Development of Shatt Al-Basra Gas Turbine: Combined or Cogeneration Cycles: Energy and Exergy Approach

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Abstract-. Most gases that burning in Basra city, one of the largest industrial cities in the south part of Iraq, have a special properties such as availability, cheap prices, high heating value, lower sulfur content and there is no ash associated with it. In Iraq and especially in Basra city, there is a significant increase in power and pure water demand. Gas turbine power plant, dual purposes systems have submitted good acceptance for power generation especially in recent years. This heat engine discards huge amount of heat energy to the atmosphere as exhaust gases. The most wide use improvement is adopting a combined cycle arrangement. However, utilization the discarded thermal energy to produce pure water seems to be good option especially in Basra city. Producing the pure water could be done using multi stage flash distillation system (MSF) technique that driven by steam. The heat recovery boiler is used to produce the steam from the exhaust gases. The availability of steam offers a fine way to improve the duty of the gas turbine using steam injection method. This papers investigate various configurations of gas turbine combined cycle power plants GTCC and evaluate thermally both the feasibility of coupling with multi stage flash (MSF) and using steam injection method to enhance the performance of gas turbine engine. Energy and exergy principles will be used to describe the theoretical models for the configuration. The analysis developed in the study will be applied to a specified gas turbine power plant that operates in Basra city as a case study.

Keywords: Power plant; Combined cycle; Multi stage flash distillation; Energy and exergy analyses; Engineering equation solver.

1. Introduction

The gas turbine engine is one of the most important technologies in the world. This engine consists of three main parts: air compressor, combustion chamber, and turbine. The operation of the gas turbine engine is based on the open thermodynamic cycle which is also called Joule-Brayton cycle [1]. Atmospheric air compressed through the air compressor to combustion chamber. The hot combustion gases were generating in the combustion chamber via burning the fuel with air. As a result, part of the chemical energy stowed in the fuel was translating into mechanical work in the turbine and the rest was discharging to the atmosphere. It is important to study the gas turbine technology and know the possibility of developing this technique to take advantage of it as much as possible to produce and develop power.

The analysis of this situation depends on the analysis of energy largely, but the development of technology and the need to take advantage of wasted energies as much as possible revealed a significant technology developed in recent years known exergy[2].

In the research works, some attention has been paid to the problem of modeling of gas turbines operated at part load and to the analysis of their performance based on the first and second laws of thermodynamics. Moreover, others researchers attention to enhancement and development this technologies to increase the power to generated electricity. **Ebadi** et al [3], performed exergy analysis for gas turbine equipped with heat exchanger between the compressor and the combustion chamber. Quantitative exergy balance for each component and for the whole system was considered. The results showed that the firing temperature represents the crucial parameters that affect the exergetic efficiency and exergy destruction of the plant. **Altayib** [4]**,** performed energy, exergy, and exergy economic for gas turbine power plant in Macca city in Saudi Arabia. the results show that the overall plant energetic and exergetic efficiencies of the plant increase by 20% and 12% respectively when the inlet air temperature is cooled down to 10℃. Besides, the exergoeconomic optimization results demonstrated that CO_2 emissions can be reduced by increasing the exergetic efficiency and using a low fuel injection rate into the combustion chamber. **Al-Gburi** [5]**,** presented exergy analysis for AL Najaf gas turbine power plant located in Iraq. The results show that, the combustion chamber is found to be the chief means of irreversibility in the plant. When 8℃ rises was done in the temperature, the exergy efficiency for combustion chamber was calculated to be 49.83%. Also, it was identified that the exergetic efficiency and the exergy destruction are considerably dependent on the alterations in the turbine inlet temperature.

2. Enhancing the Performance of Shatt AL Basra GT Power Plant with using Combined Cycle and Cogeneration (Multi Stage Flash unit) Plants.

Shatt Al-Basra gas turbine power plant is located at Shatt Al Basra at the south of Basra, Iraq, About 40 km south of the city center. The rate of power is 126.1*10 MW. Originally, the plant established in 2012 by METKA S.A. with one-generation unit rating 126.1 MW under ISO condition. The other units were commissioned in 2016. The plant may use natural gas, heavy fuel oil (HFO), or light fuel oil (LFO) in the combustion process [6].The plant currently almost uses natural gas for which the properties have given in Table (1). In the present study, unit one in Shatt Al-Basra gas turbine power plant is taken as a case study. This gas turbine unit is MS9001E (Frame 9) of single shaft arrangement. The 17-stages, axial compressor is supplied with IGV row that controls the air mass flow rate drawn by the unit at part-load operation. The combustion system is made of 14- separate combustion chamber, which are symmetrically distributed on the circumference of the gas turbine. The turbine is of impulse-reaction type. It consists of three stages and the unit operates at 3000 r.p.m[7].

For gas turbine cycle as core engine, the thermal efficiency of power plant can be increased remarkably by what is called combined cycle (CC). The overall efficiency can be in range (40%-60%). Increasing efficiency along with reducing the emissions of exhaust gases make CC is the favorite power plant worldwide. The clean and cheap natural gas as fuel consume in the CC power plant. as shown in Figure (1). The CC power plant includes in the GT cycle i.e. compressor, combustion chamber, gas turbine and the steam cycle i.e. heat recovery steam generator (HRSG), steam turbine, and condenser. Almost of the CC electric power output produces by GT cycle is about 2/3 of plant. The steam cycle produces about 1/3 of the CC power output. Hot gases exhausted from the GT (typically at 475–600 C°) are directed to HRSG to produce steam. This steam, when at high-enough temperature and pressure, was directing to steam turbine to produce more work without adding more fuel [8]. While heat in the form of steam produced from the HRSG or extracted (or discharged) from steam turbine was using for any useful process (e.g running water desalination plant) the system is called "Cogeneration". Figure (2), shows the Cogeneration Power Desalting Plant [9]. First, the gas turbine cycle is analyzed alone without combined cycle and MSF system. As can be seen in the control volume CV.1 of figure (1), the ambient air is firstly passes through the air filter from point 1 to point 2. Point 1 refers to the ambient conditionsT_o, P_o. No thermodynamic process is occurred here except pressure drop. The compressor discharges the air at point 3 at higher pressure and temperature. Usually, a percentage portion of the compressed air is extracted at different stages of the compressor. This air extraction is for cooling the turbine blades and any other application needed in the plant. Besides, these extracted air portions plays important role in the control of the plant to ensure safe operation for the compressor. The great portion of compressed air at point 3 enters to the combustion chamber and reacts with the fuel. The resulted combustion gases stream at point 4 is feed into the turbine to generate power. The exhaust combustion gases leave the turbine at point 5. In the same configuration as shown in control volume CV.2, the gas turbine cycle is coupled to steam cycle via HRSG. Firstly, turbine exhaust gas passes through after-burner duct to elevate its temperature from$T_{a,5}$ to $T_{a,6}$. Then the high temperature exhaust gas enters the HRSG to release its energy. the feed water enters to the economizer as a compressed liquid at point 16 to raise its temperature to 17, which is the saturated temperature at boiler pressure. Latent heat is acquired from point 17 to 18 throughout the evaporator and the fluid becomes saturated vapor. Finally the saturated vapor is superheated to point 10 by the super heater. At the end of the process, the exhaust gas will leave the plant at $T_{a,9}$. In this case, the pressure drop at the gas turbine exit will increase and must be updated. Then the steam enters the steam turbine atT_{10}, P_{10} and expands to T_{12} ,P_{12}. At a specified point 11, some amount of steam is extracted for the open feed water heater. the function of feed water heater is to preheat the boiler feed water and thereby minimize the thermal stresses on boiler tubes. Besides, the open FWH works as deaerator that discharges the leaked air into the steam cycle. At point 12, in the condenser, wet steam is condensed to point 13 by using cooling water. Then water pumped by two pumps at points 13 to 14 for pump 1 and 15 to 16 for pump 2.

