EJECTOR INCORPORATION TO ENHANCE THE THERMODYNAMIC PERFORMANCE OF A 285MW GAS TURBINE UNIT

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الملخص

يعتبر ارتفاع درجة حرارة المحيط الجوي من العوامل المؤثرة سلبا على أداء محطات التربينات الغازية حيث تقل الكفاءة والقدرة المحصلة بشكل ملحوظ الأمر الذي قد يشكل تحديا للمشغلين لتوفير الطاقة للمستهلكين خصوصا عند الأحمال القصوى للشبكة.

يهدف هذا العمل إلى دراسة تحسين أداء محطة السرير الغازية ذات القدرة الأسمية 285 ميجاوات وذلك عن طريق حقن بخار الماء في غرفة الاحتراق باستخدام منفث، حيث يتم استخدام غازات العادم كمصدر لتوليد بخار الماء. ولغرض الدراسة، تم اختيار درجة حرارة محيطية 15 درجة سيلسيس (ايزو) و 50 درجة سيلسيس (المناطق الحارة).

بينت النتائج انه هناك قيمة مثلى لنسبة الجر تكون عندها الكفاءة ذات قيمة قصوى والصرف النوعي للوقود ذا قيمة دنيا.

ABSTRACT

The performance of gas turbines, mainly the output power and efficiency, are affected considerably by the ambient temperature, where, they deteriorate as the ambient temperature rises. This could be a challenge especially for periods of peak load demand.

One way to enhance the performance of gas turbine power plants is by injecting steam into the combustion chamber. In this work, an ejector is integrated to inject steam into the combustion chamber of a gas turbine power unit. The exhaust gases of the gas turbine are employed as the energy source to generate the steam. For the analysis, two environmental temperatures are selected, 15ºC (ISO condition) and 50ºC (hot climate zone).

The aim of the current work is to enhance the thermodynamic performance of an existing 285MW gas turbine unit. Results show a specific value for the entrainment ratio at which the thermal efficiency and hence the specific fuel consumption emerge optimum values.

KEYWORDS: Gas turbine; Ejector; Entrainment ratio; Steam injection; Performance enhancement.

INTRODUCTION

The rousing request for gas turbines stimulates the investigations on their performance progress. Even small increases in net power or thermal efficiency have become major anxieties for both new designs and cycle reforms [1].The efficiency and net power output of a gas turbine fluctuate with ambient conditions. The net power output of a gas turbine is related to the mass flow rate of the compressed air submitted by the compressor.

Both air compressor and gas turbines have a fixed volumetric flow rate of air for a specified turning speed, however, the air mass flow rate decreases with increasing ambient temperature. By increasing the ambient temperature beyond 15°C, the power output of the gas turbine decreases beneath its rated value at the ISO (International Organization for Standardization) conditions (15°C and 101.3 kPa at sea level). An inlet air temperature rises of 1°C decreases the power output by 1%. This consequence is of anxiety to power producers, causing in the advance of many procedures to improve gas turbine performance [2].

At high ambient temperatures, a power loss of more than 20%, combined with a substantial increase in specific fuel consumption, compared to ISO standard conditions may be detected [3]. R. Hosseini [4] reported that the net power output of gas turbine units decreases by 0.50-0.90% when the ambient temperature increases by 1.0ºC.

To rise thermal power generation efficiency, there is an ongoing determination to increase the cycle's upper temperature. While the temperature of combustion reaches 1700°C, most of the power generated in the world is by steam power cycles, where the maximum steam temperature is around 560°C, while the maximum temperature of gas turbines may reach 1200°C. These temperature restrictions are sustained due to the combined effects of temperature, pressure, and dynamic forces on the cost and life of the plant [5].

During the last years, gas turbine efficiencies were effectively enhanced by raising the compressor pressure ratio and turbine inlet temperature (TIT). Developments in cooling technology and material science made high TIT promising [6].

