

Khan, M. N.; Tlili, I.

Article

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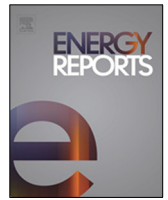
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Research paper

Innovative thermodynamic parametric investigation of gas and steam bottoming cycles with heat exchanger and heat recovery steam generator: Energy and exergy analysis

M.N. Khan^a, I. Tlili^{b,*}^a Department of Mechanical and Industrial Engineering, College of Engineering, Majmaah University, Majmaah 11952, Saudi Arabia^b Energy and Thermal Systems Laboratory, National Engineering School of Monastir, Street Ibn El Jazzar, 5019 Monastir., Tunisia

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ABSTRACT

In this study, gas and steam bottoming cycles, driven by the exhaust gas of Gas turbine cycle, are analyzed thermodynamically. The waste heat of the Gas turbine topping cycle (GT) exhaust gas is used partially to heat the air in the air bottoming cycle and partially to generate steam by passing it through HRSG in the steam bottoming cycle. The topping Gas turbine and air bottoming cycles are coupled with the heat exchanger, whereas the topping and steam bottoming cycles are coupled with Heat Recovery Steam Generator (HRSG). The effects of turbine inlet temperature ($1100\text{ K} \leq \text{TIT} \leq 1500\text{ K}$), pressure ratio ($6 \leq r_{pt} \leq 12$) and flow rate of exhaust gas on the Net work and combined thermal efficiency and exergy loss of exhaust gas are investigated. It is observed that the Net work of steam bottoming cycle is higher than the Net work of gas bottoming cycle and for higher values of turbine inlet temperature $\text{TIT} = 1500\text{ K}$, 90% of power is recovered through the bottoming cycle at lower pressure ratio. The combined thermal efficiency increases by 10% with an increase in the turbine inlet temperature from 1100 to 1500 K. It is found that the combined thermal efficiency increases with all pertinent parameters.

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1. Introduction

The availability of great amounts of energy is a crucial requirement for the well-being and development of our modern industrial societies (Angerer et al., 2017). The global consumption of energy rises continuously in parallel with population increase and economic growth (MacPhee and Beyene, 2017). To afford this ever-increasing demand, the main energy sources are fossil fuels. Due to the sustainability of oil resources and their heavy disadvantages such as air pollution including sulfur dioxides, greenhouse gases, solid particulates emission and environmental hazards, there is an urgent need to optimize energy conversion systems to promote fuel efficiencies and increase energy savings (Ibrahim and Rahman, 2015). Advanced power scenarios aim to improve performance and to operate with a wide range of fuels and with high tolerance to fuel variability, reduced pollution, and low energy cost. Thus, the development of old power plant relying on conventional cycles can be made steadily (Herrmann et al., 2017). Due to high-efficiency improvement that can be achieved by combining conventional cycles inadequate conditions such as the grouping of a gas turbine with a steam bottoming cycle pressed many engineers and

researchers to investigate in this field (Masci and Sciubba, 2017; Haseli, 2017; Fu et al., 2015).

Mahmoudi et al. (2016) recommended a new combined system composed of a Gas Turbine-Modular Helium Reactor and an augmented Kalina cycle which is found after simulations to have reduced dimensions and high economic efficiency. Wu et al. (2017) studied stand-alone triple combined cycle systems based on calcium looping technology and demonstrated after carrying on simulations that the total power efficiency of this off-the-grid electricity system reaches 60.56% and its CO₂ emissions per kWh of electricity is less than 7.18 g CO₂. Yaghet et al. (2016) considered a subcritical and supercritical organic Rankine cycle (ORC) which recovers exhaust gas waste heat of biogas-fueled combined heat and power engine and conducted a parametric optimization and found that the supercritical ORC has better net work, energy efficiency, and exergy efficiency than the subcritical ORC. Benato and Maco (2017) recommended an organic Rankine cycle (ORC) technology to recuperate the discarded heat enclosed in the exhaust gases of a 1 MWel biogas engine, using an “in-house” optimization tool, fluid and the plant configuration are optimized amongst 115 pure and mixtures fluids. Results display that proposed ORC with real data assures a 30% higher net electric power compared to one with ICE nameplate conditions. Pierobon et al. (2014) investigate by multi-objective optimization approach the furthestmost

* Corresponding author.

E-mail address: lskander.tlili@enim.rnu.tn (I. Tlili).

Nomenclature

W	Net work (W)
h	Specific enthalpy (J/kg)
\dot{m}	Mass flow rate (kg/s)
q	Heat supplied
η	Thermal efficiency
ϵ	Effectiveness of HE
P	Pressure
v	Specific volume
HE	Heat Exchanger
HRSG	Heat Recovery Steam Generator
T	Temperature (K)
r_p	Pressure ratio
s	Specific entropy (J/kg.K)
C_p	Specific heat (kJ/kg.K)
LCV	Lower Calorific Value of Fuel
I	Exergy Loss (W)
TIT	Topping cycle Turbine Inlet Temperature (K)

Subscripts

c	Compressor
gt	Gas turbine
st	Steam turbine
cc	Combustion chamber
p	Pump
a	Air
g	Exhaust gas of gas turbine of topping cycle
s	Steam
f	Fuel
γ	Ratio of specific heat
GB	Gas Bottoming Cycle
SB	Steam Bottoming Cycle
T	Topping cycle
comb	Combined cycle

appropriate waste heat recovery system for offshore services using three technologies steam Rankine cycle, the air bottoming cycle and the organic Rankine cycle. It is found that the organic Rankine cycle system guarantees higher performances more than the steam Rankine cycle. However the application of air bottoming cycle has no effect in economic and environmental viewpoint. [Benato et al. \(2015\)](#) explore and predict the most stressed devices lifetime reduction for power plant contains a gas turbine and a three pressure level Heat Recovery Steam Generator with reheat. Results show that the proposed the power plant dynamic model valuable innovative instrument to improve the plant's flexibility. [Benato et al. \(2016\)](#) inspect new technique capable to forecast the power plant dynamic behavior through running, recognize the furthest stressed section and assess their lifetime drop, they apply their procedure a 380 MW combined cycle unit, they found that for load variation around 50% faster than the reference case 52.9% reduction in superheater collectors life appears. However, for load variation around 50% a 35.8% lifetime increase is detected in superheater collectors. [Benato et al. \(2016\)](#) develop and test numerical model for case study is the Draugen off-shore oil and gas platform, they found that decreasing the core weight of the recuperator leads to limiting the frequency fluctuations, therefore reducing the hazard of failure of the power structure.

