PERFORMANCE ANALYSIS OF GAS TURBINE AT VARYING AMBIENT TEMPERATURE

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ABSTRACT

In the present work, the performance analysis of a simple open cycle gas turbine power plant which is situated at Uran in district Raigad of Maharashtra, India is carried out at varying ambient temperature and pressure ratio and a comparison is carried out from the stipulated ISO conditions. Generally, there is a difference between the actual power generated by the gas turbine at site ambient conditions and the power tagged on the gas turbine stipulated from the ISO conditions. The analysis of the cycle has been carried out by using MATLAB 13. The result shows that there is a loss of 3.28 % in thermal efficiency at 9 pressure ratio with the increase in ambient temperature from 283 K to 323 K. This loss increases up to 3.97 % at 21 pressure ratio. In addition to this, it was observed that there is a loss of 4.19 MW or 3.87 % in net power at 9 when ambient temperature rises from 283 K to 323 K, which in term increases up to 5.92 MW or 4.46 % at 21 pressure ratio. The analysis reveals that ambient temperature and pressure ratios were strongly affected the performance of gas turbine power plant.

Key words: Ambient Temperature, Compressor ratio, Gas Turbine, Net Power, Thermal Efficiency


1. INTRODUCTION

Uran Gas Turbine Power Plant is the only power station of MAHAGENCO where natural gas is fired as fuel. It has three stage turbines with total size of 912 MW. First stage comprises of four turbines with power generation capacity of 60 MW for each turbine, while second stage turbine comprises of four turbines with power generation of 108 MW for each turbine and at last, the third stage turbine comprises of two turbines with power generation capacity of 120 MW for each turbine. The present study focuses on the performance of first stage turbine only. The ISO conditions for gas turbine are 15°C ambient temperature 60 % relative humidity and 1.013 bar ambient pressure. The site is located at Uran in Raigad district of Maharashtra which is surrounded from three sides by the Arabian Sea. Fig. 1 depicts a simple schematic of the power plant.

The gas turbine power plant is well-known to trait low capital cost, high reliability, high flexibility without complexity [1]. The gas turbines are widely being used for producing electricity, operating
airplanes and for various industrial applications such as refineries and petrochemical plants [2]. Several gas turbines are being widely used for power generation in several countries all over the world. Many of these countries have a wide range of climatic conditions, which impact the performance of gas turbines [3]. The Gas Turbine power plant works on a Joule-Brayton cycle [4]. The gas turbine is a complex machine, and its performance and reliability are governed by many standards. The reliability of the turbines depends on the mechanical codes that govern the design of gas turbines [5]. The mechanical standards and codes have been written by both ASME and the American Petroleum Institute (API) amongst others.

The major variables that affect the gas turbines are:

1. Types of application
2. Plant location and site configuration
3. Plant size and efficiency
4. Types of fuel
5. Enclosures
6. Plant operation mode

The performance of a gas turbine power plant is sensible to the ambient condition. As the ambient air temperature arises, less air can be compressed by the compressor since the withdrawing capacity of compressor is given, and so the gas turbine output is reduced at a given turbine entry temperature. Additionally, the compression work increases because the limited volume of the air increases in proportionality to the intake air temperature [6]. Gas turbines are constant volume machines; at a given shaft speed, they always move the same volume of air. In gas turbines, since the combustion air is taken directly from the environment, their performance is strongly affected by weather conditions [7]. There is a loss of 3.36 % in thermal efficiency at pressure ratio 9 as ambient temperature increases from 283 K to 313 K, this loss increases up to 3.89 % at pressure ratio 21, which reveals that, as pressure ratio increases percentage loss in thermal efficiency increases on increasing the ambient temperature. The thermal performance analysis reveals that, the ambient temperature and compression ratios are strongly influence the performance of gas turbine power plant. [8]. It is well known that the performance can be qualified with respect to its efficiency, power output, specific fuel consumption as well as work ratio. There are several parameters that affect its performance including the compressor compression ratio, combustion inlet temperature and turbine inlet temperature (TIT) [9]. Simple gas turbine cycle calculations with realistic parameters are made and confirm that increasing the turbine inlet temperature no longer means an increase in cycle efficiency, but increases the work done. [10]. Further, the performance improvement of the gas turbine is dependent on the maximum temperature tolerance of the first stage blades and is also reliant on inter stage cooling at the compression stage [11]. Several methods and technologies are available to augment this power loss but this entails additional plant and equipment installation as well as additional operational requirements [12]. Many of these methods such as use of air cooler [13], regenerative steam injection [14], effusive blade cooling techniques [15-17], use of desiccant-based evaporative cooling [18] or absorption chillers [19] are commonplace. The effect of relative humidity on the gas turbine power plant addresses issues of the air cooling and enhances compressor efficiency [20-24].
2. METHODOLOGY OF RESEARCH

