THERMAL ANALYSIS OF A GAS TURBINE POWER PLANT TO IMPROVE PERFORMANCE EFFICIENCY

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ABSTRACT

The gas turbine cycle has various uses in the present scenario. The ancient and mostly use of gas turbine cycle for the generation of power. The gas turbine cycle is based on Braton cycle. In the present work the parametric study of a gas turbine cycle model power plant with intercooler compression process and regeneration turbine were proposed. The thermal efficiency, specific fuel consumption and net power output are simulating with respect to the temperature limits and compressor pressure ratio for a typical set of operating conditions. 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. Regenerative gas turbine engine cycle is presented that yields higher cycle efficiencies than simple cycle operating under the same conditions. The analytical formulae about the relation to determine the thermal efficiency are derived taking into account the effected operation conditions (ambient temperature, compression ratio, intercooled effectiveness, regenerator effectiveness, compressor efficiency, turbine efficiency, air to fuel ratio and turbine inlet temperature). The analytical study is done to investigate the performance improvement by intercooling and regeneration. The analytical formula for specific work and thermal efficiency are derived and analyzed. The simulation results shows that increasing turbine inlet temperature and pressure ratio can still improve the performance of the intercooled gas turbine cycle. The power output and thermal efficiency are found to be increasing with the regenerative effectiveness, and the compressor and turbine efficiencies. The efficiency increased with increase the compression ratio to 5, then efficiency decreased with increased compression ratio, but in simple cycle the thermal efficiency always increase with increased in compression ratio. The increased in ambient temperature caused decreased thermal efficiency, but the increased in turbine inlet temperature increase thermal efficiency.

Keywords: Gas turbine, Intercooling, Regeneration, Thermal efficiency, Power plant, Brayton cycle.
1. INTRODUCTION TO GAS TURBINES

The world energy demand has increased steadily and will continue to increase in the future: the International Energy Agency (IEA) predicts an increase of 1.7% per year from 2000 to 2030. This increase corresponds to two thirds of the current primary energy demand, which was 9179 Mtoe in 2000, and in 2030, fossil fuels will still account for the largest part of the energy demand. In addition, the IEA predicts that the demand for electricity will grow by 2.4% per year and that most of the new power generating capacity will be natural gas-fired combined cycles [1] Therefore, it is important to find improved technologies for power generation with high electrical efficiencies and specific power outputs (kJ/kg air), low emissions of pollutants and low investment, operating and maintenance costs for a sustainable use of the available fuels. Advanced power cycles based on gas turbines can meet these requirements, since gas turbines have relatively high efficiencies, low specific investment costs (USD/kW), high power-to-weight ratios and low emissions. The power markets have been deregulated in several countries and distributed generation and independent power producers have become more competitive. These changes require flexible power plants with high efficiencies for small-to-medium power outputs. As a result of this, it was estimated that more than half of the orders for new fossil-fueled power plants in the last part of the 1990s were based on gas turbines [2], since non-expensive and clean natural gas was available, and the demand for gas turbines continues to increase [4].

2. MODELLING OF THE COMPONENTS

The thermodynamic properties of air and products of combustion are calculated by considering variation of specific heat and with no dissociation. The curve fitting the data is used to calculate specific heats, specific heat ratio, and enthalpy of air and fuel separately from the given values of temperature. Mixture property is then obtained from properties of the individual component and fuel air ratio (FAR).

Combustion Products (72.54% N\textsubscript{2}, 6.48% O\textsubscript{2}, 0.86% Ar, 13.46% H\textsubscript{2}O, 6.66% CO\textsubscript{2}) Specific heat of the gases is assumed only function of temperature alone. Polynomial fits for the specific heats of each of those three components as a function of temperature are used in the calculations. The polynomial fit for specific heat is taken from [20]. Those polynomials are used to calculate the specific heats of air and gas as a function of temperatures are given by:

\[ c_{pa} = (1.0189 \times 10^3) - (0.13784 \times T_a^4) + (1.9843 \times 10^{-4} \times T_a^2) + (4.2399 \times 10^{-7} \times T_a^4) - (3.7632 \times 10^{-10} \times T_a^4) \]

\[ c_{pg} = 0.0086 \cdot c_{p,Ar} + 0.7154 \cdot c_{p,N_2} + 0.0648 \cdot c_{p,O_2} + 0.1346 \cdot c_{p,H_2O} + 0.0666 \cdot c_{p,CO_2} \]

In the above equations, \( T \) stands for gas or air temperature in deg K and \( \tau = \frac{T}{100} \).

