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Thermodynamics performance analysis of a Gas Turbine under Variable Ambient Temperatures: A case Study of South Tripoli Power Plant with Cycle Improvement Techniques

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تحليل الأداء الديناميكي الحراري لتوربين غازي في ظل درجات حرارة محيطة متغيرة: دراسة حالة لمحطة توليد الطاقة جنوب طرابلس مع تقنيات تحسين الدورة

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Abstract:
This study investigates the effect of ambient air temperature on the performance of a gas turbine power plant based on real operational data from the South Tripoli Gas Power Plant. The analysis focuses on key performance parameters including net power output, thermal efficiency, and back work ratio under different seasonal conditions. A thermodynamic evaluation of the real cycle was conducted, followed by a comparative analysis with two enhancement techniques: regeneration and the fully improved cycle (intercooling, reheat, and regeneration), while maintaining the same operating conditions to ensure a fair comparison. Concludes that advanced cycle modifications are effective in enhancing gas turbine performance, particularly in hot-climate regions such as Libya.

Keywords: South Tripoli Power Plant, Brayton Cycle, ambient air temperature, Regenerator Effectiveness

الملخص
تبحث هذه الدراسة تأثير درجة حرارة الهواء المحيط على أداء محطة توليد الطاقة بالغاز، وذلك بالاستناد إلى بيانات تشغيلية حقيقية من محطة جنوب طرابلس لتوليد الطاقة بالغاز. ويركز التحليل على معايير الأداء الرئيسية، بما في ذلك صافي إنتاج الطاقة، والكفاءة الحرارية، ونسبة العمل العكسي، في ظل ظروف موسمية مختلفة. وقد أُجري تقييم ديناميكي حراري للدورة الفعلية، تلاه تحليل مقارن باستخدام تقنيتين للتحسين: التجديد، والدورة المحسّنة بالكامل (التبريد البيني، وإعادة التسخين، والتجديد)، مع الحفاظ على نفس ظروف التشغيل لضمان مقارنة عادلة. وتخلص الدراسة إلى أن التعديلات المتقدمة على الدورة فعّالة في تحسين أداء توربينات الغاز، لا سيما في المناطق ذات المناخ الحار مثل ليبيا.

الكلمات المفتاحية: محطة جنوب طرابلس لتوليد الطاقة، دورة برايتون، درجة حرارة الهواء المحيط، فعالية المُجدِّد

Introduction

Electric power generation plays a fundamental role in supporting economic development and improving the quality of life in modern societies, as industrial, commercial, and residential sectors heavily depend on the availability of reliable and continuous electrical energy [1], [2]. In many countries, electricity demand increases significantly during the summer season due to the intensive use of air-conditioning systems and various cooling technologies. This requires power plants to operate at high-capacity levels, especially during peak demand periods.

Gas turbines operate based on the principles of the Brayton cycle, where atmospheric air is compressed in the compressor, then mixed with fuel in the combustion chamber where combustion takes place. This process produces high-temperature and high-pressure gases that expand through the turbine to generate mechanical power, which is subsequently converted into electrical energy using a generator [1], [2].

This issue becomes more critical in hot-climate regions such as Libya, where high ambient temperatures coincide with peak electricity demand. Therefore, analyzing and improving gas turbine performance under such conditions is essential to ensure reliable and continuous power generation.

1.1 Research problem

Electricity demand reaches its highest levels during the summer season due to the heavy reliance on cooling systems, particularly air-conditioning units. At the same time, gas turbine power plants operating in hot regions experience a noticeable decline in performance due to high ambient air temperatures. As the temperature increases, the density of the air entering the compressor decreases, leading to a reduction in the mass flow rate through the compression and expansion stages. This negatively affects the generated power and thermal efficiency, creating a gap between electricity supply and demand during peak periods. Therefore, the main problem addressed in this study is the deterioration of gas turbine performance under high ambient temperature conditions and the need to evaluate effective methods for performance improvement under such operating conditions.

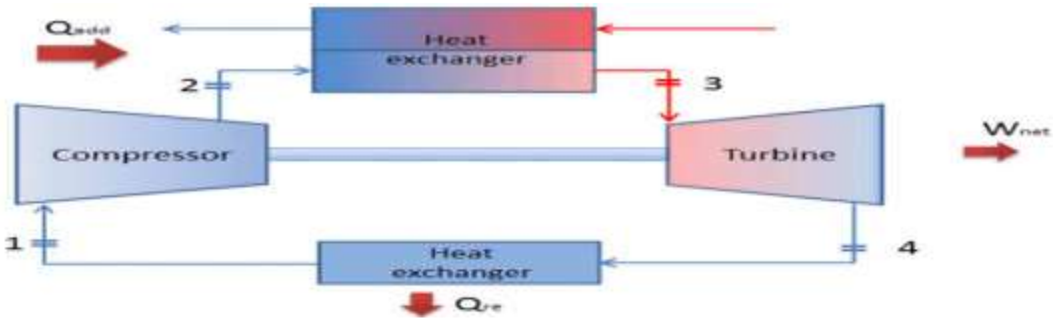
1.2 Methodology

Techniques for Improving the Thermal Performance of the Brayton Cycle in Gas Turbines. Gas turbines are subject to several deviations from ideal behavior, arising from factors such as mechanical friction, pressure losses, non-isentropic performance of compressors and turbines, as well as the influence of ambient environmental conditions including temperature and humidity. These factors collectively result in a reduction of both the net power output and the thermal efficiency when compared to their theoretical values [3],[5],[6]. The analysis of the actual Brayton cycle by identifying the main sources of deviation from the ideal cycle and examining the most important operational and design improvements that have been historically developed to enhance gas turbine efficiency and power output [3], [4], [7]. Special attention is given to performance improvement techniques suitable for high ambient temperature environments [4],[7]. This analysis constitutes a fundamental step toward understanding the real thermal performance of gas turbine power plants and serves as a basis for the subsequent case study of the South Tripoli Power Plant.

