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Performance analysis of gas turbine power plants: Effect of operating parameters

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ABSTRACT

This study aims to evaluate the performance of a simple cycle gas turbine power plant by analysing the effect of different operating parameters. These operating parameters include compressor pressure ratio and compressor & turbine isentropic efficiencies. The study quantitatively assesses the exergetic efficiency and the exergy destruction of each unit in the cycle, as well as the power used or produced by the cycle. Any change in these parameters can significantly impact the power plant's overall performance through a specific unit in the cycle. For instance, increasing the compressor pressure ratio can reduce the temperature difference across the combustor, lessening the exergy destruction and improving the cycle's overall performance. However, any decline in the compressor or the turbine isentropic efficiency results in an increase in the exergy destruction of the affected unit and can result in a decrease in the overall cycle performance. This is due to either an increase in power required by the compressor or a decrease in power produced by the turbine. The analysis suggests that the turbine isentropic efficiency has a greater impact on the net power generated than the compressor isentropic efficiency. Additionally, the turbine inlet temperature is a dependent variable as operating at different compressor pressure ratios and compressor isentropic efficiencies lead to varying turbine inlet temperatures. Therefore, increasing the turbine inlet temperature does not always lead to improved performance.

Keywords: Simple Cycle Gas Turbine; Compressor Pressure Ratio; Isentropic Efficiency; Exergy Destruction.

1. Introduction

The performance of gas turbine power plants has been the focus of numerous studies. Many of these studies have examined the effects of compressor pressure ratio (PR) and turbine inlet temperature (TIT). It is generally agreed in the literature that both of these parameters are crucial for optimising gas turbine power plant performance. Increasing both parameters has been shown to reduce net cycle exergy destruction and raise the total exergy efficiency of a simple cycle gas turbine system (SCGT) [1]. TIT is considered the most important parameter when designing gas turbine cycles, as it lowers the combustor and the overall exergy destruction, and improves cycle efficiency [1][2]. High PR and high TIT are preferable, as the heat transfer rate is reduced, the power generated is increased and an overall improvement in the efficiency of a combined cycle power plant is achieved [3]. The irreversibilities of every component of a gas turbine plant were investigated by [4]. Results revealed that the irreversibilities of the combustion chamber and the gas turbine plant finely increase and then decrease with the pressure ratio. These irreversibilities decrease as

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TIT increases. The relation between cycle thermal efficiency and PR for various TIT was shown to be linearly increased at lower PR as well as at higher TIT until a certain value of PR is reached [5][6]. Beyond this value, the thermal efficiency decreases with increased PR, and this limit is dependent on TIT. Increasing TIT increases the net power and thermal efficiency, and this is due to increasing the work produced by the turbine [6]. Optimising the performance of gas turbines through adjusting operating parameters has been studied by [7]. Increased PR results in higher thermal efficiency, whereas increased compressor and turbine isentropic efficiencies guarantee less exergy destruction in the compressor and the turbine, respectively. Increased turbine isentropic efficiency and combustion chamber inlet temperature significantly reduce exergy destruction in the combustion process [7]. The destruction across the combustion chamber has also gained much interest in the literature, where there is an agreement that the combustion chamber is the source of the greatest destruction in the cycle. An analysis based on the second law of thermodynamics was conducted by [8] to explore the exergy destruction due to combustion processes. The study found that the exergy destruction decreases as the combustion process is at higher temperatures but increases as the equivalence ratio decreases from stoichiometric. According to [9], the sources of entropy generation inside the combustion chamber were primarily due to viscous dissipation, heat transfer, mass/diffusion of species and chemical reaction. Heat transfer and chemical reactions were the most significant contributors to entropy production. High thermodynamic inefficiencies in combustion processes are caused by chemical reactions, heat transfer, friction, and mixing [10]. The exergy destruction associated with heat transfer increases with increasing differences between the average temperatures of the fuel and air. Preheating the reactants and reducing the oxidant-to-fuel ratio are two measures that result in a higher temperature of the combustion gases and less exergy destruction per unit of fuel in combustion processes. Advanced exergy-based methods split the exergy destruction into unavoidable/avoidable and endogenous/exogenous. This method provides valuable information such as the potential to improve efficiency by recognising the avoidable exogenous and endogenous sources of exergy destruction [11].

This study performs energy and exergy analyses on a 167 MW gas turbine power plant located in the west part of Tripoli, Libya. The performance of the individual units and the overall plant was quantitively assessed by analysing the effect of changing operating parameters on the exergy destruction, exergetic efficiency and power for each unit in the cycle and the overall cycle. Such an analysis helps evaluate and optimise the overall performance of gas turbine power plants.

2. System Description

The gas turbine power plant with the output power of 167 MW modelled in this study is a simple cycle gas turbine, consisting mainly of an air compressor, a combustion chamber, and a power turbine. A schematic representation of the gas turbine power plant is shown in Fig. 1. In such systems, the air is compressed to a pressure of several bars before entering the combustor where it reacts with the fuel. The high-temperature gasses from the combustion chamber drive the gas turbine. The turbine output shaft is coupled to the air compressor and operates on the same shaft.

The used fuel is diesel oil, air is assumed to enter the compressor at 34^o C (humidity of 60 %) and atmospheric pressure. The stochiometric air-to-fuel ratio is 14.8 [12] and the excess air is assumed to be 69%.

Since the effect of changing the compressor pressure ratio is investigated in this study, the ranges of compressor pressure ratio used have to be within the optimum compressor pressure ratio for the condition of maximum work as described by [13]:

$$PR_{\max work} = \frac{P_{out,\, comp}}{P_{in,\, cpmp}} = \left(\frac{T_{out,\, comp}}{T_{in,\, comp}}\right)^{\frac{Cp_{mh}}{R}} = \left(\sqrt{\left(\frac{T_{out,turb}}{T_{in,comp}}\right)}\right)^{\frac{Cp_{mh}}{R}} \tag{1}$$

The maximum PR was found to be around 14 and therefore, the model was simulated to six folds beyond this value.

