THE EFFECT OF DIFFERENT CYCLE ARRANGEMENTS ON THE PERFORMANCE OF OPEN GAS TURBINE POWER PLANT

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ABSTRACT

The use of gas turbine is increasing day by day for producing electricity and for various industrial applications. In this work a new approach of study for the effect of different cycle arrangements on the performance of simple gas turbine has been investigated. The new cycle arrangements include: reheat with heat exchanger, reheat with water injection, heat exchanger with water injection and reheat together with heat exchanger and water injection. All these arrangements are compared to the performance of gas turbine cycle with no modifications. A Matlab code was written to calculate the combustion characteristics and major performance parameters such as net work, fuel/air ratio, specific fuel consumption, and thermal efficiency ….etc. It's observed that using reheat in addition to heat exchanger with water injection gives higher thermal efficiency, maximum increment was 28.6% in compare with normal basic cycle and lower fuel consumption, maximum reduction was 22.2 in compare with normal basic cycle.

KEYWORDS: gas turbine, open cycle, modifications, water injection, heat exchanger, reheat

الخلاصة

في هذا البحث تم دراسة تاثير دمج التحسينات المختلفه على دوره التوربين الغازي بشكل جديد. شملت الدراسة استخدام اعادة التسخين مع المبادل الحراري، اعادة التسخين مع تقنية ضخ الماء، ومره اخرى المبادل الحراري مع ضخ الماء واخيرا المبادل الحراري واعادة التسخين مع ضخ الماء سوية. تم بناء برنامج بلغة ماتلاب لايجاد خصائص الاحتراق و العوامل المؤثره على اداء المحطة ومنها:الشغل الصافي ومعدل استهلاك الوقود والكفاءة الحرارية ....الخ. لوحظت اعلى كفاءة حرارية (٢٨.٦%) مع اقل معدل استهلاك للوقود (٢٢.٢%) عند استخدام نظام اعادة التسخين بالاضافة الى نظام المبادل الحراري مع تقنية حقن الماء بالمقارنة مع الدوره الاساسية للتوربين الغازي.

NOMENCLATURE

Symbol	Definition	Unit
T	Temperature	*K*
C_p	Specific Heat at Constant Pressure	*kJ/kg.K*
C_1,C_2	Parameters of Fuels	-
h	Specific Enthalpy	*kJ/kg*
f	Fuel to Air Ratio	-
W	Work	*kJ/kg*
S.F.C.	Specific Fuel Consumption	Kg/kW.hr
PR	Pressure Ratio	-

Q_{LHV}	Lower heating value per unit mass	kJ/kg
C.C	Combustion chamber	

GREEK SYMBOLS

Symbol	Definition	Unit
γ	Specific Heat Ratio	-
η	Efficiency	-

SUBSCRIPT

Symbol	Definition
a	Air
b	Burner
c	Compressor
g	Gas
m	Mechanical
max.	Maximum
n	Net
r	Ratio
t	Turbine

INTRODUCTION

In the present time a good and compact stationary gas turbine engine has been introduced with capacity of 425 MW and pressure ratio of 20 according to Siemens. But the main focus still directed to the development of metallurgic limit of alloys of turbine blades which allows to reach a high temperature at the turbine inlet which in its turn permit to burn more fuel safely, which means more power output. The demand of electricity is increasing day by day especially in such developing country like Iraq. The electricity demand also hits the maximum in the summer days which unfortunately implies a lower value of air density and lower mass of flowing air and consequently this lowers the produced power. Also, the ambient conditions play an important role which is affecting the mass flow rate and moisture content of air in the flow passages in the gas turbine power plant and if the attention is paid to the severe climate conditions of middle east countries in general and in Iraq in particular, the need of adopting a good technique for cooling the incoming air in the gas turbine power plant becomes an urgent issue. For the mentioned reasons this paper is focused to find a non-traditional way to increase both power output and thermal efficiency of the gas turbine cycle with taking into consideration the best compromised way to enhance the power output without increasing too much the amount of burning fuel to avoid any more contributions in the greenhouse effect. The adopted method is trying to apply new configurations and in the same time multiple operating conditions which grant the best result concerning the major performance parameters like power output (which is related to the electricity demand issue), specific fuel consumption (which is related to the greenhouse effect issue) and thermal efficiency (which is related to the economic justification of the gas turbine power plant).

THEORETICAL ANALYSIS

In this work the mathematical analysis falls into two sections, the first section deals with the analysis of simple cycle of open gas turbine, and the second section deals with the analysis of modified cycle open gas turbine which include (intercooling, reheat, and heat exchanger). All theoretical analysis of this study is made according to the following assumptions:

1. Both compression and expansion processes are isentropic.
2. No potential energy change between inlet and outlet of each component.

3. The pressure drop through air inlet duct, combustion chamber, nozzles…etc., are neglected.
4. Mass flow rate is constant throughout the cycle.
5. The specific heat is not strongly function of temperature variation, i.e., The specific heat is constant.

The reheat cycle allow to reach the maximum temperature in gas turbine cycle again in the combustion chamber without exceeding the metallurgical limit of turbine vanes, i.e., not to exceed the maximum allowable temperature at the inlet of the turbine.

The heat exchanger uses the energy of the exhaust gases at outlet from the turbine which serve multiple goals:

- The first is to use the wasted thermal heat in heating the incoming air from the compressor.
- The second is to prevent the thermal pollution which is in part coinciding with global trends in minimizing the greenhouse effect especially in such severe hot environmental condition like these in Iraq.
- Thirdly is to minimize the fuel flow rate which reflected positively on the economic justifications of overall operation of the gas turbine power plant which is in common has a poor thermal efficiency.

The method of water injection in the inlet air duct of the compressor implies a cooling effect on the operation condition on different parts of the gas turbine power plant. Some of the cooling effect comes as a result of evaporative cooling of water vapor during the compression process in the stages of compressor and the rest from the direct injection of very fine and atomized water droplets in the flow passages of the compressor. The cooling process majorly gives the following advantages:

- It minimizes the work of the compressor which in turn raises the thermal efficiency in overall.
- It stimulates the combustion process positively because of the role of the steam which is converted from the injected water.
- It gives a good possibility to increase the fuel rate which in turn increases the brake power output of the plant during the hot summer days when electricity demand hits the maximum.
- It decreases the NO_x emissions because of lowering the flame temperature in the combustion chamber.