1.3 Brayton cycle

The Brayton cycle, originally developed by George Brayton during the nineteenth century, forms the thermodynamic basis of modern gas turbine power plants. The ideal cycle assumes steady-flow operation, ideal-gas behavior, isentropic compression and expansion, constant-pressure heat addition, and negligible pressure losses.[1] The thermal efficiency of the ideal Brayton cycle can be expressed as:

$$\eta_{Brayton} = 1 - \frac{1}{r_p^{(\gamma-1)/\gamma}} \tag{1}$$



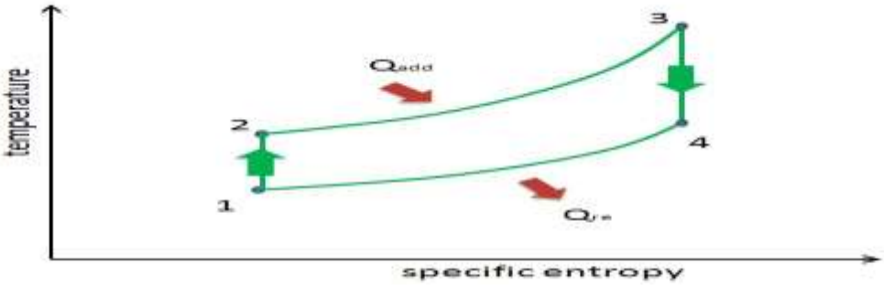


Figure (1): Schematic and T–s Diagrams of the Simple Brayton Cycle [8]

The Brayton cycle consists of four principal processes:

Process (1–2): Isentropic Compression in the Compressor: The specific work required by the compressor is expressed as:

$$w_c = h_2 - h_1 \tag{2}$$

Process (2–3): Constant Pressure Heat Addition in the Combustion Chamber: The heat supplied to the working fluid is given by:

$$q_{in} = h_3 - h_2 \tag{3}$$

Process (3–4): Isentropic Expansion in the Turbine: The turbine specific work is expressed as:

$$w_t = h_3 - h_4 \tag{4}$$

Process (4–1): Constant Pressure Heat Rejection: The heat rejected from the cycle is given by:

$$q_{out} = h_4 - h_1 \tag{5}$$

The net specific work output of the Brayton cycle is:

$$w_{net} = w_t - w_c \tag{6}$$

The thermal efficiency of the Brayton cycle can also be expressed as:

$$\eta_{th} = \frac{w_{net}}{q_{in}} \tag{7}$$

Back Work Ratio of the Brayton Cycle is :

$$BWR = \frac{w_c}{w_t} \tag{8}$$

1.4 Actual Brayton Cycle:

In practical gas turbines, compressor and turbine processes are not perfectly isentropic because of friction, pressure losses, and other irreversibilities. Consequently, the actual cycle produces lower net work and lower thermal efficiency than the ideal cycle [3]. The isentropic efficiency of the compressor is defined as:

$$\eta_c = \frac{h_{2s} - h_1}{h_{2a} - h_1} \tag{9}$$

Similarly, the isentropic efficiency of the turbine is expressed as:

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{4s}} \tag{10}$$

The Actual Brayton Cycle represented in Fig (2)

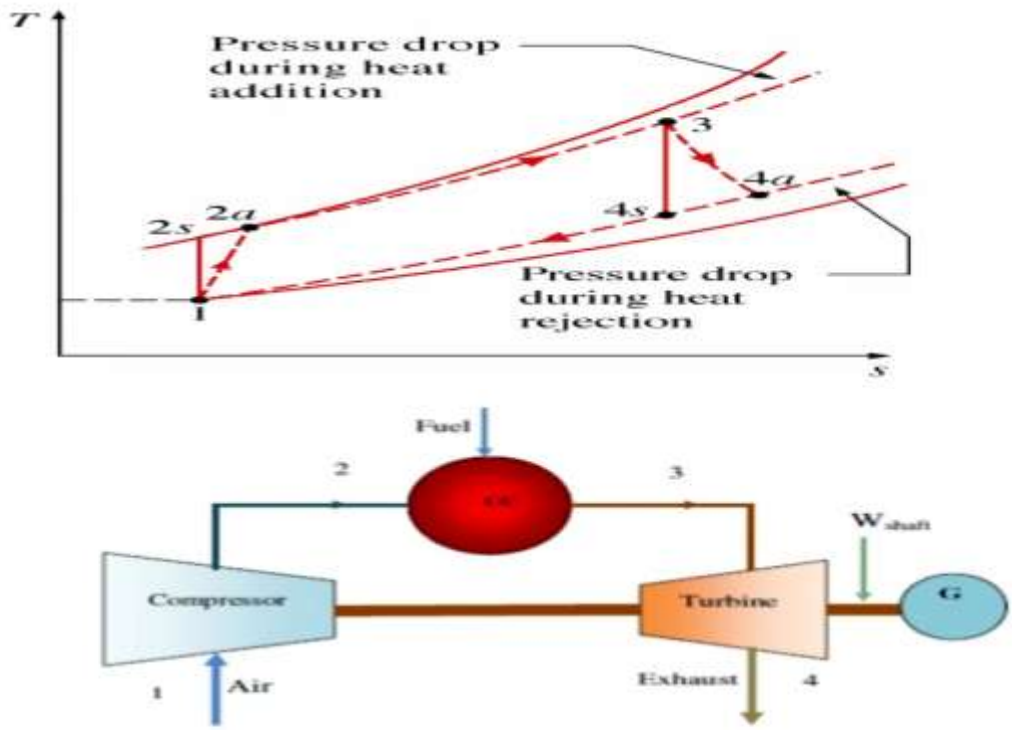


Figure (2): Schematic and T–s Diagrams of the Actual Brayton Cycle [1]

1.5 The Brayton Cycle with Regeneration

Regeneration improves cycle efficiency by recovering part of the thermal energy contained in the turbine exhaust gases to preheat the compressed air before combustion. The effectiveness of the regenerator is defined as:

$$\epsilon = \frac{q_{regen,act}}{q_{regen,max}} = \frac{h_5 - h_2}{h_4 - h_2} \tag{11}$$

Or,under the cold-air-standard assumptions,

$$\epsilon \cong \frac{T_5 - T_2}{T_4 - T_2} \tag{12}$$

The required heat input becomes

$$q_{in} = h_3 - h_5 \tag{13}$$

Higher regenerator effectiveness reduces fuel consumption and increases thermal efficiency without significantly affecting turbine or compressor work.

