CHAPTER 1.

INTRODUCTION AND LITERATURE REVIEW

1.1 INTRODUCTION:

The turbines are the best means to generate mechanical power, as they are free from reciprocating and rubbing parts thus balancing problems are few and lubrication consumption is also very low. In the initial years of its development, turbines were used with water as the working fluids. Thus the concept of hydroelectric power came into picture around the turn of the twentieth century the steam turbine begins its career. Now it has become the most important prime mover for electricity generation. But steam turbines have their own limitations as they are very bulky and employ insufficient matter of heating. It was realized that instead of water being heated indirectly to generate steam if product of combustion are directly expanded in turbine than compact power plant was feasible. Thus the concept of gas turbine came into picture. In its initial years of development gas turbines is only used for aviation and peaks load power production as its thermal efficiency was very low and it was not economical to use gas turbine for long running hours. As a result of this innovative modifications were made in conventional power plant to improve thermal efficiency and specific work output [5].

In the earliest days the combustion was possible in two ways one at constant volume and other at constant pressure. The combustion at constant volume although more efficient was more intermittent in nature and pose practical difficulties and thus constant pressure combustion was accepted as having great potential in future.

1.2 OPEN AND CLOSE CIRCUIT

In the simple open circuit gas turbine plants atmosphere air is continuously compressed in the compressor and delivered to the combustion chamber pass out to the atmosphere after expanding through the turbine. In this arrangement since the working fluid in not

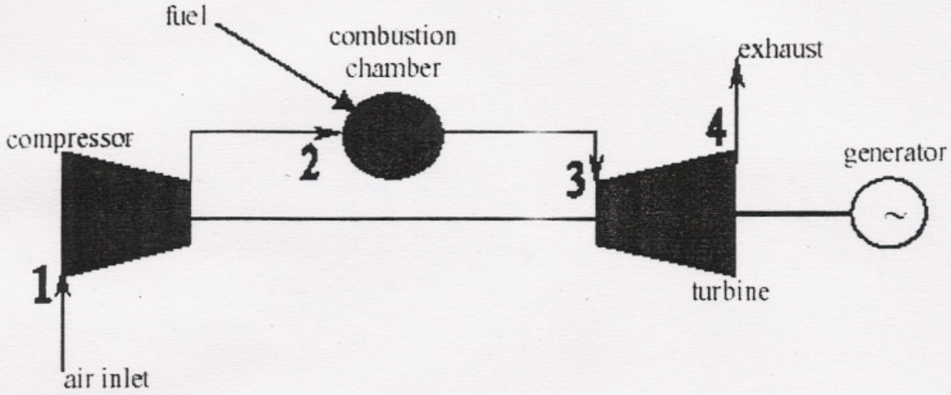
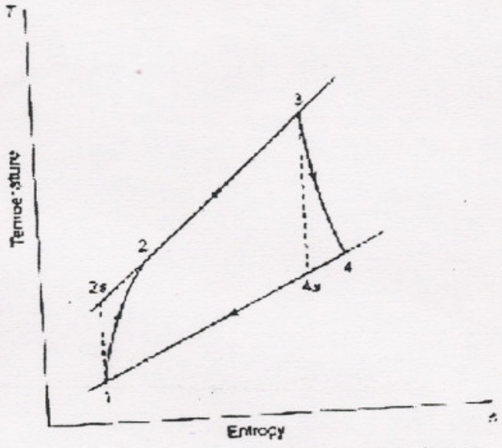


Figure 1.1

restored to its initial state, technically speaking such a plant does not execute a cycle. A cycle can only be executed in the close circuit gas turbine plant. Here the same working fluid (air or any other gas) circulates through its various components. Heat cannot be supplied to the working fluid by internal combustion; instead, it is supplied externally by employing a heat exchanger, which replace the combustion chamber of the open circuit plant. The chief disadvantage of this plant is that heat is supplied externally to the working fluid. This requires additional equipment besides being less efficient [5]



Actual cycle without pressure losses

Figure 1.2

In the actual consent pressure cycle the work in the compressor and turbine is adiabatic instead of isentropic as shown in Figure 1.2. Therefore, the compressor and turbines efficiencies are

$$\eta_c = T_{2s} - T_1 / T_2 - T_1$$

$$\eta_c = T_3 - T_4 / T_2 - T_{4s}$$

In the absence of pressure losses the pressure and ideal temperature ratio for the compressor and turbine are the same. The actual values of turbines and compressor work are given by

$$W_T = c_p (T_3 - T_4) = c_p (T_3 - T_{4s})\eta_T = c_p \eta_T T_3 [1 - T_{4s} / T_3]$$

