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Thermodynamic Analysis of Gas Turbine Power Plant

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ABSTRACT
Gas Turbine power plants are being used for producing electricity, operating airplanes and for various industrial applications such as refineries and petrochemical plants. In this paper, thermodynamic performance analysis of an open cycle gas turbine power plant has been performed. The mathematical formulation for the specific work and efficiency were derived and analyzed. The effect of operating parameters like the ambient temperature, relative humidity, compressor pressure ratio, turbine inlet temperature (TIT), isentropic efficiencies of compressor and turbine on the thermal efficiency, power output and heat rate of a gas turbine plant are studied. An in-house code in Matlab has been developed. The data generated using the code have been utilized to draw different relevant graphs. The results show that the compression ratio, ambient temperature, air to fuel ratio as well as the isentropic efficiencies can strongly influence the thermal efficiency. In addition, the thermal efficiency and power output decreases linearly with increase of the ambient temperature and air to fuel ratio. Also, the specific fuel consumption and heat rate increases linearly with increase of both ambient temperature and air to fuel ratio. Various techniques to improve the performance of gas turbine are also discussed.

Keywords
Gas Turbine, Efficiency, Heat rate, TIT, Relative Humidity

1. INTRODUCTION
Gas turbine could play a key role in future power generation addressing issues of producing clean, efficient and fuel flexible electric power. Gas turbines are the parts of the internal combustion engine in which burning of the air-fuel mixture produces hot gases to run the turbine which produces power. The gas turbines are widely used for producing electricity, operating airplanes and for various industrial applications such as refineries and petrochemical plants. In aircraft propulsion, they are advantageous as they have large power-to-weight ratio, compactness and ease of installation.

Ghaderi and Damircheli [1] investigated the effect of various factors on the efficiency of a gas turbine plant. They reported that effect of increased gas inlet temperature has limited effect on the turbine efficiency. On the other hand, the injection of water or steam into the gas turbines increases the efficiency as well as the flexibility of the turbine. Bade and Bandyopadhyay [2] observed that cogeneration or combined heat and power (CHP) improves the efficiency of the overall plant by minimizing fuel consumption thereby reducing the environmental pollution as well. Shukla and Singh [3] studied the effect of relative humidity on the performance of a gas turbine plant. They found 0.77% increase in the specific power output and 0.65% increase in thermal efficiency for every 15% increase of relative humidity of ambient air. They also reported that on reducing the compressor inlet temperature from 318K to 282K, 3.43% more fuel can be saved with the increase in specific work output by 10.12%. Basha et al. [4] simulated gas turbine plant using GT PRO software to study the effect of different parameters on the performance. They observed that the relative humidity of the inlet air did not have much impact on the gas turbine performance but reducing the ambient air temperature by 10°F increased the plant output by 5% and thermal efficiency by 2%. Furthermore, using natural gas instead of diesel or heavy bunker oil increased the output as well as the efficiency of the plant.

Oyedepo and Kilanko [5] carried out a study on the thermodynamic analysis of a gas turbine power plant fitted with an evaporative cooler based in Nigeria. They concluded that power output increased by 5-10% and thermal efficiency by around 2-5% on decreasing the inlet air temperature by 5°C. Barakat et al. [6] investigated the application of earth to air heat exchanger (EAHE) cooling system for the enhancement of a gas turbine power output. They noted that EAHE inlet air cooling system helped to increase the power output by 9% and thermal efficiency by 4.8%. Noroozian and Bidi [7] used a turbo expander connected to a mechanical chiller for decreasing the compressor inlet air temperature. They reported that the cooling system helped to reduce the temperature by 3.2% in the warmest month which led to an increase in both thermal efficiency and power output by

1.14%. A detailed analysis of inlet air cooling using indirect evaporative cooling system (IECS) was presented by Najjar and Abubaker [8]. They concluded that using IECS in combination with absorption chillers increased the power output and efficiency by 15% and 9% respectively while using IECS with mechanical chillers increased the same by 7.81% and 2.24% respectively. Sanaye and Tahani [9] studied the effects of inlet fogging and wet compression processes on the performance of a gas turbine. They observed that on increasing the amount of water injection, the net power output had an increasing trend while the heat rate had a decreasing trend for both the fogging and the wet compression processes

2. GAS TURBINE CYCLES

The Brayton cycle is a thermodynamic cycle named after George Bailey Brayton that describes the workings of a constant pressure heat engine, although it was originally proposed and patented by Englishman John Barber in 1791. It is also sometimes known as the Joule cycle. There are two types of Brayton cycles, one is open to the atmosphere and using internal combustion chamber whose schematic diagram is shown in Fig. 1 along with its T-S representation in Fig. 2. The second type is a closed cycle using a heat exchanger as shown in Fig. 3.