2.1 Gas Turbine analysis with Intercooling

Consider replacing the isentropic single-stage compression from \( p_1 \) to \( p_2 \) in figure 2 with two isentropic stages from \( p_1 \) to \( p_2 \) and \( p_2 \) to \( p_3 \). Separation of the compression processes with a heat exchanger that cools the air at \( T_2 \) to a lower temperature \( T_3 \) acts to move the final compression
process to the left on the T-s diagram and reduces the discharge temperature following compression to $T_4$. The work required to compress air from $p_1$ to $p_2$ in two stages is given by considering two compressors namely low pressure compressor and high pressure compressor. Therefore, work required by the low pressure and high pressure compressor depends upon their pressure ratios.

For low pressure compressor the work required in isentropic compression is given by,

$$W_{lp} = c_p (T_2 - T_1)$$

(3)

For the high pressure compressor, the work required in isentropic compression is given by,

$$W_{hp} = c_p (T_4 - T_3)$$

(4)

Therefore, the total work required by the compressor is given by

$$W_c = W_{lp} + W_{hp}$$

(5)

In the present work the intercooler effectiveness is given by [22].

$$\varepsilon = \frac{T_2 - T_3}{T_2 - T_1}$$

(6)

$$\eta_{lp} = \frac{T_2 - T_1}{T_2 - T_1}$$

(7)

$$\eta_{hp} = \frac{T_4 - T_3}{T_4 - T_3}$$

(8)

Note that intercooling increases the net work of the reversible cycle. Thus intercooling may be used to reduce the work of compression between two given pressures in any application. However, the favorable effect on compressor work reduction due to intercooling in the gas turbine application may be offset by the obvious increase in combustor heat addition, $(c_p T_e - c_p T_f)$, and by increased cost of compression system.

**Figure 1** Schematic of Intercooling gas turbine cycle
2.2 Gas Turbine analysis with Regeneration

Fig. 4 shows the T-S diagram for regenerative gas turbine cycle. The actual processes and ideal processes are represented in dashed line and full line respectively. The compressor efficiency ($\eta_c$), the turbine efficiency ($\eta_t$) and effectiveness of regenerator (heat exchanger) are considered in this study. These parameters in terms of temperature are defined as in [13]:

$$\eta_c = \frac{T_2 - T_1}{T_2 - T_1}$$  \hspace{1cm} (9)

$$\eta_t = \frac{T_3 - T_4}{T_2 - T_1}$$  \hspace{1cm} (10)

$$\varepsilon_{tc} = \frac{T_5 - T_2}{T_4 - T_2}$$ \hspace{1cm} (11)

The work required to run the compressor is expressed as in [13]:

$$W_c = c_p a T_1 \left[ \frac{\gamma a^{\gamma - 1} \gamma^{-1}}{\eta_c} \right]$$  \hspace{1cm} (12)

The work developed by turbine is then rewritten as in (2):

$$W_t = c_p g T_4 \eta_t \left[ 1 - \frac{1}{\gamma^{\gamma - 1} \gamma^{-1}} \right]$$  \hspace{1cm} (13)

where $T_4$ is turbine inlet temperature. The net work is expressed as [13]

$$W_{net} = W_t - W_c$$  \hspace{1cm} (14)

or

\hspace{1cm}
In the combustion chamber, the heat supplied by the fuel is equal to the heat absorbed by air, Hence,

\[W_{\text{net}} = c_{pg} T_4 \eta_t \left(1 - \frac{1}{r_{p_g}^{\gamma_f}}\right) - c_{pa} T_1 \left(\eta_c \left(1 - \frac{T_{a_1}^{\gamma_f}}{r_{p_a}^{\gamma_f} - 1}\right)\right)\]  

