Theoretical Study For The Effect Of Different Fuels On The Perfomance 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. It's well-known that thermal efficiency of open gas turbine varies with the fuel used in the combustion process. In this work an investigation for the effect of different hydrocarbon, alcohol and hydrogen fuels on the performance of simple and modified gas turbine has been studied. A computer program was written in Quick basic Language has been accomplished to calculate the thermal efficiency, work ratio, specific fuel consumption….etc. in gas turbine working with different types of fuels. It's found that specific fuel consumption is margined according to the fuel used because its dependency on the enthalpy of reaction of the fuel. Also it's noted that hydrogen fuel has higher thermal efficiency and lower fuel consumption than other fuels, but its requires high pressure to insert it to combustor. Therefore we must use methane as fuel in the plant since it is more common type than other.

الخلاصة

يتزايد انتشار التوربين الغازي يوما بعد يوم في توليد الطاقة الكهربائية وفروع الصناعة الأخرى. من المعروف ان الكفاءة الحرارية لدورة التوربين الغازي المفتوحة تتغير حسب نوع الوقود المستخدم في المحطة. تم في هذا البحث دراسة تاثير انواع مختلفة من الوقود على كفاءة وأداء دورة التوربين الغازي البسيطة والمطورة. حيث تم إعداد برنامج بلغة بيسك لحساب الكفاءة الحرارية ومعدل استهلاك الوقود ونسبة الشغل المنجز للتوربين الغازي الذي يعمل على انواع مختلفة من الوقود مثل الهايدروكاربونات بالإضافة الى الوقود الهايدروجيني كذلك تم دراسة تاثير نسبة الضغط ودرجة الحرارة العظمى على اداء المحطة. كما لوحظ إن نوع الوقود المستخدم يحدد كمية الاستهلاك النوعي له وذلك نتيجة لاعتماده على مقدار المحتوى الحراري لذلك الوقود.فقد تم التوصل الى ان الوقود الهايدروجيني يمتلك اعلى كفاءة مع اقل استهلاك نوعي ياتي بعده سائل الميثان والبروبين والبنزين ثم المجموعة الكحولية, لكن الهايدروجين يتطلب ضغط عالي لادخاله الى غرفة الاحتراق, لذا يجب استخدام الميثان كوقود بديلا عن الهايدروجين لكونه اكثر الانواع شيوعا واستخداما

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	-

Greek Symbols

Symbol	Definition	Unit
γ	Specific Heat Ratio	
η	Efficiency	
ΔH	Enthalpy of Reaction per unit Mass	*kJ/kg*

Subscript

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

Introduction

The wide use of electrical energy is one of the most important characters of each century and its large consumption considered and a feature of highly developed countries. Most of the governments and companies to complete the production of electrical energy and its distribution to help users employ it for different purposes: domestic, industrial and commercial. Although the thermal efficiency of closed gas turbine cycles was independent of the fuel used in the combustion process. **(leung E., 1985)** presented in his paper an investigation about the performance of open gas turbine. The principle result of his study was a correlation between the variation of the thermal efficiency with both hydrocarbon fuel and alcohol, and the fuel parameter C_1+C_2 taken from the equation of reaction. Practical and theoretical study of performance of simple gas turbine cycle for power generation performed by **(Badran, 1997)**, for two power stations “Rehab and Rasha stations” at different load. The result showed that an increase in compressor inlet temperature yields to decrease in the efficiency & an increase in turbine inlet temperature causes increased in the work which in turn increase in thermal efficiency and decreased in the fuel consumption. Also when pressure ratio increased, the thermal efficiency increased up to a maximu8m value and then started decreased indicating that there is an optimum pressure ratio for a given value of turbine inlet temperature. **(Rice, 1987)** studies in his investigation a heat balance method of evaluating various open gas turbine cycle systems based on the first law of thermodynamics. A useful solution is presented that can be applied to various gas turbine configurations. **(Lefebvre, 1985)** presented in his paper an analytical study for the effect of fuel properties on the performance and reliability of several gas turbine combustors. Its concluded that combustion efficiency depend on fuel chemistry, but are strongly influenced by the physical fuel properties that govern atomization quality and spray evaporation rates. **(Foster**, **1990**) studies an efficient, indirectly heated steam injected air turbine power fired with coal. The plant meet all environmental standards and generate about 35 kW. The plant offers high power without requiring condensing steam turbine and can operate efficiently. **(Schefer, 2002)** has studied the combustion process of hydrogen when blended with traditional hydrocarbon fuels in gas turbine combustors. The objectives of his work are:

1. Effect of the amount of hydrogen on flame stability, combustor acoustics, emissions and efficiency.

2. Establish a scientific and technological database for lean combustion of hydrogen-enriched fuels.

3. Establish the numerical simulation capabilities that will facilitate design optimization of lean premixed swirl burner. The approach was used to design and fabricate a lean premixed swirl burner that simulate the basic features of gas turbine combustors The results of study showed that ultra-lean premixed combustion is an effective approach to NOx emissions reduction from gas turbine engines. Hydrogen blended with traditional hydrocarbon fuels significantly improves flame stability during lean combustion and allows stable combustion at low temperature needed to minimize NOx production.

