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RESEARCH ARTICLE **ENERGY**

Effect of Ambient Temperature and Fuel Switching on the Performance and the Environmental Impact of Gas Turbine Plants: A Case Study of West Tripoli Gas Power Plant

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ABSTRACT

This paper analyses the performance and environmental impacts of gas turbine plants operating on diesel and natural gas. It provides a comprehensive assessment of how varying ambient temperatures and fuel type affect gas turbine operation and pollutant emissions. This is done under identical turbine inlet temperatures and net power to eliminate their effects and to enable a clearer comparison. Based on a case study of the Tripoli West 167 MW simple-cycle power plant, a thermodynamic model developed in Excel is used to simulate GT performance. Results indicate that as ambient temperature changes, net exergy destruction is primarily influenced by the exergy destruction changes in the turbine (~45%) and the compressor (~40%), although the combustor is responsible for over 80% of the net destruction in the cycle. The results also highlight the relationships between specific fuel consumption, net thermal efficiency, and pollutant emissions. For every 10 °C rise in ambient temperature, the net thermal efficiency declines by 1.55% and 1.5%, the specific fuel consumption rises by 1.57% and 1.52%, and pollutant emissions rise by 1.58% and 1.53% for diesel and natural gas, respectively. This may link the changes in net thermal efficiency and pollutant emissions to the specific fuel consumption. Switching from diesel to natural gas significantly reduces net exergy destruction (~73%), attributed to reduced combustor exergy destruction, and boosts overall thermal efficiency by about 1%. Switching to natural gas also reduces CO_2 emissions by (~24%) due to a lower carbon-to-hydrogen ratio (~15.2%) and lower fuel consumption (~10.2%). The emission factors determined by this study are 3.082 kg CO_2/kg fuel for diesel and 2.612 kg CO_2/kg fuel for natural gas. The change in emission factor between diesel and natural gas is similar to the change in carbon-to-hydrogen ratio between the two fuels (~15.2%). This relates the CO_2 emission factor to fuel chemistry. The findings suggest that using natural gas is preferable, especially in warmer months.

تأثير درجة الحرارة المحيطة وتغيير نوع الوقود على أداء محطات التوربينات الغازية وتأثيرها البيئي: دراسة حالة محطة غاز غرب طرابلس

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الكلمات المفتاحية

التوربينات الغازية
التحول في الوقود
الطاقة المفقودة
إستهلاك الوقود النوعي
انبعاث الملوثات

الملخص

يقوم هذا البحث بتحليل الأداء والتأثيرات البيئية لمحطات التوربينات الغازية العاملة بالديزل والغاز الطبيعي. ويقدم تقييمًا شاملاً لكيفية تأثير تغير درجات الحرارة المحيطة ونوع الوقود على تشغيل توربينات الغاز وانبعاثات الملوثات. ويتم ذلك في ظل ثبوت كلا من درجة حرارة مدخل التوربين وصافي الطاقة المتولدة للحد من تأثيراتهما وتمكين المقارنة بشكل أكثر وضوحًا. وبناءً على دراسة حالة محطة توليد الكهرباء ذات الدورة البسيطة بغرب طرابلس بقدرة 167 ميجاوات، تم استخدام نموذج ديناميكي حراري تم تطويره ببرنامج إكسل لمحاكاة أداء محطة الغاز. تشير النتائج إلى أنه مع تغير درجة الحرارة المحيطة، يتأثر صافي الطاقة المفقودة بالدورة بالمقام الأول بالتغيرات في الطاقة المفقودة في التوربين (~45٪) والضاغط (~40٪)، على الرغم من أن غرفة الاحتراق مسؤولة عن أكثر من 80٪ من صافي الطاقة المفقودة في الدورة. وتسلط النتائج الضوء أيضًا على العلاقات بين استهلاك الوقود النوعي، والكفاءة الحرارية الصافية، وانبعاثات الملوثات. حيث أن لكل ارتفاع في درجة الحرارة المحيطة بمقدار 10 درجات مئوية، تنخفض الكفاءة الحرارية الصافية بنسبة 1.55% و1.5%، ويرتفع استهلاك الوقود النوعي بنسبة 1.57% و1.52%، وترتفع انبعاثات الملوثات بنسبة 1.58% و1.53% للديزل والغاز الطبيعي على التوالي. ويؤدي التحول من الديزل إلى الغاز الطبيعي إلى تقليل صافي تدمير الطاقة أو الطاقة المفقودة بشكل كبير (~73٪)، ويعزى ذلك إلى انخفاض الطاقة المفقودة في غرفة الاحتراق، ويعزز الكفاءة الحرارية الإجمالية بنحو 1٪. ويؤدي التحول إلى الغاز الطبيعي أيضًا إلى تقليل انبعاثات ثاني أكسيد الكربون~24٪ () وذلك بسبب انخفاض

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نسبة الكربون إلى الهيدروجين في تركيبته (~15.2٪) وإلى انخفاض استهلاك الوقود (~10.2٪). وقد حددت الدراسة عوامل انبعاث 3.082 كجم ثاني أكسيد الكربون/كجم وقود للديزل و2.612 كجم ثاني أكسيد الكربون/كجم وقود للغاز الطبيعي. وتشير النتائج إلى أن استخدام الغاز الطبيعي هو الأفضل، خاصة في الأشهر الأكثر حرارة.

Introduction

Greenhouse gas emissions such as carbon dioxide, sulfur dioxide, methane and nitrogen oxides are considered the main contributors to global warming. Around 53.82 billion tons of greenhouse gases, measured in carbon dioxide equivalents, were emitted globally in 2023, 37.79 billion tons of CO_2 emissions from burning fossil fuels [1]. With the increasing urgency to reduce greenhouse gas emissions and prevent severe climate change, there is a rising focus on renewable energy sources. Various techniques, such as carbon sequestration, desulfurization, and denitrification have been applied to exhaust gases to lower emissions [2]. These techniques consume vast amounts of energy, reduce power generation efficiency and, may also contribute to CO_2 emissions, such as during desulfurization in refining activities [3]. However, transitioning from high-carbon fuels to low-carbon alternatives can reduce greenhouse gas emissions with minimal cost, effort, and time [4]. The focus is also on enhancing the efficiency of power plants. The European Commission (EC) has set an ambitious target to reduce net greenhouse gas emissions by at least 55% by 2030, compared to levels in 1990. This goal includes achieving at least a 32.5% improvement in energy efficiency [5]. Extensive studies were conducted on gas turbine performance and the environmental impact of fuel switching. Basha et al. [6] examined how inlet temperature and fuel type affect carbon emissions and plant performance. They show that a 10 oF decrease in ambient inlet air temperature boosts the net power and efficiency by about 5 and 2%, respectively where natural gas works the best among diesel and heavy bunker oil. They also show that carbon emission increases with the decrease in ambient inlet temperature. Oyedepo et.al. [7] studied the influence of compression ratio, turbine inlet temperature (TIT) and ambient temperature on the performance of GT plants. They show that TIT significantly affects the GT performance where it should be kept higher to minimize the loss in the system and to boost the output power and thermal efficiency. They also showed that specific fuel consumption relates directly with ambient temperature and inversely with compression ratio and TIT. Burnes and Camou [8] analyzed the impact of fuel composition on a solar turbine's dual-fuel combustion system. They recorded a 20% decrease in carbon emissions and a 3.5% increase in power when using natural gas instead of diesel, concluding that the heating value and H/C ratio are key factors influencing gas turbine performance and CO_2 emissions. Alhajeri et al. [9] demonstrated that mixing fuels improved air quality and reduced energy costs. Their results across four scenarios of different proportions of four fossil fuels showed significant reductions in NOx, SO2 and particulate matter, with the third scenario of natural gas as the main and only energy source achieving the highest reduction of -36.8%, -98.9%, and -72.3%, respectively. The fourth scenario, substituting natural gas with crude oil, led to increases of +5.5%, +90.1%, and +64.8%. Additionally, costs dropped by 9% when using natural gas at full capacity. Egware and Kwasi-Effah [10] developed a correlation indicates that the CO_2 mass flow rate is linearly related to the turbine inlet-to-ambient temperature ratio, ambient relative humidity, compressor pressure ratio, and exhaust gas mass flow rate. Gas combustion significantly contributes to nitrogen oxide (NOx) emissions, accounting for about 10% of global emissions [11]. Increased efficiency in gas turbine power plants can raise flame temperatures, which raises NOx concerns [12]. Techniques such as wet low-emission technology have been implemented to dilute air and lower temperatures [13]. Despite NOx issues, natural gas is preferred for power generation due to its cleaner profile and lower sulfur content, making it a viable option in areas with high sulfur dioxide levels [14].

