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Enhancing Efficiency and Sustainability: Converting Simple Cycle Gas Stations to Combined Cycle in Iraq

Husam Abed Ibrahim [a,*], Mohamed Ali Rezgui [a], Ali Trabelsi [b]

[a] *National Engineering School, University of Tunis El Manar, Tunisia*

[b] *Laboratory of Mechanics, Production and Energy, Higher National Engineering of Tunis, University of Tunis, Tunisia*

*Corresponding author: *husamabedibrahimalgbouri@gmail.com*

***Abstract*— The practical thermal efficiency of gas turbine engines ranges between 28% and 33%. The global electric power sector, particularly in Iraq, faces significant challenges, including reduced hours of electrical energy supply to consumers and difficulties in its continuous distribution. This situation necessitates exploring solutions through investments in the electric power sector, as well as the rehabilitation and maintenance of stations, networks, and transmission lines, to minimize losses and prevent misuse. Additionally, there is a pressing need to address unauthorized access to power transmission lines and networks and to harness renewable energy sources to decrease atmospheric gas emissions and protect the environment. A noteworthy approach is the utilization of exhaust gas heat from thermal gas stations, proposing a transition from simple to combined cycle systems. This research aims to explore the potential of leveraging exhaust gas heat from gas thermal stations to shift their operational system from a simple cycle to a combined cycle. Such a transition promises to enhance electrical power production in Iraq, increase station efficiency, reduce gas emissions, and contribute to environmental conservation. Through detailed study and research, this paper aims to provide viable solutions to the pressing issues facing Iraq's electric power sector, emphasizing the importance of innovative approaches in overcoming these challenges and promoting sustainable energy development. In light of the growing global concern for energy conservation and environmental protection, the authors believe that this paper will stimulate a thought-provoking discussion.**

***Keywords*—Gas station; steam station; combined-cycle power plant; power gas turbine station; power steam turbine station; power total.**

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I. INTRODUCTION

The electricity sector in Iraq plays a vital and indispensable role in fostering the country's economic growth and facilitating social advancements. As assessed by the esteemed International Energy Agency (IEA), it is evident that Iraq heavily relies on gas as its prime energy source. Remarkably, gas contributes approximately 14% to the overall electricity generation capacity, and this capacity has exhibited a notable upward trajectory in recent years [1], [2]. Fascinatingly, according to the highly reputable World Bank, Iraq boasted an impressive installed capacity of 10.39 GW to generate electricity, coupled with a commendable reserve margin of 4.41 GW in the year 2017. Yet, despite these substantial figures, the existing supply falls short of fulfilling the prevailing demand across all regions in Iraq. Notably, the gas-fired power plants established in Iraq surpass their counterparts, boasting the highest Average Load Factor (ALF) [3]. Considering Iraq's ambitious objectives, the Iraqi Ministry of Electricity (MoE) has set its sights on augmenting the Combined Cycle Gas Turbine (CCGT) or Natural Gas Combined Cycle (NGCC) capacity by an unprecedented 27 MWe by 2028 [4], [5].

Moreover, it is worth highlighting that the majority of these ambitious initiatives will be executed through self-generation approaches and build-own-operate (BOO) frameworks [6]. Thus, effectively circumvents any imminent financial encumbrance that may befall the esteemed ministry. Authors [7], as turbine power plants are the mainstay for power generation needs in Iraq. The relatively low exhaust temperature of a simple gas turbine limits the efficiency of the thermal power plant. While no cold discharge of waste heat is needed, waste heat can be easily collected and used [8]. Researchers investigated the issue of physical work that can be gained from all energy, and further, the Coefficient of Performance (COP). Using the exhaust waste heat gas recirculation with the stake cycle system, the gas turbine (GT) power plant, a considerable increase in thermal efficiency was observed. The high temperature of the combusted fuel in the combined cycle power plant (CCPP) increases the net generated power, but the costs of conventional power plants are observed. The high temperature of the combusted fuel in the combined cycle power plant increases the net generated power [9]. Still, the costs of conventional power plants are proportionally higher than those of the simple cycle. Since the first gas turbine commenced commercial operation in 1939, the evolution of gas turbines has made significant progress [10].

However, the reason the gas turbine has been successively established as the preferred choice for many applications is its configuration, which allows for different cycles. A gas turbine is equally viable for operation on its own to generate power. However, its power can be increased if it is configured to produce a combined cycle. The configuration of a combined cycle is mainly accomplished by the addition of a heat recovery steam generator (HRSG) and steam turbine (ST). The unique feature that characterizes gas turbines and prompts the option to run gas turbines on their own, and at the same time, the reason why this option is selected in many cases, is the high air-to-fuel mass ratio [11], [12].

The combustion air passes through the whole engine, and after passing through the combustion chamber, a portion of the air is employed solely to cool the turbine blades, the cap, and the manifolds. The remaining cool air is instead used by the cooling system to cool the casing and rotor, allowing the entire engine to operate at temperatures that are not detrimental to the life of its components. These systems also allow for the choice of higher air pressure ratios, enabling better turbine blade Efficiencies [13]. The high-pressure ratios, combined with the large size of industrial gas turbines, enable significant performance improvements over traditional heat recovery and combined cycle power plants that utilize air as the working fluid [14].

Simple-cycle gas turbines have characterized electrical power generation for decades due to their fast start, high reliability, and long lifespan [15]. These turbines are widely used in various industries and locations, including confined urban areas with little or no possibility of near-term conversion to the more efficient combined cycle power plants. As the population in these dense urban centers continues to grow rapidly, the need for electricity also increases. Systems requirements and challenges vary according to plant Configuration, so it is necessary to identify the plant configuration upon which the combined cycle power station will be based [16].

As the implementation of the combined cycle is (simple cycle + gas turbine + heat recovery steam generator + steam turbine) [10]. The first part is the simple gas station that will supply one or more high-pressure steam heat recovery generators for the low-pressure steam turbine. The turbine must be designed to add between 80 and 150 MW for an operating time of 24 hours per day [17], [18]. The gas station must be adequately prepared because the use of a large portion of the heat recovery steam generators leads to a significant increase in the gas turbine exhaust temperature, necessitating technical alignment in the choice of gas turbine type and cooperation between suppliers [19]. To describe the basic concept of the main topic, it is proposed here to work with a frame divided into four systems [20], namely: The fuel system, the combustion air system, the ignition system, and the emission control. The fuel supply system must ensure the flexible use of different types of fuels, with a preference for natural gas, but may also accommodate other fuels, such as diesel oil [21].

Regardless of the fuel used, the moisture must be released appropriately or separated from the natural gas(NG) before it is introduced into the burners to prevent moisture from condensing and other contaminants on the electrodes [22]. Upon entering the combustion chamber, the fuel must form a well-stabilized gas-air suspension that promotes the usual characteristics of combustion [23]. The air used in combustion must be compressed, cleaned, and heated to provide stable combustion in the Combustion chambers (CC) [24]. Due to the use of a heat recovery steam generator, the temperature of the combustion air is lower than for a simple cycle [25]. Therefore, combustion air heating is essential for improving efficiency and reducing costs. The use of supplementary or duct burners may also be an alternative to help increase the steam output [26].

Fortunately, the combination of the low construction cost of simple cycle plants, the future need to meet demand using clean gas-fired electricity generation, and the commonality of the plant design with combined cycle plants suggests that significant future interest and demand will occur [27]. The proposed work focuses on utilizing the exhaust gas heat from the AL-Mansouriya gas station [28]. It has been working on recovering the heat to operate a steam station and converting the work system of the station from the simple cycle to the combined cycle and increasing the efficiency of the station by adding a second combustion chamber and Heat Recovery Steam Generator (HRSG). in this paper, we are working on conducting an Analytical, simulation and real study of the station.

