Exergy-energy analysis for combined brayton / rankine power plant

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This paper deals with parametric energy and exergy analysis of a combined gas steam cycle plant with 165.1 MW capacity. The exergy analysis indicates how much power is rejected in exhaust gases in order to study the possibility of using it in a suggested steam unit (combined power plant) and to enhance the thermal efficiency of the plant. Analysis of exergy enhancement at different part loads of 0.2, 0.4, 0.6, 0.8, and at full load has been carried out. The exergy efficiencies were calculated according to the first and the second laws of thermodynamics. It has also been concluded that the overall thermal efficiency was improved by 9.835% and the exergy was enhanced by 9.34% at full load.

تم تطبيق طريقة التحليل الترموديناميكى بطريقة الإتاحية طبقا للقانون الثانى للديناميكا الحرارية على محطة قوى مركبة .تم التحليل للنظام ككل وكذلك للمكونات الأساسية عند أحمال تشغيل جزئية و أيضا عند الحمل الكلى. تم تحديد ظروف التشغيل المثلى والتي تشمل ضغط الغلاية ودرجة حرارة البخار المحمص وضغط المكثف وظهر أن هناك تحسن كبير فى كفاءة التشغيل طبقا للقانونين الأول والثانى للديناميكا الحرارية بنسبة تصل إلى ٩,٨٣٥% وأن الإتاحية لتوليد القدرة زادت بنسبة تصل إلى ٩,٣٤%.

Keywords: Energy, Exergy, Second law analysis, Power plant, Combined cycle

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 considered the power plant of 21^{st} century. Energy economics is a broad field. Extensive work has been done that combines energy and economics. The exergy study gives us an indication to exergy cost saving by using different delivers for the power plant wastes.

The following points are considered when the present works were done:
1. Thermodynamics efficiencies and losses.
2. Measures used for the determining the efficiencies and losses.
3. Utilization of efficiency and loss 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, considering effects, of different available parameters which affect 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. They also, reported that the maximum combined cycle efficiency was reached when the gas turbine exhaust temperature was 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] described 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. Mark A. 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. A.F. 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. Y. Sanjay et al. [33] have studied parametric energy and exergy analysis of reheat gas–steam combined cycle using closed-loop-steam-cooling. In the light of the above works the present work is aimed at: To identify and quantify the sources of losses in the selected configuration combined cycle with different means of loading rates and hence to try for minimizing these losses to achieve maximum efficiency in 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 to be used to analysis power of the choice plant (combined power plant) as a system with component by component as well as the plant as a whole 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 ($\Delta S_{HT1,2}$) and entropy production ($\Delta S_{irr1,2}$) [12, 17]:

$$\Delta S_{12} = \Delta S_{HT1,2} + \Delta S_{irr1,2} . \tag{1}$$

$$\Delta S_{HT,12} = \int_1^2 \sum_i \frac{\delta Q_i}{T_{H,i}} . \tag{2}$$

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

Where

Q Heat exchange,

T_H Temperature of heat reservoir exchanging heat with the system,

T Temperature of the system,

W_F Work to overcome friction, and

i 1 , 2 , 3 , ….

Eq. (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]

$$I_{12} = T_o \Delta S_{irr1,2} . \tag{4}$$

Where T_o be sink or surrounding temperature.

As the main object of the present work is to enhancement of the thermal efficiency of the choiced gas power plant with using an auxiliary steam power unit consumed the heat reject in the exhaust gases of the gas turbine. The following is a schematic diagram of the binary power planed can be used. Fig. 1 shows that the schematic diagram of a compound – cycle system as a binary cycle.

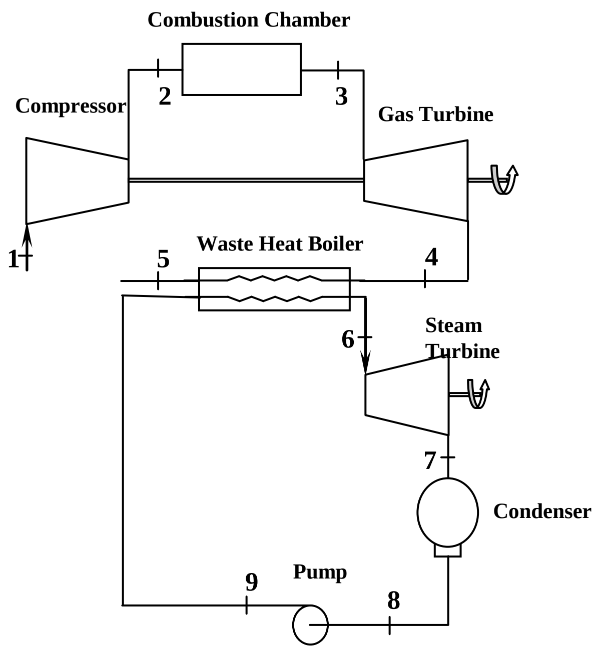

Fig. 1. Combined cycle used.

