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Estimation of Maximum Cycle Temperature and Pressure Ratio for Optimum Design and Operations of Gas Turbine Power Plant

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Authors' contributions

This work was carried out in collaboration among all authors. Author JBB designed the study, managed the literature searches, wrote the protocol and wrote the first draft of the manuscript. Author IFO managed the analyses of the study. Author DAA Performed the statistical analysis. All authors read and approved the final manuscript.

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ABSTRACT

Aims: The performance of gas turbine is influenced by a number of factors which may be classified under three main headings: design limiting conditions, environmental dependent conditions and system respond condition. The design limiting conditions have been found to have influenced gas turbine performance most. In this study, two design limiting conditions (pressure ratio and maximum cycle temperature) were evaluated and estimated for optimal design and operations of gas turbine.

Study Design: Hierarchical order in selecting the two design condition parameters was proposed.

Methodology: Steady state energy and exergy concept was used to model the behavior of the plant.

Results: Result obtained indicated that at maximum cycle temperature of 1173K, 1273K, 1373K, 1473K, 1573K, and 1673K,maximum pressure ratio for optimal performance were 19, 24, 29, 36, 44 and 51 respectively.

Conclusion: These results provide insight into the performance behavior of gas turbine and also serve as a guide for operations and design optimization.

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Keywords: Gas turbine; performance design and operations; optimization; estimation.

NOMENCLATURES

C_p : Specific heat at constant pressure [kJ/kg.K]
C_{pa} : Specific heat capacity of air at constant pressure [kJ/kg.K]
C_{pg} : Specific heat capacity of flue gas at constant pressure [kJ/kg.K]
LHV :Lower Heating Value [kJ/kg]
HR :Heat Rate [kJ/kWh]
SFC :Specific Fuel Consumption [kg/kWh]
m_a :Mass flow rate of air [kg/s]
m_f :Mass flow rate of fuel [kg/s]
m_g :Mass flow rate of flue gas (product gas) [kg/s]
p :Pressure [bar]
T :Temperature [K]
r_p :Compression pressure ratio
γ :Isentropic index of compressed air
γ_g :Isentropic index of flue gas
e_{ph} :Specific exergy [kJ/kg]
$\dot{E}x$:Exergy [MW]
$\dot{E}x^{ph}$:Physical exergy [MW]
$\dot{E}x^{ch}$:Chemical exergy [MW]
$\dot{E}x^{p}$:Potential exergy [MW]
$\dot{E}x^{k}$:Kinetic exergy [MW]
$\dot{E}_D$:Exergy distruction [MW]
e_f^{-ch} :Specific chemical exergy of fuel [kJ/kg]
s :Specific entropy [kJ/kg.K]
h :Specific entalpy [kJ/kg.K]
R :Gas constant
η :Efficiency [%]
η_p :Polytropic efficiency [%]
η_c :Compressor isentropicefficiency [%]
η_t :Turbine isentropic efficiency [%]
W_c :Compressor work [MW]
W_{gt} :Turbine work [MW]
W_{net} :Net work [MW]
W_r :Workratio
ET :Extrapolated Temperature [K]
CET :Compressor Exit Temperature [K]
ATTI :Actual Temperature at Turbine Inlet for actual fuel supply [K]
FFS :Fixed Fuel Supply [m^3/s]
AFS : Actual Fuel Supply [m^3/s]
CV : Calorific Value [kJ/kg.K]

ABBREVIATIONS

MCT : Maximum Cycle Temperature
PR : Pressure Ratio or Compression Ratio
GT : Gas Turbine
ASCE : Air Standard Circle Efficiency
SFC : Specific Fuel Consumption
IAT : Inlet Air Temperature
CDT : Compressor Discharge Tempearture
CV : Calorific Value
DLE : Dry Low Emission

1. INTRODUCTION

With the rising demand for steady electricity supply for industrial and utility services, power generation companies are considering capacity increase. One of such considerationsis through the generation of electric power via gas turbine plant. The choice for gas turbine may be due to some of its advantages which include: ability to provide clean energy(i.e., environmentally friendly), easy to setup, compactness, ability to be turned on and off within minutes, supplying power during peak or unscheduled demand and short commissioning period etc. With gas turbine becoming a major choice for increased power generation, there is need to seek critical evaluation of the performance of this plant so that informed decision can be taken while seeking an optimal design and operations of this plant.

Gas turbine performance may be evaluated base on three major indexes which are: thermal efficiency, work ratio and specific fuel consumption. However, these performance measures (indexes) may be affected by a number of factors which are: compressor inlet temperature, compressor discharge temperature, pressure ratio, Maximum cycle temperature (MCT), compressor and turbine isentropic efficiency etc. Here, attempt shall be made to classify some of these factors into three categories.The compressor inlet temperature may be classified under *environmental dependent condition*, the compressor discharge temperature may be tagged under*system respond condition* and others like pressure ratio, maximum cycle temperature, compressor and turbine isentropic efficiency may be classified as *design limiting conditions*. These factors tend to affect gas turbine performance differently.