[Luk'yanova and Trukhnii \(2012\)](#) determined the quantitative margins required to improve the economic efficiency of the steam

power unit for a given gas turbine unit and studied the flow rate, temperature, and thermophysical properties of gases supplied to the heat recovery boiler. Studies of the thermal efficiency improvement of combined cycle's power plant are of great interest. Several researches have carried out studies to enhance the gas/steam combined cycle performance. [Kovalevskii \(2011\)](#) investigated the impact of initial steam pressure variation on thermal efficiencies of condensing type circuits of binary combined cycle plants containing one, two, and three loops with different pressure levels. They reported that the combined use of steam reheating and the gas assisted air preheater leads to a higher efficiency however it requires an amplification of the heat recovery boiler heating surfaces. [Chen et al. \(2008\)](#) considered a thermodynamic model for open combined Brayton and two parallel inverse Brayton cycles. They improved the model performance by regulating the compressor inlet pressure of the bottom cycles, the mass flow rate and the distribution of pressure losses along the flow path. [Mondal and Ghosh \(2016\)](#) assessed the techno-economic performance and carried on a thermodynamic analysis of a biomass-fired combined cycle plant, employing a topping gas turbine (GT) cycle and a bottoming steam turbine cycle. They found that the total plant efficiency is maximized at topping cycle pressure ratio of 4 and Turbine inlet temperature = 1000 C, where the plant also has the least CO₂ emission. In order to make the results more accurate, some pertinent and input parameters for the combined cycle power plant are considered around real case value e.g. ambient temperature, mass flow rate of air and steam in different compartments, isentropic efficiency of turbine, compressor and pump ([Khan and Tlili, 2018](#); [Khan et al., 2017](#); [Ghazikhani et al., 2014](#); [Tiwari et al., 2013](#)). In order to enhance the combined cycle performance several solutions were studied and proposed such as, augmenting pressure ratio, inserting intercooling and re-heating in the reference cycle, increasing gas turbine inlet temperature (TIT), and decreasing the temperature differences in the heat recovery steam generator (HRSG) as well as minimizing the condenser pressure. [Sanjay and Prasad \(2008\)](#) proposed different scenarios to improve gas-steam combined cycle power plant efficiency beyond 62 percent such as augmenting pressure ratio, temperature inlet temperature, and maximum blade temperature and inserting intercooling and re-heating in the reference cycle. In a classic combined cycle, exhaust heat from the gas turbine is recovered in a HRSG. The major recognized HRSG parameters are the exhaust gas temperature, stack temperature, pinch point and steam generation pressure and temperature. A thermodynamic analysis of a combined cycle with three-pressure reheat HRSG was conducted by [Xiang and Chen \(2006\)](#). They found that the combined cycle power plant efficiency could be enhanced up to 59.05% at base load by combining HRSG optimization and gas to gas heat recuperation. [Corchero et al. \(2016\)](#) conducted a parameter study of a semi-closed combined cycle for CO₂ capture, based on the combined cycle (CC) gas turbine side. They reported that the maximum efficiency can be achieved when inlet temperature is close to the HRSG exit conditions at the gas turbine side. Energy and exergy analysis of a combined cycle power plant includes data required to determine the potential for enhancing system efficiency. [Mohtaram et al. \(2017\)](#) carried on an energy-exergy analysis on the ammonia-water combined cycle and the Rankine cycle in order to assess the impact of compressor pressure ratio on the thermodynamic performance. They found that the exergy destruction of high-pressure compressors, intercooler, and the gas turbine continuously augments in parallel with the rise of the pressure ratio of the compressor. However, the exergy destruction of recuperator declines constantly. [Angelino et al. \(1999\)](#) investigated a procedure where a fraction of the low-pressure steam is extracted and fed to an auxiliary organic Rankine cycle which is considered an alternative way of approaching optimum heat rejection conditions. A thermodynamic and exergo-economic

modeling of a combined steam-organic Rankine cycle using different organic fluid was carried on by [Nazari and Heidarnajad \(2016\)](#) who reported that steam condenser and organic vapor generator have the main distribution in total exergy destruction of the steam and organic cycle respectively. An innovative framework was proposed by [Sharifzadeh et al. \(2017\)](#) in order to guarantee the operation safety. After doing an exergy analysis, authors reported that the balance between the power generated in the solid oxide fuel cell and gas turbine was the main responsible of setting up the length of the operating window. Recently, alternative power cycles employing multicomponent working fluids have been studied intensively. [Murugan and Subbarao \(2008\)](#) focused on reducing energy loss in the condenser; power output and efficiency loss in the low-pressure turbine and irreversibility in the boiler secondary to evaporation enthalpy in order to enhance the efficiency. They reported also that ammonia–water mixture is the best working fluid regarding thermodynamic performance. [Kler and Zakharov \(2014\)](#) proposed a procedure of coordinated optimization of the parameters of cycles in 15 commercial gas turbine and combined cycle power plants. They reported that to enhance the efficiency factor of the gas-turbine plant, the compression ratio should be increased by decreasing the gas temperature whereas concerning the combined cycle power plant, it is better to increase the gas temperature by decreasing the compression ratio. A simulation procedure was proposed by [Nikhil et al. \(2011\)](#). They concluded that as the pressure ratio increases, an increase in efficiency becomes less. Both Turbine inlet temperature and superheater temperature rise increase cycle work output and efficiency. Whereas, the steam turbine and combined cycle output is reduced when the condenser pressure increases. [Kumar and Singh \(2012\)](#) considered a general model of gas–steam combined cycle with transpiration cooled gas turbine blades and found that both the cycle efficiency and specific work of topping and bottoming cycle augment in parallel with turbine inlet temperatures for each pressure ratio. [Lukyanova and Trukhnyy \(2012\)](#) demonstrated that the optimal separating pressure in designing three circuit combined cycle power plants with subcritical steam conditions varies between 2.0 to 2.5 MPa. A comparative study between a solid oxide fuel cell integrated combined power system with triple pressure reheats cycle in the bottoming steam turbine plant was conducted by [Sarmah and Gogoi \(2017\)](#). The results revealed that the system with single pressure ST cycle would be the most appropriate as it is simple with less number of components and minimum total cost with best performance. Shutdown process optimization of combined cycle gas turbine was investigated by [Wan et al. \(2016\)](#) who proposed a methodology producing an extra power of 28.41 MWh and an extra profit of 4.79 million Yuan annually. Results revealed also that the total power generation is more significant with the load-off rates rather than the load-up rates. The purpose of this work is to implement a thermodynamic evaluation and a parametric study of an energy and exergy analyses for two different bottoming cycles in operation simultaneously by gas turbine exhaust. To the best of our apprehension nobody has discussed before the effect of two different bottoming cycles in operation simultaneously by gas turbine exhaust on the combined cycle performance more than that the real using of this component is very easy, simple and considered cheapest compared to other technology used to increase efficiency and reduce greenhouse gas emission. It is important to note here that in all the discussed articles on combined cycle performance, most of the authors have used a variety of more complex technologies and different working fluids and most of them are not applicable due to their cost or complexity, and nobody provided a simple and specific example of combination leads to reduce greenhouse gas emission and increasing efficiencies. Results of the parametric investigation are reported and discussed in detail in Section 3. Finally, the results enhance the understanding of gas

and steam bottoming cycle's efficiencies and exergy destruction in order to optimize their performances and to study the effect of HRSG and HE on the total Net work, combined efficiency and exergy losses of exhaust gasses.

