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Parametric Study of Combined and Cogeneration Systems: Energy and Exergy Approach

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Abstract
In these papers a number of parameters have been studied to show their effect on performance of combined cycle and cogeneration system the in terms of energy and exergy principles. The analysis of every component in the system apart was tested in accordance with the first and second law of thermodynamics was one of the main goals of this study will explain and estimate the sites which have exergy losses. This analysis offers a substitute plan to make sure greater performance of a power plant for two parts in electricity and yield of pure water.

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

1. Introduction

The Combined cycle power plant as shown in Figure (1) includes in the left gas turbine cycle i.e. compressor, combustion chamber, gas turbine and in the right side steam cycle i.e. heat recovery steam generator (HRSG), steam turbine, and condenser. The efficiency of power plant that based on gas turbine, as core engine can be increase remarkably by what is called combined cycle (CC). The overall efficiency of combined gas –steam power plant can be in range (45%-58%) [1]. Increasing power plant efficiency along with reducing the emissions of greenhouse gases make CC is the preferred power plant worldwide. The CC power plant can utilize the clean and cheap natural gas as fuel. The GT cycle produces almost 2/3 of the CC electric power output, with mainly fuel supplied to the combustion chamber. Hot gases exhausted from the GT (typically at 475–600 C°) are directed to HRSG to produce steam[1]. This steam, when at high-enough temperature and pressure, was directing to steam turbine to produce more work without adding more fuel. The steam cycle produces about 1/3 of the CC power output[2]. When heat in the form of steam produced from the HRSG or extracted (or discharged) from steam turbine was using for any useful process, the system is called "Cogeneration" Figure (2) shows the Cogeneration Power Desalting Plant[3]. Most power plants in the gulf countries are cogeneration power-desalting plants producing both desalted seawater and electric power in single. The desalting units were suppling with its needed low-pressure (LP) steam by extracting (or discharging) steam from the steam turbine of the CC or directly from the HRSG when the steam turbine was not operating or does not exist. The steam turbine used in the cogeneration plant can be extraction -

condensing steam turbine or back - pressure steam turbine discharging all of its steam to the desalination plant[3].As combined cycle technology is imperative, it has been studied and analyzed by several research projects. **Hatem**[4]carried out parametric exergy calculations for 125MW combined power plant. The studied parameters include pressure ratio, turbine inlet temperature, steam pressure, and ambient temperature.The results show that more exergy losses are occurred in the gas turbine combustion chamber.which could reach 33% of the consumed fuel exergy. The exergy losses in other plant components are about 1% - 5% of the consumed fuel exergy.**Fathi**[5]**,** studied the outcome of repowering Beijee gas turbine plant located in Iraq to combined cycle. The plant was modeled and simulated to observe its performance in terms of output power, thermal efficiency, specific fuel consumption, and exhaust gas temperature.Results indicate that a remarkable improvement in the performance when a combined cycle mode is adopted.The power output is predicted to increase about 49.3%, the thermal efficiency could increase by 20% while specific fuel consumption may decrease nearly by 10%.**Jabboury et al**[6]**,**investigated the effect of the operating parameters of heat recovery steam generators on the performance of combined cycle coupled to sea-water desalination plant..The parameters include the pinch point, approach temperature, first and second stage pressures, and part load condition.Results indicates that the product distilled water output decreased from 8.55 mg d^{-1} to the low value of 3.00 mg d^{-1} when operating the plant under 17% part load . They recommended the use of supplementary fired HRSG since this will keep stable operation of the desalination plant under different cycle loading conditions.

2. Combined Cycle and Cogeneration (Multi Stage Flash unit) Plants.

In this combined as shown in figure (1), the gas turbine cycle is coupled to steam cycle via HRSG. Firstly, turbine exhaust gas passes through after-burner duct to raise its temperature. Then the high temperature exhaust gas enters the HRSG to release its energy[7]. At the end of the process, the exhaust gas will leave the plant at T_5. In this case, the pressure drop at the gas turbine exit will increase and must be updated.

The steam developed by the HRSG will fed to the steam turbine at point **6** to give the extra work. The condenser receives the expanded steam at point **7** with lower temperature and pressure [6].

The objective of the following section is to analyze the MSF unit that coupled to the condenser of the CC system. The selected scheme is the once through MSF system[3]. 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”[3]. A schematic diagram for the once through MSF system is shown in figure (2). 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 process variables in the flashing stages are shown in figure (2).The stage elements includes flashing chamber, which contains the condenser-preheater tubes, the demister, the brine pool, and the collecting distillate tray [3]. The temperature of intake water (saline feed water) is increased as it flows through the preheater tubes of each stage. In figure (3) 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, where its temperature is increased from stage to another. The heated brine via rejected steam from the steam turbine flashes off as it flows through the successive stages, where its temperature decreases. Simultaneously, the flashing vapor condenses around the condenser tubes in each stage, where it heats the cooling water supplied through the tubes[3]. 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. The brine

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leaving the last stage is rejected back to the water source.

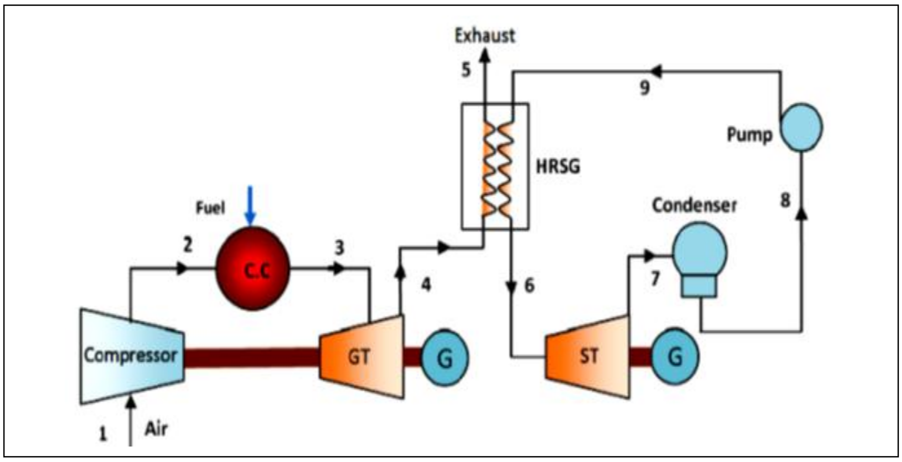

Fig. 1 Schematic Diagram of Simple Gas Turbine with Steam Turbine as a Combined Cycle.

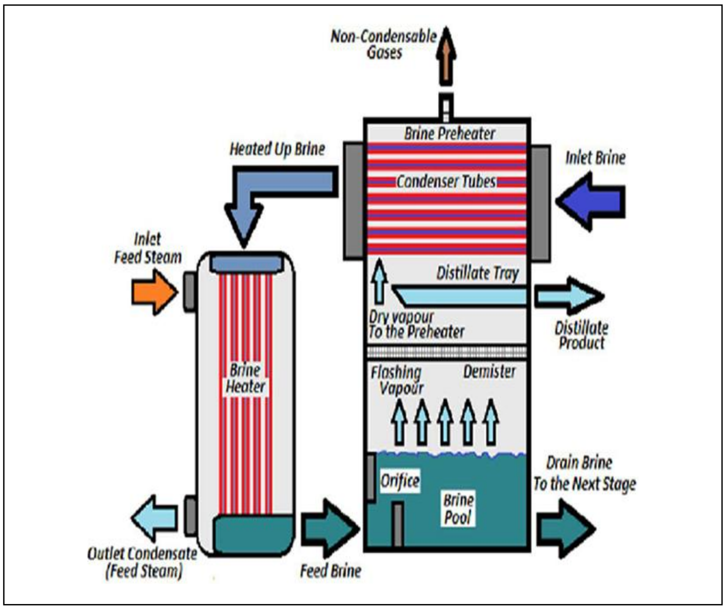

Fig. 2 The MSF-Unit as Cogeneration Cycle.