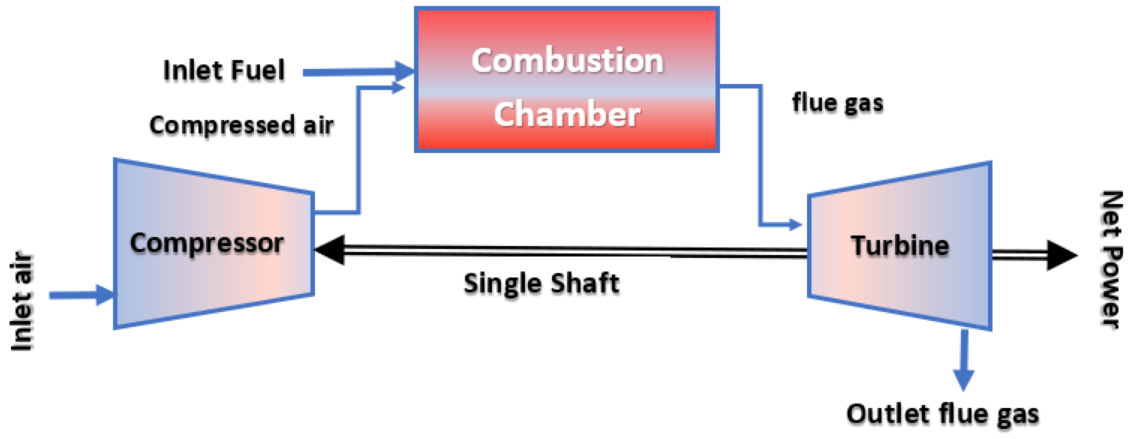

Fig. 1. Schematic representation of a simple cycle gas turbine plant

3. Thermodynamic Methodology:

To model such systems, momentum, energy and exergy analyses are required. The momentum analysis is based on the law of mass conservation which states that during a chemical reaction, the total mass of the products must be equal to the total mass of reactants. Energy and exergy analyses are proposed for modelling the individual units of the gas turbine power plant and for evaluating its overall performance.

3.1. Energy Analysis:

For steady-state flow processes, where both potential and kinetic energy changes are negligible, the energy equation is similar for the different units of gas turbine simple cycles. The energy analysis obeys the first law of thermodynamics and is defined as:

$$\Delta H = \dot{Q} - \dot{W}_s \tag{2}$$

where: ΔH is total enthalpy change (kJ/sec), $\dot{Q}$ and $\dot{W}_s$ are heat rate (kJ/sec) and shaft work (kJ/sec), respectively. For ideal gases, the enthalpy change is independent on the system pressure but is only dependent on the system temperature and is defined as:

$$\Delta H = \dot{n}\, Cp_{mh}(T_{out} - T_{in} \tag{3}$$

where: $\dot{n}$ is the molar flow rate (kmol/sec), T_{in} and T_{out} are the particular unit inlet and outlet temperatures (K), respectively. Cp_{mh} is an ideal gas mean heat capacity specific to enthalpy calculations (kJ/kmol K) and is defined as:

$$Cp_{mh} = \int_{T_{in}}^{T_{out}} \frac{Cp(T)\,dT}{T_{out} - T_{in}} \tag{4}$$

and,

$$\frac{Cp}{R} = A + BT + CT^2 + DT^{-2} \tag{5}$$

and for a mixture of pure gases of constant composition;

$$Cp_{mix} = \sum_{i=1}^{n} y_i Cp_i \tag{6}$$

where: A, B, C and D are constants characteristic of a particular component, R is the gas constant, Cp_{mix} is molar heat capacity for the mixture, Cp_i is the molar heat capacity of a pure component , y_i is the mole fraction of the component i in the mixture and n is the number of components in the mixture.

3.2. Exergy Analysis:

Since an actual process is an irreversible one, the exergy analysis obeys the second law of thermodynamics. The mechanical statement of the second law states that the total entropy change associated with any process is always positive and it approaches zero only when the process is reversible:

$$\dot{W}_{lost} = T_o \Delta S \geq 0 \tag{7}$$

where: $\dot{W}_{lost}$ is the loss of work due to entropy generation, this is also called exergy destruction.

The irreversibility reduces work output in a turbine and increases the work requirement in a compressor. It also implies that the greater the entropy generation, the greater the energy becomes unavailable for work.

3.3. Apply Energy and Exergy Analysis to Individual Units

3.3.1. Compressor and Turbine

For reversible adiabatic (isentropic) process for either a compressor or a turbine, $\dot{Q} = 0$, and $\Delta S = 0$, hence Eq. (1) reduce to:

$$\dot{W}_s(isentropic) = -\Delta H(isentropic) = \dot{n}\ Cp_{mh}\left(T_{out,isnt} - T_{in}\right) \quad (8)$$

The entropy change is calculated using:

$$\Delta S(isentropic) = Cp_{ms} \ln\frac{T_{out,isnt}}{T_{in}} - R\ln\frac{P_{out}}{P_{in}} = 0 \quad (9)$$

where: C_{pms} is the ideal gas mean heat capacity specific to entropy calculations and is defined as:

$$Cp_{ms} = \int_{T_{in}}^{T_{out}} \frac{Cp(T)\ dT/T}{\ln(T_{out,isnt}/T_{in})} \quad (10)$$

The isentropic outlet temperature $T_{out,isnt}$ is determined first for a mechanically reversible process that accomplishes the same change of state using an iterative solution applied to Eq. (9). The shaft work, in this case, is the isentropic one $W_{s,\ isnt}^{\cdot}$ and is calculated using Eq. (8).

