Exergy Analysis of a Power Plant in Abu Dhabi (UAE)

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***Abstract*-The first objective of this research was to conduct an exergy analysis for a power-generation plant in Abu Dhabi (UAE) in order to determine the main source of its irreversibilities. Our results indicated that the combustion chamber was the main contributor (70.2%) to the total exergy destruction of the plant. The compressor had the lowest contribution towards exergy destruction (12.4%). In the second part of this investigation, Aspen Hysys with the Soave-Redlich-Kwong (SRK) equation of state was utilized in order to simulate the effects of high temperatures and absolute humidity of ambient air on the exergy efficiency of different equipment in the power plant. Our data showed that the temperature of ambient air had more negative effects on the performance of the plant than the absolute humidity. The exergy efficiency of the combustor and compressor decreased with temperature while the exergy efficiency of the turbine increased with temperature. Finally, a comparative study on the performance of the power plant between design conditions (T=288K, RH=60%) and summer conditions (T=316K, RH=50%) in Abu Dhabi indicated that the power plant lost 4.66 % of its net power output and 4.61 % of its exergy efficiency.**

Keywords- Power Plant; Exergy Analysis; Exergy Destruction; Process Irreversibilities

I. INTRODUCTION

Power generation plants play a decisive role in the economic growth of the UAE. However, these power plants also contribute greatly to the annual CO_2 emissions. For example, these plants released about 43% of the 76 million tons of CO_2 during the year 2008 [1]. Recent developments show that the concept of sustainability is gaining attention among policy-makers in the UAE. Following this perspective, companies from the ADNOC group implemented a new Energy Management System ISO50001 in order to reduce CO_2 emissions by increasing the energy efficiency of all industrial facilities.

The standard conditions used for the design of gas turbines are 288K, sea level atmospheric pressure and 60% relative humidity [2]. In Abu Dhabi, the performance of gas-turbine power plants could therefore be affected because its specific atmospheric conditions are different from the ISO requirements. For example, the output of the plant power will decrease due to high temperatures, as there is a reduction in air mass flow rate; efficiency will also decrease because the compressor requires more power to compress air of a higher temperatures.

Energy balance is not useful for analyzing the performance of power plants because the real plant inefficiencies are not related to heat loss but to exergy destruction. The goal of an exergy analysis is to detect the irreversibilities in the plant's equipment. By evaluating the exergy destroyed in each component of the process, efforts will be focused on the equipment that presents the highest exergy destruction since it will offer the largest improvement of the overall efficiency of the power plant.

The first objective of this investigation is to conduct an exergy analysis of a power plant in Abu Dhabi (UAE) during a typical summer day in order to locate and evaluate the exergy destruction caused by the plant's irreversibilities. In the second part, Hysys V8.6 with the Soave-Redlich-Kwong (SRK) equation will be used asa tool to investigate the effects of the ambient air's temperature and absolute humidity on the exergy destruction in different parts of the plant as well as its net power output and exergy efficiency.

II. THEORETICAL BACKGROUND

A. Concept of Exergy

Exergy is commonly defined as the maximum theoretical work that can be extracted from a "combined system", consisting of a "system" under study and its "environment" as the system passes from an initial state to a state of equilibrium with the environment [3]. When a system is in equilibrium with the environment, the state of the system is called a "dead state", and its exergetic value is zero. According to Bejan et al. [3], the total exergy (E_T) of a stream is constituted by four main components:

$$E_T = E_{ph} + E_{Ch} + E_k + E_p \tag{1}$$

The physical exergy (E_{ph}) is often described as the maximum theoretical useful work obtainable as the system passes from its initial state (P, T) to the "restricted dead state" (P_0, T_0). The chemical exergy (E_{ch}) is the maximum useful work obtainable as the system passes from the "restricted dead state", where only the conditions of mechanical and thermal equilibrium are

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satisfied, to the "dead state" where it is in complete equilibrium with the environment [4]. The kinetic (E_k) and potential (E_p) exergies are associated with the system's velocity and height, measured relative to a given reference point. When a system is at rest in relation to the environment (E_k=E_p=0), the total mass specific exergy (e_T) of a stream is defined as:

$$e_T = e_{ph} + e_{Ch} \tag{2}$$

B. Standard Chemical Exergy of a Gas Mixture

The chemical exergy per mole of gas (k) is given by the following equation [4]:

$$e_{Ch}^k = -R.T.\ln x_e^k \tag{3}$$

For a mixture of gases, the chemical exergy per mole of the mixture can be estimated using [4]:

$$\mathrm{e_{Ch}} = \sum \mathrm{x_k}\,.\,\mathrm{e_{Ch}^k} + \mathrm{R.T.}\sum \mathrm{x_k}\,.\,\mathrm{lnx_k} \tag{4}$$

The exergy of fuel is equivalent to the calculated reversible work. The chemical exergy of a fuel can be estimated using equation (4), and the value of the exergy of hydrocarbons and other components are listed in the literature [3]. It should be noted that the value of the specific chemical exergy of a fuel at dead-state conditions is between the lower (LHV) and higher (HHV) heating values of the fuel [4].

C. Exergy Balance in Open Systems

Unlike energy, exergy is not conserved in any real process. As a consequence, an exergy balance must contain a "destruction" term, which vanishes only for a reversible process. The general form of exergy balance for a control volume can be written as [4]:

$$\frac{\mathrm{dE_{CV}}}{\mathrm{dt}} = \Sigma\mathrm{E_{heat}} + \mathrm{E_{work}} + \Sigma\mathrm{m_i}\,.\,\mathrm{e_{T,i}} - \Sigma\mathrm{m_e}.\,\mathrm{e_{T,e}} - \mathrm{E_D} \tag{5}$$

For a steady state system, equation (5) can be rewritten as:

$$0 = \Sigma\mathrm{E_{heat}} - \mathrm{W_{cv}} + \Sigma\mathrm{m_i}\,.\,\mathrm{e_{T,i}} - \Sigma\mathrm{m_e}.\,\mathrm{e_{T,e}} - \mathrm{E_D} \tag{6}$$

In equation (6), the total specific exergy transfer at the inlets and outlets can be written as:

$$\mathrm{e_T} = (\mathrm{h} - \mathrm{h_0}) - \mathrm{T_0}\,(\mathrm{s} - \mathrm{s_0}) + \sum \mathrm{x_k}\,.\,\mathrm{e_{Ch}^k} + \mathrm{R.T.}\sum \mathrm{x_k}\,.\,\mathrm{lnx_k} \tag{7}$$

h and s are properties at the inlet and the outlet of the system. h_0 and s_0 are the specific enthalpy and the specific entropy of the restricted dead state, respectively.