$$W_C = c_p (T_2 - T_1) = c_p / \eta_c (T_{2s} - T_1)$$

The output of the plant is given by,

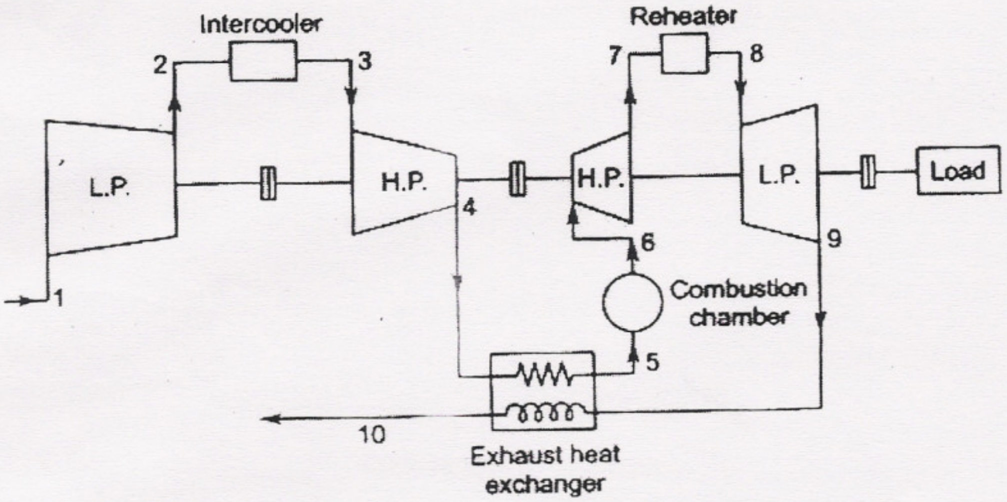
$$W_P = w_T - w_C$$

1.3 CYCLE WITH REHEAT AND EXHAUST HEAT EXCHANGER:

If we consider the figure below without intercooler, the air before entering the combustion chamber receives heat from the outgoing exhaust gases in the heat exchanger. This requires less heat to be added in the combustion chamber. The gases after expansion in the H.P.turbines are again heated to their initial temperature ($T_6 = T_8$) before entering the I.P.Turbine.

The cooling of air between two stages of compression is known as intercooling. This reduces the work of compression and increasing the specific output of the plant with the decrease in thermal efficiency. The loss of efficiency due to intercooling can be remedied by employing exhaust heat exchange as in the reheat cycle.

In view of the above it may be concluded that the best method of increasing the specific output of the plant is to employ inter-cooling and re-heating along with exhaust heat exchange. However, this increases the bulk of the plant considerably with only a marginal gain due to inter-cooling. Therefore, the designers of practical gas turbine plants have little incentive to use inter-cooling. In contrast to this, the gain due to re-heat and exhaust heat exchange are far more attractive.



Gas turbine plant with intercooling, reheating and exhaust heat exchange

Figure 1.3

Re-heating increase the specific work output but efficiency degraded marginally because of it. Re-heating is effective only for higher-pressure ratio. Inter-cooling results in increase in specific work output but thermal efficiency gets degraded because of it. Re-generation results in improvement in thermal efficiency but only when pressure ratio is less then optimum for maximum specific work output. It was realized that the thermal efficiency of gas turbine was poor, as around 60% of the total heat generated in combustion chamber goes waste in the exhaust. **Hence it was realized that thermal efficiency could improve considerably if we can make use of this waste energy, to generate steam**. Thus the concept of combine cycle came into picture.

In a combined steam and gas turbine (STAG) power plant the huge loss of energy in the gas turbine exhaust is significantly reduced by utilizing its heat in a 'bottoming cycle', here the high temperature exhaust gas transfers a large proportion of its heat to raise steam for the steam turbine power plant. Several different ways have been adopted to achieve this.

The energy in the gas turbine exhaust is used during feed water heating in the economizer, and evaporation and superheating in the evaporator and super heater respectively. The ultimate aim is to obtain higher overall thermal efficiencies, which are much higher than the values obtained in the high efficiency large steam power plants. The combined cycle plant combines the thermodynamic advantages of both the high temperature gas turbines and the lower temperature steam turbine power plants [17].

