

EFFECT OF AMBIENT TEMPERATURE, GAS TURBINE INLET TEMPERATURE AND COMPRESSOR PRESSURE RATIO ON PERFORMANCE OF COMBINED CYCLE POWER PLANT

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ABSTRACT

The performance of combined cycle power plant is strongly influenced by the operating parameters of the plant viz inlet air temperature to compressor, compressor pressure ratio, gas turbine inlet temperature, gas turbine exhaust temperature and stack temperature. This paper covers the parametric analysis of the effect of inlet air temperature to compressor, gas turbine inlet temperature and compressor pressure ratio on the performance of the combined cycle power plant by keeping the superheated steam temperature and pressure and pinch point temperature difference in HRSG constant. The results reveal that the combined cycle power plant gives best performance at low ambient air temperature, high compressor pressure ratio and high gas turbine inlet temperature.

Keywords: Gas Turbine, Steam Turbine, Combined Cycle, Ambient Air Temperature, Turbine Inlet Temperature Etc.

I. INTRODUCTION

Combined cycle plant is competitive alternative for large base load generating stations. This combined cycle recovers much of exhaust energy of gas turbine by passing high temperature exhaust gases to heat recovery steam generator to generate steam which can be further utilized to drive a steam turbine. Increased power and higher thermal efficiency obtained from this reduces the cost of additional equipments and lowers the generating cost if the number of operating hours per year substantially increased. The additional advantage of combined cycle is reduced emission of heat to the atmosphere and reduced requirement of cooling water. The design of heat recovery steam generator is always critical in the combined cycle because if more heat is recovered from

the gas turbine exhaust, the more expensive will be the heat recovery boiler. The pressure of the steam generated in the heat recovery steam generator is the most important single factor in determining the amount of steam produced in conjunction with the pinch point. In dual pressure steam cycle power plant higher efficiency can be obtained by increasing the average temperature at which heat is transferred to steam.

II. THERMODYNAMIC MODELING

The schematic diagram of dual pressure combined cycle under consideration is shown in figure.1. In this arrangement a simple gas turbine cycle is used as topping cycle and a steam cycle is used as bottoming cycle. The waste heat of gas turbine exhaust which is at high temperature is recovered in a dual pressure heat recovery steam generator.

Air from atmosphere is compressed in a compressor to a higher pressure. The fuel is injected into combustor and is burnt with compressed air at constant pressure. The hot combustion products from the combustor are allowed to expand in a gas turbine to produce useful work. The heat rejected by gas turbine is highly appreciable as the exhaust temperature is very high. This waste heat is utilized in a dual pressure heat recovery steam generator to produce steam which is expanded in steam turbine working on rankine cycle to produce additional power.

The steam expanded up to condenser pressure in LP steam turbine is condensed to saturated water state in condenser and pumped to deaerator through condensate extraction pump. In deaerator dissolved gases are removed by using the steam bled from low pressure turbine resulting in saturated feed water at deaerator pressure. The hot saturated water from deaerator is pumped to the economizer section of HRSG through LP and HP boiler feed pumps respectively and is heated to the saturation temperature corresponding to their pressures. The saturated water is converted into steam in the LP and HP evaporators and further the steam is superheated in the corresponding superheaters. The superheated steam from HP superheater is expanded in a high pressure steam turbine and is mixed with superheated steam supplied from LP superheater. The enthalpy of mixed steam is obtained by applying the adiabatic mixing process. The mixed steam is then expanded in low pressure steam turbine and appropriate part of steam is bled for deaeration process at required pressure.

The temperature – entropy diagram for the combined cycle is shown in figure 2.