For the case of the compressor inlet temperature, studies have shown that a decrease in this temperature increases the thermal efficiency of the plant. Ebadi et al. [1] conducted an energy analysis of a gas turbine with a nominal capacity of 116 MW and concluded that the impact of rising input temperature on the gas turbine improved total exergetic efficiency of the gas turbine cycle, and reduced exergy destruction. Thermodynamic appraisal of gas turbine performance in the Niger Delta region of Nigeria was carried out [2]. It was concluded that a degree centigrade rise in ambient temperature was responsible for the following: 0.83% reduction in power output, 0.17% increase in heat rate and 0.40% decrease in required air flow rate [3].

The compressor discharge temperature is measured at the end of the compressor outlet. When air passes through the compressor, it compresses and the compressed air temperature increases. Based on the compression ratio has, the isentropic index and the inlet air temperature (IAT) to the compressor, the compressor discharge temperature (CDT) is determined. The parameter CDT, is systemic because it depends on the system's condition. Decreasing this temperature by intercooling between compressor stages have been explored and results have shown improvement in the work ratio. Similar results have also been found by reheating the flue gases between turbine stages. However, there are serious consequences on this as it can cause a decrease in the cycle efficiency. However, when intercooling and reheating are used in conjunction with a heat exchanger then intercooling and reheating increase both the work ratio and the cycle efficiency [4].

The compressor and turbine isentropic efficiency also affect the performance of gas turbine power plant.The compression ratiohas been seen to inturn affect the compressor and turbine isentropic efficiencies as well. Studies have shown that as the compression ratio increases the compressor isentropic efficiency decreases while the turbine isentropic efficiency increases [5]. However, there is a limiting value for the two isentropic efficiencies as the compression ratio approaches zero. This limiting value is called "polytropicefficiency". The polytrophic efficiency is usually constant for a given system. According to [6], an increase in the compressor isentropic efficiency, turbine isentropic efficiency or both increases the thermal efficiency of combined cycle gas turbine.

For the purpose of this study, two design limiting conditions were considered and evaluated for optimal design and operations of gas turbine. These are: maximum cycle temperature and pressure ratio. The overall performance of gas turbine greatly depends on these two main factors. Setting higher values for these factors are often desirable for improved performance but care should be taken as certain limitations exit in choosing values of these factors.Thus, hierarchical order in selecting valuesof these factors is proposed as shown in Fig. 1.

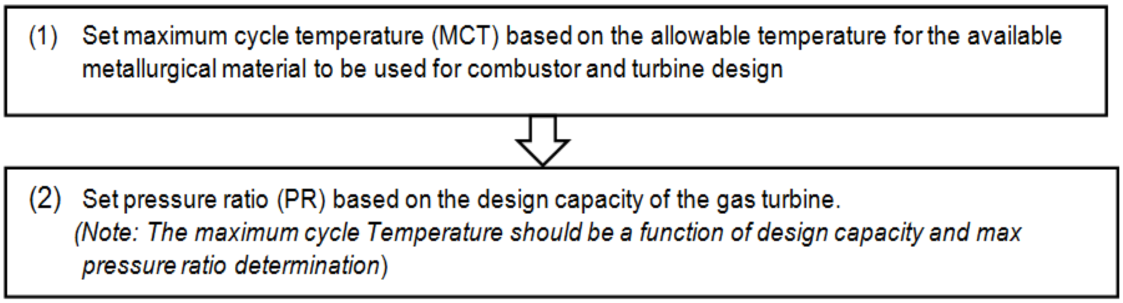

Fig. 1. Hierarchical consideration of MCT and PR for optimal design

1.1 Maximum Cycle Temperature

The MCT is one of the most critical parameters which influences gas turbine performance. It is obvious that changes in MCT influence the turbine work output and consequently affects the net-work output. Higher MCT produces higher turbine work output and hence high net-work output. Thus, for better gas turbine performance, it is desirable to have higher turbine inlet temperature. Both the power output and the thermal efficiency can be improved by increasing the MCT. Although increased MCT gives a higher thermal efficiency and a higher power output, there are some metallurgical challenges for increasing MCT. The main issue here is the material property limitation. The turbine blade elements, casing, hub and combustor elements cannot withstand higher temperature above some thresholds. Therefore, the gas path components undergo thermal and mechanical stresses at above threshold temperature, [7].

