Thermodynamic Analysis of Combined Cycle Power Plant

A.K.Tiwari[1], Mohd Islam[2], M.N.Khan[3]

[1]Greater .Noida.Institute of Technology, Greater Noida, India.
[2]Jamia Millia Islamia,New Delhi, India.
[3]Krishna Institute of Engineering & Technology,Ghaziabad, India.
[3]Corresponding Author: e-mail: khanrkgit@rediffmail.com ,

Abstract

Air Bottoming Cycle (ABC) can replace the heat recovery steam generator and the steam turbine of the conventional combined cycle plant. The exhaust energy of the topping gas turbine of existing combine cycle is sent to gas-air heat exchange, which heats the air in the secondary gas turbine cycle. In 1980's the ABC was proposed as an alternative for the conventional steam bottoming cycle. In spite of the cost of reducing hardware installations it could achieve a thermal efficiency of 80%. The complete thermodynamic analysis of the system has been performed by using specially designed programme, enabling the variation of main independent variables. The result shows the gain in net work output as well as efficiency of combined cycle is 35% to 68%.

Keywords: Topping Cycle, Bottoming cycle, Compression Ratio, Power Output

1. Introduction

The combined cycle has gained wide spread acceptance in the land based power generating industry and is definitely a proven techniogy. But due to HRSG, the cost has been found to be too high. As a consquence of high inlet temperature, a higher flue gas temperature results, which allows for improved combined cycle or cogeneration efficiency. A combined system which has not been widely investigated is the air bottoming cycle, where air, instead of steam, is used in a bottoming cycle to recover partially the energy from the turbine exhaust & convert it into useful power (Bolland O. et al, 1996). Wicks (1991) derived the concept of Air Bottoming Cycle (ABC) from the theory of ideal fuel burning engines by comparing the engines with the Carnot cycle. The air bottoming cycle may be employed to utilize the heat rejected from the gas turbine.

The ABC together with topping Gas Turbine Cycle is another type of combined cycle. It offers efficiency close to that of combined Gas Turbine/Steam Turbine Cycle and offers the potential of low weight as compared to combined Gas Turbine/Steam Turbine Cycle.

The combination of a gas turbine and an ABC represents a high-efficiency plant that provides clean, hot air for process needs. The technical-economic analysis showed that an implementation of this scheme at the industries that require hot air will result in significant fuel savings campared to the current technology and will have a payback time of 3 year. The air turbine cycle applied as a heat recovery cycle behind an industrial furnace will convert waste heat to electricity with an efficiency of upto 26% (31% on exergy basis). This result in considerable avoided costs and a payback time of 3-4 year additional fuel savings are obtained by passing the air turbine exhaust to the furnace as pre-heated combustion air. Two projects have been proposed to a governmental energy agency to demonstrate the viability of the ABC in these applications. The projects will be implemented at milk power production and industrial bakeries, and in the glass industry.(Korobitsyn, 2002)

In this system an air turbine is used to convert the exhaust energy into mechanical power, dispensing with all the hardware relvant to steam power plants (Boiler, Steam Turbine, Condenser, Pumps, Water Treatment plant, Cooling Towers etc) . The literature survey relvant to this paper are:

Korobitsyn (2002) considered and used the cycle as a compact and simple bottoming cycle in various applications such as upgrading option for simple gas turbines in the off shore industries. The technical and economical feasibility of ABC has also been evaluated; where hot air from the air turbine is supplied to food processing industries.

Kaikko (2001) presented the air bottoming cycle as an economical concept to increase the power generation efficiency of small and medium size gas turbines. A thermodynamic analysis has been presented for co-generative system where a fraction of compressed air in an inter-cooled ABC was taken to a Reversed Brayton Cycle to provide cold airflow.

Korobitsyn (1998) analyzed the performance of duel gas turbine cycle. Analysis of the duel gas turbine cycle with various topping gas turbines were performed and implementation of ABC at the gas turbine to resulted in increase in power output of 20 to 35 % when compared to steam bottoming cycle. The distinct features of this set up were its simpler and robust design.

Bolland et al (1996) performed the thermodynamic analysis of ABC and the result revealed that the ABC alternative is economical as compared with the other alternatives.

Najjar et al (1996) concluded that ABC could achieve the thermal efficiency of about 60%, which is not possible in simple gas turbine cycles.