SIMULATION OF SIMPLE CYCLE

Figure (1) shows a simplified diagram of simple gas turbine which describes basic components of the plant which are explained very briefly below:

1. Compressor: The task of compressor is to increase the pressure of incoming air so that the compression and power extraction processes after combustion can be carried out more efficiently. During the isentropic compression process:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma_a - 1}{\gamma_a}} = \left(PR\right)^{\frac{\gamma_a - 1}{\gamma_a}} \quad (1)$$

The isentropic efficiency of compressor is given by the following empirical relation **(Jaber et al., 2007)**

$$\eta_c = \left[1 - \left(\frac{0.09 + (rp - 1)}{300}\right)\right] \quad (2)$$

Also, it is defined as the ratio of work input required in isentropic compression between P_1& P_2 to the actual work required**, (Willard W,1997)**.

$$\eta_c = \frac{T_2 - T_1}{T_2^{/} - T_1} \quad (3)$$

So the work input to the compressor is:

$$W_c = Cp_a * \left(T_2^{/} - T_1\right) / \eta_m \qquad \text{; where } \eta_m \text{ mechanical efficiency} \quad (4)$$

2. Combustion Chamber: The combustion chamber is designed to burn a mixture of fuel and air to deliver the burned gases to the turbine at a uniform temperature. The gas temperature of the turbine must not exceed the allowable structure temperature of the turbine **(Jack, D.1998)**. Since the process is assumed to be adiabatic with no work transfer and neglected pressure loss, so the energy equation, is simply ;

$$\sum\left(m_i h_{i,3}\right) - \left(h_2^{/} - f.h_{f,t}\right) = 0 \quad (5)$$

Now making the enthalpy of reaction at a reference temperature of 25 °C, so equation can be expanded in the usual way to get;

$$\left(1 + f_t\right).Cp_g.\left(T_{max} - 298\right) + f_t.Q_{LHV} + Cp_a.\left(298 - T_2^{/}\right) + f_t.Cp_f\left(298 - T_f\right) = 0 \quad (6)$$

By simplifying equation (6) the theoretical fuel to air ratio will be:

$$f_t = \frac{Cp_g.\left(T_1 - T_{max}\right) + Cp_a.\left(T_2^{/} - T_1\right)}{\left(Cp_g.\left(T_{max} - T_1\right) + Q_{LHV}\right)} \quad (7)$$

The actual (fuel / air) ratio for given temperature difference is given by;

$$f_a = f_t / \eta_b \quad (8)$$

3. Turbine: The purpose of turbine is to extract kinetic energy from the expanding gases which flow from the combustion chamber **(Jack, D.1998)**. The kinetic energy is converted to shaft horse power to drive the compressor and other components. Nearly three-forth of all energy available from the product of combustion is required to drive the compressor. During the isentropic expansion process:

$$\frac{T_4}{T_{max}} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma_g - 1}{\gamma_g}} = \left(PR\right)^{\frac{\gamma_g - 1}{\gamma_g}} \quad (9)$$

The isentropic efficiency of turbine is given by the following empirical equation **(Jaber et al., 2007)**

$$\eta_t = \left[0.9 - \left(\frac{\frac{T_2}{T_1} - 1}{250}\right)\right] \quad (10)$$

Similarly the isentropic efficiency of compressor is defined as the ratio of actual work output to the isentropic work output between P_3 and P_4

$$\eta_t = \frac{T_{max} - T_4^{/}}{T_{max} - T_4} \tag{11}$$

Then the turbine work output is:

$$W_t = Cp_g * \left(T_{max} - T_4^{/}\right) \tag{12}$$

The net work output is determined by subtracting equation (12) from equation (4). The work ratio is defined as the ratio of net work output to turbine output work:.

$$W_R = \frac{W_n}{W_t} \tag{13}$$

The specific fuel consumption is given by;

$$S.F.C. = \frac{f_a \times 3600}{W_n} \tag{14}$$

Then the cycle thermal efficiency is found there from the equation below **(Jack, D.1998)**;

$$\eta_{th} = \frac{3600}{S.F.C.* Q_{LHV}} \tag{15}$$

SIMULATION OF MODIFIED CYCLE

Instead of using conventional modifications (intercooling, reheating and heat exchanger) which is added separately to the basic cycle by different works papers. A new approach of modifications are presented in this paper by adding two or three techniques together and compared to the basic cycle of gas turbine.The new arrangements are summarized below:

1. Reheat with Heat exchanger
2. Water injection and Heat exchanger
3. Water injection with Reheat
4. Reheat with heat exchanger and water injection

The objective of using such kind of arrangements is to boost the efficiency and performance levels of the plant as compared to the original basic cycle of gas turbine.

1. Reheat with Heat exchanger

The method of improvement specific power output is achieved in reheat cycle at the expense of the efficiency. This can be overcome by adding a heat exchanger to the reheat cycle. The schematic arrangement of the reheat cycle with heat exchanger is given in **Fig. (2)**. The higher exhaust temperature is now fully utilized in the heat exchanger. In fact when a heat exchanger is employed, the efficiency is higher with reheat than without. The high pressure turbine must be exactly equal to the work input for the compressor with the following equation

$$Cp_a . \frac{\left(T_2^{/} - T_1\right)}{\eta_m} = Cp_g .\left(T_{max} - T_5^{/}\right) \tag{15}$$

For the first combustion chamber, the energy equation is used

$$\sum(m_i h_{i,3}) - (h_2^{/} - f.h_{ft1}) = 0 \tag{16}$$

The first theoretical fuel/air ratio is calculated from expanding equation (16) as follows;

$$(1+f_{t1}).Cp_g.(T_{max}-298)+f_{t1}.Q_{LHV}+Cp_a.(298-T_{max})+f_{t1}.Cp_f(298-T_f)=0 \tag{17}$$

$$f_{t1} = \frac{Cp_g.(T_1 - T_{max}) + Cp_a.(T_3 - T_1)}{(Cp_g.(T_{max} - T_1) + Q_{LHV})} \tag{18}$$