II. Materials and Methods

A. Combined-cycle power plant (CCPP) Description

The creation of a gas-powered AL-Mansouriya station that operates on a basic cycle is included in the work being done, as shown in Fig. 1, to participate in the combined cycle.

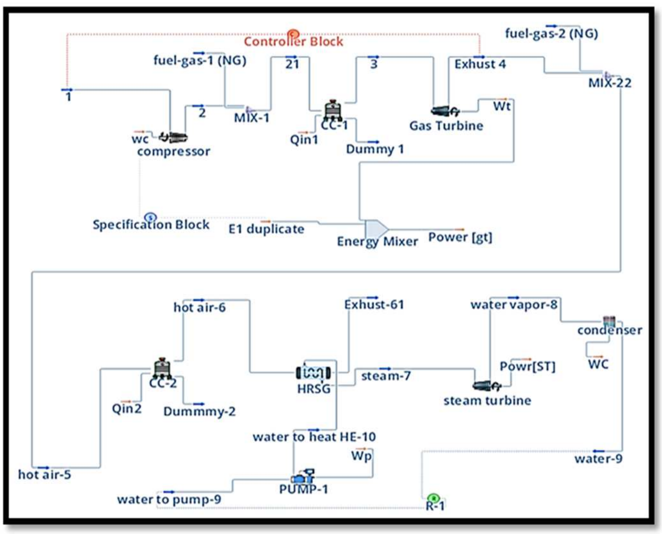

Fig. 1 Combined cycle power plant

The electricity that the plan is designed to have [181MW]. The real power [151 megawatts]. This is a diagram of the power plant that was investigated. The topping cycle and the bottoming cycle are both components of a combined cycle. The topping cycle receives the majority of the heat that is delivered. The topping cycle receives the majority of the heat that is delivered. Since the waste heat that it generates is then employed in a second process that operates at a lower temperature level, this process is referred to as a "bottoming cycle." Both the topping cycle and the bottoming cycle are examples of Brayton cycles, whereas the bottoming cycle is an example of a Rankine cycle. Four processes make up the Brayton cycle, which are as follows: Following its passage through the filters, the air is then introduced into the compressor.

As this is happening, both the temperature and air pressure are rising. After injecting liquid or gaseous fuel, the compressed air is introduced into the combustion chamber-1 (CC-1), which is where the combustion takes place at a constant pressure. As a result, the temperature and, consequently, the enthalpy of the gas are significantly increased. Following their exit from the combustion chamber-1, the hot products of combustion make their way into the turbine. As they go through the blades, they expand, which in turn causes the turbine shaft to move, which in turn causes the generator's shaft to rotate and create electricity. Following the recovery, mixing, and injection of the exhaust gases from the gas turbine into the natural gas once more, the natural gas is then introduced into the combustion chamber-2 (CC-2). Once the heated exhaust gases have exited the combustion chamber-2, they are directed into the Heat Recovery Steam Generator (HRSG), where they are subjected to the Rankine cycle, which involves the exchange of energy with water. During this cycle, water travels through the evaporators, economizers, and superheaters from beginning to end.

As a result, the production of superheated steam occurs at a single pressure stage known as high-pressure (HP). Following this, the high-pressure steam travels through the ST blades, which in turn causes the shafts of both the turbine and the generator to revolve. Following this, the expanded steam is introduced into the condenser, where it undergoes an isobaric process before condensing. Following this, the steam is preheated by passing through the condensate preheater (CPH) and the deaerator. Both high-pressure (HP) and feed-water pumps (FWP) are used to pump water into the HRSG; hence, the cycle is repeated continuously.

B. Analysis and calculation Gas Turbine Power Cycles – Brayton Cycle (Analytical, simulation, actual)

As the working fluid, gas should be utilized. In the process of combustion, the mixture of air and fuel transforms into combustion products. Compared to the vapor power plants investigated previously, these plants are both lighter and more compact [29], [30]. This type of all-gas cycle is referred to as a combustion turbine (GT) in the power-generating business. The majority of the time, it is delivered in a comprehensive package that is ready to be utilized to create electricity. A description of the gas turbine power plant can be provided by considering the air that is allowed to flow into the compressor, which is then heated in the combustor, and which generates power through its interaction with the turbine blades. In the process of being compressed in the compressor, receiving heat from an external source in the combustor, and expanding in the turbine, air is regarded to be the working fluid. A simple gas turbine. Isentropic compression, isobari combustion, isentropic expansion, and heat rejection are the examples of four different thermodynamic processes that are shown in Figure 2. Analysis and calculation Gas Turbine Power Cycles – Brayton Cycle (Analytical, simulation, actual).

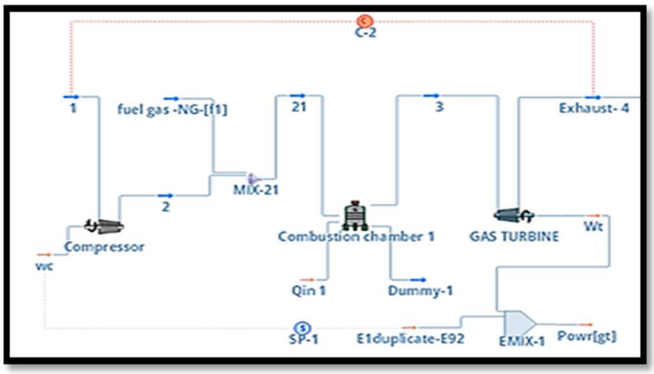

Fig. 2 Gas turbine station

C. Thermodynamic Analysis of the Gas Turbine Cycle

It has been previously shown that the performance of a gas turbine engine is a function of many parameters. The thermodynamic analysis of the engine may be simplified by making the following assumptions:

- The air used by the gas turbine, as well as the products of the combustion, behave like a perfect gas.
- The specific heat capacities are a function of the average temperature through each process.
- The loss of stagnation pressure in the compressor inlet is a constant percentage of the compressor inlet pressure.
- The loss of stagnation pressure in the combustion chamber is a constant percentage of the combustion chamber inlet pressure.

It is possible to compute the performance of the engine by conducting a methodical investigation of each component that makes up the engine, provided that certain assumptions are taken into consideration. In the next section, you will find the process used to calculate the values for specific temperatures. Because the gas turbine is a rotor-driven internal combustion engine, its operating cycle is comprised of four distinct methods, which are compression, beating, expansion, and discharge. Both the cycle pressure ratio and the turbine entrance temperature are the two primary operational factors that determine the engine's performance. This performance includes the engine's thermal efficiency as well as its specific work output [31], [32].

As a result of the fact that the basic gas turbine engine is known for its low thermal efficiency, several attempts have been made to improve thermal efficiency by altering the primary cycle parameters. A number of adjustments have been implemented, including the following: (a) increasing the temperature at which the turbine enters the system; (b) designing for a higher cycle pressure ratio; and (c) utilizing exhaust heat recovery to heat the compressed air as it enters the combustion chamber, also known as renewable heating. Although these modifications have occurred, the thermal efficiency of gas turbine engines has remained unchanged and continues to range between 28% and 38%. The range of design parameters for gas turbine power plants is determined by the ambient air temperature (AAT), in particular, the cycle pressure ratio. This is an important consideration, as it determines whether the plant is designed for optimum thermal efficiency or maximum specific work output [33], [34].