The actual temperature is determined by either multiplying or dividing the second term of the right-hand side of Eq.(11) by isentropic efficiency τ_{isnt} and this depends on whether the process produces or requires work. The shaft work in this case is the actual one $W_{s,\ actual}^{\cdot}$ which is an irreversible process and is calculated using Eq. (12).

$$T_{out} = T_{in} + \left(T_{out,isnt} - T_{in}\right) \times \frac{\tau_{isnt,\ turbine}}{\tau_{isent,\ ,\ compressor}} \quad (11)$$

and,

$$\dot{W}_s(actual) = -\Delta H(actual) = \dot{n}\ Cp_{mh}(T_{out} - T_{in} \quad (12)$$

Thus, a compressor's isentropic efficiency τ_{isnt} is:

$$\tau_{isnt}(compressor) = \frac{W_{s,\ isntropic}^{\cdot}}{W_{s,\ actual}^{\cdot}} \quad (13)$$

and a turbine's isentropic efficiency τ_{isnt} is:

$$\tau_{isnt}(\ turbine) = \frac{W_{s,\ actual}^{\cdot}}{W_{s,\ isentropic}^{\cdot}} \quad (14)$$

The actual shaft work is greater than the isentropic shaft work for compressors and is less than the isentropic shaft work for turbines. The isentropic efficiency for compressors and turbines is usually in the range of 0.7 to 0.8% [14].

Since a completely reversible process, in which all the changes within the process including heat transfer can never exist, a completely reversible process is only required for the estimation of the ideal work (reversible work) associated with it. Ideal work is the minimum work consumed by a compressor or the maximum work produced by a turbine. The only connection between ideal and actual work is that it brings the same change in temperature and pressure conditions.

$$\dot{W}_{ideal} = \left(T_o\Delta S - \Delta H_{actual}\right) \quad (15)$$

where: T_o is the surrounding temperature which is considered as 298 K.

Therefore, the exergetic efficiency $\tau_{exergetic}$ for a compressor or a turbine is defined as

$$\tau_{exergetic}(compressor) = \frac{W_{s,\ ideal}^{\cdot}}{W_{s,\ actual}^{\cdot}} \quad (16)$$

and,

$$\tau_{exergetic}(\ turbine) = \frac{W_{s,\ actual}^{\cdot}}{W_{s,\ ideal}^{\cdot}} \quad (17)$$

As a result of irreversibility or exergy destruction, some energy that becomes unavailable for work is lost.

$$\dot{W}_{lost} = \left(\dot{W}_{ideal} - W^{\cdot}_{s,\ actual}\right) = T_o\Delta S \quad (18)$$

3.3.2. Combustion Chamber

For an adiabatic reversible process applied to combustion chambers, no heat is added or removed, $\dot{Q} = 0$ and since no shaft work is produced or required by this unit, Eq. (1) reduces to:

$$\Delta H = 0 \quad \text{or}$$

$$\dot{m}_f LHV + \dot{n}_{in}Cp_{mh}\left(T_{in} - T_o\right) = \dot{n}_{out}Cp_{mh}\left(T_{out} - T_o\right) \quad (19)$$

where: $\dot{m}_f$ is fuel mass flow rate (kg/sec), $\dot{n}_{in}$ and $\dot{n}_{out}$ are the molar flow rates of the compressed air and the combustor flue gas, respectively. LHV is the lower heating value (KJ/kg). Theoretically, combustion champers run at constant pressure, but practically there is a small pressure drop within this unit. Hence P_{in} and P_{out} are slightly different.

Solving equation (19) by an iterative solution for T_{out} which is in this case, the maximum temperature that can be achieved as a result of combustion or $T_{out} = T_{max}$. This is also known as the adiabatic flame temperature in the case of stoichiometric reactions. For an irreversible process, the exergy equation described by Eq. (7) is always positive. For a non-adiabatic reversible process, the system is described by an equation similar to Eq.(15) and is defined as:

$$\dot{Q}_{rev} = T_o \Delta S - \dot{n}_{out} Cp_{mh} \left(T_{out} - T_{in} \right) \quad (20)$$

and,

$$\Delta S = Cp_{ms} \ln \frac{T_{out}}{T_{in}} - R \ln \frac{P_{out}}{P_{in}} \quad (21)$$

4. Results

In this study, a gas turbine power plant was modelled with varying compressor pressure ratios and different compressor and turbine isentropic efficiencies. The relationship between the compressor pressure ratio (PR) and the compressor & turbine isentropic efficiency was analysed with the following parameters: outlet temperature, exergy destruction, exergetic efficiency and power required or produced for the individual unit of a gas turbine power plant. Figure 2 (a-d) represents the compressor PR and compressor isentropic efficiency relationship with the previous parameters. When the compressor isentropic efficiency was 85 % and the compressor pressure ratio was raised from 8 to 14, the compressor outlet temperature showed an 18% increase from 590 K to 695 K. The exergy destruction increased by 2.1 MW (19 %) from 11.1 MW to 13.2 MW, and the power required to operate the compressor increased by 53.1 MW (38 %) from 138.5 MW to 191.6 MW at the same previous conditions. However, the exergetic efficiency showed a slight increase from 88.5% to 89.2 % despite the significant increase in the outlet temperature with pressure ratio. This exergetic efficiency's slight increase has also been reported by [15]. Decaying of the compressor isentropic efficiency at constant PR resulted in higher compressor outlet temperature, exergy destruction and power required. The outlet temperature increased by 6% from 590 K to 628 K when the compressor isentropic efficiency was reduced from 85% to 75 % at PR of 8. The exergy destruction increased by 9.3 MW (83.5 %) from 11.1 MW to 20.4 MW, and the power required increased by only 19 MW (14 %) from 138.5 MW to 157.5 MW.

For the combustion chamber, Figure 3(a and b) represents the relationship between the compressor PR and the compressor isentropic efficiency with the combustor outlet temperature and exergy destruction. When the compressor pressure ratio was raised from 8 to 14 at a constant compressor isentropic efficiency of 85 %, the outlet temperature from the combustion chamber showed a 7% increase from 1331 K to 1424 K but, the combustor destruction was reduced by about 13.84 MW (10 %) from 135.65 MW to 121.81 MW. When the compressor pressure ratio was constant at PR of 8 and the compressor isentropic efficiency decayed from 85 % to 75 %, higher combustor outlet temperature and lower exergy destruction were detected. The combustor outlet temperature was increased from 1331 K to 1365 K (2.5 %), whereas the combustor destruction was lowered by 5.34 MW (4 %) from 135.65 MW to 130.31 MW.

