} - R_a \ln \frac{P_2}{P_1}\right)\right] \\ &+ \left(\psi - 1\right) \dot{m}_f (\mathrm{LCV})_{T_1}\,. \end{aligned} \tag{7}$$

Turbine work of primary cycle is given by

$$\left(\dot{W}_t\right)_{pc} = \left(1 - z_a\right) \dot{m}_g c_{pg} T_3 \eta_t \left(1 - r_p^{-k_g}\right), \tag{8}$$

where

$$c_{pg} = 0.991615 + \left(\frac{6.9970}{10^5}\right) T + \left(\frac{2.7129}{10^7}\right) T^2 - \left(\frac{1.2244}{10^{10}}\right) T^3, \tag{9}$$

$$k_g = \frac{\gamma_g - 1}{\gamma_g}\,, \tag{10}$$

and

$$T_4 = T_3 \left[1 - \eta_t \left(1 - r_p^{-k_g}\right)\right]. \tag{11}$$

Gas turbine exergy destruction due to irreversibility is is given by

$$\begin{aligned}(I_t)_{pc} &= (1 - z) \dot{m}_g T_1 \left(s_4 - s_3\right) \\ &= (1 - z) \dot{m}_g T_1 \left(c_{pg} \ln \frac{T_4}{T_3} - R_g \ln \frac{1}{r_p}\right)\,. \end{aligned} \tag{12}$$

Primary cycle work output is given by

$$\left(\dot{W}_{net}\right)_{pc} = \left(\dot{W}_t\right)_{pc} - \left(\dot{W}_c\right)_{pc} . \tag{13}$$

Thus thermal efficiency of primary cycle is given by

$$(\eta_{th})_{pc} = \frac{\left(\dot{W}_{net}\right)_{pc}}{\dot{m}_f \mathrm{LCV}} . \tag{14}$$

4 Analysis of secondary cycle

Temperature of combustible gases entering the heat recovery steam generator (HRSG) is as follows:

$$T_x = zT_3 + (1-z)T_4 . \tag{15}$$

Rate of steam generated is given by

$$\dot{m}_s = \frac{\dot{m}_g c_{pg} \left(T_x - T_5\right)}{h_6 - h_9} . \tag{16}$$

Exergy destruction of heat recovery steam generator due to irreversibility is given by

$$I_{\mathrm{HRSG}} = T_1 \left[\dot{m}_s \left(s_6 - s_9\right) + m_g c_{pg} \ln \frac{T_5}{T_x} - m_g R_g \ln \frac{P_5}{P_x}\right] . \tag{17}$$

Turbine work of the secondary cycle is given by

$$\left(\dot{W}_t\right)_{sc} = \dot{m}_s \left(h_6 - h_7\right) , \tag{18}$$

pump work of primary cycle is given by

$$\left(\dot{W}_{pump}\right)_{sc} = \dot{m}_s \left(h_9 - h_8\right) . \tag{19}$$

and secondary cycle work output is

$$\left(\dot{W}_{net}\right)_{sc} = \left(\dot{W}_t\right)_{sc} - \left(\dot{W}_{pump}\right)_{sc} . \tag{20}$$

Then thermal efficiency of the combined cycle

$$(\eta_{th})_{sc} = \frac{\left(\dot{W}_{net}\right)_{sc}}{\dot{m}_f \mathrm{LCV}} . \tag{21}$$

5 Analysis of combined cycle

Combined cycle output is given by

$$\left(\dot{W}_{net}\right)_{comb} = \left(\dot{W}_{net}\right)_{p} + \left(\dot{W}_{net}\right)_{s} . \tag{22}$$

Thermal efficiency of the combined cycle is given by

$$(\eta_{th})_{comb} = \frac{\left(\dot{W}_{net}\right)_{comb}}{\dot{m}_f \mathrm{LCV}} . \tag{23}$$

6 Solution technique

The thermal efficiency and work output of topping, bottoming, and combined cycle, as well as exhaust gasses exergy loss, have been analyzed using the commercial software Engineering Equation Solver (EES) [34] for the configuration under investigation. The mass fraction of the combustible product, the pressure ratio of the topping cycle, and the air-fuel ratio are the functional variables for analyzing the dependent parameters described by the analytical equations