In Libya, fuel oils and natural gas are the main energy sources, with natural gas accounting for about 63% of electricity generation, while oil-based plants contribute by 29% [15]. In 2023, Libya emitted 95.39 million tons of greenhouse gases, including 60.98 million tons of CO_2, with the power generation sector responsible for 49% of emissions in 2022 [16]. Nassar et al. [17] tracked the CO_2 emission through entire energy industry from oil extraction, refining to infrastructure in Libya. They showed that power plants are the largest source of emissions, contributing by 93.6% due to reliance on fossil fuels (diesel, heavy fuel oil, natural gas) and to relatively low operating efficiency which averaging about 29%. The Tripoli West gas power plant operates on diesel in dual-fuel mode, allowing it to switch between liquid and gaseous fuels [18]. While most power plants in Libya are gas-based, some have not yet been converted to natural gas. This study aims to compare the performance and the pollution caused by the GT when operated on diesel or natural gas under the same operating conditions at different ambient conditions. A comprehensive analysis is conducted to evaluate the effects of changes in ambient temperature and fuel type on individual and overall unit performance and on the environment. The analysis is based on operating the plant at a fixed turbine inlet temperature and other internal parameters to eliminate their effects on GT performance. The analysis is also based on fixed net power to enable a better assessment of the effect of fuel switching on the environment. The study also aims to quantitatively estimate the exergy destruction from the individual units when any of the above external operating parameters is changed. Such an analysis can be beneficial in optimizing the overall plant efficiency and its environmental impact.

Methodology

The basic methodology behind the gas turbine cycle is that it consists mainly of an air compressor, combustion chamber and a turbine. In such a system, the air is compressed to a pressure of several bars before it enters the combustor, where it reacts with a fuel. The high-temperature gases from a combustor drive the gas turbine. The turbine output shaft is coupled to the air compressor and operates on the same shaft. A schematic representation of the gas turbine power plant is shown in Fig.1. Momentum, energy and exergy analyses are conducted to model the GT plant. The momentum analysis is based on a steady state process and the law of mass conservation, in which:

$$\sum \dot{m}_{out} = \sum \dot{m}_{out} \quad (1)$$

Where: $\dot{m}_{in}$ and $\dot{m}_{out}$ are mass flow rate (kg/sec) for the working fluids that is in and out of the individual uni.

The energy balance is based on the assumption of an ideal gas mixture and neglects both the kinetic and potential energy change for all the units. The first law of thermodynamics is applied in which:

$$\Delta H = \dot{Q} - \dot{W}_s \quad (2)$$

Where: ΔH is total enthalpy change (kJ/sec), $\dot{Q}$ and $\dot{W}_s$ are heat rate (kJ/sec) and shaft work (kJ/sec), respectively. For an adiabatic system, the isentropic compressor required work $W_s^{\cdot}$ or $W_{comp}^{\cdot}$ and the isentropic turbine produced work $W_s^{\cdot}$ or $W_{turb}^{\cdot}$ are calculated according to the following equation:

$$W_s^{\cdot} = \Delta H = m^{\cdot}\, Cp_{mh}\,(T_{out} - T_{in}) \quad (3)$$

where: $m^{\cdot}$ is the mass flow rate (kg/sec), T_{in} and T_{out} are the inlet and outlet temperatures (K), respectively and Cp_{mh} is an ideal gas mean heat capacity specific to enthalpy calculations (kJ/kg K) and is defined as [19]:

$$Cp_{mh} = \int_{T_{in}}^{T_{out}} \frac{Cp\,(T)dT}{T_{out} - T_{in}} \quad (4)$$

The net power $P_{net}^{\cdot}$ is defined as:

$$P_{net}^{\cdot} = W^{\cdot}_{turb} + (-W^{\cdot}_{comp}) \quad (5)$$

The energy released by the combustion chamber $Q_{cc}^{\cdot}$ is:

$$Q_{cc}^{\cdot} = m_{fuel}^{\cdot} LHV \mu_{cc} \quad (6)$$

where: $m_{fuel}^{\cdot}$ is the fuel mass flow rate (kg/sec), LHV is the lower heating value (kJ/kg), μ_{cc}.is the combustor efficiency, which is assumed to be 99%.

The exergy analysis obeys the second law of thermodynamics in which the total entropy change associated with any process is always positive and it approaches zero only when the process is reversible. This represents the work loss or the irreversibility of the system and is defined as:

$$W_{lost}^{\cdot} = T_o \Delta S \geq 0 \quad (7)$$

Where: $W_{lost}^{\cdot}$ is the loss of work due to entropy change, which is also called exergy destruction. The work lost, or the exergy destruction for a certain unit is:

$$W_{lost}^{\cdot}(\text{exergy destruction}) = T_o \Delta S \quad (8)$$

and,

$$\Delta S = Cp_{ms} \ln \frac{T_{out}}{T_{in}} - R \ln \frac{P_{out}}{P_{in}} \quad (9)$$

where: T_o is the surrounding temperature, which is considered as 298 K, R is the gas constant, P_{in} and P_{out} are the inlet and outlet pressures to and from a certain unit, and C_{pms} is the ideal gas mean heat capacity specific to entropy calculations and is defined as [19]:

$$Cp_{ms} = \int_{T_{in}}^{T_{out}} \frac{Cp(T)\; dT/T}{\ln(T_{out,}/T_{in})} \quad (10)$$

Applying energy and exergy analysis to the individual units and considering a reversible adiabatic (isentropic) process for both compressor and turbine, the isentropic shaft work $W_{s,isnt}^{\cdot}$ is calculated using equations (3), where the outlet temperature is assumed to be equal to the isentropic temperature $T_{out} = T_{out,isnt}$ and it is determined using an iterative solution applied to equation (9), where $\Delta S = 0$.