Then the thermal efficiency of the over binary cycle system is:

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

The heat balance in the economizer, evaporator and super heater to determine the ratio of the steam flow rate to the gas flow rate is explained in the following equations

$$\frac{m_s}{m_g} = \frac{c_{pg}(T_{hi} - T_{ex})}{h_{ce} - h_{iecc}} = \frac{c_{pg}(T_{hi} - T_x)}{h_{ce} - h_{w,sat}} \ . \tag{6}$$

Where

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

Cp gas specific heat, kJ/kg.K,
T_{hi} exhaust gas temperature, K,
T_x Pitch point temperature, K,
h_{ce} Super heated steam enthalpy , kJ/kg,
$h_{w.sat}$ Saturated water enthalpy, kJ/kg, and
$h_{.iecc}$ Saturated water enthalpy 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 eq. (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:

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

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

$$\varphi_1 = h_o + T_o\left(s_1\text{-}s_o\right). \tag{8}$$

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 :

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

Where:

ΔG_o Gibbs function of complete chemical reaction of fuels refer to surrounding state equal to H_o - T_o S_o, and
R gas constant = 0.287 kJ/kg.K

and subscript K refers to the component of product of combustion.

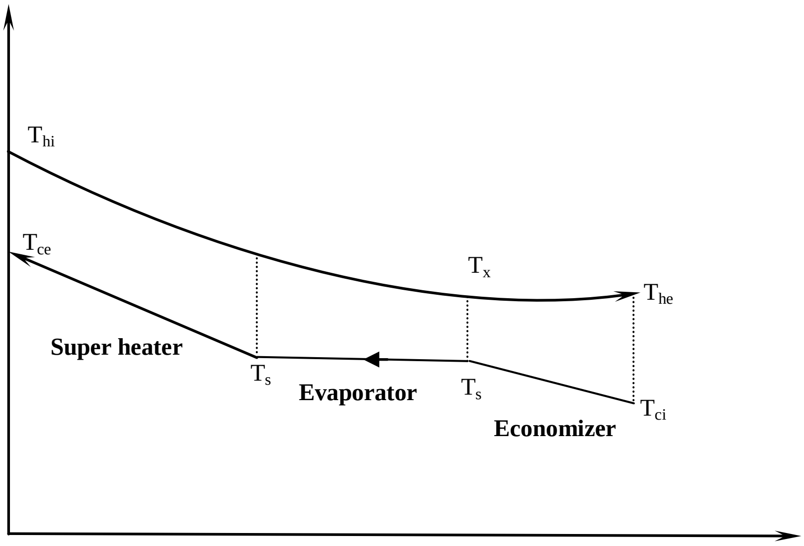

Fig. 2. Temperature variation along the waste heat boiler paths.

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* at obtains pure chemical substance having the same ratios of constituent chemicals This ratio denoted by ϵ $= \frac{\psi_{ch}}{NCV}$ estimated by [20] 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 eq. (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)

$$W_{rev} = \psi_1 - \psi_2 = h_3 - h_4 - T_o(s_3 - s_4) . \qquad (10)$$

Where:

W_{rev} The maximum reversible work net,
ψ_1 The irreversible work = c_{pg} $(T_3 - T_4)$,
ψ_2 The exergy loss =

$$c_{pg} T_o \ln\left(\frac{p_3}{p_4}\right)^{\left(\frac{K-1}{K}\right)} [(1-\eta_t)+\eta_t] ,$$

k Adiabatic process exponent, and
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:

$$\eta_{ex} = \frac{Exergygained}{Exergypaid}$$

$$= \frac{W_t}{W_{rev}} = \frac{T_3 - T_4}{[T_3 - T_4] + T_o \ln\left(\frac{p_3}{p_4}\right)^{\left(\frac{K-1}{K}\right)} [(1-\eta_t)+\eta_t]} . \qquad (11)$$