7 Result and discussion

In Fig. 1, the partial amount (z) of the combustible gases from the combustion chamber goes to the heat recovery steam generator to heat water for the steam formation, and the remaining amount of combustible gases $(1-z)$ enters the primary cycle gas turbine. When $z = 0$, the whole amount of combustible gases enters the gas turbine, whereas when $z = 1$, the full amount of combustible gases goes to the heat recovery steam generator but, under this condition, the turbine work of the primary cycle is zero which, results in a negative work output from the primary cycle. So the minimum value of z should be such that the work output in the primary cycle is positive or zero.

Figure 2a shows the variation of primary cycle work output with respect to partial amount (z) of the combustible gases at selected pressure ratio (r_p) as well as air-fuel ratio (A/F) of the primary cycle. It can be observed that the work-output from the primary cycle decreases with z and maximum at $z = 0$ in all considered cases of pressure ratio and air-fuel ratio. The reason for decrease in work output of the primary cycle with increase of z

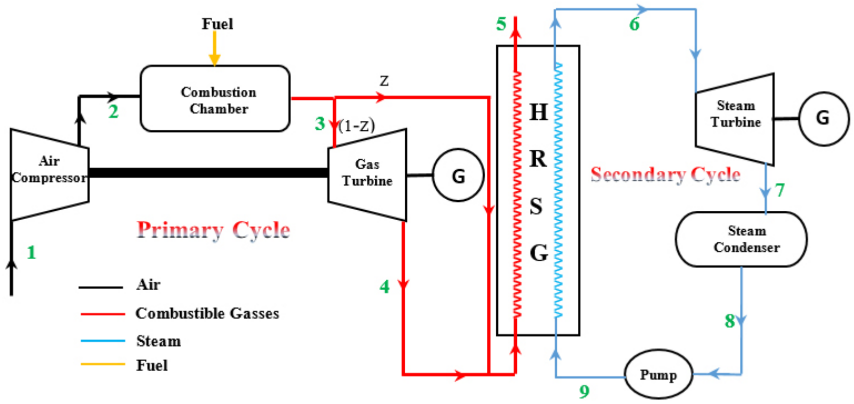
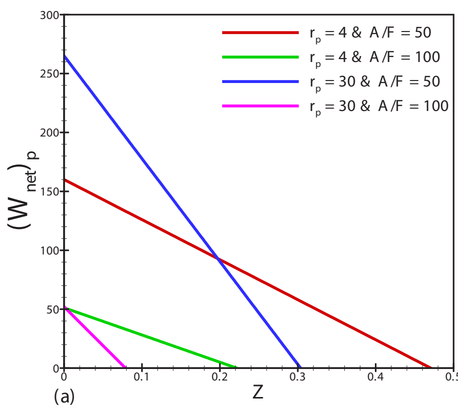


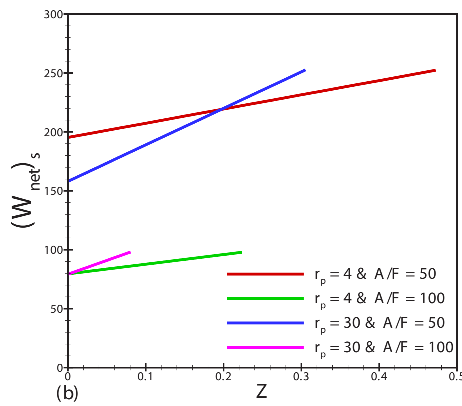
Figure 1: Schematic diagram of proposed cycle under study: G – electric generator, HRSG – heat recovery steam generator, Z – partial amount of the combustible gases.