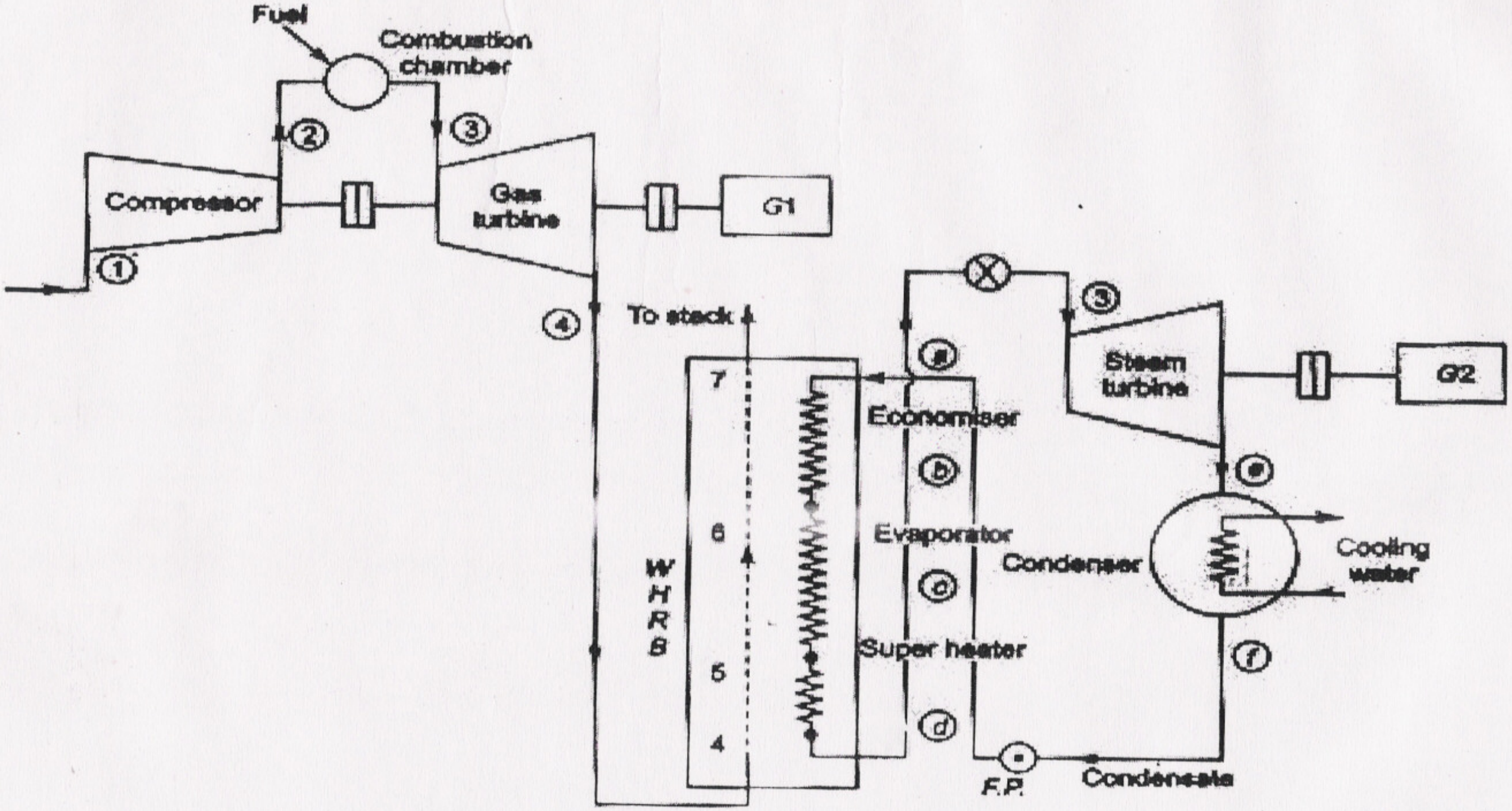
1.4 THE BASIC COMBINED CYCLE:

As stated before the principal process in a combined gas and steam turbine cycle is the recovery of heat energy in the gas turbine exhaust by the feed water and steam in the steam turbine plant.

Figure 1.4 shows the two power plant cycles, namely the Joule/Brayton cycle (gas turbine cycle) and the Rankine cycle (steam turbine cycle). The gas turbine power plant cycle (1-2-3-4-1) is the 'topping cycle: it consists of heat and work transfer processes occurring in the high temperature region. Such processes in the lower temperature region follow this. The low temperature region has the 'bottoming cycle the Rankine steam cycle (a-b-c-d-e-f-a). A waste heat recovery boiler (WHRB) transfers heat energy from the high temperature gas turbine exhaust gases to water and steam used in the bottoming cycle.

The gas turbine exhaust gases reject heat during the constant pressure process (4-1). The feed water absorbs part of this heat, and the wet and superheated steam during the processes a-b, b-c and c-d respectively. Figure shows the arrangement employed in the combined gas and steam turbine cycle power plant.

The gas turbine plant consists of the air compressor, combustion chamber, gas turbine and the electric generator (alternator) G.I. The



Combined gas and steam turbine power plants

Figure 1.4

corresponding processes of compression (1-2), combustion (2-3) and expansion (3 – 4) are also shown in figure. It is observed in figure, that the gas turbine plant does not complete a closed cycle as depicted in **Figure 1.4**.

The exhaust from the gas turbine passes through a heat exchanger i.e. WHRB, for brevity is also referred to as HRB. It is hear that the heat energy from the gas turbine exhaust gases is transferred to steam turbine plant. The lower temperature steam power plant consists of WHRB, steam turbine, condenser and the electric generator. The conventional feed water, heaters of the turbine plant, though used are shown in **Figure1.4**. The feed pumps supply the feed water to the economizer section (a-b) of the WHRB. After reaching the saturation temperature it absorbs the latent heat in the evaporator (b-c), the dry steam is heated to the design temperature in the super heater (c-d) before entering the turbine.