2. Energy analysis

2.1. Analysis of topping cycle

In topping cycle, the air is compressed from 1 to 2 (refer [Fig. 1](#)) where its temperature rises from T_1 to T_2 . The compressed air then enters the combustion chamber where the combustion of fuel takes place. This results in the rise of temperature of combustion product from T_2 to T_3 . The high-temperature gases enter the turbine where it expands to the final temperature T_4 .

Therefore, turbine work in topping cycle $(W_{gt})_T$ is given by

$$(W_{gt})_T = \dot{m}_g c_{pg} (T_3 - T_4) \quad (1)$$

where

$$T_4 = T_3 \left\{ 1 - \eta_t \left(1 - r_{pt}^{-\beta} \right) \right\} \quad (2)$$

And compressor work in topping cycle $(W_c)_T$ is given by

$$(W_c)_T = (\dot{m}_a)_T c_{pa} (T_2 - T_1) \quad (3)$$

where

$$T_2 = T_1 \left\{ 1 + \frac{(r_{pt}^\alpha - 1)}{\eta_c} \right\} \quad (4)$$

Net work topping cycle

$$(W_{net})_T = (W_{gt})_T - (W_c)_T \quad (5)$$

The energy balance in the combustion chamber is given by

$$\dot{m}_f \times LCV = (\dot{m}_f + (\dot{m}_a)_T) \times c_{pg} \times T_3 - (\dot{m}_a)_T \times c_{pa} \times T_2 \quad (6)$$

From Eq. (6), mass flow rate of flue (\dot{m}_f) is given by

$$\dot{m}_f = (\dot{m}_a)_T \cdot \left\{ \frac{c_{pg} \cdot T_3 - c_{pa} \cdot T_2}{LCV - c_{pg} \cdot T_3} \right\} \quad (7)$$

Therefore, the efficiency of the topping cycle (η_t)

$$\eta_t = \frac{(W_{net})_T}{\dot{m}_f \times LCV} \quad (8)$$

2.2. Analysis of bottoming cycles

Air at a temperature of T_6 enter the compressor from where it at higher pressure and temperature T_7 . This high-temperature air enters the heat exchanger where it absorbs heat from the exhaust gas of topping cycle, and its temperature rises to T_8 . This high pressure and high-temperature air enter the gas turbine where it expands to the final temperature of T_9 .

2.2.1. Air bottoming cycle description

The turbine work in the Gas Bottoming Cycle (GB) is given by

$$(W_{gt})_{GB} = (\dot{m}_a)_{GB} \cdot c_{pa} \cdot (T_9 - T_8) \quad (9)$$

where

$$T_9 = T_8 \left\{ 1 - \eta_t \left(1 - r_{pb}^{-\alpha} \right) \right\} \quad (10)$$

The compressor work in the Gas Bottoming Cycle (GB) is given by

$$(W_c)_{GB} = (\dot{m}_a)_{GB} \cdot c_{pa} \cdot (T_7 - T_6) \quad (11)$$

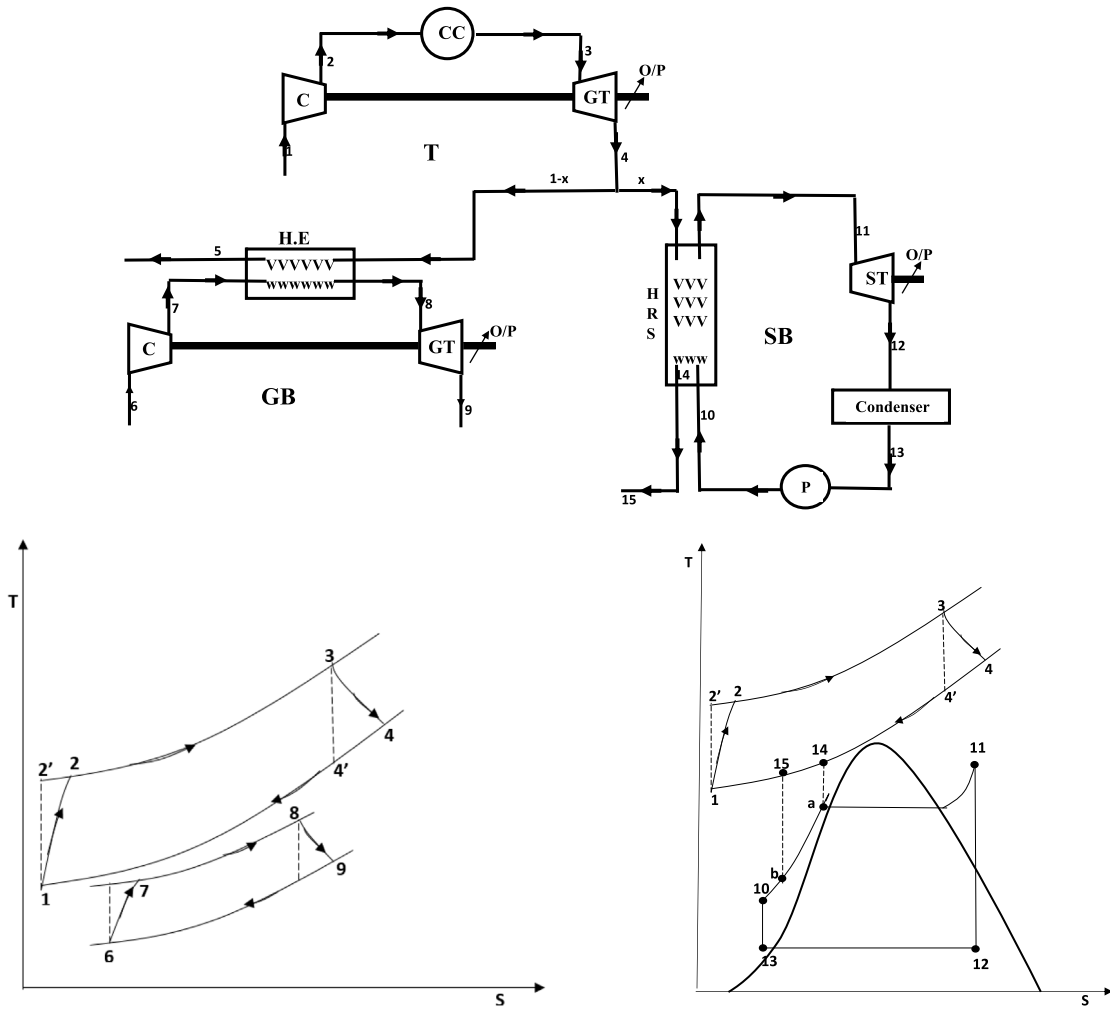


Fig. 1. Schematic diagram of steam and gas bottoming cycles in operation simultaneously by gas turbine exhaust.

