Converting Zubair oil field permanent power generation from single cycle into combined-cycle with plant exergy analysis

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Abstract: The combined-cycle power plant is the effective way to increase power and thermal efficiency which can rise thermal efficiency up to 50% corresponding to low heat value of the fuel (LHV). This study focused on converting Zubair Oil Field Gas turbine (ALSTOM GT13E2) from single cycle into a combined cycle power plant. Models suggestion design have been presented with back-pressure steam turbine by Cycle-Tempo (5.1) thermodynamics simulation program, which presents value diagram and exergy flow diagram both are useful to understanding the thermodynamic performance of complicated systems. The result of exergy analysis explicate that huge amount of exergy loss firstly occurs in gas turbine combustor (about 60% due to high irreversibility), and heat recovery steam generation (HRSG) is the second major exergy loss in combined cycle power plant (about 30%) recently apparatus concepts less than 10% of exergy (like compressors and expanders). Also, in this study explained the effect of changing the ambient temperature on the performance of power plant by using four temperature parameters on the four simulation models at constant compression ratio. The results were the increase in gas turbine temperature inlet led to decrease the net power output of the CCPPs. Thermal and exergy efficiencies increase when ambient temperature increase due to growth turbine inlet temperature but this temperature restricted by creep rupture and thermal fatigue for combustor parts and turbine blades metal. Maximum thermal and exergy efficiencies can be reached to 55.74, 53.28% respectively at worst ambient temperature. The calculation results in this study were compared with two previous studies, with considerable differences in input parameters of the gas turbine and steam turbine also, the types of the gas turbine models. finally, the results showed acceptable agreement.

Keywords: Alstom GT13E2 gas turbine, Value diagrams, Cycle-Tempo (5.1).

1. Introduction

The efficiency improvement is an essential enforcement in the Energy Engineering Field the combined cycle power plants have the maximum thermal efficiency compared with single cycles like (gas turbine or steam turbine engines) [1,2].
In the combined cycle power plants Rankine cycle represented by steam turbine which known as bottoming cycle and Bryton cycle represented by gas turbine called topping cycle [3].

Exergy is known as ultimate beneficial work that would be obtained from the system at stable state for a specify environmental conditions. Exergy cannot be preserved like to the energy but for a specify system exergy could be destroyed. Exergy calculation indicates the losses location in a system and these losses amount. The thermal efficiency of a CCPP will ordinarily be more than exergy efficiency for the same power plant [1].

Ameri et al [1] evaluated irreversibility for plant components of Neka Iran CCPP used the exergy analysis, the results shown that “the combustion chamber, gas turbine, duct burner and heat recovery steam generator” (HRSG) are the highest sources of irreversibility about 83% of the total exergy losses. Polyvakis et al [3] studied the optimizing of a CCPP described and compared four diverse gas turbine cycles. The results showed that “the reheated gas turbine” is the most eligible generally, due to highest exhaust temperature gases which is led to higher thermal efficiency of the steam turbine (bottoming cycle). Kaviri et al [2] studied “comprehensive thermodynamic modeling of a dual pressure combined cycle power plant”. The results showed that any change in “the gas turbine temperature, compressor pressure ratio, and pinch point temperature” led to a drastic change in objective functions. Woudstra et al [4] studied exergy analysis for three CCPPs system designs modeled using computer program Cycle-Tempo (5.0). The results showed that “more than 35% of the fuel exergy entering the CCPP is lost due to combustion and friction in the gas turbine, the exergy losses in the HRSG, both steam cycle and stack have only 11% for a single pressure system and are decrease to 9% for a triple pressure matter”. Ibrahim and Rahman [5] studied “thermodynamic analysis” of “the triple-pressure reheat combined cycle gas turbine with a duct burner” for an existing actual CCPP, simulated by “THERMOFLEX software”. The results showed that the total power output and overall efficiency of a CCPP drop with rise the ambient temperature. The totals power of a CCPP decrease with growth in “the compression rate”, while the whole efficiency of the CCPP rises with an increasing “the compression ratio to 2!”, then the overall efficiency will decrease. Therefore, the turbine inlet temperature has potent impact on the total performance of the CCPPs. Najjar [6] studied twelve research investigations for ten years ago, with twelve different gas turbines which have briefly reviewed, efficient use of energy produced from these power plants, they necessitate essential studies in an increment to implementations of combined systems in the industry. Author explained advantages of combined cycle power plants, how they can “boost power output and thermal efficiency as high as 60%”. Song et al [7], defined particulars of exergy based on performance features of a heavy-duty gas turbine, 150MW-class GE 7F model, their study emphasized that combustion chambers responsible on range ratio from (28.3% to 41.6%) at part-load conditions of exergy destruction at full-load in the gas turbine despite its utility in the performance of the combined cycle plant in part load operations. Hatem [8], offered that major exergy destruction occurred in the combustion chambers of the gas turbine, which may be reached 33% of the consumed fuel exergy while other apparatuses consumed fuel exergy between (1% - 5%). Jamel et al [9], dealt with “the simulation of a 200MW gas-fuelled conventional steam power plant located in Basra, Iraq”, using flow sheet computer program “Cycle-Tempo”. An
itemized analysis of exergy destruction was made. Also, they concluded that proportional exergy destruction in the combustion and evaporator were elevated compared with other components of the power plant.