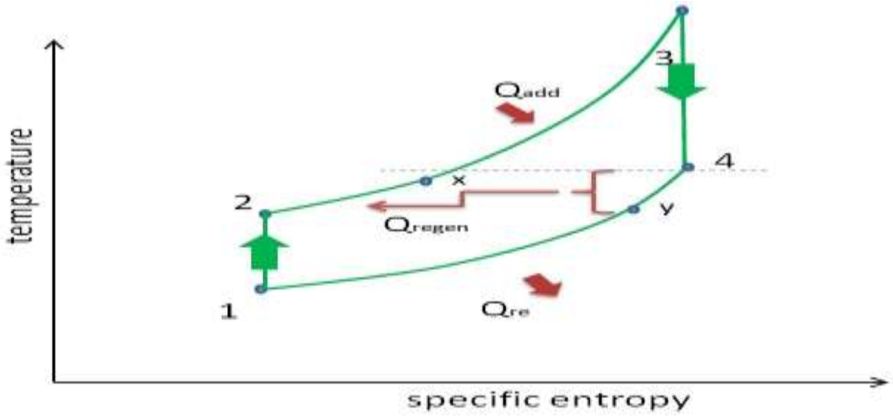


Figure (3): The T–S diagram of Brayton Cycle with Regeneration [8]

1.6 The Brayton Cycle with Reheat

The reheat cycle increases turbine work by dividing the expansion process into two stages separated by an additional combustion chamber operating at approximately constant pressure. For maximum turbine work output, the expansion pressure ratio is equally divided between the turbine stages and can be expressed as:

$$r_{p_1} = r_{p_2} = \sqrt{r_p} \tag{14}$$

The total turbine work output for the reheat cycle is expressed as:

$$w_t = (h_3 - h_4) + (h_5 - h_6) \tag{15}$$

The total heat supplied to the reheat cycle is expressed as:

$$q_{in} = (h_3 - h_2) + (h_5 - h_4) \tag{16}$$

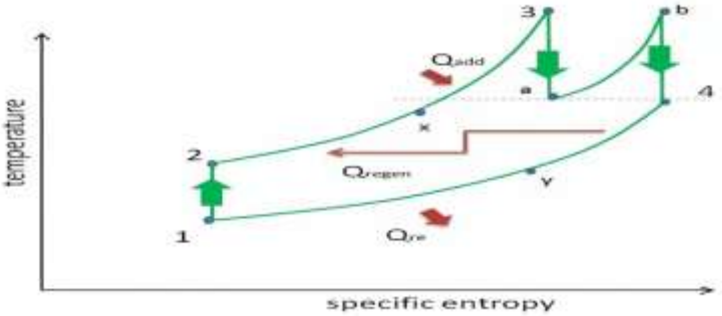


Figure (4): The T–S diagram of Brayton Cycle with Reheat [8]

Reheating increases turbine work and consequently improves the net work output of the cycle.

1.7 The Brayton Cycle with Intercooling

Intercooling reduces compressor work by dividing the compression process into two stages separated by an intermediate cooling process.

The total compressor work for the intercooling cycle is expressed as:

$$w_c = (h_2 - h_1) + (h_4 - h_3) \tag{17}$$

The optimum pressure ratio distribution is

$$r_{p_1} = r_{p_2} = \sqrt{r_p} \tag{18}$$

Although intercooling decreases compressor work and increases net work output, it is generally combined with regeneration or reheating to achieve significant improvements in thermal efficiency.

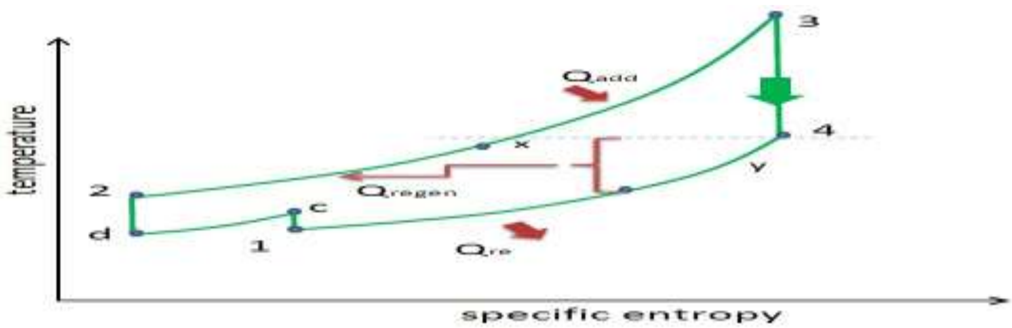


Figure (5): The T–S diagram of Brayton Cycle with Intercooling [8]

1.8 Brayton Cycle with Intercooling, Reheating, and Regeneration

The fully improved Brayton cycle combines intercooling, reheating, and regeneration to enhance both the power output and thermal efficiency of gas turbine systems. Unlike individual enhancement techniques, this integrated configuration simultaneously reduces compressor work, increases turbine work, and decreases fuel consumption, making it one of the most efficient practical Brayton cycle configurations [3], [4]. Intercooling divides the compression process into two stages separated by an intercooler, reducing the average compression temperature and consequently lowering compressor work. Reheating divides the turbine expansion into two stages with intermediate combustion, increasing turbine work without exceeding the allowable turbine inlet temperature. Regeneration utilizes the thermal energy of the turbine exhaust gases to preheat the compressed air before combustion, thereby reducing the required fuel input. The combination of these three techniques improves the cycle in three aspects:

- Reduction of compressor work through intercooling.
- Increase of turbine work through reheating.
- Improvement of thermal efficiency through regeneration.

As a result, the fully improved Brayton cycle provides significantly higher net work output and thermal efficiency than the simple and regenerative Brayton cycles, particularly under high ambient temperature conditions.