where

$$T_7 = T_6 \left\{ 1 + \frac{(r_{pb}^\alpha - 1)}{\eta_c} \right\} \quad (12)$$

Net work of Gas Bottoming Cycle (GB) is

$$(W_{net})_{GB} = (W_{gt})_{GB} - (W_c)_{GB}$$

Effectiveness of heat exchanger is given by

$$\varepsilon_1 = \frac{(m_a)_{GB} \cdot c_{pa} \cdot (T_8 - T_7)}{(1 - x) \cdot m_g \cdot c_{pg} \cdot (T_4 - T_7)} \quad (13)$$

Energy balance in the heat exchanger yields

$$T_8 = T_7 + \varepsilon_1 (1 - x) \left(\frac{\dot{m}_g}{(\dot{m}_a)_{GB}} \right) \cdot \left(\frac{c_{pg}}{c_{pa}} \right) \cdot (T_4 - T_7) \quad (14)$$

2.2.2. Heat Recovery Steam Generator description

Let the pinch point temperature difference $(T_{14} - T_f)$ be 298 K then

$$T_{14} - T_a = 298 \text{ K}$$

T_a = Saturation Temperature at P_b

and

$$h_a = (h_f)_{P_b}$$

Therefore

$$h_{11} = h_a + \frac{\varepsilon \cdot x \cdot \dot{m}_g \cdot c_{pg}}{\dot{m}_s} (T_4 - T_{14})$$

$$h_{10} = h_{13} + w_p = h_{13} + \frac{(v_f)_{P_c} \cdot (P_b - P_c)}{\eta_p}$$

$$T_{15} = T_{14} - \frac{\dot{m}_s}{\varepsilon \cdot x \cdot \dot{m}_g \cdot c_{pg}} (h_a - h_{10})$$

The expression for turbine work is given by

$$(W_{st})_{SB} = \eta_{ts} \cdot \dot{m}_s \cdot (h_{11} - h_{12}) \quad (15)$$

The expression for pump work is given by

$$w_p = \frac{(v_f)_{P_c} \cdot (P_b - P_c)}{\eta_p} \quad (16)$$

Net work of steam bottoming cycle

$$(W_{net})_{SB} = (W_{st})_{SB} - w_p \quad (17)$$

Modern HRSG surfaces operate at pinch point temperature difference of around 373 K. This value can reach a practical minimum of 243 K–253 K, while many real applications allow much higher values (in the order of 288 K–298 K) with the goal to decrease

the initial investment in heat exchangers. The simulations performed in this work assume the design pinch point temperature difference to be 373 K. This might seem a quite optimistic value for practical applications in small scale. A sensitivity analysis was therefore done for pinch point temperature differences higher than the assumed design value, ranging from 373 K up to 473 K.

2.3. Analysis of combined cycle

The Net work of combined cycle is the Net work of topping cycle plus Net work of bottoming cycle is:

$$(W_{net})_{comb} = (W_{net})_T + (W_{net})_{GB} + (W_{net})_{SB} \tag{18}$$

The expression for net efficiency of the combined plant is

$$\eta = \frac{(W_{net})_{comb}}{q} \tag{19}$$

Before starting calculations and modeling for the proposed system, and to make our results more accurate and realistic, we consider that the proposed system working under real value, for this reason some fixed and input parameters as ambient temperature, mass flow rate of air and steam in different compartments, isentropic efficiency of turbine, compressor and pump are taken around real case, this can elucidate any interpretations. Furthermore, the upper and lower limits of mass fraction of exhaust gases m were investigated for the positive work done at different TIT and r_p

Variables:	
r_{pt}	6–12
TIT	1100K–1500K
Assumptions	
Isentropic efficiency of turbine	85%
Isentropic efficiency of Compressor	85%
Mechanical efficiency of pump	95%
Ambient temperature	298K
Mass flow rate of air in topping cycle	10 kg/s
Mass flow rate of air in gas bottoming cycle	1 kg/s
Mass flow rate of steam in steam bottoming cycle	1 kg/s
Effectiveness of heat exchanger	0.8
Effectiveness of HRSG	0.9
Specific heat of gasses	1.14 kJ/kg-K
Specific heat of air	1.005 kJ/kg-K
Ratio of specific heat for air	1.4
Ratio of specific heat for gasses	1.33

3. Results and discussion

Before starting actual calculations, the upper and lower limits of the fraction of exhaust gases m were investigated for the positive work done. These limits are provided in Table 1 for each cycle. Using these limits, the variation in the Net work W_{net} with the mass fraction of exhaust gasses m for each cycle at different pressure ratios r_p for TIT = 1100 K, TIT = 1300 K and TIT = 1500 K are reported respectively in Figs. 2–4. It is important to note that the W_{net} for the combined and total bottoming cycle increase with both mass fraction of exhaust gasses m and turbine inlet temperature TIT, whereas it declines with pressure ratio. This can be explain by the fact that the increasing of pressure ratio leads to reduces the heat transfer in H.E. which in turn affect both the net work of combined and total bottoming cycle, as displayed in Fig. 1. It is found that Net work of GT cycle increases significantly with TIT which substantiates that the TIT is vital for the Net work of each cycle. It is found that for higher value of turbine inlet temperature the effect of pressure ratio on Net work becomes insignificantly especially for low mass fraction m (%) of exhaust gasses, this can

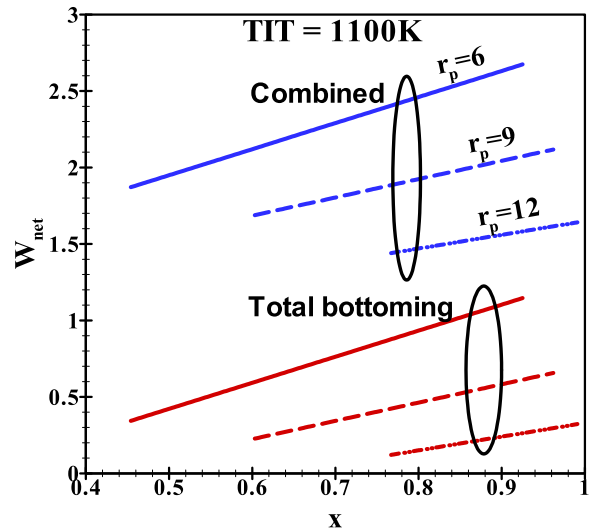


Fig. 2. Net work of combined and bottoming cycle vs mass fraction of exhaust gasses of topping cycle at TIT = 1100 K.

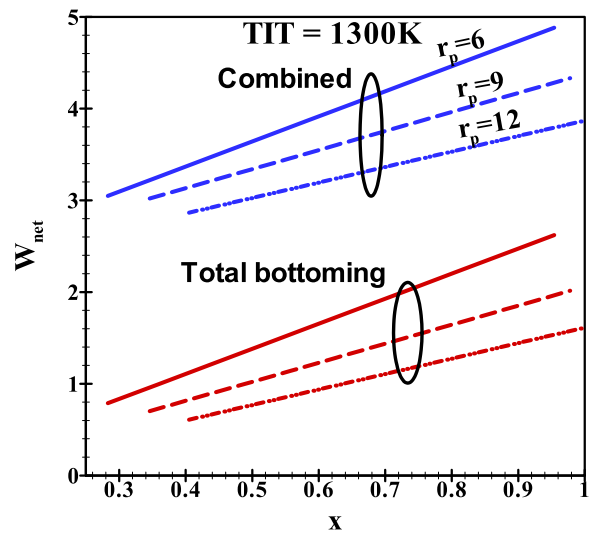


Fig. 3. Net work of combined and bottoming cycle vs mass fraction of exhaust gasses of topping cycle at TIT = 1300 K.

be attributed to the fact that the supplied heat will not be affected by pressure ratio for higher value of turbine inlet temperature.

