Study the Performance of the Combined Gas Turbine-Steam Cycle for Power Generation

Dr. Saba Yassoub Ahmed
University of Babylon/College of Engineering/Department of Mechanical/Iraq-Babylon
e-mail:saba_ya@yahoo.com

Abstract
In this research, a theoretical and analytical study of gas turbine-steam turbine combined cycle for power generation was carried out to improve the performance of the thermal power plant.
A thermodynamic analysis of the combined cycle through a typical pattern was done. Appropriate assumptions were made for two kinds of combined cycles, simple gas turbine-steam turbine and reheat gas turbine-steam turbine combined cycles, such that the results obtained would closely approximate actual data. Then the equations of the combined cycle calculations were derived. Four cases were studied to choose the optimum gas turbine and steam turbine conditions for the combined cycle.
The reheat gas turbine-steam turbine combined cycle is better as compared to the simple gas turbine-steam turbine combined cycle, because the output per unit mass of air flow is significantly improved by 35-41%, and the efficiency is higher by 4 % for the combined cycle utilizing the reheat gas turbine, which points to potential cost saving for such a cycle.
It was found that the optimum conditions of the gas turbine cycle in the combined cycle mode occur at the maximum net work conditions. The efficiency of the waste heat boiler for high exhaust temperature ranges (maximum η_B= 87%) is comparable to the efficiency of the conventional boilers. The waste heat boiler has the added advantage of simplicity, therefore this type of combined cycle using waste heat boiler has great promise for power generation.
Keywords: *Combined gas-steam cycle, gas turbine cycle*

List of Symbols:
Latin Symbols

Symbol	***Definition***
CL	Combustion pressure Loss
Cp	Specific heat at Constant pressure (kJ/kg.K)
f	Fuel to Air Ratio (kg_{fuel}/kg_{air})
h	Enthalpy (kJ/kg)
L	Combustion and Radiation Loss
m	Mass flow rate (kg/sec)
P	Pressure (Pa)
Q	Heat Supplied (KJ/Kg)
rp	Pressure Ratio
T	Temperature(K)
W	Work output (W)
Greek Symbols	
ΔH_{25}	Calorific Value at 25^{o}C
η	Efficiency
Subscript	
1,2,3,4,5,6	The state as shown in Figures(1,2,3,4)
2s,4s,6s	Isentropic case for each state
a	Air
Add	Added
B	Boiler
C	Compressor
Comb	Combined
eg	Exhaust gas
g	Gas
o	Overall
s	Steam
sg	Stack gases
sup	Superheated

T Turbine

1. Introduction

Electrical power generation has undergone a revolution since ***1990.*** The gas-steam combined power plant has come of age and there is the promise of more advanced gas turbine cycles based on aero-derivative machines. During the *1990s,* a suite of computer codes were developed at Cambridge top analyzes advanced power cycles at a more detailed level than usual. For such calculations, it is most important to handle the thermodynamics as rigorously as possible *(Horlock, J.H., et.al., 1998, 2000).*

In recent past the gas/steam combined cycle based power plants have become popular as these offer more effective utilization of the fossil fuel energy. These offer higher thermal efficiency as compared to the gas turbine based plant or the steam turbine based plants in isolation the performance of the gas/steam combined cycle power plant depends upon the performance of topping and bottoming cycle. Gas turbine is seen to offer high specific work output if the turbine inlet temperature *(TIT)* could be increased. Thus, with increased *TIT* the performance of the heat recovery steam generator *(HRSG)* and consecutively the steam turbine improves thereby, offering improvement in combined cycle performance *(Yadav, J.P., Singh,Dr O.,2006)*.

The literature has often suggested combining two or more thermal cycles within a single power plant. In all cases the intention was to increase efficiency over that of single cycles. Thermal processes can be combined in this way whether they operate with the same or with differing working media. However, a combination of cycles with different working media is more interesting because their advantage can complement one another.

Normally the cycles can be classed as a *"topping"* and a *"bottoming"* cycle. The first cycle to which most of the heat is supplied is called the *"topping cycle"*. The waste heat produces is then utilized in a second process which operates at a lower temperature level and is therefore referred to as a *"bottoming cycle"*. Up to the present time , only one combined cycle has found wide acceptance : the combination gas turbine/steam turbine power plant *(Kehlhofer, R., 1997)*.

An alternative regenerator configuration has been improved by *(Dallenback,P.A.,2002)* through improving the efficiency of gas turbine cycle.*(Ravi Kumer, et.al.,2005)* has energy losses in different gas turbine cycle components. It is observed that the irreversibility in exhaust gases is low which indicates effective utilization of heat energy, but the specific work output of the turbine decreases.

Generally with increase the turbine inlet temperature the specific work of gas turbine increases. But increase in the turbine inlet temperature has strict metallurgical limitations in terms of maximum temperature that the turbine stage could with stand. *(Ravi Kumar, N., Sita Rama, A.V., 2005)* analyzed the effect of inlet cooling on heat recovery steam generator *(HRSG)* performance. It is found that the inlet cooling reduces the work input of the compressor and increases the mass flow rate of air.

Similarly the efficiency of steam cycle can be improved by increasing the temperature of steam entered into the steam turbine. The maximum temperature of steam that can be used in steam turbine is considered from metallurgical point of view of turbine blades. The mass flow rate of steam and steam temperature depends on the amount of heat available in the gas turbine exhaust. *(Ravi Kumar, et.al., 2006)* has done performance simulation of *(HRSG)* in combined cycle power plant. They discussed the effect of various parameters like pinch point, approach point, steam pressure, steam temperature, gas flow rate on the performance of the *(HRSG)*.

(Ravi Kumar, et.al., 2007) have been analyzed different heat recovery steam generator configurations of single pressure and dual pressure. The combined cycle efficiency with different heat recovery steam generator configurations have been analyzed parametrically by using first law & second law of thermodynamics, it was observed that in the dual cycle high pressure steam turbine pressure must be high & low pressure steam turbine pressure must be low for better heat recovery from heat recovery steam generator. *(Srinivas, T., et.al., 2007)* have been analyzed the effect of 'n' feed water heaters (flow) on performance of a steam power cycle with a generalized mathematical formulation. The performance calculations were formulated separately to single *f.w.h* and extended to *'n' f.w.hs* for parametric study. The optimum bled steam temperature ratio is found at *0.4* with single *f.w.h* at given working conditions. Similarly the optimum pressure in a steam reheated is obtained at *20–25%* of the boiler pressure irrespective of the number of heaters. The results show that the maximum gain in the efficiency of cycle is obtaining with the first *fwh* and the increment diminishes with the addition of the number of heaters. Also, has been examined the improvements in efficiency with increases in boiler pressure, turbine inlet temperature and furnace temperature.

2-Calculations of Gas Turbine Cycle

2-1:Calculations of Simple Gas Turbine Cycle

A schematic diagram, ***Figure (1)***, represents the simple gas turbine cycle and ***Figure (2)*** represent the cycle on ***T-S*** diagram. Air at atmospheric pressure is compressed from state ***1*** to state ***2*** by the compressor. Fuel is added in the combustion chamber to increase the temperature at a constant pressure to state ***3***. The hot gases are then expanded through the turbine to state ***4*** (atmospheric pressure).

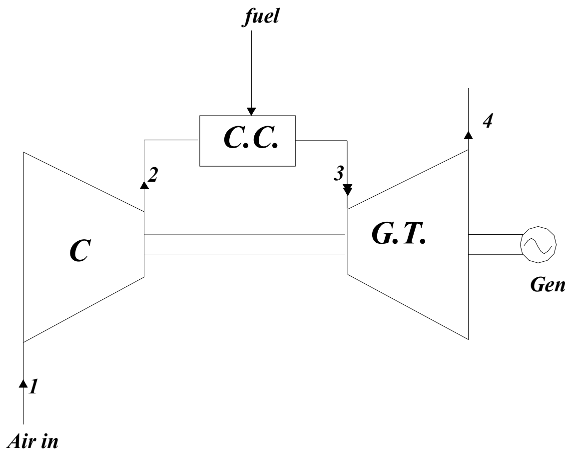


Figure (1): Schematic Flow Diagram of Simple Gas Turbine Cycle.

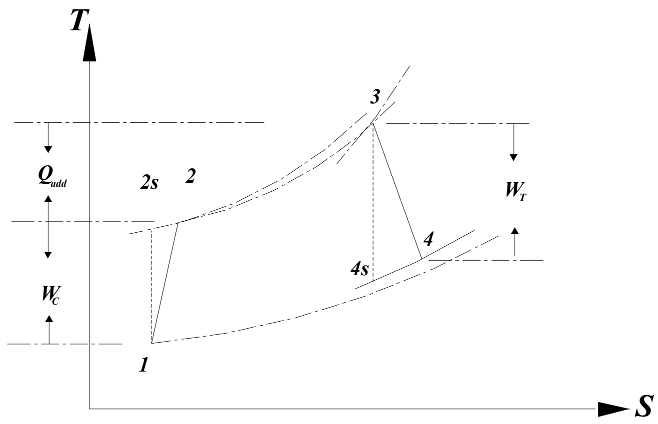


Figure (2): T-S Diagram of Simple Gas Turbine Cycle.

