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Thermo-economic analysis of a simple cycle gas turbine with intercooling based on Ogorode, sapele environment, Nigeria

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ABSTRACT

Intercooled gas turbines are necessary inclusion in the quest for improvement in power plant performance. Objectives in achieving the aim of this research were to determine thermodynamic condition by evaluating the performance measures as well as the energy generation cost of a simple cycle gas turbine with inter-cooling in comparison with the traditional simple cycle gas turbine. The methodology was to apply relevant data including available plant operational data in Ogorode to model a traditional simple cycle gas turbine and the intercooled gas turbine using the governing principles. Thereafter carry out simulation of the models to obtain values of their various performance measures and cost. This research will help gain knowledge on performance from operating simple cycle gas turbines with inter-cooling in Nigeria, especially the Sapele environment, the promotion of choice and better plant utilisation. The pressure ratio, thermal efficiency and specific fuel consumption for GT and GTIC were found to be 10.87, 0.3457, 0.2797 kg/kWh and 19.76, 0.3831, 0.2525 kg/kWh at maximum power output of 113 MW and 140.5 MW respectively. The improvement in power output was an increase of 24.34 % for the GTIC which also showed 9.72 % reduction in specific fuel consumption, 10.82 % increment in thermal efficiency and a 9.73 % lower energy cost. The enhancements will promote more energy production, lower emissions and reduced power generation cost.

KEYWORDS

Thermo-Economic Analysis, Gas Turbine, Inter-cooling, Sapele

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INTRODUCTION

Ogorode is located in Sapele, Delta State in the Niger delta region of Nigeria. At least two power plants are located in Ogorode including the National Integrated Power Project (NIPP) which is a gas turbine power plant. The Gas turbine (GT) has been playing a significant role in the power generation business for decades with improvement being achieved since its emergence, especially in the area of performance. Gas turbines are ideal for peak electricity demand as they can be started and stopped quickly, but are presently being used for base loads. Gas turbines are now also being used as baseload plants. The natural gas fuelled gas turbine is currently operated as baseload [5]. One of the gas turbine configurations is a gas turbine with inter-cooling (GTIC). An intercooled gas turbine is a technology that has been used to improve on the performance of a typical simple cycle gas turbine with the commercial success recorded. It is different from the typical simple cycle gas turbine because the technology incorporates at least two compressors which are a low-pressure compressor and a high-pressure compressor, but an intercooler is usually installed between these compressors. Intercooled gas turbines being manufactured include the General Electric's (GE's) LMS 100 and the Siemens' Industrial Trent 60, both of which are simple cycle systems.

The objectives of this work were to evaluate the thermodynamic performance measures of the GTIC and its energy generation cost in comparison to a GT configuration and the significance is in facilitating the gaining of knowledge on the performance benefits of operating a GTIC to promote optimum utilisation. Gas turbines with inter-cooling have the capability of increasing power output, raising the pressure ratio of maximum power output and creating the potential for increasing thermal efficiency above what a typical simple cycle gas turbine can offer. Ambient air is taken in by the low-pressure compressor and compressed to an intermediate pressure. The compressed air is then cooled at this pressure through the intercooler back to the ambient temperature at the high-pressure compressor inlet from where it is compressed to combustion chamber pressure. Inter-cooling can be achieved by a conventional heat exchanger or by means of a water spray inside the gas stream, such as in GE's LM 6000 [1]. Inter-cooling (IC) of the ambient air helps to raise its density and thereby increasing the mass flow of air through the machine. Since the mass flow is a factor in the power relation with direct proportionality, therefore, the increased mass flow results to higher power output.

Aside from the thermodynamic performance, it is desirable to operate gas turbines in a cost effective and environmentally sustained manner. Ensuring higher thermal efficiency of the plant has the effect of cutting fuel cost per kilowatt and reducing the amounts of emissions to the environment. Cost and environmental impact are among the factors to consider when setting up any power plant [6]. Important to achieving this is the encouragement of a highly efficient combustion process and the use of emission control technology like dry low nitrogen oxides (DLN) also known as dry low emission (DLE). These are crucial to the elimination or reduction of carbon (II) oxide (CO) and oxides of nitrogen (NOx) which formation increases with high firing temperatures. Apart from the lower energy cost that may result, more power output promotes more energy to the grid for transmission and distribution.

MATERIALS AND METHODS

Among the parameters considered were temperature, pressure, mass flow, specific fuel consumption, thermal efficiency and power output. The governing principles (equations 1 to 15) of gas turbines, as well as plant data were applied to model the typical simple cycle gas turbine and gas turbine with inter-cooling. Simulations of the models were carried out to determine values of various thermodynamic performance measures, and energy cost was found using equation (16). Tables were employed to organise the outputs results obtained for easy analysis and evaluation and comparison of the typical gas turbine model with that using IC technology.