2.1 Open Gas Turbine Cycle

As shown in Fig. 1, fresh air enters the compressor at ambient temperature where its pressure and temperature are increased. The high pressure air enters the combustion chamber where the fuel is burned at constant pressure. The high temperature and pressure gas enters the gas turbine where it expands to ambient pressure and produces work. . Some part of the power developed by the turbine is utilized in driving the compressor and other accessories and remaining is used for power generation. These turbines are used in aircraft propulsion and electric power generation. This type of cycle is mainly used in majority of gas turbine power plants as it has many inherent advantages.

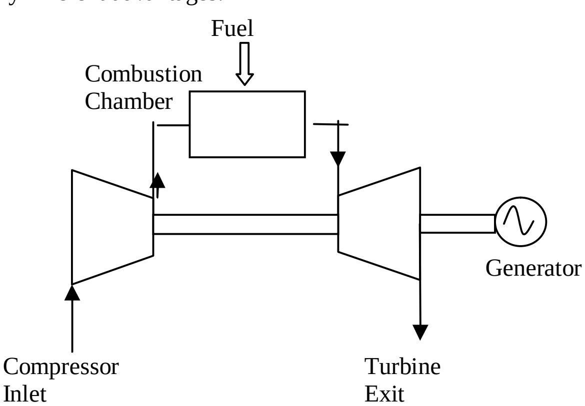

Figure 1. Schematic of open gas turbine cycle

It has high power to weight ratio, high reliability and long life. Almost any hydrocarbon fuel from high-octane gasoline to heavy diesel oils can be used in the combustion chamber of the gas turbine. When the plant is operating at its peak load the stipulation of a quick start and take-up of load frequently are the points in favor of open cycle plant. One of the most important features of open cycle gas turbine power plant, except those having an intercooler, does not need cooling water. Therefore, the plant is independent of cooling medium and becomes self-contained.

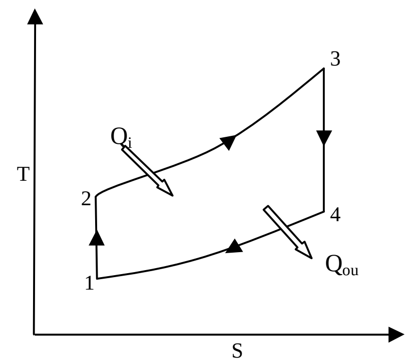

Figure 2. T-S diagram of an open gas turbine cycle

2.2 Closed Gas Turbine Cycle

Gases like nitrogen, helium, argon etc are used as working fluid in this cycle. As shown in Fig. 3, these gases enter the compressor where its pressure and temperature are increased. The high pressure air enters the high temperature heat exchanger where heat is transferred, through a combustion chamber, to the working fluid. Thus the temperature of the working fluid increases. Then this high temperature and pressure working fluid enters the turbine where it is expanded and work is produced, thereby decreasing the temperature and pressure. Then the working fluid is passed through a low temperature heat exchanger. Closed-cycle GT power plant has the potential to complement the conventional coal-fired power plant and open cycle GT power plants.