In figure (2), the MSF unit that coupled to the condenser of the CC system. The selected scheme is the once through MSF system. One important requirement to develop the cogeneration cycle is lifting the condenser pressure. This will produce the required motive steam needed to run the MSF system. In this case, the steam condenser is termed as "Brine Heater". As it is clear, the system includes a numbers of stages,n and the brine heater (in this case the condenser of CC system). The stage elements includes flashing chamber, which contains the condenser-preheater tubes, the demister, the brine pool, and the collecting distillate tray. The temperature of intake water (saline feed water) is increased as it flows through the preheater tubes of each stage from

$T_{F,k+1}$ to $T_{F,k}$ .The intake water flows from stage n to stage 1, i.e., from the low temperature to the high temperature side of the unit. The saline feed water leaving the last stage enters the brine heater at $\dot{m}_F$, where its temperature is increased from $T_{F,1}$ to $T_{B,o}$. The heated brine via rejected steam at $\dot{m}_s$ from the steam turbine flashes off as it flows through the successive stages, where its temperature decreases from $T_{B,k-1}$to $T_{B,k}$. Simultaneously, the flashing vapor condenses around the condenser tubes in each stage, where it heats the cooling water supplied through the tubes. The collected distillate in the distillate-collecting tray flows across the stages, where it leaves the plant from stage$n$. The flashing process reduces the brine temperature and increases its salinity from X_{k-1} to X_k. The brine leaving the last stage is rejected back to the water source. The temperature distribution in the once through MSF system is defined in terms of four temperatures; these are the temperatures of the steam, T_s, the brine leaving the pre-heater (top brine temperature), $T_{B,o}$, the brine leaving the last stage, T_B, and the feed saline water, T_F[9].

Table (1) Properties of Fuel Gas Used in Shatt AL Basra GTPP

Fuel Type	*Fuel Gas, %by volume*		*LFO, %by mass*		*HFO, %by mass*	
*Composition %**	CH_4 C_2H_6 CO_2 N_2 C_3H_8 nC_4H_{10} nC_5H_{12} iC_4H_{10} iC_5H_{12}	75.2 17.05 1.91 1.1 4.2 0.3 0.01 0.22 0.01	%C %H %S	85.5 11.5 3	%C %H %S	85. 5 11. 5 3
Density *LHV* *HHV*	0.86 kg/m^3 46.256 MJ/kg 53.5 MJ/kg		0.93 kg/m^3 40.6 MJ/kg 43.02 MJ/kg		0.97 kg/m^3 39.57 MJ/kg 41.83 MJ/kg	
For ISO conditions only 100% CH_4 is considered						

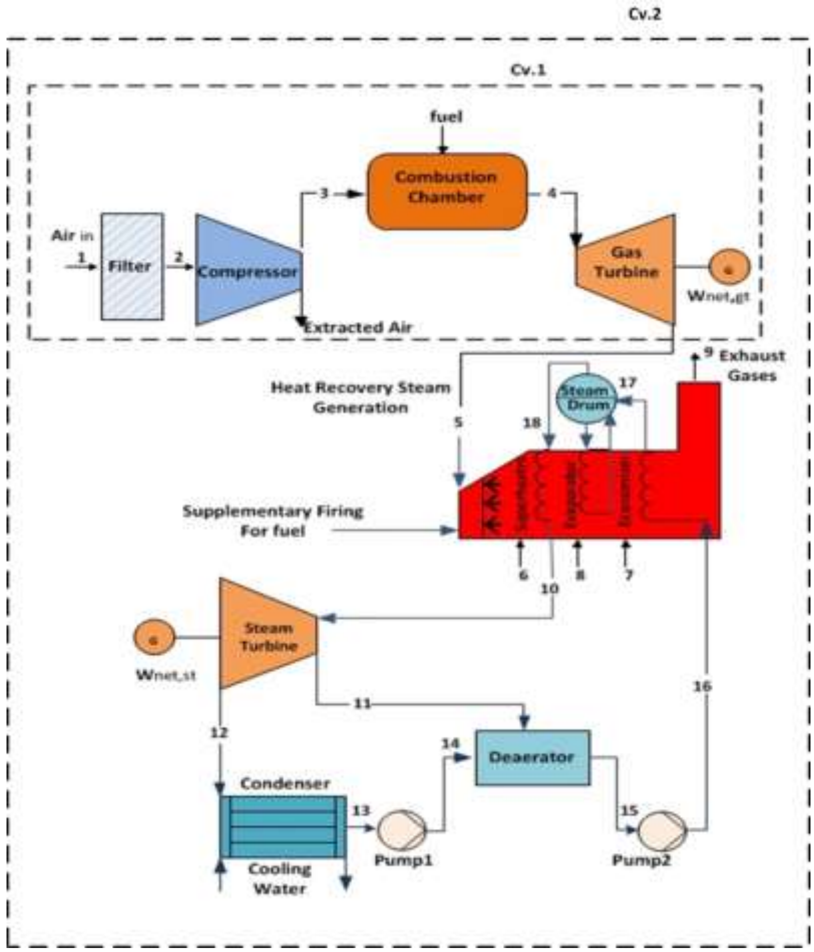


Fig.(1) Schematic Diagram of Simple Gas Turbine with Steam Turbine as a Combined Cycle.

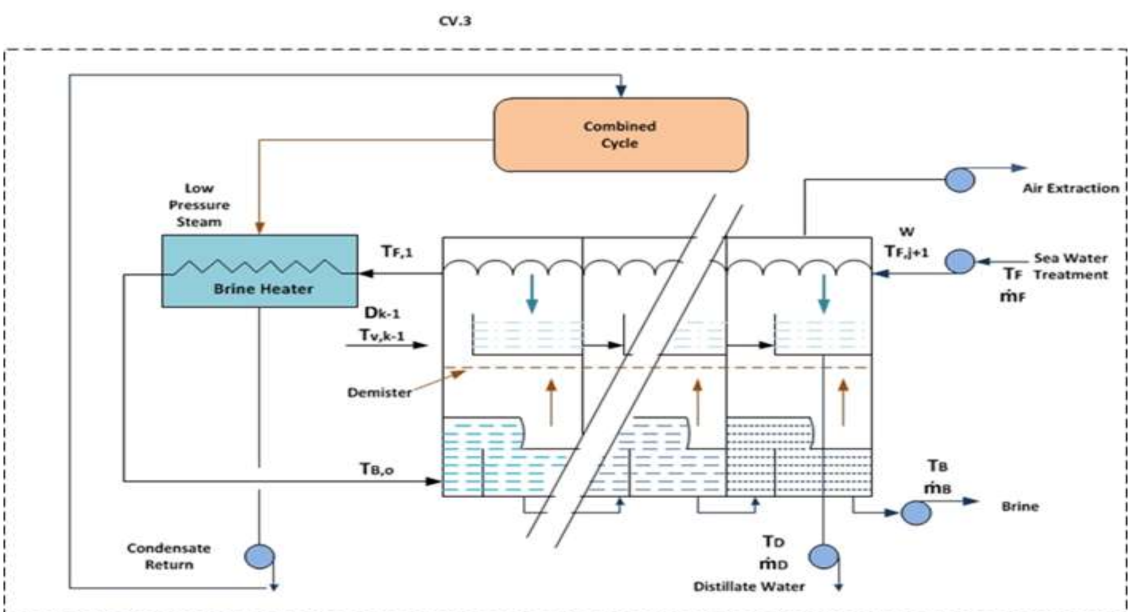


Fig.(2) The MSF-Unit as Cogeneration Cycle

3. Mathematical model

A schematic of a combined cycle power plant and its representation on the T-S diagram are shown in Fig (**3**). The following assumptions are taken into consideration:

1- For all plants modules the operation are steady state.