The same procedure is used for second combustion chamber;

$$\sum(m_i h_{i,4}) - (h_4^{/} - f.h_{ft2}) = 0 \tag{19}$$

The second fuel/air ratio is then calculated as follows;

$$(1+f_{t2}).Cp_g.(T_5-298)+f_{t2}.Q_{LHV}+Cp_g.(298-T_4^{/})+f_{t2}.Cp_f(298-T_f)=0 \tag{20}$$

$$f_{t2} = \frac{Cp_g.(T_5^{/} - T_{max})}{(Cp_g.(T_{max} - T_1) + Q_{LHV})} \tag{21}$$

Then the total fuel/air ratio is given by;

$$f_{t,tot} = f_{t1} + f_{t2} \tag{22}$$

The total actual (fuel / air) ratio for given temperature difference is given by;

$$f_{a,tot} = f_{t,tot} / \eta_b \tag{23}$$

The net work of the plant is equal to work output of low pressure turbine

$$W_n = W_{t2} = Cp_g.(T_5 - T_7^{/}) \tag{24}$$

The work ratio is given by;

$$W_R = \frac{W_n}{W_t} \tag{25}$$

The specific fuel consumption is given by;

$$S.F.C. = \frac{f_{a,tot} \times 3600}{W_n} \tag{26}$$

Finally the plant efficiency is found below **;**

$$\eta_{th} = \frac{3600}{S.F.C.*Q_{LHV}} \tag{27}$$

2. Water injection and Heat exchanger

Gas turbines are very sensible to ambient air wet bulb temperature. For instance, if the gas turbine operates with an ambient temperature of 35 °C a reduction about 20% of its capacity of generation may occur when compared to ISO standard conditions of 15°C. This is due to the direct influence of air density on the amount of air introduced in the combustion chamber (at higher temperatures, air presents low density and, therefore the air mass supplied to the turbine is reduced). Previous studies **(Guimarães 2000, Bassily 2001)** have demonstrated the advantages of the use of evaporative panels in the cooling of the air of the gas turbines.

During the gas turbine cycle the ambient air is initially cooled and humid in the direct evaporative cooler **(DEC)** due to simultaneous mass and heat exchange between the air stream and the wet surface of the panel. The mathematical model implemented for the system under study involves the following main considerations:

(i) Panel is well insulated.
(ii) Neglecting the change in the kinetic and potential energy.
(iii) The pressure drop through the inlet air duct of the compressor where the direct evaporative cooling occurs is neglected.
(iv) The change in the absolute humidity of the air is ignored.

The schematic diagram of the direct evaporative panel that is adopted in this work is shown below:

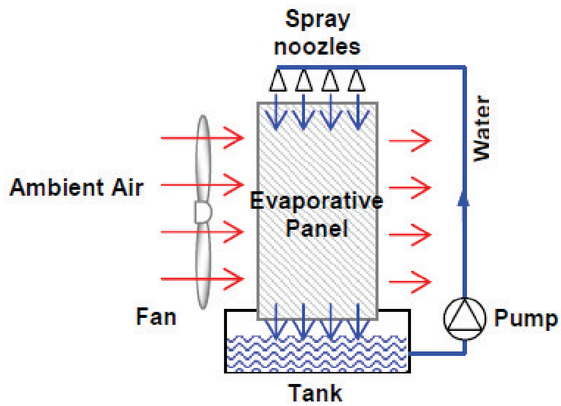

Figure (1) shows the schematic diagram of DEC

On the other hand the exhaust gases leaving the turbine at the end of expansion are still at a high temperature (high enthalpy). If these gases are allowed to pass into atmosphere, this represent a loss of available energy, this energy can be recovered by passing the gases from the turbine through a heat exchanger, where the heat transfer from the gases is used to heat the air leaving the compressor. Therefore the function of heat exchanger is to heat the outlet air from compressor (T'_2 to T_3) and to cooled the exhausted gases from turbine (T'_5-T_6) as shown in **Figure.(3) and Figure (4)**, so the ideal heat exchanger have (T'_2=T_6) and (T_3=T'_5) which is assumed in this work.

The theoretical analysis of this arrangement will be summarized in the following couples of equations

During the isentropic compression process:

$$\frac{T_{2new}}{T_{1inj}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma_a-1}{\gamma_a}} = (PR)^{\frac{\gamma_a-1}{\gamma_a}} \tag{28}$$

The isentropic efficiency of compressor is given by the following empirical relation

$$\eta_c = \left[1 - \left(\frac{0.09 + (rp - 1)}{300}\right)\right] \tag{29}$$

It is defined as the ratio of work input required in isentropic compression between P_1& P_2 to the actual work required**, (Willard W,1997)**.

$$\eta_c = \frac{T_{2new} - T_{1inj}}{T'_{2new} - T_{1inj}} \tag{30}$$

So the work input to the compressor is:

$$W_c = Cp_a * \left(T'_{2new} - T_{1inj}\right) / \eta_m \quad ; \text{where } \eta_m \text{ mechanical efficiency} \tag{31}$$

The combustion process will be the same as its in the simple cycle except the temperature entering the combustion chamber will be treated in the heat exchanger unit hence, the theoretical fuel to air ratio is given by:

$$f_t = \frac{Cp_g.(T_{1inj} - T_{\max}) + Cp_a.(T_3 - T_{1inj})}{\left(Cp_g.(T_{\max} - T_{1inj}) + Q_{LHV}\right)} \tag{32}$$

The actual (fuel / air) ratio for given temperature difference is given by;

$$f_a = f_t / \eta_b \tag{33}$$

During the isentropic expansion process

$$\frac{T_5}{T_{\max}} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma_g - 1}{\gamma_g}} = (PR)^{\frac{\gamma_g - 1}{\gamma_g}} \tag{34}$$

The isentropic efficiency of turbine is given by the following empirical equation

$$\eta_t = \left[0.9 - \left(\frac{\frac{T_{2new}}{T_{1inj}} - 1}{250}\right)\right] \tag{35}$$

Similarly the isentropic efficiency of compressor is defined as the ratio of actual work output to the isentropic work output between P_3 and P_4

$$\eta_t = \frac{T_{max} - T'_5}{T_{max} - T_5} \tag{36}$$

Then the turbine work output is:

$$W_t = Cp_g * \left(T_{max} - T_5^{/}\right) \tag{37}$$

3. Water injection with Reheat

The theoretical equations of such kind of modification will be exactly the same equations used in the reheat cycle except the cooling of air temperature which enters the compressor by using water injection technology. Therefore it's enough to observe and discuss the results without need to rewrite the governing equations of performance parameters again.