Consequently, the application of the gas turbine engine is the determining factor in selecting the optimal parameters that provide the highest efficiency while also achieving the maximum specific work. At the gas thermal power plant located in Al-Mansouriya, analytical power plant calculations were performed on the gas turbine. Equations of thermodynamics that are already known can be utilized to carry out analytical computations for power plants. Table 1 also contains a listing of the specs of the gas station, and Table 2 contains the results of the analysis.

TABLE I
SPECIFICATION OF THE GAS TURBINE STATION

Type	GT-ALSTOM[AL-Mansuriya station
Design power	GT[181MW]
Actual power	GT[151MW]
Temperature inlet compressor t_1	316[K]
Pressure inlet compressor p_1	0.988[bar]
Pressure outlet compressor p_2	13.5[bar]
Mass flow rate air inlet compressor m_a	575[kg/s]
Mass flow rate fuel gas [N.G]m_{f1}	9.1[kg/s]
Temperature fuel gas [N.G]t_{f1}	304[k]
Power gas turbine station p_{owr}[gt]	151[MW]
Thermal efficiency compressor η_c	72[%]
Thermal efficiency gas turbine η_{gt}	92.8[%]
Specific heats gas c_{pg}	2.4[kj/kg.k]
Specific heats airc_{pa}	1.005[kj/kg.k]
Ratio Specific heats air k	1.4

TABLE II
RESULT AND DISCUSSION GAS TURBINE [GT] (ANALYTICAL, SIMULATION, ACTUAL)

Parameters	Analytical	Simulation	Actual
Temperature ideal outlet compressor,t_2[k]	667	667	667
Temperature actual outlet compressor,t_{2a}[k]	802	781	719
Work compressor,W_c [kw]	281000	281000	242000
Thermal efficiency compressor, η_c[%]	72	72	83.5
Mass flow rate air and fuel inlet mix21, , m_{a1} [kg/s]	584.1	584.1	584.4
Temperature inlet combustion chamber-1,t_{21}[k]	790	761	703
Rice temperature in combustion chamber-1,TR_1[k]	730	642	604
Temperature inlet gas turbine,t_3 [k]	1520	1403	1307
Temperature ideal outlet gas turbine ,t_4 [k]	778	661	640
Temperature actual outlet gas turbine,t_{4a}[k]	832	793	793
Thermal efficiency gas turbine, η_{gt}[%]	92.8	92.8	89.3
Work turbine,W_t[kw]	435000	432000	385000
Thermal efficiency gas turbine station, η_{gts}[%]	30	26	22
Power gas turbine station ,p_{owr}[gt][MW]	151	151	143
Heat addition in combustion chamber-Q_{in1/m_a}[kj/kg.k]	718	645	607
Net work energy gas turbine station,W_{net}[kj/kg.k]	258	258	245
Back work ratio,bwr[%]	70	74	69

D. Air Compression

The best efficiency of the compressor is chosen between [72-75%], which is the best efficiency used to raise the temperature of the air coming out of the compressor. On the contrary, as the efficiency of the compressor increases, the temperature of the air coming out of the compressor decreases. Therefore, in this analytical study, we will choose the efficiency of the analytical research, we will choose the efficiency of compressor to be 72 %, where the final stagnation temperature in the compression process (t_2) equal.

$$t_2 = t_{1*} \left[\frac{p_2}{p_1}\right] \wedge [^{k-1}/_k] \tag{1}$$

Through the efficiency of the compressor, we find the actual temperature outlet compressor [t_{2a}].

$$\eta_c = \frac{[t_2 - t1]}{[t_{2a} - t1]} \tag{2}$$

The compression power (Wc) can be described as.

$$\frac{Wc}{m_a} = c_{pa} \ * \ (t_{2a-}t_1) \tag{3}$$

E. Mixture-21

The mass of hot air leaving the compressor and the mass of the fuel gas [N.G].entering the mixture -21 , are combined to determine the total hot air mass [m_{a1}].and temperature leaving the mixture-21,[t_{21}].and entering combustion chamber -1.

$$m_{a1} = \ m_a + \ m_{f1} \tag{4}$$

$$t_{21} = \frac{m_{a*}\, c_{pa*}t_{2a} + m_{f1} * c_{pg*}t_{f1}}{m_a * c_{pa} \ + m_{f1} * c_{pg}} \tag{5}$$

F. Combustion Chamber-1

Through the power gas turbine station P_{owr} *[gt]*, we find network energy gas turbine station [W_{net}]. Through we find work energy turbine [W_t]. We then get temperature inlet gas turbine [t_3].we determine then [TR_1] Rice Temperature in combustion chamber-1. Also, we find heat addition -1, $\frac{Q_{in1}}{m_a}$. The efficiency of the gas turbine[η_{gt}] [92.8%].

$$p_{owr}\ [gts] = W_{net} * m_a \tag{6}$$

$$W_{net} = W_{gt} - \ W_c \tag{7}$$

$$W_{gt} = c_{pa} \ \ * (t_{3-}t_4) \tag{8}$$

$$t_{3-}t_4 \ = \eta_{gt} * t_3 * \left[1 - \left[\frac{1}{p_3/p_4}\right]^{[^{k-1}/_k]}\right] \tag{9}$$

We find Rice Temperature in combustion chamber-1, [TR_1] [k].

$$(t_3 - t_{21}) = TR1[k] \tag{10}$$

We find heat addition -1, $\frac{Q_{in1}}{m_a}$ in the combustion chamber -1.

$$\frac{Q_{in1}}{m_a} = c_{pa} \ * \ (t_3 - t_{2a}) \tag{11}$$

G. Gas Turbine

Where the exhaust stagnation temperature is actual in the expansion process ($t4a$) equal.

$$\eta_t = \frac{[t_3 - t4a]}{[t_3 - t4]} \tag{12}$$

$$\frac{Q_{out}}{m_a} = c_{pa} \ * (t_{4a} - t_1) \tag{13}$$

Thermal efficiency gas turbine station

$$\eta_{gts} = 1 - \frac{[t_{4a} - t1]}{[t_3 - t2a]} \tag{14}$$

Back work ratio

$$bwr = \frac{t_{2a} - t_1}{t_3 - t_{4a}} \tag{15}$$

H. Analysis and Calculation Steam Turbine - Rankine Cycle (Analytical, Simulation, Actual)

Fig. 3 depicts a schematic representation of the steam power plant that operates on the Rankine cycle, along with the temperature that corresponds to it. All four components that are involved with the Rankine cycle are devices that operate with a constant flow: the pump, the boiler, the turbine, and the condenser.

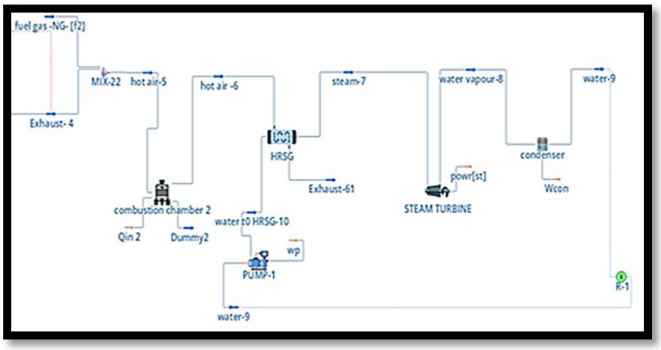

Fig. 3 Steam turbine station

As a result, each of the four processes that comprise the Rankine cycle may be examined independently as steady-flow processes. The kinetic and potential energy changes of the steam are often negligible in comparison to the work and heat transfer terms, and as a result, they are typically disregarded. There must be a state of equilibrium in pressure when carrying out the analysis procedure for the steam station. To put it another way, the pressure of the steam that is introduced into the steam turbine is equivalent to the pressure of the water that is introduced into the heat exchanger.