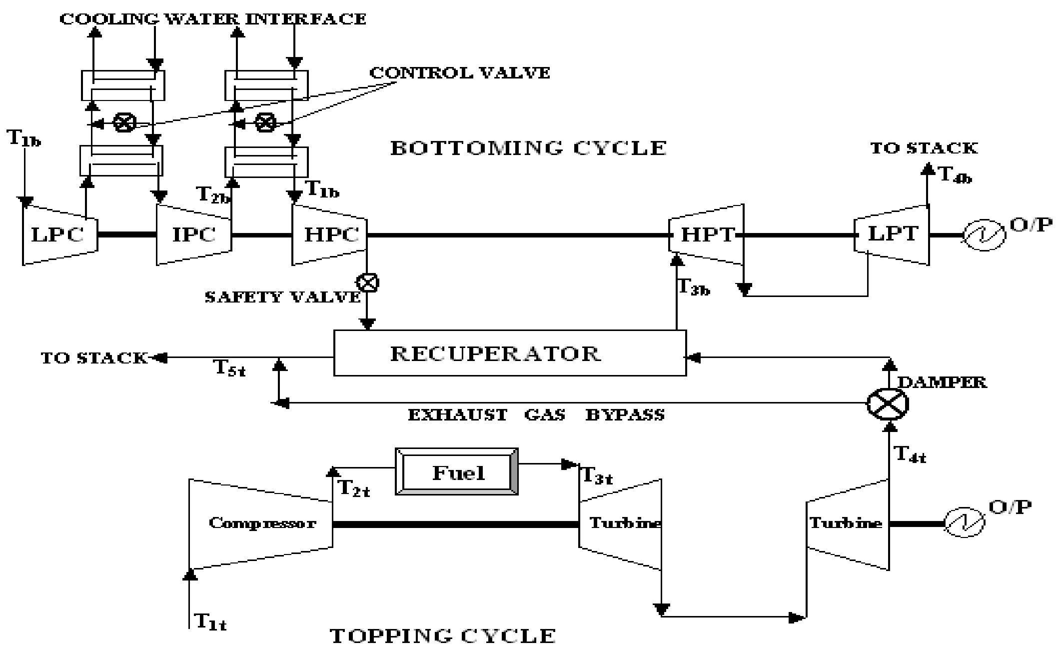


Figure 1. Schematic diagram of Air Bottoming Cycle of combined cycle power plant

Going through the literature, it is revealed that the use of Air Bottoming cycle in conjunction with the gas turbine topping cycle increases the overall efficiency of the cycle. However, the use of intercoolers and recuperators in the bottoming cycle has not been studied. So, in the present work, power optimization analysis of air bottoming cycle has been attempted.

2. Analysis of Topping Cycle

In topping cycle the air is compressed in compressor from 1 to 2 where its temperature raises from T_{1t} to T_{2t}.The compressed air then enter the combustion chamber where the combustion of fuel takes place. This results in rise of temperature of combustion product from T_{2t} to T_{3t}. The high temperature gases enter the turbine where it expands to the final temperature T_{4t}.

Therefore expression for turbine work in topping cycle (W_t) is given by

$$W_t = m_g c_{pg} (T_{3t} - T_{4t}) \tag{1}$$

Where $T_{4t} = T_{3t}\{1-\eta_t(1-r_{p1}^{-\beta})\}$ (2)

And expression for compressor work in topping cycle (W_c) is given by

$$W_c = m_a c_{pa}(T_{2t} - T_{1t}) \quad \text{Where } T_{2t} = T_{1t}\{1+\frac{(r_{p1}^{\alpha}-1)}{\eta_c}\} \tag{3}$$

Network of topping cycle $W_{nett} = W_t - W_c$ (4)

The expression for heat supplied in the combustion chamber (q) is

$$q = \frac{(m_g c_{pg} T_{3t} - m_a c_{pa} T_{2t})}{\eta_{comb.}} \tag{5}$$

Therefore efficiency of the topping cycle is $\eta_{topping} = \frac{W_{nett}}{q}$ (6)

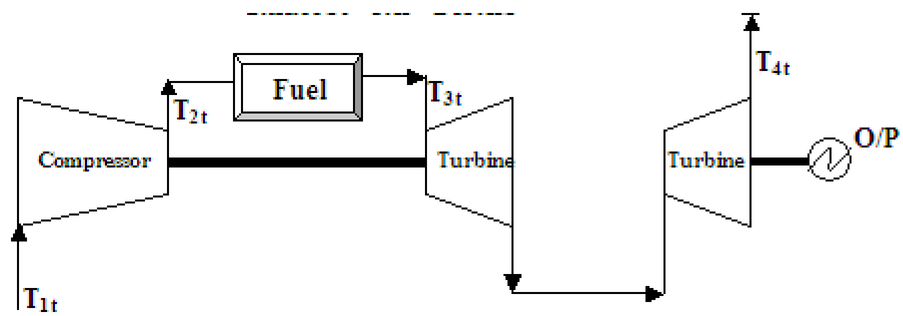


Figure 2. Schematic diagram of Topping Cycle of combined cycle power plant

3. Analysis of Bottoming Cycle

In Air Bottoming cycle air at a temperature of T_{1b}(298K) enter the Low Pressure Compressor (LPC) where it compresses and leave it at temperature higher than T_{1b}.There is Low Pressure Intercooler (LPI) between Low Pressure Compressor (LPC) and Intermediate Pressure Compressor(IPC) and High Pressure Intercooler (HPI) between Intermediate Pressure Compressor(IPC) and High Pressure Compressor (HPC). The temperature of air after leaving High Pressure Intercooler (HPI) is same as the temperature of air at the entry of Low Pressure Compressor (LPC) and after compression in High Pressure Compressor (HPC) its temperature raises to T_{2b}.This high temperature air enter the recuperator where it absorb heat from the exhaust gas of topping cycle and in this way the temperature of air raises to T_{3b}. The high pressure and high temperature air after leaving the recuperator enters the turbine where it expands to the final temperature of T_{4b}.