2-1-1: Assumption of Simple Gas Turbine Cycle:

Appropriate assumptions have been made for simple gas turbine cycle such that the results obtained will closely approximate actual data from known gas turbines. The assumptions are:

1- Compressor efficiency =***85%.***
2- Turbine efficiency=***87%***.
3- Combustion pressure loss=***4 %***.
4- Combustion and radiation loss=***3%***.
5- Air leakage=***0.5%.***
6- Inlet pressure=***101.325 Pa.***
7- Exhaust pressure=***102.73 Pa***
8- Inlet air temperature=***15 ^oC***.
9- Calorific value of fuel (CH_4)=***45500 kJ/kg***.
10- At constant A/F ratio set the excess air to ***200,300,*** and ***400%***.
11- For constant maximum temperature, the excess air set to ***400%***.

2-1-2: Analysis of Simple Gas Turbine Cycle:

$$P_2 = rp * P_1 \quad ...(1)$$

$$T_{2s} = T_1 (rp)^{\frac{(\gamma_a - 1)}{\gamma_a}} \quad ...(2)$$

The actual temperature of air at the end of compression is given by:

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

Hence work required for compression is obtained by using:

$$W_C = \int_1^2 Cp_{air} dT \quad ...(4)$$

Where the specific heat of air is given by *(Yadav,J.H., Singh, O.,2006):*

$$Cp_{air} = (28.11 + 0.1967.10^{-2} T + 0.4802.10^{-5} T^2 + 1.966.10^{-9} T^3) / 29.0$$

Similarly, the pressure after the combustion chamber and the temperature of the combustion can be calculated as follow:

$$P_3 = P_2 (1 - CL) \quad ...(5)$$

While the maximum temperature (temperature of combustion) can be calculated from the energy balance:

$$\begin{aligned} &(1 + f)(1 + \dot{m}_{aL}) Cp_{gas} (T_3 - 298) + f(1 - L)(1 - m_{aL}) \Delta H_{25} = \\ &(1 - m_{aL}) Cp_{air} (T_2 - 298) + f(1 - m_{aL}) Cp_{fuelir} (T_f - 298) \end{aligned} \quad ...(6)$$

And after expansion the temperature of the exhaust gas can be calculated as follow:

$$T_{4s} = T_3 .(1 / rp)^{\left(\frac{\gamma_{gas} - 1}{\gamma_{gas}}\right)} \quad ...(7)$$

In addition, the actual temperature:

$$T_4 = T_3 - \eta_T .(T_3 - T_{4s}) \quad ...(8)$$

Then, the work output from the gas turbine is:

$$W_T = \int_4^3 Cp_{gas} dT \quad ...(9)$$

Where Cp_{gas} is given by *(Yadav,J.H., Singh, O.,2006)*:

$$Cp_{gas} = 1.8083 - 2.3127.10^{-3} T + 4.045.10^{-6} T^2 - 1.7363.10^{-9} T^3$$

Hence the net work output is:

$$W_{net} = W_T - W_C \quad ...(10)$$

And the gas turbine cycle efficiency is:

$$\eta_{gast\ turbine} = \frac{W_{net}}{Q_{add}} \quad ...(11)$$

Where Q_{add} is the heat supply to the gas turbine and can be calculated as follows:

$$Q_{add} = f(1 - L)(1 - \dot{m}_{aL}) \Delta H \quad ...(12)$$

Or equal to

$$Q_{add} = (1 + f)(1 - \dot{m}_{aL}) Cp_{gas} T_3 - (1 - \dot{m}_{aL}) Cp_{air} T_2 \quad ...(13)$$

2-2: Calculations of the Reheat Gas Turbine Cycle:

The T-S diagram, ***Figure (3)***, represents the reheat gas turbine cycle. Air at atmospheric pressure is compressed by the compressor from state ***1*** to state ***2*** as in the previous simple gas turbine where the fuel is added in the first combustion chamber to increase the temperature at a constant pressure to state ***3***. the hot gases are then expanded to state ***4***, where again fuel is added in the second combustion chamber to heat the gases to state ***5*** (full reheat). Then, the hot gases are expanded through the second gas turbine to state ***6*** (atmospheric pressure). A schematic diagram of the reheat gas turbine represented in ***Figure (4)***, where the numbers on the schematic correspond with the numbers of the ***T-S*** diagram. The fuel added to the first combustion chamber is f_1 and for the second combustion chamber f_2.

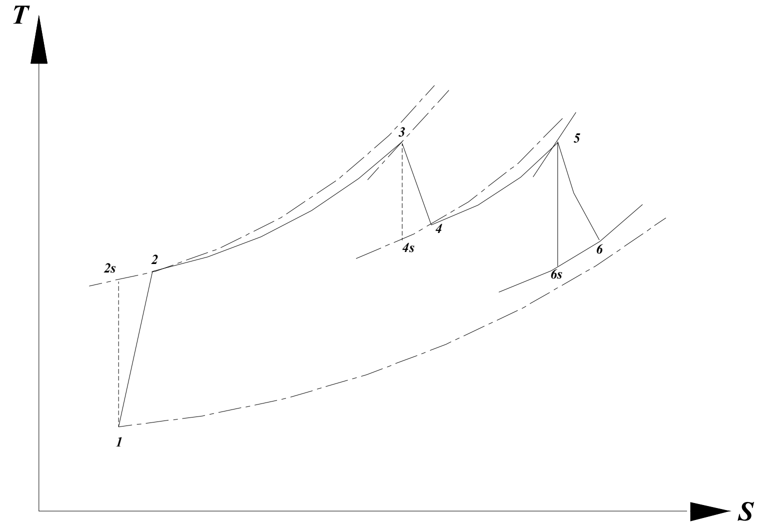


Figure (3): T-S Diagram of Reheat Gas Turbine Cycle.

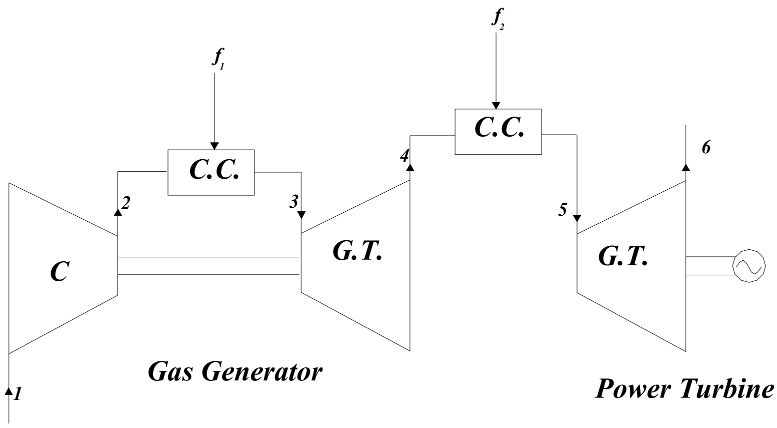


Figure (4): Schematic Diagram of Reheat Gas Turbine Cycle.

2-2-1: Assumption of Reheat Gas Turbine Cycle:

In reheat gas turbine cycle, the same assumptions were made for the simple gas turbine cycle, in addition to:

1- The second combustion pressure loss=***3%***.
2- The isentropic efficiency of the second gas turbine also was equal to =***87%***.
3- The excess air for the first gas turbine was set to ***400 %*** and ***300 %*** in the second gas turbine for expansions.

2-2-2: Analysis of Reheat Gas Turbine Cycle:

The properties of points ***1,2***, and ***3*** are the same as in the simple cycle.