Furthermore, the pressure of the water vapor that is introduced into the condenser is at the same level as the pressure of the water that is introduced into the pump. The function of the quality of the wet steam, also known as the dryness fraction, is the enthalpy of the steam. Consequently, steam that is 70 percent dry and 30 percent moist will be considered to be of high quality. It is also necessary for the index of dryness refraction to be less than one for there to be drought. Because moist steam might hurt turbine blades, this explanation is given.

To prevent corrosive condensation, the temperature of the stack exhaust should be greater than the temperature at which water vapor condenses in the exhaust gas. The temperature of the exhaust gas in the boiler ought to be at least a little bit higher than the temperature of the steam, with the exact amount of this difference being determined by the economic and design characteristics. The calculations for the steam thermal power plant, which is expected to be coupled with the gas station at Al-Mansouriya Gas Turbine, are being performed by the analytical power plant. Equations of thermodynamics that are already known can be utilized to carry out analytical computations for power plants as shown in table 3.

TABLE III
RESULT OF STEAM TURBINE [ST] (ANALYTICAL, SIMULATION, ACTUAL)

Type	ST-ALSTOM
Design power	Steam Turbine [150 MW]
Mass flow rate fuel gas [N.G] inlet MIX-22,m_{f2}	2 [kg/s]
Temperature fuel gas [N.G]t_{f2}	304 [k]
Temperature exhaust gas t_{4a}	832[k]
Mass flow rate exhaust gas , m_{a1}	584.1[kg/s]
Pressure exhaust gas p_4	0.988 [bar]
Temperature exhaust gas outlet [HRSG] t_{61}	400[k]
Pressure Steam Outlet [HRSG] p_7	150[bar]
Temperature water inlet [HRSG] t_{10}	304[k]
Pressure water inlet [HRSG]p_{10}	150[bar]
Dry refractive index [x]	0.88
Temperature water vapor inlet condenser t_8	333[K]
Mass flow rate steam m_s	110[kg/s]
Pressure inlet condenser p_8	0.2[bar]

TABLE IV
RESULT OF STEAM TURBINE

Result	Analytical	Simulation	Actual
Temperature inlet combustion chamber-2, t_5[k]	829	789	751
Temperature exit [CC-2] and inlet [HRSG], t_6[k]	982	934	898
The enthalpy exhaust gas inlet [HRSG], h_6 [kj/kg]	1025	969	928
Heat addition in combustion chamber-2, $Q_{in2/m_{at}}$ [kj/kg.k]	153	145	147
The enthalpy exhaust gas outlet [HRSG],h_{61}[kj/kg]	400	400	400
The enthalpy of the water inlet [HRSG], h_{10}[kj/kg]	129	129	129
The enthalpy of the steam outlet[HRSG], h_7[kj/kg]	3470	3416	3027
Rise temperature in combustion chamber-2,TR_2[k]	153	145	147
The energy added in the [HRSG], Q_{ins}[kj/kg.k]	3341	3287	2898
The enthalpy water vapor inlet condenser , h_8 [kj/kg]	2203	1983	1967

Result	Analytical	Simulation	Actual
The enthalpy water inlet pump, h_9[kj/kg]	127	127	127
Work energy steam turbine, W_{ST}[kj/kg]	1267	1146	1009
Power steam turbine station, p_{owr}[ST][MW]	139	126	110
Work energy condenser, W_c[MW]	228	227	202
Work energy pump, W_p [kw]	2	2	2
Dry refractive index , X	0.88	0.787	0.77
Network energy steam, W_{nets}[kj/kg]	1265	1144	1007
Thermal efficiency steam turbine station, η_{st}	0.37	0.38	0.35

I. Mixture-22

To find temperature of the gas leaving mixture -22[t_5]

$$t_5 = \frac{m_a^{\cdot} * c_{pa} * t_{4a} + m_{f2} * c_{pg} * t_{f2}}{m_a^{\cdot} * c_{pa} + m_{f2} * c_{pg}} \tag{16}$$

$$t_5 = \frac{584.1*1.4*832+2*2.24*304}{584.1*1.4+2*2.24} = 829[k]$$

$$p_5 = p_4$$

J. Combustion Chamber-2

$t_{5=}$ 829[k],Inlet temperature to combustion chamber -2, through a Fig.4. We find the increase in temperature.

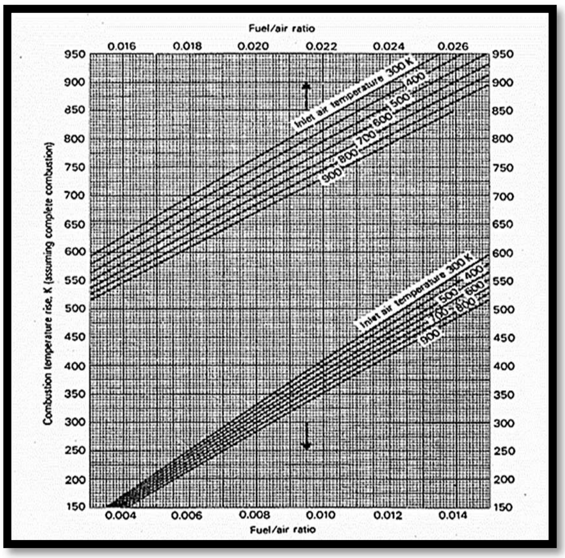

Fig. 4 Combustion temperature rise

$$F = \frac{m_{f2}}{m_a^{\cdot}} \tag{17}$$

$$F = \frac{2}{584.1} = 0.003 \; {fuel}/{air\ ratio}$$

$$t_{6=} \; t_{5+} \, t_{R2} \tag{18}$$

$t_{6=}$ $829 + 153 = 982[k]$, temperature inlet [HRSG]

$$\frac{Q_{in2}}{m_{at}^{\cdot}} = t_6 - t_5 \tag{19}$$

$$\frac{Q_{in2}}{m_{at}^{\cdot}} = 982 - 829 = 153[\text{kj/kg}]$$

$$\xRightarrow{table-A\,4\ sturated\ water} h_{10} = h_f = 129$$

$$h_8 = h_f + x\, h_{fg8} \tag{22}$$

$$h_8 = 129 + 0.88 * 2357 = 2203[\text{kj/kg}]$$

$$W_{st} = h_7 - h_8 \tag{23}$$

$$W_{st} = 3470 - 2203 = 1267[\text{kj/kg}]$$

$$p_{owr}[st] = W_{st} * m_s^{\cdot} \tag{24}$$

$$p_{owr}[st] = 1267 * 110 = 139000[kw]$$

K. Condenser

$$W_{con=} m_s^{\cdot} \,(h_8 - h_9) \tag{25}$$

The enthalpy of the water inlet of the pump

$t_9[\text{k}] = 303[\text{k}]$

$$t_{10}[\text{k}] \xRightarrow{table-A\,4\ sturated\ water} h_{10} = h_f = 129[\text{kj/kg}]$$

$h_9 = h_f = 127$

$W_{con=} 110 * (2203 - 127) = 228000[\text{kw}]$

Pump

$$W_{p\,=} (h_{10} - h_9) \tag{26}$$

$W_{p\,=} (129 - 127) = 2[kw]$

L. Thermal Efficiency Steam Turbine Station

$$W_{net\ s=} W_{st-} W_p \tag{27}$$

$W_{net\ s=} 1267 - 2 = 1265$

$$\eta_{st} = \frac{W_{net\ s}}{Q_{ins}} \tag{28}$$

$\eta_{st} = \dfrac{1265}{3341} = 0.37$

Thermal efficiency of steam turbine plants between 36% and 45%.