D. Energy & Exergy Efficiencies of Power Cycles

In the analysis of an energy conversion system, it is important to define the system's boundaries. For example, the exergy efficiency (Second law efficiency) η_{ex} of a cycle is defined as:

$$\eta_{\mathrm{ex,1}} = 1 - \frac{\mathrm{w_{net,out}}}{\mathrm{ex_{in}}} = 1 - \frac{\mathrm{ex_{dest}}}{\mathrm{ex_{in}}} \tag{8}$$

ex_{in} is the specific exergy input to the cycle, and ex_{dest} is the specific total exergy destruction in the cycle. However, in order to consider the exergy destruction due to incomplete combustion and the energy of the exhaust gases, the following equation is usually utilized to determine the exergy efficiency of a power plant:

$$\eta_{\mathrm{ex,2}} = \frac{\mathrm{W_{net,out}}}{\mathrm{m(t)_{f\,.}ex_f}} \tag{9}$$

$e_{x,f}$ is the exergy of the fuel.

III. LITERATURE REVIEW

Many researchers and scientists have investigated the effects of operating conditions on the performance of power plants. Rahman et al. [5] studied the effects of variation in some operating conditions (compression ratio, turbine inlet and exhaust temperature, air to fuel ratio, isentropic compressor and turbine efficiency, and ambient temperature) and on the performance of the gas turbine (energy efficiency, compressor work, power, specific fuel consumption, and heat rate). The results showed that the compression ratio, ambient temperature, air to fuel ratio as well as the isentropic efficiencies strongly influence the energy efficiency of a power plant. They also found that energy efficiency and power output decrease linearly with the increase of the ambient temperature.

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Altayib, K. [6] conducted an exergetic analysis on a power plant in Makkah (KSA) consisting of 18 gas turbine units with generating capacities ranging between 18 and 62 MW with 9 small diesel engines that range between 2 and 9 MW. The total installed generating capacity of the plant is about 900 MW. A parametric study and plant optimization were performed to investigate the effects of variation on specific input parameters, such as fuel mass flow rate, air volume flow rate, and compressor inlet air temperature on the overall operating efficiency of the system. The results showed that the overall plant energetic and exergetic efficiencies increased by 20% and 12%, respectively, when the compressor inlet temperature was cooled by 10K. Furthermore, the exergy analysis indicated that the largest exergy destruction occurs in the combustion chamber, followed by the turbine.

Chand, V.T. et al. [7] conducted an exergy analysis on a 112.4 MW single shaft open cycle active gas turbine plant to determine the performance of the plant. They carried out parametric analysis of the influence of various factors, namely the compressor pressure ratio (rp), compressor inlet air temperature (AT), and turbine inlet temperature (TIT), on the irreversibilities (exergy destruction) of each component of the gas turbine plant. The following conclusions were made: (1) Compressor inlet temperature has a significant effect on the exergy efficiency of gas turbine plant components. (2) Energy and exergy efficiency decreases for any increase in ambient temperature (AT). (3) The irreversibilities of the combustion chamber and gas turbine plant decrease when turbine inlet temperatures increase.

Henry, E. et al. [8] conducted an exergy analysis on a gas turbine power plant generating a capacity of 335 MW and utilizing natural gas as its combustion fuel. The obtained data showed that the gas turbine had the largest exergy efficiency at 96.17%, while that of the total plant was 41.83%. On the other hand, the combustion chamber had the largest exergy destruction at 54.15% while that of the total plant was 58.17%. The effects of the gas turbine load variation and the ambient temperature from 294K – 306K were also investigated. Exergy efficiency decreased and exergy destruction increased as the ambient temperature increased. The authors recommended that a cooling system be installed in order to decrease the effects of the high temperature of ambient air on the plant's performance.

Al Doori, W. [9] performed an exergetic analysis for a Baiji (India) power plant with a gas turbine with a capacity of 159 MW using fuel oil (LHV =42.9 MJ/kg) as the combustible. The rate of exergy destruction in the turbine was around 5.4%, whereas that in the combustion chamber was about 36.4%.

Abam, D. P. S and Moses, N. N. [10] carried out an exergy analysis of a 33 M W gas turbine power plant in Nigeria that uses natural gas. The reference temperature (T_0) and pressure (P_0) utilized were 25^0C and 1.0132 bars, respectively. An air humidity of 60% also used. The operating conditions are shown in Table 1.

TABLE 1 OPERATING CONDITIONS USED FOR THE EXERGY ANALYSIS

Operating Condition		Value
ass flow rate of air through compressor	(kg/s)	136.5
emperature of inlet air to compressor	(K)	302
ressure of inlet air to compressor	(MPa)	0.10132
utlet temperature of air from compressor	(K)	603
Outlet pressure of air from compressor	(Mpa)	0.835
Fuel gas mass flow rate	(kg/s)	2.80
Fuel-Air Ratio at full load	(on mass basis)	0.02
Fuel gas inlet temperature	(K)	302
Fuel gas inlet pressure	(MPa)	0.2279
Gas turbine inlet temperature	(K)	1087

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Exhaust gas temperature	(K)	644
Exhaust gas pressure	(MPa)	0.10132

Simulation outputs found that the exergy efficiency of the overall plant at a compressor inlet air temperature of 302K and a turbine inlet temperature of 1087K was 25.8%. Moreover, the combustion chamber had the largest exergy destruction at 65 MW, and the gas turbine had the lowest exergy destruction at 0.45 MW. The results also revealed that the exergy destruction, exergy efficiency, exergy flow rate of the power output, power to heat ratio and the specific fuel consumption depend on the ambient temperature and the turbine inlet temperature.

Abam, F. I. et al. [11] conducted an exergy analysis on a 138 MW gas turbine plant with data obtained from plant operation. The fuel used was natural gas of a low calorific value (LHV) of 39.4 MJ m^{-3}. The research investigated the effect of ambient temperature on the performance of the components of the gas turbine plant using the exergy concept, to identify components that offered significant work potential saving opportunities. The overall exergy loss was evaluated to be 83.47% and 84.56% for ambient temperature values of 295-307 K. Furthermore, a 1 ℃ increase in ambient temperature resulted in a 0.43 MW increase in the total irreversibility rate and a 0.3% decrease in exergy efficiency. Additionally, the results revealed that the combustion chamber was the highest exergy consumer; thus it offers the largest improvement potential. The authors suggested that chemical reactions between the air and fuel in the combustion process are responsible for such large exergy destruction in the combustion chamber. The second biggest consumer in exergy destruction is the turbine, with 8.19-8.99% for 295 to 307 K ambient temperatures.