Notes: the points correspond with the points of ***Figure (3).***

The actual temperature of point ***4*** is calculation from ***equation (14)*** below:

$$W_C = W_{T1}$$
$$Cp_{air}(T_2 - T_1) = (1 - \dot{m}_{aL})(1 + f_1)Cp_{ga}(T_3 - T_4) \qquad ...(14)$$

The isentropic temperature at point 4:

$$T_{4s} = \frac{(T_3 - T_4)}{\eta_T} \qquad ...(15)$$

The pressure at point ***4***

$$T_{4s} = T_3.(1/rp)^{\left(\frac{\gamma gas - 1}{\gamma gas}\right)}$$

$$\frac{P_3}{P_4} = \left(\frac{T_3}{T_4}\right)^{\frac{\gamma gas}{\gamma gas - 1}} \qquad ...(16)$$

Then the pressure at point 5 can be calculated as shown below:

$$P_5 = P_4(1 - CL_2) \qquad ...(17)$$

Where CL_2-Second combustion pressure loss percentage

Since P_6 is known from the assumption of the simple gas turbine cycle for P_4 then we must find the ratio of (P_5/P_6) in order to find the temperature at point 6 as follow:

$$\frac{T_5}{T_{6s}} = (P_5 / P_6)^{\left(\frac{\gamma gas - 1}{\gamma gas}\right)} \qquad ...(18)$$

Then, the actual temperature at point ***6***:

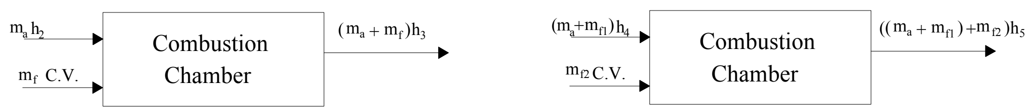
$$T_6 = T_5 - \eta_T.(T_5 - T_{6s}) \qquad ...(19)$$

Hence the net work output

$$W_{net} = W_{T1} = (1 - \dot{m}_{aL})(1 + f_1 + f_2)Cp_{gas}(T_5 - T_6) \qquad ...(20)$$

And the heat supplied is calculated as follow according to mass and energy flow diagram ***Figure (5b)*** for the second combustion chamber:

$$Q_{add} = \big((1 + f_1)(1 - \dot{m}_{aL})Cp_{gas}T_3 - (1 - \dot{m}_{aL})Cp_{air}T_2\big) +$$
$$((1 - \dot{m}_{aL})(1 + f_1 + f_2)Cp_{gas}T_5 - (1 - \dot{m}_{aL})(1 + f_1)Cp_{ga}T_4 \qquad ...(21)$$



(a) (b)

Figure (5): Mass and Energy Flow Diagram for the First and Second Combustion Chamber.

2-3: Calculations and Analysis of the combined cycle:

A schematic diagram of the combined cycle is illustrated in ***Figure (6)***. The heat available in the exhaust of the gas turbine is illustrated to generate steam by the boiler and the steam dose work through the expansion in the steam turbine. The steam turbine is mechanically connected to the generator where the cycle is a regenerative with three forward flow feed water heaters as shown in the T-S diagram ***(Figure(7)).*** The condensate steam is pumped by the feed water pumps and is heated by feed water heaters which take the bleed steam from the turbine at defined pressure. Then heat addition takes place in the economizer, evaporator and superheater. The superheated steam expands through the turbine to generate work and then the cycle is completed.

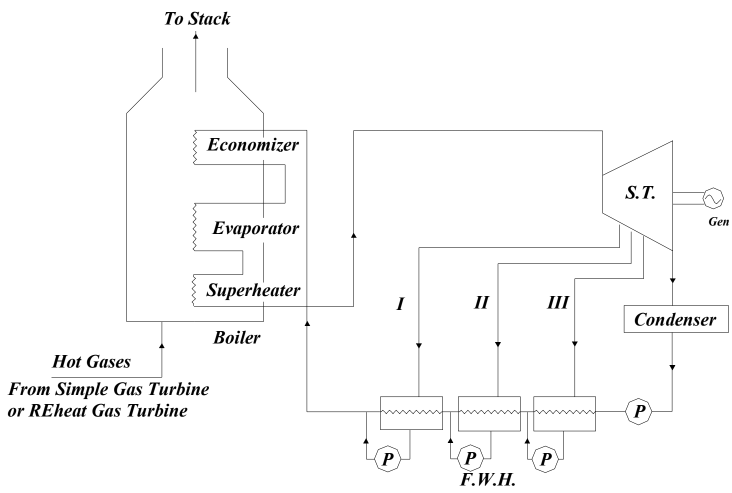


Figure (6): Schematic Flow Diagram of Steam Cycle.

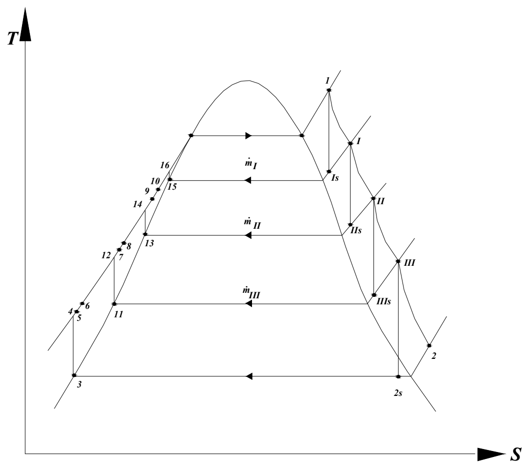


Figure (7): T-S Diagram of the Steam Cycle.

2-3-1: Assumptions of the Two Combined Cycle:

Two combined cycles were studied. The first was the simple gas turbine-steam turbine combined cycle and the second was the reheat gas turbine-steam turbine combined cycle.

The conditions and assumptions of the two combined cycles are given in table below:

Table (1): The Conditions and Assumptions of the Steam Cycle.

The conditions and assumptions	Value used in the combined cycle with	
	Simple gas turbine	Reheat gas turbine
1- feed water temp,^{o}C	121	121
2-initial steam pressure Kpa	4238	5961
3-initial temp,^{o}C	440.5	482.2
4-Condenser pressure, atm	0.05	0.05
5-Steam turbine expansion efficiency %	80	80
6-pressure loss in extraction piping %	5	5
7-Terminal difference for each feed water heater ^{o}C	2.78	2.78

2-3-2: Calculations of the Combined Cycles:

The applicable equations used in combined cycle calculations are the heat balance between the exhaust of the gas turbine and the boiler (***Holman, J.P.,2002)***:

$$\frac{mass\ of\ steam}{mass\ of\ air} = \frac{h_{eg} - h_{sg}}{h_{sup} - h_{fwh}} \quad \ldots(22)$$

To determine the mass of the bleed steam for example, consider the adiabatic mixing process at the first feed heater, in which $\dot{m}_I$ kg of steam of enthalpy h_I, mix with $(\dot{m}_s - \dot{m}_I)$ kg of water of enthalpy h_8, to give $\dot{m}_S$ kg of water of enthalpy h_{10}:

$$\dot{m}_I(h_I - h_8) = \dot{m}_s(h_{10} - h_8) \quad \ldots(23)$$

Repeated for the second and third feed water heater to find the steam work output:

$$W_S = \dot{m}_s(h_1 - h_I) + (\dot{m}_s - \dot{m}_I)(h_I - h_{II}) + (\dot{m}_s - \dot{m}_I - \dot{m}_{II}) \\ (h_{II} - h_{III}) + (\dot{m}_s - \dot{m}_I - \dot{m}_{II} - \dot{m}_{III})(h_{III} - h_c) \quad \ldots(24)$$

The steam efficiency:

$$\eta_S = \frac{W_S}{Q_S} * 100\% \quad \ldots(25)$$

$$Q_S = h_1 - h_{FWH}$$

$$Q_S\ per\ kg\ of\ air = Q_S * \frac{kg\ of\ steam}{kg\ of\ air} \quad \ldots(26)$$

$$\eta_B = \frac{Q_S\ per\ kg\ of\ air}{\left(h_{eg} - h_{sg_{at\ 15.73^o C}}\right)} \quad \ldots(27)$$

$$\eta_o = \frac{W_S\ per\ kg\ of\ air}{\left(h_{eg} - h_{sg_{at\ 15.73^o C}}\right)} \quad \ldots(28)$$

$$\eta_o = \eta_S * \eta \quad \ldots(29)$$

$$W_{comb} = W_S + W_{net} \quad \ldots(30)$$

$$Capacity\ \ Ratio = \frac{W_{net}}{W_D} \quad \ldots(31)$$

Q_{add} for Combined Cycle = Q_{add} for Gas Turbine Cycle

$$\eta_{comb} = \frac{W_{comb}}{Q_{add}} * 100\% \quad \ldots(32)$$

2-3-3: Steam Generation by Waste heat boiler:

The process of steam generation is given schematically in ***Figure (8)***. Water enters the economizer at ***121°C*** on the left side and is heated to the saturation temperature. Then, the water is boiled at a constant temperature to ***100 %*** quality in the evaporator where after the saturated steam superheated to a certain predetermined temperature. The gas turbine exhaust gives up its heat to steam/water until the "pinch point" is reached (with such boilers the pinch point is generally ***17 to 28°C*** for economical designs)***(Rice, I.C., 1980)***. The pinch point then determines the stack or boiler exit temperature.

22°C was chosen as a pinch point of the combined cycles calculations, but for the last two cases of ***1477K*** and ***1588K*** in reheat gas turbine-steam turbine combined cycle, if the pinch point at the evaporator is kept the same as for the other cases, the difference between the stack temperature and the feed water temperature (the pinch point at the economizer) becomes very small in the range of ***0 to 8 °C***. This would obviously lead to a very large economizer, which would be uneconomical and therefore not desirable. In order to avoid this, the pinch point at the economizer was made as ***22 °C*** and for this case the pinch point at evaporator comes to be ***34°C*** and ***52°C*** between the water/ steam circuit and the gas circuit in the waste heat boiler.

