Thermal Performance of Reheat, Regenerative, Inter-Cooled Gas Turbine Cycle

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Abstract

Thermal analysis of reheat, regenerative, inter-cooled gas turbine cycle is presented. Specific work output, thermal efficiency and SFC is simulated with respect to operating conditions. Analytical formulae were developed taking into account the effect of operational parameters like ambient temperature, compression ratio, compressor efficiency, regenerator effectiveness, pressure loss in inter cooling, reheating and regenerator. Calculations were made for wide range of parameters using engineering equation solver and the results were presented here. For pressure ratio of 12, regenerator effectiveness 0.95, and maximum turbine inlet temperature 1200 K, thermal efficiency decreases by 27% with increase in ambient temperature (278 K to 328 K). With decrease in regenerator effectiveness thermal efficiency decreases linearly. With increase in ambient temperature (278 K to 328 K) for the same maximum temperature and regenerator effectiveness SFC decreases up to a pressure ratio of 10 and then increases. Sharp rise in SFC is noted for higher ambient temperature. With increase in isentropic efficiency of compressor and turbine, thermal efficiency increases by about 40% for low ambient temperature (278 K to 298 K) however, for higher ambient temperature (308 K to 328 K) thermal efficiency increases by about 70%

Keywords

Gas Turbine; Reheating, Regeneration, Inter-Cooled, Thermal Analysis

I. Introduction

Oil crises in today situation have gained more importance in power generation system. Simple gas turbine cycle have low efficiency due to loss of energy at the turbine exhaust. The temperature of the gas leaving the turbine is higher than the temperature of air leaving the compressor. Hence air leaving the compressor may be heated by the turbine exhaust gases. Such cycles are referred as regenerative cycles. Regenerator with higher effectiveness is better how ever that make it larger and high pressure drop. Generally the air pressure drop is 2% of compressor discharge pressure and regenerator effectiveness is less than 85% [1]. Regeneration is most effective only at low pressure ratio and low minimum to maximum temperature ratio. Specific work output increases if expansion is two stage with intermediate heating. Reheating increases the specific work output at the expenses of efficiency. However reduction in efficiency is less severe as the maximum cycle temperature is increased. High exhaust gas temperature can be utilized in the regenerator and loss in efficiency can recover. Similarly specific output can be increased by employing inter cooling between HP and LP compressor [2-4]. All such a cycles are easy to analyze theoretically without considering the effects of component losses. The paper presented here is an attempt of accounting these losses for such analysis. Real cycle analysis consist of following considerations [4].

1. Fluid velocities are high in turbomachinery hence change in kinetic energy are considered.
2. Fluid friction results in pressure loss in combustion chamber, inter-cooler and heat exchanger hence these losses are considered
3. For the economy considerations of the heat exchanger compress air can not be heated to the exhaust gas temperature hence regenerator effectiveness is considered
4. In order to overcome bearing and windage friction in transmission and to drive ancillary components such as oil pump slightly more work is required hence transmission efficiency is considered
5. The value of cp and γ vary with the change in temperature and chemical composition. Hence effect of variable specific heat is considered
6. Fuel/air effect and combustion efficiency are considered to express the cycle performance unambiguously.

II. Cycle Description and Analysis

Fig. 1 shows gas turbine plant layout considered in this study with inter-cooler, regenerator and reheater. It is free power turbine type. The nonideal cycle is represented on TS diagram. It can be characterized by significant parameters pressure ratio, turbine inlet temperature, ambient temperature and effectiveness of a regenerator. Compression processes 1-2’ and 3-4’ are isentropic processes whereas 1-2 and 3-4 are actual ir-reversible processes. 6-7’ and 8-9’ are isentropic expansion processes for a turbine where as 6-7 and 8-9 are actual ir-reversible processes. T-S diagram also represents pressure drop at intercooler, regenerator, combustion chamber and reheater.