III. Results and Discussion

A. Combined Cycle Power Plant (CCPP), or Combined Cycle Gas Turbine (CCGT), Power Plant

The results of this research were obtained by combining the Rankine (steam) and Brayton (gas) thermodynamic cycle, (Analytical, Simulation, Actual). The analysis of thermodynamics, Thermal efficiency, and specific work are two methods that may be utilized to depict the performance of the (CCPP) in Table 5, and the outcome can be seen in Table 6. Fig. 5 illustrates this point. Steam turbine plants have a thermal efficiency that ranges from 36% to 45%, whereas gas turbine plants have a thermal efficiency that ranges between 28% and 33%. The overall thermal efficiency of the combined plant can be increased to as high as sixty percent by recovering some of the low-grade thermal energy from the exhaust gas of the gas turbine to produce high-pressure steam, and then using this steam to power a steam turbine, which will generate additional electrical power.

TABLE V
SPECIFICATION OF THE COMBINED CYCLE POWER PLANT (CCPP)

Type	Analytical	Simulation	Actual
p_{owr}[gt]	151	151	143
p_{owr}[ST]	107	92	79
Q_{in1}	718	645	607
$\dot{m}_{a1}$	584.1	584.1	583.4

TABLE VI
RESULT OF COMBINED CYCLE POWER PLANT

Type	**Analytical**	**Simulation**	**Actual**
powr[tot]	258	243	222
η_ccpp	0.61	0.64	0.62

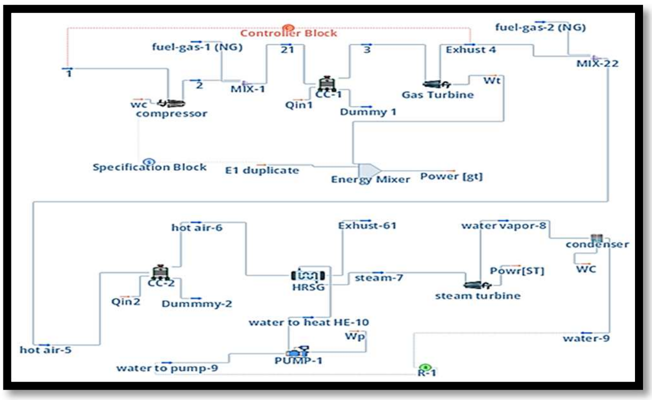

Fig. 5 Schematic diagram of a combined gas-steam power [CCPP]

This is based on theoretical studies that have been conducted. On the other hand, to achieve the highest possible level of performance from combined heat and power plants (CCPP), it is of the utmost significance to carefully pick the key design parameters for the gas and steam turbines, as well as to schedule their control systems appropriately.

B. Thermal Efficiency Combined Cycle Power Plant (η_{ccpp})

$$\eta_{ccpp} = \left(p_{owr}[tot]\right) \Big/ \left[\dot{m}_a * Q_{in\,1} + m_{f2} * Q_{in\,2}\right] \quad (29)$$

$$p_{owr}[tot] = p_{owr}[gt] + p_{owr}[ST] \quad (30)$$

$$p_{owr}[tot] = 153000 + 139000 = 292000[kw]$$

$$\eta_{ccpp} = (292000) \Big/ [584.1 * 718 + 2 * 153] = 0.69$$

Through the proposed work to convert the AL-Mansouriya gas station from a simple cycle to a combined cycle, we are modifying the inputs of the gas station and illustrating their effects in the form of diagrams for analysis, simulation, and actual operation. In the analytical case, changing the inputs to the gas station and examining their effect on the combined cycle, as shown in Table 7. It illustrates the change in inputs to the gas station resulting from the combined cycle, as shown in the following diagrams: Fig. 6, Fig. 7, Fig. 8.

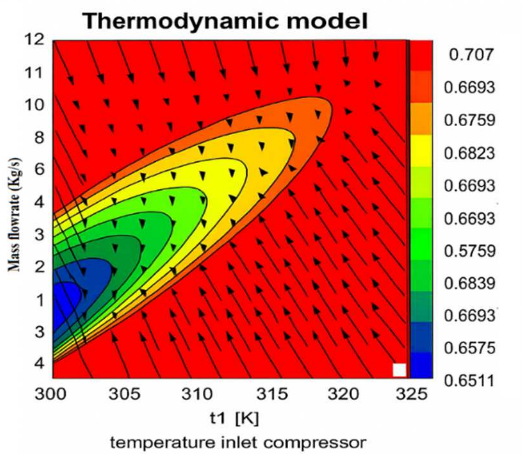

Fig. 6 Thermal efficiency [CCPP]

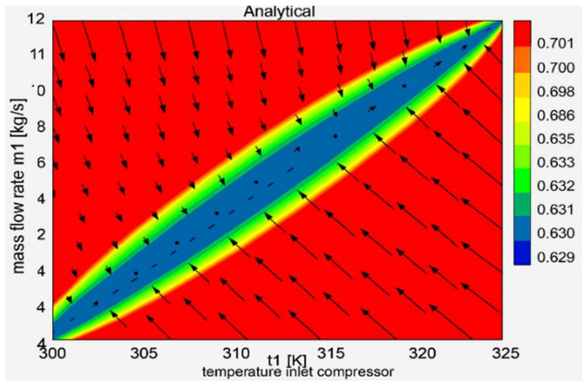

Fig. 7 Temperature of exhaust gas turbine t4a

TABLE VII
CHANGE THE GAS STATION DATA INPUTS, SHOWING THEIR IMPACT ON THE COMBINED CYCLE AND DISCUSSING THEIR RESULT USING THE ANALYTICAL

T_1[k]	m_{f1} [kg/s]	T_3[k]	$W_t[kw]$	$p_{owr}[gt]$	η_{gts}	$t_{4a}[k]$	$t_7\,[k]$	$p_{owr}[st]$	η_{st}	η_{ccpp}
300	3	1416	726.4	140000	0.2984	775	769	132602	0.3676	0.6449
301.3	3.474	1424	728.2	141419	0.297	779	773	132898	0.3681	0.6479
302.6	3.947	1433	730.1	142382	0.2956	784	778	133195	0.3686	0.6509
303.9	4.421	1441	731.9	143206	0.2942	789	783	133492	0.3691	0.654
305.3	4.895	1450	733.7	144077	0.2928	793	787	133789	0.3696	0.6571
306.6	5.368	1458	735.5	144900	0.2914	798	792	134086	0.3702	0.6602
307.9	5.842	1466	737.4	145724	0.29	802	796	134382	0.3707	0.6634
309.2	6.316	1475	739.2	146548	0.2885	807	801	134679	0.3712	0.6665
310.5	6.789	1483	741	147372	0.2871	812	806	134976	0.3717	0.6697
311.8	7.263	1491	742.8	148195	0.2856	816	810	135273	0.3722	0.673
313.2	7.737	1500	744.7	149066	0.2841	821	815	135570	0.3727	0.6762
314.5	8.211	1508	746.5	149890	0.2826	825	819	136866	0.3732	0.6795
315.8	8.684	1517	748.3	150713	0.2811	830	824	136163	0.3737	0.6828
317.1	9.158	1525	750.2	151537	0.2796	834	827	136460	0.3743	0.6862
318.4	9.632	1533	752	152361	0.278	839	832	136757	0.3748	0.6896
319.7	10.11	1542	753.8	153187	0.2765	844	837	137054	0.3753	0.693
321.1	10.58	1550	755.7	154056	0.2749	848	841	137350	0.3758	0.6965
322.4	11.05	1559	757.5	154878	0.2733	853	846	137647	0.3763	0.6999
323.7	11.53	1567	759.3	155705	0.2717	857	850	137944	0.3768	0.7035
325	12	1575	761.2	156527	0.2701	862	855	138241	0.3773	0.707