Hydrogen, as a carbon-free energy carrier, is likely to play an important role in a world with serve constraints on gas emissions. In the power industry, its utilization as gas turbine fuel can be proposed under several possible scenarios, depending on the mode of H_2 production **(Chiesa, P, 2005)**. For instance, hydrogen can be produced remotely from renewable energy sources (solar and wind) or from nuclear energy (via direct thermal conversion or by electrolysis!), but in a more realistic and near-term vision it will be derived from conventional fossil fuels by conversion process including CO_2 sequestration. Possible solutions include: (i) remote coal conversion to hydrogen (via gasification, shift, and separation from CO_2 and H_2 pipeline transport to the power station, (ii) integrated hydrogen and electricity production from coal or natural gas, exporting pure hydrogen to remote users on-side low- grade hydrogen to produce power **(Kreutz, T. G. et al,2002)**, (iii) electrical generation from combined cycles integrated to fossil fuel decarburization (applicable to coal, oil, or gas and CO_2) capture **(Lozza, G., and Chiesa, P.,2002),** Fuel cells and H_2-O_2 semi closed cycles may represent future options for power generation, but combined cycles coupled to H_2 production/ CO_2 sequestration processes can be proposed as a short/mid-term solution for massive gas emission reduction

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 are made under the following assumptions:

1.Both compression and expansion processes are isentropic.
2.No Kinetic energy change between inlet and outlet of each component.
3. The pressure drop through air inlet, combustion chamber, Nozzles…etc, are neglected.
4. Mass flow rate is constant through the cycle.

Simulation Of Simple Cycle

Fig.(1) show a simplified diagram of simple gas turbine which describes all components of the plant

1. Compressor: This part represented in process 1-2 of **Figure (1)** where the task of compressor is 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:

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

(1)

The isentropic efficiency of compressor 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)**.

$$\eta_c = \frac{T_2 - T_1}{T_2^/ - T_1}$$
(2)

So the work input to the compressor is:

$W_c = Cp_a * \left(T_2^/ - T_1\right) / \eta_m$; where η_m mechanical efficiency (3)

2. Combustion Chamber: This part is represented in process 2^--3 of (T-S) diagram. In **Figure (2)**. The combustion chamber is designed to burn a mixture of fuel and air to deliver the resulting 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)** .A schematic of combustion chamber is shown in **Figure (3)**. Since the process is adiabatic with no work transfer, so the energy equation, **(Cohen**, **1989)** is simply ;

$$\sum\left(m_i h_{i,3}\right) - \left(h_2^/ - f.h_{f,t}\right) = 0 \quad (4)$$

Now making the enthalpy of reaction at a reference temperature of 25 ^{o}c, so equation can be expanded in the usual way (**Cohen**, **1989)** to get;

$$\left(1 + f_t\right).Cp_g.\left(T_3 - 298\right) + f_t.\Delta H_{25} + Cp_a.\left(298 - T_2^/\right) + f_t.Cp_f\left(298 - T_f\right) = 0 \quad (5)$$

By simplifying equation (5) the theoretical fuel to air ratio will be:

$$f_t = \frac{Cp_g.\left(T_1 - T_3\right) + Cp_a.\left(T_2^/ - T_1\right)}{\left(Cp_g.\left(T_3 - T_1\right) + \Lambda H_{25}\right)} \quad (6)$$

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

$$f_a = f_t / \eta_b \quad (7)$$

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. This part is represented in process 3-4^- of (T-S) diagram In **Figure (2)**.

During the isentropic expansion process **(Easjop T.D.,1978)**:

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma_g - 1}{\gamma_g}} = (PR)^{\frac{\gamma_g - 1}{\gamma_g}} \qquad (8)$$

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

$$\eta_t = \frac{T_3 - T_4^{/}}{T_3 - T_4} \qquad (9)$$

Then the turbine work output is:

$$W_t = Cp_g * \left(T_3 - T_4^{/}\right) \qquad (10)$$

The net work output is determine by subtracting equation (10) from equation (3). The work ratio is defined as the ratio of net work output to turbine output work **(Easjop T.D.,1978)**:.

$$W_R = \frac{W_n}{W_t} \qquad (11)$$

The specific fuel consumption is given by;

$$S.F.C. = \frac{f_a \times 3600}{W_n} \qquad (12)$$

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

$$\eta_{th} = \frac{3600}{S.F.C. * \Delta H_{25}} \qquad (13)$$

The following Correlation was obtained to get maximum thermal efficiency **(leung E., 1985)**

$$\eta_{\max} = \left((25.47 + 29.2 * J) + (2.2 + 3.3 * J).\ln(C_1 + C_2)\right)\% \qquad \text{where } J = \ln\left(\frac{T_{\max}}{1000}\right)$$

(14)

The above equation examined the variation of maximum thermal efficiency with both hydrocarbon and alcohol fuels with fuel parameter $C_1 + C_2$ taken from the equation of reaction:

$$C(fuel) + O_2 \rightarrow C_1(CO_2) + C_2(H_2O) \qquad (15)$$

Simulation Of Modified Cycle

Possible modification to the basic cycle can be made **(Cohen**, **1989)**, such as Intercooling (use more compressor), reheating (use more combustor), and heat exchanger which uses some of the energy in the turbine exhaust gases to preheat the air entering the compressor. In this work we should summarized these modification as follows:

1. Intercooling: When the compression process is performed in two stages with an intercooler, then the work input for a given pressure ratio and mass flow is reduced. The task of intercooler is to reduce the outlet temperature of low pressure compressor which is equal to the temperature inlet to high pressure compressor at the same pressure. The system of intercooling and (T-S) diagram are shown in **Figure (4) and Figure (5)** respectively. To get the work input is a minimum, we must make the pressure in each stage of compression is the same, when the temperature of air is cooled in the intercooler, back to the inlet to the unit (plant) So:

$$T_3 = T_1 \qquad (16)$$

During the isentropic processes (1-2^-) & (3-4^-)

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} \qquad (17)$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma_a - 1}{\gamma_a}} = (PR)^{\frac{\gamma_a - 1}{\gamma_a}} \quad \& \quad \frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma_a - 1}{\gamma_a}} = (PR)^{\frac{\gamma_a - 1}{\gamma_a}} \qquad (18)$$

The isentropic efficiencies of low and high pressure compressors is given below:

$$\eta_{c1} = \frac{T_2 - T_1}{T_2^{/} - T_1} \quad \& \quad \eta_{c2} = \frac{T_4 - T_3}{T_4^{/} - T_3} \qquad (19)$$

The total work of compressors is calculated from the equation;

$$W_{ctot} = Cp_a * \left(T_2^{/} - T_1 + T_4^{/} - T_3\right) / \eta_m \qquad (20)$$

2. Reheat: The expansion process is very frequently performed in two separate stages. The high pressure turbine driving the compressor and low pressure turbine providing the useful power output. The work output of the low pressure turbine can be increased by raising the temperature at the inlet of the stage. This can be done by placing a second combustion chamber between the turbine stages **(Cohen**, **1989**), in order to heat the gases leaves the high pressure turbine. This system with the (T-S) diagram are shown in **Figure.(6), and Figure (7)**. The high pressure turbine must be exactly equal to the work input for the compressor with the following equation

$$Cp_a . \frac{\left(T_2^{/} - T_1\right)}{\eta_m} = Cp_g . \left(T_3 - T_4^{/}\right) \qquad (21)$$

For the first combustion chamber, the energy equation is used (**Cohen**, **1989**)

$$\sum\left(m_i h_{i,3}\right)-\left(h_2^{/}-f.h_{ft1}\right)=0 \qquad (22)$$

The first theoretical fuel/air ratio is calculated from expanding equation (22) as follows

$$\left(1+f_{t1}\right).Cp_g.\left(T_3-298\right)+f_{t1}.\Delta H_{25}+Cp_a.\left(298-T_2^{/}\right)+f_{t1}.Cp_f\left(298-T_f\right)=0 \qquad (23)$$

The same procedure is used for second combustion chamber;

$$\sum\left(m_i h_{i,4}\right)-\left(h_4^{/}-f.h_{ft2}\right)=0 \qquad (24)$$

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

$$\left(1+f_{t2}\right).Cp_g.\left(T_5-298\right)+f_{t2}.\Delta H_{25}+Cp_g.\left(298-T_4^{/}\right)+f_{t2}.Cp_f\left(298-T_f\right)=0 \qquad (25)$$

Then the total fuel/air ratio is given by;

$$f_{t,tot}=f_{t1}+f_{t2} \qquad (26)$$

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

$$W_n=W_{t2}=Cp_g.\left(T_5-T_6^{/}\right) \qquad (27)$$

3.Heat Exchanger: 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.(8) and Figure (9)**, so the ideal heat exchanger have ($T^{/}_2$=T_6) and (T_3=$T^{/}_5$) which is assumed in this work.
The other performance parameters are repeated with the same procedure applied in simple cycle.