2- For both energy and exergy terms the potential and kinetic are neglected.

3- The reference condition for GT cycle is taken as P_o=101.325 kPa with variable reference dead state temperature. The reference conditions for ST cycle are kept constant and not varies with ambient temperature as T_o =25℃, P_o=101.325 kPa.

4- In compressor inlet, combustion chamber, turbine exit, and HRSG the pressure losses are considered and it is not considered in the ST power plant and multi stage flash desalination.

5- The air and combustion gases are assumed to be a mixture of ideal gases with variable properties as a function of temperature and pressure.

6- The processes are considered adiabatic, in the turbines, compressor, pumps, combustion chamber and any heat exchangers,.

7- In the steam condenser, deaerator, and economizer of HRSG, the water leaving is considered a saturated liquid.

8- In HRSG, the steam leaving the evaporator is considered as saturated vapor.

9- For turbine blades, the cooling air is extracted at the final compressor stage.

10- By MSF, the distillate water produced is salt free.

11- In the MSF, the effects of the non-condensable gases are neglected.

12- Linear temperature profile for the brine water across the MSF unit.

13- The effects of the boiling point rise and non-equilibrium losses on the stage energy balance are negligible.

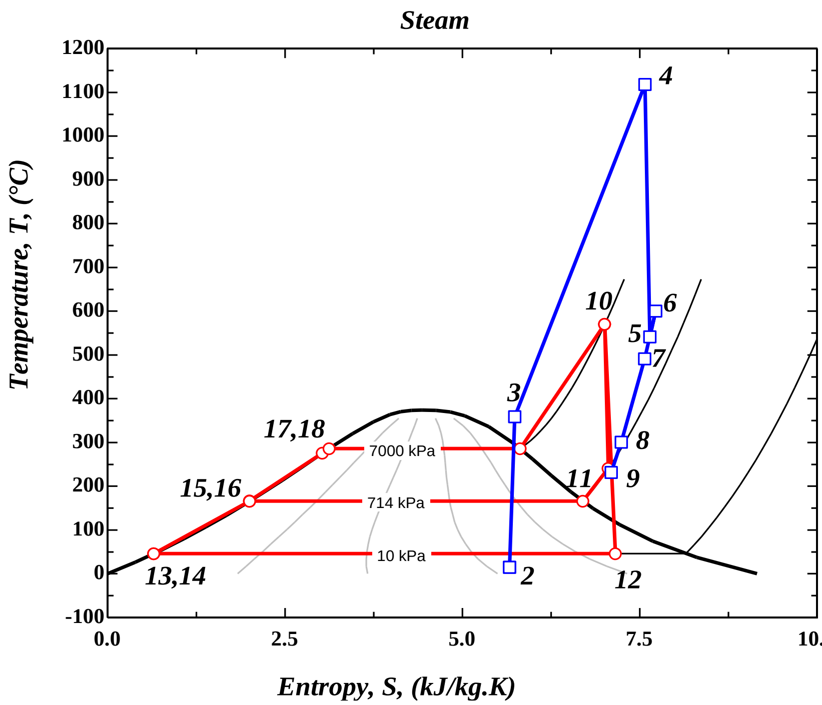


Fig.(3) T-S Diagram of Gas-Steam as Combined Cycles.

3.1. General energy analysis

The energy conservation equation for open system is depended on the first law of thermodynamics, [1]:

$$\dot{W_{Net}} = \dot{Q} + [(mh)_{inlet} - (mh)_{outlet}] \qquad ..(1)$$

The mass conservation equation is :

$$\sum m_{inlet} - \sum m_{outlet} = 0 \qquad ..(2)$$

3.2. General exergy analysis

From the second law of thermodynamics, the general equation of exergy balance is [1]:

$$\Psi_w = \sum(1 - \frac{T_o}{T_j})\dot{Q}_J + \sum[(\dot{m}\psi)_{inlet} - (\dot{m}\psi)_{outlet}]_k - T_o \cdot S_g \qquad ..(3)$$

The flow exergy of the working fluid is defined as[1]:

$$\psi = (h - h_o) - T_o(s - s_o) \qquad ..(4)$$

Where:

$T_o S_g$: represents the irreversibility I or the exergy destruction in the system.

For open system with steady state flow, the entropy generation S_g can be using entropy balance equation and is gives as[1]:

$$0 = \sum \frac{\dot{Q}_J}{T_j} + \sum(\dot{m} \cdot s)_{outlet} - (\dot{m} \cdot s)_{inlet} - S_g \qquad ..(5)$$

3.3 Performance Characteristics of the Gas Turbine Cycle

The power output of gas turbine cycle is given by[1]:

$$\dot{W}_{Net} = \frac{\dot{W}_G - \dot{W}_C}{\eta_{EG}} \qquad (6)$$

Where η_{EG} is the electric generator efficiency.

The lost heat to the environment is:

$$\dot{Q}_{Exh} = \dot{Q}_{in} - \dot{W}_{Net} \qquad (7)$$

The thermal efficiency of the gas turbine cycle is based on the first law of thermodynamic. It is gives by [1]:

$$\eta_{th,G} = \frac{\dot{W}_{Net}}{\dot{Q}_{in}} \quad (8)$$

The lost exergy from the gas turbine cycle is solely due to the exhaust gases discharged to the environment. It is given by[3]:

$$EX_{Exh} = EX_{fuel} - \dot{W}_{Net} - \sum I \quad (9)$$

Where the total exergy destructed is given by:

$$\sum I = I_C + I_G + I_{CC} \quad \ldots(10)$$

The second law efficiency of the gas turbine cycle is based on second law of thermodynamic. It is gives as[3]:

This equation can be written in terms of exergy destructed as:

$$\eta_{II,G} = 1 - \frac{\sum I_{des}}{EX_{fuel}} \quad (11)$$

3.4 Performance Characteristics Combined cycle:

The power output of combined cycle is given by[1]:

$$\dot{W}_{Net,GS} = \frac{\dot{W}_G - \dot{W}_C + \dot{W}_S - \dot{W}_P}{\eta_{EG}} \quad ..(12)$$

The heat input to the cycle is:

$$\dot{Q}_{in} = \left(\dot{m}_{fuel,CC} + \dot{m}_{fuel,AB}\right)\left(LHV + h_{fuel@T_{fuel}} - h_{fuel@15℃}\right) \quad ..(13)$$

The thermal efficiency of the combined cycle is given by[1]

$$\eta_{th,GS} = \frac{\dot{W}_G - \dot{W}_C + \dot{W}_S - \dot{W}_P}{\left(\dot{m}_{fuel,CC} + \dot{m}_{fuel,AB}\right)\left(LHV + h_{fuel@T_{fuel}} - h_{fuel@15℃}\right)} \quad ..(14)$$

The lost exergy from the combined cycle due to the exhaust gases discharged to the environment is given by[5]

$$EX_{Exh,GS} = \left(\dot{m}_{P,5} + \dot{m}_{fuel,AB}\right)\psi_{P,9} \quad \ldots(15)$$

The lost exergy from the condenser due to the hot cooling water is given by[]

$$EX_{lost,Cond} = \dot{m}_{cw}.\psi_{cw,o} \quad ..(16)$$

The total exergy destructed is given by[5]:

$$\sum I = I_C + I_G + I_{CC} + I_S + I_{Cond} + I_{F.W.H} + I_P + I_{HRSG} \quad (17)$$