Figs. 5–7 present the variation of Iso-efficiency for the combined cycle with total net work at different mass fraction respectively for TIT = 1100 K, TIT = 1300 K and TIT = 1500 K. It is found that the thermal efficiency increases with both exhaust mass flow rate m and turbine inlet temperature, but increases with turbine inlet temperature and pressure ratio. It should be pointed out that the thermal efficiency increases with combined cycle net work, however for higher value of turbine inlet temperature the thermal efficiency decreases with combined cycle net work for small mass fraction. The physical reason behind this behavior in our case is that the total Net work increases with the turbine inlet temperature and decreases with mass fraction of exhaust gasses. Since the thermal efficiency depends upon the ratio of total Net work with heat supply as presented by Eq. (19). The variation of combined thermal efficiency at the lower turbine inlet temperature TIT = 1100 K is not significant whereas this variation is more significant at the higher turbine inlet temperature TIT = 1500 K is vital

Table 1

Lower and upper limits of mass fraction m (%) of exhaust gas of topping GT Cycle for positive W_{net} at different TIT and r_p .

r_p	$m_s = m_{ab} = 1$; TIT = 1100K		$m_s = m_{ab} = 1$; TIT = 1300K		$m_s = m_{ab} = 1$; TIT = 1500K	
	Min x (%)	Max x (%)	Min x (%)	Max x (%)	Min x (%)	Max x (%)
6	45.4	92.5	28.3	95.4	20.5	96.6
9	60.3	96.2	34.6	97.8	24.2	98.5
12	76.7	99.1	40.5	99.5	27.5	99.7

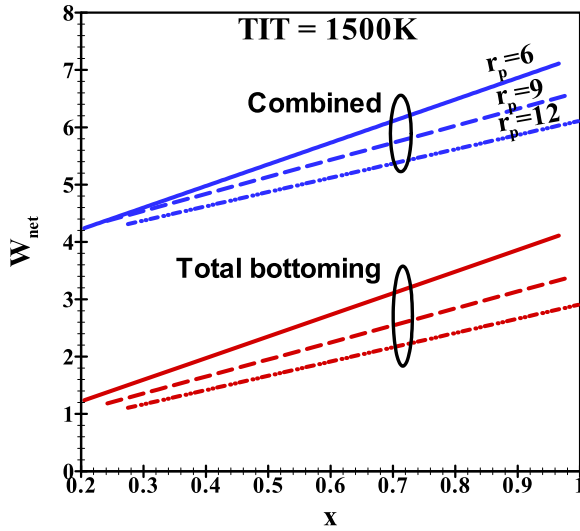


Fig. 4. Net work of combined and bottoming cycle vs mass fraction of exhaust gases of topping cycle at TIT = 1500 K.

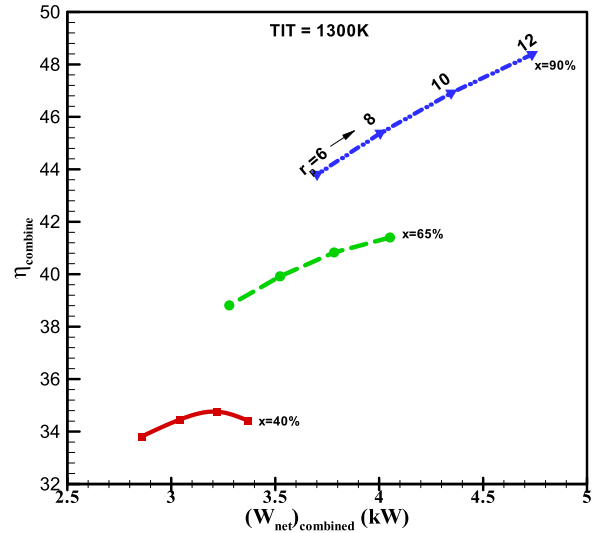


Fig. 6. Iso-efficiency curves of combined cycle at TIT = 1300 K.

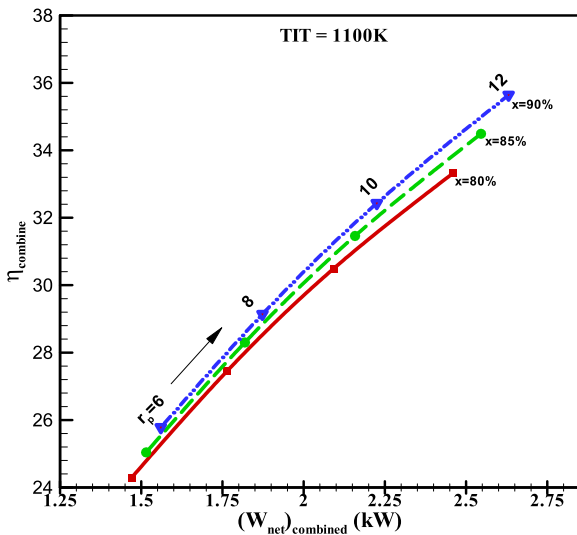


Fig. 5. Iso-efficiency curves of combined cycle at TIT = 1100 K.

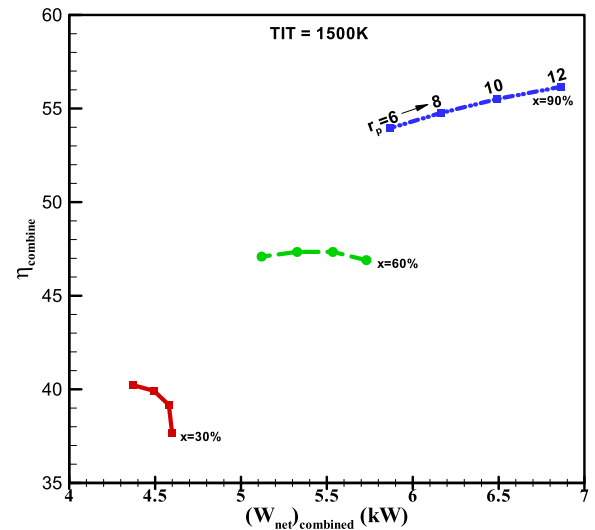


Fig. 7. Iso-efficiency curves of combined cycle at TIT = 1500 K.

for combined thermal efficiency. It demonstrates that the turbine inlet temperature is very crucial for the higher combined thermal efficiency. The combined thermal efficiency at high mass fraction increases in average from 31% to 55% with the increase of the turbine inlet temperature from 1100 to 1500 K.

