Improvement of Electric Power Generation at Khor Al-Zubair Gas Turbine Power Plant by Using Vapor Compression Cooling Cycle

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Abstract- **In this work, both energy and exergy analyses have been carried out on General Electric (GE) gas turbine unit found in Khor Al-Zubair gas turbine power plant located in Basra, Iraq. The analysis covers the ISO (international standards organization) operating conditions in addition to actual operating data recorded for one month in hot season July 2016. The feasibility of adopting a vapor compression cycle (VCC) for cooling the intake air is evaluated. Generally, the study reveals an obvious drop off for most plant performance characteristics while operating during the hot season. Energy and exergy analyses show that adopting the vapor compression cycle to enhance Khor Al-Zubair GE unit could improve the power output by 20% and 27% in case of part-load and full-load conditions respectively. Both of first and second law efficiencies could be improved by 3.5% at part-load and 9% at full load. The expected cooling load needed for the unit is in the range of 2697 to 3024.5 TR according to part-load and full-load operation respectively. Only total irreversibility of the unit is expected to increase in case of adopting VCC and this will not impair the improvement in second law efficiency of the unit. Among the unit components, combustion chamber has the largest computed irreversibility. Further improvement is recommended by utilizing the released heat energy to the atmosphere, which is characterized by significant work potential.**

Index Terms—**Power plant, Gas turbine; Vapor compression cycle; Energy and exergy analysis.**

I. INTRODUCTION

Electricity is one of the greatest blessings of life on all humanity. It is the lifeline of the civilized world and the driving force of all aspects of modern life.

Electricity in Iraq has gone through many stages. Initially, its use was limited to lighting purposes, but its uses were rapidly expanded and the demand was increased steadily especially in the last decade. The rapid development of the standard of living and population has led to this growing demand. Many power plants were established in many parts of the country. Most of these plants were run by fossil fuels [1].

The gas turbine is one of the most widely used electricity power generating technologies around the world nowadays. It is a type of internal combustion engine that can convert the chemical energy of fuels like natural gas to mechanical energy, which drives the generator that produces electrical power. The gas turbine is composed of three main components namely compressor, combustion chamber, and turbine. The gas turbine power plants compared with other types of plants like steam turbine power plant are low in installation cost and do not require a large space of the installation. Moreover, it has a high production power per unit size. Conversely, one of the most important disadvantages of gas turbine power plant is the drop in its performance due to high ambient temperatures [2].

The main problem facing gas turbine power plants in Iraq and specifically in Basra city is the rise in ambient temperature. Accordingly, one solution to overcome this problem is to cool the intake air before entering the compressor. There are different methods that can be used for cooling the compressor intake air and one of these is the vapor compression refrigeration cycle.

Actually, analysis of the gas turbine power plants and revealing their performance characteristics are very important to clarify the possible enhancement opportunities using modern techniques. Although the analysis based on energy conservation is one of the fundamentals, but the analysis of the engine according to the principle of exergy has spread in recent times. That is due to its ability to identify the real losses that hinder the access to optimal performance situation [3].

As gas turbine technology is widely used for electric power production, it has been studied and analyzed by several research projects intentionally to recover the performance. Faisal et al [4] performed energy and exergy study of Rumaila Basra gas turbine power plant. The study was during hot season for full and part-load operating conditions. The combustion chamber is found to be responsible of plant irreversibility. Erickson et al [5] studied the outcome from gas turbine inlet conditioning using an absorption refrigeration cycle. The refrigeration unit is an ammonia–water type that is powered by turbine exhaust heat. They show, based on typical summer day data, that net power was increased from 40 to 51 MW. Panyam et al [6] presented a performance comparison study of gas turbine inlet air cooling using vapor adsorption refrigeration cycle. Different refrigerants were suggested and discussed. Among the selected refrigerants, ammonia is observed to improve plant performance significantly. Kamal et al [7] discussed the feasibility of turbine inlet air cooling in Malaysia climate using mechanical chillers. They claimed that this modification is effective for power augmentation in Malaysia by 27.5% to 32.11%.

Khor Al-Zubair gas turbine power plant is located in Khor Al-Zubair area at the south of Basra city, about 43 km south of the city center. Initially, the plant was established in 1975 with five identical generating units each with a nominal rating of 63 MW under ISO conditions. The plant continued production of four units with capacities ranging from 50 MW to 60 MW during the Iran-Iraq war using gas oil fuel because of the lack of natural gas at that time [8]. In 2006, two additional GE type units (model MS9001E) were installed with a design capacity of 126.1 MW per unit. Now, the total installed generating capacity of the plant is about 500 MW. The plant can utilize natural gas or gas oil fuel in the combustion process. The plant currently receives natural gas fuel from Rumaila oil field located in Basra production field for which the properties are given in Table I [8].

In the present study, one unit of GE gas turbine will be taken as a case study. Standard model specifications under ISO operating conditions are given in Table II [9]. The MS9001E GE model (also referred as PG9171E) is of single shaft arrangement. The 17 stages, axial compressors are supplied with a single variable pitch inlet guide vane (IGV) row that controls the air mass flow rate drawn by the unit. The combustion system is made of 14 separate combustion chambers, which are symmetrically distributed. The turbine is of impulse-reaction type with a small degree of reaction. It consists of three stages and the unit operates at 3000 r.p.m.

Table I
PROPERTIES OF FUEL GAS USED IN KHOR AL-ZUBAIR GAS TURBINE POWER PLANT [8].