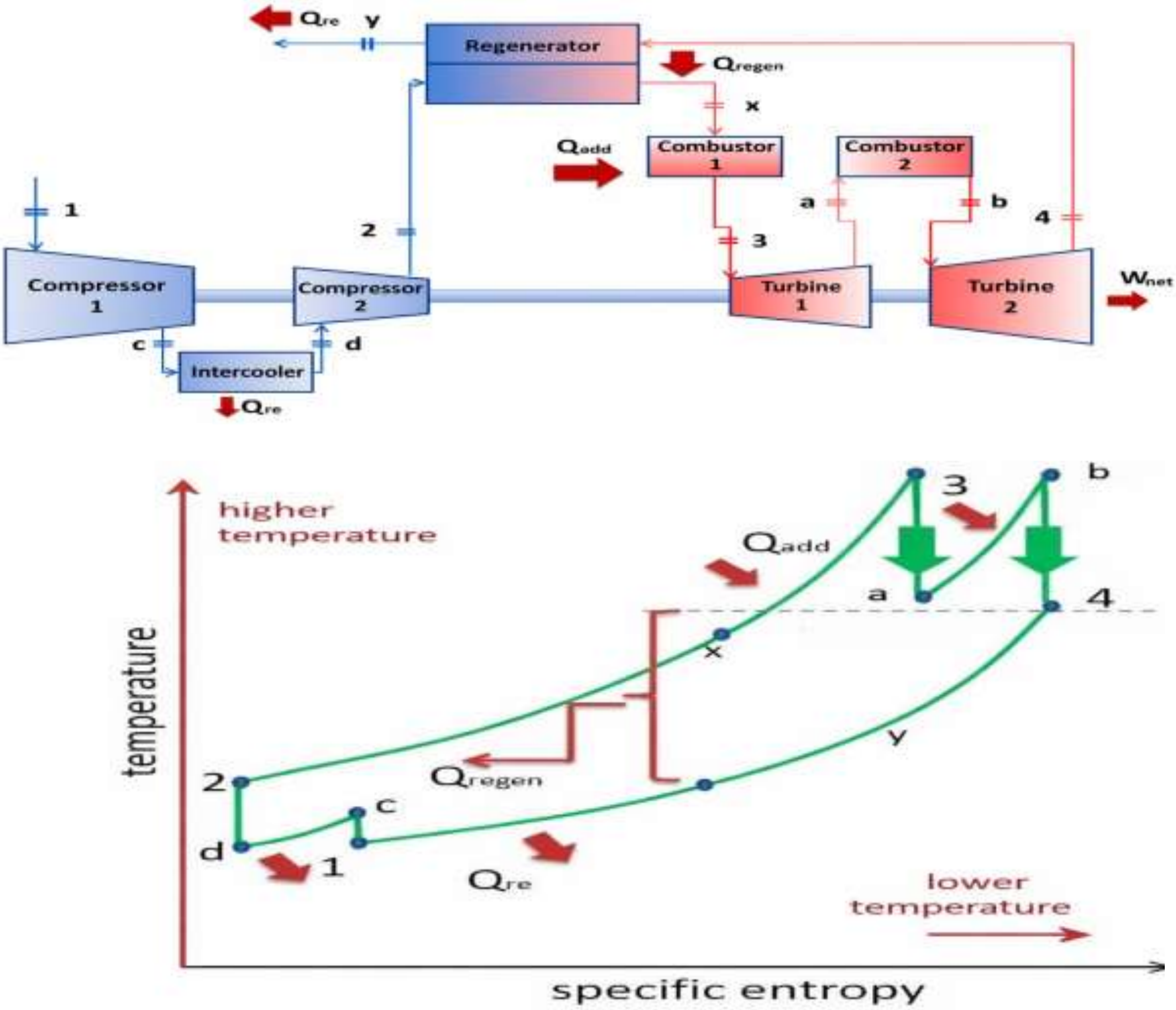


Figure (6): Schematic and T–s Diagrams of the Brayton Cycle with Intercooling, Reheating, and Regeneration [8]

The corresponding T–s diagram illustrates two-stage compression with intercooling, two-stage expansion with reheating, and regenerative heat recovery before combustion. These combined modifications reduce the compressor work area, increase the turbine work area, and decrease the external heat input, leading to the highest overall cycle efficiency among the configurations investigated in this study.

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2. Mathematical Model

Based on the thermodynamic principles the following mathematical relations are employed to evaluate the performance of the South Tripoli gas turbine power plant under different operating conditions.

2.1 Turbine Analysis

$$\dot{W}_t = \dot{m}_{exh}(h_3 - h_4) \tag{19}$$

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{4s}} \tag{20}$$

2.2 Compressor Analysis

$$\dot{m}_a = \dot{m}_{exh} - \dot{m}_f \tag{21}$$

$$\dot{W}_c = \dot{m}_a(h_2 - h_1) \tag{22}$$

$$\eta_c = \frac{h_{2s} - h_1}{h_{2a} - h_1} \tag{23}$$

2.3 Combustion Chamber Analysis

$$Q_{in} = \dot{m}_f \times LHV \tag{24}$$

2.4 Cycle Performance

The net power output is

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_c \tag{25}$$

Thermal efficiency

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \tag{26}$$

Back Work Ratio

$$BWR = \frac{\dot{W}_c}{\dot{W}_t} \tag{27}$$

Heat rejected

$$\dot{Q}_{out} = \dot{m}_{exh}(h_4 - h_1) \tag{28}$$

2.5 Regenerative Brayton Cycle

$$\epsilon = \frac{h_5 - h_2}{h_4 - h_2} \tag{29}$$

$$(\dot{m}_f + \dot{m}_a)(h_3 - h_5) = \dot{m}_f * LHV \tag{30}$$

2.6 Fully Improved Brayton Cycle

$$\dot{w}_{c_{total}} = 2 * \dot{m}_a * (h_2 - h_1) \quad (31)$$

$$\dot{Q}_{in_{total}} = \dot{Q}_{in1} + \dot{Q}_{in2} \quad (32)$$

$$\dot{w}_{t_{total}} = \dot{w}_{t1} + \dot{w}_{t2} \quad (33)$$

$$BWR = \frac{\dot{w}_{c_{total}}}{\dot{w}_{t_{total}}} \quad (34)$$

$$\dot{w}_{net} = \dot{w}_{t_{total}} - \dot{w}_{c_{total}} \quad (35)$$

$$\eta_{th} = \frac{\dot{w}_{net}}{\dot{Q}_{in_{total}}} \quad (36)$$

3. Results and Discussion

The applied thermodynamic analysis of the gas turbine performance at the South Tripoli Gas Power Plant based on actual operational data collected during selected periods representing winter and summer conditions.