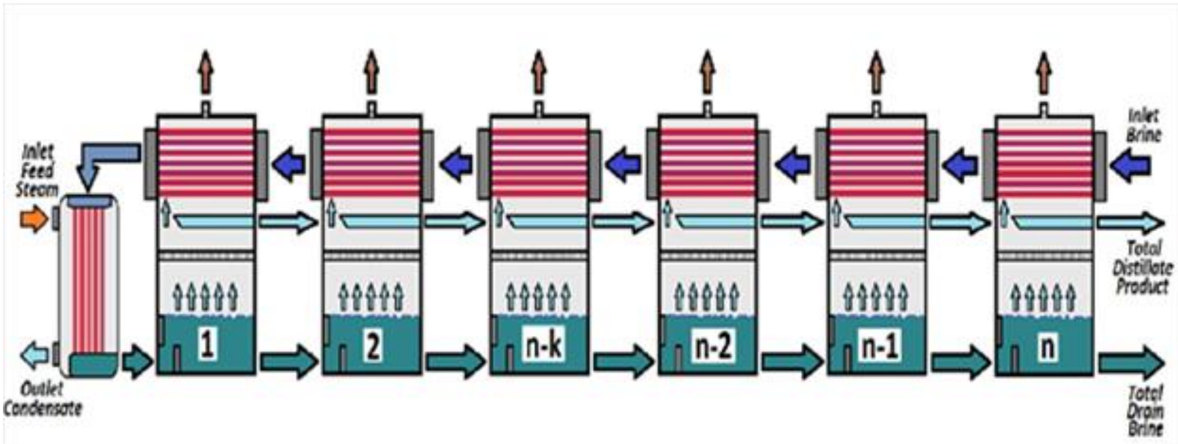

Figure 3 Scheme of a MSF once through plant

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3. Mathematical model

On the T-S diagram are shown in Fig(3). The following assumptions are taken into consideration:

1. The operation is steady state for all plant components andthe potential and kinetic terms for both energy and exergy are neglected.
2. The reference condition for GT cycle is taken as P_0=101.325 kPa with variable reference dead state temperature. The reference conditions for steam power plant are kept constant and not varies with ambient temperature as T_0=25℃, $P_{0,}$=101.325 kPa.
3. Pressure losses are considered in compressor inlet, combustion chamber, turbine exit, and HRSG. It is not considered in the steam power plant and multi stage flash desalination.
4. The air and combustion gases are assumed a mixture of ideal gases with variable properties as a function of temperature and pressure.
5. In the turbines, compressor, pumps, combustion chamber and any heat exchangers, the processes are considered adiabatic.
6. The water leaving the condenser, deaerator, and economizer of HRSG is considered as saturated liquid and The distillate water produced by MSF is salt free and the effects of the non-condensable gases in the MSF are neglected.
7. The steam leaving the evaporator of HRSG is considered as saturated vapor.
8. At the final compressor stage, the Cooling air is extracted for turbine blades.
9. Thebrine water across the MSF unit is considered linear temperature profile.
10. The effects of the boiling point rise and non-equilibrium losses on the stage energy balance are negligible.

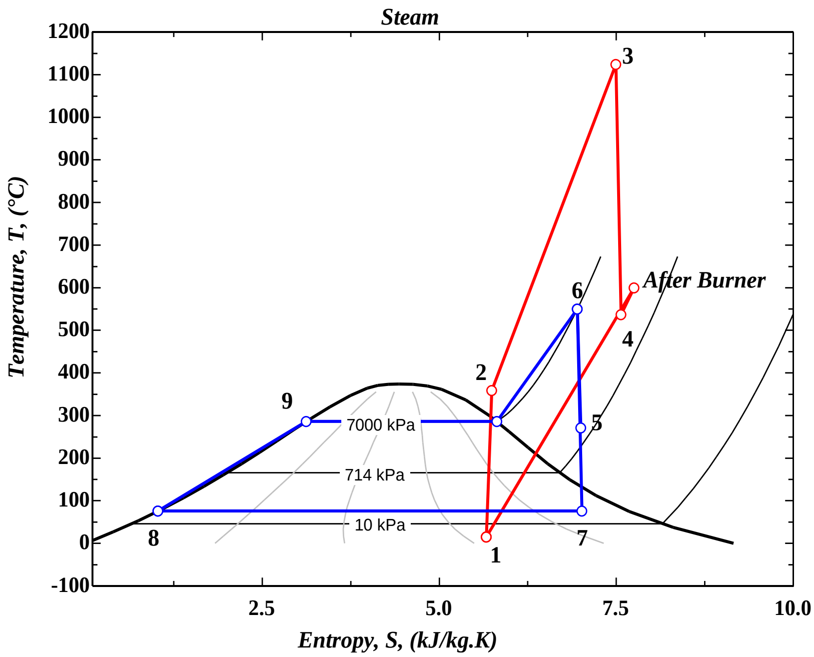

Fig. 3 The T-S Diagram of gas-steam combined cycles.

3.1. Overallenergy analysis

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

$$\dot{Q} - \dot{W}_{net} = m_e h_e - m_i h_i \qquad ..(1)$$

The mass conservation equation is :

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$$\sum m_i - \sum m_e = 0 \quad ..(2)$$

3.2. Overall exergy analysis

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

$$\Psi_w = \sum (1 - \frac{T_o}{T_j})\dot{Q}_j + \sum [(\dot{m}\psi)_i - (\dot{m}\psi)_e]_k - T_o \cdot S_g \quad ..(3)$$

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

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

The term $T_o S_g$ 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 isgives as[1]:

$$0 = \sum \frac{\dot{Q}_j}{T_j} + \sum (\dot{m} \cdot s)_e - (\dot{m} \cdot s)_i - S_g \quad ..(5)$$

3.3 Performance Of Combined cycle:

The power output of combined cycle is given by[8]:

$$\dot{W}_{net,gs} = \frac{\dot{W}_{gt} - \dot{W}_{com} + \dot{W}_{st} - \dot{W}_{pmp}}{\eta_{EG}} \quad ..(6)$$

The heat input to the cycle is:

$$\dot{Q}_{in} = (\dot{m}_{f,cc} + \dot{m}_{f,ab})\left(LHV + h_{f@T_f} - h_{f@15℃}\right) \quad ..(7)$$

The thermal efficiency of the combined cycle is given by[8]

$$\eta_{th,gs} = \frac{\dot{W}_{gt} - \dot{W}_{com} + \dot{W}_{st} - \dot{W}_{pmp}}{(\dot{m}_{f,cc} + \dot{m}_{f,ab})\left(LHV + h_{f@T_f} - h_{f@15℃}\right)} \quad ..(8)$$

The lost exergy from the combined cycle due to the exhaust gases discharged to the environment is given by[8]

$$EX_{exh,gs} = (\dot{m}_{p,4} + \dot{m}_{f,ab})\psi_{p,5} \quad ...(9)$$

The total exergy destructed is given by[8]:

$$\sum I = I_{com} + I_{gt} + I_{cc} + I_{st} + I_{con} + I_{FWH} + I_{pmp} + I_{HRB} \quad (11)$$

The second law efficiency of the gas turbine cycle is based on second law of thermodynamic. It is gives as[8]:

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$$\eta_{II,gs} = \frac{\dot{W}_{net}}{EX_{f,gs}} \qquad (12)$$

3.4 Performance 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[9]. This is:

$$\mathrm{PR} = \frac{\dot{m}_D}{\dot{m}_s} \qquad ..(13)$$

The lost exergy from the condenser due to the exit hot brine and distilled water is given by[9]:

$$EX_{lost,MSF} = \dot{m}_D.\psi_D + \dot{m}_B.\psi_B \qquad ..(14)$$

The total exergy destructed is given by[9]:

$$\sum I = I_{com} + I_{gt} + I_{cc} + I_{st} + I_{MSF} + I_{FWH} + I_{pmp} + I_{HRB} \qquad (15)$$

4. Results and Discussion from Parametric Study:

Two configurations will be adopted. The first one is CC cycle only and the second one is CC coupled with MSF unit.The assumptions and reference data underlying the present study have shown in Table (1). The general input parameters adopted have given in Table (2). 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 (1) Assumptions and Reference Data for the Parametric Study.

