Optimum Cycle Parameters of Coal Fired Closed Cycle Gas Turbine in Regenerative and Combined Cycle Configurations

This paper presents the methodology developed for the estimation of thermodynamic performance and reports the optimum cycle parameters of coal fired CCGT in regenerative and Combined Cycle Configurations using Air, Helium and Carbon-Dioxide as working gases. A rigorous approach has been followed for the determination of the cycle efficiency by assuming the specific heat of working gases as a continuous function of temperature for accurate estimation of cycle parameters. The performance evaluation of CCGT in combined cycle mode of operation is carried out to optimize the steam bottoming cycle parameters and hence the compressor pressure ratio for achieving the maximum cycle efficiency. From the study it is observed that Helium is the most suitable working gas as it produces high thermal efficiency at a low compressor pressure ratio.

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1. INTRODUCTION

Coal is the most abundant source of commercial fuel in India. In order to offset the impact of rising cost of crude oil on the economy of the country, there is an urgent need to shift to coal based energy conversion systems. The paper presents the methodology developed for thermodynamic analysis of a closed cycle gas turbine (CCGT) where the working gas is heated indirectly in an atmospheric fluidized bed coal combustor. An attempt has been made to present the optimum cycle parameters of CCGT in regenerative and combined cycle configurations using Air, Helium and Carbon Dioxide as working gases.

The method of estimating the cycle parameters of CCGT reported by Shivakura, T., et al [1] assumes constant specific heat for working gases as a first approximation and hence a needed improvisation of the method. In
the present study, a very rigorous approach is adopted wherein the specific heat of working gas is taken as continuous function of temperature throughout the thermodynamic analysis for accurate estimation of cycle parameters. For regenerative configuration with the volumetric flow rate at inlet to the compressor held constant, the method takes into account the effect of the variation of pressure loss coefficients with viscosity and molecular weight of working gases of each of the heat exchangers. The optimum compressor pressure ratio of CCGT in the above configuration is reported for varying recuperator effectiveness and for range of turbine entry temperatures (TET's).

The study is extended to include the thermodynamic analysis of CCGT in combined cycle configuration with dual pressure Waste Heat Boiler (WHB) in the steam bottoming plant. The optimum cycle parameters of combined cycle plant has been arrived at using properties of working gases [2,3] as reported latest in the literature. The optimization criterion [4,5,6] has been the maximization of Combined Cycle Plant efficiency. In the first place, the optimum cycle parameters of the bottoming steam plant have been computed for given compressor pressure ratio and TET. Finally the optimum compressor pressure ratio of CCGT is presented for the selected TET.

2. THERMODYNAMIC DESCRIPTION OF WORKING GASES

In order to achieve high and consistent level of accuracy the thermodynamic properties of working gases are calculated by considering the specific heat as function of temperature given by:

\[ C_p = C_o + C_1 T + C_2 T^2 + \ldots + C_7 T^7 \]  

The specific enthalpy and entropy function are then obtained by integration of polynomial for specific heat

\[ h = \int_o^T C_p dT + C_h; \phi = \int_o^T \frac{C_p dT}{T} + \phi \]  

During any cycle calculation it is often necessary to determine the temperature from the known values of enthalpy and entropy functions. Accordingly enthalpy and entropy equations are to be rearranged to facilitate an iterative solution for temperature. The coefficients of the polynomial of temperature for evaluating the specific heat of working gas is given in Table 1. The thermodynamic properties of water/steam for steam bottoming cycle calculation are computed using the equations given by Rivikin, S.L., et al [7] and covering pressures up to 30 MPa and temperatures up to 873 K in saturated as well as superheated regions.

3. METHODOLOGY FOR PERFORMANCE ESTIMATION OF CCGT

The thermodynamic performance of CCGT is evaluated under two conditions. One is the case where in the volumetric flow rate is varied such that the total pressure loss coefficients of each of the heat exchangers are constant for the working gases. In the second case, the working volume of the circuit is held constant and the consequent variation of total pressure loss coefficients is accounted for the working gases. For CCGT in combined cycle mode, the relative performance of different working gases is given based on constant mass flow at inlet to the compressor.