Fuel type	Fuel gas	
Composition % by volume	CH_4	76.10
	C_2H_6	16.2
	C_3H_8	4.20
	nC_4H_{10}	0.30
	iC_4H_{10}	0.22
	nC_5H_{12}	0.01
	iC_5H_{12}	0.01
	N_2	1.1
	CO_2	1.86
Density	0.86 kg/m^3	
Molar mass	20.38 kg/kmole	
LHV	46256 kJ/kg	
HHV	53500 kJ/kg	

*for ISO conditions only 100% CH_4 is considered [9].

Table II
GAS TURBINE MODEL SPECIFICATION GE GAS TURBINE MODEL (MS9001E) (STANDARD AND PREDICTED)

	Standard Value[9]	Predicted value	Absolute Error, %
Net power output, MW	126.1	126.31	0.17
Compression ratio (used as input)	12.6	12.6	-
Air mass flow rate, kg/s	407.2	409.22	0.5
Fuel mass flow rate, kg/s	-	7.48	-
Exhaust gas temperature,℃	543	537.9	0.94
Firing temperature, ℃	1124	1129	0.45
Thermal efficiency,%	33.8	36.5	7.99

The analysis that presented in this study will reveal most of the performance characteristics during a typical hot season. In addition, the improvement that occurred if a VCC is engaged will be predicted. It is worth noting that there is already a cooling system installed for the old units but has not yet entered the service. The typical operation data are selected for hot season July-2016. The plant data are recorded 12 times a day. Performance characteristics of the plant will be examined according to these data based on energy and exergy principle.

II. THERMAL MODEL

A schematic of a gas turbine power plant cycle coupled to VCC is shown in Fig. 1 together with their representations on T-S and P-H diagrams. The modeling of the plant components before and after cooling is based on energy and exergy principles. The following assumptions are adopted in the present analyses:

- Steady-state operation for all components.
- In the turbine, compressors, combustion chamber, and expansion device the processes are adiabatic.
- The kinetic and potential energy and exergy are neglected.
- Air and combustion gases products are modeled as ideal gases with variable properties.
- The reference conditions for the gas turbine cycle is taken as P_o=101.325 kPa with variable reference ambient temperature. The reference conditions for VCC it is kept constant as $T_{o,VCC}$=25℃, $P_{o,VCC}$=101.325 kPa.
- Pressure losses are considered in the compressor inlet, combustion chamber, and turbine exit, and it is not considered in the VCC.
- The refrigerant (R134a) in the VCC leaves the condenser and evaporator as saturated liquid and vapor respectively.
- Compressed air is extracted at the final compressor stage.

A. *First Law Analysis (Energy Analysis)*

Based on the first law of thermodynamics, the energy conservation equation applied to an open system is given by [10]:

$$Q + \dot{m}_i h_i = \dot{m}_e h_e + \dot{W} \quad (1)$$

The mass flow rate at inlet and exit are restricted by the mass conservation equation:

$$\dot{m}_i = \dot{m}_e \quad (2)$$

1) *Energy Balance of the Air Compressor:*

The inlet air pressure is given by:

$$P_2 = P_o - \Delta P_{ac} \quad (3)$$

The compressor pressure ratio is:

$$PR = {}^{P_3}/_{P_2} \quad (4)$$

The air enthalpy and entropy at the inlet are a function of inlet pressure and temperature. In case of isentropic compression, then the entropy at the inlet is equal to the entropy at the exit. Using the entropy and pressure, the other isentropic exit properties can be calculated. The actual exit enthalpy is given by [11]:

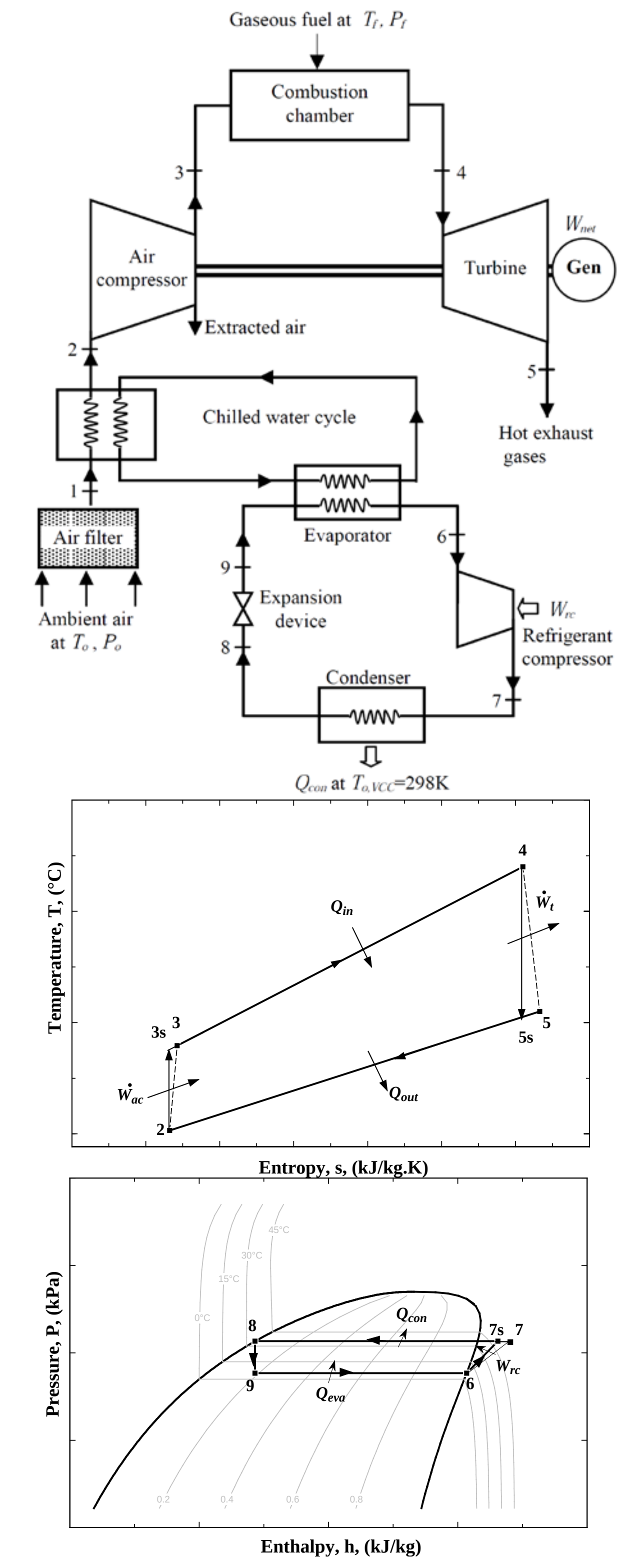

Fig. 1. Sketch of gas turbine power plant coupled with VCC and their representation on T-S and P-H diagram

$$h_{a,3} = h_{a,2} + \left(\frac{h_{a,3,is} - h_{a,2}}{\eta_{ac,is}}\right). \tag{5}$$

The work needed to drive the air compressor is:

$$\dot{W}_{ac} = \frac{\dot{m}_a\left(h_{a,3} - h_{a,2}\right)}{\eta_{ac.m}}. \tag{6}$$

2) *Energy Balance of Combustion Chamber:*

In the combustion chamber, fuel is burned with compressed air. The fuel mass flow rate needed to attain a specified firing temperature for combustion products is:

$$\dot{m}_f = F_{act} . \dot{m}_a (1 - XX) \tag{7}$$

The actual combustion reaction equation written for one kg of compressed air is given by [11]:

$$F_{act}\ kg\ Fuel + 1\ kg\ air \rightarrow$$
$$(F_{act} + \phi) stoi.produc + (1 - \phi) Excess\ air\ . \tag{8}$$

The actual fuel to air ratio is calculated with the aid of stoichiometric fuel to air ratio by:

$$F_{act} = \phi \cdot F_{stoi} \tag{9}$$

The stoichiometric reaction deals with combustion process where no excess air is supplied. The chemical reaction equation that represents the stoichiometric reaction is [10]:

$$C_xH_y + a_{stoi}\,[0.21O_2 + 0.79\ N_2] \longrightarrow b_{stoi}\,CO_2 + c_{stoi}\,H_2O$$
$$+ d_{stoi}\,N_2 \tag{10}$$

The above equation is written for any gaseous fuel. Actually, two important calculations are gained from the stoichiometric combustion equation. The first one is stoichiometric fuel to air ratio given by [10]:

$$F_{stoi} = \frac{(M_f)}{(a_{stoi}\ M_a)} \tag{11}$$

The stoichiometric kmoles of air is found by usual atomic balance for (10) [11].

The other important calculation is the enthalpy of stoichiometric products of combustions. The amount of stoichiometric products will not be affected by supplying extra air for the combustion process. The value of equivalence ratio, which is a measure for excess air, is calculated using total enthalpy concept. This principle gives the following relation [11,12]:

$$\phi = \frac{h_{a,4} - h_{a,3}}{(h_{a,4} - h_{a@15}) - \left[(1 + F_{stoi})\left(h_{stoi,4} - h_{stoi@15}\right)\right].. \quad ..+\left[(\eta_{cc} \cdot F_{stoi})\left(LHV + h_{f@T_f} - h_{f@15}\right)\right]} \tag{12}$$

The heat input to the plant is:

$$Q_{in} = \dot{m}_f\left(LHV + h_f - h_{f@15}\right) \tag{13}$$

3) *Energy Balance of Turbine:*

The pressure at the inlet and outlet of the turbine can be calculated as [2]:

$$P_4 = P_3 - \Delta P_{cc} \tag{14}$$

$$P_5 = P_1 + \Delta P_{exh} \tag{15}$$

In the case of isentropic expansion, then the entropy of combustion products at the turbine inlet is equal to the entropy at the turbine exit. By using this property along with exhaust pressure at state five, the other isentropic properties can be calculated. The actual enthalpy is given by [11]:

$$h_{p,5} = h_{p,4} - \eta_{t,is}\left(h_{p,4} - h_{p,5,is}\right) \tag{16}$$

This property together with the exhaust pressure at state five is used to evaluate the other actual properties. The output power from the turbine is calculated as:

$$\dot{W}_t = \frac{\left((1-XX)\,\dot{m}_a + \dot{m}_f\right)\left(h_{p,4} - h_{p,5}\right)}{\eta_{m,t}} \tag{17}$$

4) *Energy Analysis of Evaporator System:*

The evaporator system could contain a secondary working fluid e.g. water which transfers heat from the intake air to the VCC. The first step in performing the energy analysis of VCC is calculating the cooling load. Considering only the sensible heat, the cooling load required to cool the compressor intake air is [10]:

$$Q_{eva} = \dot{m}_a\left(h_{a,1} - h_{a,2}\right) \tag{18}$$

The heat absorbed by the refrigerant is:

$$Q_{eva} = \dot{m}_r\left(h_{r,6} - h_{r,9}\right) \tag{19}$$

The exit enthalpy from the evaporator is the saturated vapor enthalpy at evaporator temperature.