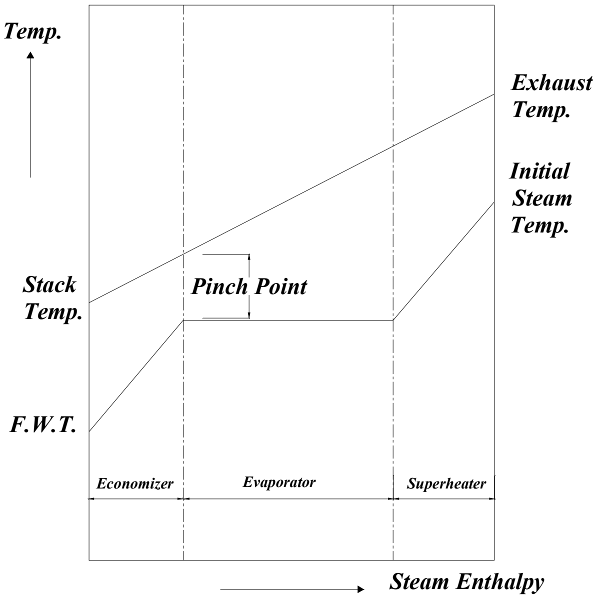


Figure (8): Enthalpy-Temperature Diagram Showing the Process of Steam Generation.

2-3-4: Anlysis the Optimum Condition of the Simple and Reheat Gas Turbine-Steam Turbine Combined Cycle:

To choose the optimum steam conditions and gas turbine conditions of the combined cycle, four cases are studied.

The cases are

1-Gas turbine conditions at the maximum net work and a certain steam conditions (as in ***Table (1)***) with assumption that:

Enthalpy increase in economizer=enthalpy increase in feed water heaters

2-The same conditions as in ***case (1)*** except the final feed water temperature=***121°C***.

3-The same conditions as in ***case (2)*** except initial steam temperature=***371°C***.

4-Gas turbine conditions at the maximum cycle efficiency and the same steam conditions as in ***case (2).***

From these four cases, ***case (2)*** seems better than the other cases and suitable, because it gives more net work and more efficiency, so all the combined cycle calculations were done based on the ***case (2).***

The same basis of calculations of the above was followed in the calculations of the reheat gas turbine-steam turbine combined cycle.

3-Results and Discussion:

3-1: Results of Simple Gas Turbine Cycle:

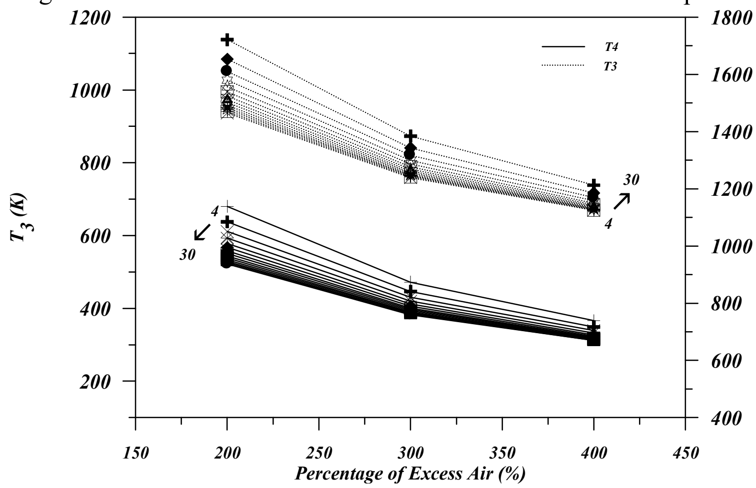
To give a clear picture of the simple gas turbine cycle, two sets of calculations have been made for constant A/F ratios and for constant maximum temperatures respectively.

3-1-1: Constant A/F Ratio:

In this set of calculations three different theoretical air percentage were chosen ***200%, 300%, and 400%.***

Figure (9) shows a plot of the theoretical air percentage versus the maximum temperature ***(T_3)*** and the exhaust temperature ***(T_4)*** for various pressure ratios. It is clear that for a constant theoretical air percentage the maximum temperature increases when pressure ratio increases, because the outlet compressor temperature increases. The maximum temperature decreases with the increase of the theoretical air percentage at a constant pressure ratio, because the amount of the burnt fuel per unit mass of air becomes less and therefore the enthalpy of the products becomes less, so that the maximum temperature decreases.

The exhaust temperature decreases when the pressure ratio increases at a constant theoretical air percentage, because of greater expansion, thus the exhaust temperature behaves opposite to the maximum temperature with the pressure ratio. For a given pressure ratio, the exhaust temperature decreases as the theoretical air percentage increases. This behavior is the same as the maximum temperature.



Figure(9): Maximum Temperature and Exhaust Temperature Versus Theoretical Air Percentage.

Figure (10) represents the relationship between the net work and the cycle efficiency as a function of pressure ratio for various theoretical air percentages. It can be shown that at a constant pressure ratio the cycle efficiency and the net work increases when the theoretical air percentage decreases, due to high maximum temperature which is reached when more fuel is burnt.

For a constant theoretical air percentage, cycle efficiency and net work increase with the increase of pressure ratio. These increases becomes less and less at high-pressure ratios, because more compressor work is needed.

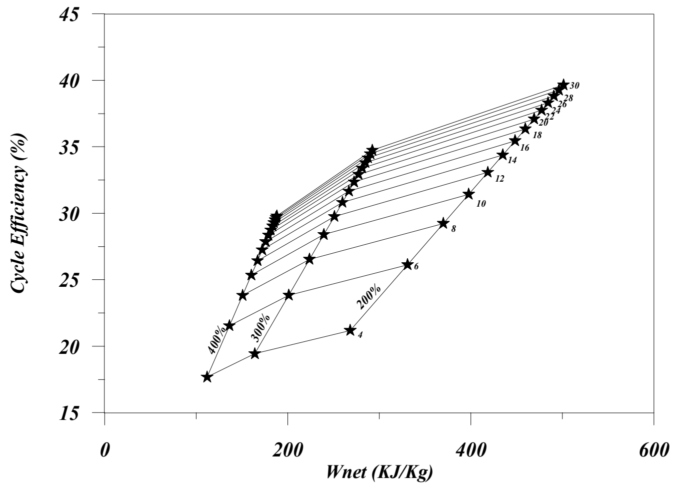


Figure (10):Cycle Efficiency Versus Net Work.

Figure (11) shows a plot of the exhaust temperature and the outlet compressor temperature ***(T_2)*** versus pressure ratio. The outlet compressor temperature increases with the increase of pressure ratio, due to more compressor work for high pressure ratio and that work makes the outlet compressor temperature increase. It is clear also that the compressor outlet temperature is independent of the theoretical air percentage.

The figure also shows that for a given pressure ratio, the exhaust temperature increases with the decrease of the theoretical air percentage, because the maximum temperature is more due to more burnt fuel.

For a given percentage of the theoretical air, the exhaust temperature decreases with the increase of the pressure ratio, because of greater expansion.

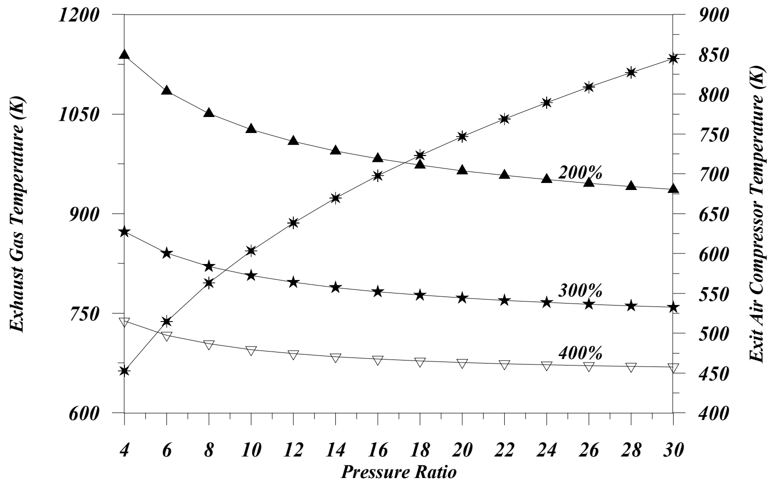


Figure (11): Exhaust Temperature and Compressor Outlet Temperature versus PR.

Figure (12) represents the relation between the work ratio and the pressure ratio for various excess air percentage. For a given theoretical air percentage, the work ratio decreases as pressure ratio increases, due to more work is needed for the compressor. At a constant pressure ratio, the work ratio increases as the theoretical air percentage decreases, because more work is produced, due to high maximum cycle temperature.