Therefore the expression for temperature of air leaving the High Pressure Intercooler (HPI) in bottoming cycle is given by

$$T_{2b} = T_{1b}\{1+\frac{(r_{p2}^{\alpha/(n+1)}-1)}{\eta_c}\} \tag{7}$$

The expression for temperature of air leaving the turbine in bottoming cycle is given by

$$T_{4b} = T_{3b}\{1-\eta_t(1-r_{p2}^{-\alpha})\} \tag{8}$$

And the expression of effectiveness for the heat exchanger (ε) is

$$\varepsilon = \frac{T_{3b} - T_{2b}}{T_{4t} - T_{2b}} \tag{9}$$

Therefore the expression for turbine inlet temperature of air in bottoming cycle is given by

$$T_{3b} = \varepsilon(T_{4t} - T_{2b}) + T_{2b} \tag{10}$$

And the expression for turbine work in bottoming cycle (W_t) is given by

$$W_t = m_a c_{pa}(T_{3b} - T_{4b}) \tag{11}$$

Expression for compressor work in bottoming cycle (W_c) is given by

$$W_c = m_a c_{pa}(T_{2b} - T_{1b}) \tag{12}$$

So net work of bottoming cycle $W_{netb} = W_t - W_c$ (13)

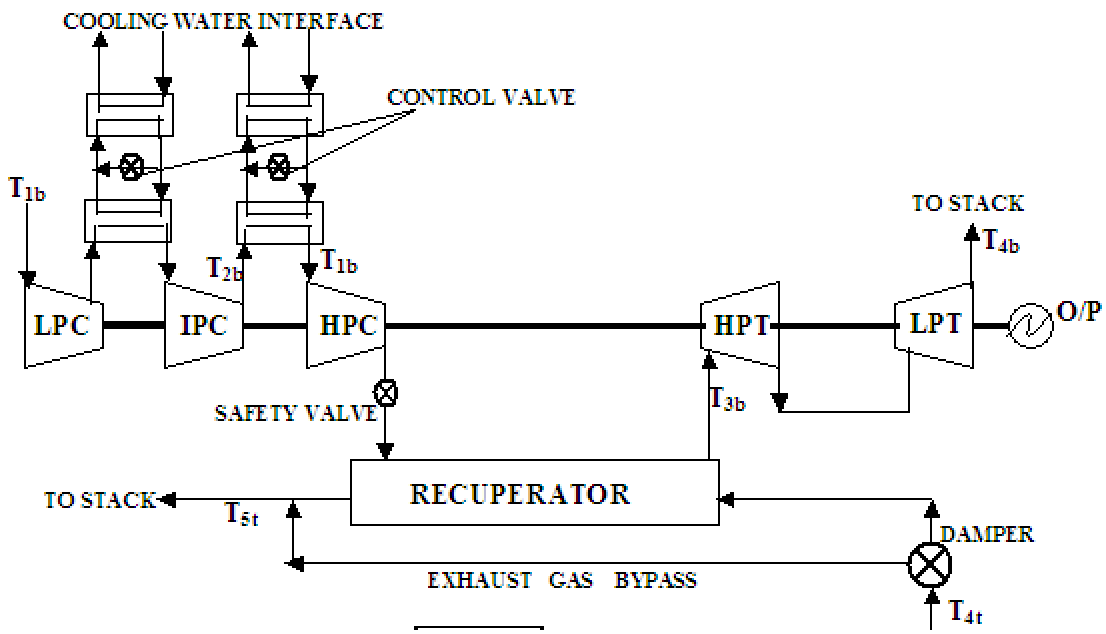


Figure 3. Schematic diagram of Bottoming Cycle of combined cycle power plant

The net work of combine cycle is the net work of topping cycle plus net work of bottoming cycle is

$$W_{net} = W_{nett} + W_{netb} \tag{14}$$

And the expression for net efficiency of the combine plant is

$$\eta_{combine} = \frac{W_{net}}{q} \tag{15}$$

4. Results and Discussion

From the above analysis it is clear that the values of work output, efficiency of topping cycle as well as bottoming cycle effected by number of independent variables like turbine inlet temperature of topping cycle (T_{3t}), pressure ratio of topping cycle (r_{p1}), pressure ratio of bottoming cycle (r_{p2}), mass flow rate in bottoming cycle (m_{ab}) and number of intercoolers (n) in the bottoming cycle. This study analyzes those factors that affect the output of topping cycle, bottoming cycle as well as combined cycle.

4.1 Topping cycle

From equations (1), (2) and (5) it is clear that the work output and efficiency of topping cycle depends on pressure ratio of topping cycle (rp1) and turbine inlet temperature (T_{3t}). From equation (1) and (2) it is easy to predict that as the turbine inlet temperature as well as pressure ratio of topping cycle (r_{p1}) increases, turbine work increases but with more increase in pressure ratio of topping cycle compressor work also increases that will results in slight decrease in work output of topping cycle.

In this study, pressure ratio of topping cycle (r_{p1}) vary from 4 to 12 and turbine inlet temperature of topping cycle (T_{3t}) vary from 1000k to 1400k, on this data results shows that on particular pressure ratio as the turbine temperature increases net work of the cycle increases as well as on the particular turbine inlet temperature as the pressure ratio increases net work of the cycle also increases. In fig.4.1 (a) the network of topping cycle are plotted against turbine inlet temperature as well as pressure ratio of topping cycle simultaneously. Here it is clear that as the turbine inlet temperature increases the rate of increase of network of topping increases with increase in pressure ratio.