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

$$T_4 = T_3[1 - \eta_t(\frac{p_3}{p_4})^{\frac{K-1}{K}}], \quad (12)$$

then the exergy efficiency is defined as:

$$\eta_{ex,T} = \eta_t(1 - (\frac{p_3}{p_4})^{\frac{K-1}{K}})[\eta_t(1 - (\frac{p_3}{p_4})^{\frac{K-1}{K}}) + \frac{T_o}{T_3}\ln[(\frac{p_3}{p_4})^{\left(\frac{K-1}{K}\right)}(1-\eta_t)+\eta_t]]^{-1}. \quad (13)$$

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:

$$\psi_{com} = m_a[c_{pa}(T_o - T_{2s}) + c_{pa}T_o[\frac{T_{2s}}{T_o} - 1 - \ln(\frac{T_{2s}}{T_o})]]. \quad (14)$$

3.3.3. Exergy of combustion chamber

The exergy of fuel or the net calorific value according as mentioned in eq. (9). From here in before the exergy of compressor and combustion chamber is equal to the exergy of the gas turbine and the exhaust gasses considering the exergy balance chart of these components as shown in fig. 3.

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:

$$\psi_{WHB} = \psi_s - \psi_{loss}$$
$$m_g.c_{pg}.\Delta T_g = m_s \Delta h_s - \psi_{loss}. \quad (15)$$

Where:

ψ_{WHB} is the exergy of the waste heat boiler, and

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

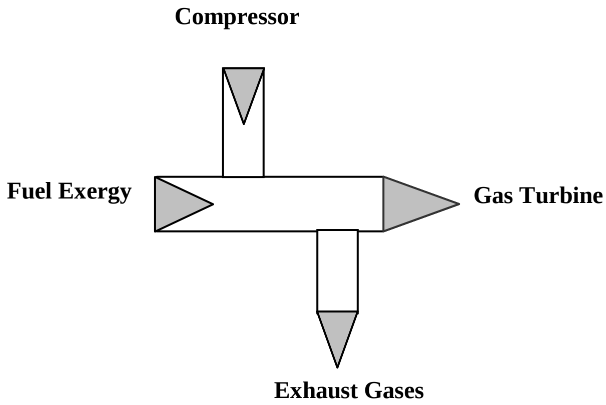

Fig. 3. Exergy balance in gas power plant.

The subscript S means to the steam generated.

The increase of exergy of feed water $\psi_{feed.w}$ to the live steam exergy ψ_s is calculated from eq. (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

$$\eta_{ex\,WHB} = \frac{\psi_s - \psi_{P_g}}{\psi_g}. \quad (18)$$

3.4.2. Exergy of steam turbine

The work of steam turbine W_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:

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

$$\eta_{ex,ST} = \frac{W_{T_g}}{\psi_S - \psi_T} \ . \qquad (20)$$

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

$$\psi_{lossCond} = \psi_{lossT} - \psi_{losssurr} \ . \qquad (21)$$

Where:

$\psi_{losssurr}$ The exergy loss in the surrounding

3.4.4. Exergy of pump

The exergy loss in the pump $\psi_{losspump}$ is illustrated in fig. 9 and the exergy efficiency of the pump can be calculated as:

$$\psi_{losspump} = W_P - (\psi_S - \psi_{lossC}) \ . \qquad (22)$$

$$\eta_{ex,ST} = \frac{{}_g\psi_S - \psi_{lossC}}{W_P} \ . \qquad (23)$$

Where the *Wp* 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 sun of individual exergy losses of the plant component and the overall exergy efficiency of the plant is calculated as:

$$\eta_{ExP} = \frac{W_N}{\psi_{Ch}} \ . \qquad (24)$$

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 name plant. Table 2-a explains the design specification of gas power station and table 2-b explains the specification of the operation conditions at part loads. Fig. 4 shows the T-S diagram according to heat balance calculation of the first law of thermodynamics relations.

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 eq. (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.

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 2-a
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 ° C

Table 2-b
The natural gas chemical composition

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

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 explain the steam flow rate at part load, then the steam turbine from the type of changeable load.

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 whish 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.