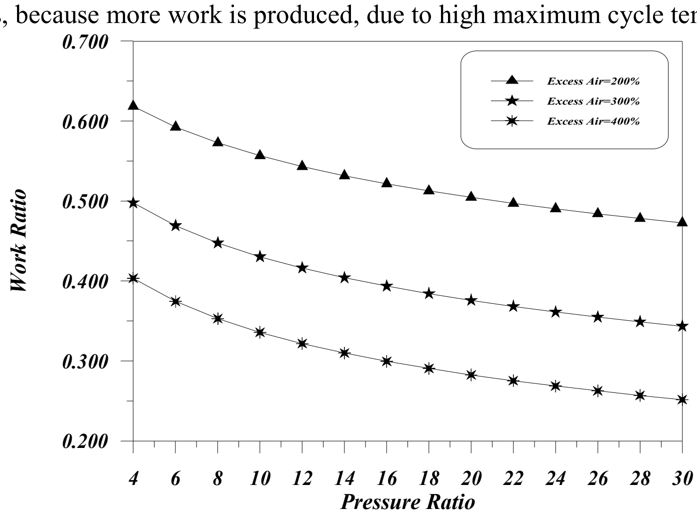


Figure (12): Work Ratio versus PR.

3-1-2: Constant Maximum Temperature:

Because the maximum temperature (firing temperature) is very important in the gas turbine cycle, so another set of calculation has been made for the constant maximum temperature.

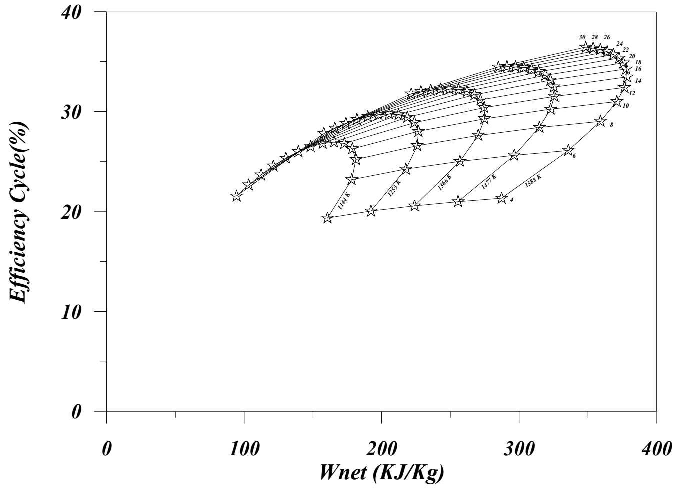
Appropriate maximum temperature used in today's and future turbines were selected to know the performance of the cycle at these temperatures, which are ***1144, 1255, 1366, 1477, and 1588 K.*** in practice, the maximum temperature is not exceeded ***1255 K*** in today's turbine.

Table (2) gives the optimum pressure ratios at which the net work and the cycle efficiency are maximum.

Table (2): Optimum Pressure Ratios for Maximum Cycle Efficiency and Maximum Net Work.

Max. Temp. K	Optimum Pressure Ratios for maximum	
	Cycle Efficiency	Net Work
1144	***14***	***8***
1255	***18***	***10***
1366	***22***	***12***
1477	***24***	***14***
1588	***30***	***16***

Figure (13) shows the relation between the cycle efficiency and the net work as a function of the pressure ratio for various maximum temperatures. It is clear from the plot that for a given maximum cycle efficiency is higher than the optimum pressure ratio for maximum net work, because at high pressure ratio the heat added becomes less which makes the efficiency increase until a certain pressure ratio, then it becomes decreases, because the net work becomes very small. At a constant pressure ratio the cycle efficiency and the net work increases as the maximum temperature increases, because greater expansion is possible.



Figure(13):Cycle Efficiency versus Net Work.

Figure (14) shows the relation between the exhaust temperature and the pressure ratio for various maximum temperature. The exhaust temperature is an important parameter when dealing with any type of heat recovery cycle. It is clear that at a constant pressure ratio, the exhaust temperature increases with the increase of the maximum temperature. At a given maximum temperature, the exhaust temperature decreases as the pressure ratio increases because of greater expansion.

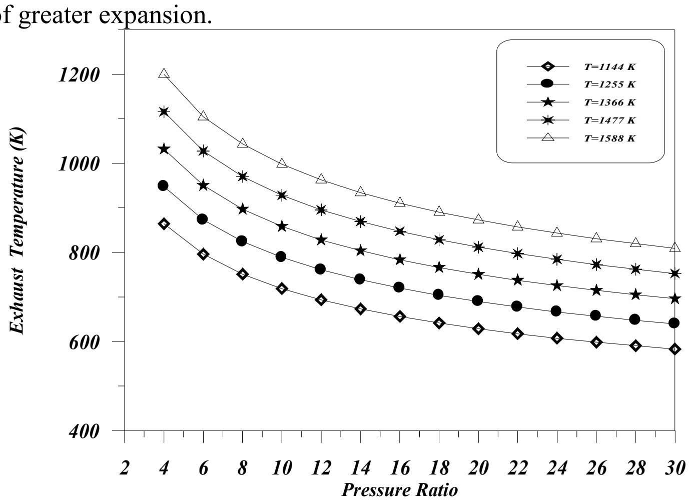


Figure (14):Exhaust Temperature versus PR

Figure (15) shows a plot of the work ratio versus the pressure ratio for various maximum temperatures. It is clear that the work ratio increases as the maximum temperature increases at a constant pressure ratio, because more net work is produced at high maximum temperatures. At a constant maximum temperature, the work ratio decreases as the pressure ratio increases, because more work is needed for the compressor.

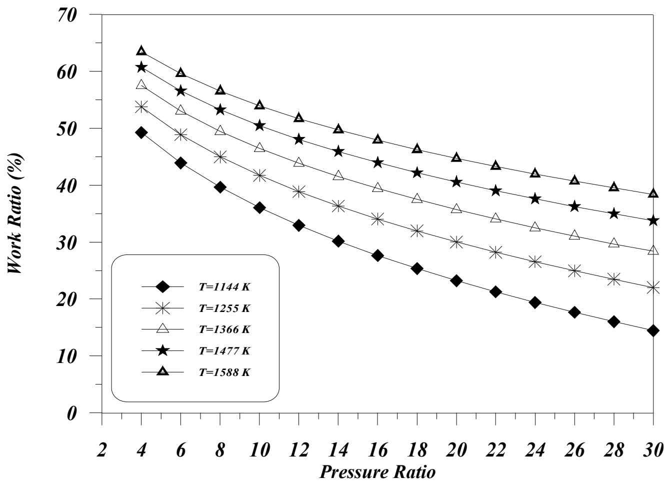


Figure (15):Work Ratio versus PR.

3-2: Results of Reheat Gas Turbine Cycle:

Two sets of calculations have been made for the reheat gas turbine cycle.

3-2-1: Constant A/F Ratio:

In this set of calculations only the ***400%*** of theoretical air was chosen.

Figure(16) shows a plot of the net work versus the pressure ratio. It can be shown that the net work increases when the pressure ratio increases, because the maximum temperature is increased due to increase of the outlet compressor temperature, the increases become less and less as the pressure ratio increases, because more work is needed for compressor.

Also it can be shown on the same ***Figure(16)*** that the cycle efficiency increases as the pressure ratio increase, because more net work is produced, but these increases become less and less, because the net work becomes less and less as more work is needed for the compressor.

The work ratio decreases as the pressure ratio increases, because more work is needed for the compressor as shown in ***Figure (16)***. The maximum temperature increases as the pressure ratio increases due to increase of the outlet compressor temperature.

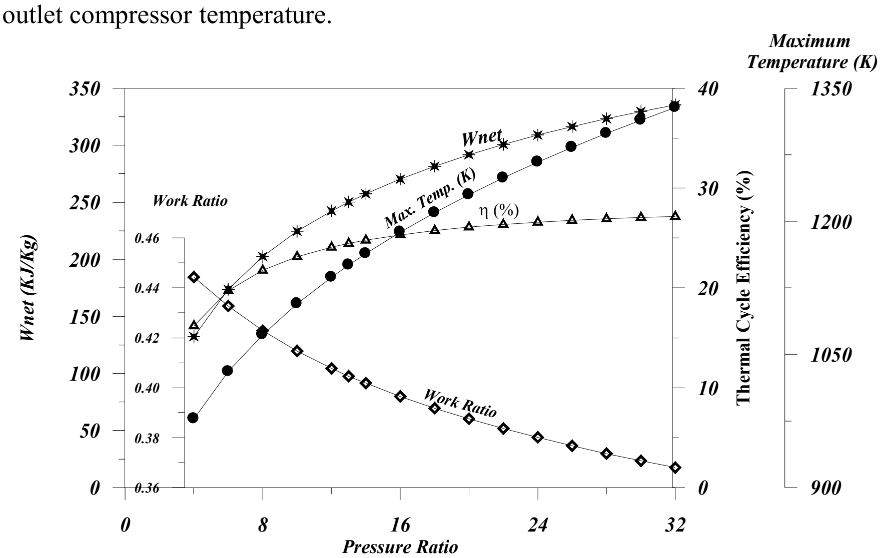


Figure (16):Cycle Efficiency, Net Work, work Ratio and Maximum Temperature versus PR.