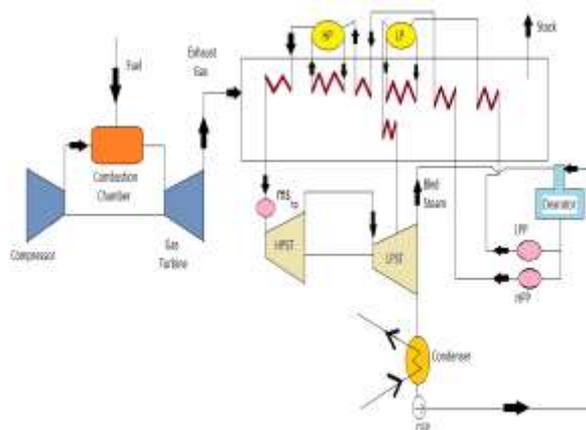


Fig.1: Schematic Diagram of Combined Cycle Power Plant

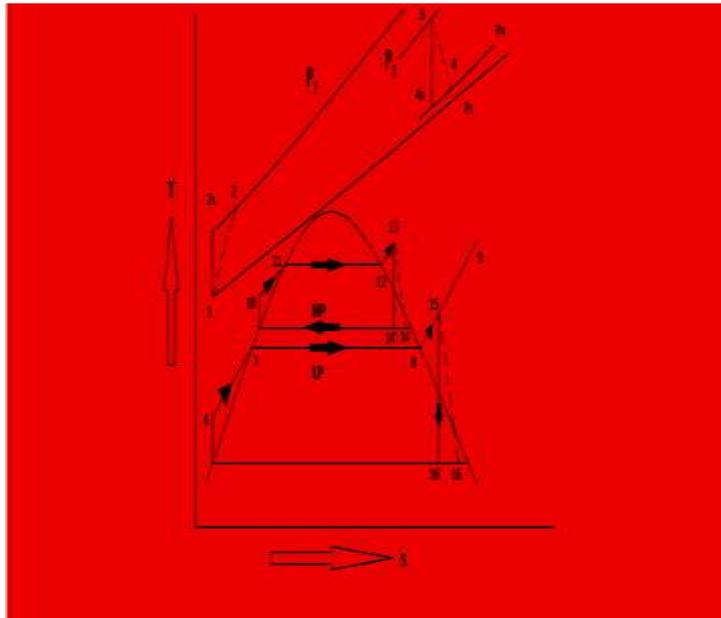


Fig.2: Temperature –Entropy Diagram of Combined Cycle Power Plant

III. GAS TURBINE CYCLE MODEL

It is assumed that the compressor efficiency and the turbine efficiency are represented by η_c and η_t respectively. The ideal and actual processes are represented by full and dashed lines on the temperature entropy diagram as shown in figure 2.

Air compressor model:

The compression ratio R_p of the compressor can be defined as

$$r_p = \frac{P_2}{P_1} \quad \text{-----(1)}$$

Where P_1 and P_2 are compressor inlet and outlet pressures respectively

The isentropic efficiency of compressor is expressed as

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1} \quad \text{-----(2)}$$

Where

T_1 = compressor inlet temperature

T_{2s} = compressor isentropic outlet temperature

T_2 = compressor actual discharge temperature

The actual discharge temperature of the compressor is calculated from Eq. 3

$$T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_c} \quad \text{----- (3)}$$

The work of the compressor (W_c) can be calculated as

$$W_c = \dot{m}_a \times c_{pa} \times (T_2 - T_1) \quad \text{----- (4)}$$

Where C_{pa} is the specific heat of air which can be fitted by Eq. 5 for the range of $200K < T < 800K$.

$$C_{pa} = 1.0189 \times 10^3 - 0.13784T_a + 1.9843 \times 10^{-4}T_a^2 + 4.2399 \times 10^{-7}T_a^3 - 3.7632 \times 10^{-10}T_a^4 \dots\dots\dots (5)$$

Combustion Chamber Model: From the energy balance in the combustion chamber

$$\dot{m}_a C_{pa} T_2 + \dot{m}_f \times LHV + \dot{m}_f C_{pf} T_f = (\dot{m}_a + \dot{m}_f) C_{pg} \times TIT \dots\dots\dots (6)$$

Where \dot{m}_f is the fuel mass flow rate (kg/s), \dot{m}_a is the air mass flow rate(kg/s), LHV is the lower heat value , TIT = T_3 is turbine inlet temperature, c_{pf} is the specific heat of fuel and T_f is the temperature of the fuel, and c_{pg} is the specific heat of flue gases.

The specific heat of flue gases c_{pg} is given by Eq. 7

$$C_{pg} = 1.8083 - 2.3127 \times 10^{-3} T + 4.045 \times 10^{-6} T^2 - 1.7363 \times 10^{-9} T^3 \dots\dots\dots (7)$$

The fuel air ratio (f) is expressed as Eq. 8

$$f = \frac{\dot{m}_f}{\dot{m}_a} = \frac{C_{pg} \times TIT - C_{pa} \times T_2}{LHV - C_{pg} \times TIT} \dots\dots\dots (8)$$

Gas Turbine Model: the isentropic efficiency of gas turbine is given by Eq. (9)

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_{4s}} \dots\dots\dots(9)$$

Where

T_3 = Turbine inlet temperature

T_{4s} = isentropic discharge temperature of gas turbine

T_4 = actual discharge temperature of gas turbine

The actual discharge temperature of gas turbine can be given by Eq.(10)

$$T_4 = T_3 - \eta_t (T_3 - T_{4s}) \dots\dots\dots (10)$$

The shaft work of the gas turbine is given by Eq. (11)

$$W_{GT} = \dot{m}_g \times C_{pg} (T_3 - T_4) \dots\dots\dots (11)$$

The net work of the gas turbine is determined by Eq. (12)

$$(W_{GT})_{net} = W_{GT} - W_C \dots\dots\dots (12)$$

The specific fuel consumption is determined by Eq (13)

$$sfc = \frac{3600 \times \dot{m}_f}{(W_{net})} \dots\dots\dots(13)$$

The heat supplied is given by Eq (14)

$$Q_{add} = \dot{m}_f \times LHV \dots\dots\dots (14)$$

The thermal efficiency of gas turbine is determined by Eq (15)

$$(\eta_{th})_{GT} = \frac{(W_{GT})_{net}}{\dot{m}_f \times LHV} \dots\dots\dots (15)$$

Steam Turbine Cycle Model: It is assumed that the steam turbine efficiency and pump efficiency are represented by η_{st} and η_p respectively. The ideal and actual processes on temperature – Entropy diagram are represented by full and dashed lines as shown in figure.