The performance of the turbine and hence the efficiency of the complete cycle were examined at different turbine isentropic efficiencies. Figure 4(a-d) shows the relationship between the compressor pressure ratio and the turbine isentropic efficiency with outlet temperature, exergy destruction, exergetic efficiency and power produced by the turbine. The turbine outlet temperature was reduced and the turbine power produced was improved as the compressor pressure ratio and the isentropic efficiency of the turbine were increased at constant compressor isentropic efficiency. Operating at high-pressure ratio conditions resulted in higher turbine exergy destruction but unexpectedly higher turbine power produced. For instance, when the turbine isentropic efficiency was 90 % and the compressor pressure ratio increased from 8 to 14, the outlet temperature decreased by 4.4 % from 861.5 K to 823.2 K. The exergy destruction increased by 3.5 MW (34 %) from about 10.2 MW to 13.7 MW, and the power produced increased by 76.76 MW

(28.52 %) from 269.1 MW to 345.86 MW. Decaying of the turbine isentropic efficiency at constant PR conditions and compressor isentropic efficiency resulted in higher outlet temperature and exergy destruction, but lower exergetic efficiency and power produced. The outlet temperature increased by 6 % from 861.5 K to 913.7 K when the turbine isentropic efficiency decayed from 90 % to 80 % at PR of 8. The exergy destruction increased by 9.7 MW (49 %) from 10.2 MW to 19.9 MW, and the power produced decreased by 28.9 MW (12 %) from 269.1 MW to 240.2 MW.

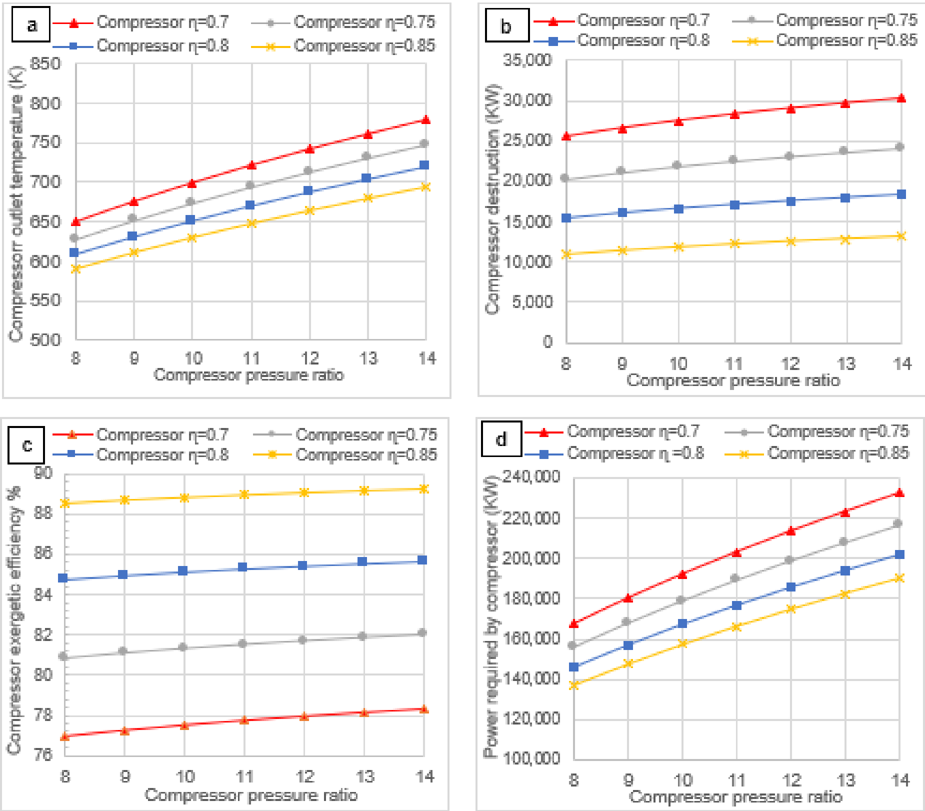

Fig. 2. Effect of compressor pressure ratio and different compressor isentropic efficiency on the compressor: (a) outlet temperature; (b) exergy destruction; (c) exergetic efficiency; (d) power required.

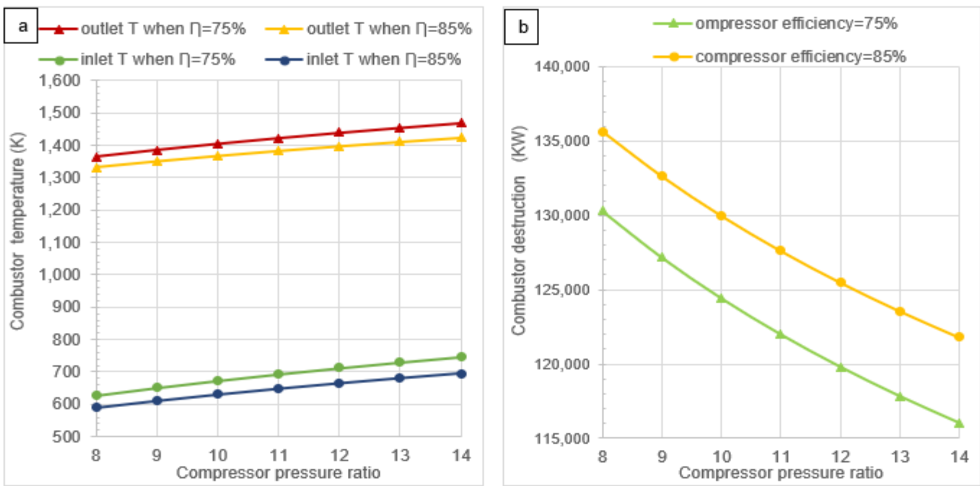

Fig. 3. Effect of compressor pressure ratio on combustion chamber (a) inlet and outlet temperatures; (b) exergy destruction at 85% and 75% compressor isentropic efficiency.