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 of open gas turbine plant respectively. In this work we studied different types of fuels which are (H_2, CH_4, C_3H_6 , C_6H_6, CH_3OH) that are used in gas turbine cycle. **Figure (10)** shows the relation between maximum plant temperature with different types of fuels. Equation (14) is used to draw this plot. It can be seen that when maximum cycle increased, the maximum efficiency also increased. When we make comparison between the fuels used, it can be noted that hydrogen and methyl alcohol have the same higher maximum efficiency than other fuels at the same operating conditions, this is because they have greater fuel parameter (C_1+C_2), see table (1). **Figure (11)** shows the effect of maximum cycle temperature which was varied from (1100-2000) K^o on the thermal efficiency for all fuels considered. In general the thermal efficiency increases with increase in temperature of cycle. The hydrogen curve is on the top of the plot because it has higher enthalpy of reaction (lower calorific value) then methane…., with methyl alcohol which came last because it has lower energy than other fuels. **Figure (12)** represent the relation between maximum cycle temperature with the specific fuel consumption. It can be noted that hydrogen fuel has minimum specific fuel consumption than other fuels and this expected because hydrogen has higher thermal efficiency and according to equation (13) hydrogen fuel must has minimum specific fuel consumption followed by methane, propen, benzene, and finally methyl alcohol. Now the graphs of thermal efficiency and specific fuel consumption are repeated for all fuels selected, but for variable pressure ratio (4-9), and for fixed maximum temperature 1600 K^o, see **Figure (13) and Figure (14).** When pressure ratio is increased, the thermal efficiency increased for all fuels but hydrogen and methane fuels remain in the top of the plot. In the graphs of modify open gas turbine power plant which include (intercooling, reheat, and heat exchanger), Only hydrogen and methane fuels is graphed, since methane fuel is the more common type used in gaseous plant, and hydrogen is promising fuel which have maximum efficiency, and minimum fuel consumption than other fuels. The use of hydrogen as a fuel appears to promise a significant improvement in performance of gaseous plants and reduces the emissions of Greenhouse gases, nitrogen oxide, and smoke. Another driving force behind the need to use hydrogen fuels is the rapid depletion rate of currently used fossil fuels. **Figure (15) and Figure (16)** shows the effect of maximum temperature and pressure ratio on the thermal efficiency and S.F.C. of intercooling cycle operating on hydrogen and methane fuels respectively. It can be noted from **Figure (15)** the thermal efficiency increase with increase in maximum temperature for a fixed pressure ratio **(PR=4)**, this increase is greater than in simple cycle about **12 %** . Also there is a reduction in S.F.C. bout **35%** as compared with simple cycle this because increase in net work and thermal efficiency for both fuels selected. **Figure (16)** shows the effect of pressure ratio variation on the thermal efficiency and specific fuel consumption with maximum temperature equal to **1600 K^o**.. Its observed that when pressure ratio increased, the thermal efficiency increased, until reaches to maximum value then starts to decrease indicating that there is an optimal pressure ratio equal **7** at which the maximum thermal efficiency is **0.360255** and specific fuel consumption is **0.08327 (kg/kW.hr)** for hydrogen fuel. The same behavior is observed for methane fuel when **PR=7**, thermal efficiency and S.F.C. are **0.3537 & .20557 (kg/kW.hr)** respectively. **Figure (17) and Figure (18)** shows the effect of maximum temperature and pressure ratio on the thermal efficiency and S.F.C. of reheating cycle operating on hydrogen and methane fuels respectively.. In the reheating cycle the increase in thermal efficiency is lower than the increased in the intercooling cycle. Also there is a reduction in the S.F.C. about **17 %** this is due to increase in the net work output and hence in the thermal efficiency. It can be seen from **Figure (18)** that thermal efficiency increased with increased in pressure ratio, also a reduction in S.F.C. is obtained. In the case of heat exchanger cycle the thermal efficiency is increased with increase in the maximum temperature reaches to **0.5233009 at 2000 K^o** for a fixed pressure ratio equal to 4, but this behavior is opposite when pressure ratio varied and maximum temperature fixed to **1600**

K°. Its observed maximum efficiency equal to **0.4760761 at PR= 4** then starts to decrease when pressure ratio increased. This is clearly observed in **Figure (19) and Figure (20)** respectively. **Figure (21)** explain the effect of maximum temperature on the work ratio for simple and modified cycle of gas turbine power plant operated with hydrogen fuel. It can noted from this figure that maximum work ratio occurred with the intercooling cycle followed by reheating cycle, then simple and heat exchanger cycle which are have the same curve because the condition of the ideal heat exchanger haves ($T'_2=T_6$) and ($T_3=T'_5$) see **Figure (8)** which gives the same turbine work and hence gives the same work ratio

Conclusion

The following conclusions can be drawn from the present work;

Hydrogen fuel is ideal promising fuel in the gaseous plant which has greater thermal efficiency, and hence minimum S.F.C. than other selected fuels. Greater improvement in the performance of modified gas turbine power plant occurred with intercooling and heat exchanger rather than simple and reheat cycle.

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Table (1) Fuels Input Data

Type of fuel	Phase	Lower calorific value (kJ/kg)	C_1+C_2
H_2	gas	120000	2
CH_4	liquid	49500	1.5
C_3H_6	liquid	45000	1.333
C_6H_6	liquid	40000	1.20
CH_3OH	liquid	19700	2

Table (2) Operating Conditions of the Plant

Compressor inlet temperature (T_1)	298 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)
Isentropic efficiency of compressor (η_c)	85 %
Isentropic efficiency of turbine (η_t)	87 %
Mechanical efficiency	98 %
Efficiency of burner	98 %
Fuel temperature (T_f)	298

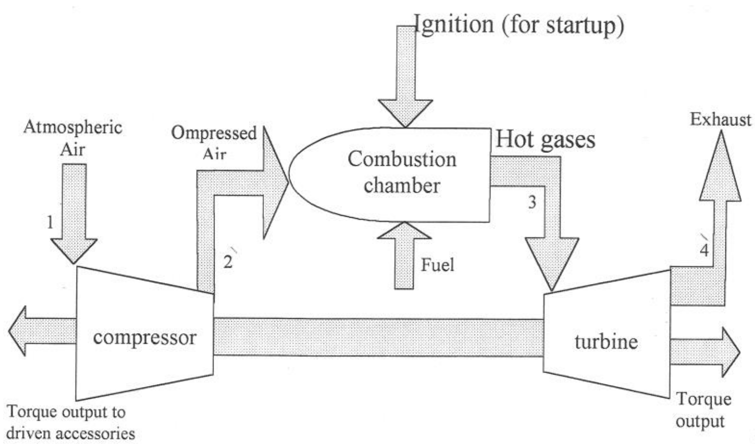


Fig.(1) Simplified diagram of simple gas turbine Power plant (Saba Y., 2000)