4. Reheat with Heat exchanger and Water injection

The last new arrangement used in this work is to add the water injection unit together with heat exchanger and reheat systems. The analysis of such type of arrangement can be summarized as shown below;

The first turbine work equal to the work input for the compressor

$$Cp_a . \frac{\left(T_{2nes}^{/} - T_{1inj}\right)}{\eta_m} = Cp_g . \left(T_{max} - T_5^{/}\right) \tag{38}$$

The first and second theoretical fuel/air ratios are calculated from expanding equation (16 and 19) as follows;

$$f_{t1} = \frac{Cp_g . \left(T_{1inj} - T_{max}\right) + Cp_a . \left(T_3 - T_{1inj}\right)}{\left(Cp_g . \left(T_{max} - T_{1inj}\right) + Q_{LHV}\right)} \tag{39}$$

$$f_{t2} = \frac{Cp_g . \left(T_5^{/} - T_{max}\right)}{\left(Cp_g . \left(T_{max} - T_{1inj}\right) + Q_{LHV}\right)} \tag{40}$$

Then the total fuel/air ratio is given by;

$$f_{t,tot} = f_{t1} + f_{t2} \tag{41}$$

The network in addition to thermal efficiency and specific fuel consumption are calculated according to the procedure mentioned in the first arrangement.

RESULTS AND DISCUSSION

The layout of the results is divided into two sections; these explain the effect of maximum temperature and pressure ratio on the performance of simple and modified cycle of open gas turbine plant respectively. In this work we studied different types of arrangements and the main target is increasing power output as much as possible without affecting negatively the value of thermal efficiency or at least not lowering it.

The study start with combining the reheat and heat exchanger systems together, keeping in mind the simple cycle as an embark line for purposes of comparison. The layout of two systems components are shown in **Figure (3)**. This type of modification tends to increase the specific fuel consumption compared with the others because of more fuel is expected to burn due to presence of two combustion chambers in such cycle , i.e, consumes more fuel. As shown in **Figure (7)** this increment by (0.1585977) kg/kW hr at pressure ratio of 4 and (0.1656321) kg/kW hr at pressure ratio of 9. **Figure (8)** shows the specific fuel consumption versus the maximum temperature in the cycle which is usually at combustion chamber exit. The arrange of heat exchanger is consuming fuel more than that of all modifications together, in exact this noticed to be (0.2124907) kg/kW hr at 1100 K of maximum temperature and (0.1585977) kg/kW hr at 1600 K of maximum temperature.

In both cases the massage says that simple cycle still indicates weak outcomes due to absence of any improvements.

The second modification is adding a water injection system in front of compressor to enhance the mass flow rate of incoming air by increasing the air density by adding more moisture to it. The numerical values of multiple performance parameters are tabulated in Table 2. The specific fuel consumption and net work are increasing with increasing pressure ratio. This may be explained because of adding more mass to the plant accompanying with the injected water and the cooling effect which permit burn more fuel without exceeding the maximum allowable temperature of turbine blades.

On the other hand the thermal efficiency and work ratio are decreasing slightly with increasing pressure ratio. This mainly because of the increasing in heat input to the cycle and the little increment in compressor work due to the increasing in mass flow rate.

The other modification is combining the reheat and water injection. The main results are listed in tables 4 and 5. This type of combination record a lower value of thermal efficiency for the same reference maximum temperature in the cycle (1600 K) compared to the previous modification i.e., heat exchanger and water injection. This mainly belongs to effective utilization of heat of exhausted gases in the heat exchanger cycle. The net work in case of reheat and water injection is higher than that of heat exchanger and water injection because of increasing in the mass added and also due to improvement in combustion characteristics by means of moisture content of pressurized air entering the combustion chamber.

Finally, the study performed all modification together which they are reheat with heat exchanger and water injection. The main parameters performance are plotted in two cases with reference to maximum temperature in the cycle holding pressure ratio constant at Pr=4 and the other performance curves are plotted versus pressure ratios holding T_{max} constant at 1600 K.

It is evident that there a good reduction of specific fuel consumption compared to simple normal cycle. The reduction was about 55% depending on operating conditions as shown in **Figures (7 & 8)**.

The thermal efficiency also improved in comparison to the simple cycle by about 48% in average. **Figures (9 & 10)** display the thermal efficiency versus pressure ratio and maximum temperature in the cycle consequently. The maximum value was 48% resulted from incorporating all types of modifications in this study.

The work ratio was sensitive to the change either in pressure ratio or in the maximum temperature as shown in **Figures (11 & 12)**. The work ratio tends to decrease rapidly with the increase in pressure ratio and tends to decrease in the increasing of maximum temperature. The maximum value obtained in this study was about 60%.

CONCLUSION

The following conclusions can be drawn from the present work;

- The use of all modifications guarantees the minimum value of specific fuel consumption.
- The high thermal efficiency is obtained from the three modifications (near 48%).
- Promising convergence between the modification of reheat and heat exchanger and the modification of adding three systems at the same time are observed from the results, so the best compromise must be chosen between the two approaches to choose the best one.