Kakaras, E. [12] reported that the gas turbine output and efficiency is a strong function of the ambient air temperature. Depending on the gas turbine type, power output is reduced by between 5 to 10 percent of the ISO-rated power output (288K) for every 10 K increase in ambient air temperature. At the same time, the specific heat consumption increases by between 1.5 and 4 percent.

Lamfon, J.N. [13] investigated the performance of a combined system consisting of a gas turbine engine, a heat pipe recovery system and an inlet-air cooling system. The net power output improved by 11% when the gas turbine engine was supplied with cold air produced by the heat-pipe recovery and utilization system. Moreover, energy efficiency rose about 6% when the ambient temperature was decreased from 313K to 303K, accompanied by a 6% drop in specific fuel consumption.

Dawaud [14] presented results from the study of gas turbine plants in two locations in Oman. In both locations, fogged cooling is accompanied with 11.4 percent more electrical energy in comparison with evaporative cooling. However, absorption cooling offered 40 and 55 percent more energy than fogged cooling.

This literature review shows that higher temperatures in warm countries have negative effects on the performance of power plants. In an on-going study, the effects of summer conditions on the performance of a power plant in Abu Dhabi (UAE) will be examined. Unlike the previous research on power plants, the difference between the effects of the temperature of ambient air and its absolute humidity on the performance of a power plant will be investigated in the present work, using Hysys V8.6 with the Soave-Redlich-Kwong (SRK) equation. Finally, the performance of a power plant under design conditions will be simulated using Hysys V8.6 and be compared to the results of the exergy analysis for a typical summer day in Abu Dhabi (UAE).

IV. METHODOLOGY

A. Plant Description

As shown in Fig. 1, the main components of a typical Open Cycle Gas Turbine (OCGT) are: (1) air compressor, (2) combustor, (3) turbine, and (4) electricity generator. The air compressor and gas turbine are aligned on a single shaft connected to an electricity generator.

The power plant under investigation is designed to produce 160 MW, and is divided into three different control volumes (Compressor, Combustor and Turbine). (1) Fresh air entering the compressor at point 1 is compressed to a higher pressure. (2) Upon leaving the compressor, air enters the combustion system at point 2, where fuel is injected and combustion occurs. The combustion process occurs essentially at constant pressure. (3) The combustion mixture leaves the combustion system and enters the turbine at point 3. (4) In the turbine section, the energy of the hot gases at point 3' is converted into shaft power in the shaft power generator.

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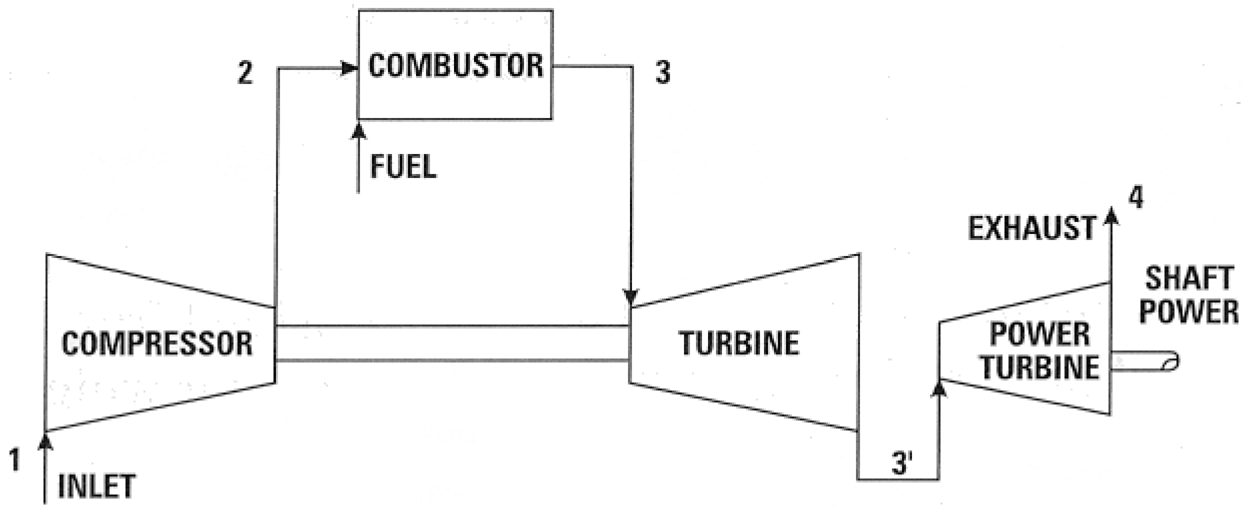

Fig. 1 Schematic Diagram of an Open Cycle Gas Turbine [15]

B. Ambient Air and Natural Gas Specifications

The power plant is located in Abu Dhabi (UAE) and the exergy analysis was conducted during summer time. The average atmospheric conditions during summer in Abu Dhabi were: P= 100.8 kPa, T= 316 K and RH= 50%, absolute humidity 0.03 kg.m-3 (The relative humidity (RH) is defined as the ratio of the water vapor's partial pressure to its saturated pressure at a particular temperature. On the other hand, the absolute humidity is the amount of water vapor present in a unit volume of dry air). Since fresh air is also the "dead state", its total exergy is equal to zero. An average mass flow rate m_{air} of fresh air is 484 kg/s and was measured at the entrance of the air compression section. Based on the given information and properties estimated using Hysys V8.6 with the Soave-Redlich-Kwong (SRK) equation of state, the specifications of fresh air are shown in Table 2.

TABLE 2 SPECIFICATIONS OF FRESH AIR

T (K)	P(kPa)	Flow (kg/s)	h (kJ/kg)	s (kJ/ kg.K)	ex_T (kJ/kg)
316	100.8	484	-352.2	5.49	0

Natural gas is used in the power plant under investigation. The composition, heating values and chemical exergy of the natural gas at standard conditions are given in Table 3.