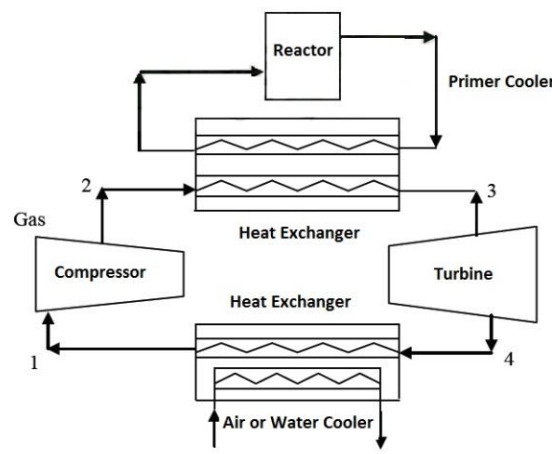

Figure 3. Arrangement for closed gas turbine cycle

In a closed cycle GT or closed Brayton cycle (CBC), the turbine exhausts are not thrown out but recirculated. The layout and temperature-entropy (T–S) diagram of a simple regenerative closed cycle GT is shown in Fig. 4. The working fluid is compressed in the compressor from point 1 to 2. Then it enters the recuperator where some of the heat content of the turbine exhaust is regenerated (process 2–5). After regeneration the fluid passes through the heat source, which could either be a nuclear reactor core, an intermediate heat exchanger (IHX) or a gas heater. In the heat source the fluid achieves the highest temperature within the cycle. This is followed by an expansion in the turbine (process 3-4). The turbine provides the work for the compressor and

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generator. Finally, heat is rejected from the cycle in the cooler, where the fluid is cooled to the initial condition.

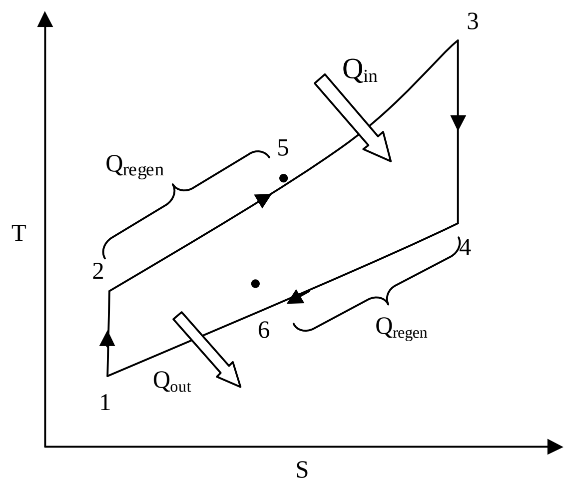

Figure 4. T-S Diagram for a regenerative closed cycle gas turbine

3. THERMODYNAMIC MODELING OF GAS TURBINE

The gas turbine plants consist of four components including the compressor, combustion chamber, turbine and generator.

The compressor pressure ratio can be defined as:

$$r_p = \frac{p_2}{p_1} \tag{1}$$

where, p_1 and p_2 are compressor inlet and outlet air pressure respectively.

In engineering analysis, isentropic efficiency involves a comparison between the actual performance of a device and the performance that would be achieved under idealized circumstances for the same inlet and exit states. The isentropic efficiencies of compressor and turbine can be calculated from figure 5. In this figure, the dotted lines represent the isentropic compression and expansion processes. The actual processes are shown by the hard lines.

The isentropic efficiency for compressor is expressed as

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1} \tag{2}$$

The temperature of the air coming out from the compressor is calculated following Rahman et al. [10] as:

$$T_2 = T_1 \left\{ 1 + \frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c} \right\} \tag{3}$$

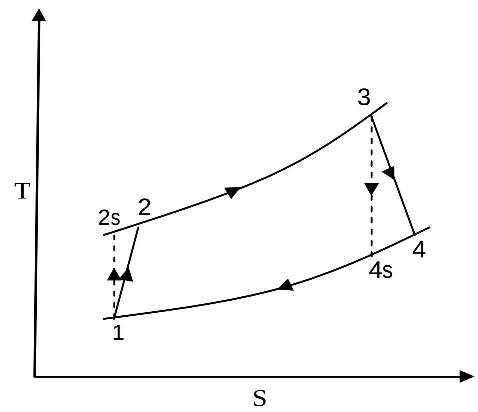

Figure 5. Isentropic efficiency for compressor and turbine

Two new variables and have been defined for simplification as:

$$R_{pa} = \left\{ \frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c} \right\} \tag{4}$$

$$R_{pg} = \left\{ 1 - \frac{1}{r_p^{\frac{\gamma_g - 1}{\gamma_g}}} \right\} \tag{5}$$

Now equation (3) can be written as

$$T_2 = T_1(1 + R_{pa}) \tag{6}$$

Compressor work can be calculated as

$$W_c = \frac{c_{pa} \times T_1 \left(r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1 \right)}{\eta_c \times \eta_m} \tag{7a}$$

$$\text{or, } W_c = \frac{c_{pa} \times T_1 \times R_{pa}}{\eta_m} \tag{7b}$$

where c_{pa} is the specific heat of air and η_m is the mechanical efficiency of the compressor.