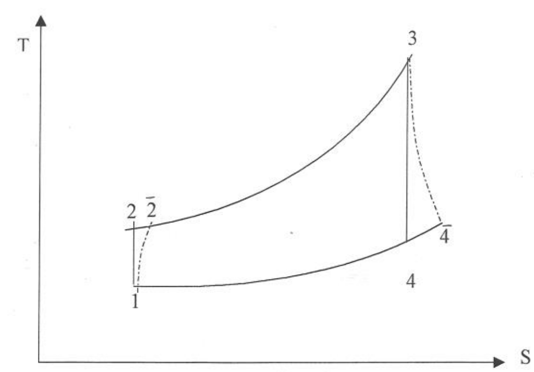


Fig.(2) (T-S) diagram of simple gas turbine cycle

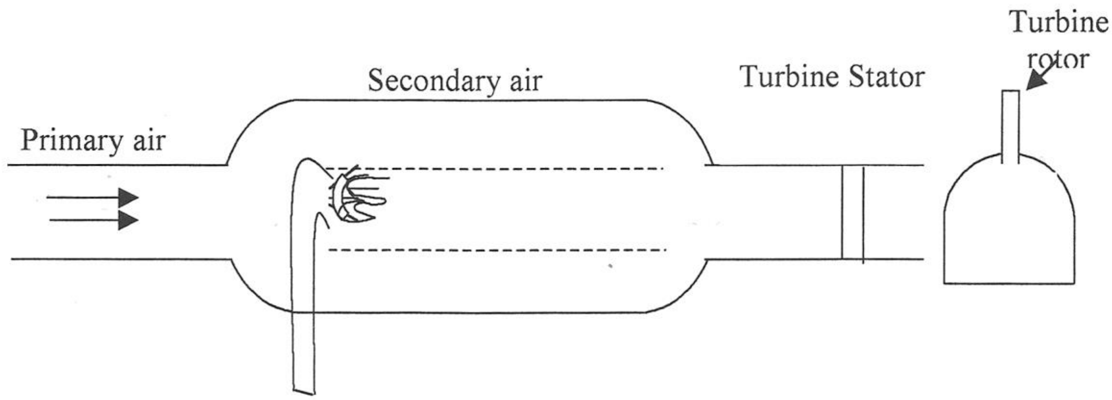
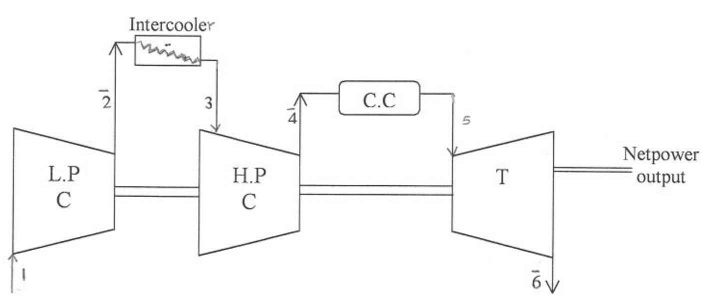


Fig.(3) A schematic of combustion chamber in gas turbine power plant (Saba Y., 2000)



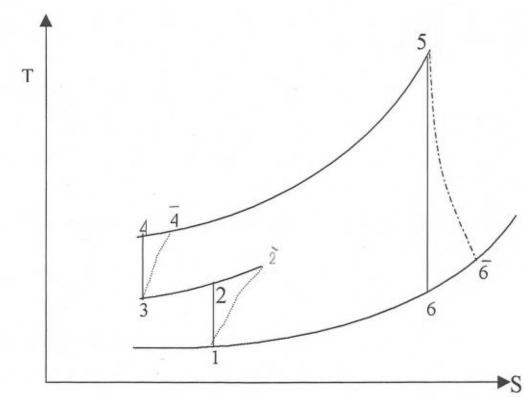
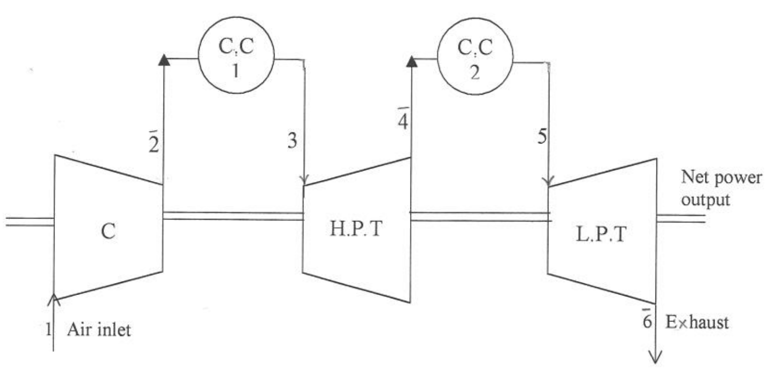