TABLE 3 COMPOSITION AND PROPERTIES OF NATURAL GAS [3]

Component	Mass (%)	LHV (MJ/kg)	HHV (MJ/Kg)	e_{Ch} (MJ/kg)
CH_4	74	50.0	55.50	51.94
C_2H_6	13.1	47.80	51.90	50.0
C_3H_8	5.1	46.35	50.35	48.86
C_4H_{10}	3.2	45.75	49.50	48.27
N_2	4.1	0	0	0.05
CO_2	0.5	0	0	0.46
Total	100	47.1	52.02	48.93

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Based on equation (4) and using the values of chemical exergies from Table 3, the chemical exergy of the natural gas at standard conditions is equal to 48.93 MJ/kg. The average values of the mass flow rate, temperature and pressure of the natural gas entering the combustion chamber are respectively 9.6 kg/s, 303 K and 2510 kPa. The value (-287.6 kJ/kg) of the thermo-mechanical energy of natural gas is evaluated using the following equation:

$$ex_{fuel}^{TM} = (h - h_0) - T_0 (s - s_0) \tag{10}$$

From the given data, the specifications of the natural gas are given in Table 4.

TABLE 4 SPECIFICATIONS OF NATURAL GAS

T (K)	P(kPa)	Flow (kg/s)	h (kJ/kg)	s (kJ/ kg.K)	ex_T (kJ/kg)
303	2510	9.6	-4053.36	8.94	48642.5

C. Compression Section

The compressor of the power plant is an axial compressor with 21 stages and the compression pressure ratio is 13.5. The average values of the measured temperature and pressure of the compressed air were respectively 714.2 K and 1361 kPa. Since the chemical exergy of air is equal to zero, the total exergy of the compressed air is calculated using equation (10). The specifications of the compressed air are given in Table 5.

TABLE 5 SPECIFICATIONS OF COMPRESSED AIR

T (K)	P(kPa)	Flow (kg/s)	h (kJ/kg)	s (kJ/ kg.K)	ex_T (kJ/kg)
714.2	1361	484	79.10	5.607	394.3

According to the simulation results from Hysys V8.6, the values of the work W_K needed for the compression and the compressor isentropic efficiency are 2.088x 10^5 kW and 0.8, respectively. Based on Fig. 1, the exergy destruction $(ED)_K$ and exergy efficiency $(EE)_K$ of the compression section were estimated using the following equations:

$$(ED)_K = W_K - m_{air} (ex_{T2} - ex_{T1}) \tag{11}$$

$$(EE)_K = 1 - \frac{(ED)_K}{W_K} \tag{12}$$

Using the collected data and equations (11) and (12), the exergy destruction (ED) in the compressor and its exergy efficiency $(EE)_K$ are 17445.31 kW and 0.916, respectively.

D. Combustion Section

The combustion chamber is equipped in ring with 72 environmental burners. The average temperature of the flue gas entering the turbine is 1480 K. The thermo-mechanical energy of the flue gas can be evaluated using equation [16]:

$$ex_3^{TM} = C_{P,fg} \left[(T_3 - T_0) - T_0 \left(\ln\frac{T_3}{T_0} - \frac{R_{av}}{C_{p,fg}} x \ln\frac{P}{P_0} \right) \right] \tag{13}$$

T_3 is the temperature of the flue gas entering the gas turbine (1480 K) and T_0 is the temperature of the ambient air (316K) in the “dead state”. The value of $C_{p,fg}$ (1.275 kJ/Kg.^{0}C) was estimated using Hysys V8.6. The thermo-mechanical exergy of the flue gas is 1110.4 kJ/kg. According to equation (4), the chemical exergy of the flue gas is 23.43 kJ/kg. The specifications of the flue gas are given in Table 6.

TABLE 6 SPECIFICATIONS OF THE FLUE GAS

T (K)	P(kPa)	Flow (kg/s)	h (kJ/kg)	s (kJ/ kg.K)	ex_T (kJ/kg)
1480	1361	493.6	-233.13	6.92	1133.8

The exergy destruction in the combustion chamber is estimated using the equation:

$$(ED)_{cc} = m_{air} x\, ex_{T2} + m_{fuel} x\, ex_{fuel} - m_{fg}.ex_{T3} \tag{14}$$

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The exergy efficiency of the combustion chamber is calculated by the following equation:

$$(\mathrm{EE})_{\mathrm{cc}} = 1 - \frac{(\mathrm{ED})_{\mathrm{cc}}}{\mathrm{m_{air}x\ ex_{T2}} + \mathrm{m_{fuel}x\ ex_{fuel}}} \tag{15}$$

The exergy destruction in the combustion chamber is equal to 109271.52 kW. Based on the data collected and equation (15), the exergy efficiency of the combustion chamber is 0.826.

E. Expansion Section

The expansion section is composed of a 5-stage gas turbine and a generator. The average temperature of the exhaust gas leaving the power turbine (4) is 886 K and the deduced turbine isentropic efficiency is 0.88. Using Hysys V8.6, the power generated by the turbine and the power needed by the compressor are 380 MW and 208.8 MW, respectively. Assuming that the mechanical efficiency of the turbine and compressor is equal to 0.98 and the generator efficiency is equal to 0.99, the net output of the power plant 155.61 MW is the difference between the power output of the turbine and the power needed for air comprssion. The value of the thermo-mechanical exergy of the exhaust gas, 314.01 kJ/kg, is calculated using equation (10), and the specifications of the compressed air are shown in Table 7.

TABLE 7 SPECIFICATIONS OF THE EXHAUST GAS

T (K)	P(kPa)	Flow (kg/s)	h (kJ/kg)	s (kJ/ kg.K)	ex_T (kJ/kg)
886	100.8	493.6	-1003	7.044	337.44

The values for the exergy destructions (24866.34 kW) and exergy efficiency (0.937) of the expansion section are calculated using the following equation:

$$(\mathrm{ED})_{\mathrm{T}} = \mathrm{m_{fg}x\ (ex_{T3} - ex_{T4}) - W_T} \tag{16}$$

$$(\mathrm{EE})_{\mathrm{T}} = 1 - \frac{(\mathrm{ED})_{\mathrm{T}}}{\mathrm{m_{fg}x\ (ex_{T3} - ex_{T4})}} \tag{17}$$

F. Analysis of the Gas Turbine

Based on equation (9), the exergy efficiency of the plant is 0.331. The exergy destruction in the power plant's equipment and its exergy efficiency are shown in Table 8.

TABLE 8 RESULTS OF THE EXERGY ANALYSIS OF THE POWER PLANT AT 316K

Equipment	Exergy destruction (kW)	% Exergy destruction	Exergy efficiency
Compressor	17445.31	12.4	0.916
Combustion chamber	98155.6	70.2	0.851
Turbine	24417.2	17.4	0.938
Power plant	140018.1	100	0.331

Similar to results by some authors [6, 8], Table 8 indicates that the combustion process is the main source (72.1%) of the exergy destruction rate in the power plant, and the compression section contributes the least (11.5%) in its total exergy destruction.