The Net work variation with mass fraction of exhaust gases of topping cycle at TIT = 1100 K, TIT = 1300 K and TIT = 1500 K are depicted respectively in Figs. 8–10. As observed for only topping cycle T Net work is independent of the mass fraction of exhaust gases m at each pressure ratio and TIT. The Net work W_{net} for the combined cycle contain only topping cycle with gas bottoming

cycle (T+GB) decreases with the mass fraction m (%) of exhaust gases at each pressure r_p and turbine inlet temperature TIT. This is due to decrease of mass fraction of exhaust gases $(1 - m)$ % in the GB which in turn reduces the heat transfer in H.E., as displayed in Fig. 1. Though, The Net work W_{net} for the combined cycle contain only topping cycle with steam bottoming cycle (T+SB) decreases with the mass fraction m (%) of exhaust gases at each pressure r_p and turbine inlet temperature TIT. This can be elucidated by the increasing of the mass fraction.

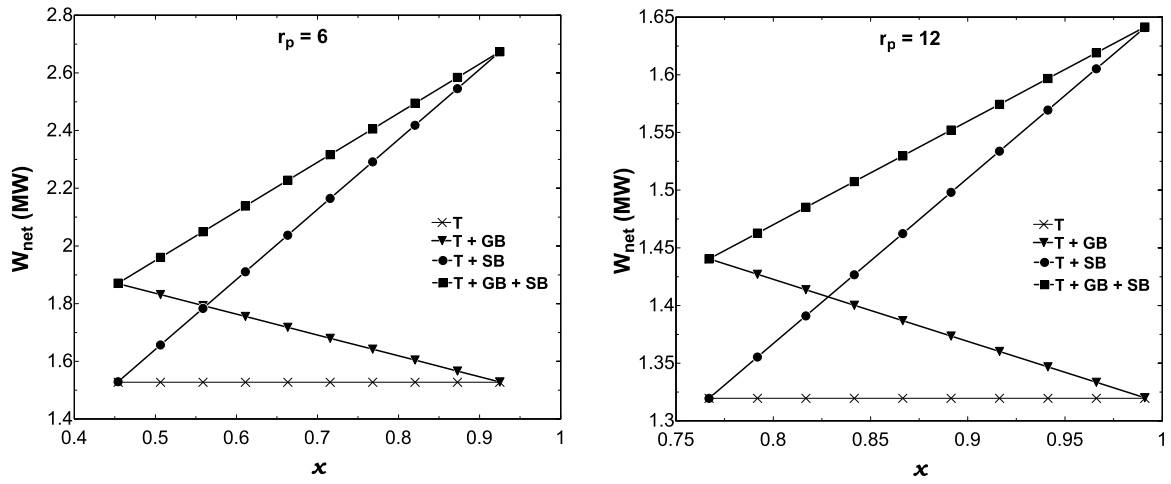


Fig. 8. Net work vs mass fraction of exhaust gases of topping cycle at TIT = 1100 K.

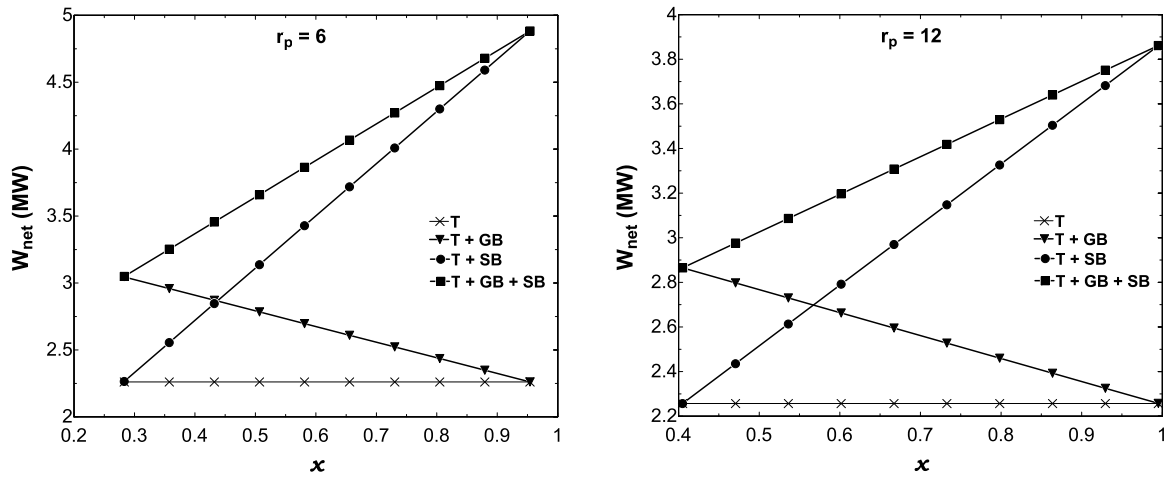


Fig. 9. Net work vs mass fraction of exhaust gases of topping cycle at TIT = 1300 K.

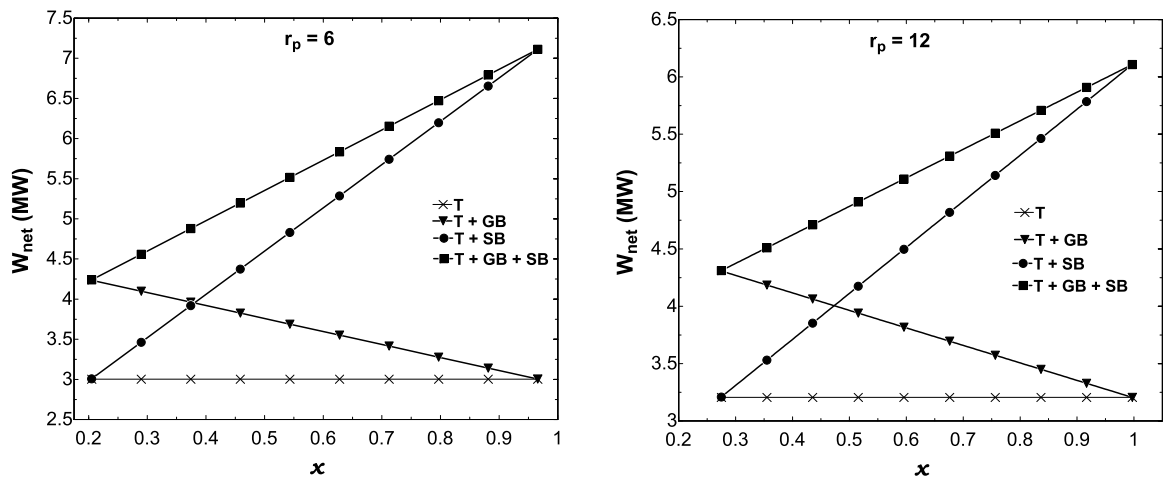


Fig. 10. Net work vs mass fraction of exhaust gases of topping cycle at TIT = 1500 K.