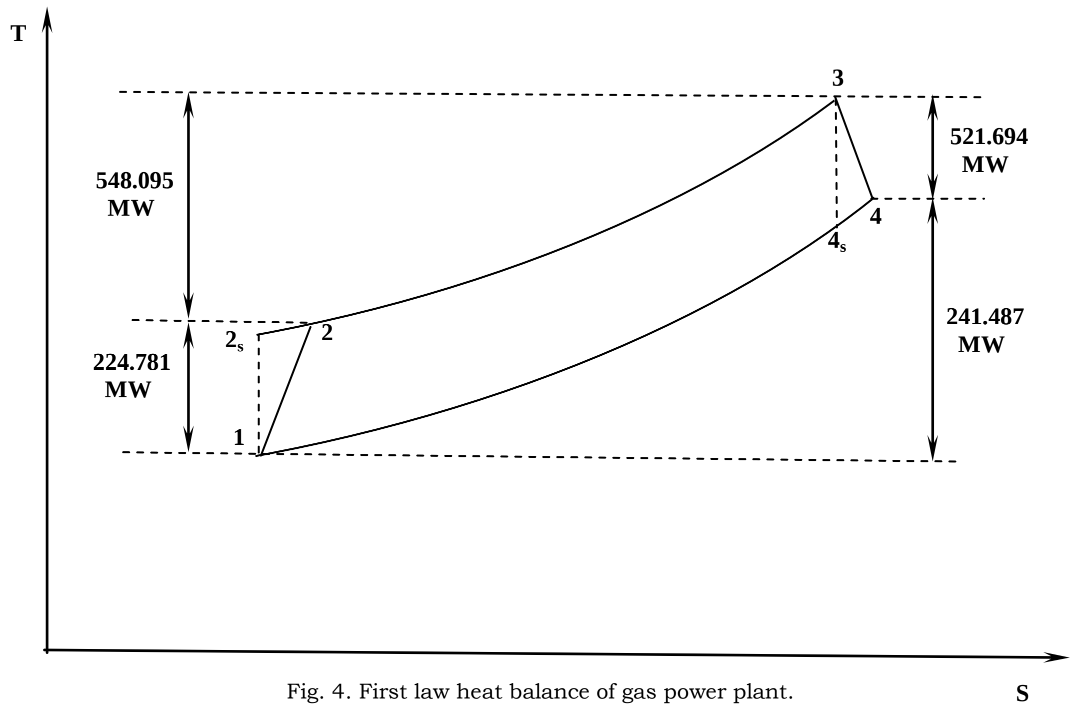

Fig. 4. First law heat balance of gas power plant.

Table 3
Design specification of the operation conditions at 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	922.31	380.54	66.04	11.303	22.41
20	395.540	760.36	307.87	33.02	10.731	14.78

Table 4
The steam flow rate at part loads

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

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 (500 C to 300 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 bar, 30 bar, 35 bar, 40 bar, 45 bar 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 nearly constant between 25 bar and 30 bar and decrease gradually until 50 bar. Fig. 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.

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 C, 350 C, 400 C, 450 C and 500 C were done to determine the suitable 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. Fig. 5 shows the effect of superheating steam temperature on the cycle thermal efficiency, it is noticed that the best temperature is 450 C.

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. Fig. 7 shows the effect of condenser pressure of the cycle thermal efficiency.

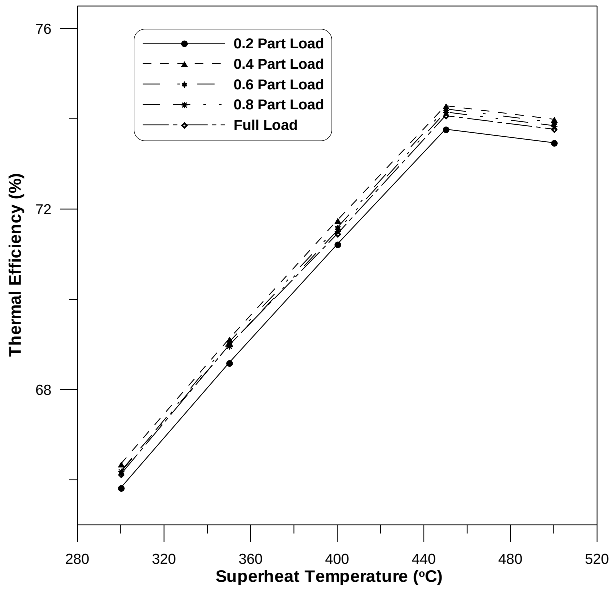

Fig. 5. Effect of live steam superheating temperature on thermal efficiency.