The energy balance in the combustion chamber gives:

$$\dot{m}_a c_{pa} T_2 + \dot{m}_f \times LHV = (\dot{m}_a + \dot{m}_f) c_{pg} \times TIT \tag{8}$$

where $\dot{m}_f$ is mass flow rate of fuel, $\dot{m}_a$ is the mass flow rate of air, LHV is low heating value and $TIT(T_3)$ is turbine inlet temperature.

After rearranging Eq. (8), the turbine inlet temperature can be expressed as:

$$T_3 = \frac{\eta_{com} \times LHV + f \times c_{pa} \times T_2}{c_{pg} \times (1+f)} \tag{9}$$

where η_{com} is the combustion efficiency.

The exhaust gases temperature T_4 from gas turbine is given by

$$T_4 = T_3 \times (1 - \eta_t \times R_{pg}) \tag{10}$$

where η_t is the isentropic efficiency of the turbine.

The shaft work (W_t) of the turbine is evaluated as

$$W_t = \frac{\left(c_{pg} \times T_3 \times \eta_t \times R_{pg}\right)}{\eta_m} \tag{11}$$

Then, the net work (W_{net}) produced by gas turbine is

$$W_{net} = W_t - W_c \tag{12}$$

The total output power from the turbine (P) is given by

$$P = \dot{m}_a \times W_{net} \tag{13}$$

Specific fuel consumption (SFC) is determined as

$$SFC = \frac{3600 \times f}{W_{net}} \tag{14}$$

The heat supplied can be calculated as

$$Q_{add} = c_{pg} \times \left(T_3 - T_1\left(1 + R_{pa}\right)\right) \tag{15}$$

The gas turbine efficiency is determined as:

$$\eta_{th} = \frac{W_{net}}{Q_{add}} \tag{16}$$

Heat rate (HR) of the gas turbine cycle is determined as:

$$HR = \frac{3600}{\eta_{th}} \tag{17}$$

The humidity of the ambient air also plays an important role on the performance of a gas turbine plant. Relative humidity is the ratio of actual vapour density to saturated vapour density. Mathematically,

$$\varphi = \frac{\text{Actual vapour density}}{\text{Saturated vapour density}} \tag{18}$$

The above mentioned equations have been used to develop the thermodynamic model and code in MATLAB. Data have been generated for different compressor inlet temperature (CIT), air fuel ratio (AFR) and turbine inlet temperature (TIT).

Some of the parameters are kept constant at a fixed value during the simulation process. The values of all the parameters used in this work have been presented in table 1.

Table 1. Summary of operating parameters

S/N	Operating parameters	Value	Unit
1	Mass flow rate of air through compressor (m_a)	125	kg/s
2	Temperature of inlet air to compressor (T_1)	268-328	K
3	Compression pressure ratio	4-32	-
4	Air- fuel ratio (on mass basis)	40-56	-
5	Inlet temperature to gas turbine (T_3)	1219	K
6	Lower heating value (LHV)	47622	kJ/kg
7	Specific heat of air	1.005	kJ/kg
8	Specific heat of gas	1.8083	kJ/kg
9	Isentropic efficiency of compressor (η_c)	85	%
10	Isentropic efficiency of turbine (η_t)	90	%
11	Efficiency of combustor (η_{com})	95	%
12	Mechanical efficiency (η_m)	90	%

4. RESULTS AND DISCUSSION

In this paper, the effects of ambient temperature, relative humidity, compression ratio, turbine inlet temperature, isentropic efficiency of components and air fuel ratio on the performance of a gas turbine are put forward. The effects of operating conditions on the net work output, specific fuel consumption, heat rate and thermal efficiency are obtained by energy balance equations using the MATLAB software. Numerical data obtained by running the code are plotted to describe the effect of various operating parameters on the performance of gas turbine.

The variations of compressor work with compression ratio for different compressor inlet temperature (CIT) ranging from 268K to 328K have been shown in Fig. 6. It is observed from the figure that compressor power input increases with the increase of compression ratio as well as compressor inlet temperature. This can be easily explained from the fact that compressor work is directly proportional to the intake air temperature and the compression ratio. From the graph, it can be noted that compressor work input increases by about 3.6% for every 10°C rise in the ambient temperature at a compressor pressure ratio of 15. Also, the power input to the compressor increases by approximately 20% on increasing the compression ratio from 7 to 11 at 298 K ambient temperature.