The second law efficiency of the gas turbine cycle is based on second law of thermodynamic. It is gives as[5]:

$$\eta_{II,GS} = \frac{\dot{W}_{Net}}{EX_{fuel,GS}} \quad (18)$$

3.5 Performance Characteristics of Cogeneration cycle:

Most of performance characteristics of the cogeneration cycle are still described by same equation given in section (3.3). However, some parameters that related to MSF unit have to be developed. The performance ratio, PR of the MSF is the defined as the amount of distillate product produced per unit mass of the heating steam. This is[10]:

$$\mathrm{PR} = \frac{\dot{m}_{Dist.}}{\dot{m}_{steam}} \quad ..(19)$$

The lost exergy from the condenser due to the exit hot brine and distilled water is given by[10]:

$$EX_{lost,MSF} = \dot{m}_{Dist}.\psi_{Dist} + \dot{m}_B.\psi_B \quad ..(20)$$

The total exergy destructed is given by[10]:

$$\sum I = I_C + I_G + I_{CC} + I_S + I_{MSF} + I_{F.W.H} + I_P + I_{HRSG} \quad (21)$$

4. Results and Discussion

Three configurations will be adopted. The first one is GT cycle only and the second one is CC. The last one is CC coupled with MSF unit. The assumptions and reference data required are shown in Table (2). The standard model specification for GT under ISO operating conditions are given in Table (3). The selected fuel for this study is 100% Methane (CH4). The complete modeling for this study was done using the (The Engineering Equation Solver (EES)) software program. By using the "Iteration Mode" for solving. The EES is able to solve rapidly large numbers of equations. The EES can been used to perform parametric studies in which the effect of a specified input variable could be easily investigated.

Table (2) Assumptions and Reference Data.

ISO Conditions 15 ℃, 101.325 kPa			
Assumptions for three plants			
1. Gas Turbine [G]		***2. Combined Cycle [GS]***	
Turbine Cooling Air Fraction	12.5%	Reference temperature	25℃
$\eta_{m,C} = \eta_{m,G} = \eta_{EG.}$	95%	Steam temperature	570℃
$\eta_{is,C}$	87%	Steam pressure	70 bar
$\eta_{is,G}$	92%	Condenser pressure	10 kpa
η_{CC}	98%	Pinch point of HRSG	15℃
ΔP_c	0.01%	Approach temperature HRSG	10℃
ΔP_{cc}	0.01%	Pump efficiency	90%
ΔP_G	0.01% for GT	for CC	0.02%
Fuel temperature	15℃	Firing temperature after burner	600℃
——	——	$\eta_{is,GS}$	90%

3. Multi Stage Flash[MSF]	
Type Condenser pressure	Once through 84.53 kpa
Number of stages	23
Motive steam temperature	95℃
Feed water temperature	25℃
Waste brine temperature	40℃
Terminal temperature difference of condenser	3℃

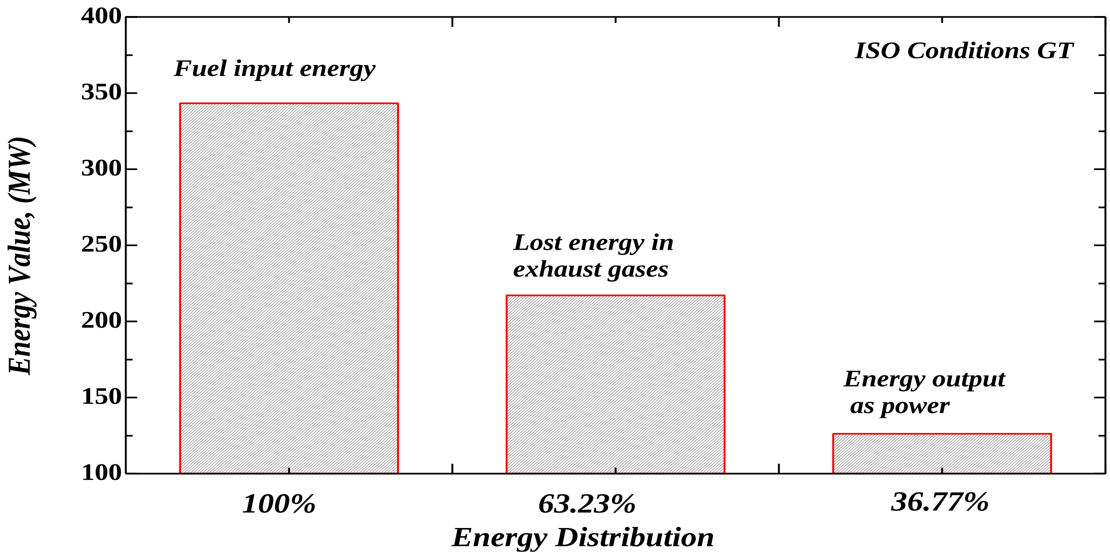
Thermodynamic loss	2℃
Salinity of feed water	42000 ppm

Table (3) GT (M9001E or Frame 9) Gas Turbine Model Specification (ISO Standard and Predicted Values)

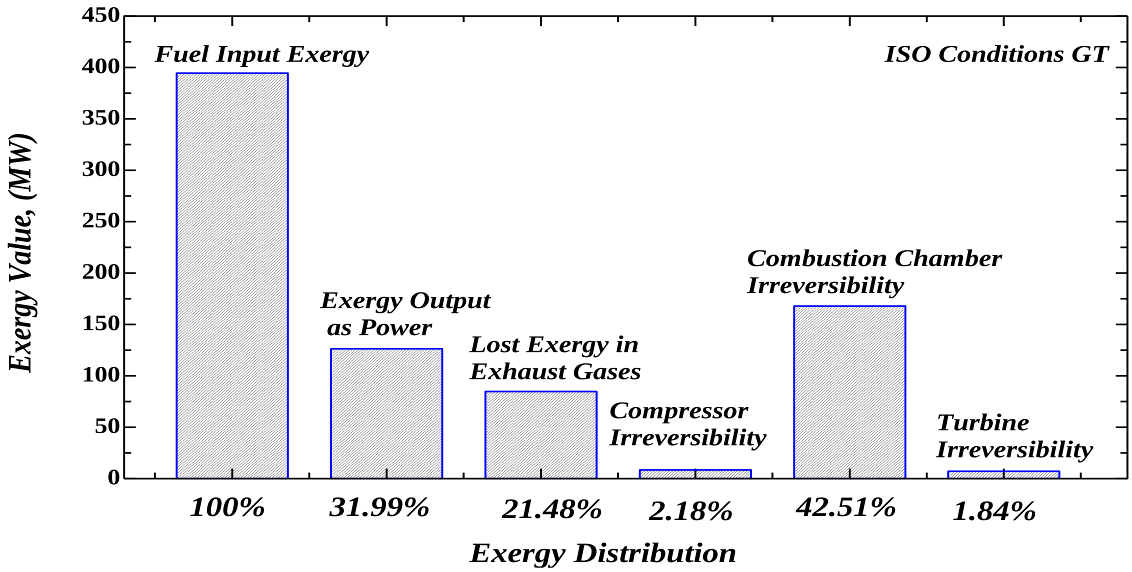
	ISO Standard value	***Predicted***	***%Error***	***Remark***
Net power output, (MW)	126.1	126.2	0.1	-
Compression ratio	12.6	12.6	-	As input
Air mass flow rate,(kg/s)	407	397.5	2.3	-
Fuel mass flow rate,(kg/s)	Not Record	6.944	-	-
Exhaust gas temperature,$°C$	543	541.3	0.31	-
Firing Temperature, ,$°C$	1124	1129	0.44	-
Thermal Efficiency	33.8	36.7	8.6	-