There are many attempts to improve the thermodynamic performance of gas turbine power plants, among which, cooling the intake air and steam injection. Fifi N.M. Elwekeel [7] studied the effect of mist cooling technique on exergy and energy analysis of steam injected gas turbine cycle. The results show that the efficiency can reach 47.2% at low coolant temperature with a mist fraction of 2%.Enhancing gas turbine performance by intake air cooling using an absorption chiller was introduced by B. Mohant [8]. The analysis was performed by taking the weather data from Bangkok (Thailand). The results show that dropping the temperature from ambient condition to 15°C may raise the power output between 8 and 13%.Yousef S. H. Najjar [9] studied the enhancement of performance of gas turbine engines by inlet air cooling and cogeneration system. Results show that the combined system achieves gains in power, efficiency and specific fuel consumption, of about 21.5, 38 and 27.7% respectively. A model to study the effect of inlet air-cooling on gas turbines power and efficiency was developed for two different cooling techniques, direct mechanical refrigeration and an evaporative water spray cooler [10]. The results show that the direct mechanical refrigeration increased the daily power output by 6.77% versus 2.57% for the spray air-cooling.

An extensive review of the various combustion turbine inlet cooling technology options open to Saudi Electric Company's has been made, and their key benefits and drawbacks in relation to the environmental conditions and generational requirements of Saudi Arabia have been identified [11].

Performance Comparison between Steam-Injected Gas Turbine and Combined Cycle during Frequency Drops was studied by Saeed Bahrami et al. [12]. The simulation results show that the steam injected gas turbine has better performance during frequency drops and it can handle relatively larger change loads. The enhancement of thermal efficiency and net power output of a simple gas turbine power plant was studied by M. De Paepe and E. Dick [13].They reported that the increase in efficiency was due to the restoration of the amount of heat in the exhaust gas, and the increase in specific power was a result of the additional mass flow through the turbine.

The effects of ambient conditions variations, such as ambient temperature and relative humidity on gas turbine unit performance at different loads were studied by I. Elfeituri [14] For this work, the power plant chosen was a gas turbine unit of 285 MW installed capacity. The results obtained showed that a degree (1°C) increase in ambient temperature, leads to 0.53% decrease in the net power output, 0.18 %decrease in the thermal efficiency and 0.195 % increase in specific fuel consumption.

The influence of ambient temperature variations on gas, steam, and combined cycles performance was simulated by G. Fellah [15]. A real-time data including ambient temperature, air mass flow rate, pressure ratio, and fuel flow rate, for the selected gas unitwere obtained at full load operation. The results of the analysis showed that for the combined cycle at an ambient temperature of 15ºC, the first and second law efficiencies raised to 42.80% and 40.20% respectively, and the percentage gain in power was 53.46%. At an ambient temperature of 40ºC, the results show that the first and second law efficiencies dropped to 41.11% and 38.60% respectively, and the percentage gain in power rises to 60.98%.

SCOPE OF THE CURRENT WORK

In this work, an ejector is incorporated to inject steam into the combustion chamber of the gas turbine power unit. The aim is to determine the effect of the entrainment ratio on the thermodynamic performance of the proposed power cycle. The technical and economic aspects are not considered in the analysis.

The gas turbine power plant which is used as a case study for this researchlocated in the SARIR power station, in Libya. The SARIR power station consists of three units, Siemens gas turbines, type SGT5-PAC 4000F, each unit with 285 MW rated capacity[14].

THE PROPOSED CYCLE

The main components of the modified cycle are shown in Figure (1). It consists of a compressor, ejector, combustion chamber, gas turbine and a heat exchanger (Heat recovery steam generator). The heat exchanger is employed to generate saturated steam by restoring energy from the exhaust gases. The compressed air from the compressor is the motive fluid to entrain the saturated steam into the ejector. The mixed fluid (air and steam) is injected directly into the combustion chamber.

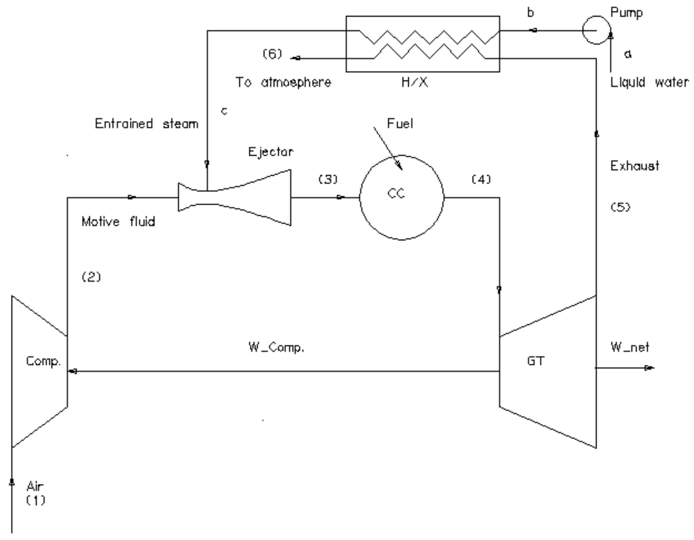

Figure 1: The gas turbine power unit with ejector

The main design data of the original unit is tabulated in the Table (1).

