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THERMODYNAMIC ANALYSIS FOR COMBINED BRAYTON / RANKINE POWER PLANT

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This paper deals with parametric of thermodynamic analysis of a gas power plant has 165.1 MW rated power capacity. This plant established as a gas power plant in Mosrata, (Libya) to feed an iron and steel factory. The thermodynamic analysis energy and exergy analysis indicate how much power has been rejected in the exhaust gases in case of gas power plant only. In order to enhance the exergy and energy efficiencies, this study propose a steam unit as a (combined power plant). The gas plant was designed to work under different part load of 0.2, 0.4, 0.6, 0.8, and at full load. Then, the thermodynamic energy and exergy analysis of the plant has been carried out. The energy and exergy efficiencies were calculated according to the first and the second laws of thermodynamics. It is concluded that the overall thermal efficiency can be improved by 9.835% and the exergy can be enhanced by 9.34% at full load.

KEYWORDS: *Energy; Exergy; Second law analysis; Power plant; Combined cycle; Pitch point*

NOMENCLATURE

C	Velocity, m/s	**Q**	Heat, kJ
cp	Specific heat, kJ / kg.K	**R**	Gas constant, kJ/kg.K
G	Gravity acceleration , m/s^2	**S**	Entropy, kJ/kg.K
h	Enthalpy , kJ / kg	**T**	Temperature, K
I	Irreversibility, kJ / kg	**W**	Work, kW.h
k	Process adiabatic exponent	**z**	Static head , m
$\dot{m}$	Mass flow rate, kg/sec	η	Efficiency
Ncv	Natural gas calorific value, kJ/kg	ϕ	Energy, kJ/kg
P	Pressure, bar	ψ	Exergy, kJ/kg

SUBSCRIPTS

0,1,2,3,4	for expansion points	loss	for losses
a	for air	loss T	for losses of turbine
add	for add heat	N	for net work
ce	for superheating temperature	Ng	for natural gas
ci	for water temperature entering the economizer	O	for oxygen
ch	for chemical	oo	Partial
cond	for condenser	p	for pump
cy	for cycle	rev	for reversibility

Ex	for exergy	S	for isentropic
f	for friction	serr	for surrounding
GT	for gas turbine	ss	for saturation steam temperature
g	for gas	sw	for saturated water temperature
hi	for high temperature of gas	sw.ec	for steam entering the economizer
he	for exhaust temperature of gas	ST	for steam turbine
ht	for high temperature	T	for turbine
hw	for hot water	x	for pitch point temperature
i	for points 1,2,3	w.h.b	for waste heat boiler
irr	for irreversibility		

1. INTRODUCTION

Combined cycle power plants continue to gain increasing acceptance throughout the world, over other energy conversion systems. In fact, gas turbines/combined cycle power plants are now referred to as the power plant of 21st century. Energy economics is a broad field. Extensive work has been done that combined energy and economics, today generally accept like energy costs, energy price, etc. The exergy study gives an indication to exergy cost saving by using different delivers for the power plant wastes.

This study deals with an established gas power plant can be converted to combined unit in order to increase the plant generated power as its income and reduce the environmental pollution in the zone.

The following points are considered when the present works were done:

1. We must be clear that, we mean when we discuss the thermodynamics efficiencies and losses.
2. At all time we ensure that measures we used for the determining the efficiencies and losses.
3. We use the utilize efficiency and losses for determining the exergy.
4. The exergy measurement is taken based on economics including quantities and costs to give the sense of values measured.

An exergy analysis of gas side and added steam side as, a binary plant, was studied with effect of different available parameters which affecting on efficiencies and exergies of the whole plant.

The early development of gas-steam turbine was described by Sieppel and Bereuter [1]. Czermak and Wunsch [2] carried out the elementary thermodynamic analysis for a practicable Brown Boveri 125 MW combined gas/steam turbine power plant. Wunsch [3] reported that the efficiencies of combined gas/steam plants were more influenced by the gas turbine parameters maximum temperature and pressure ratio than by those for the steam cycle, and also reported that the maximum combined cycle efficiency was reached when the gas turbine exhaust temperature is higher than the one corresponding to the maximum gas turbine efficiency Horlock [4] based on thermodynamic considerations outlined more recent developments and future prospects of combined cycle power plants . Wu [5] describe the use of intelligent computer software to obtain a sensitivity analysis for the combined cycle. Cerri [6] analyzed the combined gas steam plant without reheat from the thermodynamic point of view.