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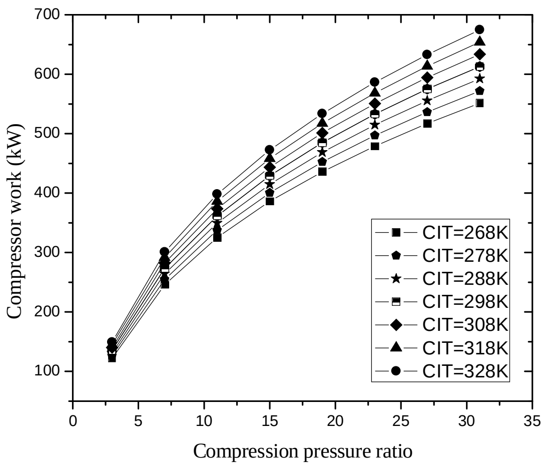

Figure 6. Effect of compression ratio and ambient temperature on compressor work

Figure 7 shows the graph between thermal efficiency and isentropic efficiency for different air fuel ratios. It can be seen that thermal efficiency increases with increasing isentropic efficiency of compressor. It is known that as the isentropic efficiency of compression increases, the effect of friction or the irreversibility in the system decreases and hence there is less loss of exergy in the compressor. On the other hand thermal efficiency decreases with increasing air fuel ratio. It is seen that thermal efficiency increases by approximately 4% on increasing the isentropic compressor efficiency by 5% and decreases by 3.2% on increasing the air fuel ratio from 40 to 44.

The effect of compression pressure ratio at different air fuel ratios on the thermal efficiency of the gas turbine has been shown in Fig. 8. The pressure at the exit of the compressor increases on increasing the compression ratio. Thus, the corresponding temperature at the exit (TIT) is also high, so more heat can be transferred. From the figure it is noted that thermal efficiency increases by 14% on increasing the compression pressure ratio from 7 to 11 at an air fuel ratio of 44. Also, it may be noted that the thermal efficiency decreases with the increase of air fuel ratio.

Figure 9 shows the variations of thermal efficiency with compression ratio at constant TIT. It can be seen that efficiency increases with compression pressure ratio up to certain value and then becomes almost constant for a fixed turbine inlet temperature. In some cases, a decrease in efficiency is also observed. It can also be noted that as the turbine inlet temperature (TIT) is increased, the thermal efficiency increases due to the increase in the net power output of the gas turbine power plant. Yet, it cannot be increased beyond a certain limit because the blades of the turbine will undergo creep failure at high temperatures. Figure 9 shows that there is an increase in the thermal efficiency of 8% on increasing the turbine inlet temperature from 1800K to 2000K for compression ratio of 15. To sustain such high temperatures, the blades of the gas turbines to be made up of single crystal material of nickel based alloys.

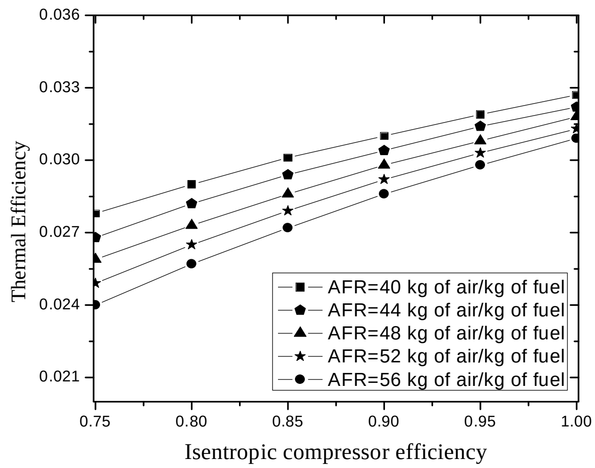

Figure 7. Effect of isentropic efficiency of compressor and air fuel ratio on thermal efficiency