In the performance estimation of CCGT, the maximum system pressure is selected to be 3039 MPa (30 ata) from the current practice [8]. As the effect of pressure of this magnitude on the thermodynamic properties is not very significant and as such, the computed thermal efficiency does not differ much, the system pressure is considered as constant throughout the analysis. The following describes the methodology adopted for computing the theoretical thermal efficiency of CCGT for the above mentioned cases.

3.1. Performance of CCGT in Regenerative Configuration

A schematic diagram of the CCGT with regenerative system is shown in Fig. 1. The working medium flows under elevated pressure through a compressor, tube side of recuperator, gas heater tubes immersed in the bed of fluidized combustor expands through turbine, enters the shell side of recuperator in the counter flow direction and finally passes through a precooler to bring back the temperature to the compressor inlet condition. The station numbers used for representing the equations of the GT component processes are given in Fig. 2. The following is the sequence of calculation performed for evaluating the CCGT performance.

3.1.1. Compressor. The compressor inlet temperature of 313 K is selected in the
Fig. 1. Schematic of closed cycle gas turbine in regenerative configuration.

The present work is based on the available cooling water temperature of 293 K to 303 K (which is due to high ambient temperatures prevailing in the country). From the known inlet temperature, corresponding enthalpy and entropy function at inlet is evaluated using equation (2). Now, to compute the actual work requirement and exit temperature of compressor, the entropy function at the outlet is obtained from the equation:

\[ \psi_2 = \psi_1 + \ln \left( \frac{P_4}{P_5} \right) \]  

The isentropic temperature at station 2 is then obtained from inverse determination of entropy function. The actual enthalpy of gases at the exit of compressor is derived from the equation of isentropic efficiency of compressor:

\[ \eta_{ic} = \frac{(h_{2i} - h_1)}{(h_2 - h_1)} \]  

Therefore, the actual work requirement of compressor is:

\[ \Delta H_c = \bar{w}_c(h_2 - h_1) \]  

3.1.3. Turbine. The TET, pressure loss coefficients of tube side and shell side of recuperator, gas heater and precooler are specified for determining the exit condition of turbine. The pressure at inlet and exit of turbine are arrived from the equations:

\[ P_4 = P_2(1 - \epsilon_{rec} - \epsilon_{gh}) \]  
\[ P_5 = P_1(1 + \epsilon_{rec} + \epsilon_{cr}) \]  

Hence the entropy function at the exit of turbine is evaluated from:

\[ \psi_5 = \psi_4 + \ln \left( \frac{P_4}{P_5} \right) \]  

As in step 3.1.1, the actual enthalpy at turbine outlet is obtained from the definition of isentropic efficiency of turbine:

\[ \eta_{it} = \frac{(h_4 - h_5)}{(h_4 - h_{5i})} \]  

Therefore, the actual turbine work output is:

\[ \Delta H_t = \bar{w}_t(h_4 - h_5) \]  

3.1.3. Recuperator. The heat in the exhaust after the turbine can be recovered through a gas to gas heat exchanger resulting in an increased cycle efficiency. The enthalpy of the gas at the outlet of recuperator is given by the recuperator effectiveness:

\[ \eta_{rec} = \frac{(h_3 - h_2)}{(h_5 - h_2)} \]  

3.1.4. Gas heater. The working gases leaving the recuperator are heated indirectly, to the maximum cycle temperature in an atmospheric fluidised bed combustor. The heat required to raise the temperature of the fluid from the exit condition of recuperator to TET is computed from the equation:

\[ Q_{in} = \frac{(h_4 - h_3)}{\theta_{gh}} \]  

3.1.5. Theoretical thermal efficiency. From the work balance between the compressor, turbine and auxiliary equipment, the power output of the GT plant is given by the equation:

\[ \Delta H_{gt} = \Delta H_t - \Delta H_c - \Delta H_{aux} \]  

Therefore, the theoretical thermal efficiency of CCGT plant is the ratio of net power output to the heat input:

\[ \eta_{gt} = \frac{\Delta H_{gt}}{Q_{in}} \]  