Andriani et al [7] carried out the analysis of a gas turbine with several stages of reheat for aeronautical applications. Polyzakis [8] carried out the first law analysis of reheat industrial gas turbine use in combined cycle and suggested that the use of reheat is a good alternative for combined applications. Rosen. [9] performed a best manner used to analyze the power plants with high quality from energy, the second law of thermodynamics permits the definitions which called the exergy as a maximum amount of work that can be produced from the energy of any power system. El-Dib [11] perform an exergy analysis technique has been applied for different applications as power plants with different types such as , steam , gas , solar , refrigeration nuclear and others [12 – 23] . Macchi and Chiesa [24], El Masri [25], Bannister et al. [26], Rice [27–29], Gambini et al. [30], Bhargava and Perotto [31], Poullikkas [32] have studied and predicted performance of reheat gas turbine using air as coolant. [33,34 and 35] have perfomed exergy – energy analysis and thermodynamic evaluation for different types of steam and gas power stations. In the light of the above works, the present work aims to identify and quantify the sources of losses in a selected configuration combined cycle with different means of loading rates. A trial can be introduced for minimizing these losses to achieve maximum efficiency of this combined cycle.

2. THERMODYNAMIC PRINCIPLES AND ANALYSIS

One of the important concepts in the second law of thermodynamics applications is the reversible work of the processes. The reversible work is the maximum work that must be supplied, thus the Second law of thermodynamics will used to analysis the choiced plant, combined power plant, as a system and its components at the steady state condition.

The necessary thermodynamics principles will be formulated in order to develop such relations. This first aim of the present work is perform a simple checking for the system according to the First law of thermodynamics as a whole system component.

The Second law of thermodynamics is performing mathematical forms to calculate the heating power of the whole system components during irreversible processes in the compressor. The entropy changing between two states, (1) at entrance and (2) at exit may be explained as the sum of two entropy exchanged as ($\Delta S_{1,2}$) and entropy production ($\Delta S_{irr1,2}$) [12,17] :

$$\Delta S_{12} = \Delta S_{hi\,1,2} + \Delta S_{irr1,2} \tag{1}$$

$$\Delta S_{hi\,,12} = \int_1^2 \sum_i \frac{\delta Q_i}{T_{h\,,i}} \tag{2}$$

$$\Delta S_{irr,12} = \int_1^2 \sum_i \delta Q_i \left[\frac{1}{T_{hi}} - \frac{1}{T} \right] + \int_1^2 \frac{\delta W_f}{T} \tag{3}$$

Where :

Q = Heat exchange,

T_{hi} = Temperature of heat reservoir exchanging heat with the system,

T = Temperature of the system,

W_f = Work to overcome friction,

i = 1 , 2 , 3 , ……

Equation (1) may be considered as a mathematical form of the second law of thermodynamics [12]. For actual thermodynamic processes and cycles, the entropy production is always positive and is a measure of the resulting irreversibility (I_{12}) which may be calculated as [12, 18]

$$I_{12} = T_o \, \Delta S_{irr,12} \tag{4}$$

Where T_o is sink or surrounding temperature.

As the main object of the present work is to enhance of the thermal efficiency of the choiced gas power plant by using an auxiliary steam power unit consumed the heat reject in the exhaust gases of the gas turbine. Figure (1) shows a schematic diagram of a compound – cycle system as a binary cycle.

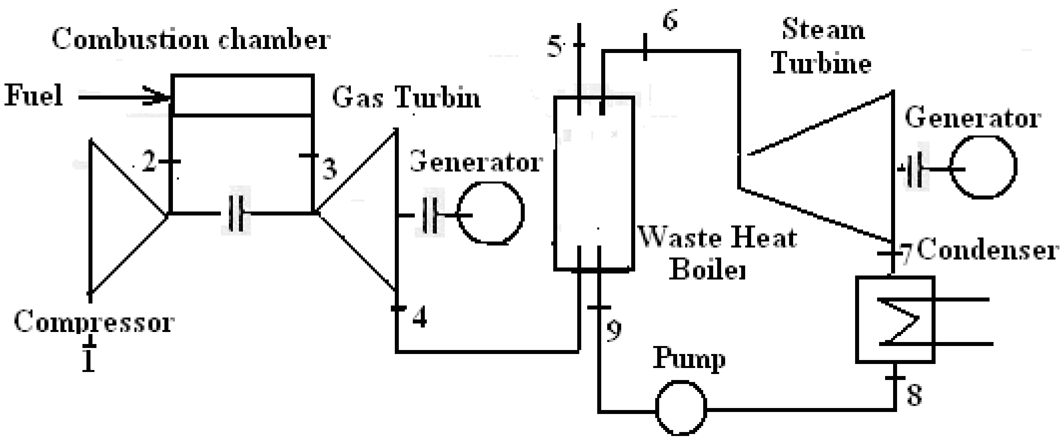

Fig. 1: Choice Combined Cycle Used

The thermal efficiency of the over combined cycle system becomes:

$$\eta_{cy} = \frac{W_{GT} + W_{ST}}{Q_{add}} \tag{5}$$

The ratio of the steam flow rate to the gas flow rate can be obtained from the balance of heats in the economizer, evaporator and superheater as shown in Fig. (2) as:

$$\frac{m_s}{m_g} = \frac{cp_g (T_{hi} - T_{he})}{h_{ce} - h_{sw.ec}} = \frac{cp_g (T_{hi} - T_x)}{h_{ce} - h_{hw}} \tag{6}$$