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Table (1) Operating conditions of the plant

Compressor inlet temperature (T_1)	(310-300) K^o
Specific heat ratio of air (γ_a)	1.4
Specific heat ratio of exhaust gas (γ_g)	1.3333
Pressure ratio (PR)	(4-9)
Mechanical efficiency	98 %
Efficiency of burner	98 %
Fuel temperature (T_f)	298 K

Table (2) Heat exchanger and water injection (T_{max}=1600 K)

Pr	SFC (kg/kW.hr)	η_{th} (%)	Net Work (kJ/kg)	Work Ratio (%)
4	0.1585977	0.4585645	286.574	0.6126595
5	0.1585015	0.4588428	312.7946	0.5912544
6	0.1596643	0.4555011	330.6349	0.5733903
7	0.1613413	0.4507666	343.3682	0.5581262
8	0.1632282	0.4455559	352.7812	0.5448701
9	0.1656321	0.4390889	358.8822	0.531661

Table (3) Heat exchanger and water injection (Pr=4)

T_{max} (K)	SFC (kg/kW.hr)	η_{th} (%)	Net Work (kJ/kg)	Work Ratio (%)
1100	0.2124907	0.342261	140.4008	0.4365955
1200	0.1941074	0.374675	169.6354	0.4835959
1300	0.1812029	0.413583	198.8701	0.5232731
1400	0.1716709	0.423643	228.1047	0.5573250
1500	0.1643626	0.442480	257.3393	0.5868366
1600	0.1585977	0.458564	286.574	0.6126595

Table (4) Reheat and water injection (T_{max}=1600 K)

Pr	SFC (kg/kW.hr)	η_{th} (%)	Net Work (kJ/kg)	Work Ratio (%)
4	0.3507607	0.2073416	324.8009	0.4641625

5	0.3142443	0.2314355	363.0045	0.6282129
6	0.2917124	0.2493116	391.5128	0.6149067
7	0.2763488	0.263172	413.7542	0.6034904
8	0.2651755	0.2742609	413.6671	0.5935003
9	0.2566752	0.2833436	446.4438	0.5846291

Table (5) Reheat and water injection (Pr=4)

T_{max} (K)	SFC (kg/kW.hr)	η_{th} (%)	Net Work (kJ/kg)	Work Ratio (%)
1100	0.4071908	0.1786073	169.908	0.4863837
1200	0.3872033	0.1878271	201.5703	0.5290678
1300	0.3735173	0.1947092	232.799	0.5647444
1400	0.3636795	0.1999763	263.701	0.5950978
1500	0.3563557	0.2040862	294.3506	0.621292
1600	0.3507607	0.2073416	324.8009	0.6441625