With the development of gas turbine technology there are some methods to overcome this limitation. Currently available material failure mitigation methods for gas turbine are air cooling, steam or water injection, use of special material such as high-performance alloys, use of single-crystal material or use of thermal barrier coating [7]. The other problem that limits the MCT of gas turbine is the environmental regulations due to NOx emission control. Normally, thermal NOx is generated in high temperature environment. At temperature above 1200^{o}C, the formation of thermal NO_x increases rapidly. With the introduction of water or steam injection combustion, lean premixed combustion, Dry Low Emission (DLE), catalytic combustion,swirl premixed combustion or compartment combustion design etc. the NOx emission is controlled successfully. Presently in most modern gas turbines the NOx emission reduces to single digit ppm value using these methods [8].

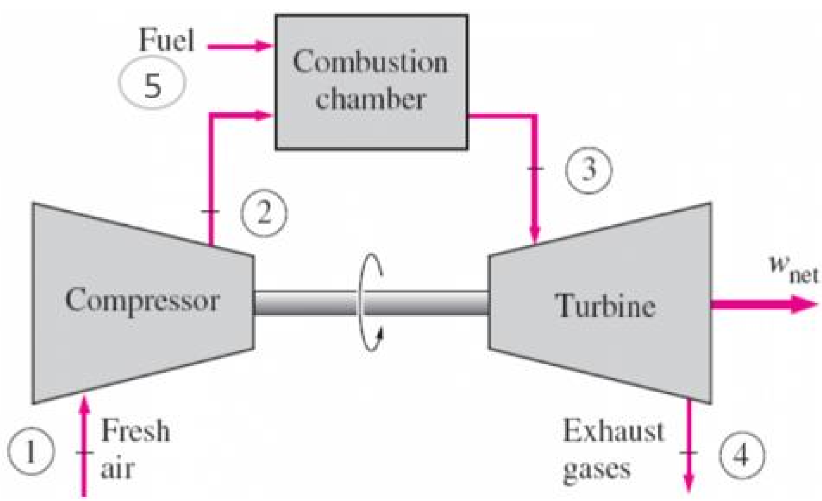

Fig. 2. Schematic of simple gas turbine power plant [10]

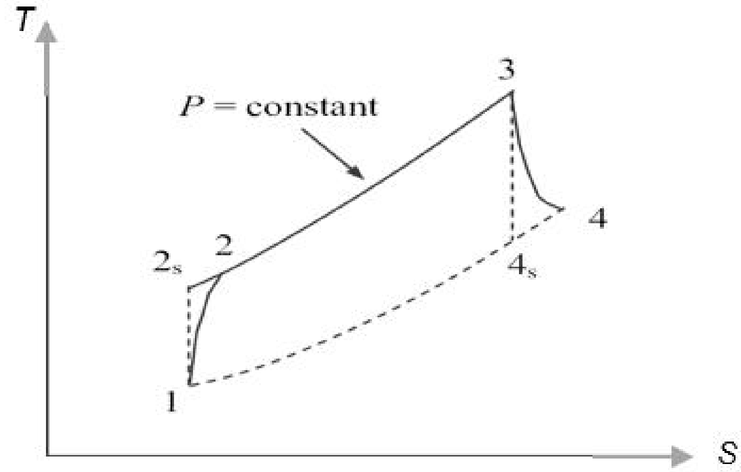

Fig. 3. Open gas turbine cycle [11]

1.2 Pressure Ratio (R_p)

The pressure ratio is a condition created during the design stage of the compressor. Most gas turbine compressors have up to seventeen compression stages which adds up to give the final compression ratio of a turbine. The Brayton cycle efficiency expression shows that an increase in pressure ratio increases the cycle efficiency. [9], claimed that thermal efficiency of gas turbine increases with the pressure ratio, but after certain value of the pressure ratio the efficiency decreases.

Thus, this suggest that the choice for pressure ratio value must be within the range of optimal performance. It is important to note here that as the pressure ratio changes, the isentropic efficiency of compressor and turbine change. Thischanges also affect the overall performance of the plant.

2. MATERIALS AND METHODS

2.1 System Description

Unit 2 of simple gas turbine power plant in Sapele, Nigeria is considered. The gas turbine (GT) power plant consists of a rotary type air compressor and a turbine, coupled along with a combustion chamber using a single shaft. Auxiliaries, such as cooling fan, water pumps etc., and the generator itself are also driven by the turbine. Other auxiliaries like the starting device, lubrication system, duct system, etc., are also part of the system. For ease of analysis, the steady-state model of the GT is presented in Fig. 2.

2.2 Methodology

The first law of thermodynamics deals with energy analysis and points to the fact that total energy in a system is constant and that it is only the quantity of that "energy" that is converted to another e.g., the conversion of thermal energy to mechanical energy. However, the second law addresses irreversibility and exergy (work potential). It also identifies efficiencies and inefficiencies in components of energy system. Thus, the first and second law of thermodynamics are used in this analysis.