Figure 4.1 (b) shows the variation of efficiency of topping cycle with respect to pressure ratio of topping cycle as well as turbine inlet temperature of topping cycle. It is indicated from the graphs that the efficiency of topping cycle increases with increase in pressure ratio as well as turbine inlet temperature of topping cycle and attain its maximum value (32%) at 1400k of turbine inlet temperature and at the pressure ratio of 12 of topping cycle.

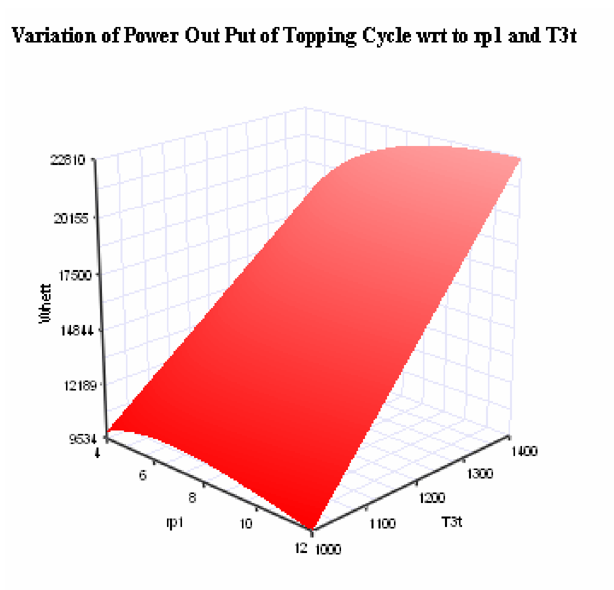


Figure 4.1a Surface of power output of topping Cycle w.r.t and T3t

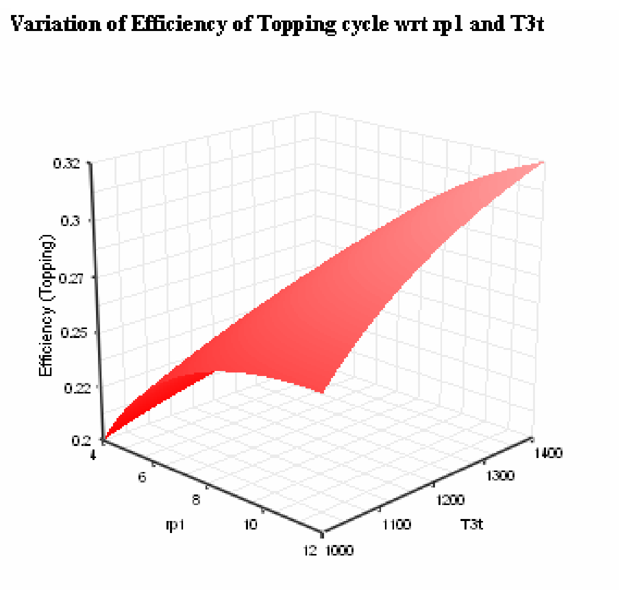
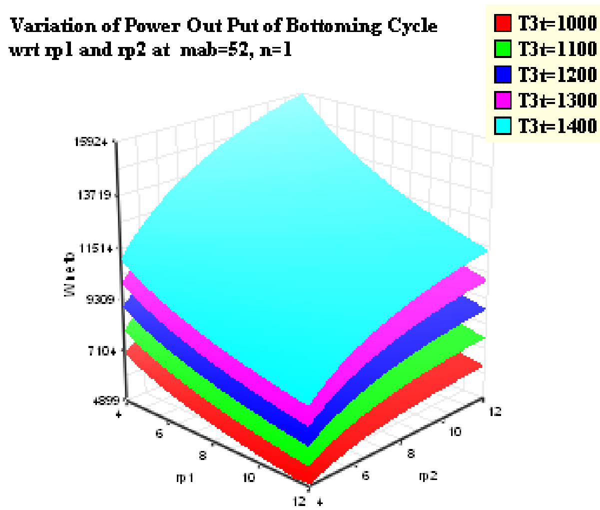


Figure 4.1b Surface of efficiency of topping cycle w.r.t rp1 and T3t

4.2 Bottoming cycle

From equations (11), (12) and (13) it is clear that the network of bottoming cycle depends on pressure ratio and turbine inlet temperature of bottoming cycle. From equation (10) the turbine inlet temperature of bottoming cycle depends on pressure ratio of topping cycle, turbine inlet temperature of topping cycle and number of intercoolers in bottoming cycle.



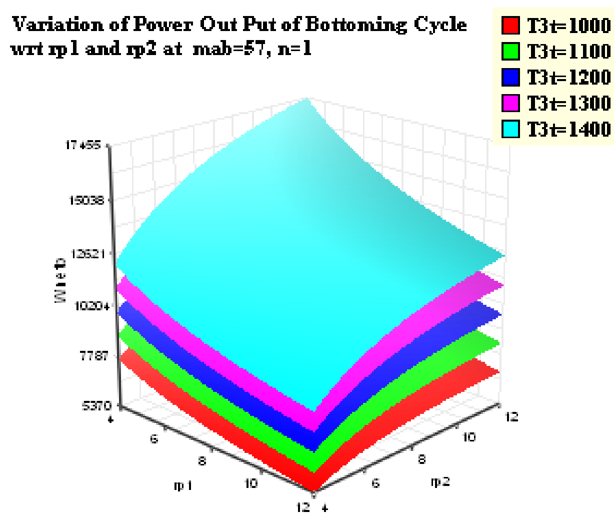


Figure 4.2a Surfaces of power output of bottoming cycle w.r.t rp1 and rp2 at mab=52, n=1