While the ***Figure (17)*** shows that the exit temperature from the first gas turbine decreases as the pressure ratio increases, because greater expansion occurs and the exhaust temperature from the second gas turbine increases as the pressure ratio increases, because the maximum temperature of the second gas turbine increases as the pressure ratio increases which leads to increase in the exhaust temperature.

On the same Figure, it can be shown that the power turbine expansion ratio ***(P_5/P_6)*** increases as the pressure ratio increases, because the maximum temperature increases which is the most parameter affecting the power turbine expansion ratio.

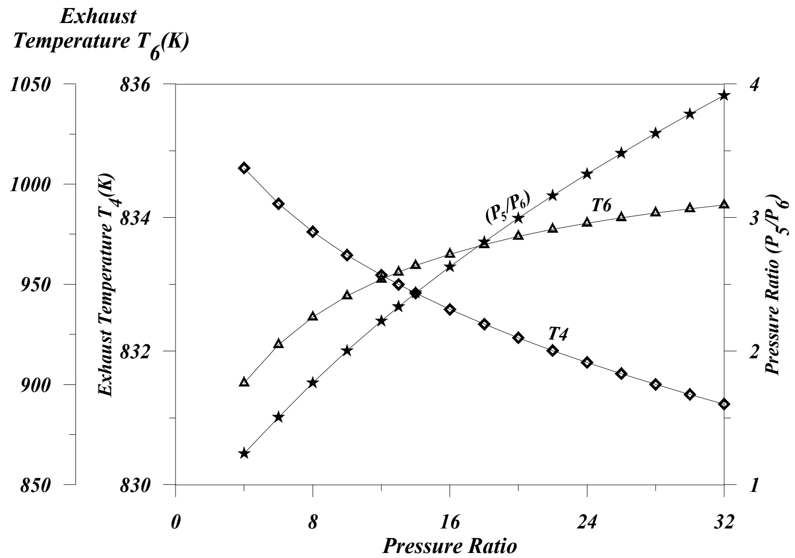


Figure (17):T_4,T_6, and P_5/P_6 versus PR.

3-2-2: Constant Maximum Temperature:

In the same of the simple gas turbine cycle, the set of calculations from ***1144 to 1588 K*** were used as the maximum temperature.

Table (3) gives the optimum pressure ratios at which the net work and the cycle efficiency are maximum.

Table (3): Optimum Pressure Ratios for Maximum Cycle Efficiency and Maximum Net Work.

Max. Temp. K	Optimum Pressure Ratios for maximum	
	Cycle Efficiency	Net Work
1144	***12***	***12***
1255	***16***	***16***
1366	***20***	***20***
1477	***24***	***24***
1588	***30***	***30***

Figure (18) shows the relationship between the net work and the pressure ratio for various maximum temperatures. At a constant maximum temperature, the net work increases when the pressure ratio increases until a certain value of pressure ratio and then it begins to decreases with the increase of the pressure ratio, because more work is needed for the compressor. At a constant pressure ratio, the net work increase with the increase of the maximum temperature due to greater expansion.

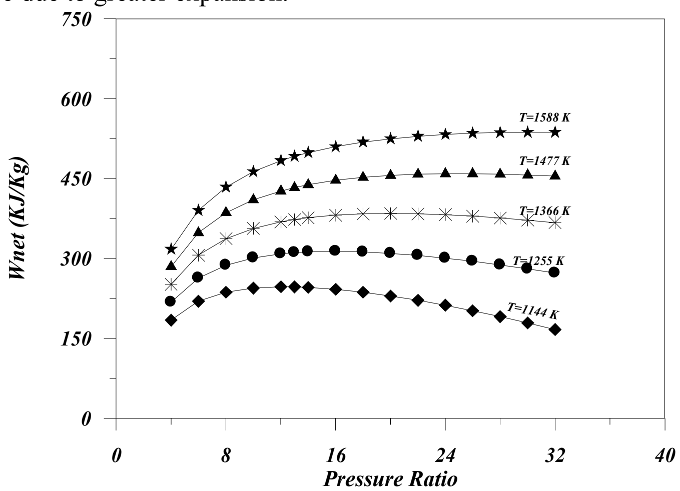


Figure (18): Net Work versus PR for Reheat Gas Turbine Cycle.

Figure (19) shows the relation between the cycle efficiency and the pressure ratios for various maximum temperatures. At a constant maximum temperature, the cycle efficiency increases when the pressure ratio increases until a certain value of pressure ratio and then it begins to decreases with the increase of the pressure ratio, because the net work becomes less. At a constant pressure ratio, the cycle efficiency increases with the increase of the maximum temperature, because net work becomes more due to greater expansion.

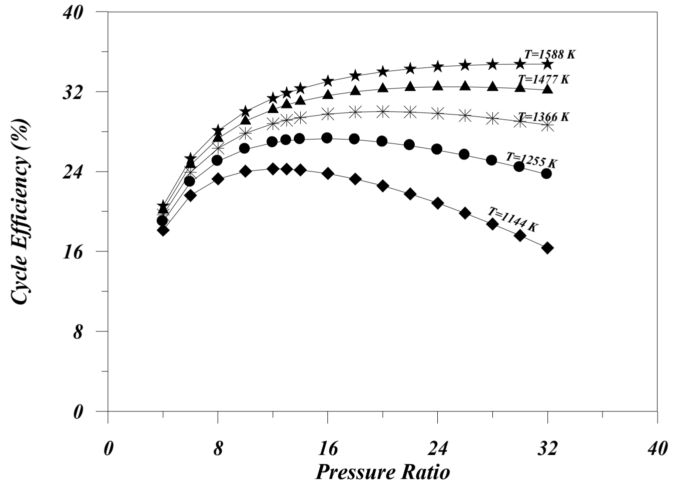


Figure (19): Cycle Efficiency versus PR for Reheat Gas Turbine Cycle.

Figure (20) shows the relation between the exit temperature from the first gas turbine and the pressure ratio for different maximum temperatures. For a constant pressure ratio, the exit temperature increases as the maximum temperature increases. At a constant maximum temperature, the exit temperature decreases when the pressure ratio increases, because greater expansion occurs in the first gas turbine as more work is needed for the compressor.

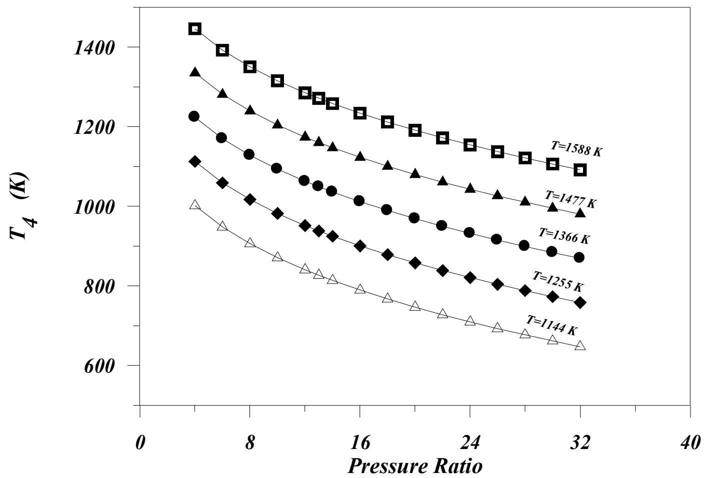


Figure (20): T_4 versus PR for Reheat Gas Turbine Cycle.

Figure (21) shows the relation between the exhaust temperature and the pressure ratio for different maximum temperatures.

At a constant pressure ratio, the exhaust temperature increases as the maximum temperature increases. At a constant maximum temperature, the exhaust temperature decreases with the increase of the pressure ratio at the beginning (low pressure ratio) and then after a certain value of pressure ratio, the exhaust temperature begins to increase, because at high pressure ratio the compressor needs more work, thus the expansion in the first gas turbine becomes more and more and consequently the heat addition in the second combustion chamber becomes more and more which leads to that state. However, the increases in the exhaust temperature are very small after the pressure ratio of ***24***.