Various stations in the gas circuit corresponding to the super heater, evaporator and economizer sections are shown as points 4.5,6, and 7 respectively [5].

1.4.1 Disadvantages of combined gas steam turbine plants: -

- Combined cycle plants are more complex; they require more skill and better-trained personnel in modern technology.

- It is less flexible in terms of fuels, which can be used.

1.5 Air Bottoming Cycle:

Like the steam bottoming cycle in a combined cycle the Air Bottoming Cycle may utilize heat rejected from a gas turbine. A general flow sheet diagram of ABC is shown in **Figure 1.5**. Ambient air (1) is drawn through a filter and is compressed in the low-pressure compressor (LPC). The air is then

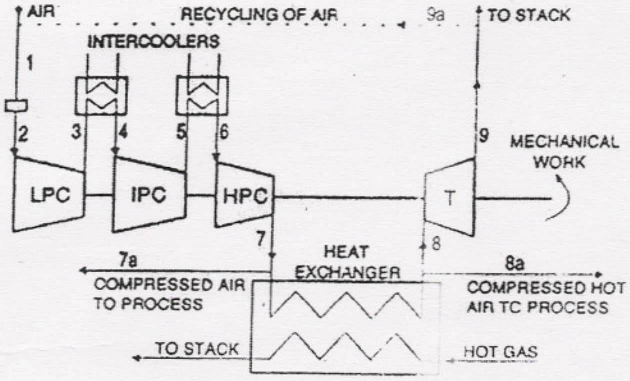


Figure 1.5

Cooled (3-4) before it is further compressed in an intermediate compressor (IPC). Again the air is cooled (5-6) before the final stage of compression in the high-pressure compressor HPC. In Figure the process shown has the compression divided into three sections with two inter-coolers. It is of course possible instead of having to inter-coolers to use one or three or ever more inter-coolers. This is a question about what is practical and balanced between cost and benefit of having more inter-coolers.

The compressed air (7) is heated in a heat exchanger before it enters (8) a turbine. In the turbine the air is expanded while shaft work is generated. After the turbine the air is exhaust to a stack. The work generated is sufficient to drive the compressor and a generator. A gear between the turbine shaft and the generator may be necessary.

The **ABC** offers the possibility to provide to an external process compressed air at moderate temperature (7a) and / or compressed hot air

(8a). The ABC may be a closed or partially closed cycle by recycling the air (9a). In the literature extensive surveys can be found on the use and thermo-dynamic potential of a variety of a power cycle working fuels where heat sources are latent and /or sensible heat. **[11]**

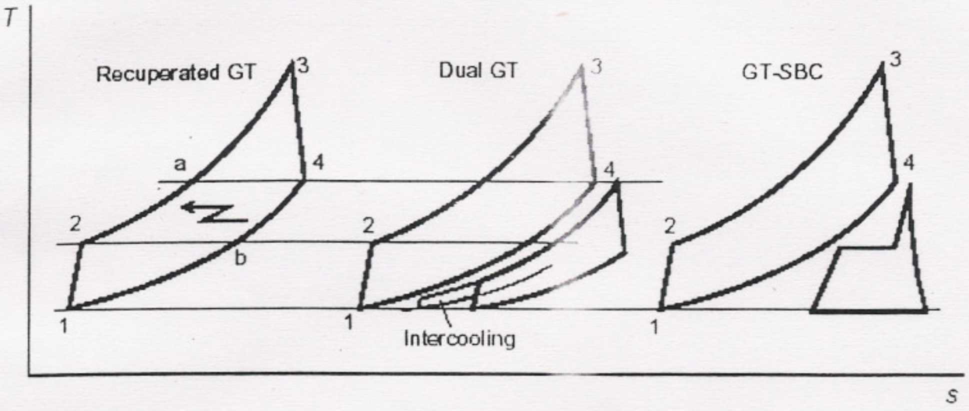
1.6 LOOKING BACK IN TO HISTORY:

G wicks **[16]** derived the concept of the air bottoming cycle from the theory of the ideal fuel-burning engine by comparing the engine with the Carnot cycle. While the Carnot cycle is ideal for heat sources and heat sinks of constant temperature, this is not the case, when considering fuel burning engines. In an ideal fuel-burning engine, the combustion products are created, and present a finite size hot reservoir, that releases heat over the entire temperature range from their maximum to ambient temperature.

When this is put into a T-s diagram, a triangular area can be drawn, instead of the rectangular area in the case of the Carnot cycle. Thus, the cycle consists of isothermal compression, heat addition, and isentropic expansion.