Where

$\frac{m_s}{m_g}$ = steam mass flow rate to gas mass flow rate

cp_g = gas specific heat, kJ/kg.K

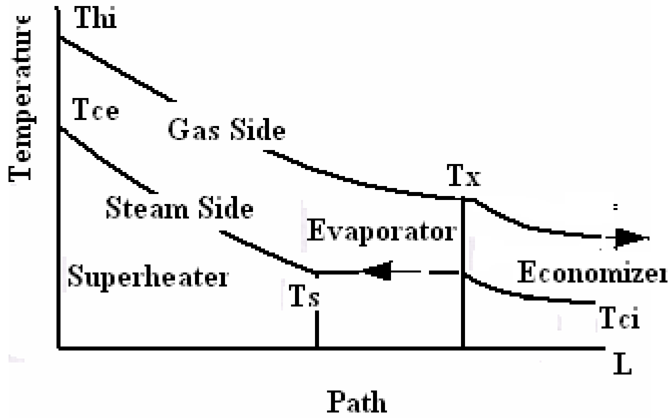

Fig. 2: Temperature Variation Along the Waste Heat Boiler Paths

T_{hi} = exhaust gas temperature, K
T_x = Pitch point temperature, exhaust gas exit temperature, K
h_{ce} = Super heated steam enthalpy , kJ/kg
$h_{h.w}$ = hot water enthalpy , kJ/kg
h_{sw} = Saturated water enthalpy, kJ/kg.
$h_{.sw.ec}$ = enthalpy of saturated (subcooled) water entering the economizer, kJ/kg

To avoid stack material from the chemical reaction between the stack metal and the exhaust gasses because this reaction will causing destructive corrosion with the stack walls. The final flow gas temperature must not to be below that 170 °C and the pitch point is 20 °C differences between the saturated liquid and the exhaust gasses [19] then the temperature of the pitch point temperature T_x can calculated by equation (6). The temperature variation along the waste heat boiler paths is explained in Fig. (2).

3. EXERGY

For steady state flow open system the specific exergy of the system can be determined as:

$$\psi = h_1 - h_o + \frac{C_1^2}{2} + g\,z - T_o(s_1 - s_o) \qquad (7)$$

Where

g = gravity acceleration, m/sec^2
z = static head, m

Whereas the rest which is not capable of doing work is termed the anergy (ϕ) may be expressed at point (1) as

$$\phi = h_o + T_o(s_1 - s_o) \qquad (8)$$

3-1 Chemical Exergy of Fuel

Chemical exergy of fuel is equals to maximum amount of heat obtained when the fuel burned with complete combustion under chemical reaction with oxygen In such processes , the initial state is surrounding state defined by T_o , P_o , and then the chemical exergy of fuel is expressed as :

$$\psi_{ch} = -\Delta G_o + RT_o[(X_{o_2} \ln(\frac{P_{oo,O_2}}{P_o}) - \sum_k x_k \ln(\frac{P_{oo,k}}{P_o})] \quad (9)$$

Where:

ΔG_o = Gibbs function of complete chemical reaction of fuels refer to surrounding state equal to H_o - T_o S_o

R = gas constant = 0.287 kJ/kg.K and subscript K refers to the component of product of combustion.

Liquid, gas, and industrial fuels are mixture of numerous chemical components of, usually, unknown nature. Szragut and Styry (see [20]) assumed that the ratio of chemical exergy of fuel ψ_{ch} is equal to the net calorific value of that fuel NCV for pure chemical substance having the same ratios of constituent chemicals This ratio denoted by $\epsilon = \frac{\psi_{ch}}{NCV}$ estimated by as $\pm$ 0.38 % .

3-2 Exergy Loss

The exergy loss for any irreversible process is obtained through exergy balance of this system when operate at steady state flow open cycle. The exergy loss is equal to the irreversibility for such which can be calculated using equation (4).