is that as the value of z increases the mass of combustible gases to the gas turbine which result in decrease in work output of primary cycle according to Eqs. (8) and (13). At $z = 0$, the work output of the primary cycle at $r_p = 4$ and 30 for $A/F = 100$ is the same. At $z \approx 0.2$, the primary cycle work output at $r_p = 4$ and 30 for $A/F = 50$ is the same. At $z = 0$, the maximum work output from the primary cycle which is at $r_p = 30$ for $A/F = 50$ is 5.17 times more than that of minimum work output from the primary cycle which is at $r_p = 4$ for $A/F = 100$. Also, at $z = 0$, the primary cycle work output at $A/F = 50$ is 220% more than the primary cycle work output at $A/F = 100$ for $r_p = 4$, while the primary cycle work output at $A/F = 50$ is 430% more than the primary cycle work output at $A/F = 100$ for $r_p = 30$. For $0 \leq z \leq 0.2$, the primary cycle work output at $r_p = 30$ is more than the primary cycle work output at $r_p = 4$ for $A/F = 50$. It also observed that for the entire considered range of pressure ratio and air-fuel ratio, the minimum value of z at which the primary cycle work output is either zero or positive at $z = 0.0788$. Figure 2b shows the variation of secondary cycle work output with respect to z for considered cases of pressure ratio as well as the air-fuel ratio of the primary cycle. From the figure, it is clear that the work output of the secondary cycle increases with z. The reason for the increase in work output of the secondary cycle with z is that the value of the combustible product temperature entering the heat recovery steam generator and the rate of steam generated increases with z, according to Eqs. (15) and (16). The secondary cycle work output is the same at $r_p = 4$

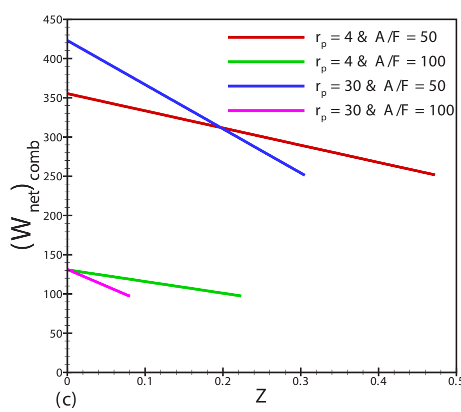
and $r_p = 30$ when $A/F = 100$ and $z = 0$, whereas the secondary cycle work output at $r_p = 4$ and $r_p = 30$ for $A/F = 50$ at $z \approx 0.2$ is the same. In the range, $0 \leq z \leq 0.2$, the secondary cycle work output at $r_p = 30$ is less than the primary cycle work output at $r_p = 4$ for $A/F = 50$. The maximum work output of the secondary cycle at $r_p = 4$ is 21.8% more than that at $r_p = 30$ for $A/F = 50$. Also, at $z = 0$, the work output of the primary cycle at $A/F = 50$ is 143.7% more than that at $A/F = 100$ for $r_p = 4$. But, the work output of the primary cycle at $A/F = 50$ is 100% more than that at $A/F = 100$ for $r_p = 30$. Figure 2c shows the variation of combined cycle



(a)



(b)



(c)

Figure 2: Variation of network output of primary cycle (a), secondary cycle (b), and combined cycle (c) with respect to partial amount (z) of the combustible gases for different pressure ratio (r_p) and air-fuel ratio (A/F).

work output (primary + secondary) with respect to z. It can be observed that the variation of combined cycle work output with respect to z is the same as the variation of work output of primary cycle that is it decreases with z and attains a maximum value at $z = 0$ for any considered range of pressure ratio and air-fuel ratio. It shows that for a maximum work output from the combined cycle, a whole amount of combustible product must pass through the heat recovery heat exchanger *via* the gas turbine of the primary cycle. The maximum work output from the combined cycle occurs at $r_p = 30$ is 39.3% more than the work output at $r_p = 4$ for $A/F = 50$. Also, at $z = 0$, the primary cycle work output at $A/F = 50$ is 227% more than the primary cycle work output of at $A/F = 100$ for $r_p = 4$ while the primary cycle work output at $A/F = 50$ is 134.6% more than the primary cycle work output at $A/F = 100$ for $r_p = 30$.