Equating (18) and (19), the refrigerant mass flow rate is given by:

$$\dot{m}_r = \frac{\dot{m}_a\left(h_{a,1} - h_{a,2}\right)}{\left(h_{r,6} - h_{r,9}\right)} \tag{20}$$

5) *Energy Analysis of Refrigerant Compressor:*

Like the air compressor in the gas turbine cycle, the real refrigerant enthalpy of compressor exit can be calculated using the isentropic efficiency definition as [10]:

$$h_{r,7} = h_{r,6} + \left(\frac{h_{r,is,7} - h_{r,6}}{\eta_{is,rc}}\right) \tag{21}$$

Using isentropic entropy and the refrigerant compressor exit pressure, the refrigerant isentropic properties at compressor exit can be found. The compression power is given by:

$$\dot{W}_{rc} = \frac{\dot{m}_r\left(h_{r,7} - h_{r,6}\right)}{\eta_{m,rc}} \tag{22}$$

6) *Energy Analysis of Condenser:*

The heat rejection in the condenser can be calculated using energy balance for the condenser as follows [10]:

$$Q_{con} = \dot{m}_r\left(h_{r,7} - h_{r,8}\right) \tag{23}$$

The exit enthalpy from the condenser is the saturated liquid enthalpy evaluated at condenser temperature.

7) *Energy Analysis of Expansion Device:*

The energy balance of expansion device is:

$$h_{r,9} = h_{r,8} \tag{24}$$

B. Second Law Analysis (Exergy Analysis)

The exergy analysis is based on the second law of thermodynamic. It is a very useful tool that gives a deeper understanding for the performance of gas turbine with and without adopting VCC. The general exergy balance equation is given as [13]:

$$\Psi_w = \sum_k \left(1 - \frac{T_o}{T_k}\right) Q_k + \sum\left[(\dot{m}\psi)_i - (\dot{m}\psi)_e\right] - T_o S_{gen} \tag{25}$$

The flow exergy is defined as [13]:

$$\psi = \dot{m}\left[(h - h_o) - T_o(s - s_o)\right] \tag{26}$$

The irreversibility or exergy destroyed in the process of the system is:

$$I = T_o S_{gen} \tag{27}$$

The entropy generation can be obtained from entropy balance as follows [14]:

$$S_{gen} = \sum(\dot{m}\cdot s)_e - (\dot{m}\cdot s)_i - \sum\frac{\dot{Q}_k}{T_k} \tag{28}$$

1) *Exergy Analysis of Air Compressor:*

The exergy balance for air compressor is given by:

$$T_o S_{gen,ac} = \dot{W}_{ac} + \dot{m}_a\left(\psi_{a,2} - \psi_{a,3}\right) \tag{29}$$

The irreversibility of air compressor is:

$$I_{ac} = T_o S_{gen,ac} = \dot{m}_a T_o\left(s_{a,3} - s_{a,2}\right) \tag{30}$$

2) *Exergy Analysis of Combustion Chamber:*

The exergy balance of the combustion chamber is:

$$I_{cc} = (1-XX)\dot{m}_a\psi_{a,3} + \dot{m}_f EX_f - \left((1-XX)\dot{m}_a + \dot{m}_f\right)\psi_{p,4} \tag{31}$$

The fuel exergy consists of two parts. The first one is the physical exergy that associated with fuel stream. The second part, which is very important, is the chemical exergy which is associated with its chemical energy [13, 14]. For an adiabatic reaction that occurs in the combustion chamber, the maximum work potential (fuel chemical exergy) can be achieved under reversible conditions. This is given by:

$$EX_f^{ch} = \sum_{Reactants} n_i\left(\bar{h}_f^o + \Delta\bar{h} - T_o\bar{s}\right)_i - \sum_{Products} n_i\left(\bar{h}_f^o + \Delta\bar{h} - T_o\bar{s}\right)_i \tag{32}$$

The total fuel exergy in this case will be [14]:

$$EX_f = \frac{1}{M_f} EX_f^{ch} + \left(\psi_f - \psi_{f@15°C}\right) \tag{33}$$

3) *Exergy Analysis of Turbine:*

The exergy balance for the gas turbine is give by:

$$T_o S_{gen} = \left((1-XX)\dot{m}_a + \dot{m}_f\right)\left(\psi_{p,4} - \psi_{p,5}\right) - \dot{W}_t \tag{34}$$

The irreversibility is given by:

$$I_t = T_o\left((1-XX)\dot{m}_a + \dot{m}_f\right)\times\left(s_{p,5} - s_{p,4}\right) \tag{35}$$

4) *Exergy Analysis of Evaporator System:*

The entropy generation rate of evaporator can be calculated by applying the entropy balance for evaporator system, which gives [13]:

$$S_{gen,eva} = \dot{m}_{a,1}(s_{a,2} - s_{a,1}) + \dot{m}_r(s_{r,6} - s_{r,9}) \quad (36)$$