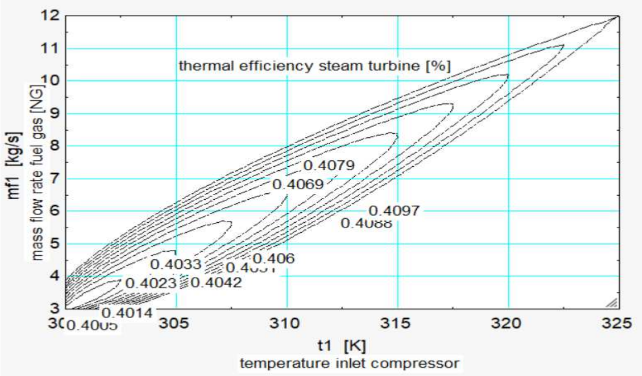

Fig. 8 Thermal efficiency steam turbine

In the Simulation case, changing the gas station data inputs and examining their effect on the combined cycle, as shown in Table 8. It shows us the change of the inputs to the gas station on the result of the combined cycle, which is shown in the following diagrams Fig. 9, Fig. 10, Fig.11, Fig.12, Fig. 13, Fig. 14.

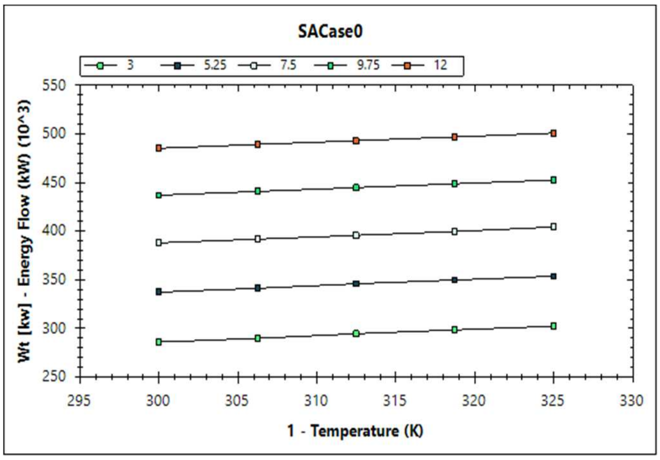

Fig. 9 Work gas turbine

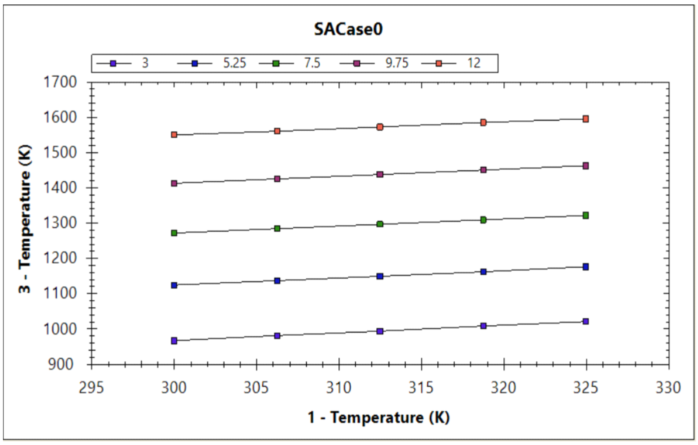

Fig. 10 Temperature inlet-t_3 [GT]

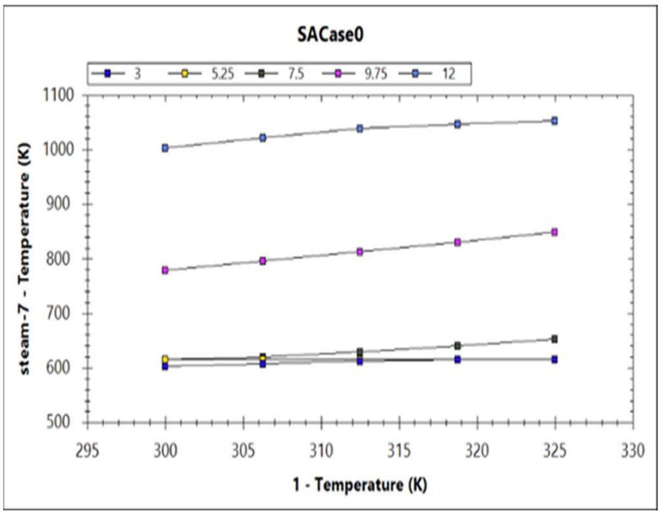

Fig. 11 Temperature steam-t7

TABLE VIII

CHANGE THE GAS STATION DATA INPUTS, SHOWING THEIR IMPACT ON THE COMBINED CYCLE, AND DISCUSSING THEIR RESULT USING SIMULATION

T1[k]	m_{f1} [kg/s]	$t_3\,[k]$	$p_{owr}[gt][kw]$	$W_t[kw]$	Exhaust-$t_{4a}[k]$	$p_{owr}[st][kw]$	Steam-t_7 [k]
300	3	966.725	4034.35	285339	520.714	45013.8	601.77
300	5.25	1122.73	69604.7	337000	616.178	71563.6	615.161
300	7.5	1270.96	119999	387394	709.345	98753.3	615.161
300	9.75	1412.4	169215	436610	800.049	129741	777.762
300	12	1547.88	217386	484781	888.207	168190	1001.58
306.25	3	980.301	22155.2	289550	528.744	47126.8	606.507
306.25	5.25	1135.78	68241.3	341097	624.131	73771	615.161
306.25	7.5	1283.57	118554	391410	717.158	101004	619.837
306.25	9.75	1424.64	167739	440595	807.652	132590	794.941
306.25	12	1559.79	215821	488677	895.696	171567	1020.2
312.5	3	993.796	20888	293744	536.737	49253	610.825
312.5	5.25	1148.76	66919	345202	532.007	75960.9	615.161
312.5	7.5	1296.12	117149	395432	724.898	103273	629.09
312.5	9.75	1436.82	166237	444520	815.282	135501	812.445
312.5	12	1571.64	214277	492560	903.145	174804	1037.89
318.75	3	1007.32	19667.2	297950	544.762	54108.7	614.731
318.75	5.25	1161.77	65570.4	349319	639.912	78162.9	615.161
318.75	7.5	1308.7	115718	399467	732.666	105596	639.695
318.75	9.75	1449.04	164730	448478	822.907	138459	830.64
318.75	12	1583.54	212718	496467	910.606	176126	1045.07
325	3	1020.76	18386.9	302135	552.754	53566.1	615.161
325	5.25	1174.72	64235.2	353415	647.783	80359.8	615.161
325	7.5	1321.21	114301	403481	740.401	107958	651.408
325	9.75	1461.2	163239	452419	830.498	141452	847.994
325	12	1595.38	211142	500321	918.082	177456	1052.26