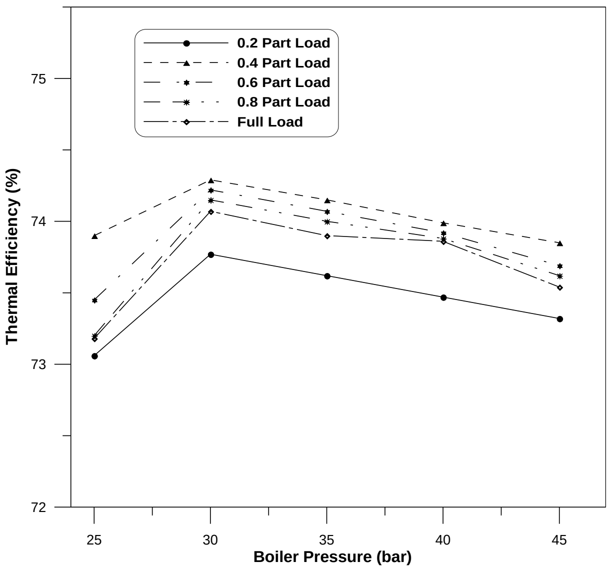

Fig. 6. Effect of live steam pressure on thermal efficiency.

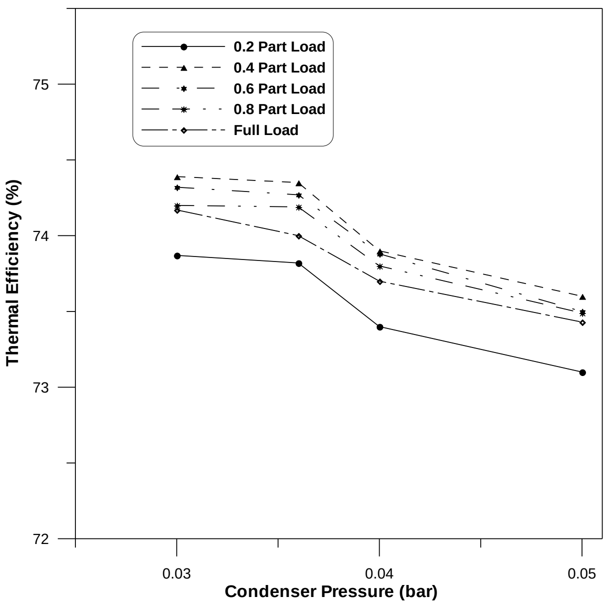

Fig. 7. Effect of condenser pressure on thermal efficiency.

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].

The calculations according to the first law of thermodynamics analysis show that the thermal efficiency of the gas power plant at different loads as mentioned in table 3 whereas the combustion chamber energy loss about 6.6 %, the heat reject in the exhaust gasses is 44.06 % from the available heat power in combustion chamber. This power is divided as mentioned in fig. 8. 17.6 % from the heat added to the cycle is converted to electrical power generation. The pumping power was about 0.12 % and the remain power was going to the stack out to the ambient.

On the other hand the exergy analysis according to the second law of thermodynamics shows the exergy input to the plant is equal to 524.113 MW as a heat added from fuel to the combustion chamber, the exergy output as net work is 219.005 MW with 41.78% moreover the exergy loss in the gas turbine equal 18.8% as shown in fig. 9. It is noticed that from fig. 9 the increase in the electrical power generated with using addition steam power unit by using the waste heat gasses from the gas power unit is 53.905 MW , also the final thermal efficiency of the binary power station is increased from 30.122% to 39.597% as can be seen from fig. 8. To choose the steam condition from the pressure, superheating temperature and condenser pressure, some points are taken into consideration to study the suitable conditions of working of steam conditions, these points are:

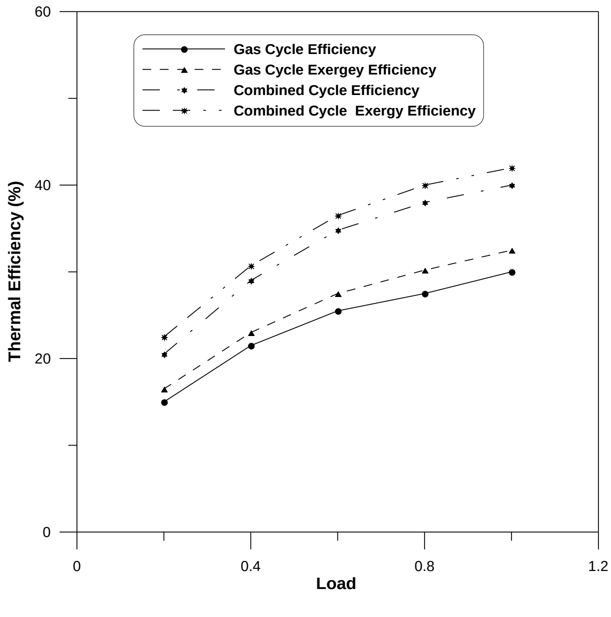

Fig. 8. Comparison between different cycles and thermal efficiency.