3-3 Exergy of Gas Turbine Power Plant

3-3-1 Gas Turbine Exergy Efficiency

Let us consider the perfect gas expand used in the gas turbine with ideal constant specific heat in adiabatic turbine. The entrance condition P_1, T_1 and the gas expand to the local atmospheric condition P_0, T_0 with ignoring the kinetic and potential energies. Now we need to evaluate the turbine performance by means exergy method. For stationary turbine the exergy method gives maximum work output as expressed in Eq. (10):

$$W_{rev} = \dot{m}_g[\psi_1 - \psi_2] = \dot{m}_g(h_3 - h_4 - T_o(s_3 - s_4)) \quad (10)$$

Where:

W_{rev} = the maximum reversible work net

ψ_1 = The irreversible work = cp_g ($T_3 - T_4$)

$\dot{m}_g$ = mass flow rate of gasses, kg/sec

ψ_2 = The exergy loss = $cp_g T_o \ln(\frac{p_3}{p_4})^{\left(\frac{K-1}{K}\right)}[(1-\eta_T) + \eta_T]$

k = process adiabatic exponent

η_T = Gas turbine isentropic efficiency

One commonly used measure of performance in the exergetic interpretation is the second law of thermodynamics efficiency, defined as:

$$\eta_{Ex} = \frac{W_T}{W_{rev}} = \frac{T_3 - T_4}{[T_3 - T_4] + T_o \ln(\frac{p_3}{p_4})^{\left(\frac{K-1}{K}\right)}[(1-\eta_T) + \eta_T]} \tag{11}$$

As for the isentropic efficiency η_T, the exergy efficiency is less than or equal to 1, and can be 1 at the best. In order to compare the two efficiencies, η_{Ex} and η_T we constitutive the temperature of exhaust gasses T_4 which expressed as

$$T_4 = T_3[1-\eta_T(\frac{p_3}{p_4})^{\frac{K-1}{K}}] \tag{12}$$

, then the exergy efficiency is defined as:

$$\eta_{Ex.T} = \eta_T(1-(\frac{P_2}{P_1})^{\frac{K-1}{K}})[\eta_T(1-(\frac{P_2}{P_1})^{\frac{K-1}{K}}) + \frac{T_0}{T_1}\ln[(\frac{P_2}{P_1})^{\frac{K-1}{K}})(1-\eta_T)+\eta_T]]^{-1} \tag{13}$$

3-3-2 Exergy of Compressor

It is known that the compressor is a machine consumed power from the main shaft of the gas turbine. This exergy power can be calculated according to the following equation:

$$\psi_{com} = m_a[cp_a(T_o - T_{2s}) + cp_a T_o[\frac{T_{2s}}{T_o} - 1 - \ln(\frac{T_{2s}}{T_o})] \tag{14}$$

3-3-3 Exergy of Combustion Chamber

As mentioned before the exergy of fuel is nearly equals to the net calorific value according to Eq. (9). The exergy of compressor and combustion chamber are equal to the exergy of the gas turbine and the exhaust gasses. The exergy balance chart of these components is shown in Fig. (3).

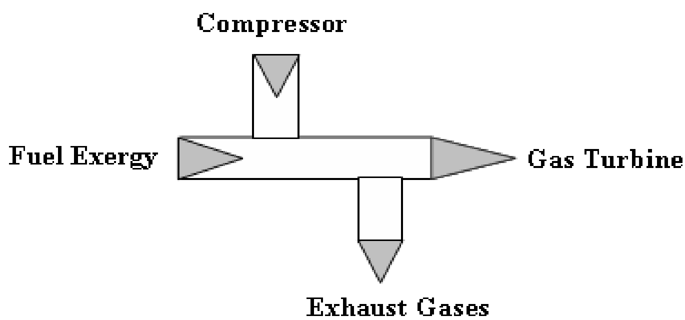

Fig. 3: Exergy Balance in Gas Power Plant

3-4 Exergy of Steam Power Plant

3-4-1 Exergy of Waste Heat Boiler

The exergy of the waste heat boiler can be calculated as a heat exchanger. It can express as:

$$\psi_{w.h.b} = \psi_s - \psi_{loss}$$

$$m_g.cp_g.\Delta T_g = m_s \Delta h_s - \psi_{loss} \quad (15)$$

Where:

$\psi_{w.h.b}$ is the exergy of the waste heat boiler

ΔT_g is calculated from Equation (6) (heat balance of waste heat boiler).

The subscript s means to the steam generated.

The increase of exergy of feed water $\psi_{s.w}$ to the live steam exergy ψ_s is calculated from equation (15) as exergy output.

$$\psi_s = h_1 - h_o - T_o(s_1 - s_o) \quad (16)$$

The exergy loss is due to irreversible heat in the heat added to the gas and can be expressed as:

$$\psi_{loss} = \psi_g - \psi_s - \psi_P \quad (17)$$

Where:

ψ_p = Exergy increase in the pump

The exergy efficiency of the waste heat boiler can be calculated as

$$\eta_{Ex\ w,h.b} = \frac{\psi_s - \psi_P}{\psi_g} \quad (18)$$

3-4-2 Exergy of Steam Turbine

The work of steam turbine Ws_T is less than the drop of exergy of steam ψ_s to $\psi_{loss,T}$ due to exergy loss as a result of irreversibility associated with the fluid flow through the turbine. The exergy loss in the turbine ($\psi_{loss.T}$) and exergy efficiency of the steam turbine ($\eta_{Ex\ S.T}$) are formulated as:

$$\psi_{loss,T} = \dot{m}_s(\psi_s - \psi_t) - W_T \quad (19)$$

$$\eta_{Ex.ST} = \frac{W_{T_g}}{\dot{m}_s(\psi_s - \psi_T)} \quad (20)$$

3-4-3 Exergy of Condenser

The exergy loss in condenser $\psi_{loss.cond}$ are mainly due the exergy dissipated in the heat reject to the surrounding as following

$$\psi_{loss.Cond} = \psi_{loss.T} - \psi_{loss.serr} \quad (21)$$

Where:

$\psi_{loss.serr}$ = the exergy loss in the surrounding

3-4-4 Exergy of Pump

The exergy loss in the pump $\psi_{loss.pump}$ is illustrated in Fig. 4 and the exergy efficiency of the pump can be calculated as :

$$\psi_{loss,pump} = W_P - \dot{m}_s(\psi_s - \psi_{loss.C}) \quad (22)$$

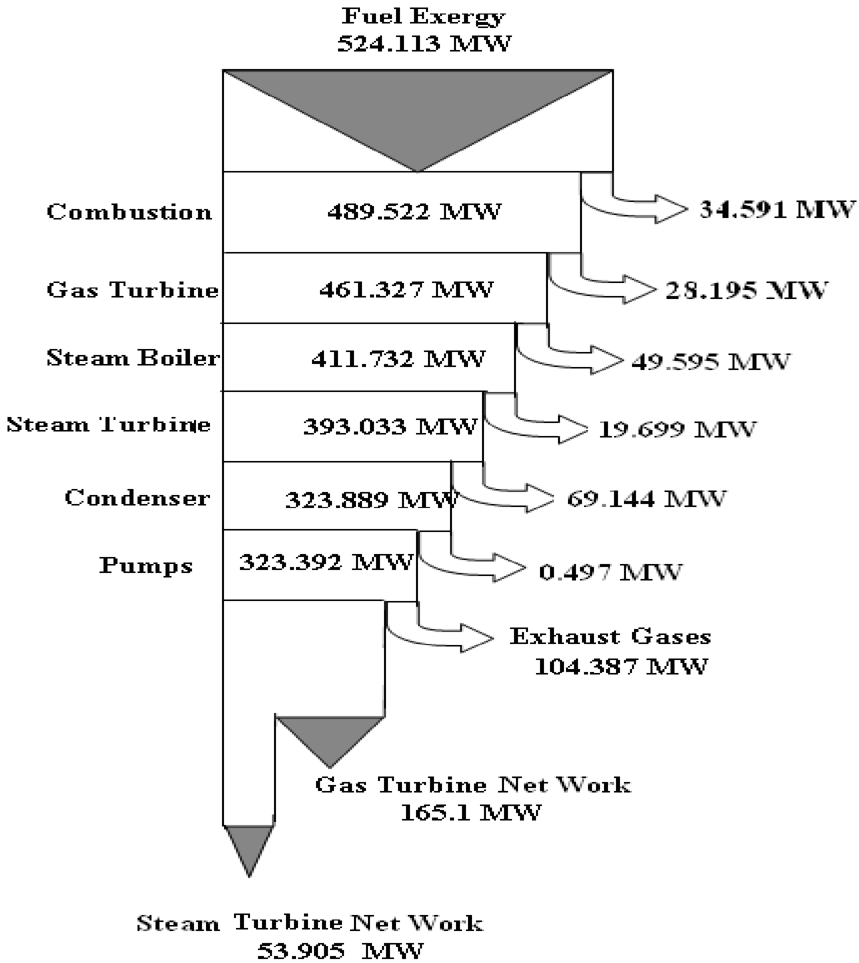

Fig. 4: Component Exergy Flow Chart of Combined Cycle

$$\eta_{Ex,ST} = \frac{m_s^{\bullet}(\psi_s - \psi_{loss.C})}{W_P} \tag{23}$$

Where the W_p is the work done input to the pump.

4- POWER PLANT EXERGY

For power plant as whole exergy input gain is the chemical exergy of fuel ψ_{Ch} whereas the exergy output is the net work produced.

The work net produced W_N is equal to the ($W_{G.T}$+ $W_{S.T}$ - W_{comp} - W_{pump}) .

The exergy losses are the sum of individual exergy losses of the plant component and the overall exergy efficiency of the plant is calculated as:

$$\eta_{Ex.} = \frac{W_N}{m_f^{\bullet}(\psi_{Ch})} \tag{24}$$

5- CASE STUDY

The design condition efficiencies and parameters for simple one shaft simple gas power station considered as a case study are having the following operation specifications which taken from the plant specification books explained in Table (2a). Table (2b) explains the specification of the operation conditions at part loads. Figure 5 shows the (T-S) diagram according to heat balance calculation of the first law of thermodynamics relations.