Figure 4.2b Surfaces of power output of bottoming cycle w.r.t rp1 and rp2 at mab=57, n=1

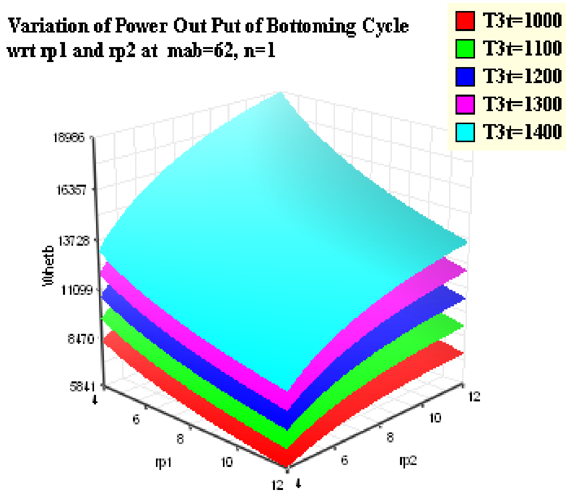


Figure 4.2c Surfaces of power output of bottoming Cycle w.r.t rp1 and rp2 at mab=62, n=1

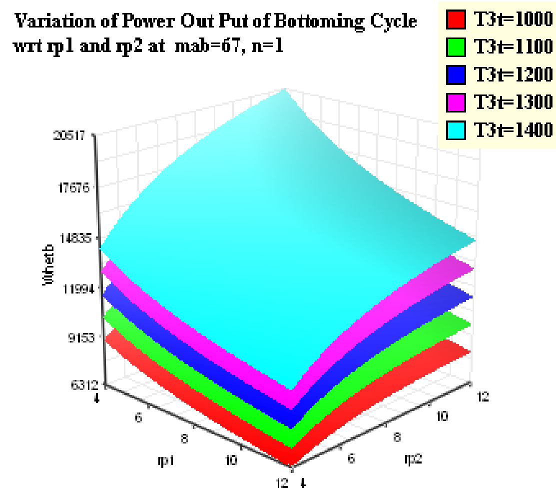


Figure 4.2d. Surfaces of power output of bottoming Cycle w.r.t rp1 and rp2 at mab=67, n=1

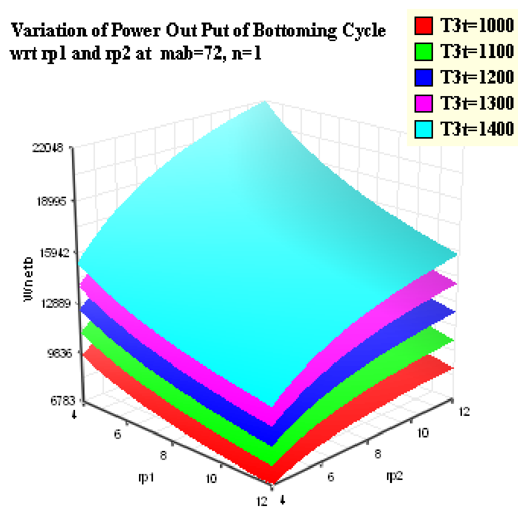


Figure 4.2e Surfaces of power output of bottoming Cycle w.r.t rp1 and rp2 at mab=72, n=1

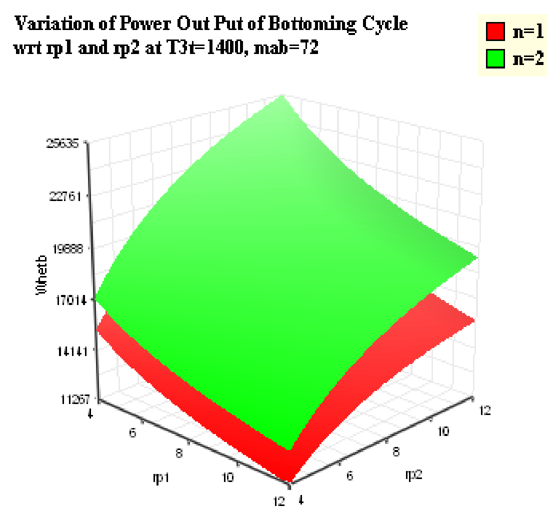


Figure 4.2f Surfaces of power output of bottoming cycle w.r.t rp1 and rp2 at T3t=1400K, mab=72

The graphs 4.2a to 4.2e shows variation of net work of bottoming cycle with respect to pressure ratio of topping cycle (r_{p1}) and pressure ratio of bottoming cycle (r_{p2}) for different turbine inlet temperature varying from 1000K 1400K and different mass flow rate in bottoming cycle (m_{ab}) varying from 52kg/s to 72 kg/s for one intercooler. It is clear from the graph that as the pressure ratio of topping cycle increases the net work of bottoming cycle decreases this is because as the pressure ratio of topping cycle increasing the exhaust temperature of topping cycle decreases which results in decrease in turbine inlet temperature of bottoming cycle. In the opposite way as the pressure ratio of bottoming cycle increases the net work of bottoming cycle increases this is because as the pressure ratio of bottoming cycle increases the turbine inlet temperature of bottoming cycle increases and exhaust temperature of bottoming cycle decreases which results in increase in net power output, further more increase in pressure ratio of bottoming will slightly decrease in net power output because of increase in compressor work. Here the maximum work output is for r_{p1}=4 and r_{p2}=12

It is shown in graph as the turbine inlet temperature of topping cycle increases the surface of net work output shifted towards above because it increases the turbine inlet temperature of bottoming cycle and attains its maximum value when turbine inlet temperature is 1400K

Similarly as the mass flow rate in bottoming cycle increases the net work output also increases because work output is directly proportional to the mass flow rate and attains its maximum value when mass flow rate is 72 kg/s.