4.1 Energy and Exergy Analysis for Full Load (ISO Operation Conditions).

Fig (4) and fig (5) shows the energy and exergy distribution among the unit equipment's in GT system. These two figures are just for ISO operation conditions. The two figures describe that the fuel input represents the main source of energy and exergy supplied to the unit. Their values depend on the fuel mass flow rate. The output power accounts 36.77% from the input energy and 31.9% from input exergy. These two percentages and exactly the definition of thermal efficiency and second law efficiency of the units. The two figures indicate the unit discharges 63.23% of the input energy to the environment. This waste heat has a great work potential that account 21.48% from the input exergy. This is due to the higher exhaust temperature of 543°C. Unfortunately, this lost energy is not utilizing yet. Concerning exergy destructed, it is founding that the combustion chamber has the highest exergy destructed among the plant equipment's, which accounts 42.51% from the input exergy. This high irreversibility in this component was attributing to the high temperature difference found in this component accompanied with no work interaction with the environment. It is founding that the exergy destruction in the air compressor and turbine are too small comparing to that in the combustion chamber due to the work interaction found in these two components

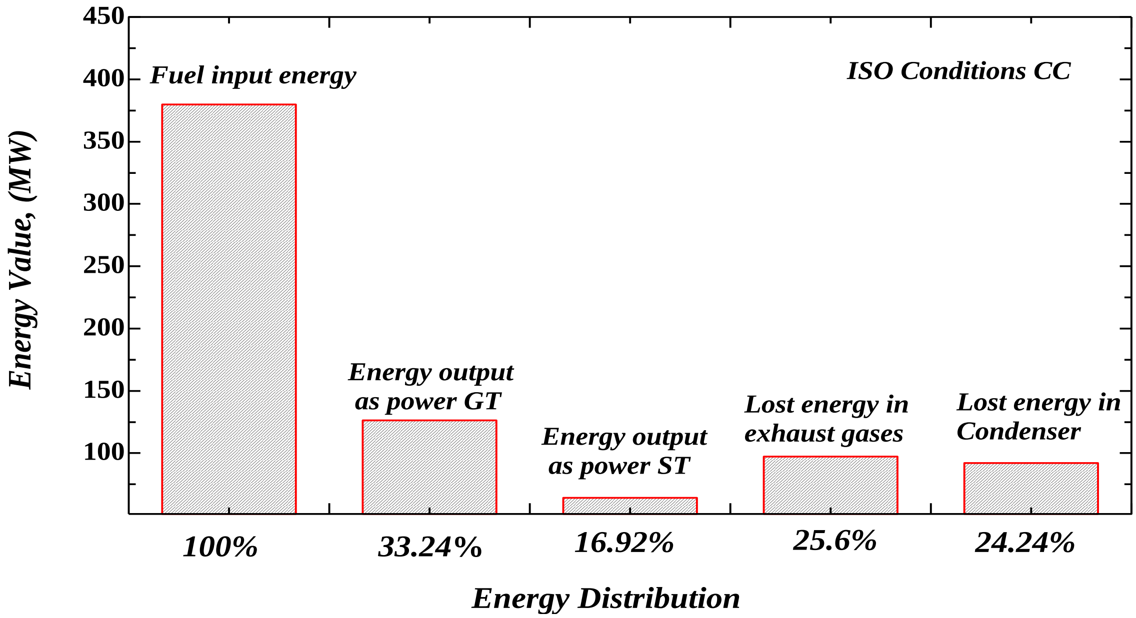


Fig(4) Energy Distribution of Shatt Al-Basra GT Power Plant.

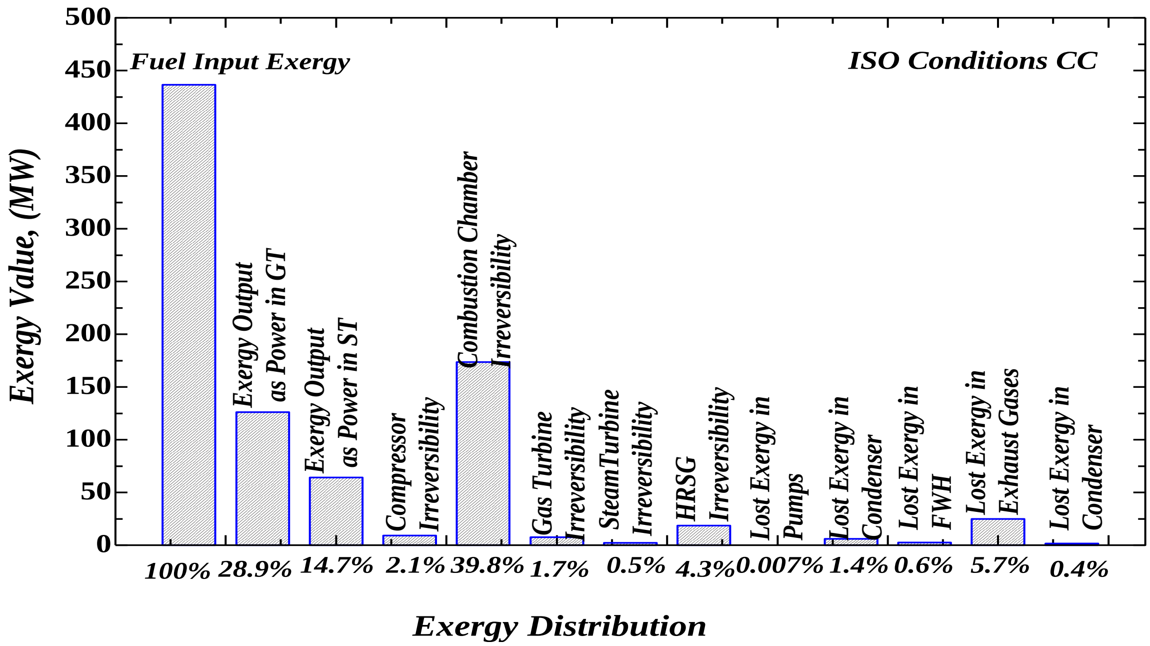


Fig(5)Exergy Distribution of Shatt Al-Basra GT Power Plant.

Fig(6) displays the energy distribution predicted when the unit is the improved to CC. By introducing HRSG to utilize the waste heat discharged to the environment, the power output can be improve to be 190.4MW while adding 64.2 MW from the steam cycle. Waste heat is now discharging from the condenser and chimney at lower temperatures. In fig (7) shows the predicted exergy distribution of gas turbine unit if CC installed. The figure shows that the total exergy destructed is 219.4MW. The lost exergy in exhaust gases was lowering to 24.9MW and the lost exergy in the condenser cooling water accounts 1.7MW.