ISO Conditions 15°C, 101.325 kPa			
1. Assumptions ofGas Turbine [G]		***2. Assumptions of Combined Cycle [GS]***	
Turbine Cooling Air Fraction	0.125	Reference temperature	25°C
$\eta_{m,C} = \eta_{m,G} = \eta_{EG.}$	0.95	Steam temperature	570°C
$\eta_{is,C}$	0.87	Steam pressure	70 bar
$\eta_{is,G}$	0.92	Condenser pressure	10 kpa
η_{CC}	0.98	Pinch point of HRSG	15°C
ΔP_c	0.0001	Approach temperature HRSG	10°C

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ΔP_{cc}	0.0001	Pump efficiency	0.9
ΔP_G	0.0001 for GT 0.0002 for CC	————	————
Fuel temperature	15°C	Firing temperature after burner	600°C
————	————	$\eta_{is,GS}$	0.9
3. Assumptions of Multi Stage Flash[MSF]			
Type	Once through		
Condenser pressure	84.53 kpa		
Number of stages	23		
Motive steam temperature	95°C		
Feed water temperature	25°C		
Waste brine temperature	40°C		
Terminal temperature difference of condenser	3°C		
Thermodynamic loss	2°C		
Salinity of feed water	42000 ppm		

Table (4.2) Input Data Range and Typical Values for General Parametric Study

Data for CC [GT-ST]	
Fuel Type	100% CH_4
ISO Air Flow rate	400 kg/s
Firing Temperatures	950°C and 1050°C
Ambient Temperature	10 → 55°C
Compressor Pressure Ratio	7 → 40

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5. Energy and Exergy Analysis.

5.1 Energy Analysis “Effect of Operation Condition”

Figure **(4)** shows the effect of pressure ratio on the power net at fixed ambient and firing temperatures. The figure shows that the power output for the CC plant will increase for all over the pressure ratio selected. The main cause of that in the sophisticated utilization of exhaust gases heat discharged by GT into a useful power via steam cycle. It is clear that the power of steam cycle is almost constant for all over the pressure ratio selected. That is because of fixing the steam cycle parameters produced and fixing air mass flow rate. In addition, optimum pressure ratio was observing for GT that was using for CC purpose. When using the MSF unit coupled with CC, the power produced will decrease from the steam cycle only. This power drop will been used to produce the useful distilled water in MSF unit. The power lost was attributing to the shifting in condenser pressure so that the steam will expand to a higher pressure than that in the CC. This is very important to the MSF unit to produce the required top brine temperature. It is found the power lost in MSF unite is about **(8to10) %**. In addition, thefigure shows the variation of power output with pressure ratio at higher firing temperature of 1050℃. Increase firing temperature from 950°C to 1050°C will shift the curves of output power to higher values. For the CC and CC coupled with MSF, the increase in firing temperature will increase the exhaust temperature discharged into the HRSG. This will minimize the fuel required for the after burner. It was founding that decreasing the amount of fuel supplied to the after burner will cause a slight increase in steam produced by the HRSG. So, no great difference in steam cycle power output due to the increase in firing temperature.

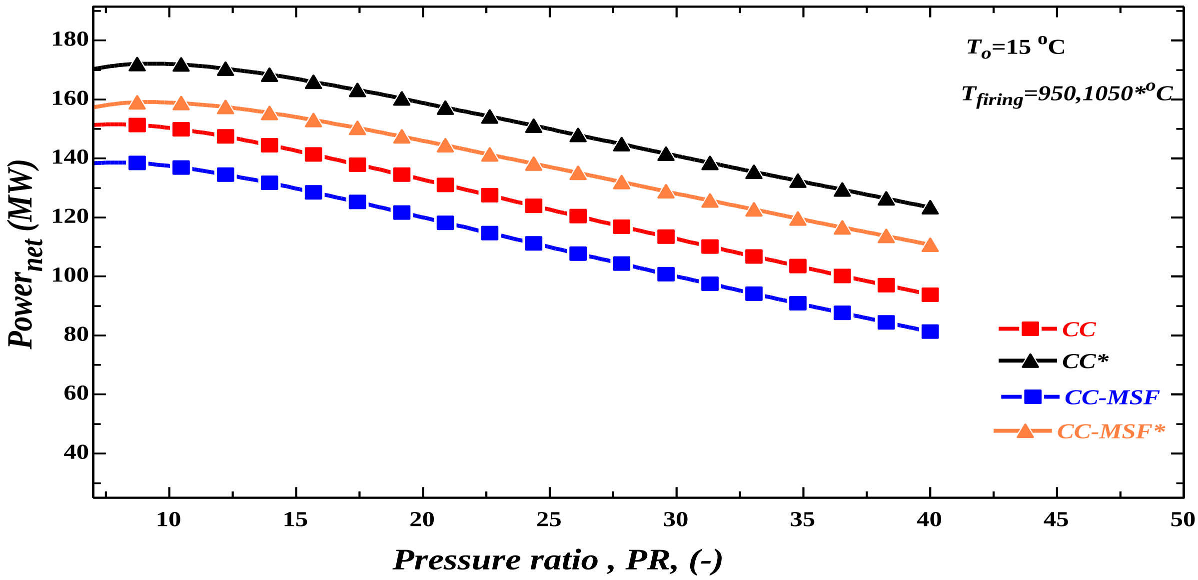

Fig 4 Variation of Power net of plant with Pressure Ratio at Constant Firing Temperature.

Figure (**5**) shows the effect of pressure ratio on the thermal efficiency .The figure explains that the thermal efficiency of the plant follows the same trend as that of power output, i.e there is an optimum pressure ratio that causes the efficiency will be maximums. However, the optimum pressure ratio that causes the efficiency to be top is higher than the optimum pressure ratio that causes the net power to be the maximum. The reason of this behavior was relating to the fuel mass flow rate. When pressure ratio is lower, then the mass flow rate of fuel will be higher, since the air temperature leaving the compressor and entering the combustion chamber, will be at a lower temperature. By increasing the pressure ratio, then the air temperature will increase and this will certainly minimize the fuel consumption. For the case of CC, the optimum pressure ratio is located at lower pressure ratios. In this case, the fuel consumption in the after burner plays important role in this behavior.For the case of CC-MSF, the thermal efficiency has the same trend as that of (CC) since

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the MSF unite will not affect the operation of the after burner. By decreasing the power developed of the steam turbine, then the thermal efficiency will certainly decrease.