2.3 Model of Design Parameters

Pressure Ratio (r_p): In open gas turbine cycle as shown in Fig. 3. Two isobaric conditions occur along process (2-3) and process (4-1). These conditions give rise to compressor pressure ratio (P_2/P_1) and the turbine pressure ratio (P_3/P_4). For the purpose of this analysis, the two pressure ratios are equal (i.e., $r_p = P_2/P_1 = P_3/P_4$). But in reality, thevariation in pressure loss in compressor and turbine create differences in pressure ratio. Thus:

$$r_p = \frac{P_2}{P_1} = \frac{P_3}{P_4} \tag{1}$$

where, P is pressure and the subscripts 1,2,3,4 represent the position as shown in Fig. 3.

Maximum cycle temperature (MCT): This temperature may be expressed as:

$$MCT = \frac{m_f \times CV}{(m_a + m_f) \times Cp_a} + T_2 \tag{2}$$

Where m_a is mass flow of air, $m_f$is mass flow of fuel, $Cp_a$is specific heat capacity of air and *CV* is the calorific value of the fuel. To get T_3 for a different fixed amount of fuel supply at each T_1, it may be necessary to extrapolate.

For steady flow steady state condition, the extrapolation function is given as:

$$ET = CET + (ATTI - CET)(FFS / AFS) \quad (3)$$

where,

ET = *Extrapolated* Temperature, K,
CET = Compressor Exit Temperature, K,
ATTI = Actual Temperature at Turbine Inlet for Actual Fuel Supply, K,
FFS = Fixed Fuel Supply, m^3/s,
AFS = Actual Fuel Supply, m^3/s.

The exhaust gas temperature, T_4can be determined from the following relation:

$$T_4 = \frac{T_3}{\left(\frac{P_2}{P_1}\right)^{\frac{(\gamma-1)}{\gamma}}} \quad (4)$$

where,$\gamma$is isentropic index of compressed air.

2.4 Exergy Model

Exergy can be divided into four distinct components. The two important ones are the physical exergy ($\dot{E}x^{ph}$) and chemical exergy ($\dot{E}x^{ch}$). In this study, the two other components which are kinetic exergy ($\dot{E}x^{k}$) and potential exergy ($\dot{E}x^{p}$) are assumed to be negligible. The physical exergy is defined as the maximum theoretical useful work that can be extracted as a system interacts with an equilibrium state. The chemical exergy is associated with the departure of the chemical composition of a system from its chemical equilibrium. The chemical exergy is an important part of exergy in combustion processes. When a system is in equilibrium with the environment, the state of the system is called 'dead'and its exergetic value is zero. According to [12], total exergy ($\dot{E}x$) of a stream consist of four main components:

$$\dot{E}x = \dot{E}x^{ph} + \dot{E}x^{ch} + \dot{E}x^{k} + \dot{E}x^{p} \quad (5)$$

As claimed by [13], when a system is at rest relative to the environment ($\dot{E}x^{k}$+$\dot{E}x^{p}$ = 0), the total exergy ($\dot{E}x$) of a stream is defined as:

$$\dot{E}x = \dot{E}x^{ph} + \dot{E}x^{ch} \quad (6)$$

$$\dot{E}x^{ph} = \dot{m}e_{ph} = \dot{m}[(h_1 - h_o) - T_o(s_1 - s_o)] \quad (7a)$$

where,

$$e_{ph} = specific\ exergy = [(h_1 - h_o) - Tos1{-}so \quad (7b)$$

where, *T* represents the absolute temperature, subscript *o* represent ambient condition, subscript 1 representsstate 1 of the system. While *h*and *s*denote the specific enthalpy and entropy respectively.

Equation (7) can be rewritten as:

$$\dot{E}x^{ph} = \dot{m}\left[C_p(T_1 - T_o) - T_o(s_1 - s_o)\right] \quad (8)$$

where,

$$s_1 - s_o = C_p \text{In}\left(\frac{T_1}{T_0}\right) - R\text{In}\left(\frac{P_1}{P_0}\right) \quad (9)$$

where, C_p=specific heat at constant pressure
R= gas constant

Putting equation (9) into (8) we have:

$$\dot{E}x^{ph} = \dot{m}\Bigg[C_p(T_1 - T_o) -$$
$$ToCp\text{In}T1T0{-}R\text{In}P1P0 \quad (10)$$

And the specific heat (C_p) can be obtained in polynomial form as a function of temperature given by the equation [14].

$$C_p = a + bT + cT^2 + dT^3 \quad (11)$$

For many fuels, the chemical exergy can be estimated on the basis of the lower heating value (*LHV*). The relation between the LHV and the chemical exergy for gaseous fuel with formulaC_nH_m based on the atomic composition is given by [15] as:

$$\varphi = \frac{e_f^{-ch}}{LHV} \cong 1.033 + 0.0169\frac{m}{n} - \frac{0.0698}{n} \quad (12)$$

Where,φ is the ratio of fuel exergy and the lower heating value of the fuel and e_f^{-ch} is the fuel exergy. For the majority of gaseous fuel, the value of φ is normally close to 1. For the fuel like methane, φ_{CH4} = 1.06 and for hydrogen fuel, φ_{H2} = 0.985 [16].