The actual temperature out is determined by either multiplying or dividing the second term of the right-hand side of equation (11) by the isentropic efficiency τ_{isnt} and this depends on whether the process produces or requires work. The shaft work in this case is the actual $W_{s,act}^{\cdot}$ which is for an irreversible process, and it is calculated using equation (3), where the outlet temperature is the actual temperature $T_{out} = T_{act}$

$$T_{out} = T_{in} + \left(T_{out,isnt} - T_{in}\right)\left(\frac{\tau_{isnt,\ turbine}}{\tau_{isent,,compressor}}\right) \quad (11)$$

The entropy change is calculated using:

$$\Delta S(isentropic) = Cp_{ms} \ln \frac{T_{out,isnt}}{T_{in}} - R \ln \frac{P_{out}}{P_{in}} = 0 \quad (12)$$

Where: C_{pms} is the ideal gas mean heat capacity specific to entropy calculations and is defined as:

$$Cp_{ms} = \int_{T_{in}}^{T_{out}} \frac{Cp(T)\; dT/T}{\ln(T_{out,isnt}/T_{in})} \quad (13)$$

For an adiabatic reversible process applied to combustion chambers, no heat is added or removed, $Q^{\cdot} = 0$ and since no shaft work is produced or required by this unit, equation (1) reduces to:

$$\Delta H = 0 \quad \text{or} \quad m_{fuel}^{\cdot} LHV \mu_{cc} + m_{air}^{\cdot} Cp_{mh,air}\left(T_{in,cc} - T_o\right) = m_{flue}^{\cdot} Cp_{mh,flue}\left(T_{out,cc} - T_o\right) \quad (14)$$

Where: $m_{air}^{\cdot}$ is the air mass flow rate (kg/sec), $m_{flue}^{\cdot}$ is the flue gas mass flow rate (kg/sec), $T_{in,cc}$ and $T_{out,cc}$the inlet and outlet temperatures to and from the combustor chamber, T_o is the surrounding temperature which is considered as 298 K, $Cp_{mh,air}$ and $Cp_{mh,flue}$ are the mean heat capacity for the air and the flue gas, respectively and can be calculated by eq (4).

Solving equation (14) by an iterative solution for $T_{out,cc}$ which is in this case, the maximum temperature that can be achieved as a result of combustion or $T_{out,cc} = T_{max}$. This is also known as the adiabatic flame temperature in the case of stoichiometric reactions. For a non-adiabatic reversible process, the system is described by the following:

$$Q^{\cdot}_{rev} = T_o\,\Delta S - m_{flue}^{\cdot} Cp_{mh,flue}(T_{out,cc} - T_{in,cc}) \quad (15)$$

and,

$$\Delta S = Cp_{ms,flue} \ln \frac{T_{out,cc}}{T_{in,cc}} - R \ln \frac{P_{out,cc}}{P_{in,cc}} \quad (16)$$

$P_{in,cc}$ and $P_{out,cc}$ are the pressure into and out from the combustion chamber which is practically a small pressure drop. The gas turbine thermal efficiency τ_{th} is expressed as:

$$\tau_{th} = \frac{P_{net}^{\cdot}}{Q_{cc}^{\cdot}} \quad (17)$$

and, the specific fuel consumption SFC (kg fuel/kWh) is expressed as:

$$SCF = \frac{3600 \times m_{fuel}^{\cdot}}{p_{net}} \quad (18)$$

Pollutant Emission

Air pollution caused by fossil-based power plants is a serious issue to humans and the planet. Emissions mainly depend on the composition of the burning fuel [20], the type and size of the boiler [12], and the operating conditions, such as compression ratio and ambient temperature [10]. Using fossil fuels and natural gas, particularly for power generation, contributes to the emission of various pollutants, such as CO_2, SOX and NOX. The power industry is a significant contributor to CO_2 emissions which contribute to global warming. The amount of CO_2 emitted per MW of power is

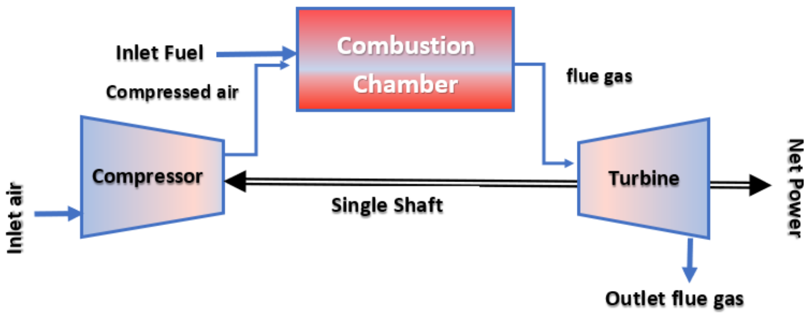

Figure 1: Schematic representation of a simple cycle gas turbine plant

affected mainly by the type of fuel and the efficiency of the plant. In a combustion process, a hydrocarbon fuel and oxygen in the air enter the combustor and are reacted by a stochiometric reaction in which all carbon is burned into carbon dioxide and all hydrogen is converted to water. Fuels may also contain some sulfur which oxidize to sulfur oxides (SOx) upon combustion. The exhaust gas turbine's sulfur oxide emissions are mainly caused by the combustion of sulfur introduced by the fuel. The sulfur oxides contain sulfur dioxide (SO2) and some sulfur trioxide (SO3). The amount of sulfur dioxide released due to combustion can be easily calculated by knowing the composition of sulfur in the fuel and the fuel flow rate [21]. SO3 can be combined with water vapor in the exhaust to form sulfuric acid, contributing to acid rain. However, natural gas has a much lower sulfur content than fuel oils, and that is why conversion to natural gas is considered in regions where high sulfur dioxide percentages were detected in the air [14].

NOx emissions, including nitric oxide (NO) and nitrogen dioxide (NO2), can cause significant harm to the environment and health. Therefore, emission standards require low NOx regulations. Nitrous oxide (N2O), which is considered a greenhouse emission, can also be formed and play the role of an intermediate in NOx formation. NO production is classified mainly into different routes: thermal-NO route, prompt-NO route, N2O route and fuel-N route.

The **thermal-NO route** is primarily affected by temperature and becomes dominant over other routes when the temperature is above 1600 ºC and with an increased air-to-fuel ratio [22]. The thermal NO route is also affected by combustor length through residence time; the more combustor length and residence time, the more NOx formation [23]. The mechanism of thermal NO is described according to the following reactions:

$N_2 + O^{\cdot} \leftrightarrow NO + N^{\cdot}$

$O_2 + N^{\cdot} \leftrightarrow NO + O^{\cdot}$

$OH + N^{\cdot} \leftrightarrow NO + H^{\cdot}$

Where: $N^{\cdot}$, $O^{\cdot}$ and $H^{\cdot}$ are nitrogen, oxygen and hydrogen radicals, respectively. These radicals are formed from other reduction reactions. The **prompt- NO Route** occurs at the fuel-rich flame consumption area where nitrogen from the flowing air is reacted with hydrocarbon radicals according to the following reactions:

$N_2 + CH^{\cdot} \leftrightarrow HCN^{\cdot} + N^{\cdot}$

$N_2 + C^{\cdot} \leftrightarrow CN^{\cdot} + N^{\cdot}$

The whole pathway is complex; equations (19) and (20) are intermediate chain reactions as $CH^{\cdot}$ and $C^{\cdot}$ radicals are formed by a series of CH_4 transformation reactions. Then $HCN^{\cdot}$ and $CN^{\cdot}$ are oxidized to $NCO^{\cdot}$, which is hydrogenated to $NH^{\cdot}$ and; $NH^{\cdot}$ is finally transferred to NO.

The **N₂O/NO Route** is similar to the thermal-NO route where radicals $O^{\cdot}$ reacts with nitrogen. This reaction is dominant at an increased air-to-fuel ratio and below 1400 ºC [22] according to the following:

$N_2 + O^{\cdot} + M \leftrightarrow N_2O + M$

$N_2O + O^{\cdot} \leftrightarrow 2NO$

$N_2O + O^{\cdot} \leftrightarrow N_2 + O_2$

The **Fuel-N Route** occurs due to nitrogen, which is chemically bound into the fuel and is called fuel nitrogen. Fuel nitrogen compounds are typically aromatic with one or more bounded nitrogen atoms.