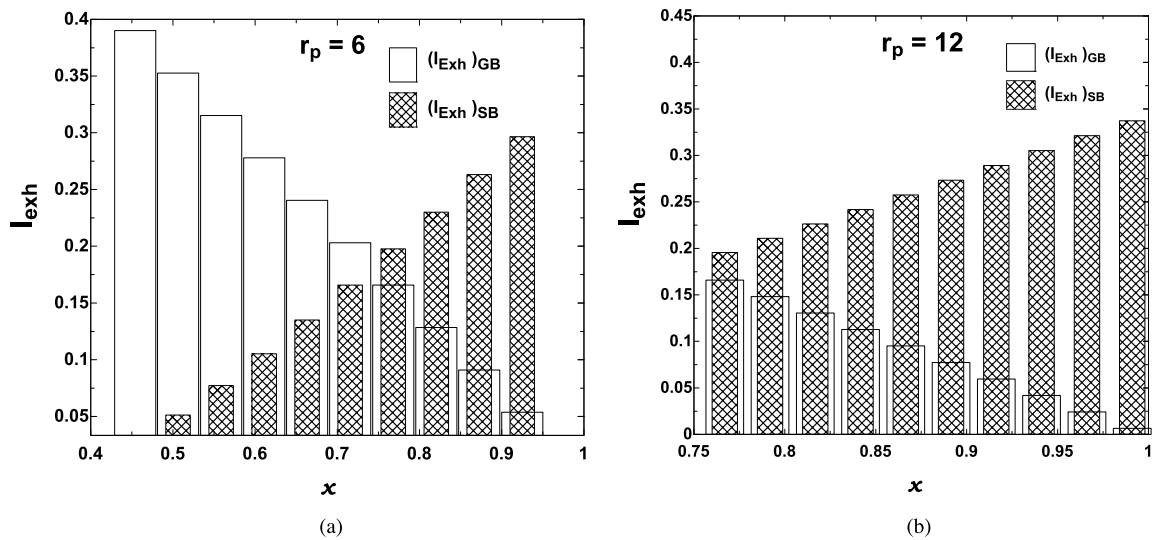


Fig. 11. Exergy loss of exhaust gases of topping cycle through Gas Bottoming Cycle (GB) and Steam Bottoming Cycle (SB) at TIT 1100 K at (a) $r_p = 6$ & (b) $r_p = 12$.

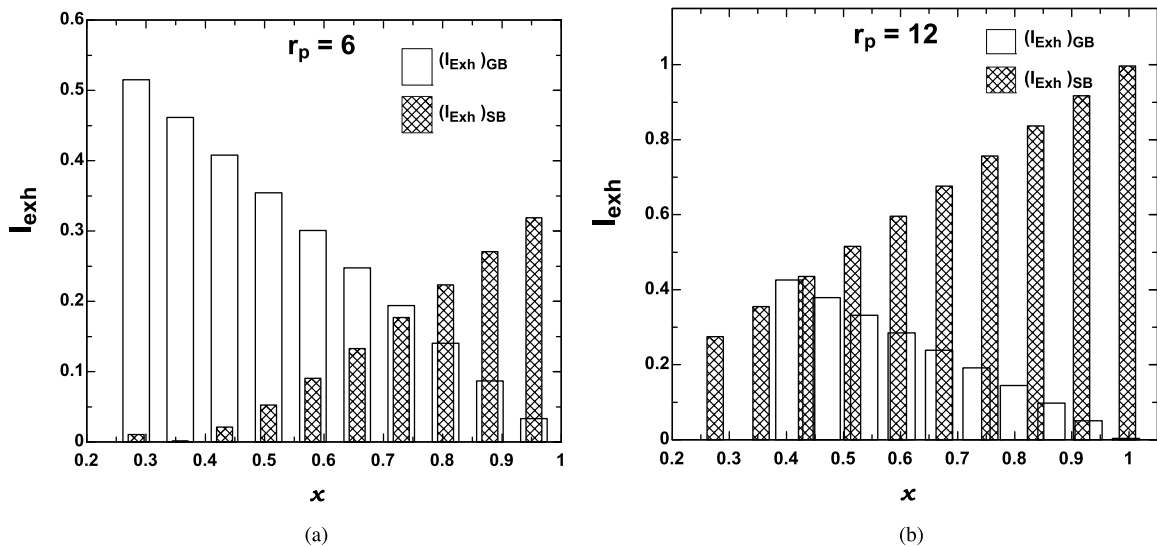


Fig. 12. Exergy loss of exhaust gases of topping cycle through Gas Bottoming Cycle (GB) and Steam Bottoming Cycle (SB) at TIT 1300 K at (a) $r_p = 6$ & (b) $r_p = 12$.

It is also found that the Net work W_{net} of the combined cycles (T+GB+SB) increases monotonically with an increase in the mass fraction m (%) of exhaust gasses at each r_p and TIT.

These results can be explained by the fact that, in case of T+GB, increasing pressure ratio leads to decrease the P–V surface for the cycle which in turn decreases the amount of Net work W_{net} since the P–V surface already exceed the optimum point in which it is in maximum value and it starts to drop. It can be seen from Fig. 1 that the performance of T+GB depends on H.E. which in turn transfers heat coming from the turbine of topping cycle and demonstrates that the Net work for both T+GB and T+SB increases meaningfully with turbine inlet temperature TIT of topping cycle.

Figs. 11–13 illustrate respectively the variation of exergy loss of exhaust gases of topping cycle with mass fraction of exhaust gasses through Gas Bottoming Cycle (GB) and Steam Bottoming Cycle (SB) at $r_p = 6$ and $r_p = 12$ for TIT = 1100 K, TIT = 1300 K and TIT = 1500 K. It can be seen that exergy loss for Gas Bottoming Cycle (GB) lessens significantly with mass fraction independent of pressure ratio and turbine inlet temperature, whereas exergy loss for Steam Bottoming Cycle (SB) augments meaningfully with mass

fraction independent of pressure ratio and turbine inlet temperature. Moreover, it is perceived that exergy loss of exhaust gases for both Gas Bottoming Cycle (GB) and Steam Bottoming Cycle (SB) decrease with pressure ratio while it is increase with turbine inlet temperature, this effect is obvious in Figs. 2–4. Figs. 14–16 display respectively the variation of total exergy loss of exhaust gases of topping cycle with mass fraction at $r_p = 6$ and $r_p = 12$ for TIT = 1100 K, TIT = 1300 K and TIT = 1500 K. It can be seen that total exergy loss of exhaust gases of topping cycle decreases significantly with mass fraction for both $r_p = 6$ and $r_p = 12$, while it increases with turbine inlet temperature.

4. Conclusion

A parametric study of energy and exergy analyses for two different bottoming cycles operated simultaneously by gas turbine exhaust has been investigated parametrically. The results enhance the understanding of gas and steam bottoming cycles efficiencies and exergy loss to optimize their performance and to study the effect of HRSG and HE usage on the total Net work, combined efficiency, and exergy losses of exhaust gasses.