Table 1: Design parameters

Component	Parameter	Value	Unit
Compressor	Inlet temperature	15	[ºC]
	Inlet pressure	101.3	[kPa]
	Pressure ratio	17.5	[-]
	Air mass flow rate	672	[kg/s]
	Air volume flow rate	550.5	[m^3/s]
Combustion Chamber	Pressure loss	30	[kPa]
	Outlet temperature	1242	[ºC]
Gas turbine	Inlet pressure	1743	[kPa]
	Inlet temperature	1242	[ºC]

The efficiencies of the compressor and turbine are taken as 87.5 and 89.29 respectively. The fuel is assumed to be methane with a lower heating value of 50,035 kJ/kg.

METHODOLOGY AND TOOLS

Throughout the analysis, the turbine inlet temperature, the volume flow rate of the intake air and the pressure ratio are fixed at the design conditions of 1242ºC, 550.5 m^3/s and 17.5 respectively. For each ambient temperature, the air flow rate to the compressor is calculated. The fuel flow rate is altered to keep the maximum cycle temperature at 1242ºC.

The first part of the analysis is oriented to find the effect of the ambient temperature on the thermodynamic performance of the original gas turbine unit (without steam injection). The mass flow rate of the intake air is calculated at each ambient temperature between 15ºC (ISO) and 50ºC (hot climate zone). Fuel flow rate, heat supply, power, and thermal efficiency are calculated at each ambient temperature.

In the second part of the analysis, a heat exchanger is incorporated to restore part of the exhaust gas's energy, where water is pumped to 1.10 bar and then heated in the heat exchanger to a saturated water vapor condition. Both streams, the motive stream (air) and the entrained stream (steam) enter an ejector. The outlet mixture (air and steam) of the ejector is injected directly into the combustion chamber. Two ambient temperatures are selected for the analysis, 15ºC (ISO) and 50ºC (hot climate condition). Steam flow rates vary from 10 kg/s to 100 kg/s, during which fuel flow rate is modified to keep the maximum temperature at 1242ºC. The effect of the entrainment ratio (mass of the motive air per mass of the steam) on the thermodynamic performance is analyzed.

Steady-state, steady flow is assumed, with negligible change in kinetic and potential energy. The conservation of mass and energy are applied, where:

$$\sum_{i} \dot{m}_i = \sum_{e} \dot{m}_e \tag{1}$$

$\sum_i \dot{m}_i$ and $\sum_e \dot{m}_e$, are the summation of the inlet and outlet flow rates, respectively.

$$\dot{Q}_{cv} + \sum_{i} (\dot{m}h)_i = \sum_{e} (\dot{m}h)_e + \dot{W}_{cv} \tag{2}$$

$\dot{Q}_{cv}$ and $\dot{W}_{cv}$, are the net heat and power flow across the control volume (cv), respectively.

To calculate the mass flow rate of air we may use the ideal gas equation of state:

$$\dot{m} = P\dot{V}/ RT \tag{3}$$

Where: P is the pressure (kPa), $\dot{V}$ is the volume flow rate (m^3/s), $\dot{m}$ is the mass flow rate (kg/s), R is the air constant (0.287 kJ/kg.K) and T is the temperature in Kelvin.

The specific fuel consumption (sfc) is defined as:

$$sfc = \frac{\dot{m}_f}{\dot{W}_{net}} \tag{4}$$

Where: $\dot{m}_f$ is the fuel mass flow rate and $\dot{W}_{net}$ is the net power output.

The entrainment ratio is defined as:

$$\text{Entrainment ratio} = \frac{\text{Motive fluid}(air)}{\text{Entrained fluid (steam)}} = \frac{\dot{m}_{air}}{\dot{m}_{steam}} \tag{5}$$

The pump power is given by:

$$\dot{W}_p = \eta \dot{V} \Delta P \tag{6}$$

Where η is the pump efficiency (taken equal to75%), $\dot{V}$ is the volumetric flow rate (m^3/s) of the injected water and ΔP is the pressure difference across the pump.