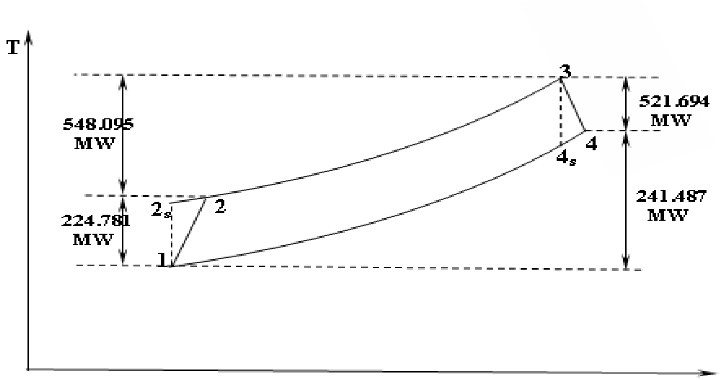

Fig. 5: First Law Heat Balance of Gas Power Plant

Table (1) the Gas Power Plant Specification Pressures and Temperatures data

Pressure (Bar)	Temperature (K)
$P_1= P_o=1.013$	$T_1= T_o=300.13$
$P_2=14.71$	$T_{2s}=644.13$
$P_3=14.71$	$T_2=693.13$
$P_4=1.041$	$T_3=1542.13$
	$T_{4s}=797.13$
	$T_4=729.13$

Table (2a) Design Specification of the Gas Power Plant at Full Load

Specification	Explanation
Fuel	Natural gas
Power output	165.1 MW
Overall efficiency	30.12%
Turbine shaft rotation	3000 r.p.m
Compression ratio	1:14.6
Exhaust gases flow rate	534.877 kg/s
Exhaust gases temperature	524 ^oC

Table (2b) the Natural Gas Chemical Composition

CH_6	CH_4	N_2	C.V(kJ/kg)
15.8	83.4	0.8	50600

Table (3) Design Specification of the Operation Conditions at Full and Part Loads

Loads %	Gas flow Rate (kg/s)	Combustion Temp (°C)	Exhaust Temp. (°C)	Power Output (MW)	pressure ratio	$\eta_{overall}$ %
100	534.877	1269.34	524	165.1	14.6	30.12
80	400.386	1269.34	547.91	132.08	13.89	27.13
60	398.774	1099.68	452.27	99.06	12.01	25.59
40	396.950	922031	380.54	66.04	11.303	22.41
20	395.540	760.36	307.87	33.02	10.731	14.78

5-1 Choice of the Steam Side Operation Condition

5-1-1 Reversible Steam Heat Generator (R.S. H.G)

The type of R.S.H.G used having three parts are economizer, Evaporator, and super heater to get the live steam pressure and superheating temperature of 30 bar and 450 °C the calculations and heat Balance was made according to the heat balance Equation (6). The mass flow Rate of live steam is changed according to the part load ratio which related to the heat added to the exhaust gasses mass flow rate.

5-1-2 Steam Turbine

Denotation the steam mass flow rate generated in the R.S.H.G at the same pressure and temperature the steam turbine will operate at the available mass flow rate whish will change according to the part load heat added. Table 4 explains the steam flow rate at part load, then the steam turbine from the type of changeable load.

Table (4) Superheated Steam Flow Rate of Part Loads at different Pressures and Temperatures

Load ratio	Steam flow rate
100%	60 kg/s
80%	48.7 kg/s
60%	33 kg/s
40%	21.35 kg/s
20%	9.6 kg/s

5-1-3 Condenser

Denoting the condenser pressure is taken to be lowest practical one Corresponding to considered ambient temperature. The live steam pressure is the maximum value which realize a practical safe exhaust Steam dryness fraction x = 0.88 and the steam process is assumed to leave the condenser at the same ambient condition.

6- DISCUSSION AND RESULTS

6-1 Discussion of the Steam Side Circuit

The system of steam circuit used was chosed according to the Maximum heat available in the exhaust gasses of the gas turbine. The Mass flow rate of steam was determined according to the temperature of the pitch point in the waste heat boiler, then the saturation temperature of the live steam circuit in the low pressure of steam (40 bar to 25 bar) and superheating temperature (300 °C to 500 °C) where the maximum dryness fraction of steam in the last stages in the steam turbine is not below 0.88. Then some trials were done to determine the suitable condition of steam as following

6-2 Determine the Live Steam Pressure

At constant condenser pressure equals 0.036 bar and superheating temperature equals 450 °C, some points of pressure of 25, 30, 35, 40, 45 and 50 bar were done. It is noticed that the dryness fraction decreased with increase the steam pressure and the enthalpy difference is increased with increase the steam pressure. Finally the total Thermal efficiency of the steam circuit is increases at 25bar until 30 bar and then decreases gradually until 50 bar. Figure 6 shows the effect of live steam pressure of the thermal efficiency of the cycle not only at Full load but at the part loads of 0.2, 0.4, 0.6, 0.8, and full load.