This three-process cycle may not be practical for a single cycle engine because very high-pressure ratio will be required. This concept is more realistic for a heat recovering bottoming cycle. Since the maximum cycle temperature will be the topping cycle exhaust temperature, rather then the topping cycle firing temperature, the corresponding pressure ratio will not be that high. The gas turbine cycle consisting of an adiabatic compressor and an expander, and a heat exchanger can be applied as an air bottoming cycle. The isothermal compression would require continuous cooling during the compression process. This can be approached with an increasing number of intercoolers, or by the introduction of wet compression.

The air bottoming cycle can be compared with both the recuperated gas turbine and the steam bottoming Rankine cycle. As seen on the T-s diagram for the same topping gas turbine **Fig. 1.6**



T-s diagram of the recuperated gas turbine cycle, the dual gas turbine combined cycle, and the gas turbine with steam bottoming cycle.

Figure 1.6

Heat recovery in the case of the recuperated cycle is limited by the compressor outlet temperature and cannot proceed below that temperature level (line 4-h). The dual gas turbine combined cycle has no such limitation, and the use of intercooling can improve the heat recovery even further.

Along with the utilization of a gas turbine exhaust, other sources of heat can be considered: such as waste heat from a chemical reactor, or an incinerator's furnace. For better ABC performance, the heat temperature, however, should exceed 400 °C. The air bottoming cycle was proposed to increase efficiency of the simple-cycle gas turbine units operating on Norwegian oil platforms in view of the C02 tax **[11]**. Converting these units into conventional combined-cycle plants was found not feasible due to the considerable weight of the steam bottoming equipment and the boiler feed water requirements. The same considerations inhibit the use of the steam injected gas turbine (STIG) cycle in off-shore applications **[8]**, whereas the air cycle offers less weight and an efficiency close to that of the SBC.

The dual gas turbine combined cycle can be implemented as a combined heat and power plant by supplying the outlet air flow, which has a temperature above 200 °C, for heating needs. The hot air may be used either directly as a product, or be seen as a heat transfer medium.

1.7 Why Air Bottoming Cycle?

Reduced fuel conjunction can be achieved by using a combined gas turbine / steam turbine. This type of combined cycle has gained widespread acceptance in the land based power generating industry and is definitely a proven technology. Even if such a combined cycle offers a high fuel to power conversion efficiency, the cost due to the high weight of the equipment especially HSRG has been found to be too high. The cost of the weight of equipment is very high on an oil platform; therefore, implementation of fuel reduction initiatives should take place with lightweight equipment.

The **ABC** together with a topping gas turbine cycle is another type of combined cycle. It offers an efficiency close to that of a combined gas turbine / steam turbine cycle, and offers the potentials for lower weight compare to the combined gas turbine / steam turbine cycle. It has been in the past not been paid much attention to, and the main reason for that is probably the rapid technological improvement during the last two decades and the commercial success of the combined gas turbine/ steam turbine cycle **[11]**.

1.8 LITRATURE REVIEW:

The combined cycle has gained wide spread acceptance in the land based power generating industry and is definitely a proven technology. But due to HRSG, the cost has been found to be too high.

As a consequence of high turbine 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 he 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 **[11]**.

The combination of a gas turbine and an ABC represents a high-efficiency CHP 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 compared to the current technology and will have a payback time of 3 yr. The air turbine cycle applied as a heat recovery cycle behind an industrial furnace will convert waste heat to electricity with an efficiency of up to 26% (31% on exergy basis). This result in considerable avoided costs and a payback time of 3–4 yr. 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 powder production and industrial bakeries, and in the glass industry. **[1]**

In this system an air turbine is used to convert the exhaust energy into mechanical power, dispensing with all the hardware relevant to steam power plants (Boiler, Steam Turbine, Condenser, Pumps, Water Treatment Plant, Cooling Towers etc.)**[2]**

Mikhail Korobitsyn, [1], (2002), considered and used this cycle as a compact and simple bottoming cycle in various applications: as an upgrading option for simple cycle gas turbines in the off shore industry; as a hot air cogeneration plant; or as heat recovery installation at high temperature furnaces. The technical & economical feasibility of ABC has also been evaluated; where hot air from the air turbine is supplied to food processing industries. The pay back time described for ABC was 3-4 Years.

J.Kaikko [2], (2001), presented the air bottoming cycle as an economical concept to increase the power generating efficiency of small & medium size gas turbines. A thermodynamic analysis has been presented for cogenerative system where a fraction of compressed air in an intercooled ABC was taken to a Reversed Brayton Cycle to provide cold airflow. System optimization procedure is discussed and potential configurations to implement this system have been investigated. For the selected configuration, characteristics are presented for power, heat and cooling output. For ABC and RBC, sensitivity of the performance is presented against primary cycle parameters.

J.Kaikko et al [3], (2001), made a comparison of thermodynamic performance between the cycle A & ordinary Rankine Cycle. The gas Turbine as a topping cycle was investigated with exhaust temperature 534°C. Two applications have been considered: power generation only, and cogeneration of heat and power. Sensitivity of the performance against primary cycle parameters has also been performed for the cycle.