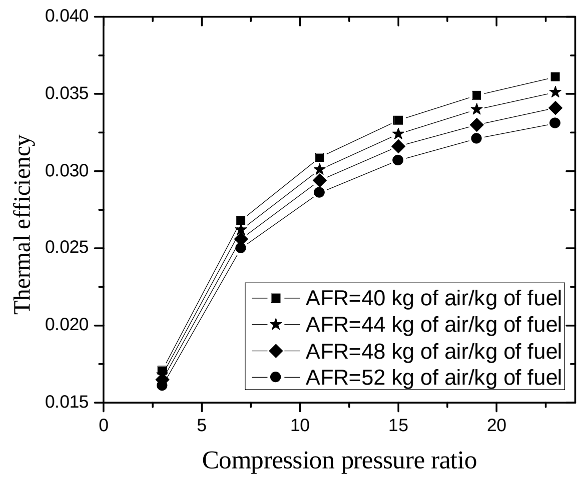

Figure 8. Effect of compression pressure ratio and air fuel ratio on thermal efficiency

The variation of thermal efficiency with the ambient temperature is shown in Fig. 10. The efficiency is linearly decreasing since the power input to the compressor increases as the ambient temperature increase. It is noted that thermal efficiency decreases by about 1.5% with a 10K increase in the ambient temperature for air fuel ratio of 40. Intake air cooling methods are therefore used in most of the plants for getting the maximum out of a GT power plant.

Figure 11 depicts the variation of heat rate with ambient temperature. It is observed that the heat rate increases linearly with the increase in ambient temperature. It can be attributed to the fact that heat rate is inversely proportional to the thermal efficiency. Thermal efficiency decreases with the increase in ambient temperature so the heat rate increases. It is observed that heat rate increases by almost 1% for 10K increase in ambient temperature for air fuel ratio of 44 and increases by 2.5% by increasing the air fuel ratio from 40 to 44 at 298K.