A. Assumptions

Following assumptions were made during the analysis and Table 1 represents summary of input data for analysis.
1. Gas is reheated in a reheater to its maximum temperature
2. The values of γ_\text{\text{air}} and γ_\text{\text{gas}} are assumed to constant as 1.4 and 1.33 respectively
3. Velocity of a fluid at inlet to the compressor is constant
4. Power output is constant and it is 5 MW

![Fig. 1: Gas Turbine Plant(Intercooled, Regenerative and Reheat)](image-url)
Table 1: Input Data

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Power</td>
<td>5000 kW</td>
</tr>
<tr>
<td>2</td>
<td>Ambient Temperature</td>
<td>20 °C</td>
</tr>
<tr>
<td>3</td>
<td>$\gamma_{\text{air}}$</td>
<td>1.4</td>
</tr>
<tr>
<td>4</td>
<td>$\gamma_{\text{gas}}$</td>
<td>1.33</td>
</tr>
<tr>
<td>5</td>
<td>Atmospheric Pressure</td>
<td>101.325 kPa</td>
</tr>
<tr>
<td>6</td>
<td>Heating Value of Fuel</td>
<td>48000 KJ</td>
</tr>
<tr>
<td>7</td>
<td>Pressure Loss in Intercooler</td>
<td>1 %</td>
</tr>
<tr>
<td>8</td>
<td>Pressure Loss in Heat Exchanger</td>
<td>2 %</td>
</tr>
<tr>
<td>9</td>
<td>Pressure Loss in Combustion Chamber</td>
<td>2 %</td>
</tr>
<tr>
<td>10</td>
<td>Heat Removed in Intercooler</td>
<td>85%</td>
</tr>
</tbody>
</table>

5. Combustion is assumed as complete combustion

B. Cycle Analysis

[5] represented a table containing the values of specific heat against temperature variation and [6] in his work fitted a polynomial to obtain the equations (1) - (5). These equations were used to calculate the values of specific heat. For a temperature range 200 - 800 K

$$c_{pa} = (1.0189 \times 10^3) - (0.13784 \times T_a) + (1.9843 \times 10^{-4} T_a^2) + (4.2399 \times 10^{-7} T_a^3) - (3.7632 \times 10^{-10} T_a^4)$$  (1)

For a temperature range 800 - 2200 K

$$c_{pa} = (7.9865 \times 10^3) + (0.5339 T_a) - (2.2882 \times 10^{-4} T_a^2) + (3.7421 \times 10^{-7} T_a^3)$$  (2)

For specific heat of gases

$$c_{pg} = c_{pa} + \left[ \frac{f}{1+f} \right] B_T$$  (3)

$$B_T = (-5.3949 \times 10^2) + (4.5164 T_a) + (2.8116 \times 10^{-3} T_a^2)$$
$$- (2.1609 \times 10^{-5} T_a^3) + (2.8689 \times 10^{-8} T_a^4) - (1.263 \times 10^{-11} T_a^5)$$  (4)

For a temperature range of 800 - 2200 K

$$B_T = (1.0888 \times 10^4) - (0.1416 T_a) + (1.916 \times 10^{-3} T_a^2)$$
$$- (1.2401 \times 10^{-6} T_a^3) + (3.0669 \times 10^{-10} T_a^4)$$
$$- (2.6117 \times 10^{-13} T_a^5)$$  (5)

Inlet stagnation temperature to the compression process is calculated as

$$T_o1 = T_a + \left( \frac{c_{p1}}{2 C_{pa}} \right)$$  (6)

$$T_o2 = T_o1 + \left( \frac{T_o1}{\eta_c} \right) \left[ \left( \frac{P_{o2}}{P_{o1}} \right)^{\frac{n_s-1}{n_s}} \right]$$  (7)

Intermediate pressure is the pressure with reference to minimum work required for the compression. Isentropic efficiency of both LP and HP compressor is same and given by the equation

$$\eta_{LPC} = \frac{T'_{o2} - T_{o1}}{T'_{o2} - T_{o1}}$$  (8)

$$\eta_{HPC} = \frac{T'_{o4} - T_{o3}}{T'_{o4} - T_{o3}}$$  (9)