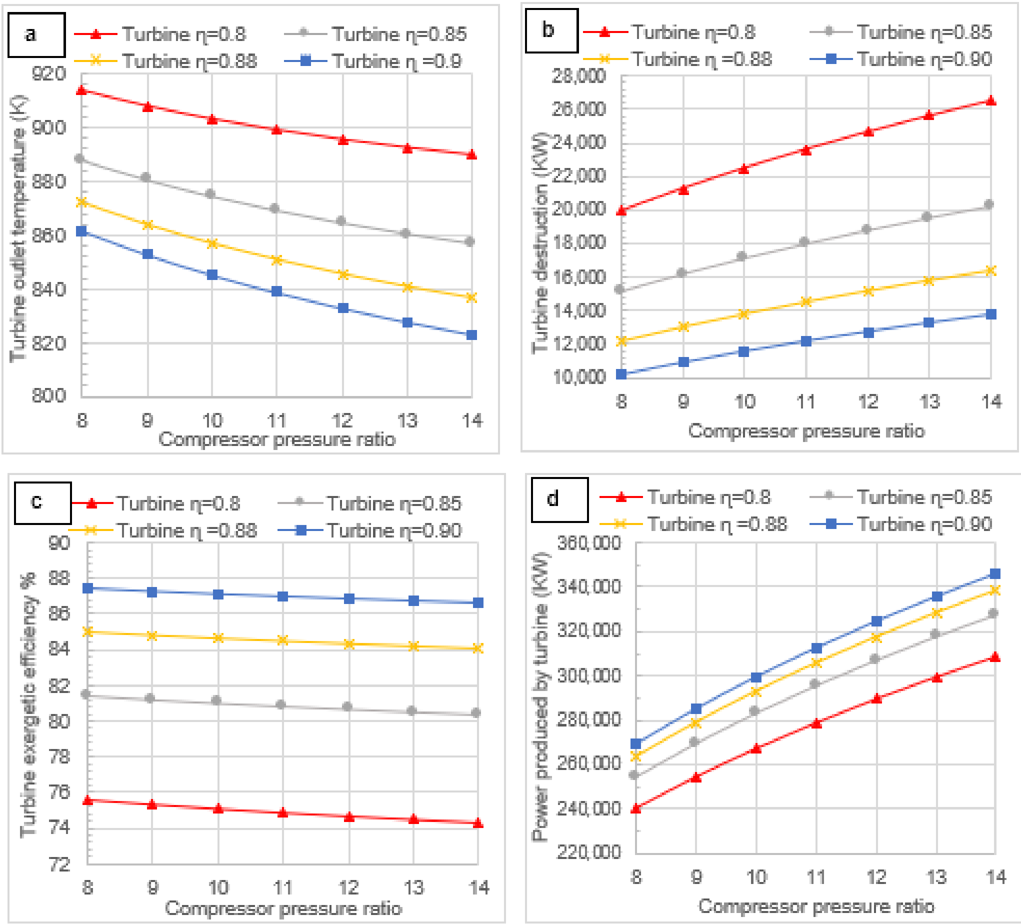

Fig. 4. Effect of compressor pressure ratio and turbine isentropic efficiency on turbine: (a) outlet temperature; (b) exergy destruction; (c) exergetic efficiency; (d) power produced; (the compressor isentropic efficiency is 85%).

The net cycle efficiency, the net power generated and the net exergy destruction by the gas turbine power plant were analysed at 85% and 75% compressor isentropic efficiencies and various PR conditions and turbine isentropic efficiency as shown in Figs. (5 and 6). The figures clearly showed that increasing PR led to an improvement in net cycle efficiency and net power generated and reduced net cycle destruction. Figure 5 showed that with 85% compressor and turbine isentropic efficiencies and raising the pressure ratio from 8 to 14, the net cycle efficiency was boosted from 27.96% to 32.7%. Moreover, the net output power was enhanced by 19.72 MW while the net destruction was reduced by 6.62 MW. However, the overall performance declined when the power plant was operated at 75% compressor isentropic efficiency. Figure 6 showed that with 75% compressor and 85% turbine isentropic efficiency and raising the pressure ratio from 8 to 14, the net cycle efficiency was increased from about 24.5% to 28.3%. The net output power was enhanced by only 16 MW, while the net power destruction was reduced by 5.40 MW.

Furthermore, operating at a constant PR of 8 and constant turbine isentropic efficiency of 85 % and considering a lowering in compressor isentropic efficiency from 85% to 75% would reduce the net exergetic efficiency from 27.96 % to 24.5 %. The net power would also decrease by 14.58 MW while the net exergy destruction would increase by 3.8 MW. On the other hand, operating at a constant PR of 8 and constant compressor isentropic efficiency of 85% and considering a lowering in Turbine isentropic efficiency from 90 % to 80 % would reduce the net exergetic efficiency from 31.44 % to 24.48 %. The net power would also decrease by 28.9 MW while the net exergy destruction would increase by 9.7 MW.