Heat Recovery Steam Generator Model: A dual pressure HRSG is considered here for combined cycle gas turbine plant. By applying energy balance for gas and water in each part of the HRSG the gas temperature and water properties are calculated by solving the following equations,

Heat available with the exhaust gases from gas turbine is given as

$$Q_{av} = \dot{m}_g \times C_{pg} \times (T_4 - T_5) \times h_{f1} \quad \dots\dots (16)$$

Where T_5 is the exhaust temperature of the flue gases from HRSG, and h_{f1} is the heat loss factor whose value ranges from 0.98 to 0.99.

The HP superheater duty is expressed as

$$(Q_{sh})_{HP} = m_{S_{HP}} (h_{13} - h_{12}) = \dot{m}_g \times c_{pg} \times (T_4 - T_g) \quad \dots\dots(17)$$

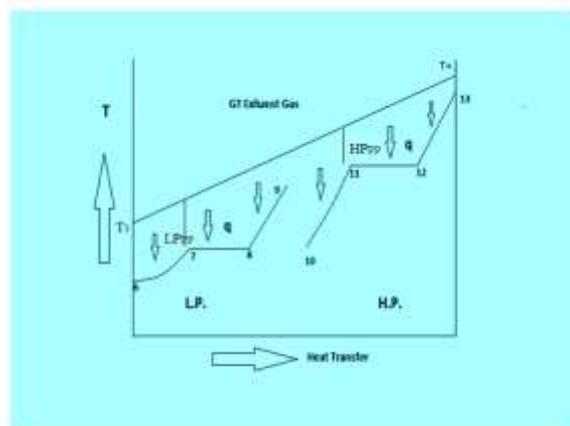


Fig.3: Temperatures – Heat Transfer Diagram of HRSG

The thermal analysis of HRSG depends on the designed pinch point $(\Delta T)_{PP}$ and approach point $(\Delta T)_{AP}$. The temperature of the gas leaving the HP evaporator is given by Eq. 18

$$T_{g1} = T_{11} + (\Delta T_{PP})_{HP} \quad \dots\dots\dots(18)$$

Where T_{11} is the saturation steam temperature corresponding to HP superheater pressure and $(\Delta T_{PP})_{HP}$ is the pinch point temperature difference in the HP side of HRSG.

The LP superheater duty is expressed as

$$(Q_{sh})_{LP} = m_{S_{LP}} (h_9 - h_8) = \dot{m}_g \times c_{pg} \times (T_{g1} - T_{g2}) \quad \dots\dots\dots (19)$$

The temperature of the hot exhaust gases leaving the HRSG is given as

$$m_{S_{LP}} (h_9 - h_6) + m_{S_{HP}} (h_{11} - h_{10}) = \dot{m}_g \times c_{pg} \times (T_{g1} - T_5) \quad \dots\dots\dots (20)$$

Steam turbine model: The steam at high pressure and high temperature obtained from HP superheater is expand in the HP steam turbine and then mixed with LP steam obtained from LP super heater and expand up to the condenser pressure in LP steam turbine. The energy balance gives

$$W_{HPST} = m_{SHP} (h_{13} - h_{14}) \dots\dots\dots (21)$$

And

$$W_{LPST} = m_{SHP} + m_{SLP} (h_{15} - h_{16}) \dots\dots\dots(22)$$

Condenser model: The heat rejected from the condenser is expressed as

$$Q_{rej} = (m_{SHP} + m_{SLP} - m_{bled}) (h_{16} - h_{f17}) \dots\dots\dots (23)$$

Pump model: The condensate from the condenser is extracted by the HP and LP pump and raised to LP and HP economizer pressure. The corresponding work is given by

$$W_{hpp} = m_w \times v_{17} (p_{10} - p_c) \dots\dots\dots (24)$$

$$W_{lpp} = m_w \times v_{17} (p_6 - p_c) \dots\dots\dots(25)$$

Therefore the net work of ST power plant is

$$(W_{net})_{ST} = W_{ST} - W_P \dots\dots\dots(26)$$

The efficiency of steam turbine power plant is

$$\eta_{st} = \frac{(W_{net})_{st}}{Q_{av}} \dots\dots\dots(27)$$

The overall thermal efficiency of the combined cycle gas turbine plant is given by

$$\eta_{cc} = \frac{(W_{net})_{GT} + (W_{net})_{ST}}{m_f \times LHV} \dots\dots\dots(28)$$

IV. RESULTS AND DISCUSSION

The power output, efficiency and mass flow rate for gas turbine, steam turbine and combined cycle were calculated for different ambient temperature, gas turbine inlet temperature and compressor pressure ratio and results of analysis are presented in figures given below.