V. EFFECTS OF AIR AMBIENT ON THE PERFORMANCE OF THE POWER PLANT

The performance of gas turbine power generation plants varies according to ambient conditions. Higher temperatures greatly affect electricity production and exergy efficiencies. The second purpose of the present study is to use Aspen Hysys V8.6 with the Soave-Redlich-Kwong (SRK) equation of state in order to simulate the effects of a decrease in ambient temperature from 316K to 306K, with the same absolute humidity of 0.003 kg.m^3, on the exergy destruction in all the plant's equipment and the net power plant's output. The effects of decreasing ambient temperature on the exergy destruction in each section of the process and their exergy efficiencies are shown in Tables 9 and 10.

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TABLE 9 RESULTS OF THE EXERGY ANALYSIS OF THE POWER PLANT AT 311K

Equipment	Exergy destruction (kW)	% Exergy destruction	Exergy efficiency
Compressor	17203.0	11.3	0.918
Combustion chamber	95551.7	62.6	0.855
Turbine	39890.4	26.1	0.902
Power plant	152645.1	100	0.333

TABLE 10 RESULTS OF THE EXERGY ANALYSIS OF THE POWER PLANT AT 306K

Equipment	Exergy destruction (kW)	% Exergy destruction	Exergy efficiency
Compressor	16350.4	10.4	0.919
Combustion chamber	86895.8	54.9	0.866
Turbine	54952	34.7	0.888
Power plant	158198.2	100	0.336

Based on Tables 8-10, Figs. 2-4 show the effects of decreasing temperature on the exergy destruction and efficiency of each equipment, the net power output and the exergy efficiency of the power plant.

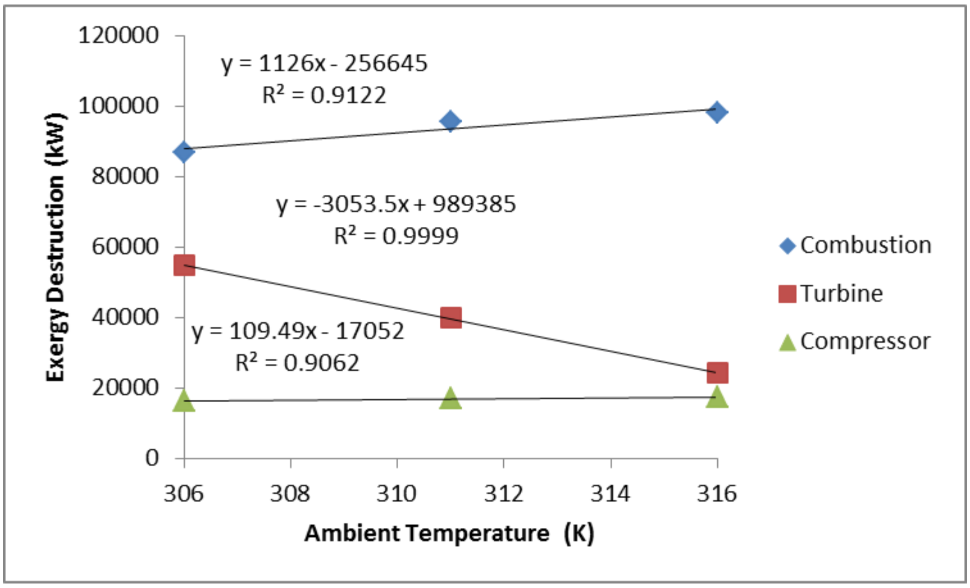

Fig. 2 Effects of ambient temperature on the exergy destructions

According to Fig. 2, there is a linear relationship between the air temperature and the exergy destruction in the plant's main equipment. Increasing the temperature of ambient air will increase the exergy destruction in the combustion chamber and decrease the exergy destruction in the turbine. An increase in air temperature also causes a small increase in the exergy destruction of the compressor.

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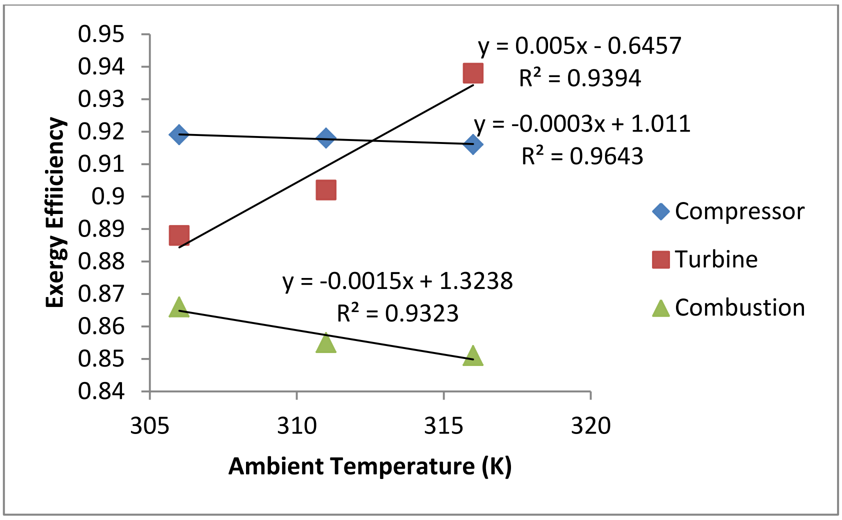

Fig. 3 Effects of ambient temperature on the exergy efficiencies

According to Fig. 3, increasing the air temperature will result in a large decrease in the exergy efficiency of the combustion chamber and a small decrease in the compressor section, but it will highly increase the exergy efficiency of the turbine.

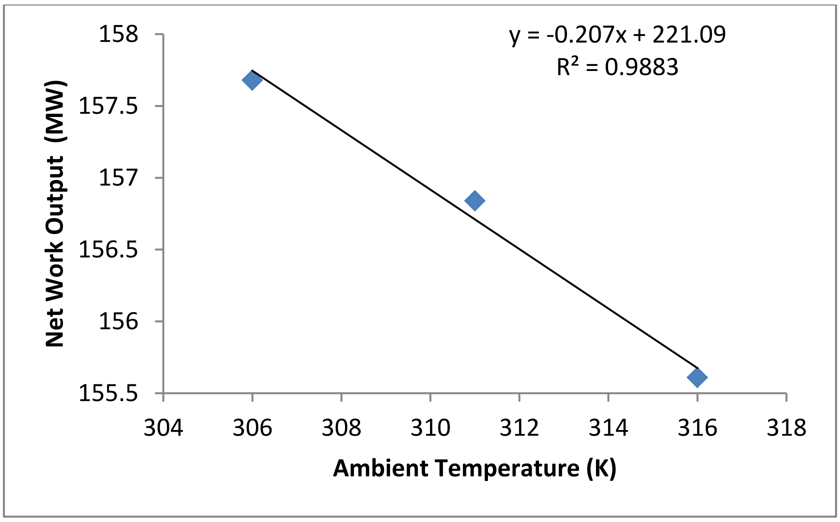

Fig. 4 Effects of ambient temperature on the net output of the plant

Fig. 4 shows a linear relationship between the ambient air temperature and the net power output. A 10 K increase in the ambient temperature from 306K to 316K will result in a 1.31 % decrease of the net power output.