(15)

Power output is given by:

\[P = \dot{m}_a \times W_{\text{net}}\]  

(17)

Air to fuel ratio is given by

\[\text{AFR} = \frac{\text{LCV}_f}{Q_{\text{add}}}\]  

(18)

and Specific Fuel consumption

\[\text{SFC} = \frac{3600}{\text{AFR} \times W_{\text{net}}}\]  

(19)

Fuel to air ratio is given by

\[\text{FAR} = 1/\text{AFR}\]  

(20)

Thermal Efficiency is given by

\[\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_{\text{add}}}\]  

(21)

Fig. 3 Schematic of a Regenerative gas turbine
3 RESULTS AND DISCUSSION

3.1 Intercooling Gas Turbine Cycle

In the present work, two compressors high pressure (HP) and low pressure (LP) and a single turbine have been used for intercooling gas turbine cycle. For regenerative gas turbine cycle, one compressor and one turbine have been used for their analysis. The cycle was modeled using the thermodynamic analysis for the simple gas turbine, Intercooling gas turbine and regenerative gas turbine. The pressure losses are assumed in this work in various components.

The effect of thermal efficiency, specific fuel consumption, pressure ratio across the compressor, turbine inlet temperature (TIT), ambient temperature (Tamb), effectiveness of intercooler and effectiveness of regenerator on the first-law efficiency and power are obtained by the energy-balance approach or the first-law analysis of the cycle programming using C++ software.

Figure 5 shows the effect of ambient temperature on the efficiency of gas turbine cycle with intercooler effectiveness at a given value of turbine inlet temperature (TIT=1500 K) and compressor pressure ratio (\(r_p=30\)). It is clear from the figure that decreasing the ambient temperature increases the gain in efficiency.

![Figure 4](image1.png)

Figure 4 T-s representation of Regenerative gas turbine cycle

![Figure 5](image2.png)

Figure 5 Effect of Ambient temperature and intercooler effectiveness on thermal efficiency
Power output decreases on increasing the ambient temperature as shown in Figure 6.

Figure 6 Effect of ambient temperature and intercooler effectiveness on power output

Figure 7 shows the variation of compressor work with compressor pressure ratio for different values of intercooler effectiveness. It is to be noted that the compressor work increases on increasing the pressure ratio for a given value of atmospheric temperature and low pressure ratio. It also observed that the compressor work decreases on increasing the intercooler effectiveness for a fixed value of compressor ratio.

Figure 7 Effect of compression ratio and compressor work on intercooler effectiveness

It is shown in the figure 8 that work ratio increases on increasing the turbine inlet temperature for a given intercooler effectiveness, compressor pressure ratio and ambient temperature. It is also noticed from that figure work ratio increases on increasing the intercooler effectiveness for a given TIT, compressor ratio and ambient temperature.
3.2 Regenerative Gas Turbine Cycle

The figure 9 to figure 12 is drawn for the regenerative gas turbine cycles. Figure 9 shows the effect of ambient temperature and regenerative effectiveness on thermal efficiency of gas turbine cycle. Turbine inlet temperature (TIT) and compressor ratio (r_p) are of 1700 K, 20. It can be seen that the thermal efficiency decreases with increases of ambient temperature while decreases of regenerative effectiveness.

The variation of specific fuel consumption with ambient temperature is also shown in Figure 10. It shows that when the ambient temperature increases the specific fuel consumption increases too. This is because, the air mass flow rate inlet to compressor increases with decrease of the ambient temperature. So, the fuel mass flow rate will increase, since (AFR) is kept constant. The power increase is less than that of the inlet compressor air mass flow rate (m_a) therefore, the specific fuel consumption increases with the increase of ambient temperature.
Figure 11 Influence of ambient temperature on SFC with various compression ratio

Figure 12 Effect of isentropic turbine efficiency and ambient temperature on thermal efficiency

Figure 4.30 Variation of thermal efficiency with ambient temperature for various cycles
Figure 11 Effect of isentropic turbine efficiency and ambient temperature on thermal efficiency. It is found that the thermal efficiency increases on increasing the isentropic turbine efficiency for a given value of ambient temperature. At the same time, the thermal efficiency decreases on increasing the ambient temperature for a given value of isentropic turbine efficiency.