M.A.Korobitsyn, [7], (1998), analyzed the performance of a dual gas turbine cycle. Energy & Exergy analysis of the dual gas turbine combined cycle with various topping Gas Turbine were also performed. Implementation of ABC at a Gas Turbine adds 7 to 10 points to simple cycle efficiency and rise in power output of 20-35% when compared to steam bottoming cycle, the ABC showed performance value close to an exceeding to those of SBC. The distinct features of this setup were its simpler & robust design, smaller dimension and the absence of water treatment processes.

CHAPTER 2.

ANALYSIS OF GAS TURBINE/AIR BOTTOMING CYCLE

2.1 CYCLE ANALYSIS:

The Combined Gas Turbine/ Air Bottoming Cycle is shown below in **figure 2.1** [11].

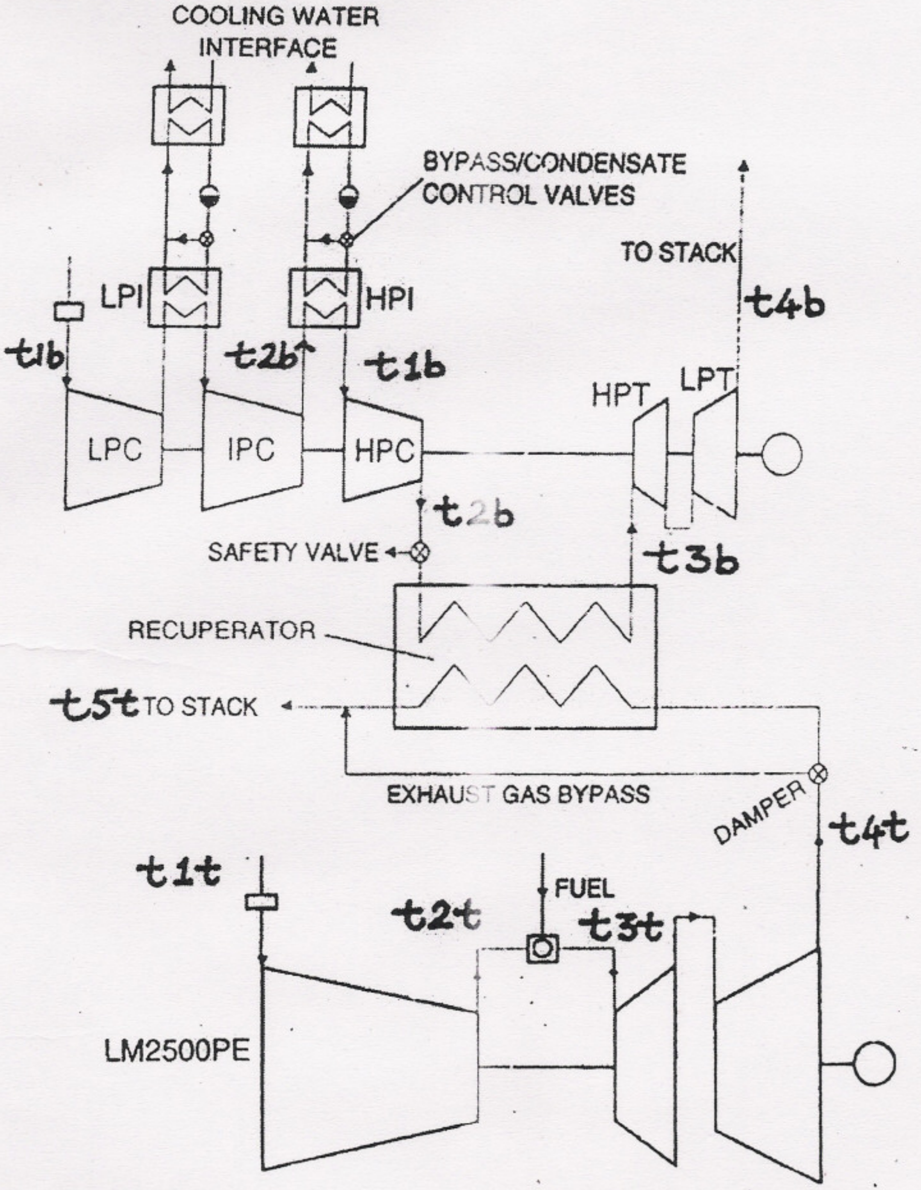


Figure 2.1

The analysis of combined Gas Turbine/Air Bottoming Cycle is shown below in **figure** 2.2.