3.1.6. Corrected theoretical thermal efficiency. For the case, when the volume flow rate at inlet to the compressor is held constant the pressure loss in each of the heat exchangers vary widely with the working gases. The method of correcting the theoretical thermal efficiency taking into account the pressure loss variation is described below. The corrected theoretical thermal efficiency is then used for comparing the performance of various working gases.
Generally, the pressure loss of a gas through the shell and tube type of heat exchanger [9] is given by:

\[
\Delta P = 0.0094 \left( \frac{0.2}{\gamma} \right) (\frac{L D - 1.2}{n/4 D^2 z})^{1.8} \left( \frac{w}{\mu_0 D^2 z} \right) \tag{15}
\]

This equation can be simplified and written as:

\[
\Delta P = 0.015 \left( \frac{0.2}{\gamma} \right) (\frac{L D - 4.8}{n/4 D^2 z})^{1.8} \left( \frac{w}{\mu_0 D^2 z} \right) \tag{16}
\]

By definition the pressure loss coefficient with air as working medium flowing through the heat exchanger is:

\[
A P_a^2 = \frac{\Delta P_a}{P_a} \tag{17}
\]

The pressure loss in the ducts which connects the heat exchangers to compressor or turbine is to be added to that of heat exchanger at the appropriate station.

The pressure loss coefficient for any other gas flowing through the same heat exchanger can be obtained by applying correction to the pressure loss coefficient of air:

\[
A P_g = A P_a \frac{(\Delta P_g/\Delta P_a)(P_a/P_g)}{\gamma} \tag{18}
\]

By substituting the pressure loss equation (16) in equation (18) and rearranging, we obtain:

\[
A P_g = A P_a \frac{\mu_0^{0.2}}{\mu_a} \left( \frac{M_a}{M_g} \right)^{0.8} \tag{19}
\]

Where viscosity of gases is to be corrected for temperature using equation:

\[
\mu_g = \mu_0 \left( \frac{T + C_s}{T_0} \right)^{3/2} \tag{20}
\]

As seen from equation (19) the pressure loss coefficient and hence power loss due to flow resistance varies with molecular weight of gas and viscosity. Hence the thermal efficiency is to be corrected taking the variation of pressure loss coefficient and repeating the steps from equation (6) to (14).

### 3.2. Performance of CCGT in combined cycle configuration

In the combined cycle plant shown schematically in Fig. 3 the sensible heat in the exhaust of GT is recorded in a WHB where steam is generated to produce additional power in a bottoming steam cycle. In order to recover greater amount of heat from the exhaust of GT, a dual pressure WHB is considered [10] for achieving higher overall cycle efficiency.

The performance calculation of CCGT in combined cycle mode is similar to the method described in section 3.1 except that the gases leaving the compressor enters directly the gas heater as there is no recuperator. After the performance computation of topping GT is carried out an energy balance between GT exhaust gases and water/steam flowing through the WHB is performed for determining the bottoming steam plant efficiency.

#### 3.2.1. The bottoming steam cycle

A general objective is to cool the exhaust gas of GT to the lowest practical temperature while passing through the WHB. The gas and water/steam temperature profiles in a dual pressure WHB shown in Fig. 4 indicates a closer approach of the gas cooling curve and water/steam heating curve at two points which usually occurs as the gas leaves the HP and LP evaporator sections. Knowing the enthalpy and steam flow rates through HP and LP sections of the steam turbine, the bottoming steam plant output is computed. Therefore, the overall efficiency of the steam bottoming plant is evaluated which is the product of steam Rankine cycle efficiency and heat recovery effectiveness, where the heat recovery effectiveness of WHB is defined as the ratio of the exhaust heat recovered to the total sensible heat of the exhaust gases above ambient condition.

#### 3.2.2. The combined cycle efficiency

The overall thermal efficiency of CCGT in combined cycle configurations is:
Hence variables considered for optimization are compressor pressure ratio and TET, compressor effectiveness, and pressure losses in the cycle mode is obtained by search method.

4. OPTIMIZATION

The univariate search methods viz. Dichotomous, Fibonacci, Bolzano etc. are commonly used to obtain the optimum value of a polynomial function. In the study cycle thermal efficiency is fitted as a polynomial of compressor ratio of CCGT using least square method and Fibonacci's search scheme [11] is adopted as it is found to be fast convergent and requires less computer time compared to the other methods.