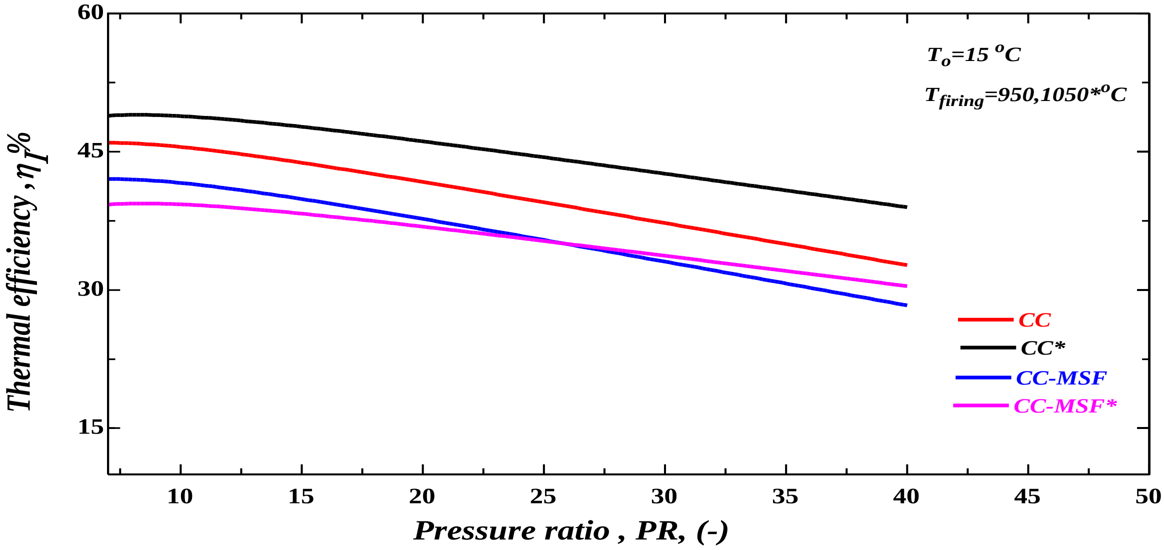

Fig 5 Variation of Thermal Efficiency of Plant with Pressure Ratio at Constant Firing Temperatures.

Figure **(6)** shows the variation of distilled water production with pressure ratio. The figure shows a little reduction in the mass flow rate of distilled water production. This little decrease is belonging to the reduction in fuel mass flow consumed in the combustion chamber. Also, figure shows the variation of distilled water mass flow rate with pressure rate at lifted firing temperature of 1050℃. By comparing the results of firing temperature(950 and1050℃)respectively, it is clear that there is a little increase in distilled water production, due to increase the firing temperature. That is mainly due to the little improvement in steam mass flow rate resulted from improving the quality of exhaust gas discharged by the gas turbine.

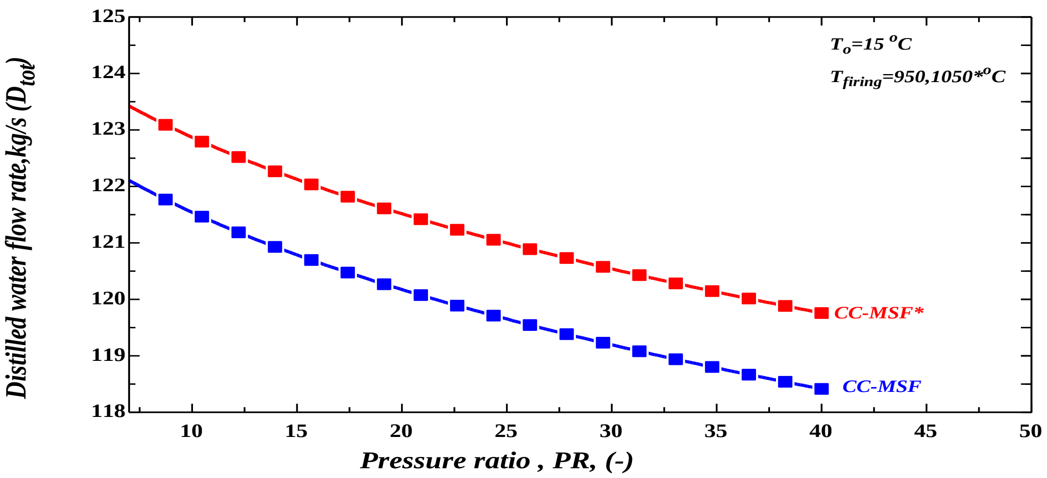

Fig6 Effect of Pressure Ratio on Distilled water flow rate, kg/s (D_tot) atConstant Firing Temperatures.

Figure **(7)** shows that the amount of distilled water produced per kg of steam discharged from the steam turbine (DPS) are independent of pressure ratio. That is mainly because of DPS is the function of MSF system specification not GT cycle specification. Both of steam mass flow rate and distilled water mass flow are almost constant and not affected by the pressure ratio. This is because that fuel mass flow is very

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low relative to the air mass flow rate and the later has the biggest effect on the steam mass flow rate produced by HRSG

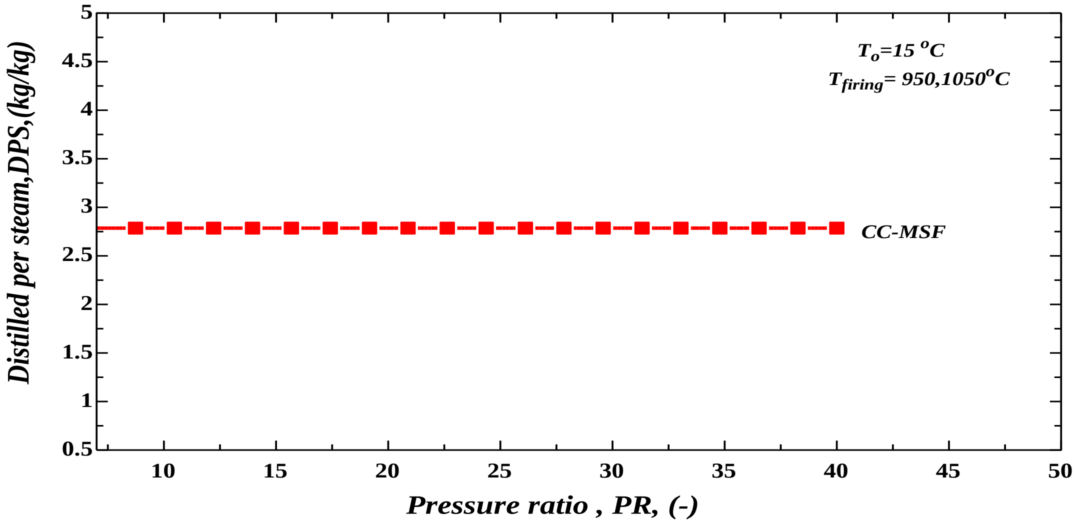

Fig 7 Effect of Compressor Pressure Ratio on the Distilled Per Steam, DPS, (kg/kg) at Constant Firing Temperatures

Figure **(8)** shows the variation of optimum pressure ratio with firing temperature. For the GT cycle, the optimum pressure ratio required for thermal efficiency to be maximums is higher than that required for maximum power output. That is mean that there are two choices for best design of GT engine, either choosing thermal efficiency to be maximum or output power to be maximum. Of course, the procedure is to choose pressure ratio between these two lines and making compromising. When gas turbine engine is utilized in CC mode (no difference if MSF is found or not) then the best pressure ratio design rang is lowered and best pressure ratio required for power output to be max is somewhat higher than that required for the thermal efficiency to be the best. This behavior was attributing to the fuel mass flow rate supplied to the after burner. It is worth noting that coupling the HRSG on the gas turbine, then the power of gas turbine was lowering by about 5**% to 7%** due to increase the backpressure or pressure drop across the HRSG. Fortunately, the power produced by steam cycle will be greater than this loss in power and the result is the increase in pressure ratio required for power output to be maximums.