THEORY/CALCULATION

In Figure 1 is the temperature-isentropic (T-S) diagram of the ideal and actual processes that takes place in the simple gas turbine and Figure 2 is the T-S diagram of the intercooled gas turbine.

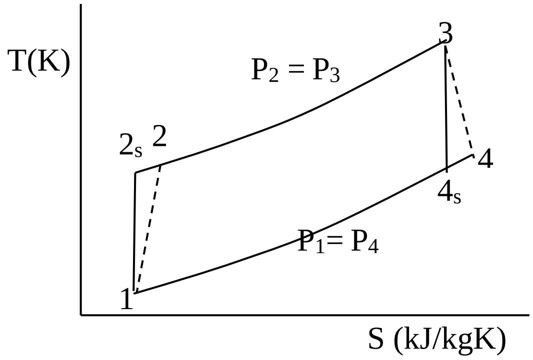

FIGURE 1: T-S Diagram of Ideal and Actual Simple Cycle Gas Turbine

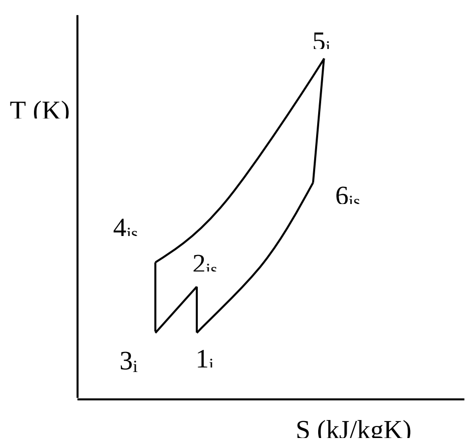

FIGURE 2: T-S Diagram of Simple Cycle Gas Turbine with Inter-Cooling

- **Compression Process**

The output pressure of compressor is

$P_2 = P_1(T_{2s}/T_1)^{(\gamma/\gamma-1)}$ (1)

where, T_1 = ambient temperature of air at compressor inlet; T_{2s} = isentropic temperature of air at compressor outlet; P_1 = ambient pressure; P_2 = outlet pressure of compressed air, and γ = ratio of specific heats.

The ratio of the compressor outlet pressure to the inlet pressure is the pressure ratio, r, mathematically related as

$r = P_2/P_1$

The value for ratio of specific heats for air is given as

$\gamma = \gamma_a = 1.4$ [3].

The compressor isentropic work is

$W_{cisen} = c_{pa}(T_{2s} - T_1)$ (2)

and the actual work is

$W_c = W_{cisen}/\eta_c$ (3)

where, T_{2s} = compressor isentropic outlet temperature, W_c = actual compressor work; c_{pa} = specific heat at constant pressure for air, W_{cisen} = isentropic compressor work; η_c = isentropic efficiency of compressor.

Actual compressor power is

$P_c = \dot{m}_a c_{pa}(T_2 - T_1)$ (4)

where, $\dot{m}_a$ = mass flow of air, and T_2 = compressor actual outlet temperature

- **Combustion Process**

Actual combustion process may be modelled by applying the relation given as

$Q_{in} = \dot{m}c_{pg}(T_3 - T_2)$ (5)

where, c_{pg} =specific heat at constant pressure for flue gas

where, Q_{in} = energy in combustion chamber; T_3 = turbine inlet temperature; $\dot{m}$ = mass flow.

- **Expansion Process**

Expansion takes place through the turbine to extract mechanical power, and from the isentropic relation we have,

$T_3/T_{4s} = (P_2/P_1)^{(\gamma-1)/\gamma}$ (6)

The value for ratio of specific heats for flue gas is given as $\gamma = \gamma_g = 1.33$ [3]

where, T_{4s} = isentropic temperature at turbine outlet, and T_3 = turbine outlet temperature; and the isentropic turbine work and actual turbine work are given in equations (7) and (8),

$W_{gtisen} = c_{pg}(T_3 - T_{4s})$ (7)

$W_{gt} = \eta_t W_{gtisen}$ (8)

where, W_{gt} = actual gas turbine work output; W_{gtisen} = isentropic gas turbine work output; η_t = isentropic efficiency of turbine.

Turbine actual power output is

$P_t = \dot{m}c_{pg}(T_3 - T_4)$ (9)

where, P_t = actual turbine power output.