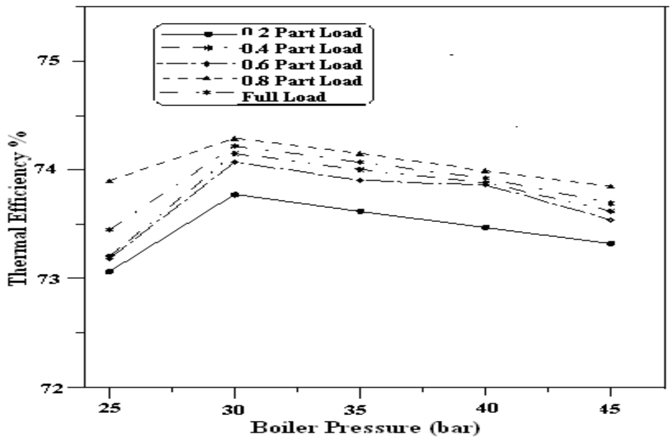

Fig 6: Effect of Live Steam Pressure on Thermal Efficiency

6-3 Determining the Live Steam Superheating Temperature

It is noticed that from Fig. 6, the best pressure of live steam is 30 bar, then at constant pressure of live steam in boiler of 30 bar and constant condenser pressure of 0.036 bar, some point of superheating Temperature of 300, 350, 400, 450 and 500 °C were done to determine the suitable of superheating temperature. The readings explain that the dryness fraction is increased with increase of superheating temperature until 0.88 at 450 °C and 0.912 at 500 °C. The enthalpy difference increases with increase of the superheating temperature. Figure 7 shows the effect of superheating steam temperature on the cycle thermal efficiency, it is noticed that the best temperature is 450 °C.

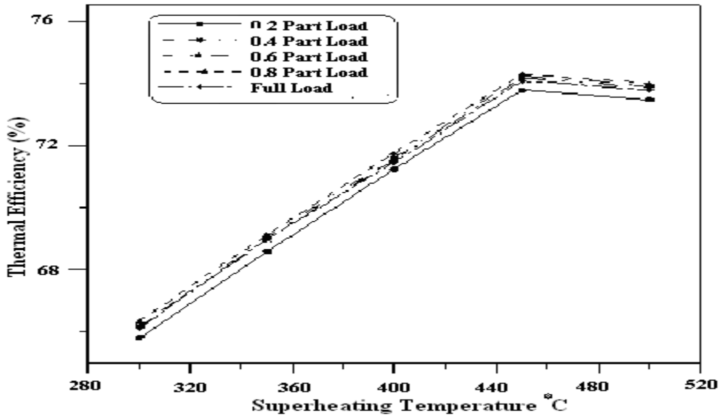

Fig. 7: Effect of Live Steam Superheating Temperature on Thermal Efficiency

6-4 Determining the Suitable Condenser Pressure

On the same way at constant live steam pressure of 30 bar and superheating temperature of 450 °C ,some points of condenser pressure 0.025 bar ,0.03 bar,0.036 bar, 0.04 bar 0.05 bar and 0.055 bar were done. The reading explains that the dryness fraction is increases rapidly with increase of condenser pressure but the enthalpy difference is decreased with increase of the condenser pressure, then the best condenser pressure is 0.036 bar. Figure 8 shows the effect of condenser pressure on the cycle thermal efficiency.

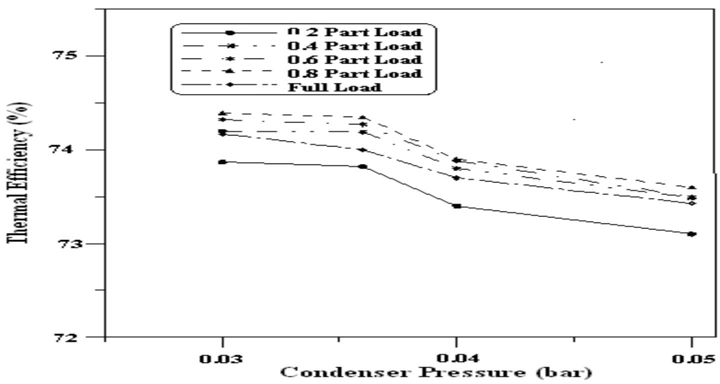

Fig. 8: Effect of Condenser Pressure on Thermal Efficiency

Choosing the steam condition including the pressure, superheating temperature and condenser pressure, some requirements are taken into consideration as follows:-