Figure 4 shows the effect of TIT on steam generation rate in dual pressure HRSG. It is clear from the figure that with increasing TIT the HP steam generation rate increases and the LP steam generation rate decreases. The increase in HP steam generation rate with increasing TIT is greater than the decrease in LP steam generation rate and therefore total steam generation rate increase with increasing TIT. The reason behind this is that for a fixed maximum temperature and pressure of HP and LP steam and pinch point the heat recovery in HP section is greater than that in LP section for fixed value of HP pinch point temperature. For a increase in TIT from 1273K to 1373 K the HP steam generation per kg of gas flow increases from 0.112 to 0.145(126.35kg/s to 163.99kg/s) and the LP steam generation per kg of gas flow decreases from 0.034 to 0.025(38.97kg/s to 28.84kg/s). Similarly for an increase of TIT from 1373 K to 1473 K the HP steam generation per kg of gas flow increases from 0.145 to 0.178(163.99kg/s to 201.85kg/s) and the LP steam generation per kg of gas flow decreases from 0.025 to 0.016(28.84kg/s to 19.16kg/s).The total steam generation rate increases with increasing TIT.

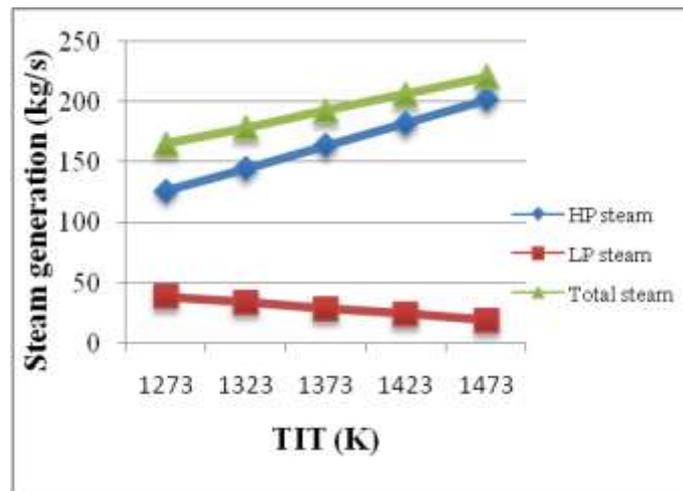


Fig.4: Effect of TIT on Steam Generation Rate at $R_p = 12.2$

Figure 5 shows the effect of TIT on power output of gas turbine, steam turbine, and combined cycle. It is clear from the figure that the net power output of topping cycle i.e gas turbine cycle, bottoming cycle i.e steam turbine cycle as well as of combined cycle increases with increases in TIT. The reason behind this is that the gas turbine work increases with TIT because of increasing temperature difference of TIT and exhaust gas turbine temperature whereas the compressor work remain unaffected therefore net power output of gas turbine increases with TIT. The steam turbine output increases with TIT because of increasing mass flow rate of total steam in HRSG due to increasing exhaust temperature of gas turbine with increasing TIT. With High exhaust temperature of gas turbine more heat is available at the gas turbine exhaust to generate steam in HRSG. The net total output of combined cycle increases with increasing TIT due to improvement in power output of both topping and bottoming cycles. It is also observed that the net power output of topping cycle is higher than that of bottoming cycle at each value of TIT. The net power output of topping cycle increases by 42.63% for an increase of TIT from 1273 K to 1473 K. Similarly the net power output of bottoming cycle increases by 46.37% for the same increase in TIT from 1273 K to 1473 K. The net power output of combined cycle increases by 44.05% for an increase of TIT from 1273 K to 1473 K.