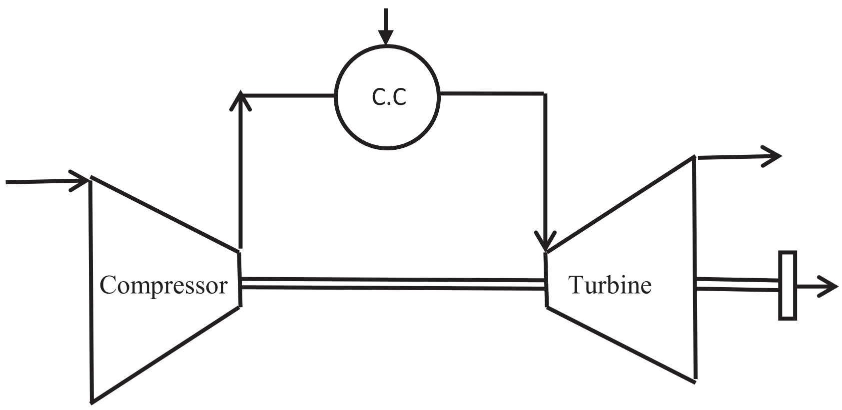

Figure 2 shows the basic components of simple cycle of gas turbine

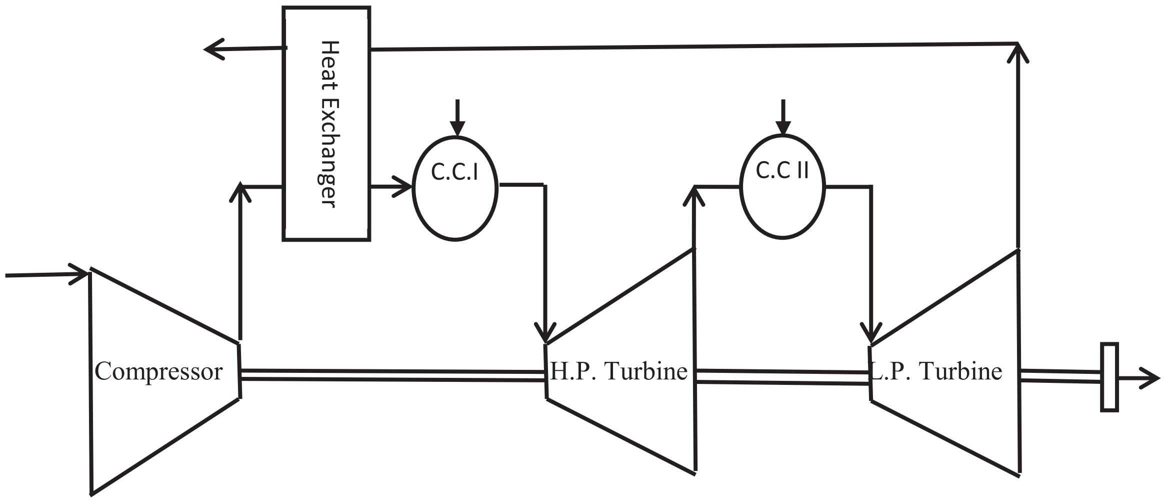

Figure 3 shows the modified cycle of gas turbine with reheat and heat exchanger

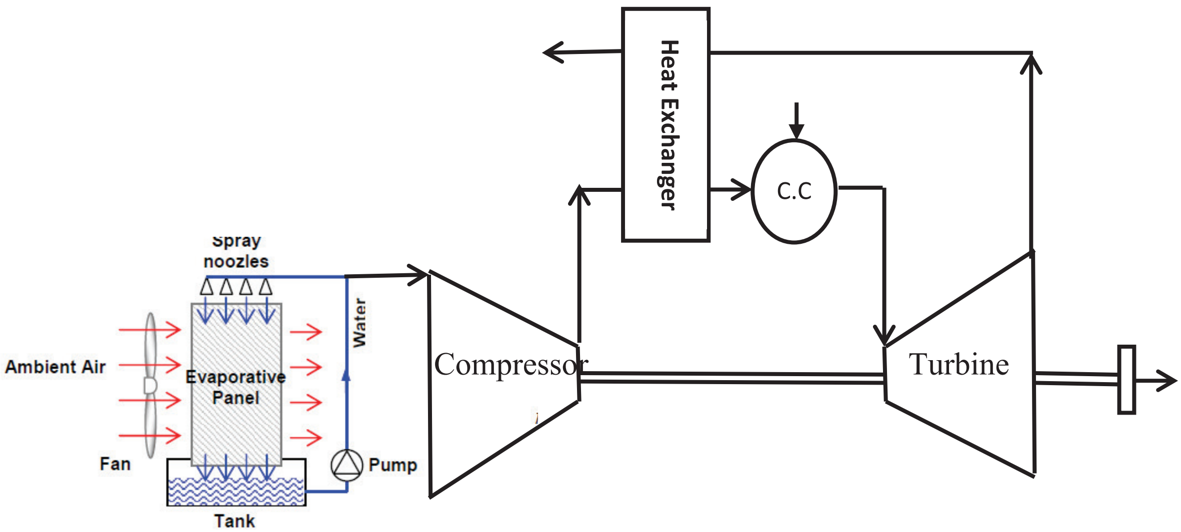

Figure 4 shows the modified gas turbine cycle with heat exchanger and water injection

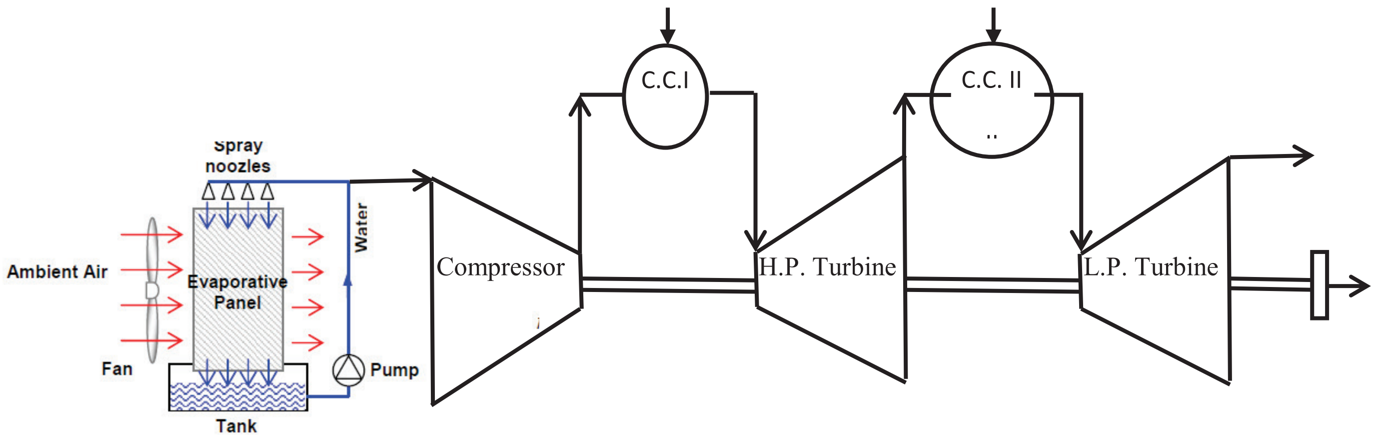

Figure 5 shows the modified cycle of gas turbine with adding two systems: reheat and water injection

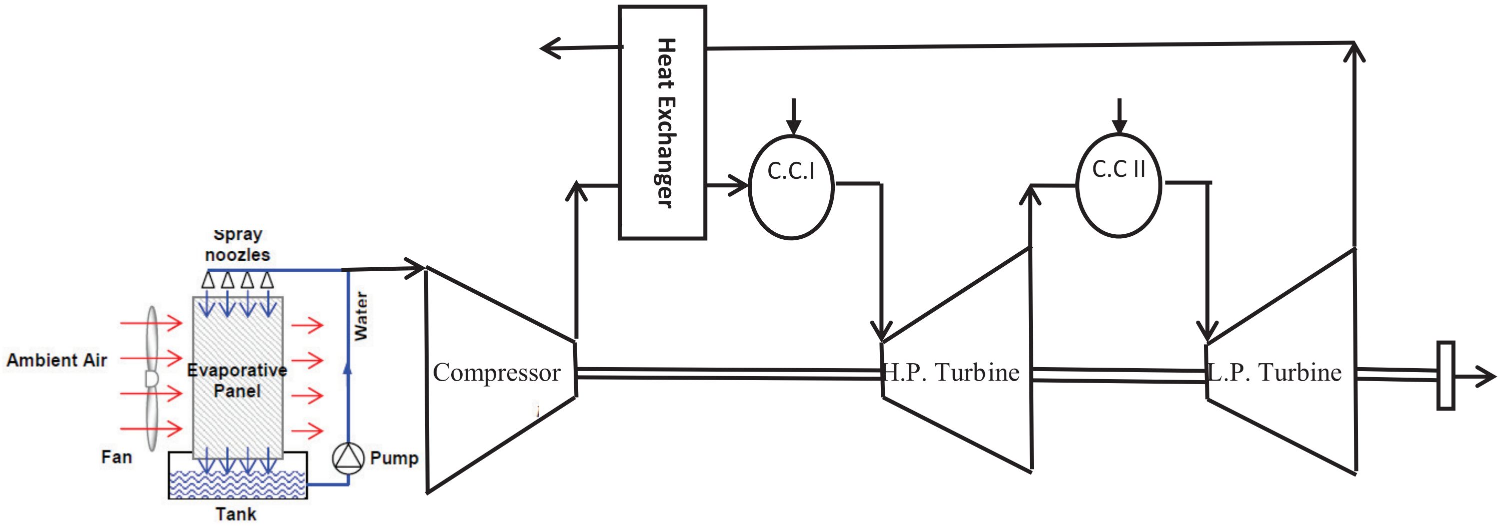

Figure 6 shows the gas turbine cycle considering all systems of adopted modifications: reheat, heat exchanger and water injection