2.1. Reversible Simple (Joule-Brayton) Cycle

The Joule-Brayton cycle is the most idealised cycle for the simple gas turbine power plant. Gas turbine power plant consist of four components i.e. compressor, combustion chamber, turbine and generator. T-s diagram for an actual Brayton cycle is demonstrated in fig. 2. The pressure loss in the combustion chamber is represented by $P_2 - P_3$ and pressure loss in exhaust hood is represented by $P_4 - P_1$.

![Figure 2: Actual gas turbine cycle presentation on T-s diagram.](image)

In this cycle

1-2 is isentropic compression. 1-2' is actual compression.

3-4 isentropic expansion 3-4' is actual expansion.

**Compressor**

The compressor efficiency, also known as isentropic compressor efficiency, $\eta_{cis}$ is:

$$\eta_{cis} = \frac{\text{Isentropic compression work}}{\text{Actual compression work}} = \frac{w_c}{w_{ca}}$$
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\[ \eta_c = \frac{T_2 - T_1}{T_2' - T_1} \]  

(1)

For compression process 1-2, we have

\[ T_2 = T_1 \left( \frac{P_2}{P_1} \right)^{\frac{(\gamma - 1)}{\gamma}} \]

Where \( \gamma = 1.4 \) for air

\[ T_2 = T_1 (r_p)^{(\gamma - 1)/\gamma} \]

Where \( r_p \) (pressure ratio) = \( \frac{P_2}{P_1} \)

\[ T_2' = T_1 + \frac{T_2 - T_1}{\eta_c} \]  

(2)

For compression process 1-2, we also have

\[ \frac{T_2'}{T_1} = \left( \frac{P_2'}{P_1} \right)^{\frac{(\gamma - 1)}{\gamma} \eta_p c} \]  

(3)

**Combustion Chamber**

For combustion process 2-3, we have

\[ T_3 = \frac{(C.V.) \times \eta_{comb} + c_{pa} \times A/F \times T_2'}{(c_{pg}) \times (A/F + 1)} \]  

(4)

**Turbine**

The turbine efficiency, \( \eta_t \) = \( \frac{\text{Actual turbine work}}{\text{Isentropic turbine work}} = \frac{W_{ta}}{W_t} \)

\[ \eta_t = \frac{T_3 - T_4'}{T_3 - T_4} \]  

(5)

For expansion process 3-4, we have

\[ \frac{T_3}{T_4} = \left( \frac{P_3}{P_4} \right)^{\frac{(\gamma - 1)}{\gamma}} \]

Where \( \gamma = 1.34 \) for gas

\[ T_4' = T_3 - \eta_t (T_3 - T_4) \]  

(6)

For expansion process 3-4’, we also have

\[ \frac{T_3}{T_4'} = \left( \frac{P_3}{P_4} \right)^{\frac{(\gamma - 1) \eta_{pt}/\gamma}{\gamma}} \]  

(7)

For pressure calculations, we have

\[ p_2 = p_3 = r_p \times p_1 \]  

(8)

\[ p_2' = p_2 \times 0.97 \]  

(9)

\[ p_4 = p_1 \times 1.02 \]  

(10)

The thermal efficiency (\( \eta_{tha} \)), is defined as ratio of net work done to the heat supplied. Thus,

\[ \eta_{tha} = \frac{(c_{pg}(T_3 - T_4') - c_{pa}(T_2' - T_1))}{(c_{pg}T_3 - c_{pa}T_2')} \]  