Table (1): Measured Operating Parameters of the South Tripoli Gas Power Plant

Indicators	Dec	Jan	Feb	Jun	Jul	Aug
The temperature at compressor inlet(C°)	15	8	10	30	37	45
The pressure at compressor (KPa)	101.5	101.0	101.00	101.6	101.5	101.1
Pressure ratio	11.4	11.4	11.5	11.3	11.3	11.2
the temperature at turbine inlet(C°)	990	990	990	989	990	988
Fuel flow (Kg/s)	7.65	7.902	7.83	7.09	6.82	6.51
Exhaust flow()(Kg/s)	387.8	396.48	394	369	360.2	350.2
Exhaust temperature (C°)	493	491.04	491.6	498	500.4	503.2
Gross electrical output (KW)	94770	98690	97570	85950	81570	76480

3.1 Thermodynamic Analysis

Mathematical model to the measured data yields the following results.

Table (2): Thermodynamic Performance Parameters of the Real Brayton Cycle under Different Ambient Temperatures.

Parameter	Dec	Jan	Feb	Jun	Jul	Aug
$\dot{W}_t$ (Mw)	220.025	225.8	224.231	206.906	201.454	193.960
η_t (%)	85.85	86.169	85.9	85.129	84.62	84.37
$\dot{m}_a$ (kg/s)	380.15	388.578	386.17	361.91	353.38	343.69
$\dot{W}_c$ (Mw)	125.255	127.11	126.661	120.956	119.884	117.480
η_c (%)	87.97	86.514	87.363	90.7	91.36	92.05
$\dot{Q}_{in}$ (Mw)	298.503	308.336	305.526	276.651	266.116	254.020
η_{th} (%)	31.75	32	31.935	31.07	30.65	30.108
BWR (%)	56.93	56.29	56.49	58.46	59.51	60.57
$\dot{Q}_{out}$ (Mw)	192.574	198.819	196.941	179.698	173.825	167.254

Effect of Ambient Air Temperature on Gas Turbine Performance:

Variation of Compressor and Turbine Work with Ambient Air Temperature

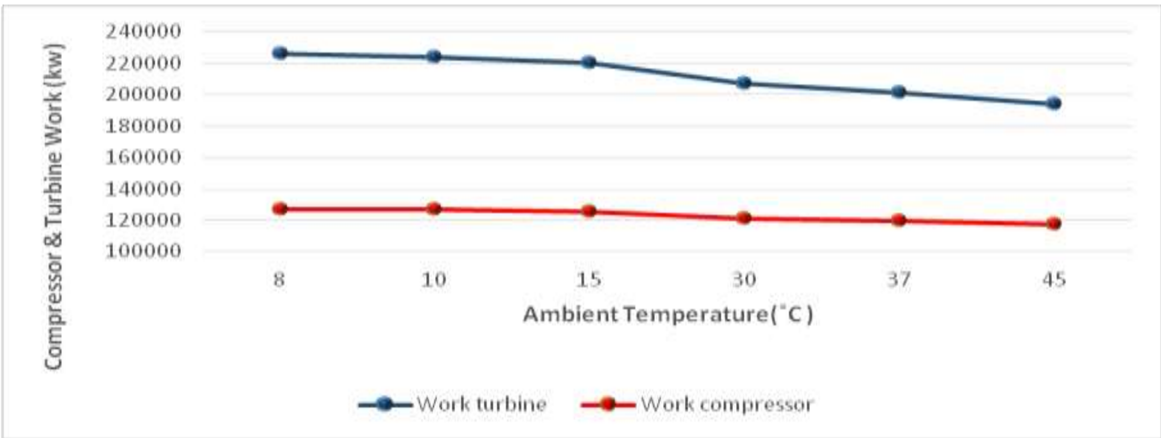

Figure (7): Compressor & Turbine Work vs Ambient Temperature

Fig (7) presents the variation of turbine and compressor work with ambient air temperature. As the temperature increases, the work produced by the turbine decreases noticeably due to the reduction in the available expansion energy. The compressor work also shows a slight decrease; however, the reduction in turbine work is more pronounced. This difference leads to a reduction in the network output of the cycle.

Effect of Ambient Air Temperature on Gross Electrical Power Output:

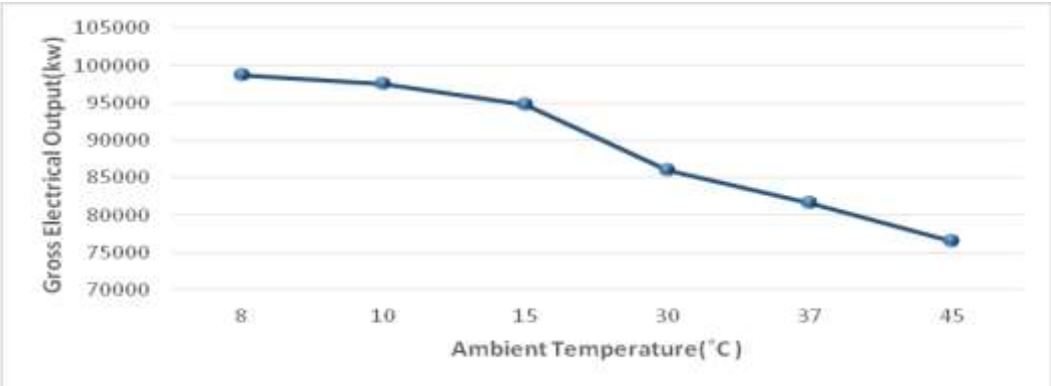

Figure (8): Gross Electrical Output vs Ambient Temperature

Fig.(8) illustrates the relationship between ambient air temperature and the gross electrical power generated by the gas turbine. The results indicate a steady decline in power output as the ambient temperature rises. The reduction in turbine work directly influences the generator output, leading to lower electrical production at higher environmental temperatures.