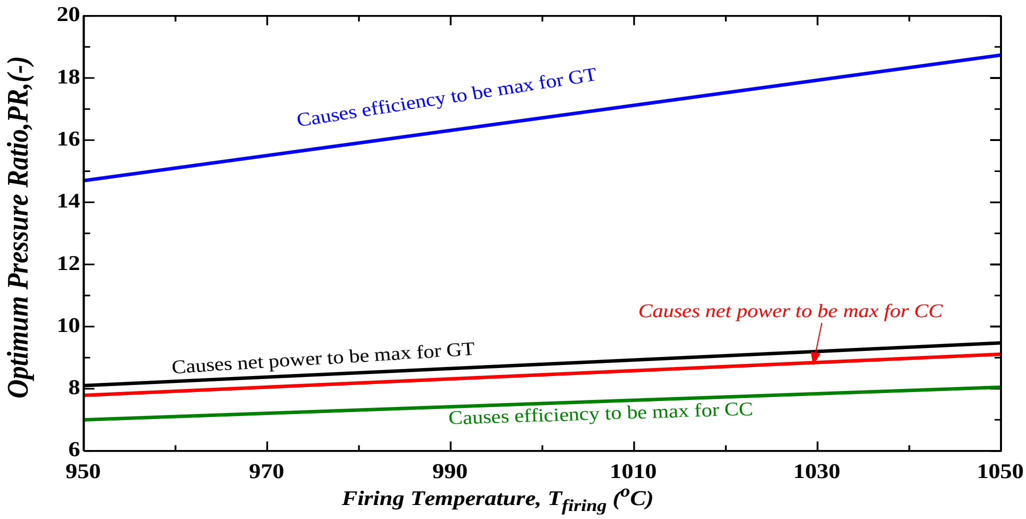

Fig8 Optimum Pressure Ratio for Thermal Efficiency and output power Versus Firing Temperature

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Figure **(9)** shows the relation between ambient temperature and power developed for the combined cycle. The figure reveals a drop in power at higher ambient temperature. The relation of power generation and ambient temperature is linear and this indicates that there is main parameter that causes this behavior. Actually, this factor is the air mass flow rate. The increase in ambient temperature will decrease the air density and since the gas turbine engine was operating at constant volume flow rate, then this will certainly decrease the air mass flow rate. By decreasing air mass flow rate, the steam mass flow rate will also decrease and thereby the power developed by steam cycle will decrease. As a result, the power developed by CC will decrease. By attaching MSF unit, the decrease in power is constant for all the range of ambient temperatures selected. That is because the assumption of the steam cycle is independent of ambient temperature.

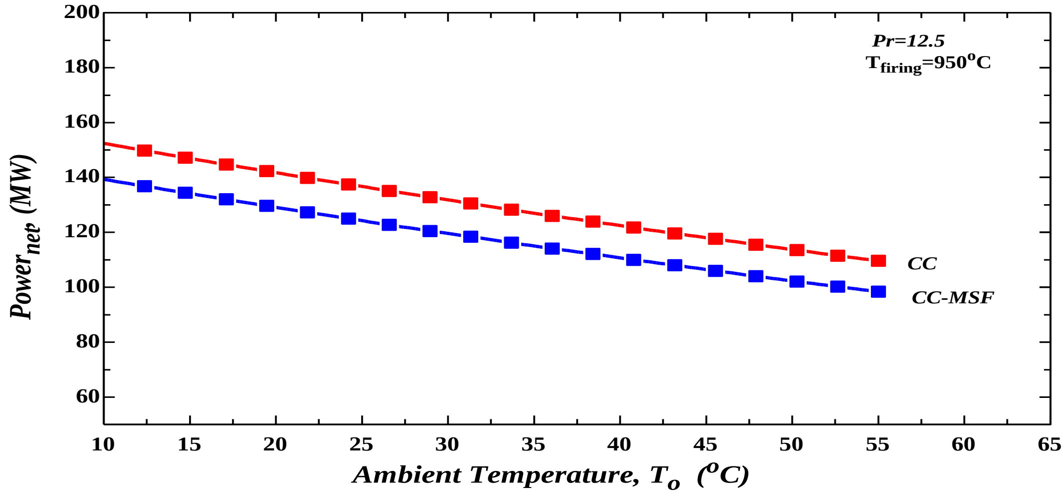

Fig 9 Power of Plant Versus Compressor Inlet Air Temperature at Constant Pressure Ratios and Firing Temperatures.

Figure**(10)** explains the variation of thermal efficiency with ambient temperature. The figure shows that thermal efficiency was also affecting negatively in the hot summer. However, the curves of thermal efficiency for the two proposed cycles have a lower gradient than that for the power output. Although, thermal efficiency is directly proportional to the power output, but it is inversely proportional to the fuels mass flow rate. It was founding that increasing ambient temperature will decrease the fuel mass flow rate and this is the reason why thermal efficiency has a lower gradient than power output.

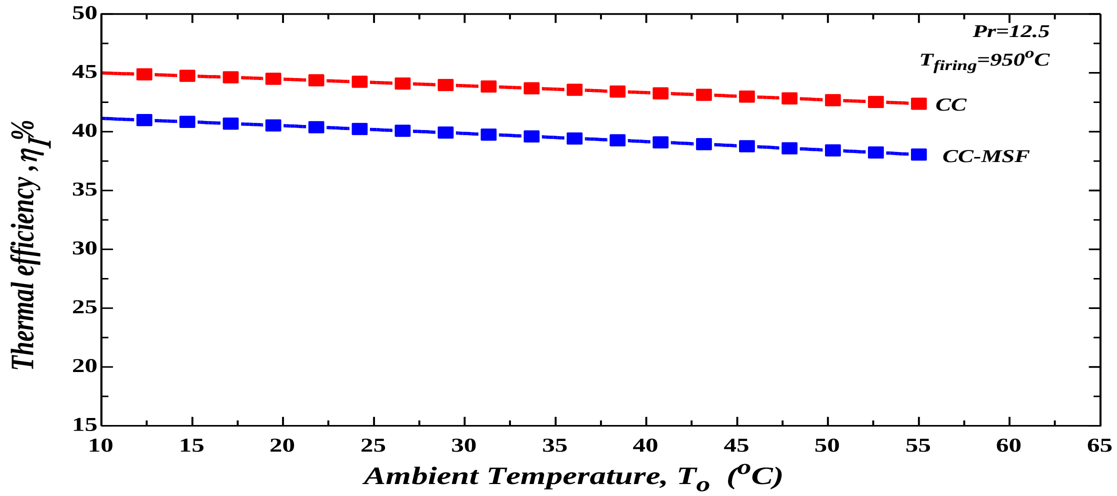

Fig10 Thermal Efficiency of Plant Versus Compressor Inlet Air Temperature at Constant Pressure Ratios and Firing Temperatures.

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Figure **(11)**shows the effect of ambient temperature on the production of distilled water mass flow rate. The figure shows a negative effect of ambient temperature on the mass flow rate of distilled water. That is mainly due to the reduction in steam mass flow rate produced by the HRSG. The decrease in steam mass flow rate is due to lower air and fuel mass flow rate that are joined together to form combustion gases.