This paper aimed to convert Zubair oil field gas turbine GT13E2 from single cycle into simplest design form of combined-cycle power plant, calculate exergy efficiency and exergy loss for most important apparatus of combined cycle and specify the exergy loss intensity. Using Cycle-Tempo thermodynamics simulation program to explain how different ambient temperature affects the power plant performance. The models have been compared with two previous studies with acceptable agreement results.

2. Zubair oil field gas turbine Power Plant

It is one of the modern project in Iraq which located in the west southern of Basra city, till now project progressing for installation with following specifications:

Two dual fuel gas turbine generation units, composed of:
1x 176 MW ISO gas turbine Alstom GT13E2 dual fuel (raw gas/diesel oil);

Two single fuel gas turbine generation units, composed of:
1 x 176 MW ISO gas turbine Alstom GT12E2 single fuel (raw gas);

Evaporative cooling system for each gas turbine (GT), Balance of plant systems (600 MW site condition), sized for six gas turbine-generator units.

2.1 System Specification

The thermal block consists of:

21-stage, subsonic axial compressor with one row of variable inlet guide vanes (VIGV), single annular combustor with 48 advance environment burners (AEV), Five stage air-cooled turbine, Turbine diffuser (or diffuser one) to recover remaining hot gas kinetic energy, Equalizing section (diffuser two) [11].

![Fig. (1): front view of Zubair Oil Field Power Plant](image_url)
2.2. Operational design data (boundary conditions)

Table (1): GT13E2 ambient conditions [12]

| Ambient conditions: |   |
|---------------------|---|
| Ambient ISO conditions | Tmp=+15°C  
|                       | Pamb =101.3kPa 
|                       | Rel. Humidity=60% |
| Maximum ambient temperature | + 55°C  |
| Minimum ambient temperature | -5°C  |
| Ambient pressure | 100.7 kPa (1007 mbar) |
| Relative ambient humidity | 10% to 100% |
| Wind maximum speed | 45 m/s |

2.3. GT Thermal block (mechanical data) manufacter: ALSTOM Mannheim (Germany).

Table (2): Thermal black performance [12].

| Performance: |   |
|--------------|---|
| Normal frequency | 50Hz |
| GT gross electrical power (generator terminals) | 125.3MW at 55°C ambient temperature |
| GT gross electrical efficiency (generator terminals) | 32% |
| Gross heat rate | 11114kJ/kWh |
| Compressor Rate | 14:1 |
| Exhaust gas flow | 456.6kg/s |
| Exhaust gas temperature | 550 °C |

Table (3): GT13E2 specifications [12].

| GT specifications: |   |
|---------------------|---|
| Simple cycle model | 4x GT13E2 |
| Model series | GT13E2 (2005) |
| Rotation speed | 300rpm |
| Rotation direction | Clockwise in flow direction |
| GT weight | 357 tones |
| GT dimensions (length/height/width) | 11.5/5.4/5.5 m |

3. Theoretical analysis
3.1. Work and thermal efficiency of thermodynamics cycle

Thermodynamic is the science of numerous processes concerned with one form of energy being change into another shape of energy. Generally, the heat is supplied at one end and taken out from another end of the cycle. The difference between the heat supplied and heat taken out from the cycle expresses the work in equation (1)

\[ W = Q_{\text{in}} - |Q_{\text{out}}| \] (1)

Thermal efficiency is expressed by quotient of work and supplied heat.

\[ \eta_{\text{th}} = \frac{W}{Q_{\text{in}}} = \frac{Q_{\text{in}} - |Q_{\text{out}}|}{Q_{\text{in}}} = 1 - \frac{Q_{\text{out}}}{Q_{\text{in}}} \] (2)

The main target is to get higher thermal efficiency from the cycle [13].