Effect of Ambient Air Temperature on Back Work Ratio:

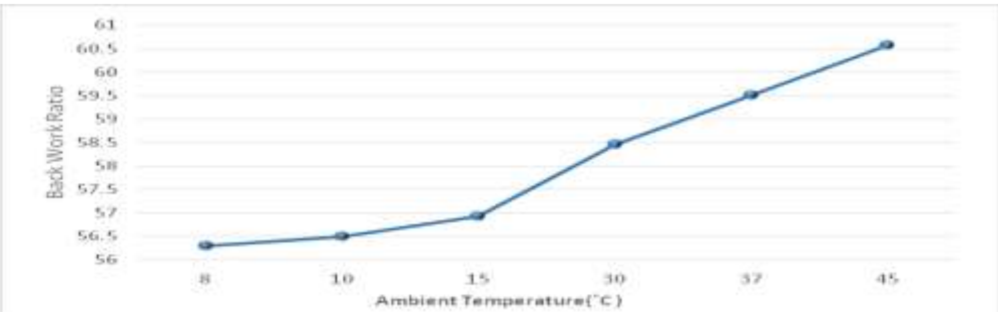

Figure (9): Back Work Ratio vs Ambient Temperature

Fig(9) shows the change in back work ratio with increasing ambient air temperature. The results reveal a gradual increase in this ratio as temperature rises. This indicates that a larger fraction of the turbine work is required to operate the compressor, which reduces the effective useful work available from the system.

Effect of Ambient Air Temperature on Thermal Efficiency:

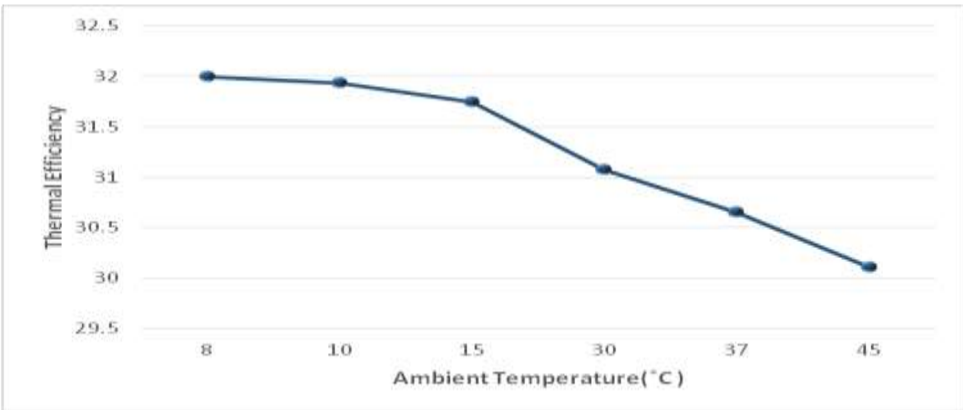

Figure (10): Thermal Efficiency vs Ambient Temperature

This chart presents the variation of thermal efficiency with ambient air temperature. The results show that the efficiency gradually decreases as the temperature increases. This behavior reflects the reduction in the effective energy conversion within the cycle as operating conditions deviate from the optimal thermal environment.

Effect of Ambient Air Temperature on Mass Flow Rate:

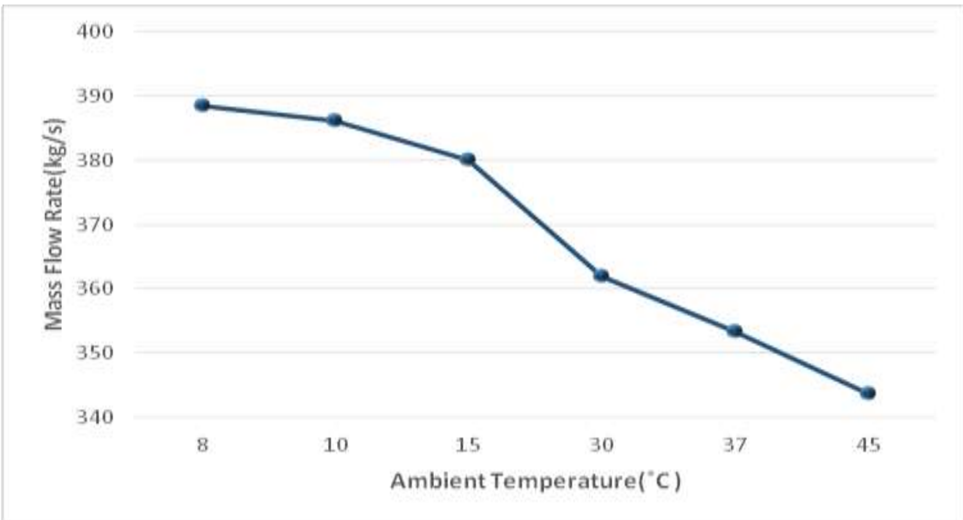

Figure (11): Mass Flow Rate vs Ambient Temperature

The figure (11) demonstrates how the mass flow rate varies with ambient air temperature. As temperature increases, the mass flow rate decreases progressively. This trend affects the overall operating conditions of the turbine and contributes to the observed variations in power output and system performance.

3.2 Analysis of Regeneration and Fully Improved Cycle Model

To evaluate possible performance improvements, the regenerative Brayton cycle and the fully improved Brayton cycle were analyzed under the same operating conditions adopted for the actual cycle. Maintaining identical operating parameters ensures that the observed performance differences result solely from the implemented cycle modifications.

3.3 Regeneration Cycle Model Analysis

The regenerative cycle utilizes a heat exchanger to recover thermal energy from turbine exhaust gases and transfer it to the compressed air before combustion. This process reduces the required fuel input while maintaining approximately the same turbine inlet temperature.

Table (3): Thermodynamic Performance Parameters of the Regenerative Brayton Cycle under Different Ambient Temperatures.

Parameter	Dec	Jan	Feb	Jun	Jul	Aug
$\dot{m}_f\ (kg/s)$	6.1127	6.293	6.242	5.7146	5.54	5.3182
$\dot{m}_{exh}\ (kg/s)$	386.263	394.871	392.412	367.625	358.92	349
$\dot{W}_t\ (Mw)$	219.153	224.88	223.328	206.135	200.738	193.3
$\dot{Q}_{in}\ (Mw)$	238.517	245.552	243.562	222.983	216.170	207.516
$\eta_{th}\ (\%)$	39.37	39.82	39.69	38.2	37.4	36.54
BWR(%)	57.15	56.52	56.72	58.68	59.722	60.78
$\dot{W}_{net}\ (\mathrm{Mw})$	93.898	97.773	96.666	85.178	80.854	75.820

The results indicate that regeneration significantly improves thermal efficiency because less external heat is required in the combustion chamber. However, the reduction in total fuel flow produces a slight decrease in turbine work and consequently a small reduction in net power output.