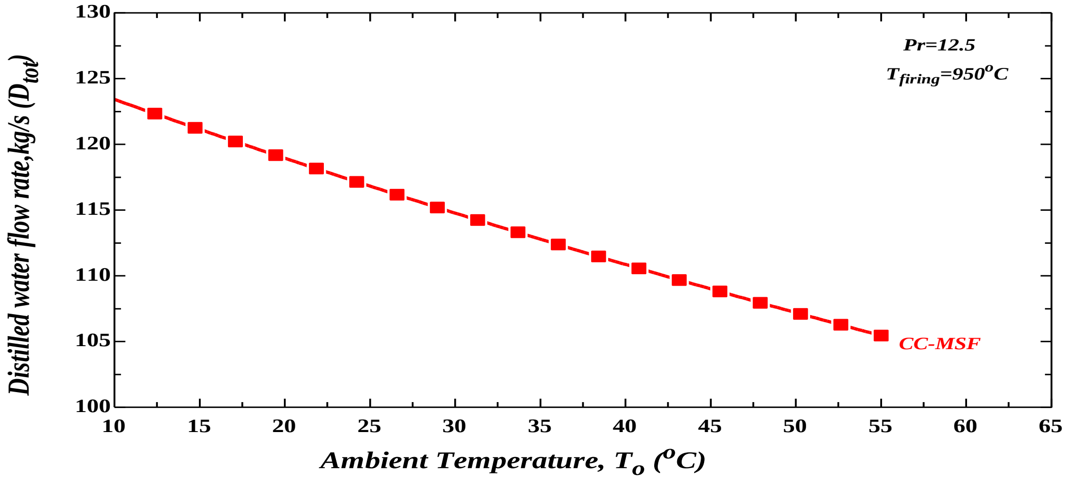

Fig 11 Effect of Ambient Temperature on Distilled water flow rate, kg/s (D_tot) at Constant Pressure Ratios and Firing Temperature.

5.2Exergy Analysis "Effect of Operation Condition"

Figure **(12)** shows the variation of second law efficiency with pressure ratio. The figure explains that the variation of this definition follows the same style as that of first law efficiency but with lower values. Generally, the main cause is the difference between the energy and exergy content of fuel. The fuel has a higher exergy than the energy. This will make the second law efficiency is lower than the first law efficiency.Also, the variation of second law efficiency with pressure ratio at higher firing temperature 1050℃. The figure follows the same style like that at lower firing temperature of 950℃. However, with higher values of second law efficiency, this improvement was attributing to the increase in recovered exergy from combined cycle. The figure also explains that there is optimum pressure ratio at which second law efficiency could be optimum. The CC still has the top value of second law efficiency followed by CC-MSF.

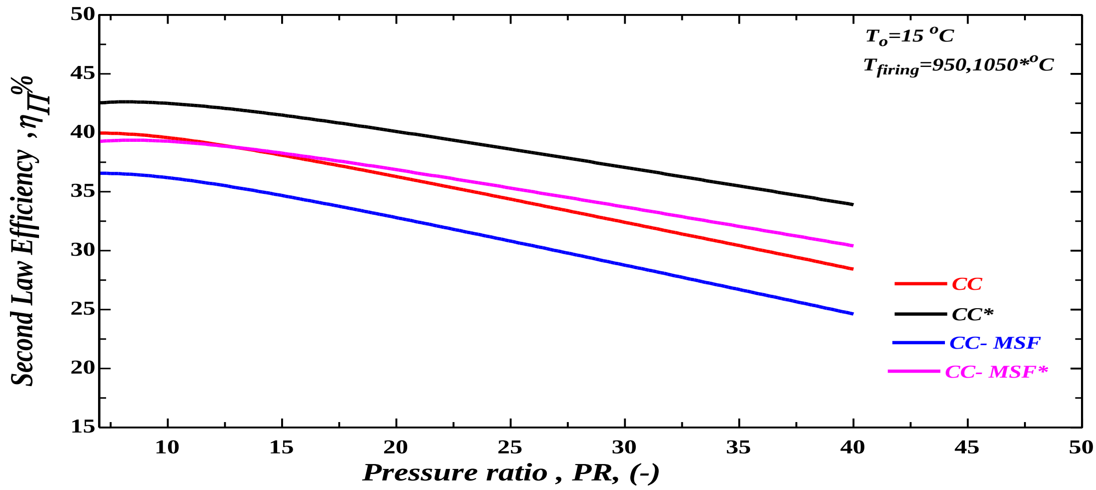

Fig12 Second Law Efficiency Versus Pressure Ratio.

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Figure **(13)** shows the variation of total exergy destructed in the cycle components with pressure ratio. First, it is clear that the pressure ratio causes a decrease in the total exergy destructed for the two systems. Actually, the exergy destructed in combined components (compressor, combustion chamber, turbine and HRSG) are the main components that affect by pressure ratio. Increasing pressure ratio will certainly increase the air temperature input to the combustion chamber. This will minimize the temperature difference and as a result, the exergy destructed will decrease. Total exergy destructed for CC and CC-MSF has a higher value since the numbers of equipment have increased. In addition, Figure shows the effect of pressure ratio on the total exergy destructed in the cycle components at elevated firing temperature of 1050℃. The figure reveals that total exergy destructed will increase as firing temperature are getting rise. This quite understood due to the increase of temperature difference between inlet and exit for the combustion chamber. A temperature difference is higher than the lost work will be more. Beside, increasing firing temperature will somewhat increases the steam mass flow rate generated by the HRSG and this will increase the lost work in this component. Keep in mind that increase the firing temperature will increase the fuel exergy input to the two cycles.

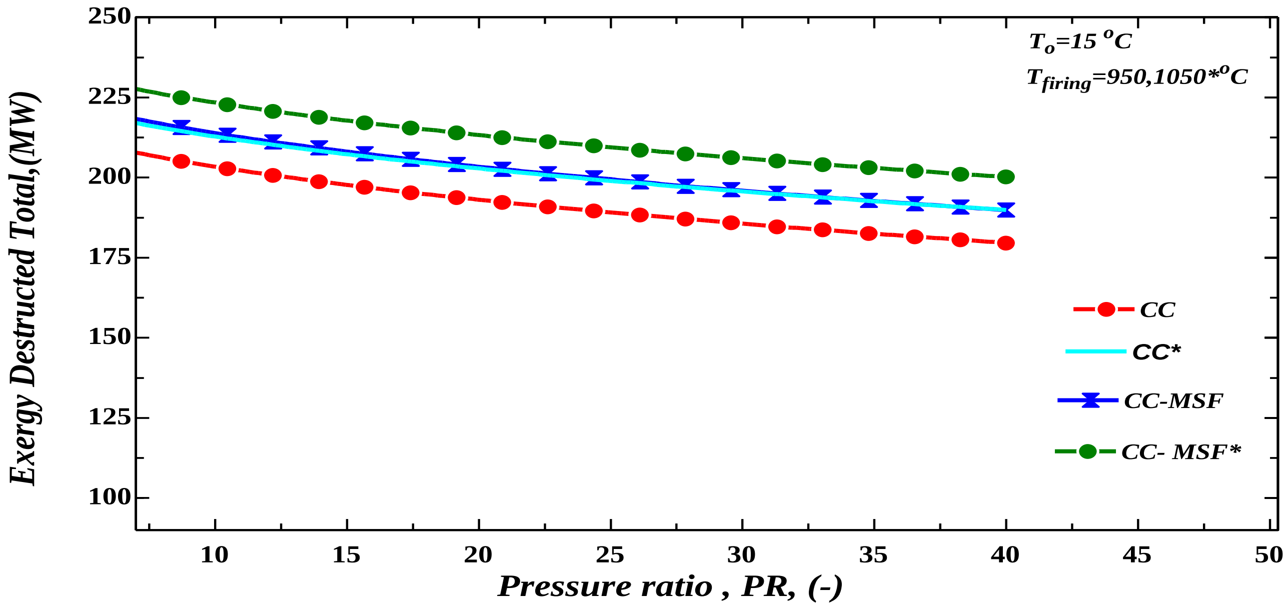

Fig 13 Variation of Exergy Destructed Total, (MW) of Plant with Pressure Ratio at Constant Firing Temperatures.