Work output of the compressor turbine is sufficient to drive the HP and LP compressor. Air leaving the HP compressor is pass through heat exchanger where it is heated by turbine exhaust gases. There is a pressure drop of 2% in heat exchanger and effectiveness is given by the equation

$$\epsilon = \frac{T_5 - T_4}{T_9 - T_4}$$  (10)

Net specific heat supplied, specific work output and thermal efficiency is given by equation

$$q_s = c_{pg}(T_6 - T_5) + c_{pg}(T_8 - T_7)$$  (11)

$$w_{CT} = \frac{c_{pg}(T_6 - T_7)}{\eta_{\text{transmission}}}$$.  (12)

$$w_{PT} = c_{pg}(T_8 - T_9)$$  (13)

$$\eta_{\text{thermal}} = \frac{w_{PT}}{q_s}$$  (14)

The hydrocarbon fuel chemical reaction is determined by following expression.

$$C_n \: H_m = \lambda \left( \frac{n + m}{4} \right) [O_2 + 3.76 N_2] + (s) H_2O$$
$$+ (n) CO_2 + \left[ (\lambda - 1) \left( \frac{n + m}{4} \right) \right] O_2$$
$$+ \left[ \frac{3.76\lambda (n + m)}{4} \right] N_2 + (m/2 + s) H_2O$$  (15)

Hydrocarbon fuel model is assumed as $C_8H_{18}$. The specific fuel consumption is expressed as

$$SFC = \frac{3600}{(1/f)W_{PT}}$$  (16)
III. Results and Discussion

The cycle was modeled using the thermodynamic analysis for the reheat regenerative gas turbine. The pressure losses are assumed in this study. All the equations were solve using engineering equation solver [7]. The effect of thermal efficiency, power and specific fuel consumption on operation conditions are analyzed in the following section.

A. Effect of Pressure Ratio and t

Fig 3 represents the effect of pressure ratio on thermal efficiency of the plant for different turbine inlet temperature. It is observed that for the same value of t (ratio of minimum temperature to the maximum turbine inlet temperature (TIT)) with increase in pressure ratio efficiency increase up to pressure ratio of 4 then decreases. At higher TIT efficiency is approximately constant after a pressure ratio 4. About 50% increase in thermal efficiency is observed at a pressure ratio of 4 when t decrease from 0.33 to 0.23. Fig 4 represents the effect on specific output of the power turbine. It is observed that for the higher value of t (low TIT) specific output increase up to a pressure ratio of 5.5 and then decreases. However for low values of t specific output of the power turbine increase continually with increase in pressure ratio. About 25% increase in specific output is observed as the t value decrease from 0.25 to 0.23 for a pressure ratio 8 to 12. For a t value of 0.23 and 0.25 as pressure ratio increases up to 6 sharp rise in specific output is observed.

Fig. 5 represents the effect on specific heat added to the cycle. With increase in pressure ratio and decrease in t value (increase in TIT) heat required to be added for the same power output is increasing. Rate of increase in heat is more for the pressure ratio about 5.5 and then decreases. For a maximum pressure ratio of 12 about 65% increase is observed when the t value decreases from 0.33 to 0.23.

Fig. 6 to 10 represents effect of pressure ratio on specific output and thermal efficiency for the given value of t. Following observations where made:

- At t = 0.33 maximum specific output is observed at a pressure ratio of 5.5 and the maximum efficiency is at 4.5.
- At t = 0.3 maximum specific output is observed at a pressure ratio of 8 and the maximum efficiency is at 4.5.
- At t = 0.27 maximum specific output is observed at a pressure ratio of 10 and the maximum efficiency is at 5.
- At t = 0.25 maximum specific output is observed at a pressure ratio of 12 and the maximum efficiency is at 5.
- At t = 0.23 maximum specific output is observed at a pressure ratio of 12 and the maximum efficiency is at 6.
B. Effect of Ambient Temperature