Also by increasing the number of intercooler in bottoming cycle from one to two it saves in compressor work and net work output also increases. Here the maximum value of net work output is 25635KW for r_{p1}=4, r_{p2}=12, T_{3t}=1400K, m_{ab}=72kg/s and n=2 shown by green surface in figure 4.2f.

4.3 Combined cycle

Equation (14) gives the expression for network of combine cycle, and from the above analysis the results for network of combined cycle are shown in figures 4.3(a) to 4.3(e).

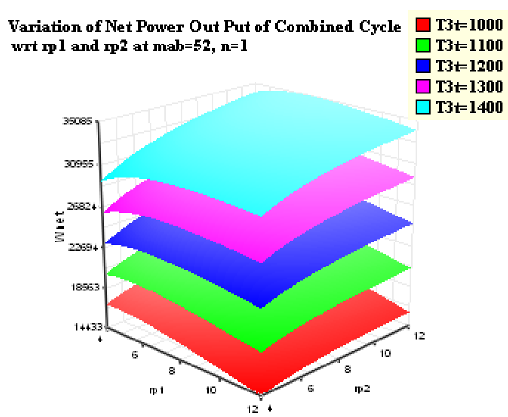


Figure 4.3b. Surfaces of power output of combined
Cycle w.r.t rp1 and rp2 at mab=52, n=1

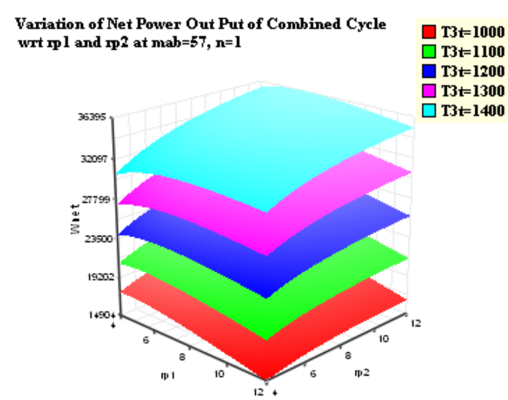


Figure 4.3a Surfaces of power output of combined
Cycle w.r.t rp1 and rp2 at mab=57, n=1

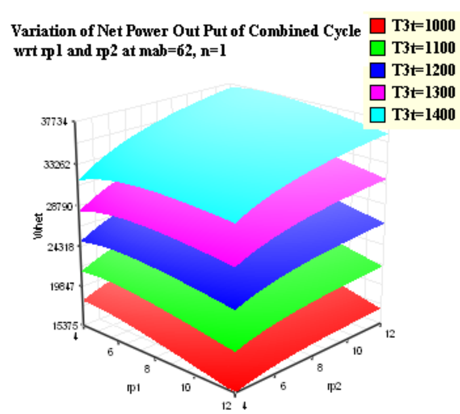


Figure 4.3c Surfaces of power output of combined
Cycle w.r.t rp1 and rp2 at mab=62, n=1

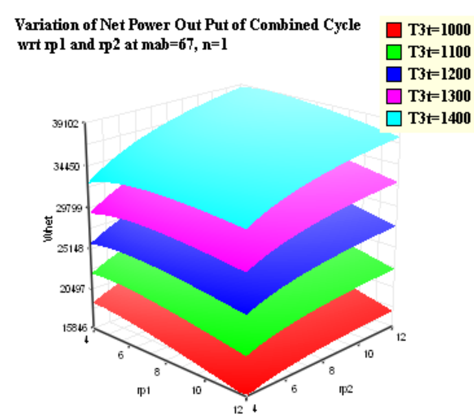


Figure 4.3d. Surfaces of power output of combined
Cycle w.r.t rp1 and rp2 at mab=67, n=1

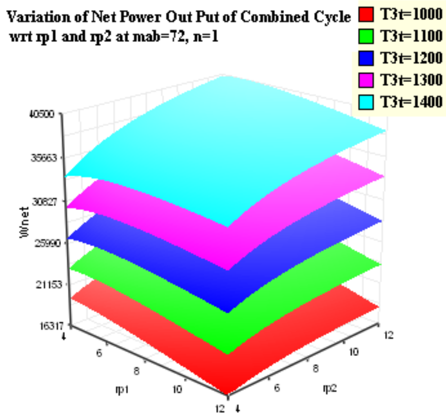


Figure 4.3 e Surfaces of power output of combined Cycle w.r.t rp1 and rp2 at mab=72, n=1