Figure (14) shows the relation between pressure ratio and lost exergy for the two cycles CC and CC-MSF at 950 and 1050℃respectively. It is clear that the lost exergy was lowering due to good utilizing of the useful heat via HRSG and producing extra work by steam turbine. Best utilizing has done in the CC only since the condenser pressure was lowering than that of CC-MSF. In these two cycles, the lost exergy is nearly independent of pressure ratio due to using after burner that fixes the exhaust temperature.

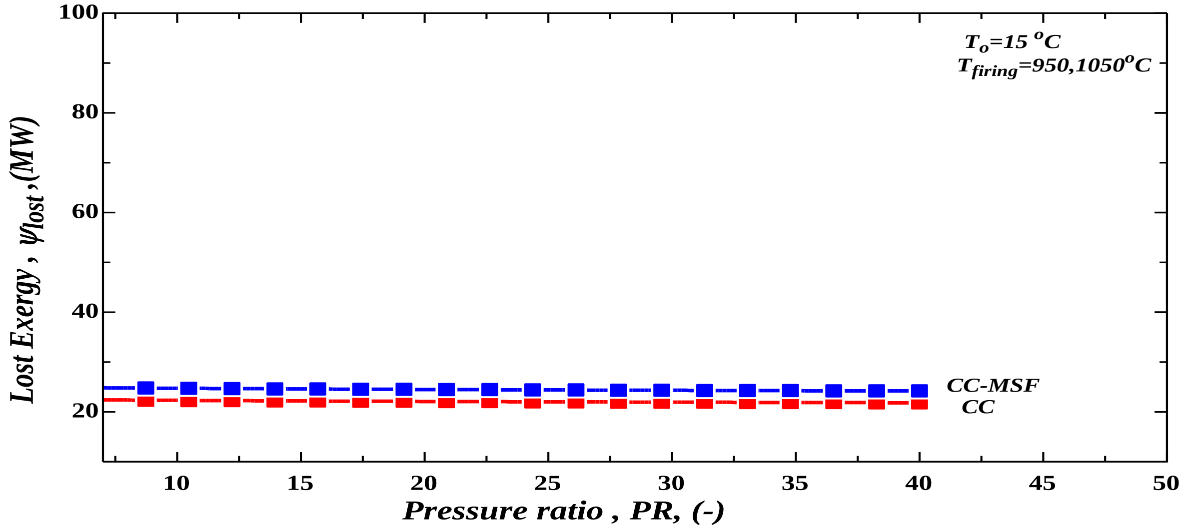

Fig 14 Variation of Lost Exergy, Ψ_lost , (MW) of Plant with Pressure Ratio at Constant Firing Temperatures.

Figure**(15)**shows the variation of second law efficiency (exergy efficiency) with ambient temperature. The figure has the same trend as that for first law efficiency. The lower values of second law efficiency is attributed to the higher exergy contend in the fuel. The decrease in exergy output as useful work is responsible for this behavior.

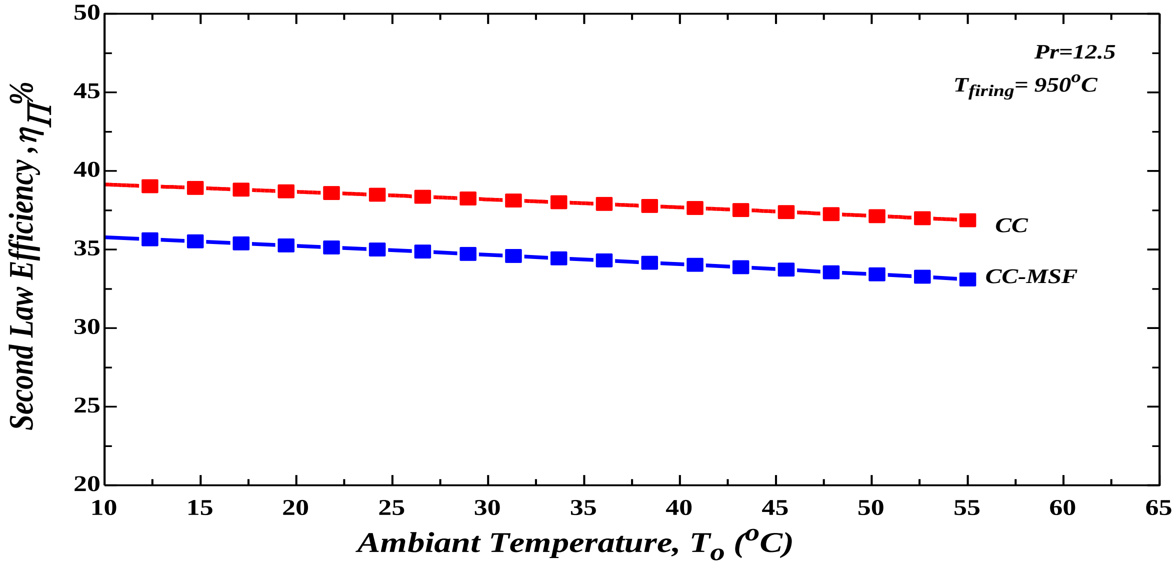

Fig15 Second Law Efficiency Versus Ambient Temperature.

Figure**(16)** shows the effect of ambient temperature on the total exergy destructed in the cycle components. The figure reveals a drop in total exergy destructed in hot summer season. Unfortunately, that is not meaning an improvement in the cycle duty from the second law of thermodynamic point of view. Since this accompanied with a severe drop in the exergy recovered as work and lowering exergy input to the cycle due to decreasing the fuel mass flow rate.