1. Effect of pitch point in the waste heat boiler to safe the stack from the chemical reaction between the stack metal and the exhaust gasses. This reaction will cause destructive corrosion with the stack walls. The final flow

gas temperature must be less than 170 °C [19] the pitch point is 20 °C difference between the saturated liquid and the exhaust gasses

2. Effect of steam superheating temperature, it is noticed from Fig. 7 that with increasing this temperature the thermal efficiency increased rapidly. The choice points are 300, 350 , 400 , 450 and 500 °C .the variation in the final thermal efficiency is varied with 10% from 300 °C until 450 °C and the increasing is slow until to 500 °C , then the choice temperature is 450 C.
3. Effect of generated steam pressure, Fig. 6 shows that, the suitable pressure of the steam cycle is chosen 30 bar.
4. Effect of condenser pressure, it can be seen that after determining the steam pressure, it is necessary to determine the condenser pressure. The choiced points were 0.025, 0.03, 0.036, 0.04, 0.045, 0.05 and 0.055 bar. It can be seen that from Fig. 8 the thermal efficiency is increased with decreasing the condenser pressure. The suitable condenser pressure is 0.036 bar.

6-5 Energy and Exergy

Considering the calorific value of the natural gas as a fuel used is 50600 kJ /kg, the fuel to air ratio is 0.021. This ratio is allowable ratio for simple gas power station having driven single shaft [23].

Table (3) shows the designed thermal efficiency of the gas power plant at different load conditions. The combustion chamber energy loss is about 6.6 %, whereas the rejected heat in the exhaust gasses is about 44.06 %. The remaining part of available fuel energy consumed in the running of the generating unit. Figure 9 shows the thermal energy and exergy efficiencies of the gas cycle. It is noticed from Fig. 9 that the increase in the electrical power generated using additional steam power plant id 53.905 MW. This means that, the final thermal efficiency increased from 30.12% to 39.597% .

On the other hand the exergy analysis shows that the exergy input to the plant is equal to 524.113 MW, whereas the exergy output as net work is 219.005 MW with 41.78% while, the final exergy of the gas power plant is 32.44 % .

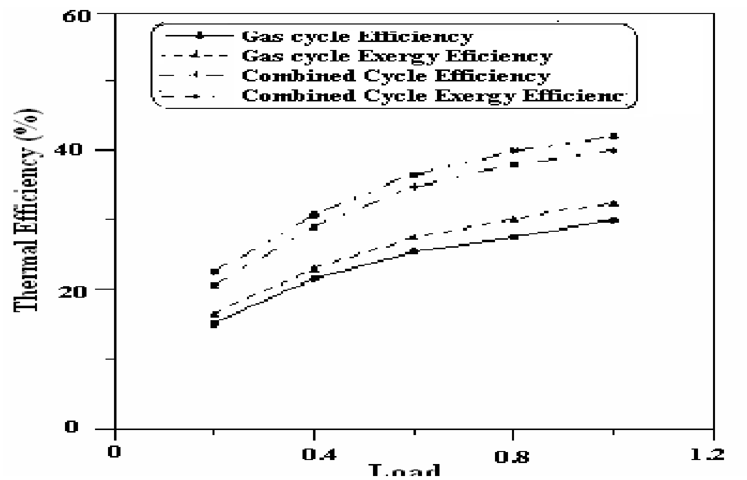

Fig. 9: Comparison between Different Cycles and Thermal Efficiency

7- CONCLUSIONS

1. The using combined plant improves the whole efficiency with percentage more than the gas power plant only.
2. The generated power remained merely constant with changing live steam pressure and superheating temperature and increases with reducing the condenser pressure.
3. The suitable operation conditions of steam plant are 30 bar live boiler pressure, 450 °C superheating temperature , 0.036 bar condenser pressure and dryness fraction is 0.88
4. The maximum power produced in the binary plant at 0.8 of part load.
5. The steam turbine can be operate at any part load of gas side of binary plant.
6. the overall thermal efficiency was improved by 9.477% and the exergy was enhanced by 9.34% at full load

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التحليل الثرموديناميكى لمحطة قوى مركبة برايتون / رانكين

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تم تطبيق طريقة التحليل الثرموديناميكى بطريقة الحودة الحرارية و الإتاحية طبقا للقانون الثانى للديناميكا على محطة توليد قوى غازية تم دراسة تجويلها الى محطة قوى مركبة. تم التحليل الثرموديناميكى للنظام ككل وكذلك للمكونات الأساسية عند أحمال تشغيل جزئية و أيضا عند الحمل الكلى , تم تحديد ظروف التشغيل المثلى و التي تشمل ضغط الغلاية و درجة حرارة البخار المحمص وضغط المكثف وظهر أن هناك تحسن كبير فى كفاءة التشغيل طبقا للقانون الأول و الثانى للديناميكا الحرارية بنسبة تصل إلى 9.835% و أن الإتاحية لتوليد القدرة زادت بنسبة تصل إلى 9.34% .