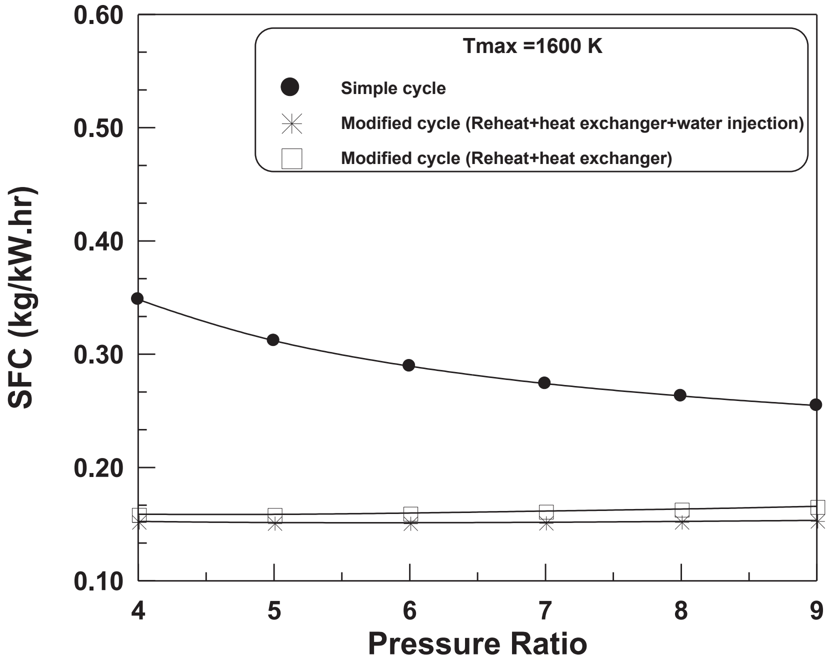

Figure 7 shows the specific fuel consumption (SFC) versus the pressure ratio for different types of adopted arrangements

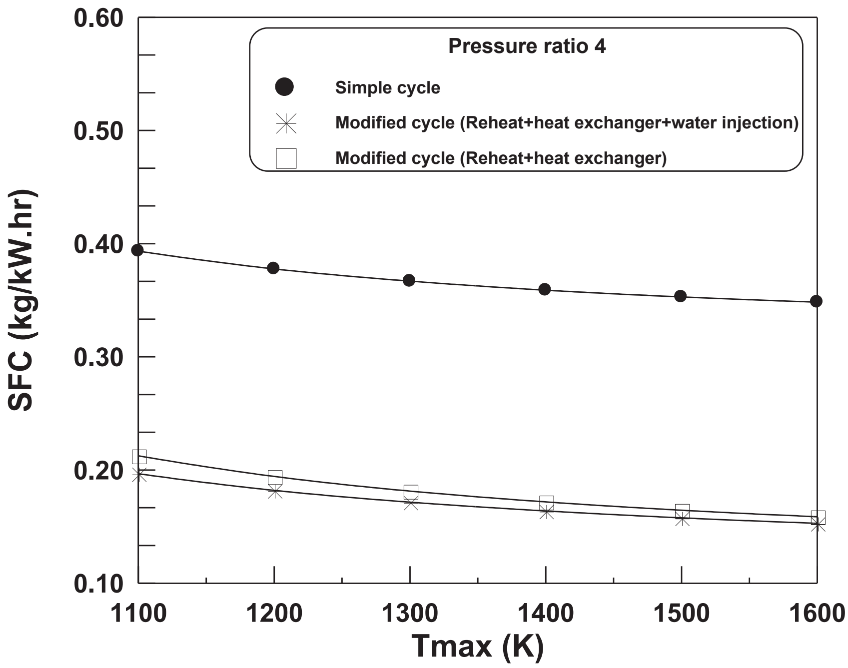

Figure 8 shows the specific fuel consumption (SFC) versus the maximum temperature in the cycle for different types of modifications

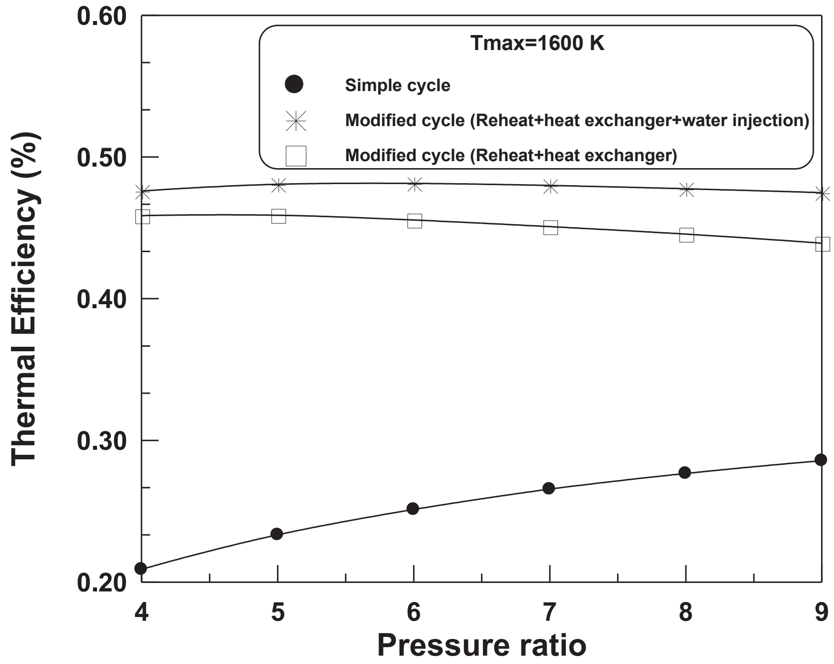

Figure 9 shows the thermal efficiency versus the pressure ratio for different types of adopted arrangements