System Description

The Tripoli West 671 MW simple-cycle power plant is located in north-western Libya, approximately 30 km west of Tripoli. The power plant comprises four 167 MW gas turbines that operate in dual-fuel mode. The turbines use light distillate oil as a primary fuel, while natural gas is supposed to be used as a secondary fuel. Gas is sourced from the onshore Wafa Field, through the Mellitah Complex, where it is treated and transported via pipelines to the domestic market. The operating data used to model the power plant is listed in Table (1). The amount of air enters the compressor is adjusted according to ambient temperature change to maintain constant turbine inlet temperature. The fuel mass flow rate is also adjusted to maintain a constant net power. Diesel and natural gas fuels with the specification given in Tables 2 and 3 are used:

Table 1: Model operating input data

Operating parameters	**Value**	
Ambient temperature (ºC)	5, 15, 25, 35	
Atmospheric pressure (atm)	1	
Humidity (%)	60	
Compressor pressure ratio	11	
Compressor isentropic efficiency	0.85	
Turbine inlet temperature (K)	1313	(19)
Turbine isentropic efficiency	0.87	(20)
Net power (MW)	137.52	(21)

Results

The gas turbine power plant studied is simulated using MS Excel spreadsheet, where momentum, energy and exergy balance equations are required. The GT plant performance and environmental impact are assessed based on the same net power achieved at various ambient temperatures. (22) Since the ambient temperature inversely affects the air (23) density and hence the air mass flow rate, the air mass flow rate through the compressor is reduced as the ambient temperature rises. This reduces the air-to-fuel ratio and causes a drop in the net power in hotter weather conditions.

Table 2: Diesel oil fuel physical properties and composition[*] * Source: Libyan Petroleum Institute

Lower heating value (kJ/kg)	**Molecular weight**	**Specific gravity at 15 ºC**	**Stoichiometric air-to-fuel ratio**		
46,841.87**	18.446***	0.64	16.4		
Fuel Composition (mol %)					
Methane	**Ethane**	**Propane**	**i-butane**	**n-butane**	**n-pentane**
86.071	9.953	1.442	0.032	0.082	0
Carbon dioxide	**Nitrogen**	**Sulfur**	**Helium**	-	-
1.898	0.483	0.0	0.093	-	-

Calculated by $LHV = \sum_i^n LHV_i / MW_i$ * Calculated by $MW = \sum_i^n y_i \times MW_i$

Table 3 Chemical composition of typical Libyan natural gas [24]

Higher heating value (kJ/kg)	**Temperature (º C)**	**Density (kg/l) at 15 ºC**	**Stoichiometric air-to-fuel ratio**		
42,510.0	15	0.843	14.8		
Fuel Composition (Wt %)					
C	**H**	**N**	**O**	**S**	**AR**
86.589	13.315	0.015	0.021	0.06	-

Therefore, to keep the same net power for different ambient conditions, the plant has to be operated at increased turbine inlet temperatures, as shown in Fig. 2 (a and b). This can be achieved by increasing the amount of fuel burned, but this led to an elevated TIT reaching 1396K and 1383K when the plant is operated on diesel and natural gas at 35 ºC, respectively. These high combustion temperatures may be unsuitable given the operating temperature limits of the turbine materials. High combustion temperatures may also increase nitrogen oxide emissions [22,25].

In addition, varying combustion temperatures can lead to different overall GT performance across net power, efficiency, exergy destruction, and specific fuel consumption [26-28]. Therefore, to eliminate this effect and enable fair analysis across ambient conditions and fuel types, all assessments are conducted at a fixed TIT and net power, with the air mass flow rate adjusted accordingly. The air mass flow rate was adjusted to maintain a fixed TIT of 1313K at different ambient temperatures, as shown in Fig. 2(c and d). However, this TIT is close to the optimum TIT of 1328 K, which was selected by [25] based on energy, exergy, environmental and economic analysis.

Exergy analysis reveals that exergy destruction increases with rising ambient temperature across all GT components and overall cycle (Fig. 3 (a–d) and Tables 4 and 5). The combustion chamber accounts for over 80% of total exergy destruction. However, the incremental increase in exergy destruction per 10 °C rise is primarily attributed to the turbine (0.75 ± 0.06 MW, 45%), followed by the compressor (0.61± 0.05 MW, ~40%). The combustor contributed by only (0.30±0.2 MW and 0.23±0.2 MW, ~15%) for diesel and natural gas, respectively. Fig. 4 (a and d) and Tables 4 & 5 show that the incremental increase in compressor power consumption and turbine power production is 13.44±1.17 MW per 10 °C rise. However, this incremental increase is similar for both units due to fixed net power operation.

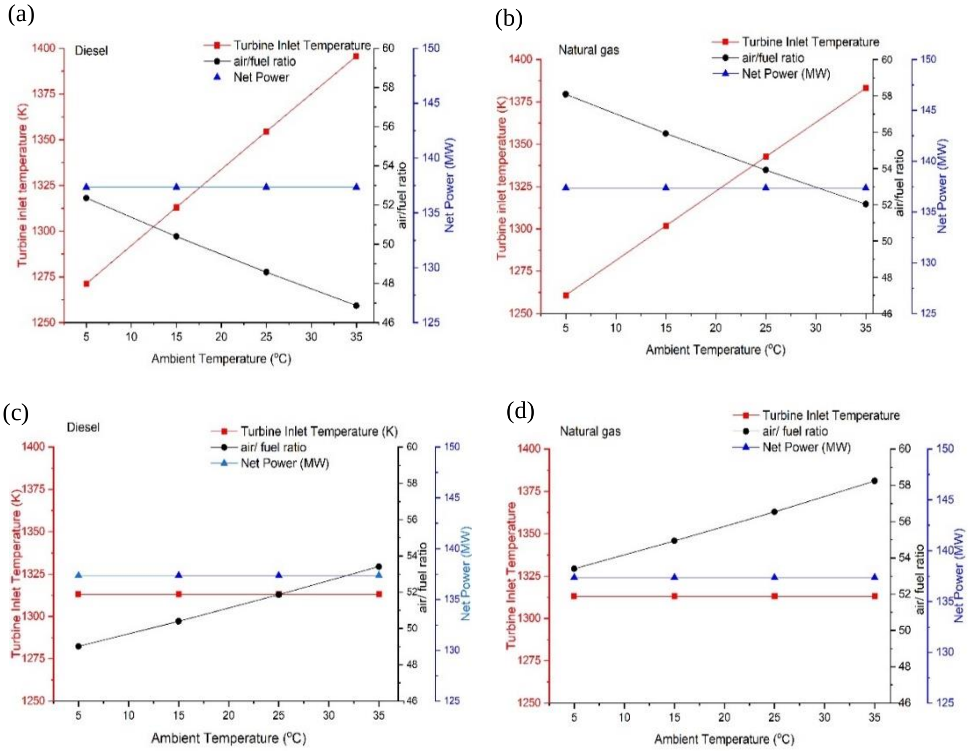

Figure 2: The GT operating parameters based on fixed net power when running by:(a) diesel, (b) natural gas; based on fixed net power and TIT when running by: (c) diesel, (d)natural gas