3.2. Combined thermodynamic cycles

Major idea of combination different thermodynamics cycles is employing heat potential in a fluid which is leaving the topping cycle (gas turbine) which has higher temperature level, to the bottoming cycle (lower temperature level) as explained in figure (1). This fact using with aids in coupling “a gas turbine as a topping cycle and steam turbine as bottoming cycle”. Gas turbine is identical machine that had been developed from aircraft engines, steam turbine has been in use for decades as well.

Gas turbine associated with a generator produces electrical energy and hot exhaust gases directed by heat recovery steam generator to convert water into steam. This steam can be used as process steam and/ or expands in a steam turbine which generates electrical energy too, co-generation is an application can be used from district heating cycle. [13]

Gas turbine efficiency is defined as

\[ \eta_{GT} = \frac{P_{GT}}{Q_{GT}} \] (3)

And efficiency of a steam turbine (with or without additional firing burner) is:

\[ \eta_{ST} = \frac{P_{ST}}{Q_{ST} + Q_{SF}} \] (5)

Where

\[ Q_{EG} \approx Q_{GT} \cdot (1 - \eta_{GT}) \] (6)

By merging equation (4), (5) and (6) we get gross thermal efficiency of a CCPP:

\[ \eta_{CCPP} = \frac{P_{GT} + P_{ST}}{Q_{GT} + Q_{ST} + Q_{SF} \text{ (if exist)}} \] (7)

Reducing total power output by plant auxiliary consumption yields net thermal efficiency:
\[ \eta_{CCPP} = \frac{\eta_{GT} \cdot \eta_{ST} \cdot \eta_{aux}}{\eta_{GT} + \eta_{ST} (i \text{ exist})} \]  

(8)

![T-s diagram of combined cycle power plant][13].

3.3. Exergy analysis

Exergy are collected from two essential parts, physical exergy and chemical exergy. For this study the exergy of kinetic and potential phase is ignored. The chemical exergy is related with the chemical composition of fuel in combustion chambers and additional fuel burner in the HRSG (as optional). Exergy balance formula could be written from the first and second lows of thermodynamics:

\[ \dot{E}_Q + \sum \dot{m}_i \dot{e}_i = \sum \dot{m}_e \dot{e}_e + \dot{E}_w + \dot{\dot{I}} \]  

(9)

\( \dot{E}_Q \) is the total specific exergy and \( \dot{\dot{I}} \) is the exergy loss rate

\[ \dot{E}_Q = (1 - \frac{T_0}{T}) Q_i \]  

(10)

\[ \dot{E}_w = W \]  

(11)

\[ e_{ph} = (h - h_0) - T_0 (S - S_0) \]  

(12)

“where \( T \) is the absolute temperature” in (K), and \( (i) \) and \( (0) \) refer to ambient conditions.

“The mixture chemical exergy” is define as:

\[ e^{\text{mix}}_{\text{ch}} = \sum_{i=1}^{n} X_i e^{\text{ch}}_i + R T_0 \sum_{i=1}^{n} X_i \ln X_i + G^F \]  

(13)

The last term, \( G^F \), free Gibbs energy is trivial at lower pressure for a gas mixture.

To evaluate the exergy of fuel, the above formula difficult be used. The ratio of fuel chemical exergy represented by this equation:
\[ \xi = \frac{e_x}{LHV_f} \]  

(14)

Most of the gaseous fuels have ratio of chemical exergy closed to one.

\[ \xi_{CH_4} = 1.06 \]

\[ \xi_{H_2} = 0.985 \]  

(15)

For gaseous fuel with “C x H y” ; the following experimental equation is used to calculate \( \xi \):

\[ \xi = 1.033 + 0.0169 \frac{y}{x} - 0.0698 \frac{x}{y} \]  

(16)

For combined-cycle power plant thermal and exergy efficiencies are calculated from these equations:

\[ \eta_{th} = \frac{\sum n \dot{W}_n}{\dot{q}_f} \]  

(17)

\[ \eta_{ex} = \frac{\sum n \dot{W}_n}{\dot{E}_f} \]  

(18)

where \( \dot{q}_f \) : energy and \( \dot{E}_f \) : exergy of the natural gas [1].