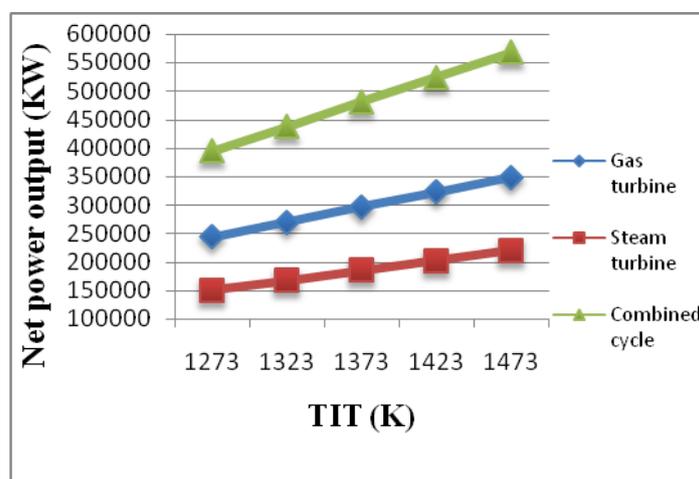


Fig.5: Effect of TIT on Net Power Output at $R_p = 12.2$

Figure 6 shows the effect of TIT on gas cycle efficiency, steam cycle efficiency and combined cycle efficiency. It is observed that the efficiency of each of the topping cycle i.e. gas turbine cycle, bottoming cycle i.e. steam turbine cycle and combined cycle increases with increasing TIT. This is due to the reason that the power output of each of the topping cycle, bottoming cycle and combined cycle increases with increasing TIT. The heat addition in each of the topping cycle, bottoming cycle and combined cycle also increases with increasing TIT but the increase in power output more than increase in heat addition therefore result

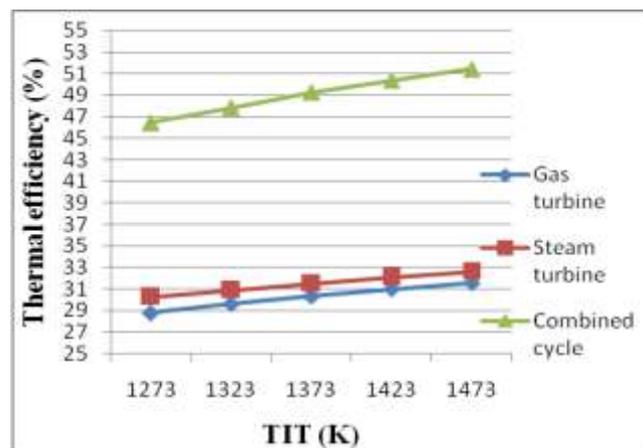


Fig.6; Effect of TIT on Cycle Efficiency at $R_p = 12.2$

in increase in efficiency of each cycle. The topping cycle efficiency increases by 5.45% for an increase in TIT from 1273 K to 1373 K and increases by 3.95% for an increase of TIT from 1373 K to 1473 K. The improvement in topping cycle efficiency is slower in higher range of TIT of 1373 K to 1473 K. The efficiency of bottoming cycle increase by 4% for an increase of TIT from 1273 K to 1373 K and it increases by 3.49% for an increase of TIT from 1373 K to 1473 K. This is due to increasing exhaust gas temperature and enthalpy with increasing TIT resulting into increased HP steam generation rate as conclude by the figure. The efficiency of combined cycle increase by 6.04% for an increase of TIT from 1273 K to 1373 K and increases by 4.4% for an increase of TIT from 1373 to 1473 K. This is due to the improvement in both topping and bottoming cycle efficiency with increasing TIT

Figure 7 shows the effect of pressure ratio on steam generation rate in dual pressure HRSG. It is clear from the figure that with increasing pressure ratio the HP steam generation rate decreases and the LP steam generation rate increases. The decrease in HP steam generation rate with increasing pressure ratio is greater than the increases in LP steam generation rate therefore total steam generation rate decreases with increasing pressure ratio. The reason behind this is that with increasing pressure ratio the temperature at the exhaust of gas turbine decreases which results in reduction in heat recovery in the HP section of the HRSG and increase in heat recovery in LP side of HRSG for a fixed maximum temperature of HP and LP steam and pinch point. The net result is reduction in total heat recovery in HRSG. For an increase in pressure ratio from 8 to 12 the HP steam generation per kg of gas flow decreases from 0.182 to 0.145 (206.2kg/s to 164kg/s) and the LP steam generation per kg of gas flow increases from 0.015 to 0.025 (17.2 kg/s to 28.8 kg/s). Similarly for an increases in pressure ratio from 12 to 16 the HP steam generation per kg of gas flow decreases from 0.145 to 0.122 (164 kg/s to 137.8 kg/s) and the LP steam generation per kg of gas flow increases from 0.025 to 0.031(28.8 kg/s to 35.5 kg/s). The total steam generation rate decreases with increasing pressure ratio.