In this paper the optimization has been based on maximisation of cycle thermal efficiency. In regenerative configuration of CCGT cycle parameter such as efficiency of compressor, gas heater, turbine, recuperator effectiveness, and pressure losses in heat exchangers are treated as constants. Hence variables considered for optimization are compressor pressure ratio and TET. Considering TET as parameter, the optimum \( \eta_{cc} \) is then arrived by adopting Fibonacci's search technique.

In the optimization of combined cycle configuration of CCGT, besides the component efficiencies and pressure losses of GT, the cycle parameters of bottoming steam plant viz., HP and LP evaporator pinch/approach points, superheater approach point, condenser pressure are assumed as constant. This leaves the cycle variables HP and LP steam generation pressures in WHB, compressor pressure ratio, to be optimized for maximisation of thermal efficiency of cycle for given TET.

Initially the effect of varying LP boiler pressure of a dual pressure WHB on combined cycle efficiency has been tested [2] for fixed HP boiler pressure, compressor pressure ratio and TET with different working gases viz., Helium and Carbon Dioxide. From the above it is observed that the variation in combined cycle efficiency due to changes in LP generation pressure is not appreciable, thus leaving behind the parameters HP boiler pressure and compressor pressure ratio for optimization for given TET.

The optimization is then carried out in two steps. In the first step the optimum HP steam generation pressure of WHB is determined to achieve maximum cycle efficiency for given compressor pressure ratio and fixed TET. The process is then repeated for different compressor pressure ratios at the same TET. In the second step the optimum compressor pressure ratio of CCGT operating in combined cycle mode is obtained by search method.

5. RESULTS AND DISCUSSIONS

The design data used for evaluating the thermodynamic performance of CCGT is given in Table 2. The Fig. 5 shows the comparison of thermal efficiency of CCGT in regenerative configuration computed by the present method with that of Shivakura, T., et al [1], which assumes constant specific heat for the working gas during compression and expansion processes. For the TET of 1073 K and assuming same pressure loss coefficients, the maximum cycle efficiency reported in Ref.[1] differs by 8 to 10% from the present method. And the optimum compressor pressure ratios using Helium, Air and Carbon Dioxide from Ref. [1] are 2.53, 5.0 and 6.3, respectively, while the corresponding values by the present method are 2.6, 3.75 and 6.3 at the same TET.

The thermal efficiency of CCGT in regenerative configuration is presented for two different cases. In the first case, the thermal efficiency is computed assuming the constant pressure loss coefficient for each of the heat exchangers with different working gases. While in the second case the volume flow rate at inlet is constant so that the pressure loss coefficient accordingly varies with the working gases for the same heat exchangers and hence the thermal efficiency is corrected. The cycle efficiency for the above cases is shown against compressor pressure ratio in Fig. 6 and 7 for recuperator effectiveness of 0.8 and 0.9 respectively with TET as parameter varying from 873 K to 1073 K. After taking into account the correction for thermal efficiency due to variation of pressure loss coefficient, it is observed that with Helium the efficiency increases by about 15 percent at TETs of 873 K, 973 K and 1073 K respectively. Whereas with Carbon Dioxide the same decreases by about 1.5
percent, 0.75 percent and 0.45 percent for the above TET's.