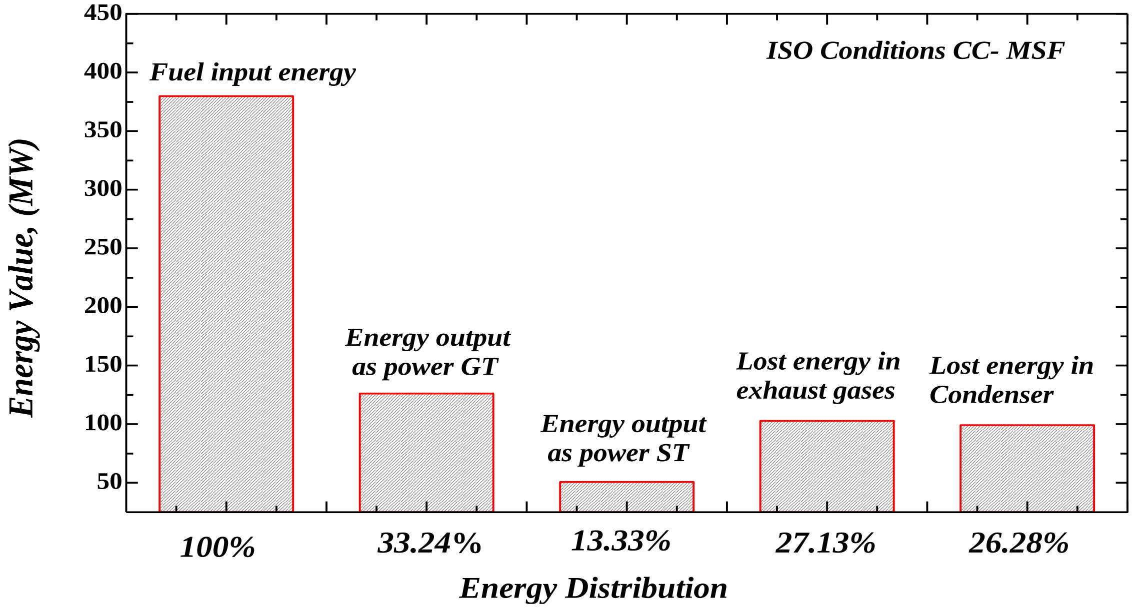


Fig(6) Energy Distribution of Shatt Al-Basra as Combined Cycle Power Plant.

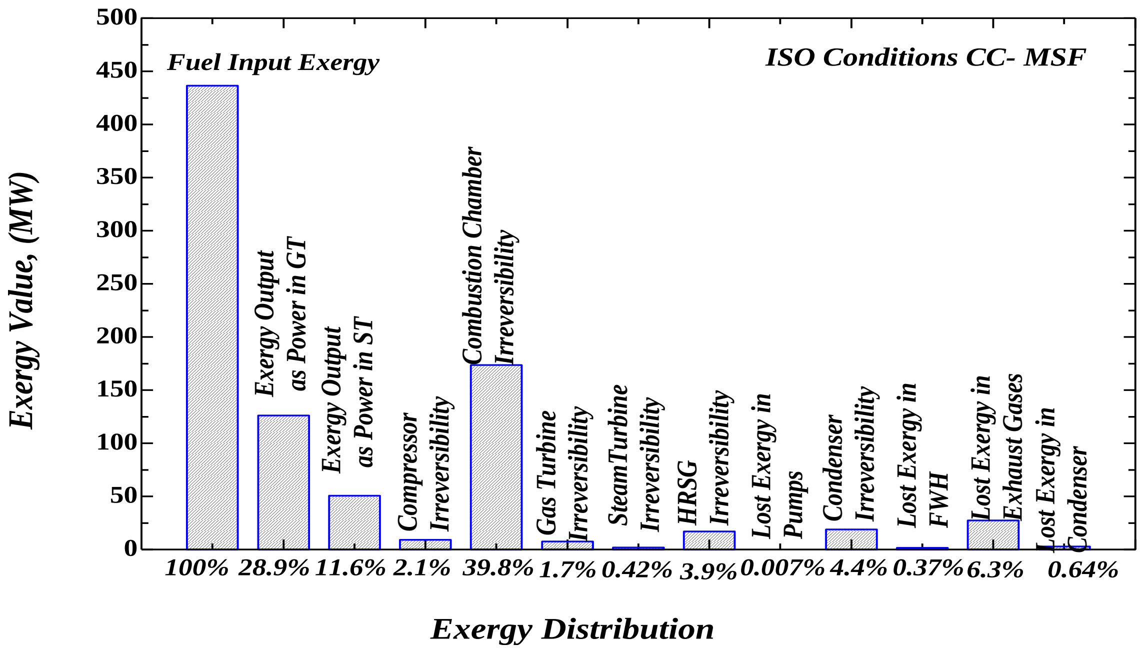


Fig(7) Exergy Distribution of Shatt Al-Basra as Combined Cycle Power Plant.

Fig(8) shows the energy distribution for the CC-MSF unit. By comparing with fig (6), it is clear that the energy distribution concerning GT unit is the same. The main change was occurring in the steam turbine power. It is decreased from (16.92% to 13.33%). In the other side, the heat discharged in the condenser is increased from (24.24% to 26.28%). This extra heat discharged is very important to run the MSF unit. Moreover, the waste heat lost in exhaust gases is also increase from (25.6% to 27.13%). Fig (9) shows the exergy distribution for CC-MSF. By comparing, the results obtained in case of CC cycle only (fig (7)), it is clear that the power output from the steam turbine is decreased from (64.24 MW) to (50.62MW) due to this modification. Besides, exergy destruction in steam turbine and HRSG has been decrease. All of these aspects are resulted from increasing condenser pressure to higher values. The most important aspects can been see on the condenser unit that plays important role in the operation of MSF unit. Exergy lost from the condenser will increase due to the production of distilled water and brine at higher temperatures. The increase in temperature difference in condenser will certainly increase the exergy destructed in this component.



Fig(8)Energy Distribution of Shatt Al-Basra as CC-MSF Power Plant.



Fig(9)Exergy Distribution of Shatt Al-Basra as CC-MSF Power Plant.

4.2 Energy and Exergy Analysis for Part Load(Actual Operation Conditions)

The variation of actual power production from the GT, CC and CC-MSF unit during one year describe in fig (10). Maximum power production was recording in February and minimum power was recording in November. The variation in power production from three units was attributing to two main reasons. The first one is the change in ambient temperature as indicated in fig(11). Low ambient temperatures were recording in January and February while the highest ambient temperatures were recording in July and August. The low ambient temperatures will result in higher air density. This will certainly improve the air mass flow rate passing through the compressor and thereby the unit produces higher power. Inlet guide vans (IGV) was the second reason of variation that effect on the power production. These vans has fixed on the compressor inlet to control the air mass flow rate drawn by the unit and in that way controlling the load applied on the engine. Machinists always choose the cold season to carry out periodic maintenance of the unit preparing it for the stressful service during the harsh summer. After the summer season, unit meets many problems that lead the machinists to reduce the load on the unit. This is the reasons for why October and November have low power production. Ambient temperature and IGV are effect on the compressor pressure ratio and this factor is very critical on the behavior of gas turbine performance.

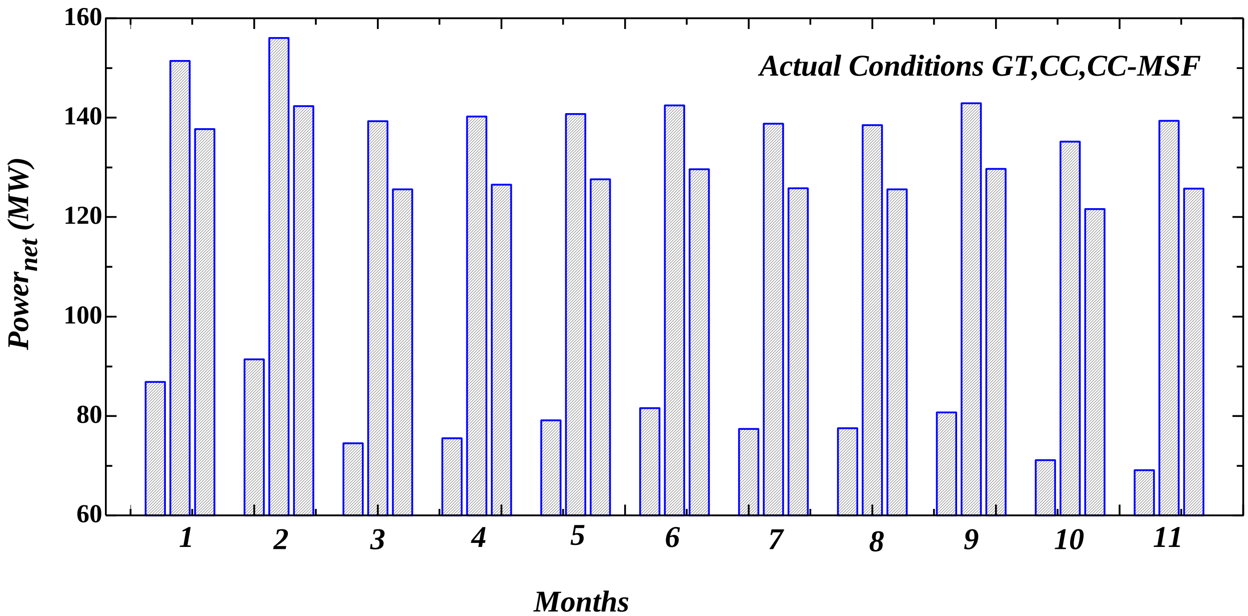


Fig (10)Power Distribution of Shatt Al-Basra GT, CC, CC-MSF Power Plant for one Year.