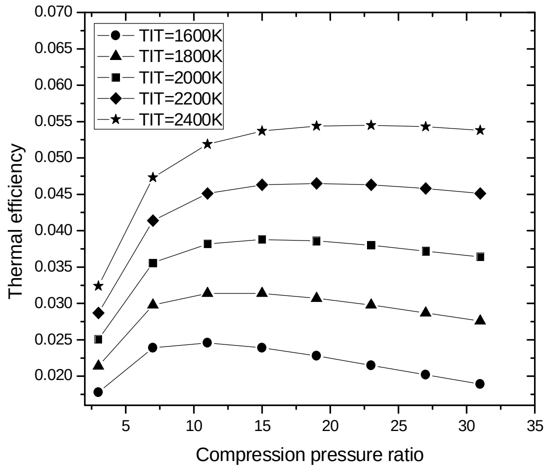

Figure 9. Effect of turbine inlet temperature (TIT) on thermal efficiency

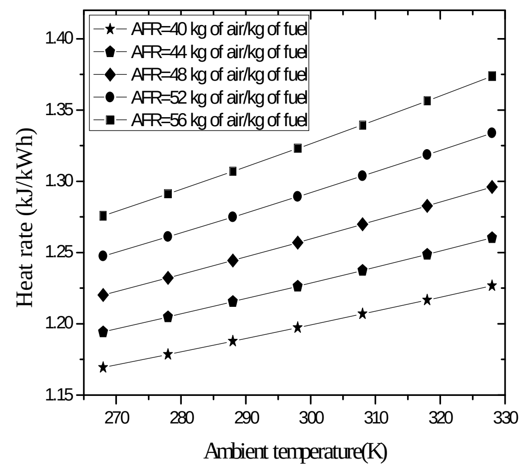

Figure 11. Effect of ambient temperature on heat rate

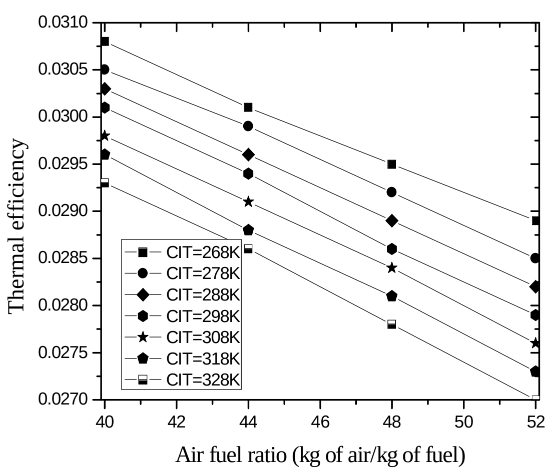

Figure 10. Effect of ambient temperature on GT thermal efficiency

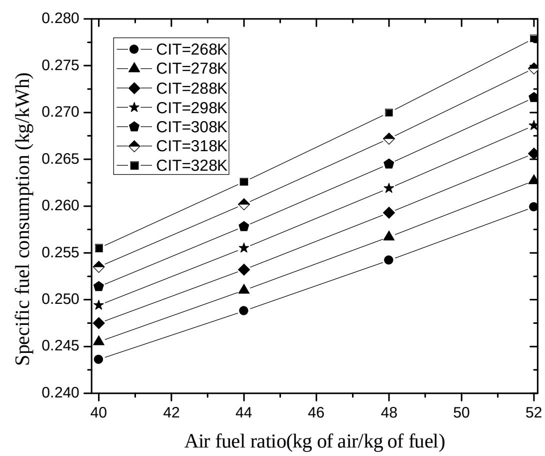

Figure 12: Effect of air fuel ratio on SFC for different ambient temperatures

Specific fuel consumption (SFC) is one of the most important performance parameters of a gas turbine power plant. The variation of specific fuel consumption with air fuel ratio for different compressor inlet temperature is shown in Fig. 12. It is observed that specific fuel consumption increases with increasing ambient temperature. It can be understood that specific fuel consumption is inversely proportional to net work output and on increasing ambient temperature compressor work increases, hence decreasing work output and thereby increasing specific fuel consumption. It is seen from the graph that specific fuel consumption increases by 1% for 10K increase in ambient temperature for air fuel ratio of 44.

In Fig. 13, the variation of specific fuel consumption is shown with respect to relative humidity at various compressor inlet temperature (CIT) of gas turbine power cycle. SFC decreases by a marginal amount with the increase in the relative humidity of the intake air. It is observed that specific fuel consumption decreases by 1.3% by increasing relative humidity by 20% for a compressor inlet temperature of 293K.

5. CONCLUSION

The performance evaluation of a simple gas turbine showed that the ambient temperature and relative humidity, compressor pressure ratio, air fuel ratio, turbine inlet temperature and the isentropic efficiencies of the components influence the thermal efficiency, power output, heat rate and specific fuel consumption of a gas turbine power plant.

The ambient conditions have a significant impact on the gas turbine performance therefore inlet air cooling with the help of evaporative coolers, mechanical chillers and absorption chillers can be very beneficial in augmenting its performance.

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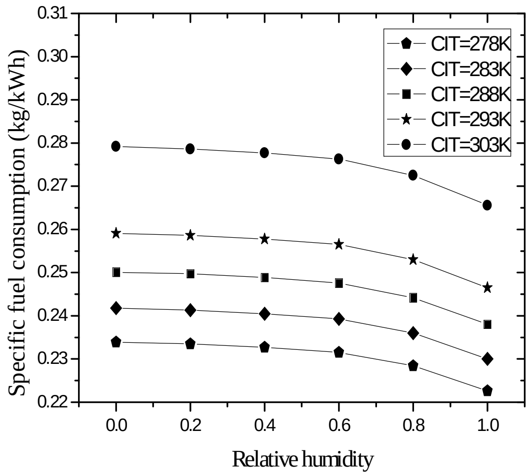

Figure 13. Effect of relative humidity & ambient air temperature on specific fuel consumption

6. NOMENCLATURE

Cp Specific heat at constant pressure [kJ/kg°C]

LHV Lower heating value [kJ/kg]

HR Heat rate [kJ/kWh]

SFC Specific fuel consumption [kg/kWh]

$\dot{m}$ Mass flow rate [kg/s]

p Pressure [Pa]

T Temperature [K]

r_p Compression pressure ratio

W Specific work [kW/kg]

Q Heat transfer rate [kW]

P Power output [kW]

f Fuel to air ratio [ ]

η Efficiency [%]

7. ABBREVIATION

TIT Turbine Inlet Temperature

MWh Mega Watt Hour

GT Gas Turbine

CBC Closed Brayton Cycle

IECS Indirect Evaporative Cooling System

EAHE Earth to Air Heat Exchanger

CHP Combined Heat and Power

HR Heat Rate

IHX Intermediate Heat Exchanger

SFC Specific Fuel Consumption

CIT Compressor Inlet Temperature

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