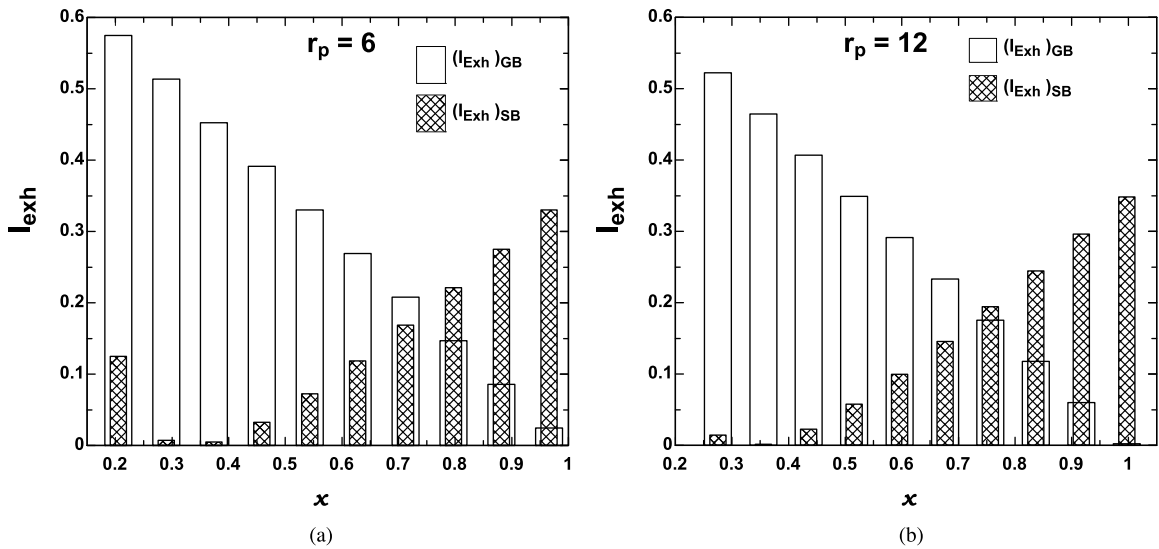


Fig. 13. Exergy loss of exhaust gases of topping cycle through Gas Bottoming Cycle (GB) and Steam Bottoming Cycle (SB) at TIT 1500 K at (a) $r_p = 6$ & (b) $r_p = 12$.

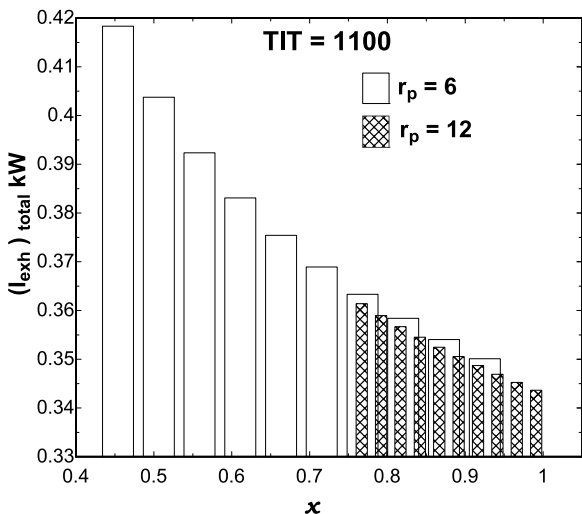


Fig. 14. Total exergy loss of exhaust gases of topping cycle at TIT 1100 K at $r_p = 6$ & $r_p = 12$.

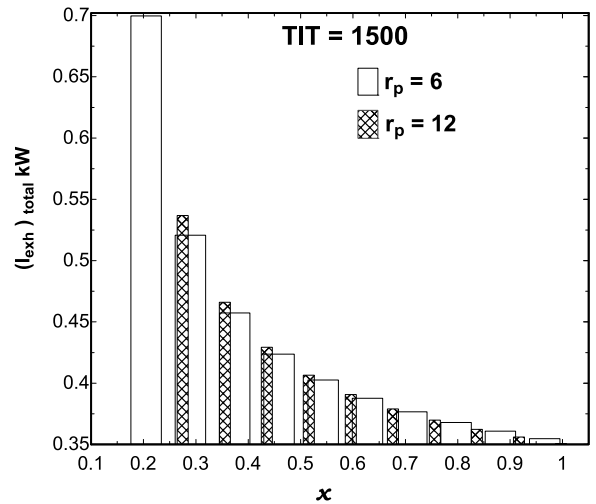


Fig. 16. Total exergy loss of exhaust gases of topping cycle at TIT 1500 K at $r_p = 6$ & $r_p = 12$.

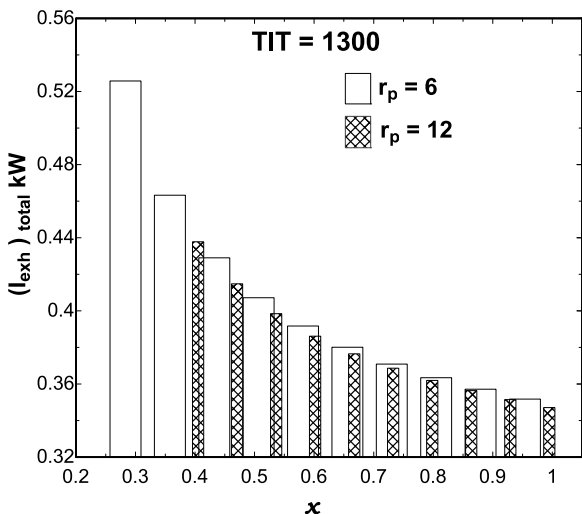


Fig. 15. Total exergy loss of exhaust gases of topping cycle at TIT 1300 K at $r_p = 6$ & $r_p = 12$.

The succeeding conclusions have been drawn from this work:

- The W_{net} for the combined and total bottoming cycle increase with both mass fraction of exhaust gases m and turbine inlet temperature TIT, whereas it declines with pressure ratio.
- Turbine inlet temperature affects Net work and efficiency of each combined cycle.
- For higher turbine inlet temperature, the topping cycle Net work increases with an increase in pressure ratio.
- The thermal efficiency increases continually with combined cycle net work for smaller value of TIT, however for higher value of turbine inlet temperature the thermal efficiency decreases with combined cycle net work for small mass fraction.
- TIT, pressure ratio and flow rate of exhaust gases affect significantly the Net work of bottoming cycles more than the Net work of topping cycle.
- At the higher value of TIT = 1500 K, more than 90% of power is recovered through the bottoming cycle.
- The thermal efficiency of steam and gas bottoming cycles increases with exhaust flow rate, turbine inlet temperature and pressure ratio.

– The combined thermal efficiency at high mass fraction increases in average from 31% to 55% with the increase of the turbine inlet temperature from 1100 to 1500 K.

Finally, this paper proves that for higher turbine inlet temperature it is the highly required utilization of recovered process HE and HRSG in bottoming cycles.

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