3.5 Fully Improved Cycle Model (Intercooling + Reheat + Regeneration)

The fully improved cycle combines intercooling, reheating, and regeneration to maximize overall cycle performance. Intercooling decreases compressor work, reheating increases turbine work, and regeneration reduces fuel consumption by utilizing waste heat from the turbine exhaust gases.

Table (4): Thermodynamic Performance Parameters of the Fully Improved Brayton Cycle under Different Ambient Temperatures.

Parameter	Dec	Jan	Feb	Jun	Jul	Aug
$\dot{W}_{c_{total}}\ (Mw)$	103.878	105.356	104.906	100.479	99.622	98.240
$\dot{m}_{f1}\ (kg/s)$	5.01	5.1523	5.113	4.688	4.544	4.371
$\dot{m}_{exh1}\ (kg/s)$	385.16	393.73	391.283	366.598	357.924	348.061
$\dot{W}_{t1}\ (Mw)$	126.08	129.364	128.534	118.534	115.1	111.102
$\dot{Q}_{in1}\ (Mw)$	195.49	201.042	199.509	182.925	177.306	170.552
$\dot{m}_{f2}\ (kg/s)$	3.26	3.343	3.322	3.0632	2.9743	2.871
$\dot{m}_{exh2}\ (kg/s)$	388.42	397.073	394.605	369.66	360.9	350.932
$\dot{W}_{t2}\ (Mw)$	127.147	130.462	129.626	119.525	116.056	112.019
$\dot{Q}_{in2}\ (Mw)$	127.205	130.443	129.624	119.526	116.057	112.018
$\dot{W}_{t_{total}}\ (Mw)$	253.227	259.826	258.161	238.059	231.157	223.122
$\dot{Q}_{in_{total}}\ (Mw)$	322.695	331.486	329.133	302.451	293.364	282.571
$\eta_{th}\ (\%)$	46.28	46.6	46.563	45.49	44.837	44.195
BWR (%)	41.022	40.55	40.64	42.21	43.097	44.03
$\dot{W}_{net}\ (Mw)$	149.348	154.470	153.254	137.580	131.535	124.881

Compared with the actual and regenerative cycles, the fully improved cycle provides the highest net power output and thermal efficiency together with the lowest back work ratio.

3.7 Comparative Evaluation of Cycle Performance under Varying Ambient Conditions

Comparison of Back Work Ratio for Different Cycles:

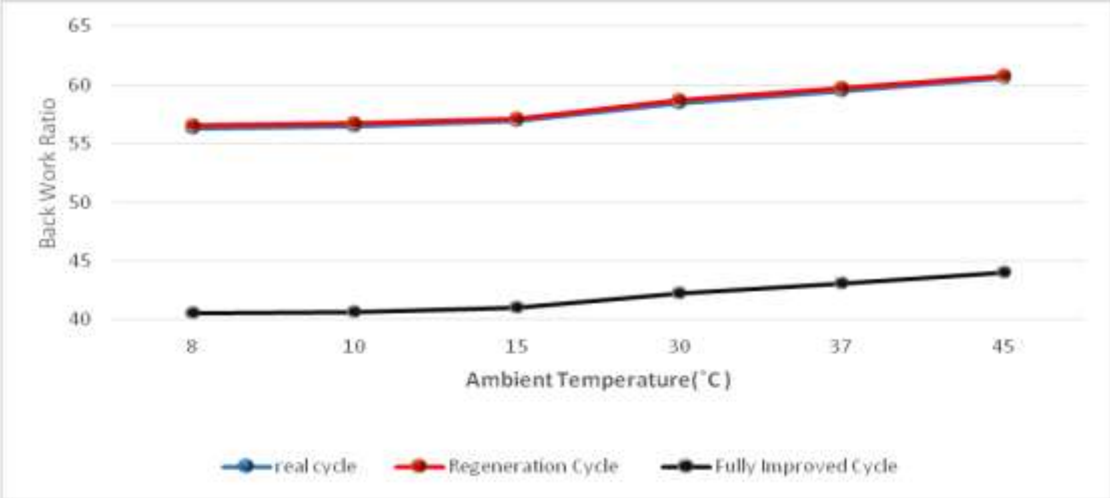

Figure (12): Comparison of Back Work Ratio Variation with Ambient Temperature for Different Cycles

Figure (12) shows that the back work ratio increases with ambient temperature for all investigated configurations. Nevertheless, the fully improved cycle maintains the lowest BWR because intercooling decreases compressor work while reheating increases turbine work.

Comparison of Thermal Efficiency for Different Cycles

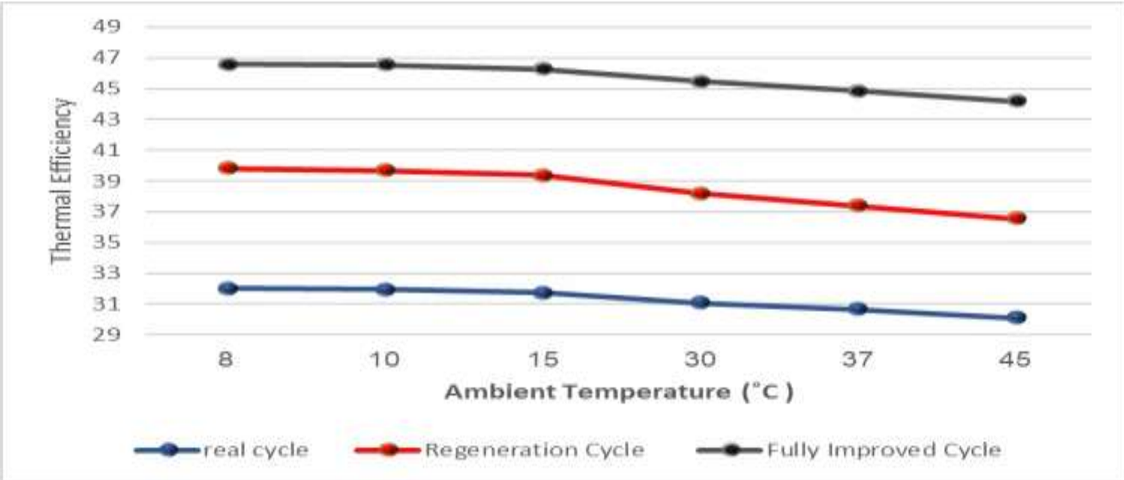

Figure (13): Comparison of Thermal Efficiency Variation with Ambient Temperature for Different Cycles