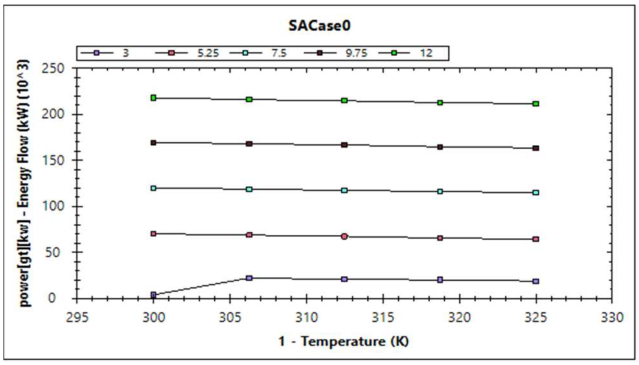

Fig. 12 Power gas turbine

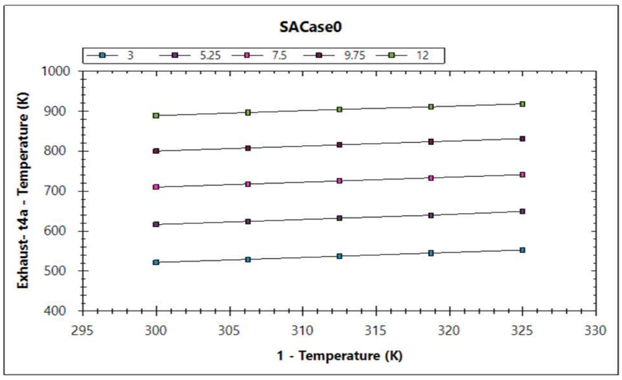

Fig. 13 Exhaust gas –t4a[GT]

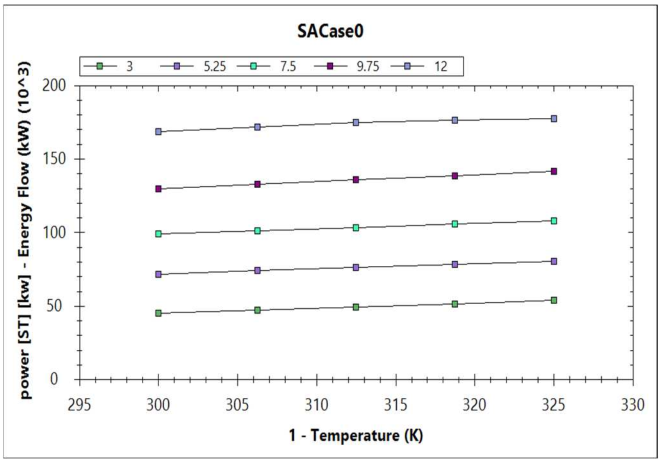

Fig. 14 Power steam turbine

In the Actual case, changing the gas station data inputs ,and showing their effect on the combined cycle, according to Table 9. It shows us the change of the inputs to the gas station on the result of the combined cycle.

TABLE IX
CHANGE THE GAS STATION DATA INPUTS, SHOWING THEIR IMPACT ON THE COMBINED CYCLE, AND DISCUSSING THEIR RESULT USING ACTUAL.

$t_1\,[k]$	m_{f1} [kg/s]	$t_3\,[k]$	$p_{owr}[gt][kw]$	$W_t[kw]$	Exhaust-$t_{4a}[k]$	$p_{owr}[st][kw]$	Steam-$t_7\,[k]$
300	3	911.468	15542.7	258104	504.933	40091.1	591.403
300	5.25	1069.68	77581.4	308149	603.473	66641.4	615.161
300	7.5	1219.75	126408	356976	699.478	93946.6	615.161
300	9.75	1362.75	174093	404661	792.79	124446	761.62
300	12	1499.58	220729	451297	883.394	162613	989.615
306.25	3	924.05	31279.5	261848	512.52	41998.5	596.553
306.25	5.25	1081.75	76534.9	311812	610.944	68666.2	615.161
306.25	7.5	1231.4	125287	360563	706.811	96003.8	615.161
306.25	9.75	1374.03	172902	408179	799.981	127038	777.592
306.25	12	1510.55	219493	454770	890.417	165691	1007.05
312.5	3	936.56	30297.2	265574	520.077	43921.9	601.377
312.5	5.25	1093.75	75501	315458	618.384	70686	615.161
312.5	7.5	1242.98	124178	364134	714.15	98057.2	616.588
312.5	9.75	1385.26	171723	411680	807.143	129665	793.767
312.5	12	1521.47	218227	458184	897.475	168817	1024.6
318.75	3	949.092	29354.6	269311	527.663	45874.2	605.89
318.75	5.25	1105.79	74444.8	319114	625.849	72716.1	615.161
318.75	7.5	1254.6	123046	367715	721.442	100144	624.779
318.75	9.75	1396.52	170523	415192	814.328	132345	810.227
318.75	12	1532.43	216965	461634	904.518	171438	1039.21
325	3	961.553	28359.8	273029	535.222	47840.3	610.036
325	5.25	1117.76	73416.2	322769	633.26	74734.8	615.161
325	7.5	1266.16	121940	371293	728.72	102255	634.144
325	9.75	1407.44	169337	418689	821.483	135057	826.82
325	12	1543.35	215716	465069	911.534	172656	1045.96

IV. CONCLUSION

The following are some significant observations that may be drawn from these findings: There is a direct correlation between raising the temperature of the gas turbine inlet and an increase in both the combined cycle efficiency and the combined specific work. The pressure ratio difference has a bigger impact on the combined cycle efficiency for any combination of combined cycle configurations, and this influence increases in proportion to the temperature of the gas turbine inlet. Only in the presence of high exhaust temperatures (high turbine intake temperature and/or low-pressure ratio) is it possible to justify high steam boiler pressure. Only in situations where the turbine intake temperatures are low (i.e., low gas exhaust temperatures) and the combined specific work output is higher, may reheating of gas turbines be considered justifiable.

The combined cycle efficiency and specific work output will both increase as a result of utilizing dual-pressure steam. Even though intercooling gas turbines improves the performance of gas turbines, it has only a marginal impact on the combined cycle efficiency and the combined specific work output. The most influential parameters on the performance of the gas turbine cycle are the compressor pressure ratio and the temperature at which the gas turbine is introduced into operation. When it comes to a basic gas turbine cycle, the maximum efficiency and the maximum particular work have distinct performance characteristics.

On the other hand, when it comes to reheating the gas turbine cycle, the maximum efficiency and the maximum specific work have the exact performance requirements. If the additional heating increases the steam turbine cycle efficiency, then the combined cycle efficiency will always be reduced. However, supplemental heating will always lower the combined cycle efficiency. There is a direct correlation between raising the temperature of the gas turbine inlet and an increase in both the combined cycle efficiency and the combined specific work. When the input temperature of the gas turbine is higher, the pressure ratio difference has a more substantial impact on the combined cycle efficiency. This holds regardless of the layout of the combined cycle.

REFERENCES

[1] H. D. Alhassany et al., "Review of bioenergy potential from the agriculture sector in Iraq," *Energies*, vol. 15, no. 7, p. 2678, 2022, doi:10.3390/en15072678.

[2] R. Mills and M. Salman, *Powering Iraq: Challenges Facing the Electricity Sector in Iraq*, Al-Bayan Center for Planning and Studies, 2020.

[3] M. Jamal et al., *Iraqi Electricity Sector Overview*, KAPITA's Research Team, 2021.

[4] S. Srinivasan et al., "Business models for accelerating phase-out of coal-based generation: Developing typologies and a discussion of the relative merits of alternative models," *Electr. J.*, vol. 35, no. 8, p. 107185, 2022, doi: 10.1016/j.tej.2022.107185.

[5] L. A. Al-Rubaye et al., "Characteristics study of gas turbine performance with using surface geothermal energy: Case study of Iraq," *IOP Conf. Ser.: Mater. Sci. Eng.*, vol. 1076, no. 1, p. 012069, 2021, doi: 10.1088/1757-899X/1076/1/012069.

[6] K. Takeishi and R. Krewinkel, "Advanced gas turbine cooling for the carbon-neutral era," *Int. J. Turbomach. Propuls. Power*, vol. 8, no. 3, p. 19, 2023, doi: 10.3390/ijtpp8030019.