The rate of chemical exergy flow can be expressed as:

$$\dot{E}^{ch} = \dot{m} e_f^{-ch} \quad (13)$$

Exergy destruction of each component is given as:

$$\dot{E}_D = \dot{E}x_{in} - \dot{E}x_{out} \quad (14)$$

While the exergetic efficiency of each component of the gas turbine power plant is given as:

$$\eta = \frac{\dot{E}x_{out}}{\dot{E}x_{in}} \quad (15)$$

The inlet exergy rate, the outlet exergy rate, the exergy destruction and exergetic efficiency of each component is calculated by the following equation in Table 1.

The overall exergetic efficiency of the power plant is given as:

$$\eta_{PowerPlant} = \frac{\dot{W}_{net}}{\dot{E}x_3} \quad (16)$$

2.5 Energy Model of Gas Turbine Components

2.5.1 Compressor work

The amount of work done by the compressor on the compressed air may be expressed as:

$$W_c = m_a C_{pa}(T_2 - T_1) \quad (17)$$

Where, m_a is the mass of air flowing into the compressor (kg/s), C_{pa} is specific heat, T_1 is the ambient temperature and T_2 is compressor discharge temperature.

Expressing the compressor work in terms of compressor isentropic efficiency(η_c)and pressure ratio (r_p) we have,

$$W_c = \frac{m_a C_{pa} T_1}{\eta_c}\left[(r_p)^{\left(\frac{\gamma-1}{\gamma}\right)} - 1\right] \quad (18)$$

Where, γ is the isentropic index for air. The compressor isentropic efficiency is expressed as,

$$\eta_c = \frac{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{(\gamma)\eta_p}} - 1} \quad (19)$$

where, η_p is the polytropic efficiency

2.5.2 Turbine work

As the hot gases pass through the turbine blade, it expands and does work on the turbine. The work here may be expressed as.

$$W_{gt} = m_g C_{pg} (T_4 - T_3) \quad (20)$$

where, m_g is mass of flue gas (kg/s), C_{pg} is the specific heat capacity of the flue gas, T_3 is maximum cycle temperature and T_4 is the exhaust gas temperature.

Expressing the turbine work as a function of the isentropic efficiency of the turbine (η_t) and polytropic index for flue gas (γ_g) we have:

$$W_{gt} = m_g C_{pg} T_3 \eta_t \left[1 - (r_p)^{\left(\frac{1-\gamma_g}{\gamma_g}\right)}\right] \quad (21)$$

Where

$$\eta_t = \frac{1 - \left(\frac{P_4}{P_3}\right)^{\frac{\eta_p(\gamma_g - 1)}{\gamma_g}}}{1 - \left(\frac{P_4}{P_3}\right)^{\frac{(\gamma_g - 1)}{\gamma_g}}} \quad (22)$$

Net work done: The net work done may be expressed as:

$$W_{net} = W_{gt} - W_c \quad (23)$$

Work ratio: The work ratio may be expressed as:

$$W_r = \frac{networkoutput}{grossworkoutput} = \frac{W_{net}}{W_{gt}} \quad (24)$$

Specific Fuel Consumption (SFC): This is the ratio of fuel used by a machine to the amount of power the machine produces at a given instant of time (hour). And it can be determined by the equation:

$$SFC = \frac{3600 \times m_f}{W_{net}} \quad (25)$$

where, m_f is the fuelmass flow rate (kg/s)

The design analysis was performed by simulating the behaviour and performance of GT unit 2 of NDPHC gas turbine power plant in Sapele, Nigeria. The following design/operational parameters in Table 2 were used.

Assumptions: The following assumptions were made in carrying out thermodynamic modeling and simulation of the power plant.

1. All the thermodynamic processes in the gas turbine (GT) cycle are considered base on the steady state model.
2. The principle of ideal gas mixture was applied for the air and combustion products with variable specific heat.
3. The fuel injected to the Combustion Chamber is assumed to be natural gas.
4. Complete combustion of fuel occurs in Combustion Chamber.