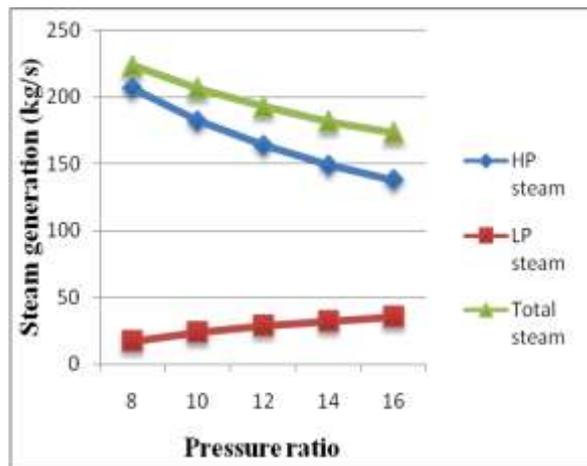


Fig.7: Effect of Pressure Ratio on Steam Generation in HRSG

Figure 8 shows the effect of pressure ratio on the net power output of gas turbine, steam turbine and combined cycle. It is clear from the figure that the net power output of topping cycle i.e. gas turbine cycle first increases with pressure ratio up to a certain value and then decreases with increasing pressure ratio, the net power output of bottoming cycle i.e. steam turbine continuously decreases with increasing pressure ratio and the net power output of the combined cycle continuously decreases with increasing pressure ratio. The reason behind this is that the for topping cycle both turbine work and compressor work increases with increasing pressure ratio but at higher pressure the increase in compressor work is more than the increase in turbine work therefore net power output of gas turbine start decreasing in the higher range of pressure ratio. Similarly the net output of steam turbine decreases with increasing pressure ratio due to reduction in total steam generation rate with increasing pressure ratio as concluded in the figure 5.27. The net power output of the combined cycle also decreases with increasing pressure ratio because of reducing output of both topping cycle and bottoming cycle. The net power output of gas turbine increases by 1.06% for an increase in pressure ratio from 8 to 12 and decreases by 0.31% for an increase in pressure ratio from 8 to 16.

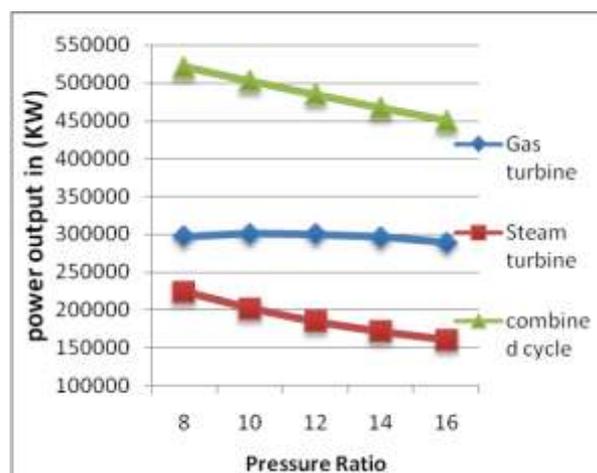


Fig.8: Effect of Pressure Ratio on Power Output at TIT 1100°C

Figure 9 shows the effect of pressure ratio on thermal efficiency of gas turbine cycle, steam cycle, and combined cycle. It is clear from the figure that the thermal efficiency of gas turbine cycle continuously increases

with increasing pressure ratio and of steam cycle continuously decreases with increasing pressure ratio and of combined cycle first increases with pressure ratio up to a certain value and then starts decreasing with increasing pressure ratio. The reason behind this is that the mass of fuel added in the combustor of topping cycle decreases and net power output of gas turbine increases with increasing pressure ratio. The steam cycle efficiency decreases due to reducing mass flow rate of total steam in the HRSG as explained in figure 5.27. Due to the combined effect of both cycles the combined cycle efficiency first increases and then decreases with increasing pressure ratio. For an increase in pressure ratio from 8 to 12 the gas cycle efficiency increases by 8.7% and it increases by 12.9% for an increase in pressure ratio from 8 to 16. Similarly for an increase in pressure ratio from 8 to 12 the steam cycle efficiency decreases by 4.3% and it decreases by 2.27% for an increase in pressure ratio from 8 to 16. The combined cycle efficiency increases by 1.1% for an increase in pressure ratio from 8 to 12 and it increases by 0.32% for an increase in pressure ratio from 8 to 16. The improvement is more in lower range of pressure ratio.