The Carbon dioxide emission factor (F) is calculated as:

$$F \left[\frac{tCO_2}{MWh}\right] = \frac{Emitted\ carbon\ dioxide\ \left[\frac{t}{h}\right]}{Net\ power\ output\ [MW]} \tag{7}$$

The temperature at the turbine outlet is estimated from:

$$T_4 = T_3 - \frac{\dot{W}_{turbine}}{\dot{m}_4 c_P} \tag{8}$$

c_P is the specific heat of the exhaust gases.

RESULTS AND DISCUSSION

In the first part of the analysis, the performance of the original gas power unit without steam injection is obtained and tabulated in Table (2). It is found that, as the ambient temperature increases from 15 to 50ºC, the specific fuel consumption increases by 0.319g/kWh/ºC, the net power output drops by 1.81 MW/ºC (22.16% reduction) and the thermal efficiency falls by 0.0355 %/ºC, the results are in a fair agreement with that given in the literature[3,4,14,15].The reduction in the fuel flow rate and the drop in net power output have an adverse effect on the carbon dioxide emission factor, the results show a slight rise in the carbon dioxide emission factor.

Table 2: The effect of the ambient temperature on the performance of the gas power unit

Temp. [ºC]	mass [kg/s]		QH [MW]	Power [MW]			Eff.	SFC [g/kWh]	F [tCO2/ MWh]
	Air	Fuel		Comp.	Turb.	Net			
15	672.00	19.83	992	280.32	565.51	285.19	0.2874	250.46	0.5150
20	660.54	19.25	963	280.21	555.41	275.20	0.2858	251.89	0.5180
25	649.46	18.69	935	280.16	545.64	265.48	0.2840	253.38	0.5213
30	638.75	18.14	908	279.99	536.19	256.20	0.2823	254.91	0.5245
35	628.38	17.61	881	279.82	527.05	247.23	0.2805	256.50	0.5277
40	618.35	17.11	856	279.66	518.20	238.54	0.2787	258.15	0.5311
45	608.63	16.61	831	279.49	509.63	230.14	0.2769	259.86	0.5346
50	599.22	16.13	807	279.33	501.33	222.00	0.2750	261.64	0.5383

In the second part of the analysis, a heat exchanger and an ejector are added. Two environmental temperatures are selected for the analysis 15ºC (ISO condition) and 50ºC (hot climate zone). Figure (2) shows the effect of the entrainment ratio on the fuel flow rate.

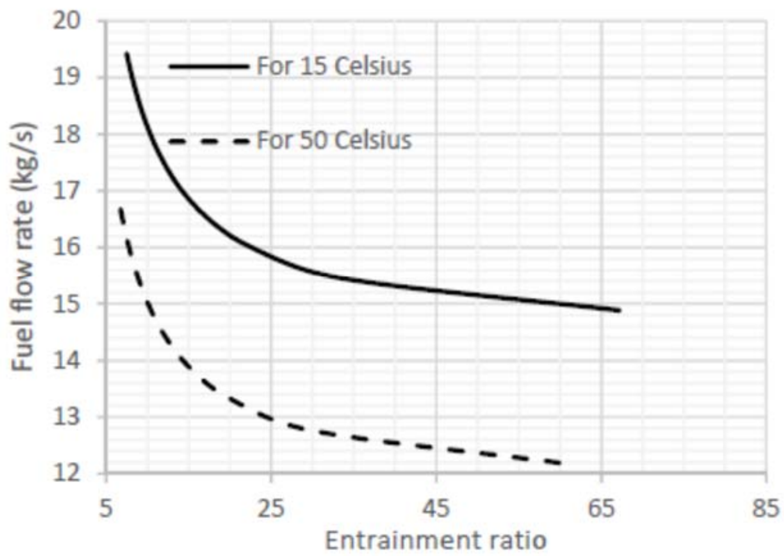

Figure 2: The effect of the entrainment ratio on the fuel flow rate

The temperature of the entrained saturated steam (around 100ºC) is considerably lower than the temperature of the motive air, and hence the temperature of the air at the combustion chamber inlet rises with the increase in the entrainment ratio (i.e. with a reduction in the steam flow rate). To keep the turbine inlet temperature constant at

1242ºC, the fuel flow rate (and hence the supplied heat) declines with the increase in the entrainment ratio.