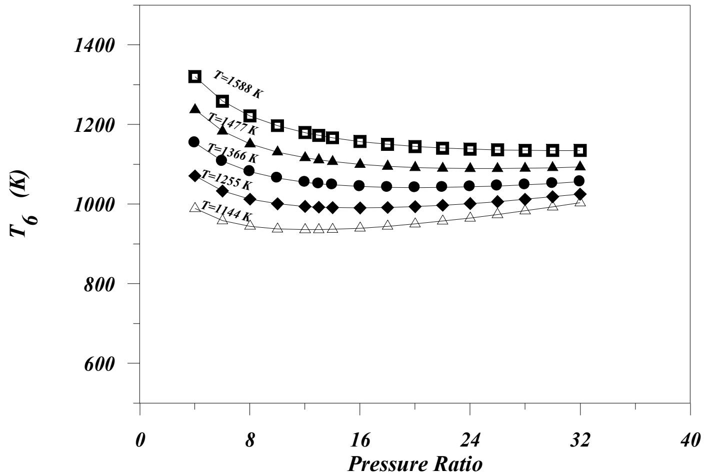
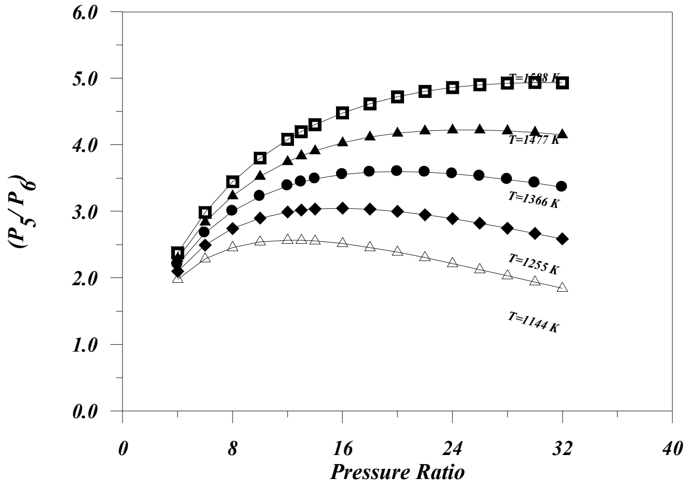


Figure (21): T_6 versus PR for Reheat Gas Turbine Cycle.

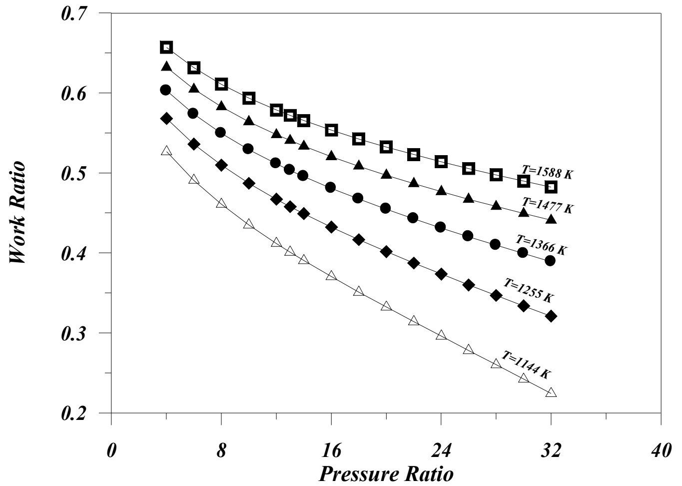
Figure (22) shows the relation between the power turbine expansion ratio and the pressure ratio for different maximum temperatures.

At a constant pressure ratio, the power turbine expansion ratio increases as the maximum temperature increases, because the exit enthalpy from the first gas turbine increases, so that the exit relative pressure increases which causes the increase in the pressure of state ***5***. At a constant maximum temperature, the power turbine expansion ratio increases when the pressure ratio increases until a certain value of pressure ratio, the power turbine expansion ratio begins to decreases, because the greater expansion occurs in the first gas turbine due to more compressor work makes the exit enthalpy becomes less, so that the exit relative pressure becomes less which leads to the decrease in the pressure of state ***5***.



Figure(22): P_5/P_6 versus PR for Reheat Gas Turbine Cycle.

Figure (23) shows the relation between the work ratio and the pressure ratio for various maximum temperatures. It is the same as in the simple gas turbine cycle.



Figure(23): Work Ratio versus PR for Reheat Gas Turbine Cycle.

3-3: Results of the Combined Cycle:

The results of the simple gas turbine-steam turbine and the reheat-steam turbine combined cycles are given in the same plots to make a comparison between them.

The results are illustrated in ***Figures (24) to (29)***. ***Figure (24)*** shows the relation between the exhaust temperature and the maximum temperature for both simple and reheat gas turbine cycle. The exhaust temperature increases as the maximum temperature increases and it is clear that the exhaust temperature for the reheat gas turbine is greater than the exhaust temperature for the simple gas turbine, which means that the heat energy available in the exhaust gases is more in the reheat gas turbine cycle.

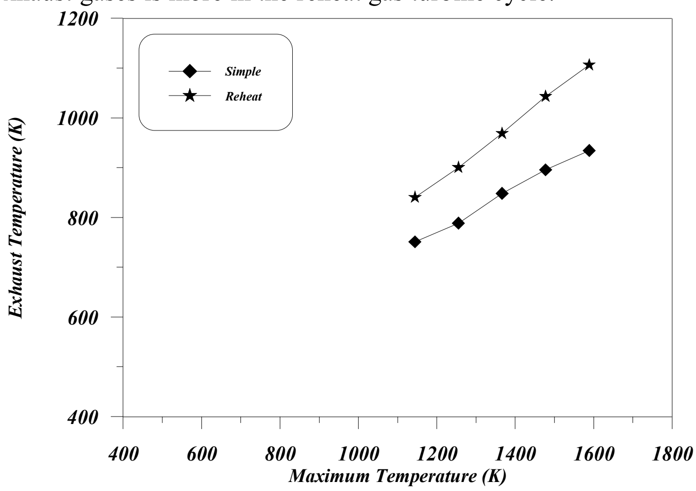


Figure (24):Exhaust Temperature versus Max. Temperature.

Figure (25) shows a plot of the stack temperature versus the maximum temperature for both combined cycles. It is clear that the stack temperature decreases as the maximum temperature increases for both combined cycles and the stack temperature of the combined cycle with the reheat gas turbine is less than the stack temperature of the combined cycle with the simple gas turbine at a given maximum temperature, which means that the heat available is utilized with greater efficiency when it was more.

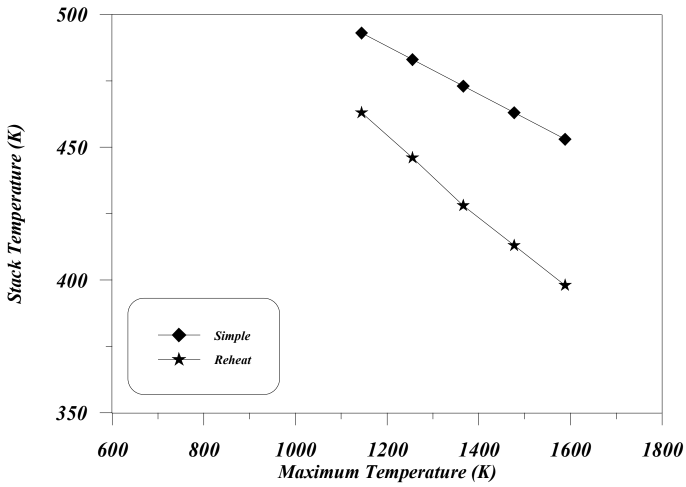


Figure (25): Stack Temperature versus Max. Temperature.

Figure (26) shows a plot of the boiler efficiency versus the maximum temperature. When the maximum temperature increases the boiler efficiency increases in the both combined cycles. The boiler efficiency of the combined cycle using the reheat gas turbine is greater than the boiler efficiency of the combined cycle using the simple gas turbine by ***(10-17) %***, because of the lower stack temperature in the combined cycle using reheat gas turbine.

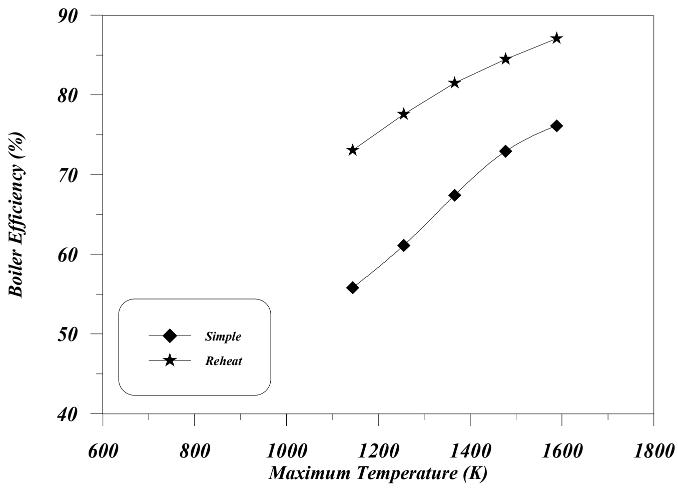


Figure (26): Boiler Efficiency versus Max. Temperature.

Figure (27) shows a plot of the overall steam cycle efficiency versus the maximum temperature. The overall steam cycle efficiency increases when the maximum temperature increases. The overall steam cycle efficiency of the combined cycle with the reheat gas turbine is greater than the overall steam cycle efficiency of the combined cycle with the simple gas turbine by ***(5-10)%,*** because of the high boiler and steam cycles efficiencies.