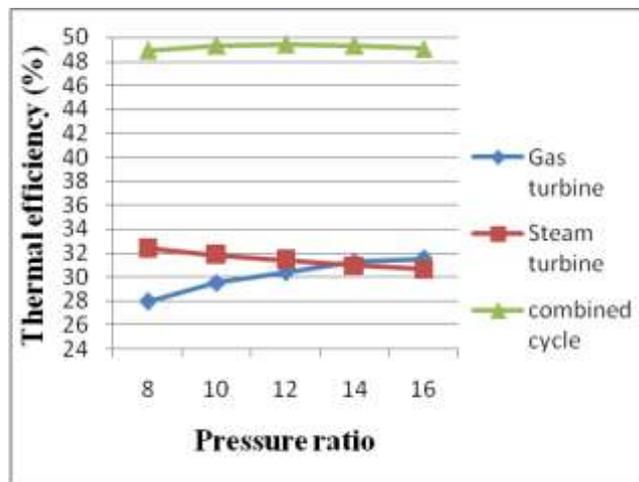


Fig.9: Effect of Pressure Ratio on Cycle Efficiency at TIT 1100°C

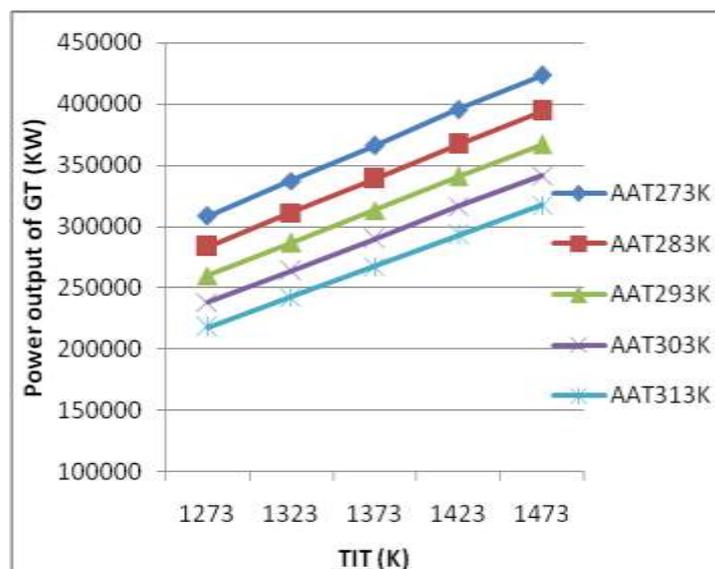


Fig.10: Effect of AAT and TIT on Power Output of GT

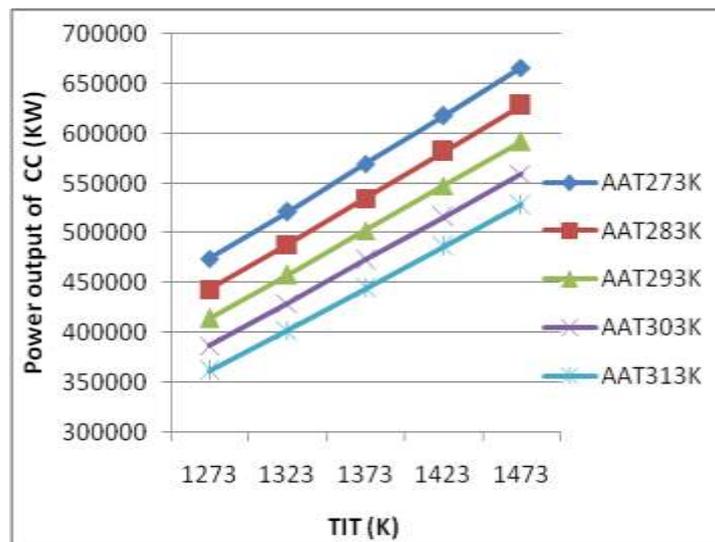


Fig.11: Effect of AAT and TIT on Power Output of CC

Figure 10 and 11 shows the effect of inlet air temperature to compressor and turbine inlet temperature on the power output of gas turbine cycle and combined cycle. For a given turbine inlet temperature the power output of gas turbine cycle decreases with increasing ambient air temperature. The reason behind this is that with increasing ambient air temperature the mass flow rate of air through the gas turbine decreases due to decreased density of air which results in reduction in net power output of gas turbine. The improvement in power output is more at low ambient air temperature and high turbine inlet temperature.

The similar trend is seen for the power output of combined cycle. The reason behind this is that with increasing ambient air temperature the energy available at the exhaust of gas turbine decreases due to reduced mass flow rate of air through the gas turbine which results in reduction in steam generation rate in HRSG and hence reduced power output of bottoming cycle. The overall effect is to decrease the power output of combined cycle.

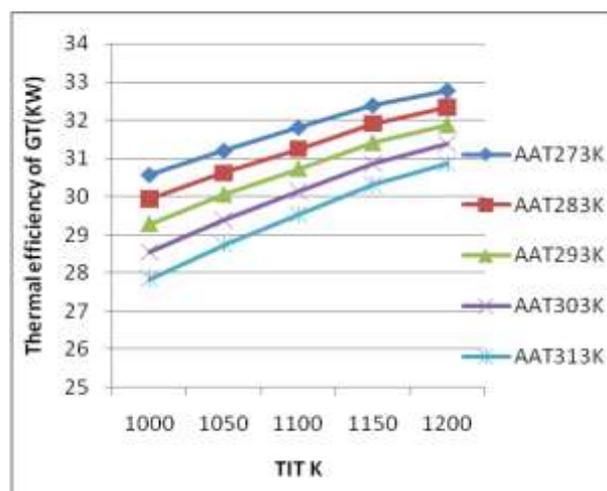


Fig.12: Effect of AAT and TIT on Efficiency of GT

Figure 12 and 13 shows the effect of inlet air temperature and turbine inlet temperature on the thermal efficiency of gas turbine and combined cycle efficiency. For a given turbine inlet temperature the efficiency of

gas turbine cycle decreases with increasing ambient air temperature. The reason behind this is that with increasing ambient air temperature the net power output of gas turbine decreases due to reduced mass flow rate through it and heat supplied is also reduced but the reduction in heat supplied is less than the reduction in power output. Therefore the overall effect is to reduce the efficiency. The improvement in efficiency is more in the lower range of TIT with decreasing AAT.