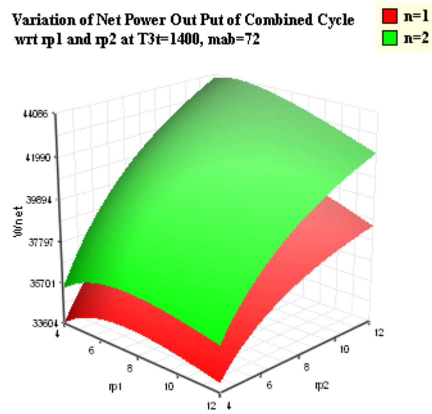


Figure 4.3f. Surfaces of power output of combined cycle w.r.t rp1 and rp2 at T3t=1400K, mab=72

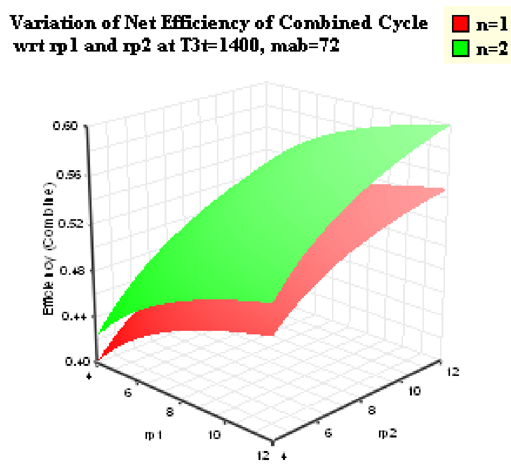


Figure 4.3g. Surfaces of net efficiency of combined cycle w.r.t rp1 and rp2 at T3t=1400K, mab=72

The above graphs (4.3a to 4.3e) indicate variation of net work of combined cycle with respect to pressure ratio of topping cycle and pressure ratio of bottoming cycle for different turbine inlet temperature of topping cycle and mass flow rate in bottoming cycle for one intercooler. It is clear from the graph as the pressure ratio of topping cycle increases the net work output of combined cycle increases and attains its maximum value at r_{p1}=6 then decreases slightly. This position shifted towards above when increases in turbine inlet temperature; mass flow rate in bottoming cycle and number of intercooler in bottoming cycle.

Similarly as the pressure ratio of bottoming cycle increases the net work output of combined cycle increases and attains its maximum value at r_{p2}=12. This position shifted towards above as shown in graphs by increase in turbine inlet temperature of topping cycle; mass flow rate in bottoming cycle and number of intercooler in bottoming cycle. Figure 4.3f shows the optimum position of network output of combined cycle i.e 44086 kW for r_{p1}=6, r_{p2}=12, T_{3t}=1400K, m_{ab}=72 and n=2 by green surface.

The graph 4.3g shows the variation of net efficiency of combined cycle w.r.t rp1 and rp2 corresponding to maximum power output condition ie at T3t =1400K, mab = 72 kg/s for one and two intercooler in ABC. It is clear from graph that as pressure ratio of topping cycle and bottoming cycle increases the net efficiency increases and attains its maximum value at rp1=12, rp2=12. Also by adding two intercooler in ABC the net increase in efficiency is 8%.

5.0 Conclusions

On the basis of above analysis on topping, bottoming as well as combined cycle outputs the following conclusion are made by varying the pressure ratio of topping cycle (r_{p1}), pressure ratio of bottoming cycle (r_{p2}), turbine inlet temperature of topping cycle (T_{3t}), mass flow rate in bottoming cycle (m_{ab}) and number of intercoolers in the bottoming cycle (n):

a) The network of combined cycle increases as increases in turbine inlet temperature of topping cycle for every particular value of mass flow rate of ABC and pressure ratio of topping cycle of ABC.
b) The network of bottoming cycle increases as increase in pressure ratio of bottoming cycle and decreases as increase in pressure ratio of topping cycle for every particular value of turbine inlet temperature of topping cycle as well as mass flow rate in bottoming cycle and attains its maximum value for r_{p1}=4, r_{p2}=12, T_{3t}=1400K, m_{ab}=72kg/s for two intercooler.
c) The use of two intercooler considerable improve the performance of ABC and increasing the bottoming cycle power output by 16% in the relation to the one intercooler case
d) For any particular value of pressure ratio of topping cycle any particular value of pressure ratio of bottoming cycle, turbine inlet temperature of topping cycle, and any mass floe rate in bottoming cycle of ABC, the network of combined cycle attain its maximum value when two intercoolers are used in bottoming cycle.
e) A combined system with air bottoming cycle can improve the net power output as well as efficiency by about 35% to 68%.
f) The optimum point for this system is found to be, r_{p1}= 6, r_{p2}=12, m_{ab}=72 of air bottoming cycle with two intercoolers at turbine inlet temperature of 1400k.
g) The net power output and efficiency of combined cycle increases up to 8% and 18% respectively, by adding two intercooler in ABC in relation to the one intercooler case.

The performance of this system can also be analysed by considering the pressure drop in combustion chamber, introducing intercooler in topping cycle and with reheat in topping and bottoming cycle both in future work.