Figure 3a, b, and c show the variation of thermal efficiency of the primary cycle, secondary cycle, and combined cycle, respectively. The trend of thermal efficiency variation in Fig. 3a is almost the same as the trend of primary cycle work output. The thermal efficiency of the primary cycle decreases with z and A/F and attains maximum at $z = 0$ and $A/F = 50$ when $r_p = 30$. The thermal efficiencies of the primary cycle at $r_p = 4$ and 30 are the same when $A/F = 100$ at $z = 0$, whereas the thermal efficiencies of the primary cycle at $r_p = 4$ and 30 are equal when $A/F = 50$ at $z = 0.2$. For $0 \leq z \leq 0.2$, the thermal efficiency of the primary cycle at $r_p = 30$ is greater than the thermal efficiency at $r_p = 4$. Figure 3b shows that the thermal efficiency of the secondary cycle increases with z due to increase in temperature of combustible product and rate of steam generation according to Eqs. (15) and (16). The minimum thermal efficiency under the considerable range of parameters is similar at $r_p = 4$ and 30 when $A/F = 100$ and at $r_p = 30$ when $A/F = 50$. The maximum thermal efficiency of the secondary cycle is 28.6% when $A/F = 50$ at $r_p = 30$ for $z = 0.3033$ and also at $r_p = 30$ for $z = 0.4707$. Figure 3c shows that the thermal efficiency of the combined cycle decreases with z and A/F.

Plant exergy destruction is the addition of each component of the plant. It means that total exergy destruction increases with the increase in the total number of accessories and necessaries of the plant. The number of components in the combined cycle power plant is more than a simple gas turbine power plant and, due to which the total exergy destruction of a simple gas turbine plant is always less than that of the combined cycle power plant. It also observed that the exergy destruction of cycle components except the air compressor is the function of the air-fuel ratio. Figure 4 shows

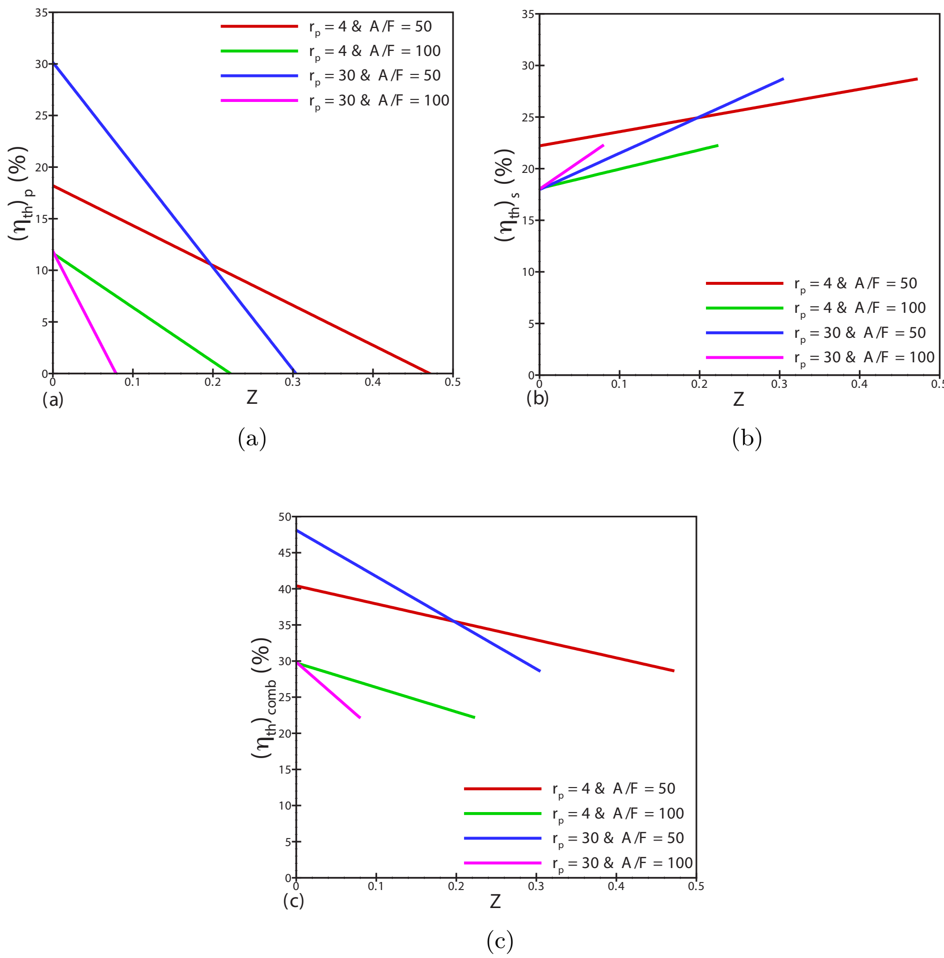


Figure 3: Variation of thermal efficiency of primary cycle (a), secondary cycle (b), and combined cycle (c) with respect to partial amount (z) of the combustible gases for different pressure ratio (r_p) and air-fuel ratio (A/F).