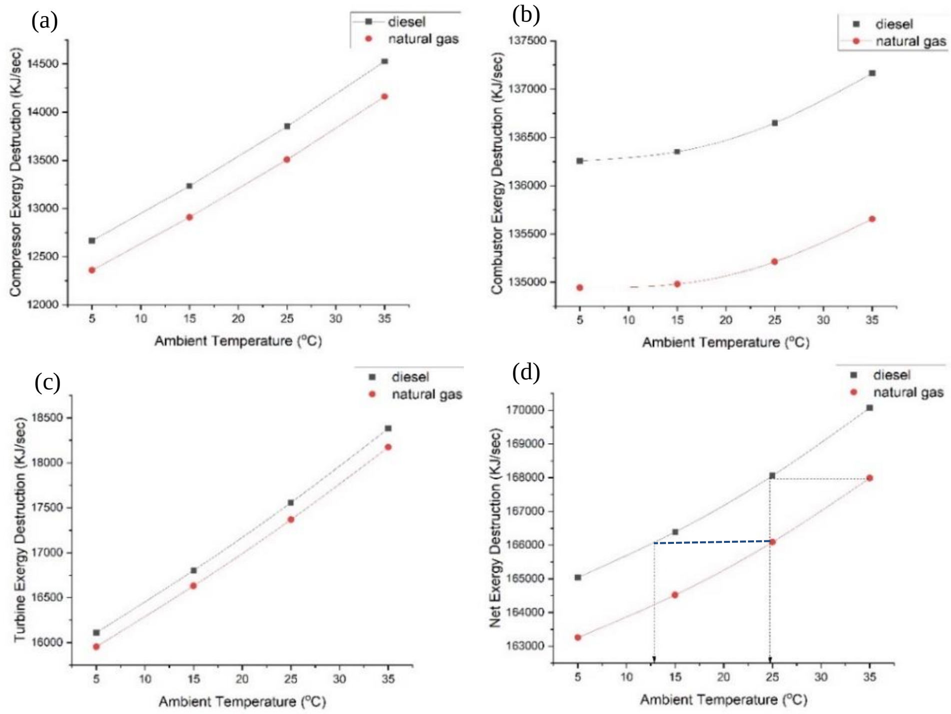

Figure 3: The exergy destructions at different ambient temperatures when running by either diesel or natural gas for: (a) compressor; (b) combustor; (c) turbine and (d) overall cycle

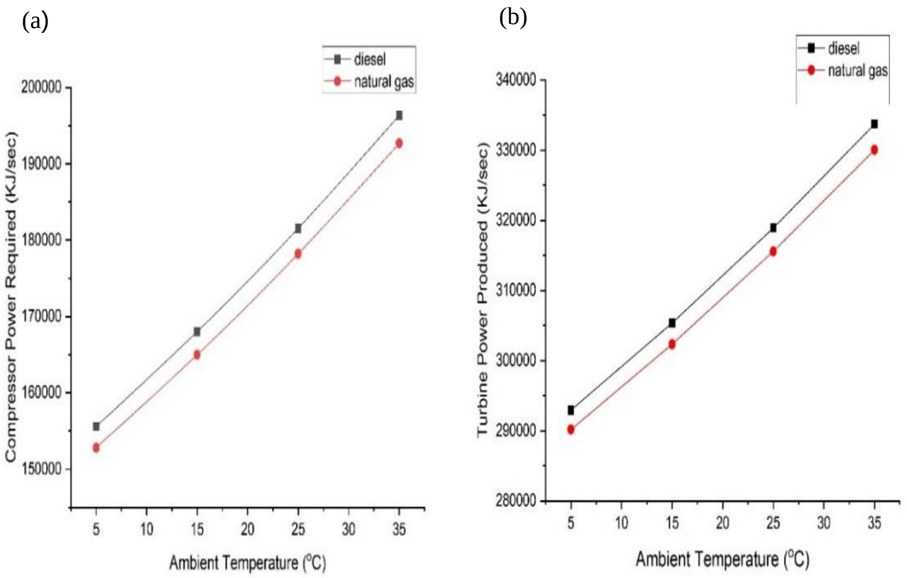

Figure 4: The power at different ambient temperatures when operating by either diesel or natural gas: (a) required by the compressor; (b) produced by the turbine

Table 4: The MW of exergy destruction and power that is added or removed when the ambient temperature changes by 10 °C for diesel

Ambient Temperature	Compressor		Combustor	Turbine		Overall cycle	
	destruction	power Required	destruction	destruction	power produced	net destruction	net power
5 to 15 °C	+0.57	+12.43	+0.10	+0.69	+12.43	+1.36	0.00
15 to 25 °C	+0.62	+13.54	+0.30	+0.75	+13.55	+1.67	0.00
25 to 35 °C	+0.68	+14.80	+0.51	+0.82	+14.80	+2.01	0.00

Table 1: The MW of exergy destruction and power that is added or removed when the ambient temperature changes by 10 °C for natural ga

Ambient Temperature	Compressor		Combustor	Turbine		Overall cycle	
	destruction	power Required	destruction	destruction	power produced	net destruction	net power
5 to 15 °C	+0.55	+12.16	+0.04	+0.68	+12.16	+1.26	0.00
15 to 25 °C	+0.60	+13.23	+0.23	+0.74	+13.23	+1.57	0.00
25 to 35 °C	+0.65	+14.46	+0.44	+0.81	+14.46	+1.90	0.00

Table 2: The MW of exergy destruction and power that is added or removed when converting fuels from diesel to natural gas

Ambient Temperature (ºC)	Compressor		Combustor	Turbine		Overall cycle	
	destruction	power Required	destruction	destruction	power produced	net destruction	net power
5	-0.31	-2.76	-1.31	-0.16	-2.76	-1.78	0.00
15	-0.32	-3.03	-1.37	-0.17	-3.02	-1.86	0.00
25	-0.34	-3.33	-1.44	-0.19	-3.34	-1.97	0.00
35	-0.37	-3.68	-1.51	-0.21	-3.68	-2.09	0.00

Switching from diesel to natural gas reduces exergy destruction across all units (Fig. 3(a-d) and Table 6), primarily due to a significant reduction in combustor exergy destruction (−1.41 ± 0.1 MW, ~73%). The compressor exergy destruction declines by (−0.34 ± 0.03 MW, ~18%), whereas the turbine destruction declines by (−0.18 ± 0.025 MW, ~9%). Fig. 4(a and b) and Table 6 show that both compressor power consumption and turbine power production decrease by (−3.20 ± 0.3 MW) upon switching. The net thermal efficiency of the GT in Fig. 5 decreases linearly with increasing ambient temperature. It declines by ~1.55% for diesel and ~1.50% for natural gas for every 10 °C rise in ambient temperature. A study conducted at the Tripoli West power plant [29] showed an average of 1.33% reduction in thermal efficiency for every 5 ºC increase in ambient temperature. However, the operating conditions applied is different, and this will be discussed later. Another study conducted for the Middle East desert conditions [30] found that for every one-degree rise in ambient temperature above an iso condition of 15 °C, the gas turbine efficiency loses 0.1% of its thermal energy. The net thermal efficiency is boosted by about 1% upon switching to natural gas.