Fig.(4) Simplified diagram of Intercooling gas turbine Power plant

Fig.(5) (T-S) diagram of intercooling cycle



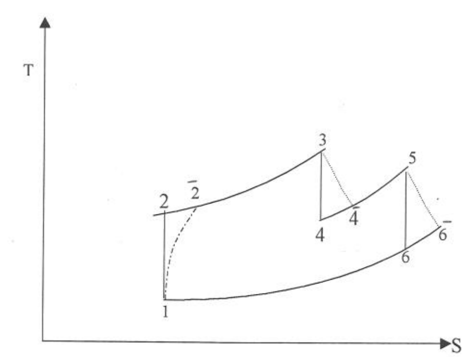


Fig.(6) Simplified diagram Reheating gas turbine Power plant

Fig.(7) (T-S) diagram of reheating cycle

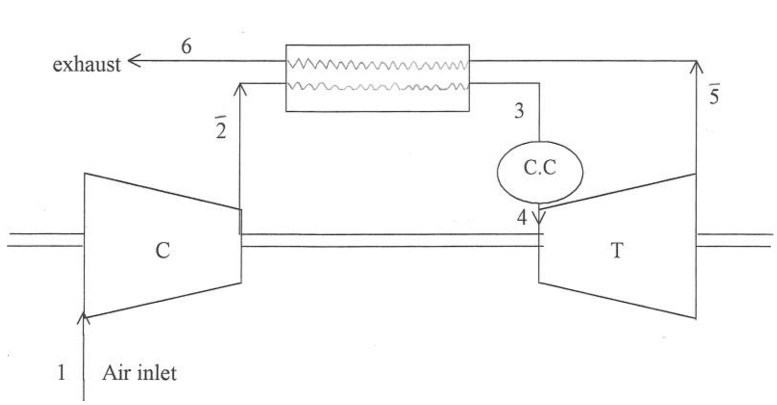


Fig.(8) Simplified diagram Heat exchanger gas turbine Power plant

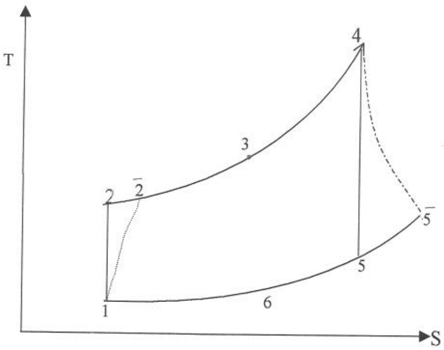


Fig.(9) (T-S) diagram of Heat exchanger cycle

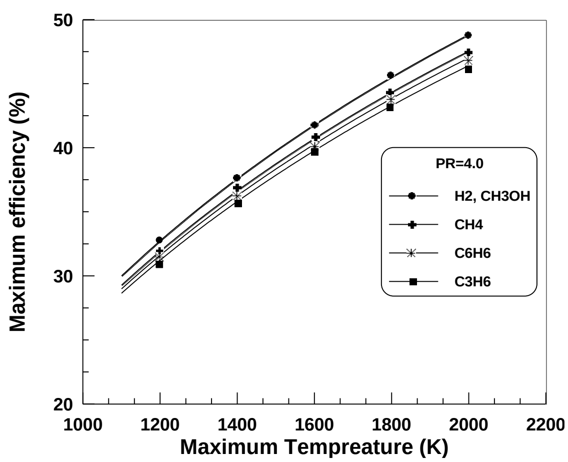


Fig.(10) Effect of maximum cycle Temperature on the max. Efficiency for simple cycle

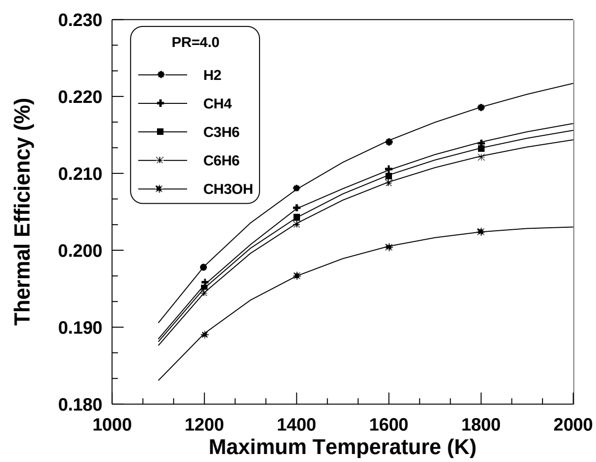


Fig.(11) Effect of maximum cycle Temperature on the thermal Efficiency for simple cycle