• Performance Measures

The net power output is

$P_{net} = \dot{m}c_{pg}(T_3 - T_4) - \dot{m}c_p(T_2 - T_1)$ (10)

where, P_{net} = net power output in kJ/s or kW.

Relating the rate of energy supplied, Q_{in} (kJ/s) and the thermal efficiency and converting to mega Watt, we may obtain

$P_o = (\eta_o * Q_{in}/1000) * PF$ (11)

where, P_o = power output (MW); PF = power factor.

The thermal efficiency for the typical simple cycle and an intercooled simple cycle gas turbines are given in equations (12) and (13) respectively.

$\eta_{th} = ((\dot{m}c_{pg}(T_3 - T_4) - \dot{m}c_p(T_2 - T_1))/(\dot{m}c_{pg}((T_3 - T_2))$ (12)

$\eta_{th} = (c_{pg}(T_{5i} - T_{6is}) - c_{pa}[(T_{2is} - T_{1i}) + ((T_{4is} - T_{3i})])/c_{pg}(T_{5i} - T_{4is})$ (13)

where, η_{th} = thermal efficiency; T_{1i} = inlet temperature of low pressure compressor; T_{2is} = isentropic outlet temperature of low pressure compressor; T_{3i} = inlet temperature of high pressure compressor; T_{4is} = isentropic outlet temperature of high pressure compressor; T_{5i} = turbine inlet temperature for GTIC; T_{6is} = isentropic turbine outlet temperature for GTIC and $T_{5i} = T_3$; T_3 = turbine inlet temperature traditional simple cycle.

How specific fuel consumption is related to thermal efficiency and mass flow rate of fuel is given in equations (15) and (16) respectively.

$\eta_{th} = 3600/SFC.FHV$ (14)

$SFC = 3600\, \dot{m}_f/\dot{W}_{net}$ (15)

[2],

where, SFC = specific fuel consumption; FHV = fuel heating value; $\dot{m}_f$ = mass flow of fuel; $\dot{W}_{net}$ = net work rate.

• Energy Cost

The fuel or energy costs, E_c (mills/kWh) for natural gas fired power station is given by [8]:

$$E_c = \frac{F_c\left(\frac{dollar}{kWh}\right) \text{x } 100\left(\frac{mills}{dollars}\right)}{\eta_{oc}Q_{net}\left(\frac{kJ}{kWh}\right) \text{x } kWh/3600kJ} = \frac{3.6*10^6 F_c}{\eta_{oc}Q_{net}} \quad (16)$$

as cited by [4]

where, F_c= fuel cost (dollar/kWh); Q_{net} = net energy released per unit fuel mass, (kJ/kWh); $\eta_{oc} = \eta_o$ = overall thermal efficiency; mills/kWh = standard expression for cost of electricity (1 mill/kWh = \$1/MWh). That is 1mill = 1/1000 US dollar or 1/10 cent [7].

(a) Unit Cost of fuel: $1\ kWh = \$0.0025/0.3013 = \$0.00830$.

Using exchange rate of $\$1 = N389.01$ as at May 3, 2020, then

$\$0.00830 * 389.01 = N3.23$,

The unit cost of fuel, $F_c = N3.23/kWh$.

(b) Energy Cost: Choosing 1 cubic feet (CF) = 1,034 British thermal unit (BTU)

$1034\ BTU/CF = (1.055 * 1034 * BTU/0.29)/BTU = 3{,}761.62$

$Q_{net} = 3{,}761.62 kJ/kWh$.

Evaluating the cost of energy for the various gas turbine configurations by using equation (16),
the typical simple gas turbine energy cost considering overall thermal efficiency of 0.3457 is

$E_c = 3.6 * 10^6 * 3.23/0.3457 * 3761.62 = N8{,}941.92/MWh\ (i.e. N8.94/kWh)$,

the intercooled gas turbine energy cost considering overall thermal efficiency of 0.3831 is
$E_c = 3.6 * 10^6 * 3.23/0.3831 * 3761.62 = N8{,}068.97/MWh\ (i.e.\ N8.07/kWh)$