Figure 12 shows the variation of thermal efficiency with ambient temperature for all the three different types of cycles which have been taken for the analysis in the present work. It is observed from the figure that the thermal efficiency is highest with the regenerative gas turbine cycle for any values of ambient temperature. The intercooled cycle has minimum thermal efficiency comparing with regenerative cycle at all the values of ambient temperatures. In both the cases the effectiveness of regenerator and intercooler has been taken as 0.95 and turbine inlet temperature is 1700K. Thermal efficiency for simple cycle gas turbine is smaller than both of them at all the values of ambient temperature and same turbine inlet temperature.

CONCLUSION

The present work determined the performance of a regenerative and intercooled gas turbine power plant. A design methodology has been developed for parametric study and performance evaluation of a regenerative and intercooled gas turbine. Parametric study showed that compression ratio \( r_p \), ambient temperature and turbine inlet temperature (TIT) played a very vital role on overall performance of a regenerative and intercooled gas turbine. The simulation result from the analysis of the influence of parameter can be summarized as follows:

1. The heat duty in the regenerator decreases with the pressure ratio but increases with the decreases ambient temperature and increases TIT this mean increased thermal efficiency.
2. The thermal efficiency of the simple gas-turbine cycle experiences small improvements at large pressure ratios as compared to regenerative gas turbine cycle.
3. In general, peak efficiency, power and specific fuel consumption occur at compression ratio \( r_p = 5 \) in the regenerative gas turbine cycle.
4. The thermal efficiency increases and specific fuel consumption decreases with the regenerator effectiveness.
5. The thermal efficiency increases and specific fuel consumption decreases with increase in the intercooler effectiveness.
6. The thermal efficiency of the simple gas-turbine cycle experiences small improvements at large compression ratios as compared to gas turbine cycle with intercooler.
7. The peak efficiency, power and specific fuel consumption occur when compression ratio increases in the gas turbine cycle with intercooler.
8. Maximum power for the turbine inlet temperature is selecting an optimum value of compression ratio and turbine inlet temperature, which will result in a higher thermal efficiency.

NOMENCLATURE

1, 2, 3, …. are the state points
\( P_{amb} = \) Ambient Pressure
\( Q_{add} = \) Heat added to combustor
\( \eta_t = \) Turbine Isentropic Efficiency
\( \eta_c = \) Compressor Isentropic Efficiency
\( \eta_h = \) Combustor or Burner Efficiency
\[ \eta_m = \text{Mechanical Efficiency} \]
\[ R_g = \text{Gas constant for gas} \]
\[ R_a = \text{Gas constant for air} \]
\[ LCV_f = \text{Lower calorific value of fuel} \]
\[ T_{amb} = \text{Ambient temperature} \]
\[ m_a = \text{Mass of air} \]
\[ T_{IT} = \text{Turbine inlet temperature} \]
\[ \text{EFFC} = \text{Intercooler Effectiveness} \]
\[ \text{RGEFF or } \alpha_r = \text{Regenerator effectiveness} \]
\[ \text{LPR} = \text{Low pressure ratio} \]
\[ \text{OPR} = \text{Overall or compressor pressure ratio} \]
\[ \text{GT} = \text{Gas turbine} \]
\[ \text{IGT} = \text{Basic gas turbine with inter cooling} \]
\[ \text{RGT} = \text{Basic gas turbine with regeneration} \]
\[ W_{th} = \text{Network output} \]
\[ W_c = \text{Compressor Work} \]
\[ W_t = \text{Turbine Work} \]
\[ c_{pg} = \text{Specific Heat of gases} \]
\[ c_{pa} = \text{Specific heat of air} \]
\[ r \text{ or } r_p = \text{compressor pressure ratio} \]
\[ P = \text{power} \]
\[ \Delta p_{cc} = \text{Pressure drop in combustor} \]
\[ p_b = \text{Pressure in combustor} \]
\[ P_n = \text{Net Power developed} \]
\[ W_f = \text{Work ratio} \]

REFERENCES