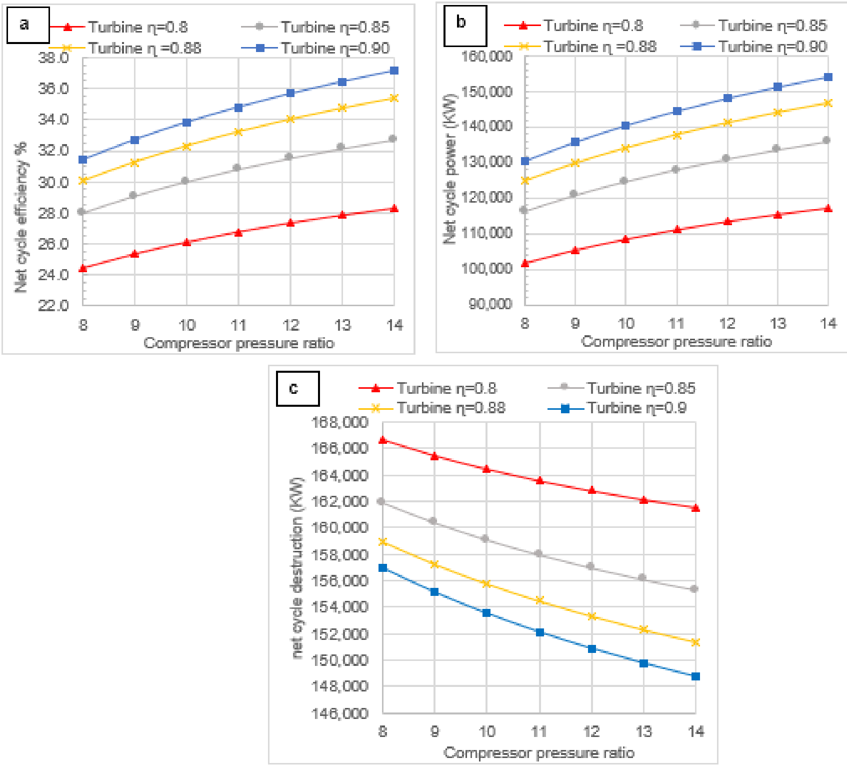

Fig. 5. Effect of compressor pressure ratio and turbine isentropic efficiency on the gas turbine cycle: (a) net efficiency (b) net power produced and (c) net exergy destruction. (the compressor isentropic efficiency is 85%)

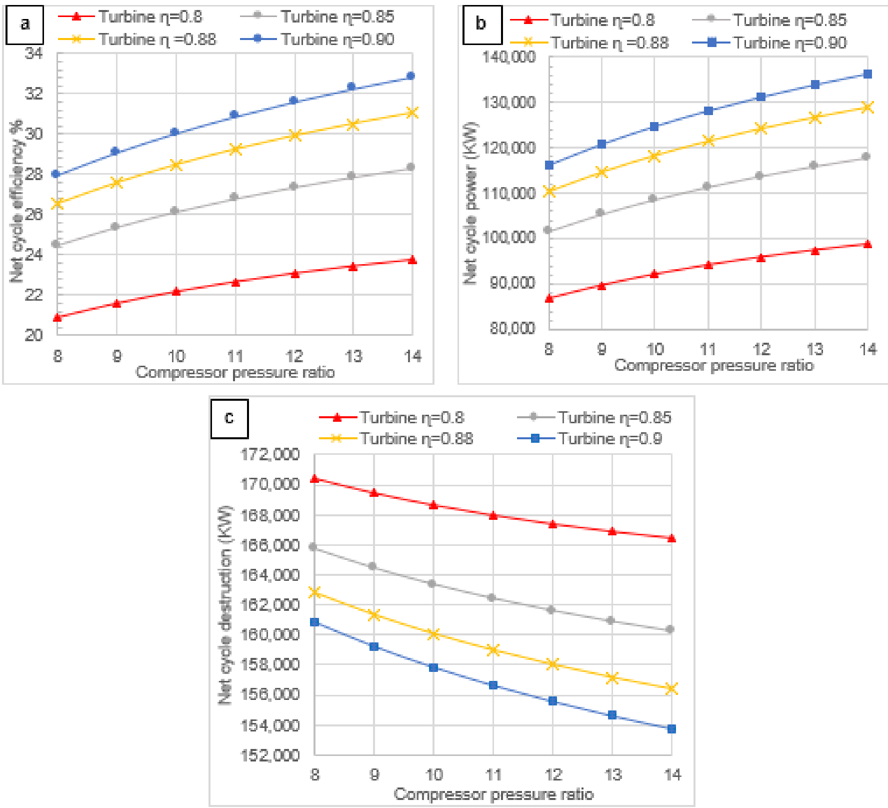

Fig. 6. Effect of compressor pressure ratio and turbine isentropic efficiency on the gas turbine cycle: (a) net efficiency (b) net power produced and (c) net exergy destruction. (the compressor isentropic efficiency is 75%)

The amount of energy that is added to (+ sign) or reduced from (- sign) the individual units and the overall cycle when exhibiting different operating conditions were illustrated in Tables (1) to Table (3). These tables summarise the effect of increasing compressor pressure ratio, and decaying of the compressor or the turbine isentropic efficiencies on the overall performance of the gas turbine power plant. Table (1) shows that energy is added (+ sign) to the exergy destruction for both, the compressor and the turbine when PR is increased at constant compressor and turbine isentropic efficiencies. The energy is also added (+ sign) to the power required to operate the compressor, the power produced by the turbine and the net cycle power. Table (1) also shows that the only reduced amount of energy (- sign) was due to the exergy destruction across the combustion chamber. Therefore, the net cycle destruction and the net power are affected in this case, by the decline of the combustor exergy destruction and the increase of the power produced by the turbine, respectively.

Table (2) shows that energy is added (+sign) to the compressor exergy destruction and power required to operate the compressor when the efficiency of the compressor is decayed from 85% to 75% at constant compressor PR and constant turbine efficiency. Table (2) also shows that the amount of combustor and turbine exergy destruction is reduced (- sign) whereas, the power produced by the turbine is increased (+ sign). As a result of that, the net exergy destruction of the cycle is increased (+ sign) whereas, the net power generated is decreased (-sign) when the compressor's efficiency is decayed at constant PR conditions.