The effect of fuel flow rate on the net power output is shown in Figure (3). The maximum net power output increases by **11.2%** and **16.65%**above the original cycle for 15ºC and 50ºC ambient temperature, respectively. These findings verify the advantageous of the modified cycle over the original one.

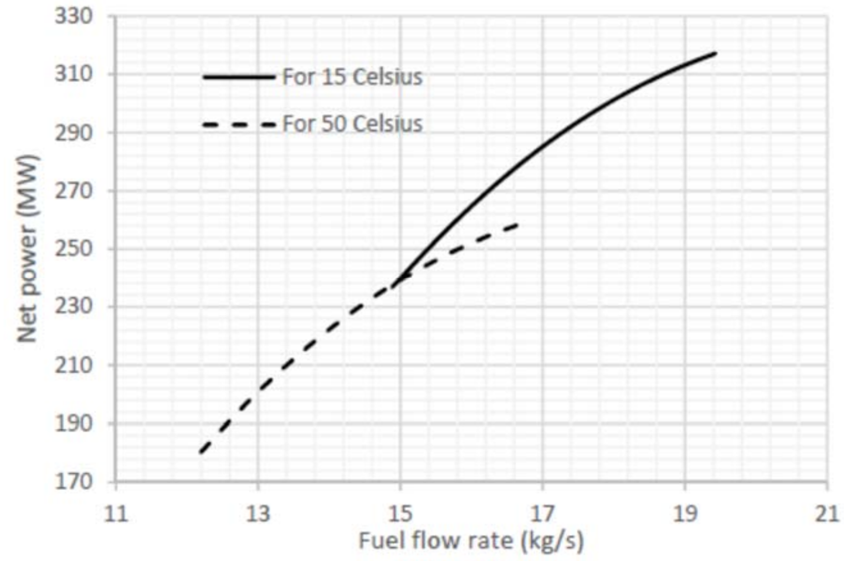

Figure 3: The effect of fuel flow rate on the net power output

The power output of the gas turbine (and hence the net power output) depends on the mass flow rate of the exhaust gases and its temperature at the turbine outlet (since the inlet temperature is kept constant). Both of them drop with the increase in the entrainment ratio, as shown in Figures (4, 5).

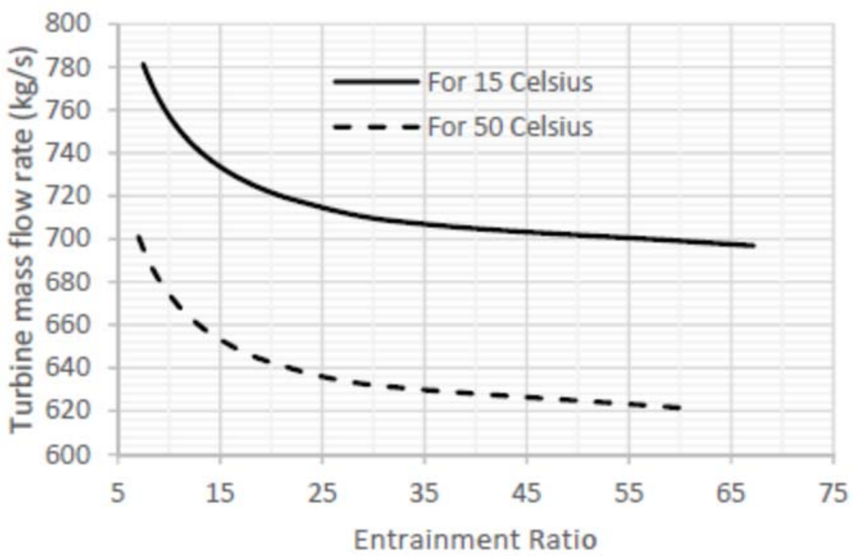

Figure 4: The effect of the entrainment ratio on the mass flow rate across the gas turbine

The drop in the mass flow rate is due to the reduction in the entrained flow. The variation of the exhaust temperature is controlled by the ratio $\frac{\dot{W}_{turbine}}{\dot{m}_4 c_P}$ as given by equation (8).