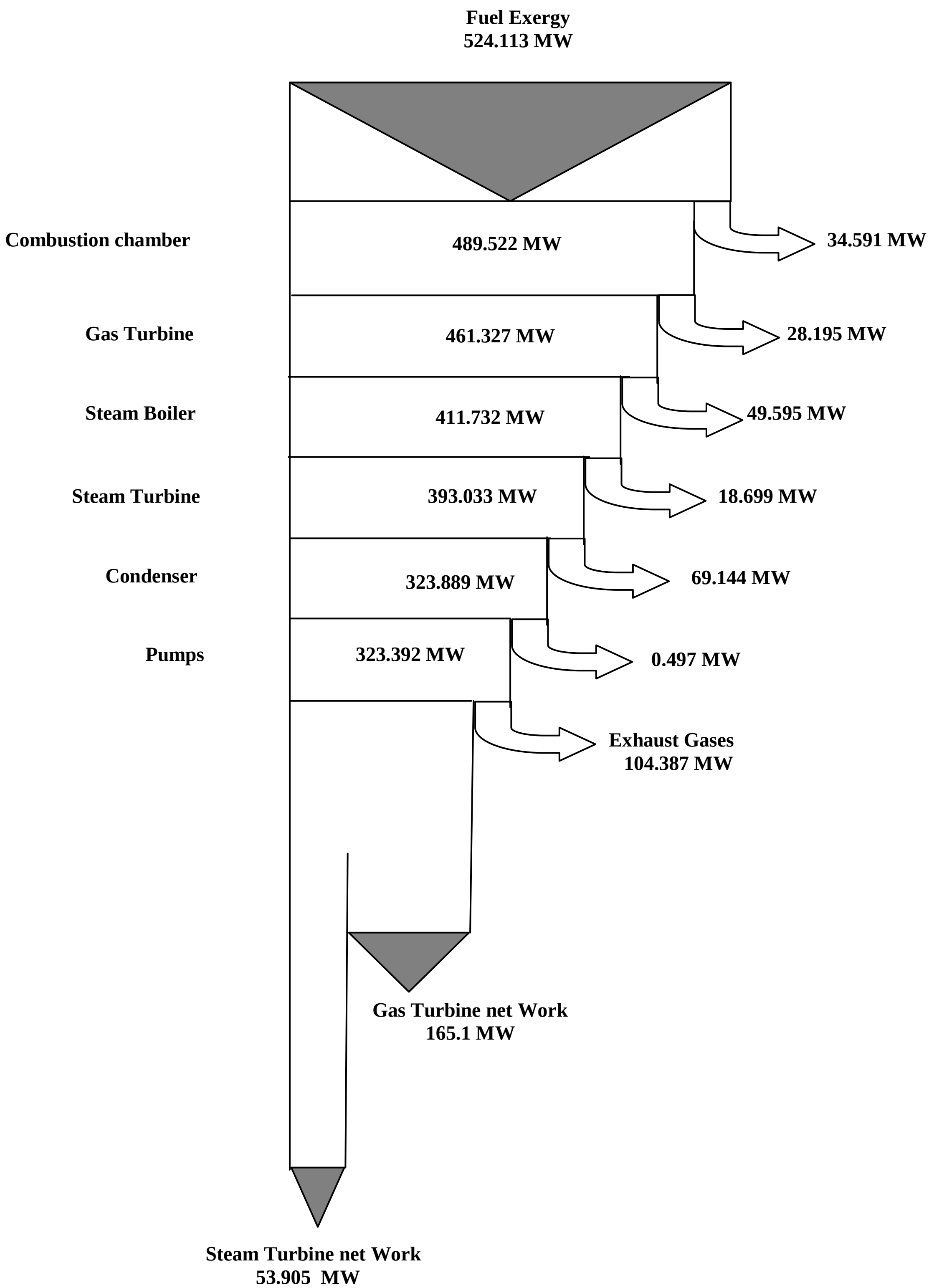

Fig. 9. Component exergy flow chart of combined cycle.