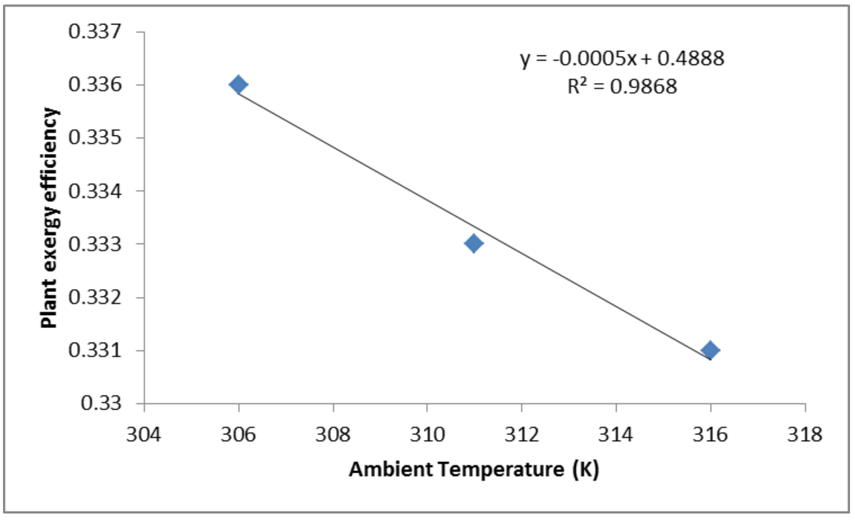

Fig. 5 Effects of ambient temperature on the exergy efficiency of the plant

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According to Fig. 5, increasing the ambient temperature linearly decreases the exergy efficiency of the plant. A 10K increase in the ambient temperature (from 306K to 316K) will result in a 1.49% decrease of the exergy efficiency of the plant.

VI. EFFECTS OF TEMPERATURE AND HUMIDITY ON THE PERFORMANCE OF THE POWER PLANT

The relative humidity (RH) in Abu Dhabi (UAE) typically ranges from 21% (dry) to 89% (very humid) over the course of the year, rarely dropping below 9% (very dry) and reaching as high as 100% (very humid during December). With an absolute humidity of 0.03 $kg.m^{-3}$ and an atmospheric pressure in Abu Dhabi (UAE), water from the air will condense in the compressor at temperatures below 303.8 K. In this third part of our investigation, Hysys V8.6 is utilized to study the effects of decreasing the temperature of a saturated air ambient (RH=100%) from 301K to 290K. Table 11 shows the results of the exergy analysis of the power plant at the ambient air temperature of 301K and a maximum absolute humidity of 0.0269 $kg.m^{-3}$. Table 12 shows the results of the exergy analysis of the power plant at an ambient air temperature of 296 K and a maximum absolute humidity of 0.0203 $kg.m^{-3}$. Finally, Table 13 shows the results of the exergy analysis of the power plant at the ambient air temperature of 290 K and a maximum absolute humidity of 0.0142 $kg.m^{-3}$.

TABLE 11 RESULTS OF THE EXERGY ANALYSIS OF THE POWER PLANT AT 301K (ABSOLUTE HUMIDITY= 0.0269 KG.M-3)

Equipment	Exergy destruction (kW)	% Exergy destruction	Exergy efficiency
Compressor	16593	11.3	0.916
Combustion chamber	90114	61.2	0.862
Turbine	40416	27.5	0.90
Power plant	147123	100	0.339

TABLE 12 RESULTS OF THE EXERGY ANALYSIS OF THE POWER PLANT AT 296 K (ABSOLUTE HUMIDITY = 0.0203 KG.M-3)

Equipment	Exergy destruction (kW)	% Exergy destruction	Exergy efficiency
Compressor	15979.6	11.3	0.917
Combustion chamber	84755.49	59.9	0.869
Turbine	40699	28.8	0.898
Power plant	141434.09	100	0.342

TABLE 13 RESULTS OF THE EXERGY ANALYSIS OF THE POWER PLANT AT 290 K (ABSOLUTE HUMIDITY = 0.0142 KG.M-3)

Equipment	Exergy destruction (kW)	% Exergy destruction	Exergy efficiency
Compressor	14939	10.7	0.920
Combustion chamber	80922.7	58.2	0.874
Turbine	43295	31.1	0.91
Power plant	139156.7	100	0.346

Data from Tables 11-13 are utilized to investigate the effects of both the ambient air's temperature and absolute humidity on the exergy destruction of the plant's different equipment and their exergy efficiency.

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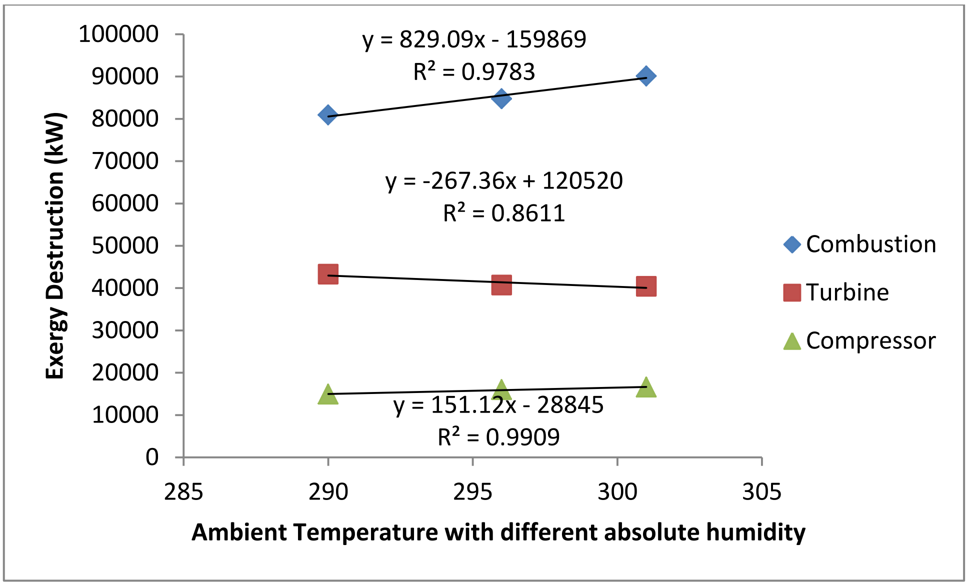

Fig. 6 Effects of temperature and absolute humidity on the exergy destructions

According to Fig. 6, increasing the air temperature and absolute humidity increases the exergy destruction in the combustion chamber. However, the exergy destruction in the compressor increases only slightly with temperature and absolute humidity. On the other hand, exergy destruction in the turbine decreases slightly with temperature and absolute humidity.