3.4. The exergy loss in the steam cycle [1]

The exergy loss in steam cycle will occur in components like: HRSG, stack, steam turbine and condenser which caused by irreversibility process and exergy losses can obtained from

\[ \dot{I} = \sum_i \dot{m}_i e_i - \sum_e \dot{m}_e e_e \]  

(19)

Specific exergy loss define by following equation:

\[ \xi = \frac{\dot{I}}{\dot{q}_i} \]  

(20)

4. Results and discussion

4.1 Model system description

As shown in figure (3) which consists of four heat exchangers (18, 17, 14, 32), condenser 33 deaerator 12, steam drum 15, sink 30, four pumps (11, 13, 16, 31) and steam turbine operating at ISO condition.

In this power plant natural gas is fed to the gas turbine (ALSTOM GT13E2) via source 1 with (17 bar & 107°C), air via source 2 with (1.013 bar & 15°C). The gas turbine consists of compressor 3 with output (14.18 bar & 375°C), combustor 4 have output parameters (14.08 bar & 1103°C) and turbine 5 has (1.033 bar & 545°C). The flue gases from turbine are passed to stack 6 via surface heat exchangers 18, 17, 14 and 32.

The flue gases are used to heat the water/steam in the water/steam cycle to an inlet temperature for steam turbine 10 of (530°C). The steam expands in the turbine and the heat is transferred via condenser
33 to the district heating system. Through pump 11, deaerator 12 and pump 13 the steam condensed to water is passed back to the waste heat boiler (heat exchangers 18, 17, 14 and drum 15) where steam is again produced.

Water of the district heating system is heated in condenser 33 and in an extra heat exchanger 32 in the gas turbine cycle. The heated district heating system is characterized by heat sink 30.

The electric power is supplied by generators which are driven by the gas turbine and steam turbine produce about (256.905 MW).

![Diagram](image-url)

**Zubair Field ALSTOM GT13E2 STAG unit with back pressure steam turbine**

**Fig. (3) Power plant diagram by Cycle-Tempo program**
Fig. (4) shows the enthalpy-entropy diagram for steam cycle model, maximum enthalpy occurs in superheater point 1 equal to (3482.12 kJ/kg). After steam turbine expansion steam take two directions (to deaerator 12 and condenser 33) which appears as two convergent lines due to enthalpy closed values (2296.56 kJ/kg and 2251.42 kJ/kg respectively).

Figure (5) showed temperature-entropy diagram for steam cycle related to saturated line these diagrams drawing by Cycle-Tempo (5.1). Maximum temperature at point 1 via steam turbine.

![Fig. (4): Enthalpy-entropy (Mollier) diagram](image1)

![Fig. (5): Temperature-entropy diagram](image2)

For both Heat-temperature (Q-T) and Value diagrams maximum heat transfer occurs in condenser 33 (about 120MW), its value in evaporator 17 (about 104MW), then heat exchanger 32 (about 72MW), economizer 14 (about 52MW), superheater 18 (about 49MW) finally heat loss to stack about (100MW) as explained in figure (6 and 7).

Table (4) System efficiencies, power input and output

| No. | Apparatus  | Energy [MW] | Totals [MW] | Exergy [MW] | Totals [MW] |
|-----|------------|-------------|-------------|-------------|-------------|
| Absorbed | 1 Fuel source | 503.313 | 526.871 |
| Power | | 503.313 | 526.871 |
| Delivered | 1 Generator | 174.3 | 174.3 |
| Gross power | 2 Generator | 82.605 | 82.605 |
| Aux. power | 11 Pump | 0.00564 | 0.00564 |
| | | | | | |