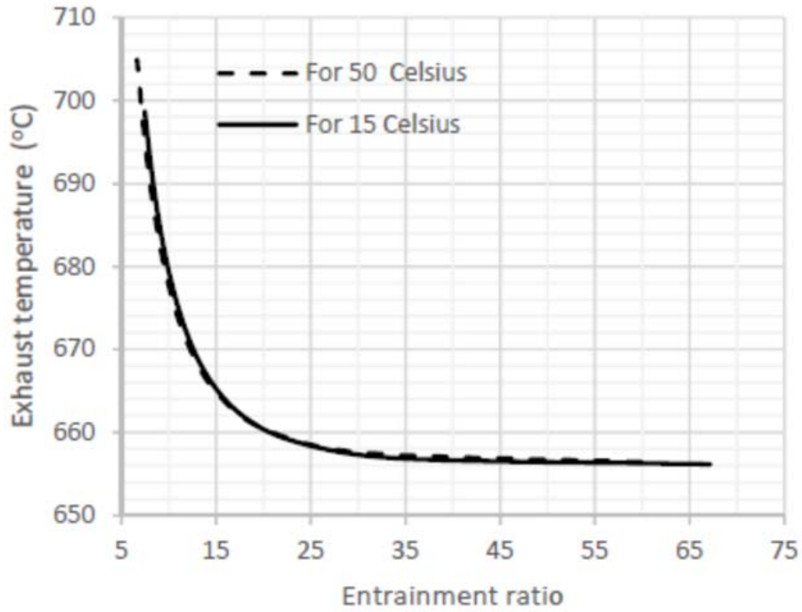

Figure 5: The effect of the entrainment ratio on the exhaust temperature

The drop in the mass flow rate across the gas turbine and the fall in the exhaust temperature have an adverse effect on the net power output. The impact of those two parameters on the net power output is depicted in Figure (6), where a drop in the net power occurs with the increase in the entrainment ratio.

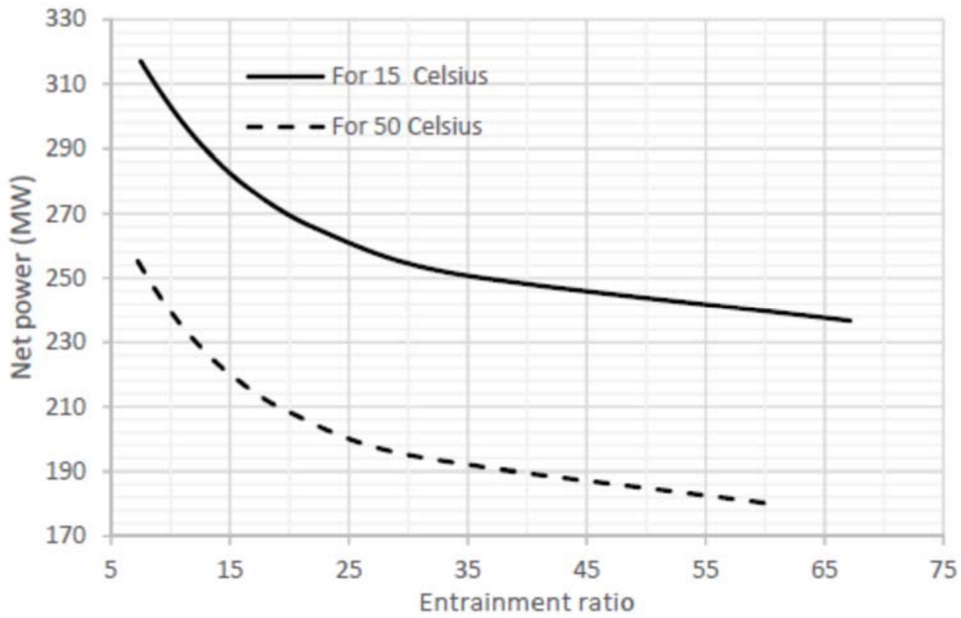

Figure 6: The effect of the entrainment ratio on the net power output

The specific fuel consumption (and hence the thermal efficiency) is a function in the fuel mass flow rate (Figure (2)) and in the net power output (Figure (6)). Both of them decline with different and variable rates. Figures (7, 8) show that there is a specific value for the entrainment ratio at which the specific fuel consumption (and hence the thermal efficiency) has an optimum value, beyond which the specific fuel consumption increases, and the thermal efficiency drops with the increase in the entrainment ratio.