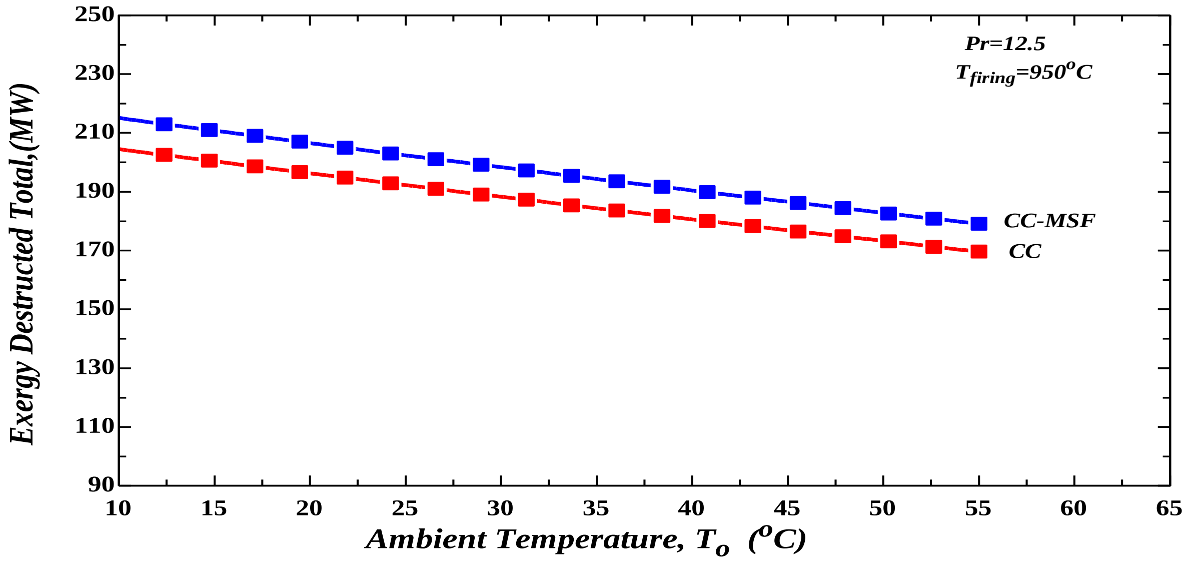

Fig 16 Variation of Exergy Destructed Total, (MW) of Plant with Ambient Temperature at Constant Firing Temperatures.

Figure **(17)** shows the variation of lost exergy with ambient temperature. The figure explains that increasing ambient temperature will result in a drop for lost exergy to the environment for the two proposed systems. It is worth nothing that the decrease in lost exergy was not attributing to the exhaust temperature. The exhaust temperature is independent of ambient temperature and depends on pressure ratio and firing temperature. The decrease in air mass flow rate which affecting the mass flow rate of exhaust gases is the main reason of this manner.

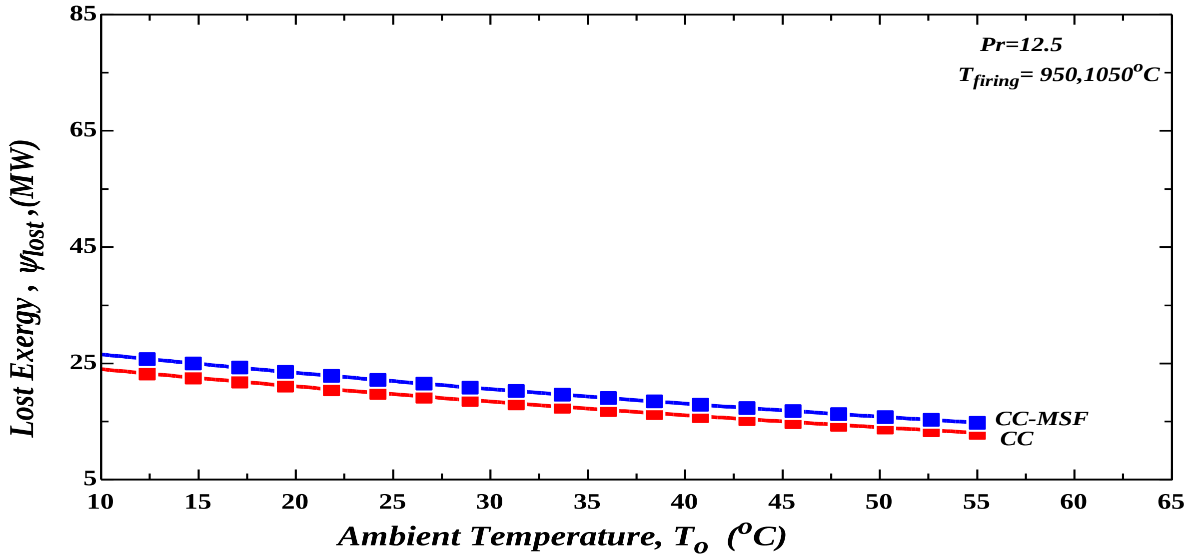

Fig 17 Variation of Lost Exergy, Ψ_lost , (MW) of Plant with Ambient Temperature at Constant Firing Temperatures.

7. Conclusions

This study reveals the effect of the operations conditions on the performance of the combined cycle and multi stage flash unit in terms of energy and exergy .

1- The performance of CC depended on operation conditions of power plant. These conditions are pressure ratio, ambient temperature, firing temperature.

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2- 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.

8. References

[1] Y.A.Cengel, and M.A. Boles, ***"Thermodynamics, An Engineering Approach"***, eighth Edition, McGraw Hill Companies, New York, USA, 2006.
[2] A. Khaliq, and S.C. Kaushik, " Second-law based thermodynamic analysis of Brayton/Rankine combined power cycle with reheat", Applied Energy, Vol. 78, pp. 179-197, 2003.
[3] H.T. EL-Dessouky, and H.M. Ettouney," Fundamentals of Salt Water Desalination" Elsevier Science B.V., 2002
[4] F.F. Hatem, "Exergy Losses Calculation for a 125MW Combined Power Plant", Eng. &Tech. Journal,Vol. 27, No.2, 2009.
[5] A.Y. Fathi, " Thermal Evaluation of Applying Combined Cycle Mode to Beijee Gas Turbine Generators", Al-Rafidain Engineering, Vol. 20, No. 3, 2011
[6] B.G.JABBOURY, and M.A.DARWISH," THE EFFECT OF THE OPERATING PARAMETERS OF HEAT RECOVERY STEAM GENERATORS ON COMBINED CYCLE/SEA-WATER DESALINATION PLANT PERFORMANCE", Heat Recover Systems & CHP, Vol. 10, No. 3, pp. 255-267, 1990.
[7] A. K. Tiwari, M. M. Hasan, and M. Islam," EFFECT OF AMBIENT TEMPERATURE ON THE PERFORMANCE OF A COMBINED CYCLE POWER PLANT", Transactions of the Canadian Society for Mechanical Engineering , Vol. 37, No. 4, 2013.
[8]A. Almutairi, P. Pilidis, and N. Al-Mutawa, " Energetic and Exergetic Analysis of Combined Cycle Power Plant: Part-1 Operation and Performance", Energies, Vol. 8 ,pp. 14118–14135, 2015
[9] M.A. Al-Weshahi, A. Anderson, and G. Tian, "Exergy efficiency enhancement of MSF desalination by heat recovery from hot distillate water stages",Applied Thermal Engineering, Vol. 53, pp. 226-233, 2013.

Nomenclature

English symbols **Subscripts**

EX	Exergy term.	ac	Air compressor.
h	Specific enthalpy.	cc	Combustion chamber.
I	Irreversibility.	gen	Generator.
m	Mass flow rate.	gt	Gas turbine.
P	Pressure.	I	First law.
Q	Heat power.	II	Second law.
s	Specific entropy.	in	Input.
$Sgen$	Entropy generation.	i,e	Inlet and exit.
T	Temperature.	$lost$	Lost.
W	Work power.	m	Mechanical.
PR	The Performance Ratio	net HRB pmp gs	Net value. Heat Recovery Boiler Pump Gas- Steam turbine
Greek symbols			
η	Efficiency.	o	Denotes to the dead state.

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Δ	Difference .	out	Out.
ψ	Flow physical exergy.	t	Turbine.
Ψw	Exergy due to work.	tot	Total.

Abbreviations

EES	Engineering equation solver.
CC	Combined Cycle.
GT	; Turbine
HRSG	Heat Recovery Steam Generator.
ISO	International standards organization.
HFO	Heavy fuel oil.
LFO	Light fuel oil.
HHV	Higher heating value.
LHV	Lower heating value.
MSF	Multi Stage Flash
ST	Steam Turbine

Superscripts

ch	Chemical