The exergy destroyed in the evaporator is given by:

$$I_{eva} = \dot{m}_a T_o(s_{a,2} - s_{a,1}) + \dot{m}_r T_{o,VCC}(s_{r,6} - s_{r,9}) \quad (37)$$

The reference temperature of VCC is $= 298K$

5) *Exergy Analysis of Refrigerant Compressor:*

The entropy balance of the refrigerant compressor is given by [5]:

$$S_{gen,rc} = \dot{m}_r(s_{r,7} - s_{r,6}) \quad (38)$$

The irreversibility in the refrigerant compressor is:

$$I_{rc} = T_{o,VCC} \cdot S_{gen,rc} \quad (39)$$

6) *Exergy Analysis of Condenser:*

The entropy balance of the condenser is written as:

$$S_{gen,con} = \dot{m}_r(s_{r,8} - s_{r,7}) + \frac{Q_{con}}{T_{o,VCC}} \quad (40)$$

The irreversibility in the condenser is:

$$I_{con} = T_{o,VCC}\, S_{gen,con} \quad (41)$$

4) *Exergy Analysis of Expansion Device:*

In the refrigerant expansion valve, the entropy generation is:

$$S_{gen,exp} = \dot{m}_r(s_{r,9} - s_{r,8}) \quad (42)$$

Thus, the exergy destroyed is:

$$I_{exp} = T_{o,VCC}\, S_{gen,exp} \quad (43)$$

C. *Overall Plant Performance Characteristics:*

The net output power from the plant is calculated as:

$$\dot{W}_{net} = \eta_{gen}(\dot{W}_t - \dot{W}_{ac}) - \dot{W}_{rc} \quad (44)$$

The exhaust heat rejected by the gas turbine to the environment is:

$$Q_{out} = Q_{in} - \dot{W}_{net} \quad (45)$$

The first law efficiency of the gas turbine power plant is the ratio of net output power to the input heat, i.e.:

$$\eta_{I,gt} = \frac{\dot{W}_{net}}{Q_{in}} \quad (46)$$

The second law efficiency for the plant can be written as:

$$\eta_{II,gt} = \frac{\dot{W}_{net}}{\dot{m}_f EX_f} \quad (47)$$

The above equation can be written in terms of irreversibilities as:

$$\eta_{II,gt} = 1 - \frac{I_{tot}}{\dot{m}_f EX_f} \quad (48)$$

The total irreversibility is given by:

$$I_{tot} = I_{ac} + I_{cc} + I_t + I_{eva} + I_{rc} + I_{con} + I_{exp} \quad (49)$$

The lost exergy by exhaust gases to the environment is:

$$EX_{lost} = \dot{m}_f EX_f - \dot{W}_{net} - I_{tot} \quad (50)$$

III. Results and Discussion

The entire mathematical model presented in this study was set in computer code using "Engineering Equation Solver (EES)". In the GE unit operation data, there are no records for air mass flow rate and firing temperature. However, other data are recorded. Therefore, two main iterations are prepared in the solution algorithm. Firstly, the air mass flow rate is to be found from the recorded net power output. The second one is to find the firing temperature from the recorded exhaust temperature. Assumptions and reference data needed to complete the solution for both full and part-load are given in Table III.

Table II shows the predicted values of standard ISO specifications of the GE unit. All the results show good agreement except thermal efficiency, which has the maximum absolute error of 7.99%. However, this absolute error percentage is acceptable given that the fuel mass flow rate is too low compared with the air mass flow rate. This procedure is considered as a verification for the model.

Figs. 2 and 3 show the energy and exergy distribution for Khor Al-Zubair GE unit. These two distributions are restricted for the designed full-load ISO operation conditions. The input fuel represents the source of energy and exergy supplied to the unit. The power output accounts 36.5% and 31.47% from the input energy and exergy respectively. Actually, these two percentages are the-

Table III
Assumptions and Reference Data.

Assumptions for gas turbine	
ISO conditions	15 ℃, 101.325 kPa
Fuel condition	25 ℃, 2500kPa
$\eta_{ac,m} = \eta_{t,m} = \eta_{gen}$	0.95
$\eta_{ac,is}$	0.87
$\eta_{t,is}$	0.92
η_{cc}	0.98
$\Delta P_{ac} = \Delta P_t$	10 kPa
ΔP_{cc}	0.04* P_3
Compressor air extraction	13%
Assumptions for VCC	
Refrigerant	R134a
Condenser temperature	35 ℃
Evaporator temperature	5 ℃
$\eta_{rc,is}$	0.95
Dead state temperature	25℃

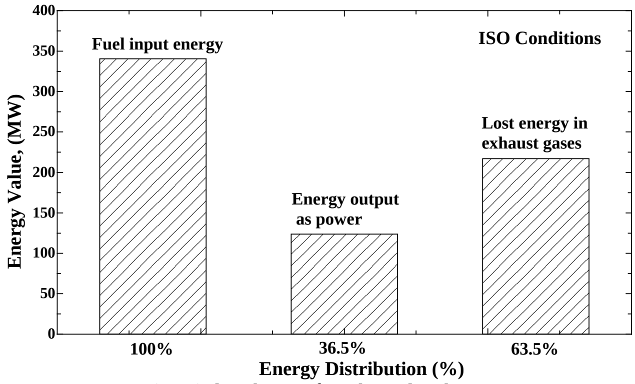

Fig. 2. Energy (MW) distribution for Khor Al-Zubair GE unit at ISO conditions.