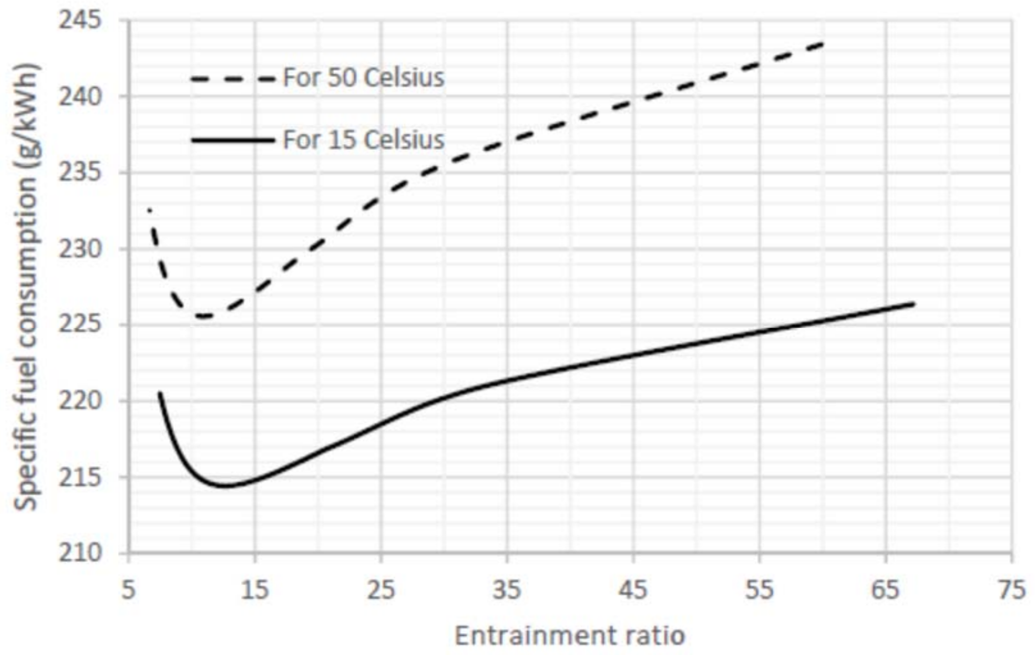

Figure 7: The effect of the entrainment ratio on the specific fuel consumption

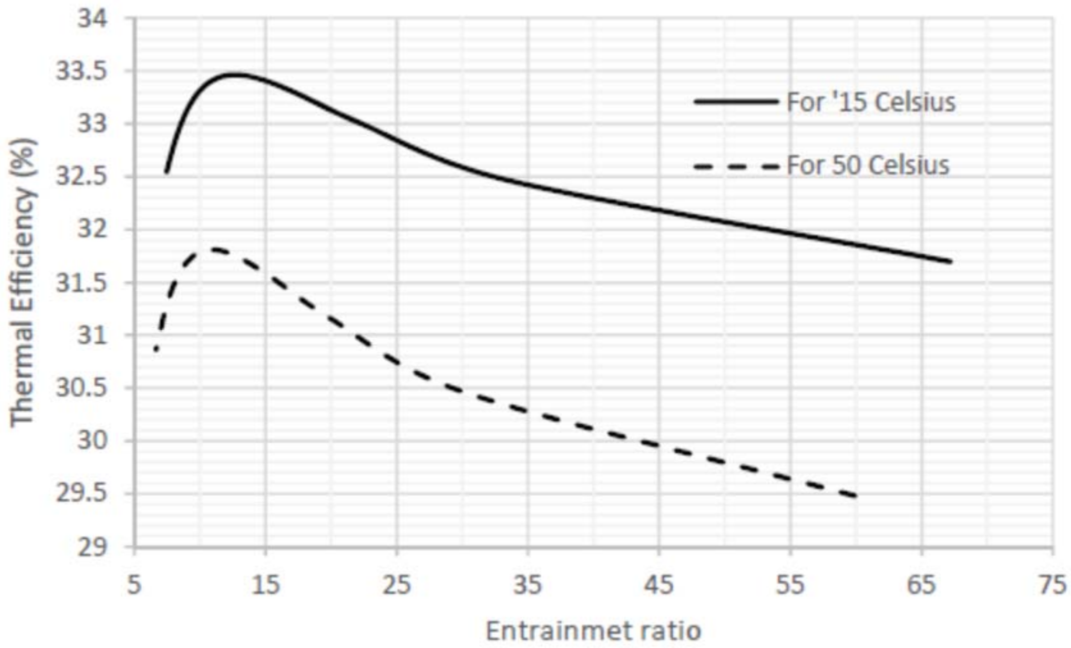

Figure 8: The effect of the entrainment ratio on thermal efficiency

Results show that the CO_2 emission factor has a minimum value corresponds to the point of the best thermal efficiency, as shown in Figure (9).