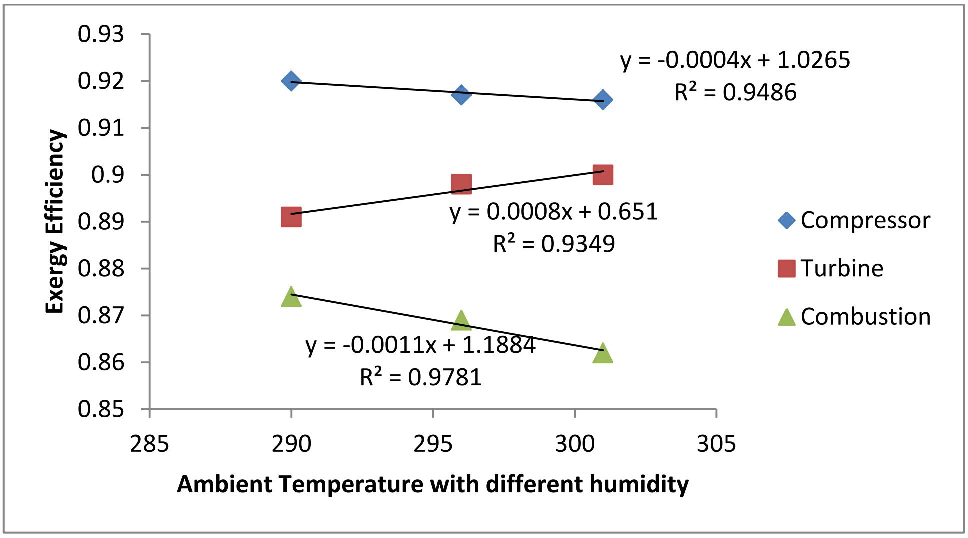

Fig. 7 Effects of temperature and absolute humidity on exergy efficiencies

According to Fig. 7, increasing the air temperature and absolute humidity decreases the exergy efficiency of the combustion chamber and compressor. On the other hand, the exergy efficiency of the turbine increases with the air temperature and absolute humidity.

The third stage of this investigation is to utilize Hysys V8.6 to conduct an exergy analysis of the power plant at design conditions (T=288K and RH=60%) with an absolute humidity of 0.008 kg.m^{-3}. Table 14 shows the comparison between the effects of ambient air at design conditions and during a typical summer day (T=316K and RH=50%) in Abu Dhabi (UAE) on the exergy destruction of all the equipment and its exergy efficiency.

TABLE 14 COMPARATIVE ANALYSIS ON THE POWER PLANT

Equipment	Exergy Destruction (kW)		Exergy Efficiency (%)	
	Design conditions	Summer conditions	Design conditions	Summer conditions
Compressor	15437.42	17445.31 (+13%)	0.916	0.916 (equal)

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Combustion chamber	82234.25	98155.6 (+19.4%)	0.872	0.851 (-2.4%)
Turbine	40760	24417.2 (-40.1%)	0.89.6	0.938 (+4.7%)
Power plant	138431.67	140018.1 (+1.1%)	0.347	0.331 (-4.6%)

Table 14 shows that the summer conditions increase the exergy destruction in the combustion chamber and the compressor by 19.4% and 13%m respectively. However, there is a decrease of 40.1% in the exergy destruction in the turbine. The summer conditions decreased the exergy efficiency of the combustion chamber and increased the exergy efficiency of the turbine. However, there is no effect on the exergy efficiency of the compressor. The final stage of this third investigation is to analyze the effects of ambient air temperature and absolute humidity on the net power output and the exergy efficiency of the power plant.