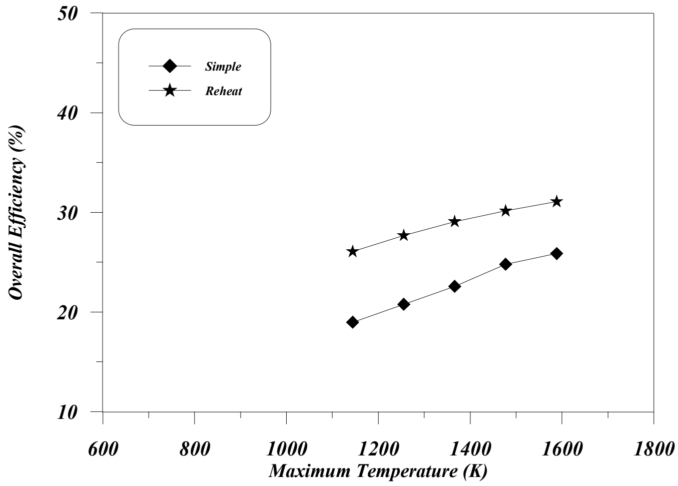


Figure (27): Overall Steam Cycle Efficiency versus Max. Temperature.

Figure (28) shows a plot of the capacity ratio versus the maximum temperature. It is clear that the capacity ratio increases as the maximum temperature increases, because the gas turbine net work is large due to high maximum temperature. At a given maximum temperature the capacity ratio of the combined cycle using the reheat gas turbine is less than the capacity ratio of the combined cycle using the simple gas turbine, because of the work done by the steam turbine of the combined cycle using the reheat gas turbine is more due to more heat energy is utilized, which leads to generate more steam and hence more work of steam turbine.

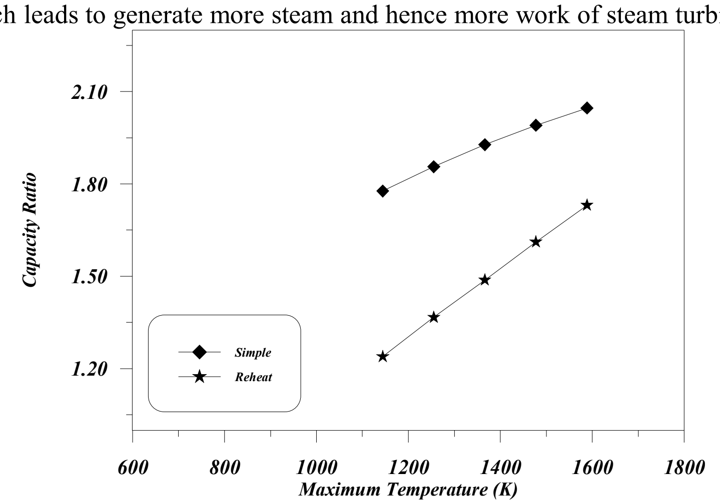


Figure (28): Capacity Ratio versus Max. Temperature.

Figure (29) shows a plot of the combined cycle's efficiencies versus the maximum temperature. The efficiency of the combined cycle using the reheat gas turbine is greater than the efficiency of the combined cycle using the simple gas turbine by ***(4)%*** due to the better utilization of the waste heat in the boiler in the combined cycle with the reheat gas turbine. It was found from the heat balance for all combined cycles calculations that the percentages of errors were about ***0.91-3.2%,*** so that the assumptions of the combined cycles seem suitable and reasonable.

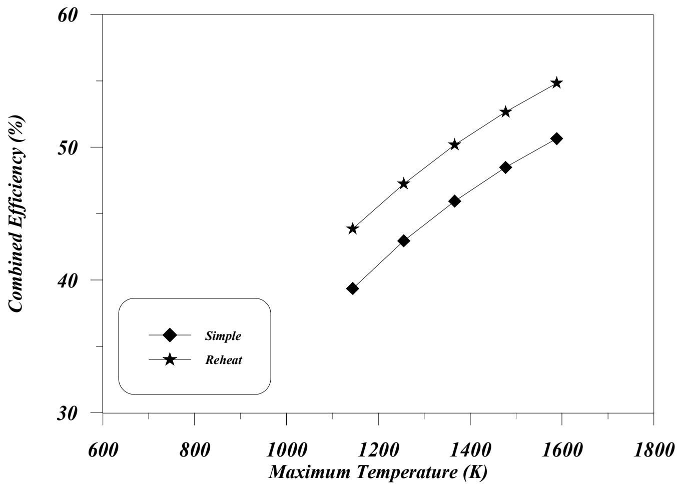


Figure (29): Combined Cycle Efficiency versus Max. Temperature.

4-Conclusions

1- The reheat gas turbine-steam turbine combined cycle is better as compared to the simple gas turbine-steam turbine combined cycle, because the output per unit mass of air flow is significantly improved by ***(35-41) %*** , and the efficiency is greater by 4% for the combined cycle utilizing the reheat gas turbine, which points to potential cost saving for such a cycle.

2- It is found from the present investigations that the optimum conditions of the gas turbine cycle in the combined cycle mode occur at the maximum net work conditions.

3- It is found that the efficiency of the waste heat boiler for high exhaust temperature ranges ***(maximum η_B=87%)*** is quite comparable to the efficiency of the conventional boilers. The waste heat boiler has the added advantage of simplicity; therefore, this type of combined cycle using waste heat boiler has great promise for power generation.

5-References

-Dallenback, P.A.,2002, *"Improved Gas Turbine Efficiency Through Alternative Regenerator Configuration",* Transaction ASME, Journal of Engineering for Gas Turbine and Power, vol.124, pp.441-446.

-Holman, J.P., 2002*"Heat Transfer"*,9th edition, McGraw Hill series in mechanical Engineering, pp.511-556.

-Horlock, J. H., Young, J.B., Manfirda, G., 2000, *"Exergy Analysis of Modern Fossil-Fuel Power Plants",* ASME J. Eng. For Gas Turbine and Power 128, pp-1-7.

-Horlock, J. H., Young, J.B., Manfirda, G., 1998, *"The Rational Efficiency of Fossil-Fuel Power Plants",* ASME Symposium on thermodynamics Int. Mech. Eng., Nov., California.

-Kehlhofer, R., 1997, *"Combined-Cycle Gas and Steam Power Plant",* PennWell Publishing Company, Tulsa-Oklahoma.

-Ravi Kumar, N., et.al., 2005, *"Exergy analysis of Gas Turbine Power Plant with Alternative Configuration of Regenerator",* Proceeding on CD, 2nd International Exergy Energy Environmental Symposium (IEEES2), Kos, Greece, VI-13.

-Ravi Kumar, N., et.al., 2006, *"Performance Simulation of Heat Recovery Steam Generator in Combined Cycle Power Plant",* Proceeding 18th National and 17th ISHMT-ASME Heat and Mass Transfer Conference, India, pp.1781-1787.

-Ravi Kumar, N., et.al., 2007, *"Thermodynamic Analysis of Heat Recovery Steam Generator in Combined Cycle Power Plant",* Thermal Science, Vol.11, No.4, pp.143-156.

-Ravi Kumar, N., Sita Rama Raju, A.V., 2005, *"The Study of the Effects of Gas Turbine Inlet Cooling on Plant and HRSG Performance",* Proceedings, National Conference on Advances in Mechanical Engineering, India, pp.55-60.

-Rice, I.G., 1980, *"The Combined Reheat Gas Turbine Steam Turbine Cycle",* ASME, Vol.102, January, pp. 35-49.

-Srinivas, T., Gupta, A.V.S.S.K.S., Reddy, B.V.,2007, *"Generalized Thermodynamic Analysis of Steam Power Cycles with 'n' Number of Feedwater Heaters",* Int. J. of Thermodynamics, December,Vol.10 (No.4), pp. 177-185,

-Yadav, J.P., Singh, O., 2006, *"Thermodynamic Analysis of Air Cooled Simple Gas and Steam Combined Cycle Plant",* Journal of Institution of Engineers (India)-Mechanical Engineering, Vol.86, pp.222.

This academic article was published by The International Institute for Science, Technology and Education (IISTE). The IISTE is a pioneer in the Open Access Publishing service based in the U.S. and Europe. The aim of the institute is Accelerating Global Knowledge Sharing.

More information about the publisher can be found in the IISTE's homepage: http://www.iiste.org

CALL FOR JOURNAL PAPERS

The IISTE is currently hosting more than 30 peer-reviewed academic journals and collaborating with academic institutions around the world. There's no deadline for submission. **Prospective authors of IISTE journals can find the submission instruction on the following page:** http://www.iiste.org/journals/ The IISTE editorial team promises to the review and publish all the qualified submissions in a **fast** manner. All the journals articles are available online to the readers all over the world without financial, legal, or technical barriers other than those inseparable from gaining access to the internet itself. Printed version of the journals is also available upon request of readers and authors.

MORE RESOURCES

Book publication information: http://www.iiste.org/book/

Recent conferences: http://www.iiste.org/conference/

IISTE Knowledge Sharing Partners

EBSCO, Index Copernicus, Ulrich's Periodicals Directory, JournalTOCS, PKP Open Archives Harvester, Bielefeld Academic Search Engine, Elektronische Zeitschriftenbibliothek EZB, Open J-Gate, OCLC WorldCat, Universe Digtial Library , NewJour, Google Scholar