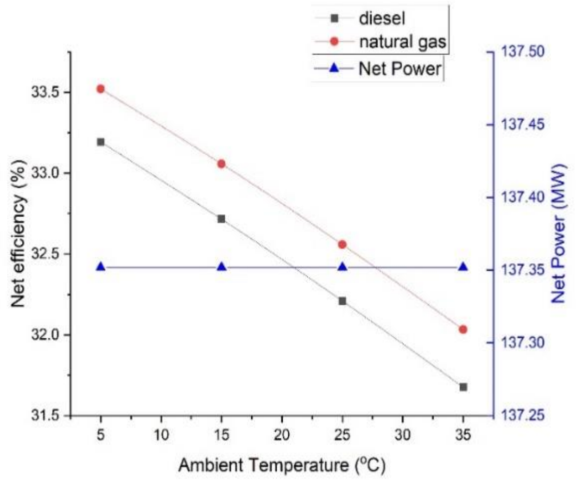

Figure 5: Net efficiency and net power for the overall cycle when running by either diesel or natural gas at different ambient temperatures

The relationship between the air-to-fuel ratio (a/F) and the specific fuel consumption (SFC) with ambient temperature is linear, as shown in Fig. 6 (a and b). For every 10 °C rise in ambient temperature, SFC increases by ~1.57% for diesel and ~1.52% for natural gas. Switching to natural gas lowers SFC by ~10.2%. This is quite similar to [31], who showed that SFC for natural gas is lower than that of diesel by about 13.5% when the net power is fixed. The air-to-fuel ratio increases by ~2.9% for every 10 °C rise, whereas when switching to natural gas, it increases by (~9%). This reflects the need for more air to maintain constant TIT and power output. Pollutant emissions per MW produced also vary with ambient temperature as shown in Fig.7. For diesel, CO_2, SO_2, and NOx emissions increase by about 13.22, 0.00488, and 0.00134 kg/MW, respectively, representing about 1.58% increase for every 10 ºC rise. For natural gas, CO_2 and NOx emissions rise by 9.8 kg/MW and 0.057 kg/MW, representing about 1.53% increase for every 10 ºC. Switching from diesel to natural gas reduces CO_2 emissions by 200 kg/MW (~23.95%) and lowers the CO_2 emission factor from 3.082 to 2.612 kg/kg fuel (~15.25%).

These emission factors are quite close to those determined by other studies in Libya. Kraedegh [32] estimated an emission factor of 3.114kg CO_2/kg of heavy fuel for a steam power plant. Nassar et al. [17] estimated a factor of 2.984 kgCO_2/kg of diesel from the electricity generation sector. SO_2 emissions decline by 99% upon switching. This is attributed to fuel composition, where the composition of natural gas studied shows no sulfur content. However, NOx emissions elevated upon switching, contradicting the perception of natural gas as a cleaner fuel and this will be discussed later.

Discussion

The performance and environmental impact of a gas turbine (GT) plant operating on diesel or natural gas are evaluated under varying ambient temperatures, with fixed net power and turbine inlet temperature (TIT), where the air and fuel mass flow rates are adjusted accordingly. The study also assesses the impact of switching from diesel to natural gas. The results reveal that as ambient temperature rises by 10 °C, the additional compressor and turbine exergy destructions contribute to overall net exergy destructions of about 40% and 45%, respectively. The exergy destruction in the combustor has the least impact among the three units, even though it accounts for more than 80% of the net exergy destruction. Ambient temperature increase reduces the air inlet to the compressor and hence reduces the GT net power and the overall thermal efficiency. The GT net power is kept constant by consistently adjusting the air-to-fuel ratio as ambient temperature changes. Since overall thermal efficiency is significantly influenced by both ambient temperature and TIT [33], the GT is also operated at a constant TIT to eliminate its effect on overall efficiency. The results reveal a linear relationship, with a similar percentage change of (~1.55% and ~1.57% for diesel) and (~1.50% and ~1.52% for natural gas) for the overall thermal efficiency drop and SFC rise, respectively, for every 10 °C increase in ambient temperature. However, many studies have determined a similar percentage change between SFC and thermal efficiency with ambient temperature change, despite the differences in operating conditions. Shukla and Singh [27] showed that when the compressor inlet temperature drops from 318K to 282K, there is a 3.43% decrease in SFC and a 3.45% gain in thermal efficiency. Kumar et al. [34] showed that for every 10K increase in ambient temperature, the thermal efficiency decreases by about 1.5% for an air-to-

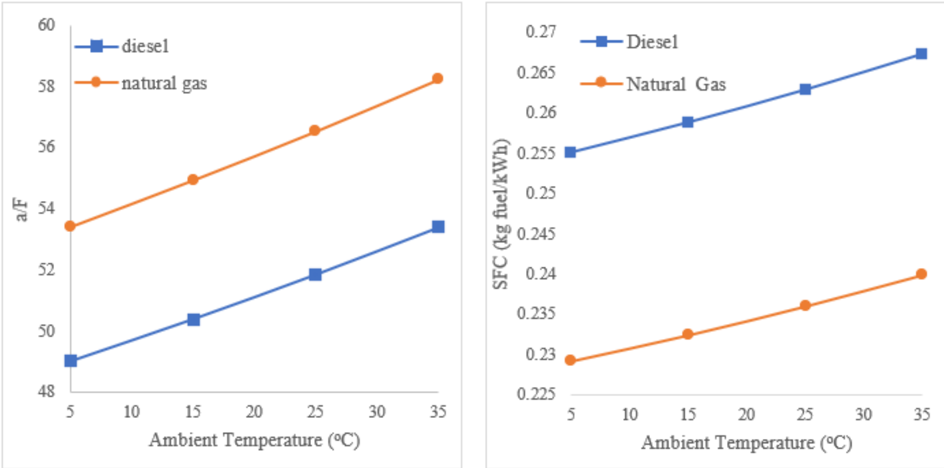

Figure 6: (a) air-to-fuel ratio (b) Specific Fuel Consumption (SFC); when running by either diesel or natural gas at different ambient temperatures

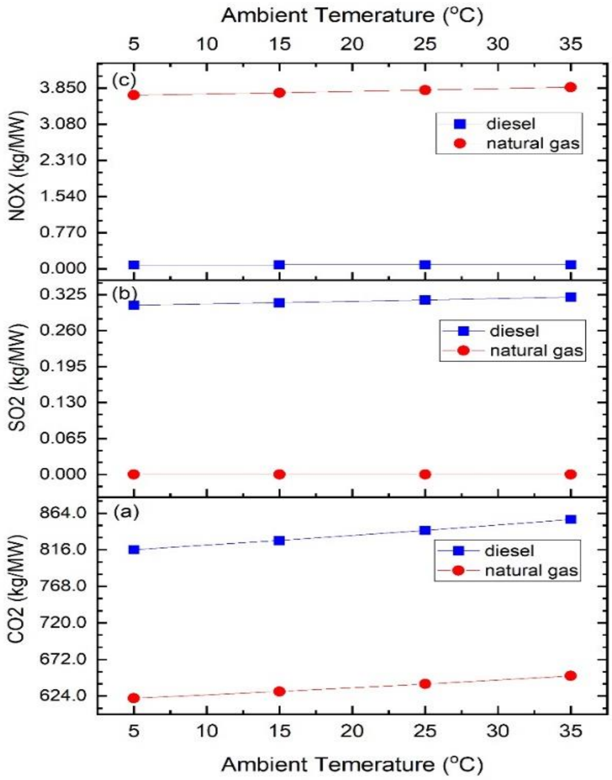

Figure 7: Pollutant emission in kg/MW by both fuels at different ambient temperatures: (a) CO_2; (b)SO_2; (C) NOx

fuel ratio of 40, and the SFC increases by 1% for an air-to-fuel ratio of 44. Montasser and Eshoul [29] also observed a similar trend, where for every 5 °C rise in ambient temperature, the SFC increases by 1.35%, and the thermal efficiency decreases by 1.33%. Surprisingly, the emission of pollutants also showed a close percentage change to that for thermal energy and SFC upon temperature change. Pollutant emission increased by ~1.58% and ~1.53 % for diesel and natural gas, respectively, for every 10 ºC. This may link the changes in net thermal efficiency and pollutant emissions to the specific fuel consumption. The specific fuel consumption (SFC) is the amount of fuel consumed in kg per kWh of net power produced. Since the net power is fixed in this study, this may lead to the conclusion that the GT performance and its environmental impact are directly affected by the amount of fuel being consumed. The relation between the air-to-fuel ratio and the ambient temperature also showed a direct linear relationship but, with a higher percentage of ~ 2.9% rise for every 10 °C increase. This implies that the gas turbine (GT) requires a higher airflow and consumes more fuel as ambient temperature increases. Therefore, it is of interest to investigate the optimum air-to-fuel for optimum performance and less harm to the environment. Converting from diesel to natural gas, on the other hand, provides significant environmental and efficiency advantages. The net exergy destruction of the gas turbine (GT) is reduced by approximately 73%, attributed to the reduction in combustor exergy destruction. The transition also boosts the overall thermal efficiency by about 1%. Furthermore, a comparable net exergy destruction occurs when the gas turbine operates on natural gas compared to diesel at a lower temperature by about 10-12 °C, as illustrated by the dashed lines in Fig. 3(d).