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 because this reaction will causing destructive corrosion with the stack walls. The final flow gas must be more than 170 C [19] the pitch point is 20 C difference between the saturated liquid and the exhaust gasses.
2. Effect of generated steam pressure, fig. 5 shows that, the effect of the changing of the live pressure on the thermal efficiency of the steam side. It can be seen that the variation in the thermal efficiency is +0.01% this influence is not important to determine the best pressure, the suitable pressure of the steam cycle is chosen 30 bar.
3. Effect of steam superheating temperature, it is noticed 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.
4. Effect of condenser pressure, it can be seen that after determining the steam pressure, it is necessary to determine the condenser pressure. The choice points are 0.025, 0.03, 0.036, 0.04, 0.045, 0.05 and 0.055 bar. It can be seen that from fig. 7 the thermal efficiency is increased with decreasing with the condenser pressure. The suitable condenser pressure is 0.036 bar.

7. Results

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 maximum power produced in the binary plant at 0.8 of part load.
4. The steam turbine can be operate at any part load of gas side of binary plant.
5. the overall thermal efficiency was improved by 9.835% and the exergy was enhanced by 9.34% at full load

Nomenclature

C	Velocity, m/s,
c_p	Specific heat, kJ / kg.K,
g	Gravity acceleration , m/s^2,
h	Enthalpy , kJ / kg,
I	Irreversibility, kJ / kg,
k	Adiabatic exponent process,
m	Mass , kg,
Ncv	Natural gas calorific value, kJ/kg,
P	Pressure, bar,
Q	Heat, kJ,
R	Gas constant, kJ/kg.K,
s	Entropy, kJ/kg.K,
T	Temperature, K,
W	Work, kW,
ψ	Exergy, kJ/kg, and
Φ	Anergy, KJ/kg.

Subscripts

Ce	Superheating temperature,
Ci	for water temperature entering the economizer,
f	for friction,
g	for gas,
hi	for high temperature of gas,
ht	for high temperature,
i	for points 1, 2, 3,
irr	for irreversibility,
iecc	for steam temperature entering the economizer,
s	for saturation steam temperature,
w.s.t	for saturated water temperature,
x	for pitch point temperature,
1	for high temperature,
0	for ambient temperature,
GT	for gas turbine,
ST	for steam turbine,
add	for add heat,
O	for oxygen,
oo	Partial,
Ch	for chemical,
Rev	for reversibility,
T	for thermal,
S	for isentropic,
Loss	for losses,
w.h.b	for waste heat boiler,
s	for steam,
p	for pump,
loss T	for losses of turbine,
serr	for surrounding,

C for condenser,
Ex for exergy,
N for net work, and
Ng natural gas.