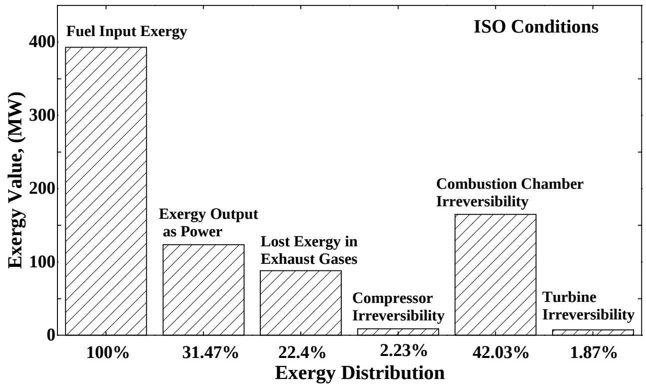

Fig. 3. Exergy (MW) distribution for Khor Al-Zubair GE unit at ISO conditions.

definitions of first law and second law efficiencies of the unit. As seen from figures, the plant releases 63.5% of the input energy to the environment as hot exhaust gases. This lost energy is accompanied by work potential that accounts for 22.4% from the input fuel exergy. That is mainly due to the high exhaust temperature. Unfortunately, this energy accompanied with its hidden exergy is not utilized at the present time. It certainly offers a good opportunity to develop GE unit performance. Fig. 3 reveals that the maximum irreversibility occurs in the combustion chamber of the unit, which accounts for 42.03% from the input fuel exergy. This is due to the excessive temperature resulted from the combustion process that occurs adiabatically in this component besides the absence of work interaction. Lower irreversibility is found to have occurred in compressor and turbine. The irreversibilities of these two components account about 2% of the input fuel exergy. The losses that occurred in the compressor and turbine are termed as internal. That means there is no heat interaction with surroundings.

Actually, the irreversibility analysis explained before is very useful. The discussion explains that if there is a prospect to develop the plant performance toward the top, then this will certainly be in the combustion chamber. Although there is much interest in fuel cell technology nowadays, unfortunately, traditional combustion process still the dominant way to extract the chemical energy of fuel.

In fact, the plant operates at part-load and many specifications are not at their standard ISO values. Fig. 4 shows the variation of actual and predicted temperatures at inlet and outlet of the GE unit components during July 2016. It is clear that the ambient temperature suffers a wide range of variation away from the ISO operation condition.

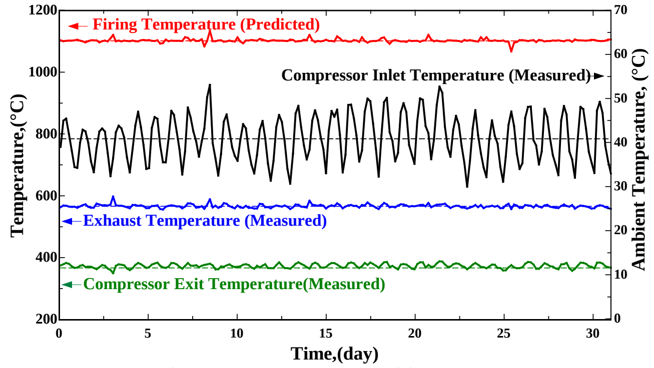

Fig. 4. Variation of operating temperatures of for Khor Al-Zubair GE unit during July 2016.

The minimum recorded ambient temperature is 29°C which occurs on July 23rd at 6:00 AM. The maximum recorded one is 56°C which occurred on July ninth at 4:00 PM and the mean value is 40.9°C. In fact, the change in ambient temperature above ISO condition along with the load demand on the plant (characterized by the IGV position) is responsible for degradation the entire performance specifications of the unit. The compressor outlet temperature variation is affected by the variation of both the pressure ratio and the ambient temperature. The exhaust temperature from the turbine is nearly constant at an average value of 565°C. Actually, the control system of the engine tries to keep the exhaust temperature at a constant level. This situation is reached by adjusting the fuel mass flow rate, which linked to the air mass flow rate drawn to the unit. Because of this, the firing temperature will vary so that it will be always on the safe side. It is found that the firing temperature is in the range of 1055-1118°C with an average value of 1102°C.

Fig. 5 shows the variation of actual output power from Khor Al-Zubair GE unit during July 2016. The figure reveals that the average power production during the selected time period is just 98.44 MW. This indicates a penalty of 22% drop from that at ISO conditions. This manner is attributed to the effect of air mass flow rate consumed by the unit. The air mass flow rate is affected by the ambient temperature as well as the position of IGV that control the load on the engine. For a given position of IGV, increasing the ambient temperature will decrease the density of air and hence air mass flow rate will be lower.

The figure also indicates that adopting VCC for cooling the inlet air will improve the net output power in case of part and full-load operation. In part-load condition, the IGV is assumed to be at the same recorded one (same load demand) and hence, the predicted increase in power is solely due to the improvement in air density.