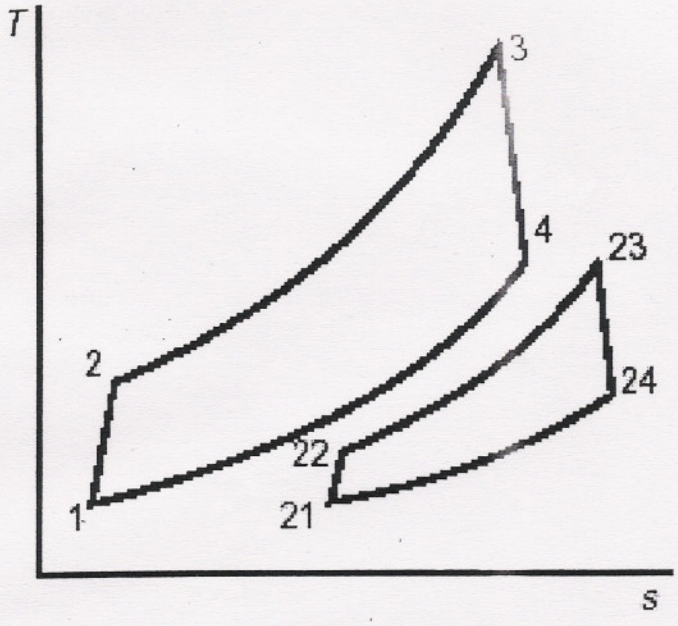


Figure 2.2

The processes are: -

For Topping Cycle (1-2-3-4):

Process 1-2: Ambient Air Compression.

Process 2-3: Heat Addition in Combustion Chamber.

Process 3-4: Expansion of Gas in Gas Turbine.

Process 4-1: Heat Rejection in Heat Exchanger.

For Bottoming Cycle (21-22-23-24):

Process 21-22: Ambient Air Compression.

Process 22-23: Heat Addition in Heat Exchanger.

Process 23-24: Expansion of Air in Air Turbine.

Process 24-21: Rejection of Hot Air.

2.2 COMPUTATIONAL PROCEDURE:

2.2.1 Topping Cycle.

It is assume that t1t = 298 K.

$t2t = t1t*[1+((rp1)^{a}-1)/\eta_c]$……………………(i)

Where rp1=pressure ratio of topping cycle

$a = (\gamma-1)/\gamma$

"η_c" Efficiency of compressor

$t4t = t3t*[1-\eta_t(1-(rp1)^{-b}]$……………………(ii)

where $b = (\gamma'-1)/\gamma'$

"η_t" Efficiency of turbine

Wct = ma * Cpa (t2t-t1t)……………………(iii)

Where 'Wct' compressor work in topping cycle

"Cpa" specific heat of air

Wtt = ma *Cga * (t3t-t4t)……………………(iv)

Where 'Wtt' turbine work in topping cycle

"Cpg" specific heat of gas.

(Wnet) = Wtt-Wtc…………………………(v)

Where '(Wnet)' net work in topping cycle

$q = [mg *cpg * t3t - ma * cpa * t2t]/\eta_{comb}$….. (vi)

Where 'q' heat supplied in combustion chamber in topping cycle

"η_{comb}" Efficiency of combustion chamber.

neta_t = (Wnet)/q……………………………… (vii)

$= (Wtt-Wtc)/ ([mg *cpg * t3t - ma * cpa * t2t]/\eta_{comb})$

2.2.2 Bottoming Cycle:

It is assume that $t1t = 298$ K.

$t2b = t1b*[1+((rp_2)^{(a/(n+1))}-1)/\eta c]$.................................(viii)

Where "rp_2" pressure ratio of bottoming cycle

$t4b = t3b*[1-\eta_t*(1-(rp_2)^{-a}]$...(ix)

$e = (t3b-t2b)/(t4t-t2b)$...(x)

Where "e" is the effectiveness of heat exchanger.

Suppose $x = (mg*Cpg*b*\eta_t*\eta_c*t3b)/(mab*Cpa*a*t1b) = k*t3b$

Where $k= (mg*Cpg*b*\eta_t*\eta_c)/(mab*Cpa*a*t1b)$

And $rp_2 = (x)^{(1/(a+b)}$ $= (k)^{(1/(a+b)}*(t3t)^{(1/(a+b)}$.............(xi)

From equ. (ix) and (x)

$t3b =(1-e)*t1b*[1+[(k)^{(1/(a+b)}*(t3t)^{(1/(a+b)}]/\eta_c]+e*t3t*[1-\eta_t(1-(rp1)^{-b}]$....(xii)

On solving equ. (xii)

$t3b = \alpha+\beta(t3b)^{a/((a+b)*(n+1))}$

Where

$\alpha= (1-e)(1-1/\eta_c)t1b+e*t3t*[1- [1-\eta_t(1-(rp1)^{-b}]]$

$\beta= \{(1-e)*t1b*k^{a/((a+b)*(n+1))}\}/\eta_c$

2.3 EXHAUST GAS TO AIR HEAT EXCHANGER:

The heat of the gas turbine exhaust is recovered and transferred to the compressed air between the compressor and turbine of the ABC. For this application there are two different types of heat exchanger that can be used: regenerators and recuperator. In a recuperator, heat is transferred through walls that separate the flows. In a regenerator, the heat transfer surface is alternately exposed to the two flows. List three advantages that regenerators have over recuperator:

- ❖ A much more compact heat transfer surface can be employed.
- ❖ The heat transfer surface is substantially less expensive per unit of transfer area.
- ❖ Because of the periodic flow reversals, there are no permanent flow-stagnation regions and, consequently, the surface tends to be self-cleaning. **[11]**

A fourth advantage is that regenerators require only light gage containment for the low-pressure exhaust gas over most of their volumes. Recuperator requires heavy-gage containment over their entire volumes **[12]**. Also list two disadvantages of regenerator, compared to recuperator:

1 There is some mixing of the hot and the cold fluids because of leakage and carryover.

2 If the fluids are at different pressures, as in the ABC, the sealing problem of a regenerator is a difficult one.

Additionally one can say that recuperator design for this type of application and size range is a much more proven technology compared to that of regenerators.

Some fluid mixing is acceptable for regenerators in gas turbine systems, however; leakages have been unacceptably high over four percent of the compressor flow in many cases. This is because the seal designs have been poor. This is why recuperator has been favored over regenerators for gas turbine systems. With a proper seal design it is possible to have leakages at about three percent. An air leakage of 3 percent reduces the combined gas turbine and ABC efficiency approximately 1 percentage point.

In the present work it was chosen to go for a recuperator and not a regenerator. The most important reasons for this decision are the efficiency reduction caused by air leakage from a regenerator, and that large recuperator is a more proven technology compared to regenerators for this application.

2.4 INDUSTRIAL APPLICATION OF AIR BOTTOMING CYCLE:

The ABC can be used in cogeneration scheme, where hot air from the air Turbine can be supplied to food processing industries. This is referred as Hot Air Cogeneration (HOT-COGEN). Another application is the heat recovery unit in a glass-melting furnace (Heat Recovery Air Turbine Cycle, HERAC). [1]

CHAPTER 3.

RESULTS AND DISCUSSION

- RESULTS AND DISCUSSION-I
- RESULTS AND DISCUSSION-II
- RESULTS AND DISCUSSION-III

4.1 RESULTS AND DISCUSSION-I

A computer program have been generated for the analysis of:

1. Net work of Topping Cycle, Wnett
2. Thermal efficiency of Topping Cycle, neta_t
3. Net work of Bottoming Cycle, Wnetb
4. Net work of Combined Cycle, Wnet
5. Thermal efficiency of Combined Cycle, neta_net
6. Specific fuel consumption of Combined cycle, sfc.

The following variables have been varied.

1. Turbine inlet temperature of topping cycle,t3t
2. Mass flow rate of Air bottoming cycle. mab.
3. Number of intercooler, n.
4. Pressure ratio of topping cycle. rp1.

Assumptions:

Mass flow rate of air in topping cycle, ma = 69kg/sec.

Mass flow rate of fuel, m_f = 1.32kg/sec.

Efficiency of combustion chamber $\eta_{comb.}$ = 0.96

Efficiency of Compressor η_c = 0.91

Efficiency of Turbine η_t = 0.92

Effectiveness of heat exchanger, e = 0.9 to 0.7

Specific heat of air Cpa = 1.005kj/kg k

Specific heat of gas Cpa = 1.14 kj/kg k

Isentropic index of air,γ = 1.4

Isentropic index of air,γ = 1.33

Inlet temperature to compressor, t1t, t1b = 298 K.

Discussion on Results:

a) Efficiency of Topping Cycle:

- The efficiency of topping cycle increases with increase in the turbine inlet temperature, t3t for every particular value of pressure ratio of topping cycle (rp1) of ABC.

- The efficiency of topping cycle increases with increase in with pressure ratio of topping cycle (rp1) of ABC for any particular value of turbine inlet temperature of ABC

- The efficiency of topping cycle is independent of effectiveness of heat exchanger as well as mass flow rate of ABC.

b) Network of Topping Cycle:

- The network of topping cycle increases with increase in the turbine inlet temperature, t3t for every particular value of pressure ratio of topping cycle (rp1) of ABC.

- The network of topping cycle increases with increase in with pressure ratio of topping cycle (rp1) of ABC for any particular value of turbine inlet temperature of ABC

- The network of topping cycle is independent of effectiveness of heat exchanger as well as mass flow rate of ABC.

a) **Heat supplied of Topping Cycle:**

- The heat supplied in combustion chamber of topping cycle increases with increase in the turbine inlet temperature, t3t for every particular value of pressure ratio of topping cycle (rp1) of ABC.
- The heat supplied in combustion chamber of topping cycle increases with increase in with pressure ratio of topping cycle (rp1) of ABC for any particular value of turbine inlet temperature of ABC
- The heat supplied in combustion chamber of topping cycle is independent of effectiveness of heat exchanger as well as mass flow rate of ABC.