Table 1. Exergy destruction rate and efficiency equations of each component

Component	$\dot{E}x_{in}$ (MW)	$\dot{E}x_{out}$ (MW)	$\dot{E}_D$ (MW)	η (MW)
Compressor	$\dot{W}_c + \dot{E}x_5$	$\dot{E}x_2$	$\dot{W}_c + \dot{E}x_1 - \dot{E}x_2$	$\frac{\dot{E}x_2}{\dot{W}c + \dot{E}x_1}$
Combustion Chamber	$\dot{E}x_2 + \dot{E}x_5$	$\dot{E}x_3$	$\dot{E}x_2 + \dot{E}x_5 - \dot{E}x_3$	$\frac{\dot{E}x_3}{\dot{E}x_2 + \dot{E}x_5}$
Gas Turbine	$\dot{E}x_3$	$\dot{W}_{gt} + \dot{E}x_4$	$\dot{E}x_3 - (\dot{W}_{gt} + \dot{E}x_4)$	$\frac{\dot{W}_{gt} + \dot{E}x_4}{\dot{E}x_3}$

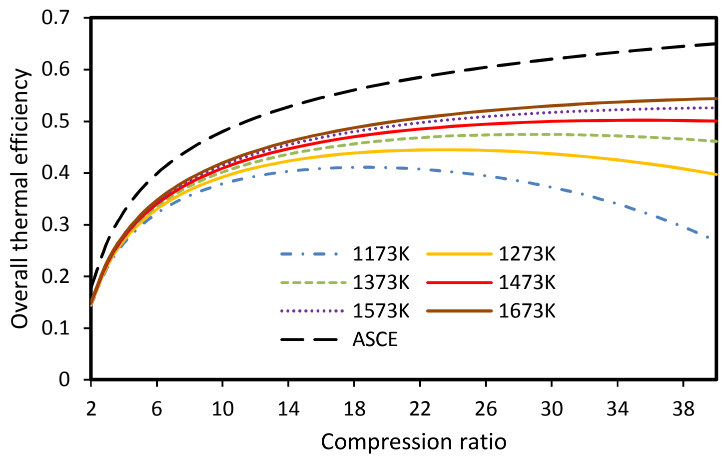

Fig. 4. Variation of compression ratio and MCT on overall thermal efficiency of the plant