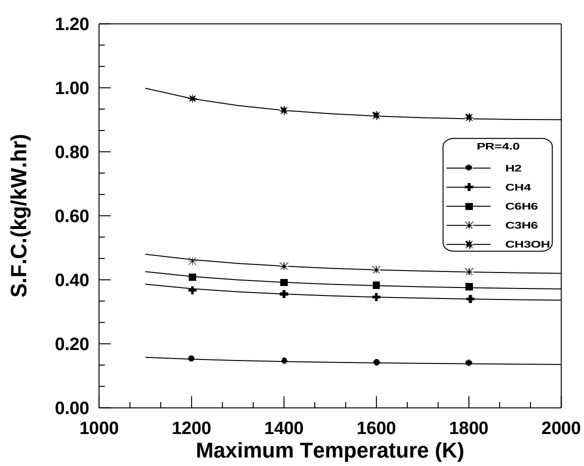


Fig.(12) Effect of maximum cycle Temperature on the S.F.C. for simple cycle

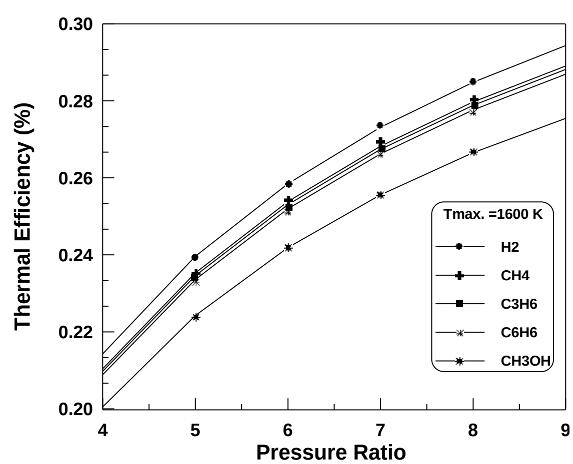


Fig.(13) Effect of Pressure ratio on the thermal Efficiency for simple cycle

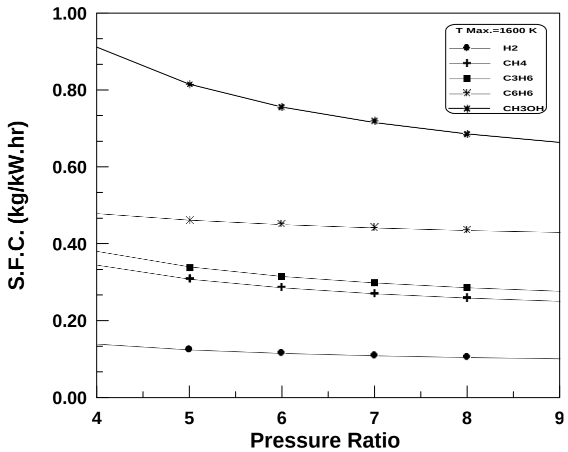


Fig.(14) Effect of Pressure ratio on the S.F.C. thermal for simple cycle

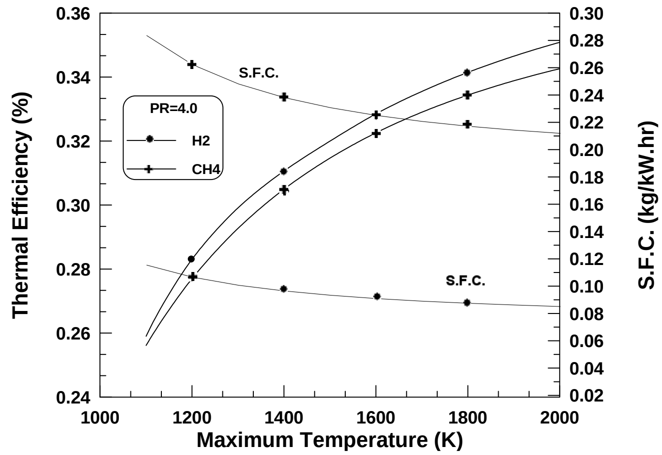


Fig.(15) Effect of Max. Temperature on the efficiency & S.F.C. for intercooling cycle

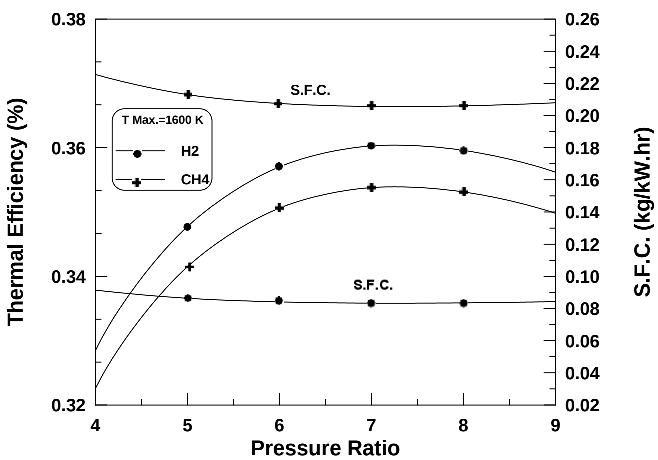


Fig.(16) Effect of Pressure ratio on the S.F.C. effi. for intercooling cycle

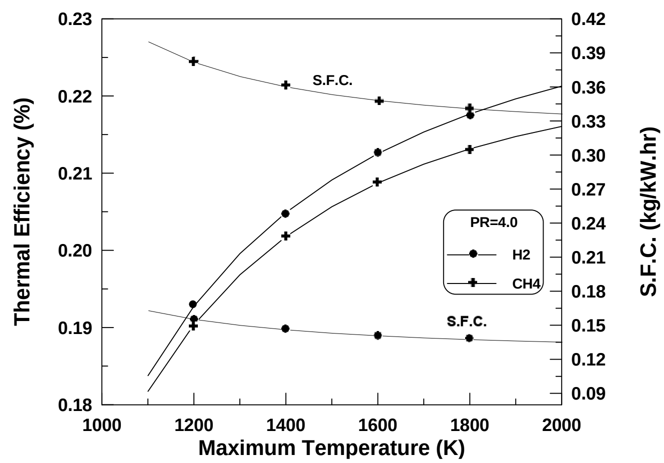


Fig.(17) Effect of Max. Temperature on the thermal thermal efficiency & S.F.C. for reheating cycle