[7] H. E. Bower et al., "Experimental analysis of a micro gas turbine combustor optimized for flexible operation with various gaseous fuel compositions," *J. Eng. Gas Turbines Power*, vol. 142, no. 3, p. 031015, 2020, doi: 10.1115/1.4044901.

[8] M. F. Ezzat and I. Dincer, "Energy and exergy analyses of a novel ammonia combined power plant operating with gas turbine and solid oxide fuel cell systems," *Energy*, vol. 191, p. 116571, 2020, doi:10.1016/j.energy.2019.116750.

[9] O. Mahian et al., "Exergy analysis in combined heat and power systems: A review," *Energy Convers. Manage.*, vol. 226, p. 113467, 2020, doi: 10.1016/j.enconman.2020.113467.

[10] M. Nourpour and M. H. K. Manesh, "Evaluation of novel integrated combined cycle based on gas turbine-SOFC-geothermal-steam and organic Rankine cycles for gas turbo compressor station," *Energy Convers. Manage.*, vol. 251, p. 114917, 2022, doi:10.1016/j.enconman.2021.115050.

[11] P. R. Silva, A. J. Leiroz, and M. E. Cruz, "Evaluation of the efficiency of a heat recovery steam generator via computational simulations of off-design operation," *J. Braz. Soc. Mech. Sci. Eng.*, vol. 42, no. 11, p. 569, 2020, doi: 10.1007/s40430-020-02655-1.

[12] G. Fortino et al., "Internet of things as system of systems: A review of methodologies, frameworks, platforms, and tools," *IEEE Trans. Syst., Man, Cybern., Syst.*, vol. 51, no. 1, pp. 223-236, 2020, doi:10.1109/tsmc.2020.3042898.

[13] R. P. Babakhodjaev and D. A. Mirzaev, "Numerical investigation of hydrodynamic processes in combustion chamber with an intensified fluidized bed," *J. Phys.: Conf. Ser.*, vol. 1683, no. 2, p. 022025, 2020, doi: 10.1088/1742-6596/1683/2/022025.

[14] E. V. Jithin et al., "A review on fundamental combustion characteristics of syngas mixtures and feasibility in combustion devices," *Renew. Sustain. Energy Rev.*, vol. 146, p. 111178, 2021, doi:10.1016/j.rser.2021.111178.

[15] H. Hu et al., "A case study of thermal performance of gas-steam combined cycle with gas turbine inlet air cooling and condenser deep cooling," *Case Stud. Therm. Eng.*, vol. 58, p. 104747, 2024, doi:10.1016/j.csite.2024.104747.

[16] V. Muniandy, M. S. Aziz, and H. F. Latip, "Study on the improvement of heat recovery steam generator efficiency-A review," *J. Adv. Res. Fluid Mech. Therm. Sci.*, vol. 94, no. 2, pp. 89-98, 2022, doi:10.37934/arfmts.94.2.8998.

[17] A. Ahmadi et al., "Applications of geothermal organic Rankine Cycle for electricity production," *J. Clean. Prod.*, vol. 274, p. 122950, 2020, doi: 10.1016/j.jclepro.2020.122950.

[18] R. Y. Dahham, H. Wei, and J. Pan, "Improving thermal efficiency of internal combustion engines: Recent progress and remaining challenges," *Energies*, vol. 15, no. 17, p. 6222, 2022, doi:10.3390/en15176222.

[19] M. A. Habib, G. A. Abdulrahman, A. B. Alquaity, and N. A. Qasem, "Hydrogen combustion, production, and applications: A review," *Alexandria Eng. J.*, vol. 100, pp. 182-207, 2024, doi:10.1016/j.aej.2024.05.030.

[20] S. Yan, M. A. Fazilati, D. Toghraie, M. Khalili, and A. Karimipour, "Energy cost and efficiency analysis of greenhouse heating system enhancement using phase change material: An experimental study," *Renew. Energy*, vol. 170, pp. 133-140, 2021, doi:10.1016/j.renene.2021.01.081.

[21] N. V. Burnete, F. Mariasiu, C. Depcik, I. Barabas, and D. Moldovanu, "Review of thermoelectric generation for internal combustion engine waste heat recovery," *Prog. Energy Combust. Sci.*, vol. 91, p. 101009, 2022, doi: 10.1016/j.pecs.2022.101009.

[22] J. Zhao et al., "Industrial reheating furnaces: A review of energy efficiency assessments, waste heat recovery potentials, heating process characteristics and perspectives for steel industry," *Process Saf. Environ. Prot.*, vol. 147, pp. 1209-1228, 2021, doi:10.1016/j.psep.2021.01.045.

[23] X. Feng et al., "Progress and key challenges in catalytic combustion of lean methane," *J. Energy Chem.*, vol. 75, pp. 173-215, 2022, doi:10.1016/j.jechem.2022.08.001.

[24] C. Ononogbo et al., "Opportunities of waste heat recovery from various sources: Review of technologies and implementation," *Heliyon*, vol. 9, no. 2, 2023, doi:10.1016/j.heliyon.2023.e13590.

[25] M. Zhou et al., "Environmental benefits and household costs of clean heating options in northern China," *Nat. Sustain.*, vol. 5, no. 4, pp. 329-338, 2022, doi: 10.1038/s41893-021-00837-w.

[26] A. J. Pimm, T. T. Cockerill, and W. F. Gale, "Reducing industrial hydrogen demand through preheating with very high temperature heat pumps," *Appl. Energy*, vol. 347, p. 121464, 2023, doi:10.1016/j.apenergy.2023.121464.

[27] M. Farghali et al., "Strategies to save energy in the context of the energy crisis: A review," *Environ. Chem. Lett.*, vol. 21, no. 4, pp. 2003-2039, 2023, doi: 10.1007/s10311-023-01591-5.

[28] Y. Qi et al., "A review on ammonia-hydrogen fueled internal combustion engines," *eTransportation*, vol. 18, p. 100288, 2023, doi:10.1016/j.etran.2023.100288.

[29] M. Mayrhofer et al., "Assessment of natural gas/hydrogen blends as an alternative fuel for industrial heat treatment furnaces," *Int. J. Hydrogen Energy*, vol. 46, no. 41, pp. 21672-21686, 2021, doi:10.1016/j.ijhydene.2021.03.228.

[30] Z. Yao, C. Romero, and J. Baltrusaitis, "Combustion optimization of a coal-fired power plant boiler using artificial intelligence neural networks," *Fuel*, vol. 344, p. 128145, 2023, doi:10.1016/j.fuel.2023.128145.

[31] E. C. Okafor et al., "Liquid ammonia spray combustion in two-stage micro gas turbine combustors at 0.25 MPa; Relevance of combustion enhancement to flame stability and NOx control," *Appl. Energy Combust. Sci.*, vol. 7, p. 100038, 2021, doi:10.1016/j.jaecs.2021.100038.

[32] Z. Yang et al., "Experimental study and demonstration of pilot-scale, dry feed, oxy-coal combustion under pressure," *Appl. Energy*, vol. 285, p. 116367, 2021, doi: 10.1016/j.apenergy.2020.116367.

[33] L. Chen, Y. Liao, and X. Ma, "Economic analysis on sewage sludge drying and its co-combustion in municipal solid waste power plant," *Waste Manage.*, vol. 121, pp. 11-22, 2021, doi:10.1016/j.wasman.2020.11.038.

[34] F. Pruvost, S. Cloete, C. A. del Pozo, and A. Zaabout, "Blue, green, and turquoise pathways for minimizing hydrogen production costs from steam methane reforming with CO2 capture," *Energy Convers. Manage.*, vol. 274, p. 116458, 2022, doi:10.1016/j.enconman.2022.116458.