Table 2 Design data for performance estimation of CCGT

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor inlet temperature K</td>
<td>313</td>
</tr>
<tr>
<td>Polytropic efficiency of compressor</td>
<td>0.88</td>
</tr>
<tr>
<td>Isentropic efficiency of turbine</td>
<td>0.90</td>
</tr>
<tr>
<td>Efficiency of AFB Combustor</td>
<td>0.80</td>
</tr>
<tr>
<td>Mechanical Efficiency of Transmission</td>
<td>0.99</td>
</tr>
<tr>
<td>Max. pressure in the circuits MPA</td>
<td>3.039</td>
</tr>
<tr>
<td>Tube side pressure loss coefficient of recuperator</td>
<td>0.025</td>
</tr>
<tr>
<td>Shell side pressure loss coefficient of recuperator</td>
<td>0.025</td>
</tr>
<tr>
<td>Pressure loss coefficient of precooler</td>
<td>0.025</td>
</tr>
<tr>
<td>Pressure loss coefficient of gas heater</td>
<td>0.03</td>
</tr>
<tr>
<td>Duct losses from compressor exit to gas heater inlet</td>
<td>0.01</td>
</tr>
<tr>
<td>Duct losses from gas heater exit to turbine inlet</td>
<td>0.01</td>
</tr>
<tr>
<td>Recuperator effectiveness</td>
<td>0.75 to 0.9</td>
</tr>
<tr>
<td>Percentage of bleed for bearing/structural cooling</td>
<td>1</td>
</tr>
<tr>
<td>The approach temperature difference at superheater exit °C</td>
<td>50</td>
</tr>
<tr>
<td>The pinch point as the gas leaves the HP and LP evaporator sections in WHB °C</td>
<td>25</td>
</tr>
<tr>
<td>The approach point as the gas enters HP and LP evaporator sections in WHB °C</td>
<td>20</td>
</tr>
<tr>
<td>The Deaerator pressure MPA</td>
<td>0.2026</td>
</tr>
<tr>
<td>The condenser pressure MPA</td>
<td>0.01013</td>
</tr>
<tr>
<td>Isentropic efficiency of steam turbine</td>
<td>0.80</td>
</tr>
<tr>
<td>Percentage pressure losses in WHB</td>
<td>5</td>
</tr>
<tr>
<td>Lower calorific value of coal KJ/Kg</td>
<td>17640</td>
</tr>
<tr>
<td>Heat losses from the gas heater by radiation etc.</td>
<td>-</td>
</tr>
<tr>
<td>Turbine entry temperature K</td>
<td>873-1173</td>
</tr>
<tr>
<td>Compressor pressure ratio</td>
<td>2-12</td>
</tr>
</tbody>
</table>

The optimum $r_c$ of CCGT in regenerative cycle increases as TET increases from 873 K to 1073 K as follows (a) with air as working medium the optimum $r_c$ increases from 4.0 to 5.0 for $\eta_{rec}$ of 0.8 and from 2.5 to 4.0 for $\eta_{rec}$ of 0.9 (b) with Helium, the optimum $r_c$ increases from 2.6 to 3.25 for $\eta_{rec}$ of 0.8 and from 2.4 to 2.75 for $\eta_{rec}$ of 0.9 (c) with Carbon Dioxide the optimum $r_c$ increases from 8.0 to 12.5 for $\eta_{rec}$ of 0.8 and from 5.5 to 8.5 for $\eta_{rec}$ of 0.9. Also the optimum $r_c$ and the corresponding maximum corrected thermal efficiency are shown in fig. 8 for varying recuperator effectiveness from 0.75 to 0.9 at a selected TET of 993 K.

For thermodynamic optimization of CCGT in combined cycle configuration, the cycle variables considered are HP and LP generation pressures of dual pressure WHB, the compressor pressure ratio and TET. From the study made it is observed that the effect of varying LP generation pressure on combined cycle efficiency is not significant. As shown in fig. 9, combined cycle efficiency varies by less than 1 percent as LP generation pressure is varied from 0.2026 MPA to 1.013 MPA at a selected HP generation pressure of 4.66 MPA and TET of 993 K. In general Helium, Air and Carbon Dioxide have shown optimum LP generation pressure at 0.4052 MPA, 0.6078 MPA and 0.8104 MPA respectively.

Hence the cycle variables for optimization reduces to HP generation pressure of WHB, compressor pressure ratio and TET. Initially for a fixed compressor pressure ratio and given TET, HP generation pressure is varied from 2.03 MPA to 12.66 MPA and the optimum value is arrived at by adopting search technique. The procedure is then repeated for different compressor pressure ratios and the maximum combined cycle efficiency is plotted against $r_c$ as shown in fig. 10 for range of TET's from 993 K to 1173 K. The optimum $r_c$ of Helium increases from 2.75 to 3.5, and the optimum $r_c$ of air increases from 4.5 to 6.0, while that of Carbon Dioxide increases from 7.5 to 12.0 as TET increases from 993 K to 1173 K. The optimum HP generation pressures corresponding to the best thermal efficiency of the cycle with Air, Helium and Carbon Dioxide are 4.71 MPA, 4.66 MPA and 11.4 MPA respectively at the
TET of 993 K. The optimum HP generation pressures for the same working media are 4.92 MPa, 4.74 MPa and 11.14 MPa at the TET of 1073 K respectively.