Table 1. The amount of energy (MW) that is added (+sign) or removed (-sign) when increasing the compressor PR from 8 to 14 at constant compressor and turbine isentropic efficiencies

Compressor isentropic efficiency	Compressor		Combustion Chamber	Turbine (80% efficiency)		Overall cycle	
	destruction	power Required	destruction	destruction	power produced	net destruction	net power
70 %	+4.71	+65.25	-14.44	+6.56	+74.93	-3.16	+9.68
75 %	+3.78	+60.64	-14.27	+6.58	+72.52	-3.92	+11.88
80 %	+2.92	+56.63	-14.06	+6.59	+70.26	-4.55	+13.63
85 %	+2.11	+53.12	-13.84	+6.61	+68.87	-5.11	+15.75

Turbine (85% efficiency)		Overall cycle		Turbine (90% efficiency)		Overall cycle	
destruction	power produced	net destruction	net power	destruction	power produced	net destruction	net power
+5.06	+79.94	-4.66	+14.69	+3.48	+84.24	-6.24	+18.99
+5.08	+76.69	-5.41	+16.05	+3.49	+80.82	-7.00	+20.18
+5.09	+74.30	-6.05	+17.67	+3.50	+78.30	-7.64	+21.67
+5.11	+72.84	-6.62	+19.72	+3.51	+76.76	-8.21	+23.64

Table 2. The amount of energy (MW) that is added (+sign) or removed (-sign) when the efficiency of the compressor is decayed from 85% to 75% at constant compressor PR and turbine isentropic efficiency

PR	Compressor (Efficiency decays from 85% to 75%)		Combustor	Turbine (85% efficiency)		Overall cycle	
	destruction	power required	destruction	destruction	power produced	net destruction	net power
8	+9.29	+18.98	-5.34	-0.15	+4.40	+3.79	-14.58
9	+9.66	+20.46	-5.46	-0.17	+5.10	+4.03	-15.35
10	+9.98	+21.83	-5.55	-0.18	+5.78	+4.25	-16.05
11	+10.27	+23.10	-5.62	-0.19	+6.43	+4.46	-16.67
12	+10.52	+24.30	-5.69	-0.20	+7.06	+4.63	-17.24
13	+10.75	+25.43	-5.74	-0.22	+7.67	+4.80	-17.76
14	+10.96	+26.50	-5.78	-0.23	+8.25	+4.95	-18.25

Table 3. The amount of energy (MW) that is added (+sign) or removed (-sign) when the efficiency of the turbine is decayed from 90% to 80% at constant compressor PR and compressor isentropic efficiency

PR	Compressor (85% Efficiency)		Combustor	Turbine (Efficiency decays from 90% to 80%)		Overall cycle	
	destruction	power required	destruction	destruction	power produced	net destruction	net power
8	0	0	0	+9.70	-28.90	+9.70	-28.90
9	0	0	0	+10.34	-30.54	+10.34	-30.54
10	0	0	0	+10.92	-32.02	+10.92	-32.02
11	0	0	0	+11.45	-33.36	+11.45	-33.36
12	0	0	0	+11.93	-34.59	+11.93	-34.59
13	0	0	0	+12.38	-35.73	+12.38	-35.73
14	0	0	0	+12.80	-36.79	+12.80	-36.79

Table (3) displays the efficiency decay of the turbine, assuming that the pressure ratio and the compressor isentropic efficiency remain constant. No change in exergy destruction across the compressor and the combustor are found. This means that no added or reduced exergy losses lead to a stable power required to operate the compressor. However, the turbine's exergy destruction increases (+ sign), whereas, the turbine power produced decreases (- sign). As a result, the net cycle destruction is increased (+ sign), and the net power is decreased (- sign) by the same amount of energy exhibited by the turbine.

5. Results and Discussion

The results present the relationships between the individual unit's outlet temperature, exergy destruction rate, exergetic efficiency and power required or produced with the compressor pressure ratio and the compressor & the turbine isentropic efficiencies. It was observed that the compressor outlet temperature was increased with the increase of PR at constant compressor isentropic efficiency. This increases the temperature difference between the inlet and outlet temperature of the compressors assuming fixed inlet temperature to the compressor. As a result, the exergy destruction by the compressor is increased. The increased PR yielded also, an increase in the combustor outlet temperature but a reduction in the temperature difference between the outlet and the inlet temperatures to the combustor. This led to a clear decline in the exergy destruction inside the combustion chamber with PR as shown in Fig. 3(b). As a result, the turbine's outlet temperature is decreased and the power produced by the turbine and the net power are increased as seen in Fig. 4(a and d) and Fig. 5(b) & Fig. 6(b).

Considering the decaying of the compressor isentropic efficiency at constant PR and constant turbine isentropic efficiency, the compressor and combustor outlet temperatures in this case were increased, see Fig. 3(a). As a result, the combustor exergy destruction was decreased as shown in Fig. 3(b) and power produced by the turbine was increased despite the compressor's efficiency decay and this is quite interesting. The decline in the combustor's exergy destruction and the rise in the turbine's power produced in this case is due to the compressor outlet temperature rise which is also the turbine inlet temperature (TIT). Nevertheless, the net cycle's exergetic efficiency and net power declined, compare Fig. 5(a and b) to Fig. 6(a and b). According to [16], the power and efficiency of gas turbine cycles result from a complex interaction of different turbomachines and a combustion system. As the turbine drives the compressor, reducing the compressor's efficiency at constant compressor PR, increases the percentage of power required to drive the compressor as shown in Fig. 7(a). This also increases the power produced by the turbine as discussed earlier. Therefore the reduction in the overall power generated is because of an increased percentage of power consumed by the compressor as it decays. This suggested that reducing the combustor exergy destruction due to increasing the turbine inlet temperature is not always crucial for improving gas turbine plant performance. However, The effect of turbine inlet temperature and compressor pressure ratio on Bryton cycle performance was analysed by [13] and claimed that TIT is not involved in affecting the efficiency and that only increasing PR can augment gas turbine thermal efficiency.