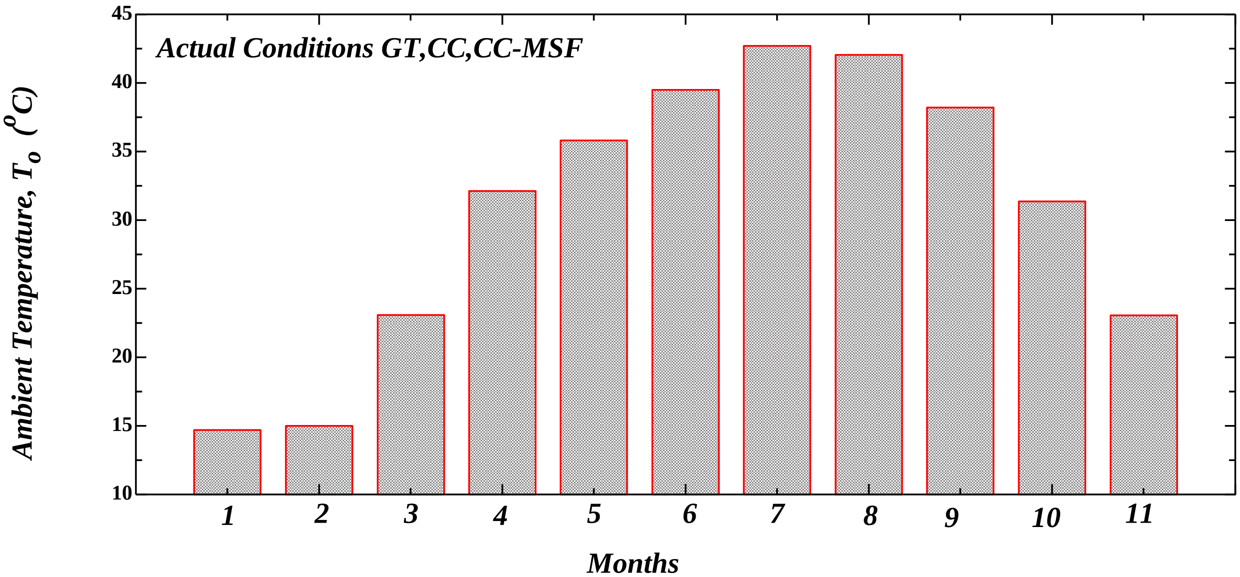
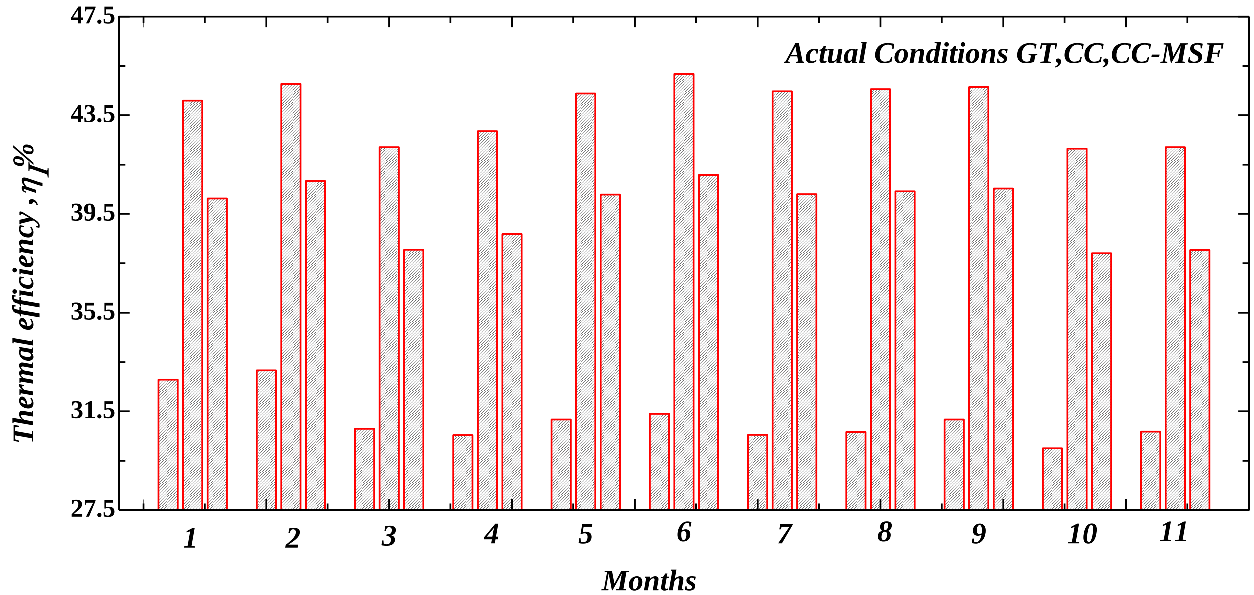
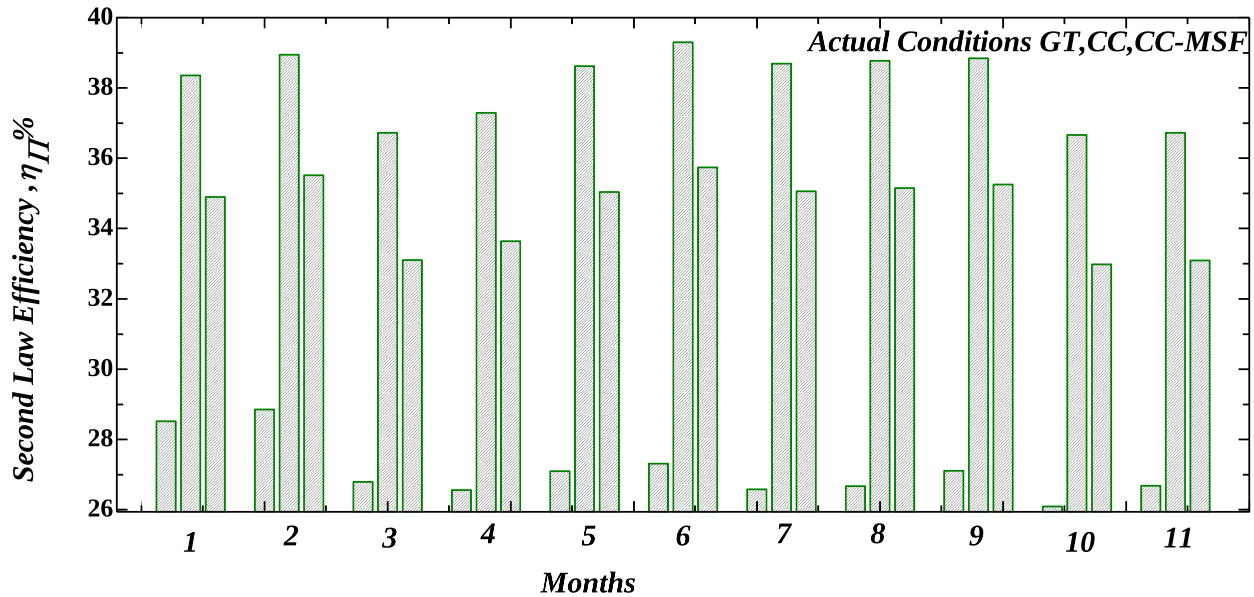


Fig (11) The variation of ambient temperature during the year 2017 (average).