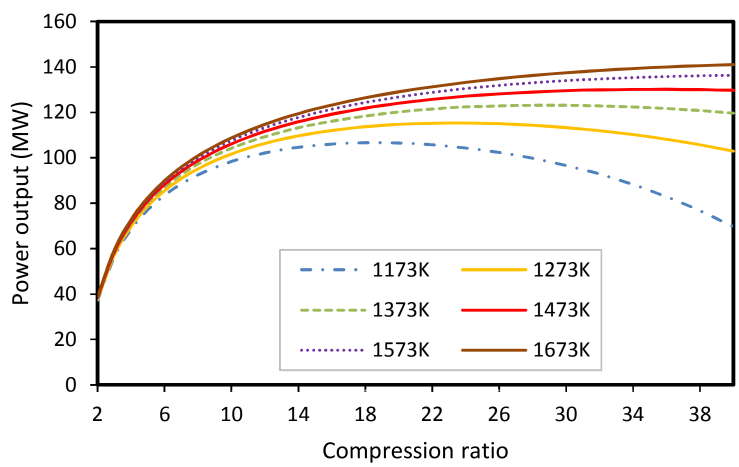

Fig. 5. Variation of compression ratio and MCT on power output

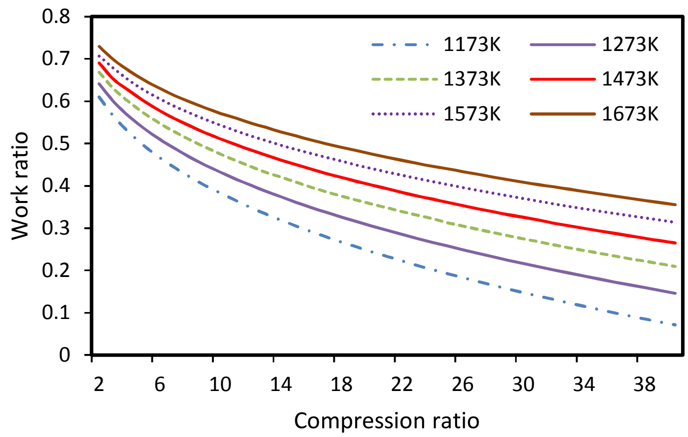

Fig. 6. Variation of compression ratio and MCT on Work ratio

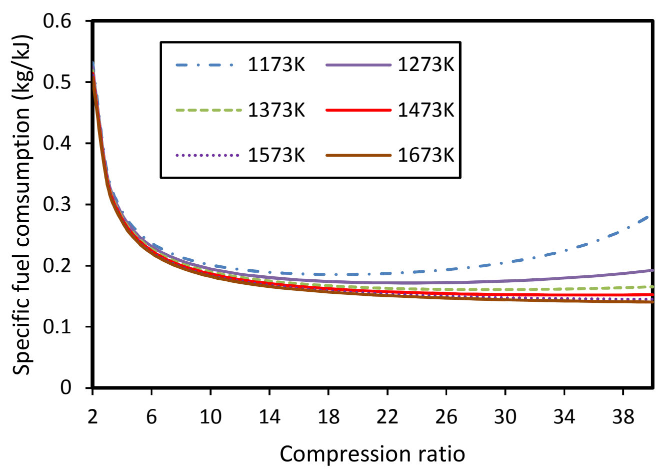

Fig. 7. Variation of compression ratio and MCT on Specific Fuel Consumption (SFC)

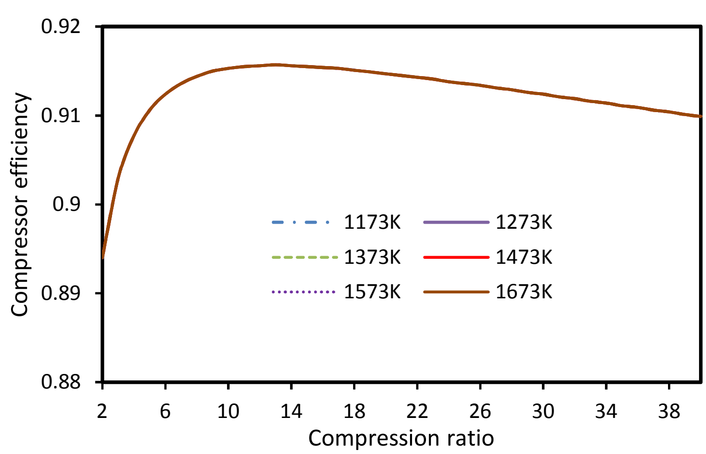

Fig. 8. Variation of compression ratio and MCT on compressor efficiency

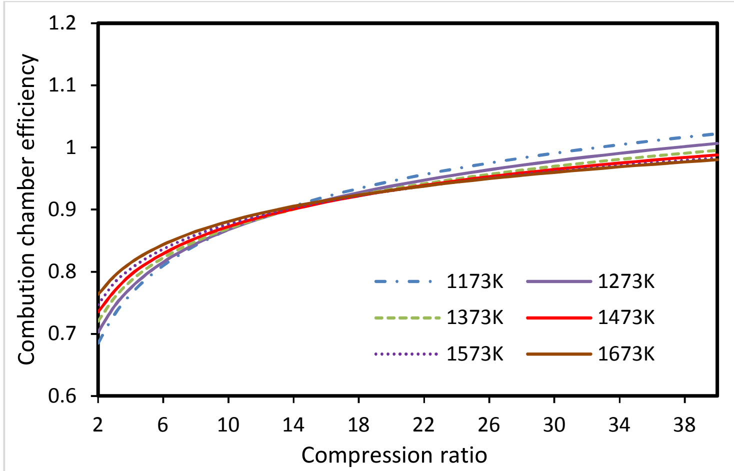

Fig. 9. Variation of compression ratio and MCT on combustor efficiency

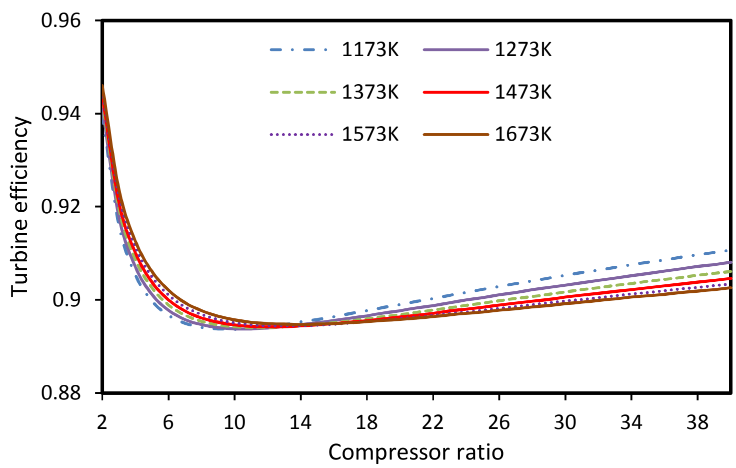

Fig. 10. Variation of compression ratio and MCT on turbine efficiency

Table 2. Design parameters

S/N	Parameters	Units
1	Ambient air Temperature T_o (K)	298.15
2	Ambient Pressure P_o (bar)	1.013
3	Mass flow rate of fuel (kg/s)	5.5
4	Fuel type	Natural gas
5	Calorific value of Natural gas (kJ/kg)	47141
6	Polytropic efficiency of turbine and compressor	0.87

3. RESULTS AND DISCUSSION

The variables in this analysis are: pressure ratio, mass flow rate of air, maximum cycle temperature, compressor isentropic and turbine efficiencies. The effect of these variables on the work ratio, cycle efficiency, specific fuel consumption of the gas turbine and components exergy destruction rate was evaluated.