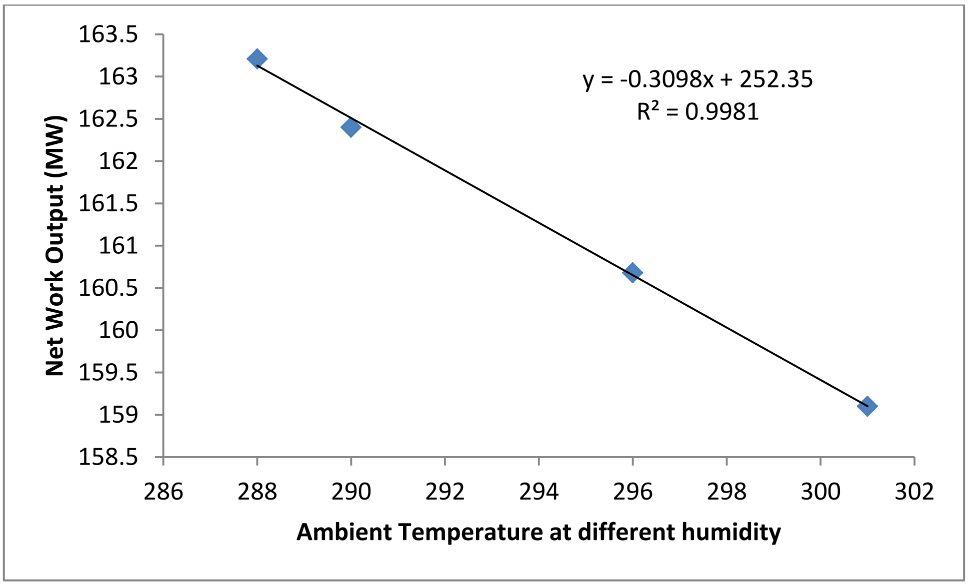

Fig. 8 Effects of ambient temperature and absolute humidity on the net output

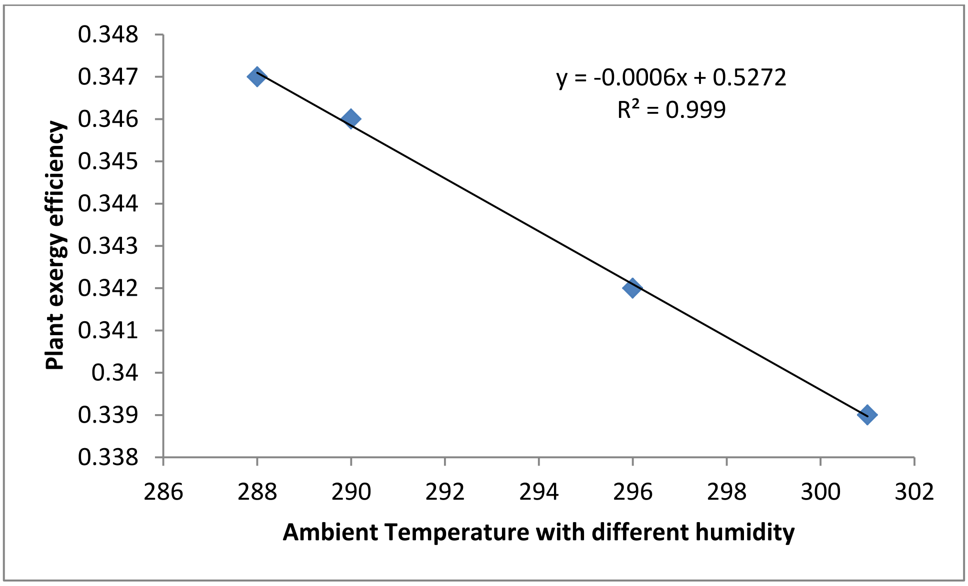

Fig. 9 Effects of ambient temperature and absolute humidity on the exergy efficiency of the plant

As indicated by other authors [5], Figs. 8 and 9 indicate that both net power output and exergy efficiency of the power plant decrease linearly with temperature and absolute humidity.

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VII. DISCUSSION

When comparing the slopes in Fig. 2 and Fig. 6 (109.49 and 151.12), both the temperature of ambient air and its absolute humidity increase slightly with the exergy destruction in the compressor. Based on equation (11), these results could be explained by the fact that the compressor requires more work (W_K) to compress air of a higher temperature because air density is lower. Moreover, due to molecular weight, air absolute humidity decreases the density of air; by consequence, more work is needed to compress the humid air.

Because the slope (-3053.5) of the effects of ambient air on the exergy destruction in the turbine is higher than the slope (-267.36) of the corresponding curve in Fig. 6, it could be concluded that the exergy destruction in the turbine decreases with temperature but increases with air absolute humidity. However, the effects of temperature could be more important. Equation (16) shows that, the findings are due to the fact that the work output (W_T) of the turbine increases with the compressor inlet temperature and decreases with air humidity.

Finally, comparing the slopes (1126 and 829.09) related to the exergy destruction in the combustion chamber, the temperature of ambient air increases the exergy destruction of the combustion but humidity decreases it with smaller effects. According to equation (14), the exergy destruction in the combustor increases with the exergy of the compressed air. Both the temperature of air and its absolute humidity increases the exergy of the compressed air; however, humidity has a cooling effect as well, so it could increase the efficiency of combustion.

Comparing Fig. 3 with Fig. 7, the slope related to combustion (-0.0015) is higher in Fig. 3 than the corresponding slope (-0.0011) in Fig. 7, suggesting that temperature decreases the exergy efficiency of combustion but absolute humidity increases it. For the exergy efficiency of the compressor, a comparison between the slope (-0.0003) of Fig. 3 and the slope (-0.0004) of Fig. 7 indicates that the temperature of ambient air slightly decreases the exergy efficiency of the compressor but the absolute humidity increases it. The results related to the exergy efficiency of the turbine (0.005 higher than 0.0008) indicate that temperature increases the exergy efficiency of the turbine but absolute humidity decreases it.

Figs. 4 and 8 show that, both the temperature and humidity of ambient air decrease the net-work output of the power plant but the effects of temperature are more important. Finally, Figs. 5 and 9 indicate that the exergy efficiency of the power plant decreases with increasing air temperature and humidity, but temperature has much stronger effects. These results could be explained by the fact that the negative effects of ambient air temperature on the compressor are higher than its positive effects on the turbine.

Exergy analysis of the power plant during summer time (Table 14) shows that the summer conditions had no effect on the exergy efficiency of the compressor (the negative effects of higher temperatures could be cancelled by the positive effects of higher absolute humidity), they decreased by 2.4% the exergy efficiency of the combustion chamber (the negative effects of high temperature are more important than the positive effects of higher absolute humidity) but increased by 4.7% the exergy efficiency of the turbine (due to the important positive effects of high temperature).

Finally, the results related to the power plant's net power output and exergy efficiency at summer conditions are respectively 155.61 MW and 33.1 %, respectively. For the design conditions (T=288K, RH=60%), the net output and exergy efficiency are 163.21 and 34.7%, repectively. As a consequence, due to the summer conditions in Abu Dhabi (UAE), the power plant lost 4.66 % of its net power output at a rate of 0.27 MW per degree Kelvin. The power plant also lost 4.61 % of its exergy efficiency at a rate of 0.06% per degree Kelvin.

VIII. CONCLUSION

The first objective of the present investigation was to conduct an exergy analysis of a power plant in Abu Dhabi (UAE). As mentioned in the literature, the combustor contributed the most to the total exergy destruction of the plant (70.2%) while the compressor had the lowest contribution at of 12.4%. Aspen Hysys V8.6 with the Soave-Redlich-Kwong (SRK) equation of state was then utilized to simulate the effects of high temperatures and absolute humidity during summer on the exergy destruction in different parts of the plant as well as on the plant's net-power output and exergy efficiency. For the overall performance of the power plant, both temperature and absolute humidity decreased the net-work output of the power plant. Regarding the exergy efficiency of the plant, both temperature and absolute humidity decreased it but temperature had more negative effects than the absolute humidity of air. From the design conditions, our results show that the power plant lost 7.6 MW (4.66 %) with a rate of 0.27MW per 1K. and 4.61% of its exergy efficiency. It was also found that irreversibilities in the combustion chamber increased with temperature but decreased with the absolute humidity of air. Irreversibilities in the turbine decreased with increasing temperature but increased with increasing air humidity. However, temperature had stronger effects than the absolute air humidity. Finally, irreversibilities in the compressor increased with both increase in the ambient air temperature and absolute.

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