The figure shows that thermal efficiency gradually decreases as the ambient air temperature increases in all cases. This is due to the reduction in air density, which leads to a lower mass flow rate entering the cycle and consequently a decrease in net work output. In the regeneration cycle, an improvement in thermal efficiency is observed compared to the real cycle. This is mainly attributed to the reduction in heat input (Qin) by utilizing exhaust heat, which lowers fuel consumption while keeping the network output relatively close to its original value. In the fully improved cycle, the highest thermal efficiency is recorded. This is due to the combined effect of several factors, most importantly the reduction in compressor work through intercooling, the increase in turbine work via reheating, and the reduction in fuel consumption using regeneration. These improvements collectively lead to a significant increase in net work output and better energy utilization within the cycle.

Comparison of Gross Electrical Output for Different Cycles:

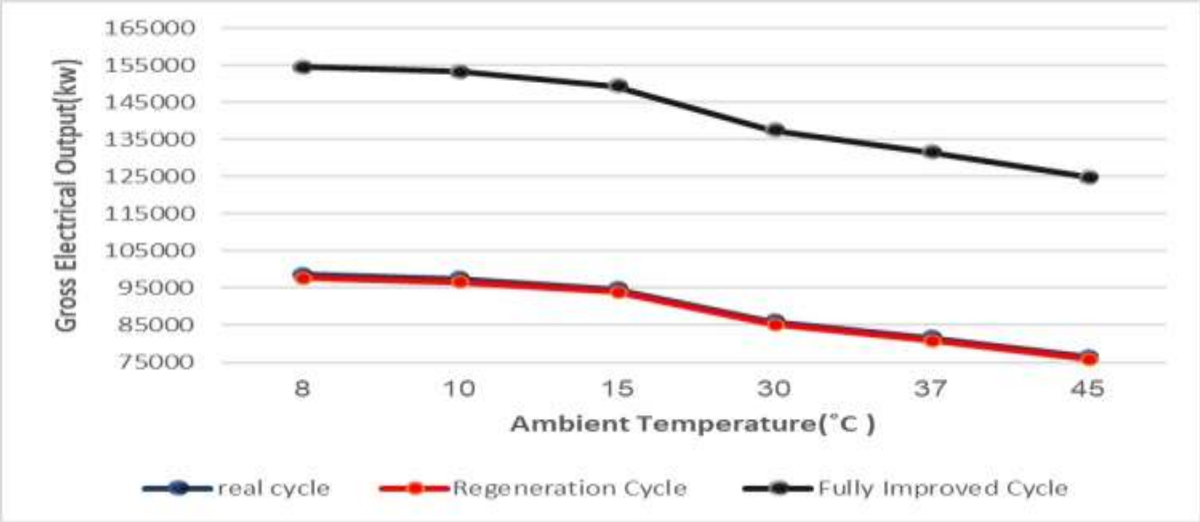

Figure (14): Comparison of Gross Electrical Output Variation with Ambient Temperature for Different Cycles

Figure (14) indicates that electrical power decreases with increasing ambient temperature for all cycles. However, the fully improved cycle consistently produces the highest power output owing to reduced compressor work and increased turbine work. In contrast, the regenerative cycle exhibits only a slight improvement in efficiency with a marginal reduction in power output due to lower fuel consumption.

3.8 Discussion of Results

The obtained results demonstrate that ambient air temperature has a significant influence on the thermodynamic performance of gas turbines. As the compressor inlet temperature increases, the air density decreases, resulting in a lower air mass flow rate. Consequently, turbine work, net power output, and thermal efficiency decrease, while the back work ratio increases because a larger fraction of the turbine output is consumed by the compressor. These findings explain the noticeable reduction in power generation during the summer months in hot-climate regions such as Libya.

The regenerative Brayton cycle provides a clear improvement in thermal efficiency by recovering a portion of the waste heat from the turbine exhaust gases and utilizing it to preheat the compressed air before combustion. This reduces the required fuel input while maintaining nearly the same operating conditions. Although a slight reduction in turbine work is observed due to the lower fuel flow rate, the overall cycle efficiency is considerably higher than that of the actual Brayton cycle. Among the investigated configurations, the fully improved Brayton cycle incorporating intercooling, reheating, and regeneration achieves the best overall performance. Intercooling reduces compressor work, reheating increases turbine work, and regeneration decreases fuel consumption by utilizing exhaust heat. The combined effect of these enhancement techniques produces the highest net work output, the highest thermal efficiency, and the lowest back work ratio under all investigated ambient conditions.

The comparison indicates that advanced Brayton cycle modifications can effectively reduce the adverse effects of high ambient temperatures on gas turbine performance. Therefore, such improvement techniques represent a promising option for gas turbine power plants operating in hot regions, although their practical implementation should also consider economic factors, equipment complexity, and maintenance requirements.

Conclusion

Based on the results obtained from the present analysis, the following conclusions can be drawn:

1. Ambient air temperature has a direct negative impact on gas turbine performance, as its increase leads to a reduction in net power output and thermal efficiency, along with an increase in the back work ratio.
2. The reduction in air density and the resulting decrease in mass flow rate are the main factors responsible for performance deterioration at high ambient temperatures.
3. The regeneration cycle improves thermal efficiency by reducing the required heat input, although its effect on net power output is limited.

4. The fully improved cycle achieves the best overall performance in terms of increased network output, improved thermal efficiency, and reduced back work ratio.

The combination of intercooling, reheating, and regeneration proves to be highly effective in enhancing gas turbine performance under high-temperature operating conditions.
Preventive and Predictive Maintenance Scheduling should be utilized to determine optimal maintenance schedules for generation units based on temperature-related performance degradation, thereby minimizing unexpected failures during peak load periods.
Multi-objective optimization employing artificial intelligence algorithms such as genetic algorithms could be used to identify the balance between maximum thermal efficiency of Brayton cycle against minimum operating costs or emissions, accounting for fluctuations in ambient temperatures.

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