The CO_2 emissions reduction when switching to natural gas was about 24%. This is probably due to the combined effect of a lower carbon-to-hydrogen ratio in the fuel chemistry, which led to approximately a 15.2% reduction in the CO_2 emitted per kg of fuel and a reduction in fuel consumption of about 10.2%. This is in agreement to [35] in which their results reveal a reduction of 9.45% in SFC and of 22.2% in CO_2 emissions upon using natural gas comparing to diesel. However, natural gas is noted for its low carbon-to-hydrogen ratio. According to [36], fuel CO_2 emissions primarily depend on their carbon content and hydrogen–carbon ratio. The CO_2 emission factor estimated in this study reduces from 3.082 to 2.612 kg/kg fuel (~15.25%) when switching and this is the same percentage detected due to carbon-to-hydrogen ratio change. Therefore, the CO_2 emission factor can be considered as a measure of fuel chemistry. Since conversion to natural gas showed substantial benefits comparing to diesel. Therefore, it is preferred to convert to natural gas in the warmer months of the year. The significant decrease in SO_2 emissions when using natural gas results from its low sulfur content. While CO_2 and SO_2 emissions can be directly calculated from the fuel flow rate and composition, NOx emissions are more complex and strongly influenced by operating conditions and GT design [21]. The overestimation of NOx emitted from natural gas is likely due to a simplified combustion reaction of nitrogen and oxygen that is used in the model, where over 95% of the emitted NOx is considered in the form of NO [37]. The remainder exists as nitrogen dioxide (NO_2), produced by the NO oxidation reaction. In fact, hundreds of elementary reactions, through various pathways, are needed to model the NOx reaction process for natural gas [38] as described in pollutant emissions subsection. Solving these reactions for NO, NO_2, and N_2O requires simulating the temperature distribution along with the concentrations of reactants and NOx formation, all of which relate to retention time within the combustor. This is a complex mathematical problem which needs comprehensive computational modelling to address, such as Computational Fluid Dynamics (CFD) or the Chemical Reaction Network Method (CRNs) [39], or both [40].

Conclusions

This study evaluates the thermodynamic performance and environmental impact of gas turbine plants operating on diesel or natural gas and the effect of conversion from diesel to natural gas, considering varying ambient temperatures while maintaining fixed net power and turbine inlet temperature (TIT). The main findings include:

a) When ambient temperature increases:

- The net exergy destruction increases, driven primarily by the turbine and compressor.
- The combustor has less impact, despite being responsible for over 80% of net exergy destruction.
- Similar percentage changes were observed for specific fuel consumption (SFC), thermal efficiency and pollutant emission.
- The GT performance and its environmental impact are directly affected by the amount of fuel being consumed.

(b) when switching from diesel to natural gas:

- Switching from diesel to natural gas offers notable environmental and efficiency benefits.
- Natural gas reduces net exergy destruction due to reduced combustor exergy destruction.
- Natural gas reduces specific fuel consumption and CO_2 emissions and improves thermal efficiency.
- Natural gas CO_2 emissions are reduced due to both lower carbon-to-hydrogen ratio and lower fuel consumption.
- The CO_2 emission factors is affected by fuel chemistry (Carbon-to hydrogen ratio).

Recommendations

The findings suggest that gas turbine performance declines as ambient temperatures rise, while operating on natural gas improves efficiency and reduces environmental impact, making it a preferable option in hotter conditions. The study also highlights the need for optimizing air-to-fuel ratios for better performance and reduced environmental impact.

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References

[1] Our World in Data, CO_2 and Greenhouse Gas Emissions Data Explorer. [Online] Available at: https://ourworldindata/explorers/co2?Gas+or+Warming.

[2] R. Hannun, and A. Abdul Razzaq. “Air Pollution Resulted from Coal, Oil and Gas Firing in Thermal Power Plants and Treatment: A Review.” *In: IOP Conference. Series: Earth and Environmental Science*, 1002, 2022. https://doi.org/10.1088/1755-1315/1002/1/012008.

[3] A. Szklo, and R. Schaeffer. “Fuel specification, energy consumption and CO_2 emission in oil refineries.” *Energy*, 32 (2007): 1075–1092. https://doi.org/10.1016/j.energy.2006.08.008

[4] L. Strömberg. “Combustion in a CO_2/O_2 mixture for a CO_2 emission free process.” *Second Nordic Minisymposium on Carbon Dioxide Capture and Storage*, Göteborg, October 26, 2001.http://www.entek.chalmers.se/~anly/symp/symp2001.html

[5] M. Hayes, W. Steurs, S. Cossu, and H. Iglesias. “European Green Deal Policy Guide, Focus on the ‘Fit for 55’ package.” *KPMG International*, 2021. https://assets.kpmg.com/content/dam/kpmg/xx/pdf/2022/01/green-deal-policy-guide-web-2022.pdf

[6] M. Basha, S. Shaaid, and L. Al-hadhrami. “Impact of Fuels on Performance and Efficiency of Gas Turbine Power Plants.” *Energy Procedia*, vol. 14, pp. 558 – 565, 2012. https://doi.org/10.1016/j.egypro.2011.12.975

[7] S. Oyedepo, R. Fagbenle, and S. Adefila. “Modelling and Assessment of Effect of Operation Parameters on Gas Turbine Power Plant Performance using First and Second Laws of Thermodynamics.” *American Journal of Engineering and Applied Sciences*, vol. 10, no. 2, pp. 412-430, 2017. https://doi.org/10.3844/ajeassp.2017.412.430

[8] D. Burnes, and A. Camou. “Impact of Fuel Composition on Gas Turbine Engine Performance.” *Journal of Engineering for Gas Turbines and Power*, vol. 141, 2019. https://doi.org/10.1115/1.4044238

[9] N. Alhajeri, M. Dannoun, A. Alrashed, and A. Aly. “Environmental and economic impacts of increased utilization of natural gas in the electric power generation sector: Evaluating the benefits and trade-offs of fuel switching.” *Journal of Natural Gas Science and Engineering*, vol. 71, p. 102969, 2019. https://doi.org/10.1016/j.jngse.2019.102969

[10] H. Egware, and C. Kwasi-Effah. “A novel empirical model for predicting the carbon dioxide emission of a gas turbine power plant.” *Heliyon*, vol. 9, no. 3, p. e14645, 2023. https://doi.org/10.1016/j.heliyon.2023.e14645

[11] T. Gould, and C. McGlade. “The environmental case for natural gas.” *IEA* 2017. https://www.iea.org/commentaries/the-environmental-case-for-natural-gas.