In regenerative configuration, the maximum cycle efficiency of CCGT with Air, Helium and Carbon Dioxide at a TET of 1073 K is 32.4%, 29.1% and 32.1% respectively and is shown in Fig. 7. The corresponding values for combined cycle configuration are 32.5%, 31.75% and 33.5% as given in Fig. 10. From this it can be inferred that at the same TET, the CCGT in combined cycle configuration has only about 2% to 8% higher cycle efficiency than regenerative configuration. This can be explained from the fact that, inspite of greater recovery of heat from the exhaust of CCGT in combined cycle mode than regenerative mode, substantial increase in thermal efficiency may not be possible from the former since the isentropic efficiency of steam turbine is less than that of gas turbine for converting the available heat to power. Further increase in thermal efficiency of combined cycle plant can be achieved by increasing the heat recovery effectiveness of WHB through the use of three pressure system which may not be economical. However, the technoeconomic studies conducted by a number of investigators [12,13] have revealed that coal fired CCGT in neither regenerative nor combined cycle modes are competitive to the other coal based advanced power cycles using open cycle GT as topping plant. But AFB coal fired CCGT adopting cogeneration system is shown [14] to be economical for industries where there is simultaneous requirement of heat and power. Further, the application of fluidized bed coal fired CCGT cogeneration systems depend on the heat and power loads of an industry, and the case to case study of the economics which is to be evaluated and compared with that of alternative means of meeting the demand.

7. CONCLUSION

a) An accurate method of estimation of thermodynamic performance of CCGT in regenerative and combined cycle configurations has been presented. The optimum compressor pressure ratio of CCGT for varying recuperator effectiveness and L/V pressure ratio is given. Also the optimum HP generation pressure and compressor pressure ratio of CCGT in combined cycle mode is reported for selected TETs.

b) Helium is found to be the most suitable working gas for CCGT as it can develop high cycle efficiency at a very low compressor pressure ratio. Air ranks next to Helium achieving nearly same thermal efficiency although at relatively higher pressure ratio.

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REFERENCES


5 Rao, J.S., Bindra, G.S., Gas tur-
bines for power generation using combined
cycles, Proceedings of Fourth Seminar on
Gas Turbines, Gas Turbine Research Establish-
ment, Bangalore, India, November, 1979.

6 Rao, J.S., Optimization studies of
combined cycle gas turbine with integrated
coal gasifier, Sixth National Conference on
IC Engines and Combustion, IIT, Bombay, India,

7 Rivikin, S.L., Kremanevskaya, E.A.,
Equations of State Water and Steam for Compu-
ter Calculations for Process and Equipment
at Power Stations, Thermal Engineering,
No. 24(3) 69-73, 1977, pp.50-54.

8 Sawyer, J.W., Sawyers Gas turbine
Engineering Handbook, 2nd ed. vol.2, Gas
turbine publications, Stamford, Conn. 1976.

9 Cardwell, F.D., 'Optimum tube size
for shell and tube type heat exchangers,

10 Foster Pegg, R.W., Steam Bottoming
Plants for Combined Cycles, Journal of Engi-
neering for Power, ASME, vol.100, April,
1978.

11 Marlin, H. Mickle, Szn, T.W., Opti-
mization in systems engineering, Intext

12 , Energy Conversion Alterna-
tive Study,Vol. VI, Closed Cycle Gas Tur-

13 Corman, J.C., Fox, G.R., Performance
and Economics of Advanced Energy Conversion
systems for Coal and Coal derived fuels, ASME
Journal of Engineering for Power, vol.100,
April 1978, pp.252-259.

14 Foster-Pegg, R.W., 'A Coal Fired
Fluid Bed Gas Turbine Cogeneration System,
Combustion, June 1979, pp.29-35.