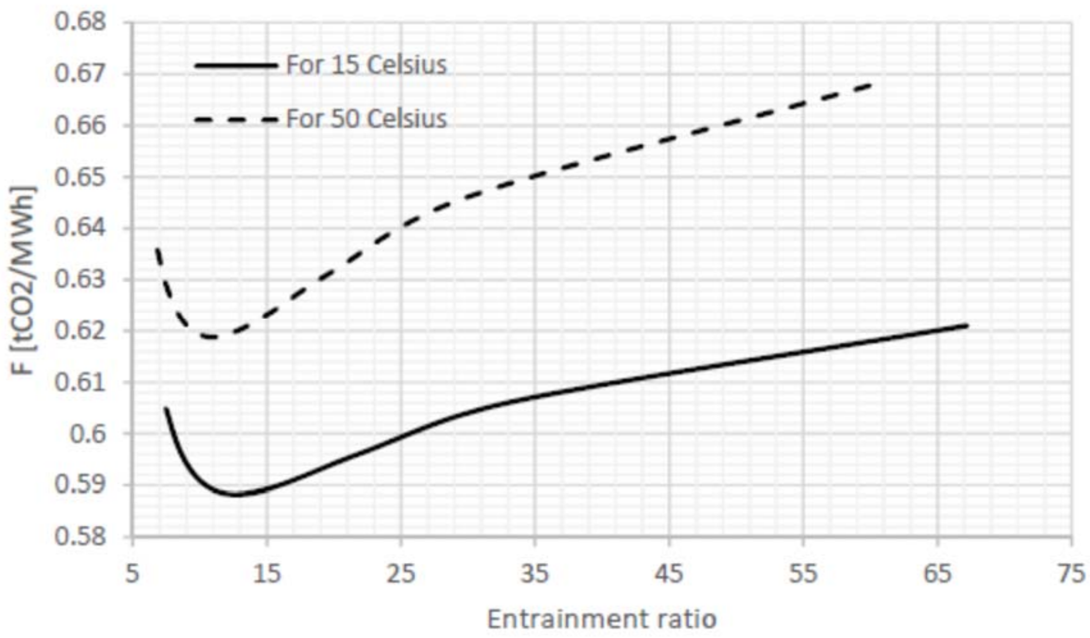

Figure 9: The effect of the entrainment ratio on the CO2 emission factor

Comparisons between the original and modified cycles are summarized in Table (3). The modified cycle has better thermodynamic performance than the original one, where the efficiency and the specific fuel consumption are improved. Increasing the entrainment ratio beyond the point of maximum efficiency causes a drop in the net power output.

Table 3: comparison between the original cycle and the modified one.

	Original cycle		**Modified cycle (at best efficiency)**		**Modified cycle (at max. power output)**	
	15ºC	**50ºC**	**15ºC**	**50ºC**	**15ºC**	**50ºC**
Entrainment ratio	--	--	13.44	11.98	7.47	6.66
Specific fuel consumption [g/kWh]	*250.46*	*261.64*	214.48	225.73	220.50	232.54
Efficiency [%]	*28.74*	*27.50*	33.46	31.79	32.54	30.86
Net power output [MW]	285.19	222.00	287.93	230.61	317.14	258.99
Fuel mass flow rate [kg/s]	19.83	16.13	17.15	14.46	19.43	16.73
CO2 emission factor [tCO_2/MWh]	0.5150	0.5383	0.5884	0.6192	0.6049	0.6379

CONCLUSIONS

The thermodynamic analysis of SARIR gas turbine power plant is divided into two parts:

In the first part of the analysis, the original cycle is analyzed. The ambient temperature is raised from 15ºC (ISO) to 50ºC (hot climate temperature). It is found that the net power is reduced by 1.810MW/ºC, and the thermal efficiency by 0.0355%/ºC. The results are in fair agreement with that given in the literature.

In the second part of the analysis, the original cycle is modified, such that an ejector and a heat exchanger (heat recovery steam generator) are incorporated. Two environmental temperatures are selected for the analysis, 15ºC, and 50ºC. The effect of the entrainment ratio on the thermodynamic performance is presented. It is found that the thermal efficiency, the specific fuel consumption and net power output are improved over the original cycle.

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