TABLE 1: Assumptions

Gas turbine compressor inlet temperature, T_1 (K)	300.74
Gas turbine inlet temperature, T_{5i} (K)	1395.33
Gas turbine compressor inlet pressure, P_1 (bar)	1.012
Intermediate (outlet pressure of low-pressure compressor), P_{2is} (bar)	3
Mass flow of fuel, $\dot{m}_f$ (kg/s)	7.4
Mass flow of air, $\dot{m}_a$ (kg/s)	414.4
Net calorific value for natural gas, C_{net} (kJ/kg)	37,889
Combustion efficiency, η_{comb}	1
Generator power factor, PF	0.9
Negligible combustion fractional pressure loss, f_{pl}	-
Negligible loss of heat of vaporization of water vapour, $q_{loss,vap}$	-
Ratio of specific heats for air, γ_a	1.4
Ratio of specific heats for flue gas, γ_g	1.33
Compressor isentropic efficiency, η_c	0.85
Turbine isentropic efficiency, η_t	0.9
Specific heat at constant pressure for air gas, c_{pa} (kJ/kgK)	1.005
Specific heat at constant pressure for flue gas, c_{pg} (kJ/kgK)	1.15

RESULTS AND DISCUSSION

TABLE 2: Simple Cycle Gas Turbine Simulation

r	SFC (kg/kWh)	η	P_o (MW)
4.94	0.3708	0.2608	101.30
5.93	0.3418	0.2830	106.20
6.92	0.3218	0.3005	109.40
7.91	0.3071	0.3149	111.30
8.89	0.2959	0.3269	112.40
9.88	0.2870	0.3370	112.90
10.87	0.2797	0.3457	113.00
11.86	0.2737	0.3533	112.80
12.85	0.2687	0.3599	112.20
13.83	0.2644	0.3658	111.60
14.82	0.2607	0.3709	110.80

TABLE 3: Intercooled Simple Cycle Gas Turbine Simulation

r	SFC (kg/kWh)	η	P_o (MW)
4.94	0.4023	0.2404	109.10
5.93	0.3670	0.2635	117.00
6.92	0.3429	0.2820	122.80
7.91	0.3253	0.2973	127.20
8.89	0.3118	0.3101	130.50
9.88	0.3011	0.3212	133.10
10.87	0.2923	0.3308	135.10
11.86	0.2850	0.3393	136.70
12.85	0.2788	0.3469	137.90
13.83	0.2735	0.3536	138.80
14.82	0.2689	0.3597	139.50
15.81	0.2648	0.3652	139.90
16.80	0.2612	0.3703	140.30
17.79	0.2580	0.3749	140.40
18.77	0.2551	0.3792	140.50
19.76	0.2525	0.3831	140.50
20.75	0.2501	0.3867	140.40
21.74	0.2474	0.3901	140.20

TABLE 4: Performance Improvement

	η	Im (%)	P_o (MW)	Im (%)	SFC (kg/kWh)	Im (%)	E_c (N/kWh)	Im (%)
GT	0.3457		113.00		0.2797		8.94	
GTIC	0.3831	10.82	140.50	24.34	0.2525	- 9.72	8.07	- 9.73

The pressure ratio of maximum power output was found to be higher for the GTIC compared to the GT and with greater power output at the same pressure ratio of 10.87 (Tables 2 and 3). This raised pressure ratio of the GTIC means the potential for higher thermal efficiency and reduced emissions since thermal efficiency increases with increasing pressure ratio. GT and GTIC had pressure ratios of 10.87 and 19.76 at maximum power outputs of 113 MW and 140.5 MW respectively (Tables 2 and 3). The results indicated that at maximum power output, GT and GTIC thermal efficiencies and specific fuel consumption were 0.3457 and 0.3831 and 0.2797 kg/kWh and 0.2525 kg/kWh respectively. The figures presented in this analysis showed increment of 10.82 % in thermal efficiency of the GTIC over the GT configuration (Table 4). The improvement in power output was an increase of 24.34 % for the GTIC which also showed 9.72 % reduction in specific fuel consumption. This increased power output from the GTIC means more energy to the grid. Moreover, there was 9.73 % reduction in energy cost per kWh (Table 4) from operating the GTIC which can promote a sustainable plant operation in the face of depleting natural gas reserves.

CONCLUSION

It was observed that the IC influenced the pressure ratio of maximum power output, with greater power output at the same pressure ratio of 10 87 compared to the GT. The pressure ratio, thermal efficiency and specific fuel consumption for GT and GTIC were found to be 10.87, 0.3457, 0.2797 kg/kWh and 19.76, 0.3831, 0.2525 kg/kWh at maximum power output of 113 MW and 140.5 MW respectively. The improvement in power output was an increase of 24.34 % for the GTIC which also showed 9.72 % reduction in specific fuel consumption, 10.82 % increment in thermal efficiency and a 9.73 % lower energy cost. The GTIC should be operated at maximum power output to raise pressure ratio to encourage higher thermal efficiency and lower specific fuel consumption. The enhancements mean more energy production, lower emissions and reduced power generation cost.

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