Fig 11 represents the effect of ambient temperature and regenerative effectiveness on thermal efficiency. Turbine inlet temperature is 1200 K and pressure ratio is 12 as specific output is maximum at this ratio as discussed in previous section. With increase in ambient temperature thermal efficiency of the cycle decreases. For regenerator effectiveness of 0.95 efficiency decrease by about 37%. For regenerator effectiveness of 0.75 efficiency decreases by about 25%. For regenerator effectiveness of 0.5 efficiency decreases by about 35%. This implies that with increase in temperature rate of decrease is minimum for regenerator with effectiveness in the range 0.65 - 0.75. Also in order to limit the cost most of the regenerators with an effectiveness of 0.75 can be selected. Further analysis is therefore done with an effectiveness of 0.75.

Fig 12 represents the effect of pressure ratio and ambient temperature on thermal efficiency of the cycle. Regenerator effectiveness is 0.75, turbine inlet temperature (TIT) is 1200K and isentropic efficiency of the compressor and turbine is 0.85. It is observed that for a temperature range 278 - 298 K thermal efficiency first increases up to a pressure ratio of 12 and then decreases with increase in pressure ratio. At higher ambient temperature thermal efficiency decrease with increase in pressure ratio. At a low temperature of 278 K only about 10% of change in efficiency is observed with increase in pressure ratio. At an ambient temperature of 318 K change in efficiency is about 25% whereas at 328 K change in efficiency is about 45% with increase in pressure ratio.

Fig 13 represents effect of pressure ratio and ambient air temperature on specific fuel consumption. Regenerator effectiveness is 0.75, turbine inlet temperature (TIT) is 1200K and isentropic efficiency of the compressor and turbine is 0.85. It
up to a pressure ratio of 12 and then again increases. At higher ambient air temperature SFC curve looks a hook curve where as low ambient air temperature SFC is approximately constant with increase in pressure ratio beyond 12. Minimum SFC at 328 K is about 0.22 kg/kWh and at 278 K is about 0.15 kg/kWh Fig 14 represents effect of pressure ratio and regenerator effectiveness on thermal efficiency of the cycle. At higher effectiveness thermal efficiency decreases with increase in pressure ratio. For an effectiveness of 0.75 to 0.55 thermal efficiency increases upto a pressure ratio of 12 and then decreases. At a pressure ratio of 12 for an effectiveness of 0.75 maximum efficiency is about 0.33. Fig 15 represents effect of isentropic efficiency of compressor and ambient air temperature on thermal efficiency of the compressor. It is observed that thermal efficiency of the cycle increases with increase in isentropic efficiency of the compressor and decreases with increase in ambient temperature. Similar effect is observed for isentropic efficiency of turbine. Fig 16 represents effect of isentropic efficiency of turbine.

**IV. Conclusion**

Gas turbine cycle with reheat, regeneration and inter cooling is analyzed taking in to considerations of component losses. Analysis was done with engineering equation solver. Effect of pressure...
ratio, ratio of turbine inlet temperature to minimum temperature, regenerator effectiveness, ambient temperature, isentropic efficiency of compressor and turbine over thermal efficiency was analyze and following conclusions were made

1. About 50% increase in thermal efficiency is observed at a pressure ratio of 4 when t decrease from 0.33 to 0.23.
2. About 25% increase in specific output is observed as the t value decrease from 0.25 to 0.23 for a pressure ratio 8 to 12.
3. For a maximum pressure ratio of 12 about 65% increase is observed when the t value decreases from 0.33 to 0.23
4. With increase in ambient temperature thermal efficiency of the cycle decreases. For regenerator effectiveness of 0.95 efficiency decrease by about 37%. For regenerator effectiveness of 0.75 efficiency decreases by about 25%.
5. At higher ambient air temperature SFC curve looks a hook curve where as low ambient air temperature SFC is approximately constant with increase in pressure ratio beyond 12. Minimum SFC at 328 K is about 0.22 kg/kWh and at 278 K is about 0.15 kg/kWh
6. For an effectiveness of 0.75 to 0.55 thermal efficiency increases up to a pressure ratio of 12 and then decreases.

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References