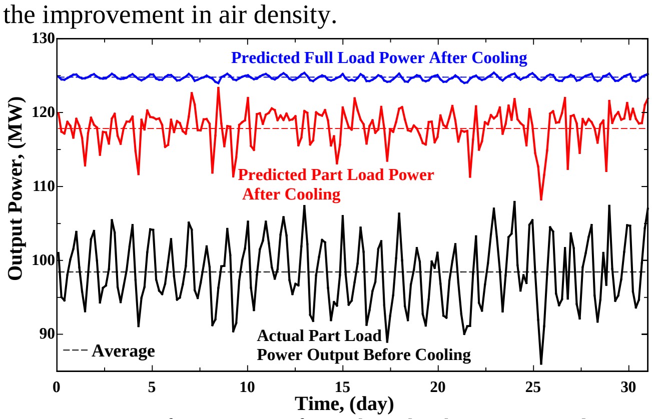

Fig. 5. Variation of power output from Khor Al-Zubair GE unit and power enhancement predicted after cooling with VCC.

It is found that the percentage enhancement in power output after cooling inlet air in case of part-load operation varies as 10.6 - 25.7 % according to the load demand and ambient temperature. Based on the average value, the power could improve to 117.83 MW, i.e. the percentage increase of 20 %. In case of enhancement of the unit to full-load operation, the intake air temperature will be restored back to ISO operating temperature of 15 °C while the IGV position is assumed to be fully opened. Consequently, the predicted increase in power will be higher. Based on the average value, the power could improve to 124.8 MW, i.e. the percentage increase of 27%. The reason that the maximum ISO net power will not be attained is the power required to drive the VCC. Generally, this case is impractical; since the unit is always operating on part-load condition.

Fig. 6 explains the variation of thermal efficiency that based on the first law of thermodynamics with time for Khor Al-Zubair GE unit both before and after unit modification. The figure explains that the variation of curves before and after modification follows the same style. This behavior is attributed to the fluctuation of air mass flow rate resulted from unit augmentation. Without a doubt, the proposed intake cooling technique will improve thermal efficiency. This expected due to the increase in power developed. The value of thermal efficiency without unit modification is ranged from 31.7- 34.3% with average value of 33.07%. After unit modification using VCC, the thermal efficiency is predicted to improve to 34.24% for part-load and to 36.04% for full-load operation based on average values.

Fig. 7 represents the prediction of cooling load required for the GE unit. The only sensible part of the cooling load is calculated that is needed to cool the inlet temperature back to ISO temperature of 15 ^{o}C. As explained before, two cases are studied which are the full and part-load condition.

The results show that the average cooling load needed is 3024.5 T.R for full-load operation and 2697 T.R for part-load operation.

The last discussion revealed that Khor Al-Zubair GE unit energy performance characteristics are improved after applying the VCC for cooling the inlet air.

The variation of total exergy destruction in Khor Al-Zubair GE unit components with time before and after the adopting VCC is shown in Fig. 8. The figure exhibits an increase in the unit total exergy destruction as the inlet temperature is restored to 15 oC in the new cycle configuration. This behavior is entirely due to the increase of plant components. Before unit modification, irreversibility occurred in the compressor, combustion chamber, and turbine. After unit modification, new irreversibility is found in VCC components, which are the compressor, condenser, expansion device, and evaporator. Moreover, it is found that the irreversibilities of the original unit components (compressor, combustion chamber, and turbine) are all increased after unit modifications. This is well understood since irreversibility is directly proportional to ambient temperature in spite of restoring specific entropy and air and combustion products.

On the other hand, the intake air cooling using VCC causes a rise in the overall second law efficiency of the unit and this is shown in Fig. 9. Although the increase in unit irreversibility after applying VCC has a side effect of the second law efficiency, the increase in the fuel exergy input to the unit after cooling along with boosting the exergy recovered as useful work are the main reasons for this behavior. Results show that the actual second law efficiency is 32.34% and it can be improved to 33.48% for part-load and to 35.24% for full-load operation all based on the average value.

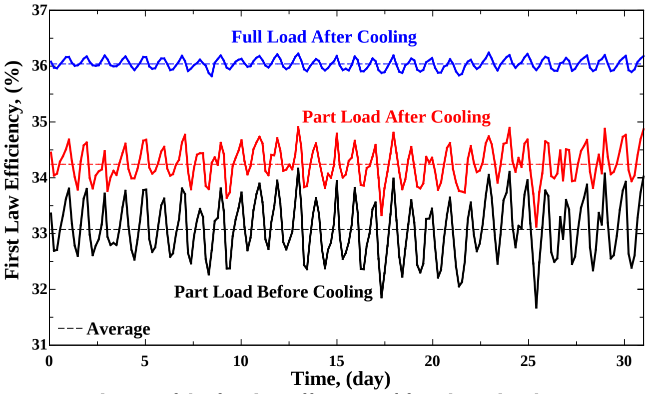

Fig. 6. Prediction of the first law efficiency of for Khor Al-Zubair GE unit before and after cooling with VCC.

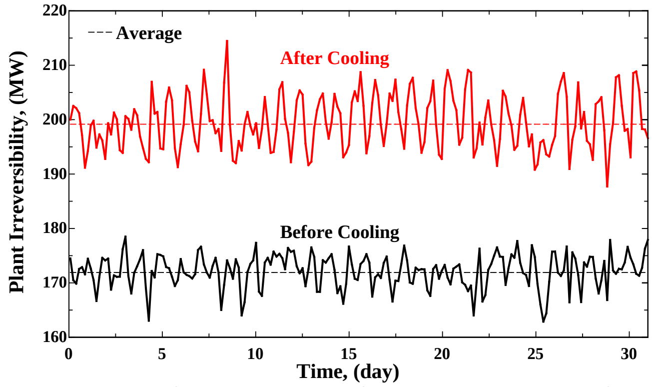

Fig. 8. Variation of total irreversibility of Khor Al-Zubair GE unit before and after cooling with VCC.