(11)
Actual Net Work
Actual compressor work = $w_{ca} = h'_2 - h_1$

$$= c_{pa}(T'_2 - T_1) \text{ kJ/kg}$$

Actual turbine work = $w_{ta} = h_3 - h'_4$

$$= c_{pg}(T_3 - T'_4) \text{ kJ/kg}$$

Actual net work, $w_{net} = w_{ta} - w_{ca}$

$$= c_{pg} (T_3 - T'_4) - c_{pa}(T'_2 - T_1)$$

The Net Power
Total net power available = $w_{net} \times m_a \text{ MW}$ (12)

The Specific Fuel Consumption
It is one of the most important parameters expressed in kg/kWh. Thus,

$$sfc = \frac{(3600)m_f}{w_{net}} \frac{\text{kg}}{\text{kWh}}$$ (13)

The Cycle Work Ratio
A good design of gas turbine not only requires a high thermal efficiency and a low air rate, but it must also have a high work ratio. The work ratio (WR) is defined as the ratio of net work to the turbine work. Thus, it is clear that for a good gas turbine the work ratio should be high.

$$WR = \frac{w_{net}}{w_t}$$ (14)

The Cycle Air Rate
The size of plant depends on the rate of flow of air in relationship to the useful power output. Air rate (AR), is defined as the air flow required per kWh output. Thus,

$$AR = \frac{3600}{w_{net}} \frac{\text{kg of air}}{\text{kWh}}$$ (15)

Heat supplied
Heat supplied or heat input represents the heat supplied in combustor chamber.

Heat input $Q_H = (c_{pg} \times T_3 - c_{pa} \times T'_2)$ (16)

2.2. Software Analysis
The analysis of gas turbine cycle to examine the performance of plant has been done using software developed in MATLAB13. The results are shown in the form of graphs of various parameters calculated using developed software.

3. RESULTS AND DISCUSSION
In the following section the results of the present work have been discussed.
Fig. 3 depicts the variation of ambient temp and different pressure ratio on thermal efficiency. It is evident from Fig. 3 that, as pressure ratio increases from 6 to 21, the thermal efficiency also increases for a given ambient temperature; however it is also concluded from fig. if there is increase in the ambient temperature from 283 K to 323 K, there is decrease in the thermal efficiency for all range of pressure ratios being considered. Thermal efficiency is higher at lower temperatures but its value slightly decreases as there is increase in temperature. It is also observed that, there is a loss of 3.28 % in thermal efficiency at 9 pressure ratio as ambient temperature increases from 283 K to 323 K, and this loss increases up to 3.97 % at 21 pressure ratio, which conflicts that, as pressure ratio and ambient temperature increases, there is a significant drop in thermal efficiency. The thermal efficiency is affected by ambient temperature due to the change of air density and compressor work. Since a lower ambient temperature leads to a higher air density and a lower compressor work gives a higher gas turbine output which in turn provides a higher thermal efficiency. Nevertheless higher ambient temperature leads to a lower air density and a higher compressor work. In this study compressor work is more as compared to turbine work leading to a decrease in net work done, which may give a way to lower the thermal efficiency at higher ambient temperatures.

Fig. 4 depicts the variation of ambient temp and different pressure ratio on net power. It is also evident from the Fig. 4 that, as pressure ratio increases from 6 to 21 the net power also increases for a given ambient temperature. However it is also concluded from fig. if there is increase in the ambient temperature from 283 K to 323 K, there is decrease in the net power for all range of pressure ratios being considered. The net power is higher at lower temperatures but its value slightly decreases as there is increase in
temperature. It is also observed that, there is a loss of 3.87% in net power at 9 pressure ratio as ambient temperature increases from 283 K to 323 K, and this loss increases up to 4.46% at 21 pressure ratio, which conflicts that, as pressure ratio and ambient temperature increases, there is a significant drop in net power. It has been observed that the loss of net power is more as compared to thermal efficiency at higher pressure ratios.