Considering the decaying of the turbine's efficiency when operating the compressor at constant PR and constant compressor isentropic efficiency increases both the exergy destruction by the turbine as shown in Fig. 4(b) and the percentage of power required to drive the compressor as shown in Fig. 7(b). This causes a decline in both the power produced by the turbine as seen in Fig. 4(d) and the net cycle power as seen in Figs. 5(b) and 6(b).

Tables (1) to (3) summarize the results and show that the net cycle power is improved only when operating at increased PR at constant compressor and turbine efficiencies but it is declined by the decay of any of the compressor or the turbine efficiencies. The drop in net cycle exergy destruction (- sign) when the compressor pressure ratio is increased at constant compressor and turbine isentropic efficiencies is due to the reduced (- sign) exergy destruction across the combustor which is in this case is higher than the amount of energy that is added (+ sign) to both the compressor and the turbine exergy losses. This led to an improvement of the power produced by the turbine and hence to the overall cycle net power. The rise in net cycle exergy destruction (+ sign) when the compressor isentropic efficiency decayed at constant PR and turbine isentropic efficiency is due to the added (+ sign) exergy destruction of the compressor. The compressor exergy destruction in this case is higher than the reduced (- sign) exergy destruction exhibited by both the combustor and turbine. The net power generated in this case is declined (- sign) and this is due to the increased amount of energy (+ sign) that is required to operate the compressor. The net cycle exergy destruction also showed an increase (+ sign) when the turbine isentropic efficiency decayed at constant PR and compressor isentropic efficiency. This is affected by the increase (+ sign) in turbine exergy destruction. The net power generated by the cycle is reduced (- sign) in this case due to the reduced amount of energy (- sign) that the turbine produces. Tables (2 and 3) show that turbine decay has a more significant negative impact on the cycle's net power than compressor decay (about double). This negative effect is more pronounced at higher compressor pressure ratios. This supports [17] who claimed that improving the turbine's isentropic efficiency has a greater positive impact on the the net work per cycle and the thermal efficiency of a Brayton cycle than the same improvement in the compressor's isentropic efficiency. This finding is also consistent with [18] who examined an equation for calculating the thermal efficiency of a simple cycle gas turbine which exhibits irreversible compression and expansion processes and concluded that the sensitivity is greater to expansion inefficiency.

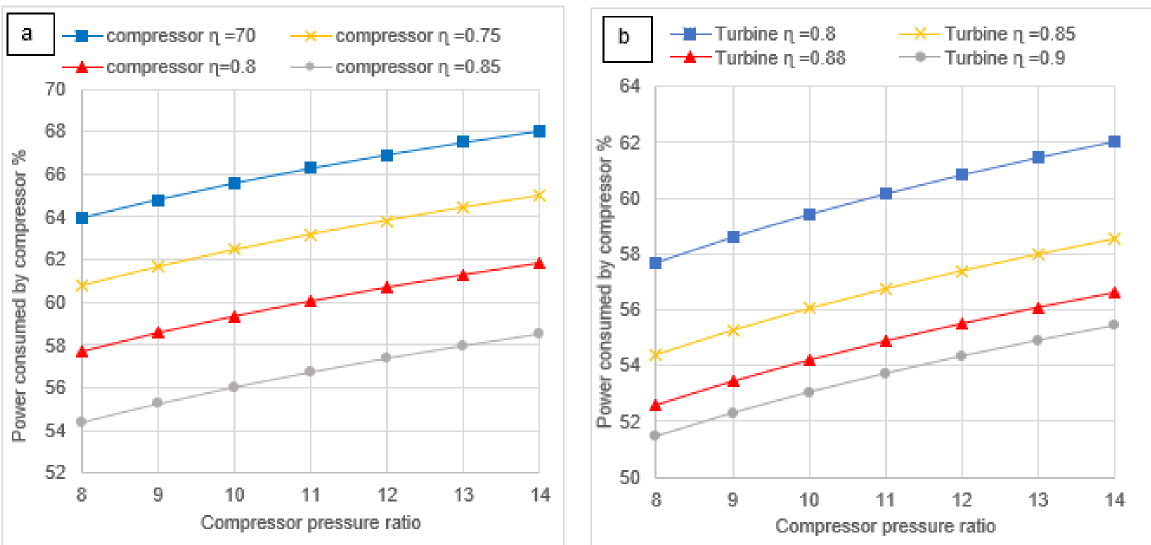

Fig. 7. Effect of compressor pressure ratio on the percentage power consumed by the compressor from the gross power; (a) at various compressor isentropic efficiencies and constant turbine isentropic efficiency of 85%, (b) at various turbine isentropic efficiencies and constant compressor isentropic efficiency of 85%

6. Conclusions

This study analyses the effect of altering operating parameters on the performance of simple cycle gas power plants. The analysis focuses on three scenarios: (1) Increasing compressor pressure ratio when compressor and turbine isentropic efficiencies remain constant. (2) Decreasing compressor isentropic efficiency while keeping the compressor pressure ratio and turbine isentropic efficiency constant. (3) Decreasing turbine isentropic efficiency while keeping the compressor pressure ratio and compressor isentropic efficiency constant.

The study found that the net cycle exergy destruction is decreased when the plant operated at an increased compressor pressure ratio while maintaining constant compressor and turbine isentropic efficiencies. This was due to the reduced exergy destruction across the combustor caused by a reduction in the temperature difference between the inlet and outlet to the combustor. As a result, the turbine's power production is improved and the net cycle power is increased. When the isentropic efficiencies of the compressor or the turbine are decreased at a constant compressor pressure ratio, the net cycle exergy destruction increases, leading to a decline in net power generated. These effects are caused by the increased compressor exergy destruction, which requires more power to operate, or by the increased turbine exergy destruction, which results in decreased power produced by the turbine. However, the results show that declining turbine isentropic efficiency has a more negative impact than declining compressor isentropic efficiency on the net power generated by the plant.

The findings also indicate that turbine inlet temperature is affected by other operating parameters, making it a dependent variable. Therefore, it has a less important role in studying the performance of power plants. Moreover, increasing the turbine inlet temperature does not always improve its performance.

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