Fig **(**12**)** shows the variation of actual thermal efficiency of three units during the year 2017. Maximum thermal efficiency of GT unit was founding to be in January and February while minimum thermal efficiency was founding to be in October. The behavior of this parameter was directly relating to the power output and fuel mass flow rate. In CC unit, Best improvement was noting in June. That is mainly due to the exhaust temperature. In June, the exhaust temperature is higher and this will minimize the fuel required to operate the after burner, which causes the thermal efficiency to be the top. For the CC-MSF, again maximum thermal efficiency was observing in June due to the high exhaust temperature discharged by the GT unit. In fig **(**13**)** shows the distribution of second law efficiency during one year for GT, CC, and CC-MSF units. The change in these parameters are exactly same as thermal efficiency due to the difference in energy and exergy definition of fuel

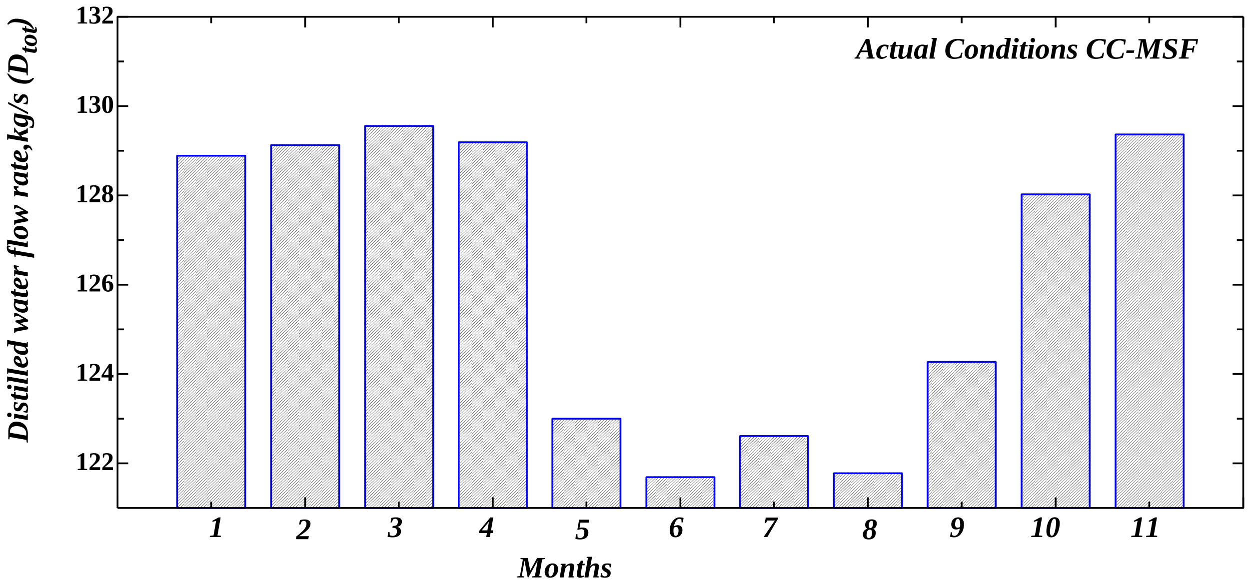


Fig(12)Variation of Thermal Efficiency of Shatt Al-Basra GT, CC, CC-MSF Power Plant for one Year.



Fig(12)Second Law Efficiency Variation of Shatt Al-Basra GT, CC, CC-MSF Power Plant for one Year.

when MSF unit was attaching to the combined cycle. In this case, the power enhancement will decrease as shown in figure (10). This power was used to produce pure distill water. Fig (13) shows the distribution of distilled water production predicted from the MSF unit. It was founding that the range of flow rate is 122 to 129.5 kg/s.



Fig(13)Distilled water flow rate, kg/s (D_tot) Variation of Shatt Al-Basra as CC-MSF Power Plant for one Year.

5. Conclusions

This study display the development of the performance of Shatt Al-Basra GT power plant in terms of energy and exergy .

1- The system of GT has affected by the operation conditions of this engine. These conditions are pressure ratio ambient temperature and firing temperature effect on the performance from load and hours operation .
2- The performance of CC depended on operation conditions of power plant. These conditions are pressure ratio, ambient temperature, firing temperature
3- The performance characteristics of MSF which is (the thermal performance ratio (PR)) in depend of the pressure ratio, ambient temperature and firing temperature and could vary only with the ratio of distilled flow rate to the heating steam
4- The results and analysis of Shatt Al- Basra GT power plant for one year show that the load capacity ranged has varied from season to season and for part load and full load operation. The maximum value of power output in summer is about 81.59 MW. While in winter is about 91.36 MW.
5- relative to ISO operation conditions, the results show the capability of enhancing of Shatt Al-Basra GT power plant by inserting the CC. this modification has improved the power output by 190.44 MW (126.2 MW for GT and 64.24 MW for ST) in case of full load condition. While the first law efficiency has improved from 36.77% to 50.16% at full-load. For case 3, CC coupled with MSF unit, the power output is 176.8 MW (126.2 MW for GT and 50.62 MW for ST). Also, the thermal efficiency has changed from 50.16% to 46.57%.
6- The amount of energy and exergy lost in GT, CC, and CC-MSF, for ISO operation conditions are (63.23%, 49.84%, 53.41%) and (68.01%, 56.4%, 59.5%) respectively. The results show the capability of modifying the actual plant by inserting the technology of MSF desalination with

CC plant, dual purpose plant or combined plant, for production of distill water and electricity together.

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Nomenclature

English symbols		Subscripts	
EX	Exergy term.	AB	After burner
		C	Air compressor.
h	Specific enthalpy.	CC	Combustion chamber.
		$Dist.$	Distilled
		$Exh.$	Exhaust
I	Irreversibility.	gen	Generator.
m	Mass flow rate.	G	Gas turbine.
P	Pressure.	I	First law.

Q	Heat power.	II	Second law.
s	Specific entropy.	in	Input.
		is	Isentropic
Sg	Entropy generation.	$inlet$, $outlet$	Inlet and exit.
T	Temperature.	$lost$	Lost.
W	Work power.	m	Mechanical.
PR	The Performance Ratio	Net	Net value.
		$HRSG$	Heat Recovery Steam Generator
		P	Pump
		GS	Gas- Steam turbine

Greek symbols

η	Efficiency.	o	Denotes to the dead state.
Δ	Difference .	out	Out.
ψ	Flow physical exergy.	t	Turbine.
Ψw	Exergy due to work.	tot	Total.

Abbreviations

EES	Engineering equation solver.
EG	Electrical Generator
CC	Combined Cycle.
HRSG	Heat Recovery Steam Generator.
ISO	International standards organization.
HFO	Heavy fuel oil.
LFO	Light fuel oil.
F.W.H	Feed water heater
HHV	Higher heating value.
LHV	Lower heating value.
MSF	Multi Stage Flash

Superscripts

ch	Chemical