Figure 5 Variation of ambient temp and different pressure ratio on specific fuel consumption

Fig. 5 shows variation of ambient temp and different pressure ratio on specific fuel consumption. There is a reverse phenomenon of sfc as compared to thermal efficiency and net power. It has been observed that as pressure ratio increases the value of sfc decreases but for a particular pressure ratio as ambient temperature increases value of sfc increases. This is because of the fact that the air mass flow rate at compressor input increases with increase in ambient temperature. So, the fuel mass flow rate will increase, since air to fuel ratio is kept constant. The power increase is less than that of inlet compressor air mass flow rate, therefore, the specific fuel consumption increases with the increase of ambient temperature. As the pressure ratio increases, the air exiting the compressor is hotter; therefore less fuel is required to reach the desired turbine inlet temperature (TIT) and specific fuel consumption decreases as pressure ratio increases.

Figure 6 Variation of ambient temp and different pressure ratio on work ratio
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Fig. 6 shows variation of ambient temp and different pressure ratio on work ratio. It is evident from the Fig. 6 that as pressure ratio increases from 6 to 21 the value of work ratio decreases but from the graph it also reflect that as ambient temperature increases from 283 K to 323 K, the value of work ratio also decreases. It is higher at lower temperatures but its value slightly decreases as temperature increases.

![Heat Input vs Ambient Temperature](image)

**Figure 7** Variation of ambient temp and different pressure ratio on heat input

Fig. 7 shows variation of ambient temp and different pressure ratio on heat input. It is evident from the Fig. 7 that as pressure ratio increases from 6 to 21 the value of heat input decreases but from the graph it also reflect that as ambient temperature increases from 283 K to 323 K, the value of heat supplied decreases. It is higher at lower temperatures but its value slightly decreases as temperature rises.

![Air Rate vs Ambient Temperature](image)

**Figure 8** Variation of ambient temp and different pressure ratio on air rate

Fig. 8 shows variation of ambient temp and different pressure ratio on air rate. It has been observed that as pressure ratio increases the value of air rate decreases but for a particular pressure ratio as ambient temperature rises, the value of air rate increases. It is higher at higher temperatures but its value slightly decreases as temperature gets down.
4. CONCLUSION
The simple gas turbine power plant has been analyzed for various parameters. The most important parameters which have been covered in this work are the varying ambient temperature and pressure ratio of compressor. The graphical representation of data as given in preceding paragraphs, demonstrate that the gas turbine thermal efficiency and its useful power output varies with the ambient temperature. At higher ambient temperatures (other than ISO conditions) the thermal efficiency, useful power output, work ratio and heat input tend to decreased but specific fuel consumption and air rate will be increases. As the ambient temperature decreases, the thermal efficiency and net work done increases. As the pressure ratio increases the thermal efficiency increases and specific fuel consumption decreases.

5. NOMENCLATURE
T₁ Temperature at gas turbine compressor inlet-K
T₂ Temperature at gas turbine compressor outlet-K
T₂' Actual temperature at gas turbine compressor outlet-K
T₃ Temperature at gas turbine inlet-K
T₄ Temperature at gas turbine outlet-K
T₄' Actual temperature at gas turbine outlet-K
T-s Temperature-entropy-K-j/kg K
γ Ratio of specific heats
η Cycle efficiency
p₁ Pressure at gas turbine compressor inlet-bar
p₂ Pressure at gas turbine compressor outlet-bar
p₃ Pressure at gas turbine inlet-bar
p₃' Actual pressure at gas turbine inlet-bar
p₄ Pressure at gas turbine outlet-bar
η_c Compressor efficiency
η_t Gas turbine efficiency
r_p Pressure ratio
c_p,a Specific heat constant for air-KJ/kg K
c_p,g Specific heat constant for gas-KJ/kg K
A/F Air to fuel ratio
η_comb Combustion chamber efficiency
WR Work ratio
sfc Specific fuel consumption
C.V. Calorific value of fuel used-KJ/kg
η_pc Polytropic efficiency of compressor
η_pt Polytropic efficiency of turbine
w_net Net power available
η_tha Actual thermal efficiency
w_c Isentropic compression work
w_ca Actual compression work
w_ta Actual turbine work
w_t Isentropic turbine work
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\[ w_{\text{net}} \quad \text{Net power available-MW} \]
\[ Q_H \quad \text{Heat input at constant pressure-(kJ)} \]
\[ \dot{m}_a \quad \text{mass flow rate of air-kg/s} \]
\[ \dot{m}_f \quad \text{mass flow rate of fuel-kg/s} \]

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REFERENCES