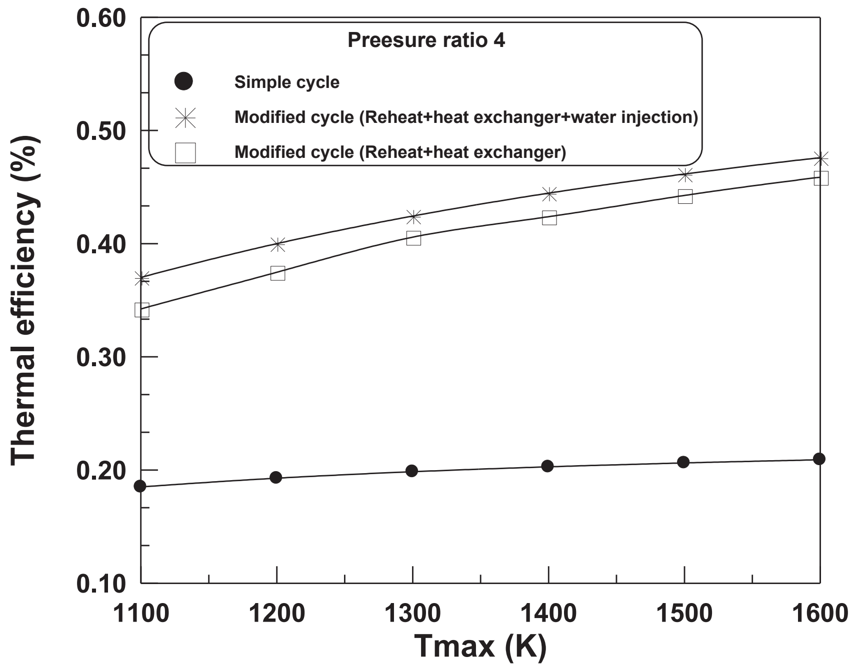

Figure 10 shows the thermal efficiency versus the maximum cycle temperature for different types of adopted arrangements

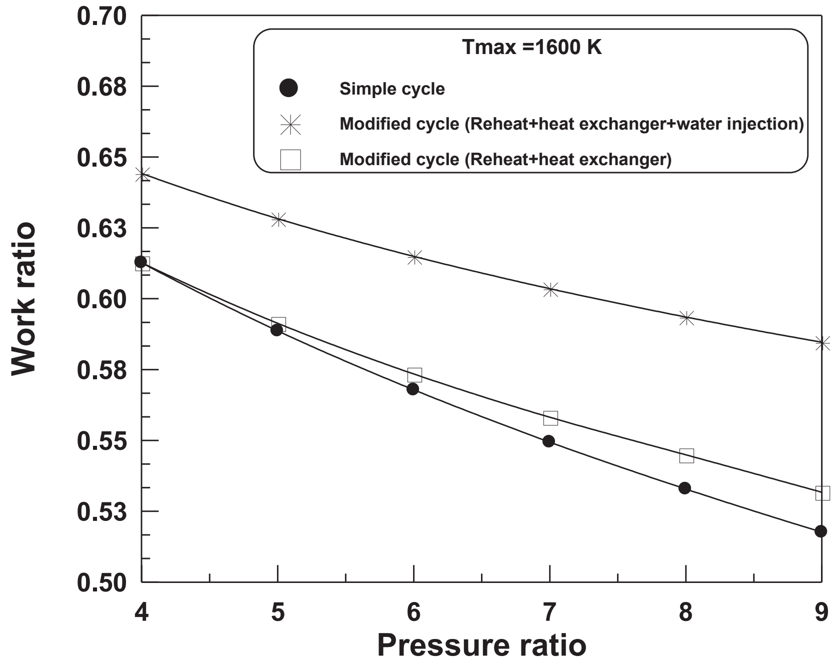

Figure 11 shows the work ratio versus the pressure ratio for different types of adopted arrangements

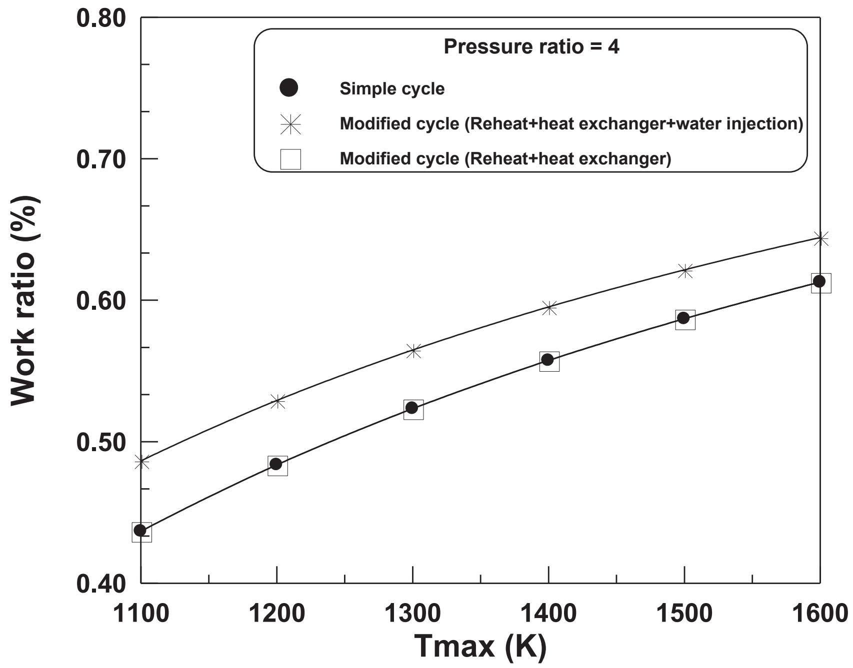

Figure 12 shows the work ratio versus the maximum temperature in the cycle for different types of modifications