the variation of total exergy destruction of the cycle understudy with respect to the pressure ratio and the air-fuel ratio. It is noted from the figure, that the plant exergy destruction increases with the decreases in air-fuel ratio at the particular value of pressure ratio, whereas the plant exergy destruction increases with pressure ratio at particular value of air-fuel ratio. The maximum rate of increase is at $r_p = 30$ when $A/F = 50$. At $z = 0$, the total exergy destruction at $r_p = 4$ and $A/F = 100$ is least whereas the maximum is attained at $r_p = 30$ and $A/F = 50$.

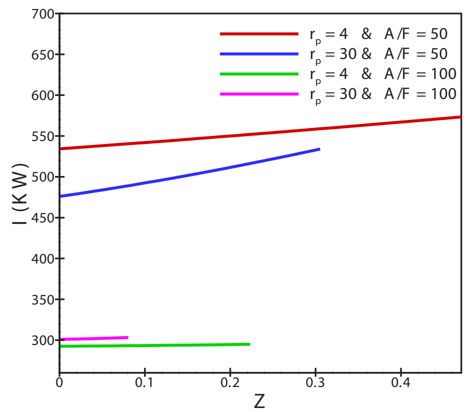


Figure 4: Variation of total exergy destruction with respect to partial amount (z) of the combustible gases for different pressure ratio (r_p) and air-fuel ratio (A/F).

8 Conclusion

Based on the above analysis, the following conclusions are listed.

1. The work output of the primary cycle and combined cycle decreases, whereas the work output of the secondary cycle increases with partial amount of the combustible gases under the considered range of pressure ratio and air-fuel ratio. In both the primary cycle and combined cycle the maximum work output was observed, at $z = 0$, $r_p = 30$, and $A/F = 50$. And under these conditions, the work output of the proposed cycle increased by 59.6% as compared to the simple gas turbine cycle. Therefore, for the maximum work-output of the primary cycle and combined cycle, the entire mass of combustible products from the combustion chamber is released to the environment through a heat exchanger via the gas turbine of the primary cycle.

2. Thermal efficiency of the primary cycle and combined cycle also decrease with partial amount of the combustible gases under the considered range of pressure ratio and air-fuel ratio. It is noted that at $z = 0$, $r_p = 30$, and $A/F = 50$, the thermal efficiency of the primary cycle and combined cycle attain its peak value. The thermal efficiency of the proposed cycle also increased by the same percentage as noted for the work output with respect to the simple gas turbine cycle. The maximum rise in thermal efficiency is reflected under the condition of $z = 0$, $r_p = 4$, and $A/F = 100$.

3. As compared to the pressure ratio, the effect of the air-fuel ratio is more dominating on thermal efficiency and work output in the primary, secondary and combined system. Therefore, for maximum thermal efficiency, and maximum work output of the combined cycle, the air-fuel ratio should be as minimum as possible.

4. For minimum total exergy destruction, the whole mass of the combustible product is released to the environment through a heat exchanger *via* a gas turbine of the primary cycle with an air-fuel ratio as minimum as possible.

This paper proves that for optimum work output and thermal efficiency of topping as well as combined cycle, the whole amount of combustible gases passes through the gas turbine of the topping cycle with a maximum pressure ratio and minimum air-fuel ratio. The full amount of combustible gases passes through the gas turbine of the topping cycle is also favorable for minimum exergy losses through the exhaust gases with the only change that both the pressure ratio as well as air-fuel ratio should be at maximum. Finally, the extraction of combustible gases from the passage of the combustion chamber and gas turbine of the topping cycle to operate the bottoming-cycle is not suitable to enhance the combined-cycle power plant performance.

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