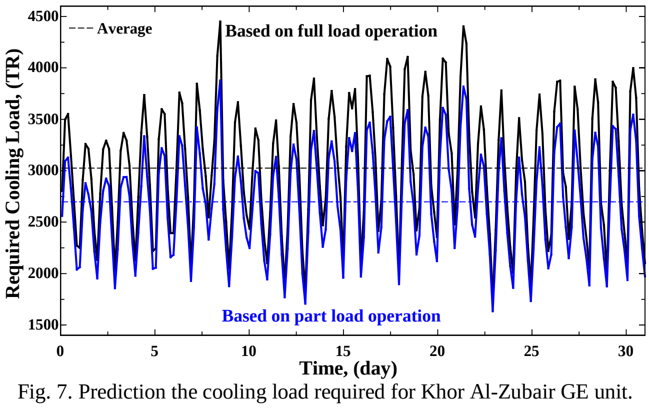

Fig. 7. Prediction the cooling load required for Khor Al-Zubair GE unit.

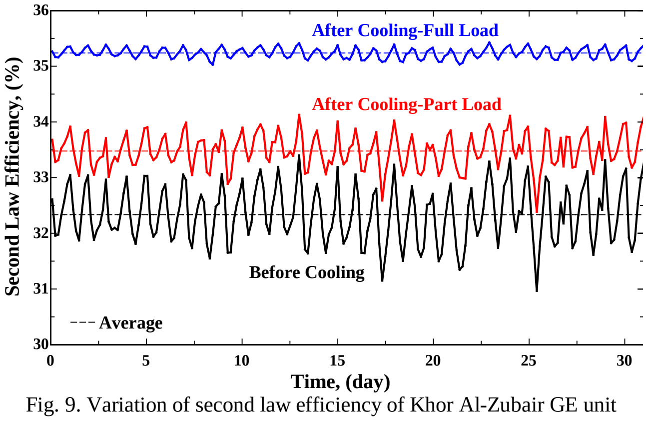

Fig. 9. Variation of second law efficiency of Khor Al-Zubair GE unit before and after cooling with VCC.

IV. CONCLUSIONS

The present work highlights the energy and exergy aspects of GE unit found in Khor Al-Zubair gas turbine power plant under Basra conditions. The study enables to quantify the actual energy and exergy losses in the unit and predict the enhancement that could be achieved when adopting VCC for cooling the intake air. The following conclusions are reached:

- Ambient temperature has a clear effect of the performance on the GE unit operation. All the studied performance specifications are found to be declined with increase the ambient temperature.
- Studying Khor Al-Zubair GE unit shows that the required cooling load capacity is ranged from 2697 TR to 3024.5 TR according to part-load and full-load operation respectively.
- Enhancing Khor Al-Zubair GE unit by using VCC is possible and this modification could improve the power output by 20% and 27% in case of part-load and full-load condition respectively. Both first and second law efficiencies could be improved by 3.5% at part-load and 9% at full load.
- In the case of adopting VCC, only total irreversibility of the unit will increase and this will not affect the improvement in second law efficiency of the unit.

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VII. NOMENCLATURE

English symbols

$a, b, c, and\ d$	Number of kmoles (kmole).
C_xH_y	General fuel formula.
EX	Exergy term (kW).
F	Fuel to air ratio (kg_f/kg_a).
h	Specific enthalpy (kJ/kg).
$\bar{h}_f^o$	Enthalpy of formation (kJ/kmole)
HHV	Higher heating value (kJ/kg_f).
I	Irreversibility (kW).
LHV	Lower heating value (kJ/kg_f).
M	Molecular weight (kg/kmole).
$\dot{m}$	Mass flow rate (kg/s).
n	Number of kmoles.
P	Pressure (kPa).
PR	Pressure ratio (-).
Q	Heat power (kW).
s	Specific entropy (kJ/kg.K).
S_{gen}	Entropy generation (kW/K).
T	Temperature (K).
$\dot{W}$	Power (kW).
XX	Fraction of compressed air extraction.

Greek symbols

η	Efficiency (-).
Δ	Difference (-).
ϕ	Equivalence ratio (-).
ψ	Flow physical exergy(kJ/kg).
Ψ_w	Exergy due to work (kw).

Subscripts

$1, 2, \ldots 9$	Points at gas turbine and VCC cycles.
a	Air.
ac	Air compressor.
act	Actual.
cc	Combustion chamber.
con	Condenser.
eva	Evaporator.
exh	Exhaust.
exp	Expansion device.
f	Fuel.
gen	Generator.
gt	Gas turbine.
I	First law.
II	Second law.
i, e	Inlet and exit.
in, out	Input and output.
is	Isentropic.
$lost$	Lost.
m	Mechanical.
net	Net value.
o	Denotes to the dead state.
out	Out.
p	Products of combustion.
r	Refrigerant.
rc	Refrigerant compressor.
sp	Stoichiometric products.
$stoi$	Stoichiometric.
t	Turbine.

tot	Total.
VCC	Vapor compression cycle.
Superscripts	
ch	Chemical.
$-$	Denotes per kmole.
Abbreviations	
EES	Engineering equation solver.
GE	General Electric Co.
IGV	Inlet guide vans.
ISO	International standards organization.
T.R	Ton refrigeration
VCC	Vapor compression cycle.