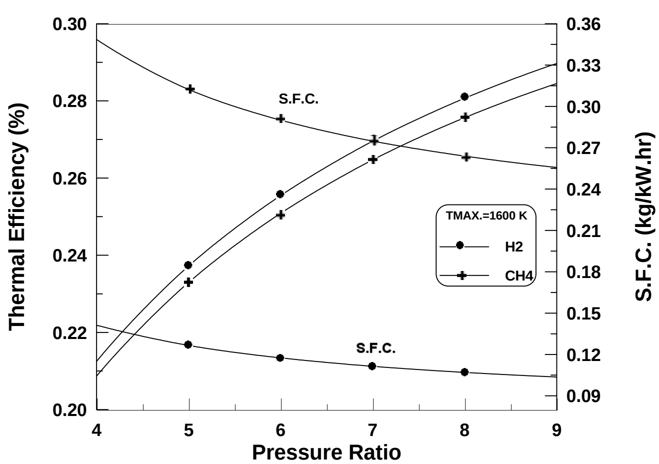


Fig.(18) Effect of Pressure ratio on the S.F.C. effi. for reheating cycle

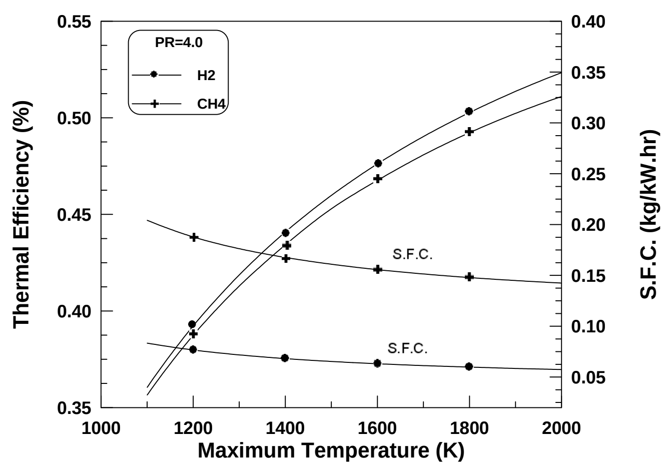


Fig.(19) Effect of Max. Temperature on the thermal thermal efficiency & S.F.C. for heat exchanger cycle

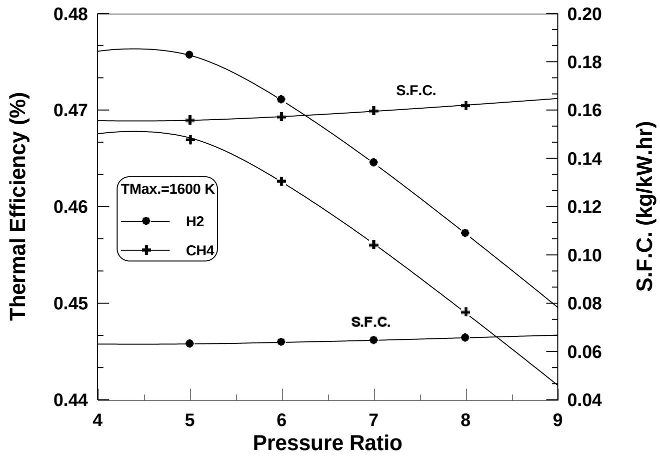


Fig.(20) Effect of Pressure ratio on the S.F.C. & efficiency for heat exchanger cycle

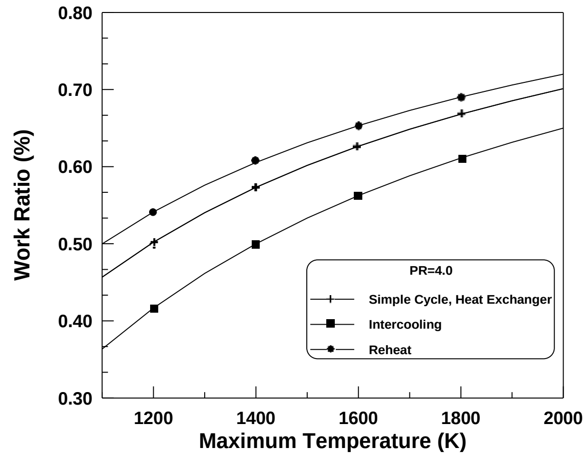


Fig.(21) Effect of Max. temperature on the work ratio for simple and modify gas turbine cycle