MATLAB R2014b was used in simulating the behavior of the GT plant. From the result obtained, Fig. 4 shows the variation of compression ratio and MCT on the thermal efficiency of the gas turbine plant. The overall thermal efficiency increased as the compression ratio and MCT increased. At compression ratio of 40, the overall efficiency increased from 0.2686 to 0.6501 as the MCT increased from 1173K to 1673K. Though, the variation of overall thermal efficiency is minor at lower compression ratio, it is very significant at higher compression ratio. However, at MCT of 1173K, 1273K, 1373K, 1473K, 1573K, and 1673K, highest thermal efficiencies were recorded at compression ratio 19, 24, 29, 36, 44 and 51 respectively, after which the thermal efficiency begins to deteriorate with further increase in compression ratio. Thus, the compression ratios of 19, 24, 29, 36, 44 and 51 represent the maximum compression ratios that yield the highest thermal efficiencies for the gas turbine at specific MCT of 1173K, 1273K, 1373K, 1473K, 1573K, and 1673K respectively.These results indicatethatthe efficiency of gas turbine operating at specified MCT cannot be increased further by increasing the compression ratio beyond it optimal range. Overall efficiency of the turbine was also presented along with the Air standard cycle efficiency (ASCE) to assess their variability. The ASCE shows a steady and continuous increase with compression ratio while the overall efficiency of the gas turbine at various maximum cycle temperature increased but began to decreaseafter reaching certain compression ratio. The power output of the turbine in Fig. 5 shows similar result as in the case of the overall efficiency of the plant. Fig. 6 presents the variation of the compression ratio and MCT on the work ratio of the plant. The result shows that the work ratio increasedwith MCT and decreases as the compression ratio Increased. At compression ratio of 26, the work ratio changed from 0.1829 to 0.4335 as MCT changed from 1173K to 1673K.

In Fig. 7, SFC decreased as the compression ratio Increased. However, as the compression ratio reached its maximum point for each MCT, the SCF began to decrease. At lower compression ratio, there was no significant variation in the SFC as the MFC Increased, but at higher compression ratio the variation become significant. At compression ratio of 30:1, the SFC changed from 0.2051 to 0.1441 as MCT changed from 1173K to 1673K.

The exergetic efficiency of compressor was presented in Fig. 8, the result shows that the MCT has no effect on its efficiency. However, compressor exergetic efficiency increased with the compressor ratio. At compressor ratio of 13:1 the maximum exergetic efficiency was attained and as the compressor ratio increased further, the exergetic efficiency began to deteriorate.

Fig. 9 shows the exergetic efficiency of the combustor. The result shows that combustor exergetic efficiency increased with compression ratio. The MCT variation show interesting behaviour. At lower compression ratio, combustor exergetic efficiency variation with MCT varies significantly at increasing order. At compression ratio of 14:1 the variation became insignificant and at higher compression ratio away from 14:1 the combustor exergetic efficiency decreases (decreased) as MCT increased with significant variation.

The turbine exergetic efficiency is presented in Fig. 10. The result shows that this efficiency decreased with increasing compression ratio until at compression ratio of 13. Further increase in compression ratio led to increase in turbine efficiency. At lower compression ratio, increase in MCT increased the turbine efficiency but at compression ratio of 13:1 the variation between the efficiency and the MCT became insignificant and at higher compression ratio away from 13:1, the efficiency decreased with increase in MCT.

4. CONCLUSION

This study evaluated the effect of two design limiting conditions (MCT and compression ratio) on the performanceof simple cycle gas turbine power plant. Sapele gas turbine in Nigeria was modelled and simulated using the steady state energy and exergy equations. The behavior of the turbine was used to predict optimum design and operations parameters for optimum performance. The result showed that the performance (overall thermal efficiency, work ratio and the specific fuel consumption) of the gas turbine power plant depend strongly on MCT and compression ratio. As such, optimum design and performance requirehigher values of MCT and compression ratio. Thus, an estimate shows that maximum compression ratio that can be used for optimum design and operations of gas turbine at MCT of 1173K, 1273K, 1373K, 1473K, 1573K, and 1673K are 19, 24, 29, 36, 44 and 51 respectively.

COMPETING INTEREST

Authors have declared that no competing interest exist.

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