References

[1] C. Sieppel and R. Bereuter "The Theory of Combined Steam and Gas-Turbine Installation", Brown Boveri., Review, 47:83–799 (1960).
[2] H. Czermak, A. Wunsch "The 125 MW Combined Cycle Plant Design Features, Plan Performance and Operating Experience", Paper No. 82 GT-323. ASME (1982).
[3] Wunsch A. "Highest Efficiencies Possible by Converting Gas-Turbine Plants Into Combined Cycle Plants", Brown Boveri Review, 10:455–63 (1985).
[4] J.H. Horlock "Combined Power Plants- Past, Present, and Future", Trans ASME, Journal of Engg. for Gas Turbine and Power, 117:608–16 (1995).
[5] WuC., "Intelligent Computer-aided Sensitivity Analysis of Multi-stage Brayton/Rankine Combined Cycle", Energy Conv Mgmt, 40:215–32 (1999).
[6] G. Cerri "Parametric Analysis of Combined Gas–Steam Cycles", Trans, ASME J Engng Gas Turbine Power, 109:46–54 (1987).
[7] R. Andriani, U. Ghezzi, Anntoni LFG. Jet Engines with Heat Addition During Expansion: A Performance Analysis pp., 99-0744. AIAA (1999).
[8] A. Polyzakis "Industrial Gas-Turbine for Combined Cycle Plant", MSc thesis, Cranfield University (1995).
[9] A. Bejan "Fundamentals of Exergy Analysis, Entropy Generation Minimization, and the Generation' of flow architecture", Int. J. Energy Res, 26:545–65 (2002).
[10] M.A. Rosen, "Clarfying Thermodynamic Efficiencies and Losses Via Exergy", Exergy an International journal 2, pp. 3-5 (2002).
[11] A.F. El-Dib "An Exergy Analysis of Steam power and Cogeneration Plants" Journal of Engineering and Applied Science, Faculty of Engineering, Cairo University, Vol. 45, pp. 791-809 (1998).
[12] D.k. Anand, K.W. Linder, S. Schweitzer and W.J. Knish, "Second Law Analysis of Solar Powered Absorption Cooling Cycles and Systems Transactions of the ASME", Journal of solar Energy Engineering, Vol. 106, pp. 291-298 (1984).
[13] K.F. Knoche, "Technishe Thermodynamic Vieweg ", Brauschweig (Germany) (1978).
[14] R. Natarajan, "Thermodynamic Analysis of Energy Systems and Processes", Indian Institute of Technology, Madras, (1986).
[15] D.E. Winterbone, "Advanced Thermodynamics for Engineers", Jone Wily and Sons. Inc., New York (1997).
[16] K. Krith and J.F. Kreider, Principles of Solar Engineering. McGraw-Hill Book Company, New York (1998).
[17] V.K. Kirillin, V.V. Sychev and A.E. Sheindlin, "Engineering Thermodynamics, Mir Publishers Moscow (1976).
[18] R.E. Sonntag, G.J. Van Wylen, Introduction to Thermodynamics : Classical and Statistical. Jone Wily and Sons. Inc., New York (1991).
[19] W. Li Kam and A.P. priddy, Power plant System Design, Jone Wily and Sons. Inc., New York (1985).
[20] T.J. Kotas, "The Exergy Method of Thermal Plant Analysis", Butterworths, London (1985).
[21] H. Struchtrup and M.A. Rosen. How Mush Work is Lost in an Irreversible Turbine. Exergy an International Journal 2, pp. 152- 158 (2002).
[22] G. Rogers and Y. Mayhew .Engineering thermodynamics Work and Heat Transfer. Longman England (2000).
[23] H. Cohan, G.F.C. Rogers and H.I. Saravanamuttoo, "Gas Turbine Theory Longman House", England (1972).
[24] Chiesa, Macchi, "A Thermodynamic Analysis of Different Options to Break 60% Electrical Efficiency in Combined Cycle Power Plants, ASME J. Eng. Gas Turbine Power 126, pp. 770–785 (2004).
[25] M.A. El Masri, "On Thermodynamics of Gas Turbine Cycles: Part 3- Thermodynamic Potential and

Limitations of Cooled Reheated-Gas Turbine Combined Cycles, ASME J. Eng. Gas Turbine Power, Vol. 108, pp. 160–169 (1986).

[26] R.L. Bannister, R.A. Newby and W.C. Yang, "Final Report on the Development of A Hydrogen-Fueled Combustion Turbine Cycle for Power Generation, ASME J. Eng. Gas Turbine Power, Vol. 121, p. 38 (1999).

[27] I.G. Rice, "The Combined Reheat Gas Turbine/Steam Turbine Cycle: Part I – A Critical Analysis of the Combined Reheat Gas Turbine/Steam Turbine Cycle", ASME J. Eng. Gas Turbine Power, Vol. 102, pp. 35–41(1980).

[28] I.G. Rice, "The Combined Reheat Gas Turbine/Steam Turbine Cycle: Part II – The LM5000 Gas Generator Applied to Combined Reheat Gas Turbine/Steam Turbine Cycle", ASME J. Eng. Gas Turbine Power 102, pp. 42–49 (1980).

[29] I.G. Rice, "The Reheat Gas Turbine with Steam Blade Cooling – A Means of Increasing Reheat Pressure, Output and Combined Cycle Efficiency, ASME J. Eng. Gas Turbine Power 104, pp. 9–22 (1982).

[30] M. Gambini, G.L. Guizzi and M. Vellini, "H2/O2 Cycles: Thermodynamic Potentialities and Limits", ASME Gas Turbine Power, Vol. 127, pp. 553– 563 (2005).

[31] R. Bhargava and A. Perotto, A Unique Approach for Thermo Economic Optimization of An Intercooled, Reheat, and Recuperated Gas Turbine for Cogeneration Applications, ASME J. Eng. Gas Turbine Power, Vol. 124, pp. 881–891 (2002).

[32] Andreas Poullikkas, "An Overview of Current and Future Sustainable Gas Turbine Technologies, Renew. Sustain. Energy Rev., Vol. 9, pp. 409– 443 (2005).

[33] Y. Sanjay et al., "Energy and Exergy Analysis of Steam Cooled Reheat Gas–Steam Combined Cycle", Applied Thermal Engineering, Vol. 27, pp. 2779–2790 (2007).

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