Nomenclature

T_{1t}	Compressor inlet temperature in topping cycle
T_{2t}	Compressor exit temperature in topping cycle
T_{3t}	Turbine inlet temperature in topping cycle
T_{4t}	Turbine exit temperature from topping cycle
T_{1b}	Compressor inlet temperature in bottoming cycle
T_{2b}	Compressor exit temperature in bottoming cycle
T_{3b}	Turbine inlet temperature in bottoming cycle
T_{4b}	Exhaust Temperature from air turbine
W	Work output
W_{nett}	Net work of topping cycle
W_{netb}	Net work of bottoming cycle
W_{net}	Net work of Combine Cycle
ϵ	Effectiveness of Heat Exchanger
n	Number of intercooler
r_{p1}	Pressure ratio in topping cycle
r_{p2}	Pressure ratio in bottoming cycle
η_c	Compressor efficiency
η_t	Turbine efficiency
η_{comb}	Combustion Efficiency
$\eta_{topping}$	Efficiency of Topping Cycle
$\eta_{combine}$	Efficiency of Combined cycle
q	Heat Supplied in combustion chamber
C_p	Specific Heat

Variables

r_{pt} (Pressure Ratio Of Topping Cycle)	: 4 to 12
r_{pb} (Pressure Ratio Of Bottoming Cycle)	: 4 to 12

T_{3t} (Turbine Inlet Temperature of Topping Cycle) : 1000 to 1400K
m_{ab} (Mass flow rate of air in bottoming Cycle) : 52 to 72 Kg/s
n (Number of Inter-Cooler) : 1to 2

Constants

$\eta_{comb} = 0.96$, $\eta_t = 0.90$, $\eta_c = 0.90$
T_{1t} (Compressor inlet temperature of topping cycle) = 298K
T_{1b} (Compressor inlet temperature of Bottoming Cycle) = 298K
m_a (Mass flow rate of air in Topping Cycle) = 69Kg/s
m_f (Mass flow rate of fuel) = 1.32Kg/s
m_g (Mass flow rate of gas in topping cycle) = $m_a + m_f$ =70.32Kg/s
ϵ (Effectiveness of heat exchanger) = 0.90
C_{pg} (Specific Heat Of Gas) = 1.14KJ/KgK
C_{pa} (Specific Heat of air) = 1.005KJ/KgK
γ (For air) = 1.4
$\alpha = (\gamma-1)/\gamma = 0.285$
γ'(For Gas) = 1.33
$\beta = (\gamma'-1)/\gamma' = 0.248$

References

[1] Anonymous, 1991 "Low Cost 'Air Bottoming Cycle' for Turbines", Gas Turbine World, May-June, 1991.
[2] Bathie W.W., 2000 'Fundamentals of Gas Turbines', John Wiley & Sons, 2000
[3] Boland O., Forde M., and Hande B., 1996. "Air Bottoming Cycle: Use of Gas turbine Waste Heat for Power Generation", ASME Paper 95-CTP-50, 1996.
[4] Cohen H., Rogers G.F.C. and Satavanmutto H.H., 1996 "Gas Turbine Theory." 4th Edn., Longmans, 1996.
[5] Happenstall T., 1998 "Advance Gas Turbine Cycle for Power Generation: A critical Review" ASME-98-GT-846.
[6] Horlock J.H; 1995 "Combined power plant-past, present and future" Trans of the ASME, J. of Engg. For Gas Turbine and Power, Vol. 117, 1995.
[7] Kaikko J., 2001. "Air Bottoming cycle for Cogeneration Power, Heat and cooling" Department of Energy Technology, Stockholm, Sweden, 2001.
[8] Kakaras E., Doukelis A., Saharfe J.,2001 "Application of gas turbine plant with cooled compressor intake air", ASME 2001-GT-110.
[9] Korobitsyn M.A., 1998. "New and Advance Energy Conversion Technologies. Analysis of Cogeneration, Combined and Integrated Cycles", CHP 7 (107-117), 1998.
[10] Korobitsyn Mikhail, 2002. "Industrial application of Air Bottoming Cycle" pentagon, Vol.43, pp 1311-1322.
[11] Najjar Y.S.H. and Zaamout M.S., 1996. "Performance Analysis of Gas Turbine Air Bottoming Combined System", Energy Conversion and Management, vol. 37, pp. 399-403, 1996.
[12] Wicks F., 1991 "The Thermodynamic Theory and Design of an Ideal Fuel Burning Engine", Proceedings of the 25th Intersociety
[13] Energy Conversion Engineering Conference IECEC'90, Boston, Vol.2, pp.474-481, 1991.
[14] Yahaya S.M., 2000. "Turbine Compressor and Fans" 2000

Biographical notes

Mr. A.K. Tiwari is Senior Lecturer in Department of Mechanical Engineering, Greater Noida Institute of Technology, Gr.Noida, India. He is presently pursuing M.Tech from Jamia Millia Islamia, New Delhi. He has engaged in teaching and research activities since last 7 years. His field of specialization is Thermal Engineering.

Mr. M.N.Khan is Associate Professor, Department of Mechanical Engineering, Krishna Institute of Engg. & Tech. Ghaziabad, India. He has done his M.Tech. from Delhi College of Engineering and presently pursuing Ph.D. from Jamia Millia Islamia, New Delhi. He has engaged in teaching and research activities since last 11 years. His field of specialization is Thermal. Mr. Mohd Nadeem Khan has published several papers in various national, international conferences and journals. He has guided students for their M. Tech. work.

Prof. Mohd. Islam is working as a professor in Department of Mechanical Engineering, Jamia Millia Islamia, New Delhi, India. He has engaged in teaching and research activities since last 25 years. Prof. M. Islam has published several papers on various National and Internal Journals and guided several M.Tech and Ph.D. thesis in Thermal areas.