9
| No. | Name          | Exergy transmitted from system [kW] | Related exergy loss [%] |
|-----|---------------|------------------------------------|-------------------------|
|     |               | Total                              | Power/Heat | Losses |                   |
| 5   | Gas turbine   | 405508.38                          | 384115.34  | 21393.03 | 4.06               |
| 10  | Steam turbine | 68147.70                           | 84723.75   | 16576.05 | 3.15               |
| 33  | Condenser     | 3864.44                            | 0.00       | 3864.44  | 0.73               |
| 14  | Economizer    | 3762.11                            | 0.00       | 3762.11  | 0.71               |
| 18  | Superheater   | 3726.50                            | 0.00       | 3726.50  | 0.71               |
| 32  | Heat exchgr.  | 11409.90                           | 0.00       | 11409.90 | 2.17               |
| 12  | Deaerator     | 14.07                              | 0.00       | 14.07    | 0.00               |
| 30  | Heat sink     | 40908.55                           | -40908.55  | 0.00     | 0.00               |
| 17  | Evaporator    | 8073.50                            | 0.00       | 8073.50  | 1.53               |
| 4   | Combustor     | 161705.16                          | 0.00       | 161705.16| 30.69              |
| 15  | Drum          | 2.72                               | 0.00       | 2.72     | 0.00               |
| 3   | Compressor    | -193251.55                         | -205346.09 | 12094.55 | 2.3                |
| 2   | Air source    | -64.35                             | 0.00       | -64.35   | -0.01              |
|   | Fuel source | -526871.19 | -526871.19 | 0.00 | 0.00 |
|---|-------------|------------|------------|------|------|
| 6 | Stack       | 14928.68   | 0.00       | 14928.68 | 2.83 |
|   | Total       | -0.05      | -225081.72 | 225081.68 | 42.72 |

Figure (6) explains the quantity of heat transfer for each apparatus in the HRSG, condenser, heat exchangers and stack. Evaporator has the highest irreversibility in the HRSG which means inefficient heat transfer in this device.

Figures (8) and (9) explain the exergy destruction and exergy efficiency for the apparatuses of the Zubair gas turbine cycle. The overall exergy destruction and exergy efficiency of the gas turbine plant is moreover presented. The result shows that the highest exergy destructions occurred in the combustion chamber of the gas turbine engine due to the chemical composition of combustion process and huge temperature differences between the flame and fluid flow. Also, combustion chamber has less amount of exergy efficiency compared with other apparatuses. However, the results exposed that the highest exergy efficiency occurred in the gas turbine wheel due to its huge work produced value and represented as the second exergy loser in GT engine as shown in figure (9).

Exergy destruction explained in figure (10) for combined cycle power plant each apparatus. The HRSG has the biggest exergy loss after combustion chamber in the CCPP. Also, shows exergy destruction for resent components like condenser, additional heat exchangers and stack.
Thermal and exergy efficiencies of the CCPP are calculated by using equations (17) and (18). They are $\eta_{th} = 51\%$ and $\eta_{ex} = 48.76\%$ respectively as shown in table (4). However, HRSG has the major exergy loss after the combustion chamber. Equation (17) could be used to calculate exergy loss for each parts of the HRSG. Therefore, every apparatus of the HRSG has two input (hot gases and water flow) and two output (gases with less temperature value and steam). Figure (11) shows that stack, evaporator (EV), and heat exchanger 32 have higher irreversibility.

Exergy destruction could be reduced by applying air intercooling for gas turbine, which can increase compressor efficiency and gas turbine output power.

The specific exergy destruction of evaporator 17 and the heat exchanger 32 are the highest value compared with other apparatuses like (superheater 18, economizer 14, and condenser 33) due to inefficient heat transfer process in these devices.

Exergy destruction could be reduced by applying air intercooling for gas turbine, which can increase compressor efficiency and gas turbine output power.

### 4.2 Effect of changing the ambient temperature

Figure (12) shows a slightly change in thermal and exergy efficiency for CCPPs according to changes in ambient temperature. After numbers of iteration process with the CCPP suggestion model using ambient temperatures (-5, 15, 35, 55 $^\circ$C) for the gas turbine input parameters selected from performance data GT13E2 [12]. As shown in tables (6).

Gas turbine output power decreases with increasing ambient temperature due to increasing the compressor consumption work for falling air density. Assuming constant compression pressure ratio ($Pr = 14:1$). Thermal efficiency of the CCPP model increase with ambient temperature rise due to rapidly increase in turbine inlet temperature.
Also, exergy efficiency increasing with rising ambient temperature according to decrease the \((T_o/T)\) value in equation no. (10). Steam turbine power output increase with an increase in exhaust gases (temperature and mass flow), due to the additional steam generated by additional heat amount in the HRSG of the CCPPs. However, the total CCPP output power decreases when ambient temperature increase due to the decrease in the gas turbine output power which is constructed about 70% from the total plant produced power as illustrated in figure (13).