[12] S. Algayyim, et al. “Influence of natural gas and hydrogen properties on internal combustion engine performance, combustion, and emissions: A review.” *Fuel*, vol. 362, p. 130844, 2024. https://doi.org/10.1016/j.fuel.2023.130844

[13] G. Choi, and M. Katsuki. “Advanced low NO combustion using highly preheated air.” *Energy Conversion and Management*, vol. 42, no. 5, pp. 639-652, 2024. https://doi.org/10.1016/S0196-8904(00)00074-1

[14] R. Sosa, et al. “Reduction of atmospheric emissions due to switching from fuel oil to natural gas at a power plant in a critical area in Central Mexico.” *Journal of the Air & Waste Management Association*, vol. 70, no. 10, pp. 1043–1059, 2020. https://doi.org/10.1080/10962247.2020.1808113

[15] M. Almaktar, and M. Shaaban. “Prospects of renewable energy as a non-rivalry energy alternative in Libya.” Renewable and Sustainable Energy Reviews, vol. 143, p. 110852, 2021.https://doi.org/10.1016/j.rser.2021.110852

[16] The International Energy Agency (IEA). https://www.iea.org/countries/libya/electricity

[17] Y. Nassar, et al “Estimation of Environmental Damage Costs from CO2e Emissions in Libya and the Revenue from Carbon Tax Implementation.” *Low Carbon Economy*, vol. 8, pp. 118-132, 2017. https://doi.org/10.1007/s10668-021-01248-9

[18] ENKA, “Tripoli West 671 MW Simple Cycle Power Plant”. https://www.enka.com/portfolio-item/tripoli-west-671-mw-simple-cycle-power-plant-2/

[19] J. Smith, H. Van Ness, M. Abbott, and M. Swihart. “*Introduction to Chemical Engineering Thermodynamics*”. McGrow-Hill Book, Eighth Edition, 2018.

[20] S. Özturk, and M. Eyriboyun. “NOx Formation in Combustion of Natural Gases Used in Turkey under Different Conditions.” *Journal of Thermal Science and Technology*, vol. 30, no. 2, pp. 95-102, 2010.

[21] R. Pavri, and G. Moore. “Gas Turbine Emissions and Control.” *GE Power Systems GER-4211* (03/01). https://www.eplptx.com/wp-content/uploads/2020/05/Gas-Turbine-Emissions-and-Control.pdf

[22] G. Löffler, et al “NOx Formation in Natural Gas Combustion—A New Simplified Reaction Scheme for CFD Calculations.” *Fuel*, vol. 85, pp. 513–523, 2005. https://doi.org/10.1016/j.fuel.2005.07.012

[23] C. Hwang, S. Hyun, and C. Lee. “Effects of heat loss on NOx emission in the post flame region of premixed CH4—Air combustion.” *Energy Fuels*, vol. 22, no. 2, pp. 996–1003, 2008. https://doi.org/10.1021/ef700604f

[24] A. Qarash, A. Atia, and A. Elshoboky. “Analysis of Natural Gas Properties (a Case Study of Wafa Gas Field).” *2nd Conference for Engineering Sciences and Technology*. (2019) Sabratha –Libya: 29- 31.

[25] P. Kazemiani-Najafabadi, and E. Rad. “Introducing an Optimum Gas Turbine Inlet Temperature (TIT) Based on a 4E Analysis: A Case Study of Northeastern Iran.” *Journal of Thermal Analysis and Calorimetry*, vol. 146, pp. 403–413, 2021. https://doi.org/10.1007/s10973-020-09955-7

[26] D. Abam, and M. Moses. “Computer Simulation of a Gas Turbine Performance.” *Global Journal of Researchers in Engineering*, vol. 11, no. 1, 2011. https://scispace.com/pdf/computer-simulation-of-a-gas-turbine-performance-51jpfobe95.pdf

[27] A. Shukla, and O. Singh. ‘Effect of Compressor Inlet Temperature & Relative Humidity on Gas Turbine Cycle Performance.” *International Journal of Scientific & Engineering Research*, vol. 5, no. 5. 2014. https://www.scribd.com/document/491587282/Effect-of-Compressor-Inlet-Temperature-Relative-Humidity-on-Gas

[28] A. Abdullah, et al. "Optimization Study of Converting the Derna Steam Power Plant to a Combined Gas/Steam Cycle." *waujpas*, vol. 4, no. 2, pp. 47-58, 2026. https://doi.org/10.63318/waujpasv4i2_07

[29] F. Montasser, and N. Eshoul, “Enhancing the Performance of West Tripoli Power Plant Gas Turbine Unit using Absorption Chiller.” *African Journal of Advanced Pure and Applied Sciences (AJAPAS)*, vol. 3, no. 1, pp. 300-314, 2024. https://doi.org/10.65418/ajapas.v3i1.745

[30] A. De Sa, and S. Al Zubaidy. “Gas turbine performance at varying ambient temperature.” *Applied Thermal Engineering*, vol. 31, pp. 2735-2739, 2011. https://doi.org/10.1016/j.applthermaleng.2011.04.045

[31] M. Elgohary, I. Seddiek. “Comparison between Natural Gas and Diesel Fuel Oil Onboard Gas Turbine Powered Ships.” *JKAU: Mar. Sci.*, vol. 2, no. 23, pp. 109-127, 2012. https://doi/10.4197/Mar.23-2.7

[32] A. Kraedegh. “Performance and Emission Assessment of Solar-Assisted Feed-Water Heating in Steam Power Plants.” *Journal of Technology Research (JTR)*, vol. 3, no. 2, pp. 81-91, 2025. https://doi.org/10.26629/jtr.2025.10

[33] M. Rahman, et al. “Influence of Operation Conditions and Ambient Temperature on Performance of Gas Turbine Power Plant.” *Advanced Materials Research*, pp. 189-193, 2011. https:// www.scientific.net/AMR.189-193.3007

[34] A. Kumar, et al. “Thermodynamic Analysis of Gas Turbine Power Plant.” *International Journal of Innovative Research in Engineering & Management*, vol. 4, no. 3, 2017. https://www.scientific.net/AMR.189-193.3007

[35] I. Mangir, et al. “The Impact of Fuel Type on the Economic, Environmental, and Energy Performance of Power Generation Plants: A Case Study of the Ubari Gas Power Plant-Libya.” *Wadi Alshatti University Journal of Pure and Applied Sciences*, vol. 4, no. 2, pp. 145-154, 2026. https://doi.org/10.63318/waujpasv4i2_17

[36] M. Hamad, M. Hamdan, and N. Fathi. "Energy and Exergy Analysis of an Al-Hussein Steam Power Plant in Jordan Using Engineering Equation Solver (EES)." *Pure and Applied Sciences*, vol. 4, no. 1, pp. 199-209, 2026. https://doi.org/10.63318/waujpasv4i1_21

[37] D. Jarvis, et al. “Nitrogen dioxide. In: WHO Guidelines for Indoor Air Quality: Selected Pollutants. Geneva: World Health Organization; 2010. 5. Available at: https://www.ncbi.nlm.nih.gov/books/NBK138707/.

[38] S. Hill, and L. Smoot. “Modeling of nitrogen oxides formation and destruction in combustion systems.” *Progress in Energy and Combustion Science*, vol. 26, pp. 417-458, 2000. https://doi.org/10.1016/S0360-1285(00)00011-3

[39] Z. Wang, and X. Yang “NOx Formation Mechanism and Emission Prediction in Turbulent Combustion: A Review.” *Applied Science*, vol. 14, p. 6104, 2024. https://doi.org/10.3390/app14146104

[40] T. Nguyen, et al. “A numerical study on NOx formation behavior in a lean-premixed gas turbine combustor using CFD-CRN method.” *Journal of Mechanical Science and Technology*, vol. 33, no. 10, pp. 5051-5060, 2019. https://doi.org/10.1007/s12206-019-0944-3