The change in thermal efficiency of combined cycle with increasing ambient air temperature for a given turbine inlet temperature is very small as compared to the change in efficiency of the gas turbine. The combined cycle efficiency insignificantly varies with AAT but significantly vary with TIT for any given value of AAT.

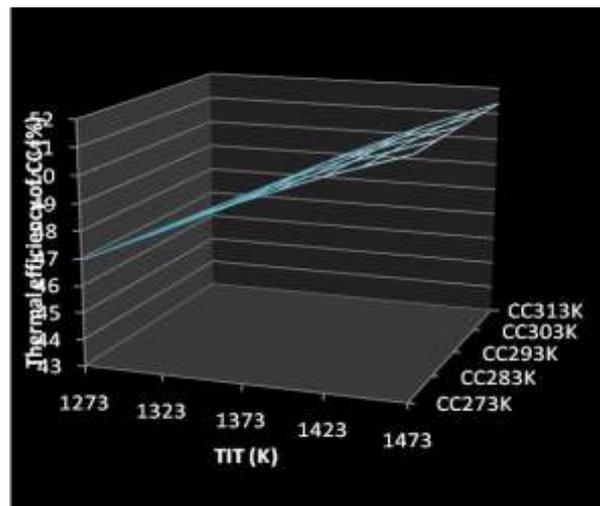


Fig.13: Effect of AAT and TIT on Efficiency of CC

V. CONCLUSION

A dual pressure HRSG combined cycle power plant has been critically examined and the effect of various operating parameters on its performance is studied. On the basis of results obtained the following conclusions are made:

- For a fixed maximum steam temperature in HRSG the difference between gas turbine exhaust temperature and steam generation temperature increases with increasing TIT which increases the heat recovery in HRSG and hence steam generation rate.
- For a increase in TIT from 1273K to 1473K the power output of combined cycle increases by 44% and thermal efficiency is found to increase by 6.4%.
- For a fixed TIT the power output and thermal efficiency of combined cycle increases with decreasing ambient air temperature. For an increase in AAT by 40⁰C at TIT 1200K, the power output of combined cycle decreases by 23.8% and thermal efficiency is found to decrease by 1.5% for the same increase in AAT.
- The efficiency of combined cycle increases with pressure ratio up to a certain value and after that it decreases with increasing pressure ratio for a fixed TIT. The optimum pressure ratio in this case is 12. The optimum pressure ratio decreases with an increase in HP steam pressure in HRSG.

- For a fixed steam temperature and TIT there is a feasible range of pressure ratio. For a gas turbine inlet temperature of 1100⁰C , the feasible pressure ratios are 8-16. This range increases with increasing TIT.
- The conclusion remains that the performance of combined cycle plant is affected by these parameters and the need for plant optimization is necessary for optimum values of operating parameters that offers the best plant performance.

REFERENCES

- [1]. Sanjay Singh, O., Prasad B.N, “Thermodynamic modeling and simulation of advanced combined cycle for performance enhancement,” J. power and Energy proc. IMechE, vol. 222 part A. 2008. PP.541-555.
- [2]. Bassily AM2005 Modeling, numerical optimization and irreversibility reduction of a dual pressure reheat combined cycle. Applied Energy 81: 127-151,2005
- [3]. De S, Biswal SK 2004 performance improvement of a coal gasification and combined cogeneration plant by multi pressure steam generation. Applied thermal engineering 24: 449-456
- [4]. Kotas T J 1995 The Exergy method of thermal plant analysis. Malawar, FL. Krieger publishing company
- [5]. Ganapathi V. heil, B. Rentz. J., “ Heat Recovery Steam Generator for cheng cycle application” international power conference PWR San Diego, vol 4, 1998, pp. 61-65.
- [6]. Nag, P.K and Raha, D. “Thermodynamic analysis of a coal based combined cycle power plant,” Heat recovery system and CHP, Vol.15 (2), 115-129, 1995.
- [7]. Sanjay. “Investigation of effect of variation of cycle parameters on thermodynamic performance of gas-steam combined cycle”. Energy, Vol.36, 157-167, 2011
- [8]. Cerri, G. “Parametric analysis of combined gas – steam cycles,” J. Eng. Gas turbine power, Vol.109 (1), 46-54, 1987.
- [9]. Wunsch, A., “Highest efficiencies possible by converting gas turbine plant into combined cycle plants”. Brown Boveri Review, Vol.10, 455-463, 1985.
- [10]. Naradasu,R.K, Konijeti,R.K and Alluru, V.R. 2007.Thermodynamic analysis of heat recovery steam generator in combined cycle power plant. Thermal Science, 11(4): 143-156.