![Fig.10. Exergy loss for combined-cycle power plant apparatus](image1)

![Fig.11. Exergy loss in HRSG apparatus](image2)

**Table (6) Ambient conditions versus plant behavior by Cycle-Tempo [14,15]**

| Parameters | T=15°C | T =35°C | T =55°C | T =-5°C |
|------------|-------|--------|--------|--------|
| GT output power (bar) input data | 178.332 | 163.773 | 125.3 | 189.3 |
| GT exhaust temperature(°C) | 506.4 | 519 | 550 | 497.1 |
| Humidity (%) | 60 | 40 | 40 | 40 |
| GT pressure input (bar) | 1.013 | 1.007 | 1.007 | 1.007 |
| ST input temperature(°C) | 492 | 504 | 535 | 482 |
| ST power output (MW) | 83.872 | 83.717 | 77.736 | 65.563 |
| TIT (°C) | 1042.58 | 1054.97 | 1101.73 | 1016.64 |
| GT gases mass flow (kg/s) | 651.173 | 589.822 | 473.465 | 515.652 |
| CCPP gross thermal efficiency (%) | 49.762 | 50.262 | 51.903 | 48.258 |
| CCPP gross exergy | 47.567 | 48.045 | 49.614 | 46.129 |
4.3 Validations

The validation test for this study is taken by compared with two previous studies by Woudstra [4] and Ameri [1]. Both studies calculated the exergy analysis with different power plants model for single pressure heat recovery steam generation (HRSG). There is a difference in a calculation value of the exergy efficiency and the exergy loss in these studies due to the differences in the turbines model. Woudstra [4] investigated gas turbine type Siemens V94.3A and Ameri [1] used Siemens V94.2. present study with ALSTOM GT13E2 type. Also the difference in ambient conditions, in general, the results differences are reasonable compared with other studies.

Figure (14) consists of two comparative parameters, net electrical power in (MW) blow line and overall exergy loss in (MW) red line, for three combined power plants single pressure HRSG for all models, Ameri [1] power plant had highest exergy loss value because it has supplementary burner in the stack (HRSG) type (additional chemical exergy process).

Figure (15) consists of two comparative parameters power plants, thermal efficiencies blow line and exergy efficiencies red line. Ameri [1] power plant had the lowest thermal efficiency because it has less amount of the turbine inlet temperature (971°C) which cause drops in the thermal and exergy efficiencies.
Figure (16) consists of two comparative parameters fuel exergy in (MW) blow line and combustion exergy loss in (MW) red line. Woudstra [4] gas turbine combustion chamber has the highest exergy loss due to the highest gas turbine power output 251 MW related to other models.

Figure (17) consists of two comparative parameters produced power from steam turbine in (MW) blow line and produced power from gas turbine in (MW) red line. Woudstra [4] gas turbine had the highest gas turbine produced electrical power due to the model Siemens (V94.3 A) compared with another gas turbine models.
5. Conclusion

This study can be concluded with an important point in bellow:

1- Convert gas turbine power plant (single cycle) into combined-cycle power plant efficiency can approximately rise from 32 to 55%. Also, net output power growth by 45%.

2- The irreversibility has been specified for each part of Zubair Oil Field CCPP through the exergy analysis by using designing data for GT13E2 ALSTOM gas turbine through Cycle-Tempo 5.1 thermodynamic analysis program.

3- The results explain that the exergy efficiency of the combustion chamber is the lowest components exergy efficiencies in the gas turbine for alliteration models (60-70%) due to higher irreversibility and the second large exergy loss occur in the HRSG about 12%.

4- Optimization of the HRSG has an essential part to dropping the exergy destruction of overall combined cycle one of these is establish multi-pressure HRSGs.

5- Ambient condition is the effective factor on the output produced electrical power for both gas and steam turbine, maximum power produced when approach to the ISO condition, in this study, have been taken various models of combined cycle with different ambient temperature (-5, 15, 35, 55 °C) for each condition different outlet power the results are gas turbine power reduces with increase in ambient temperature while steaming turbine output power growth with increase in ambient temperature due to HRSG temperature rises but the decreasing in gas turbine output power more affected than steam turbine power which led to decrease in the net output power of the CCPP.

6- Thermal and exergy efficiency rises with an increase in ambient temperature due to increasing of turbine inlet temperature.

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