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Optimizing gas-steam combined cycle with solar energy integration: A thermodynamic and economic analysis

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Abstract
The depletion of energy reserves worldwide has driven studies on optimizing existing power systems. Among all renewable clean energies, solar energy might offer a potential solution for gas-steam combined cycle systems widely used in power generation. The current study explores the thermodynamic and economic benefits resulting from the integration of solar energy into a combined cycle in an Integrated Solar Combined Cycle system. Based on the first law of thermodynamics, a thermodynamic model was developed in the present work, and simulations were run employing EBSILON Professional software. In this work, the ISCC system is compared to a traditional GTCC system: variations produced by the most relevant variables have been analyzed, namely DNI and ambient temperature. The results highlight that, with an input of natural gas of 10 kg/s, the ISCC system gives a cycle thermal efficiency of 44.1%, whereas that from GTCC is 39.8%. Further, ISCC decreases natural gas consumption by 2.638 kg/s at identical output power. Eventually, solar energy integration results in increased system efficiency and reduced fuel consumption, promising some encouraging economic benefits. Thus, ISCC systems are positioned as feasible options for future energy requirements.

Keywords: Integrated solar combined cycle, gas-steam combined cycle, solar energy integration, thermodynamic efficiency

1. Introduction

The energy sector has been facing excessive stress with the exponential draining of fossil fuel reserves and the increased demand for cleaner and greener sources. GTCC systems have emerged as one of the widely adopted solutions for power generation; these are popular owing to their high efficiency and lower emissions compared to conventional power plants [1]. Regardless of these advantages, further improvements in efficiency and fuel saving are urgently required with the view to the energy demand that will need to be covered in the future by sustainable sources only [2, 3]. Solar energy, renewable and clean, represents another possibility to enhance GTCCs' performance further. Integrated Solar Combined Cycle systems will significantly increase thermal efficiency and decrease fossil fuel consumption by incorporating solar power with conventional gas turbines [4]. Various studies have determined that ISCC systems increase overall system efficiency and offer significant environmental and economic benefits by reducing greenhouse gas emissions and dependence on non-renewable energy sources [5-7].

Various works have been done on the advantages of integrating solar energy into combined cycle systems. Amani and Ghenaiet, 2020 [8], proposed a new hybridization of solar central receiver systems with combined cycle power plants and showed the possibility of efficiency enhancement. Shaaban, 2024 [9], optimized ISCC systems for the cogeneration of electricity and fresh water and demonstrated the advantages of multi-energy complementarity. Lu *et al.* (2024) [10], proposed dynamic models for coordinated control in combined cycles to improve the performance and flexibility of these systems. Settino *et al.* (2023) [11], performed energy analyses on hybrid solar-natural gas systems, focusing on the efficiency gains in ISCC systems. Assareh *et al.* (2023) [12], worked on solar-assisted combined gas power cycles in diverse climates, which also speaks to the adaptability of ISCC systems. Part-load operation and exergoeconomic analysis of ISCC systems were performed by Diaz Pérez *et al.* in 2023 [13] and Elmorsy *et al.* in 2022 [14], highlighting their flexibility in operation and viability from an economic point of view. Nourpour and Khoshgoftar Manesh

(2022) [15], investigated new cycle combinations, such as geothermal and organic Rankine cycles, for hybrid systems. Wang *et al*. (2022) [16] performed the thermodynamic and economic analyses for the integrated power and steam generation systems using solar photovoltaic/thermal collectors. Khandelwal *et al*. (2022) [17] conducted a comparative performance evaluation of different ISCC plants using various heat transfer fluids focusing on energy storage and efficiency improvements. Al-Hamed and Dincer (2022) [18], proposed a novel integrated solar combined cycle (ISCC) system with carbon capture using a supercritical CO_2 promising system for effective carbon emission reduction.

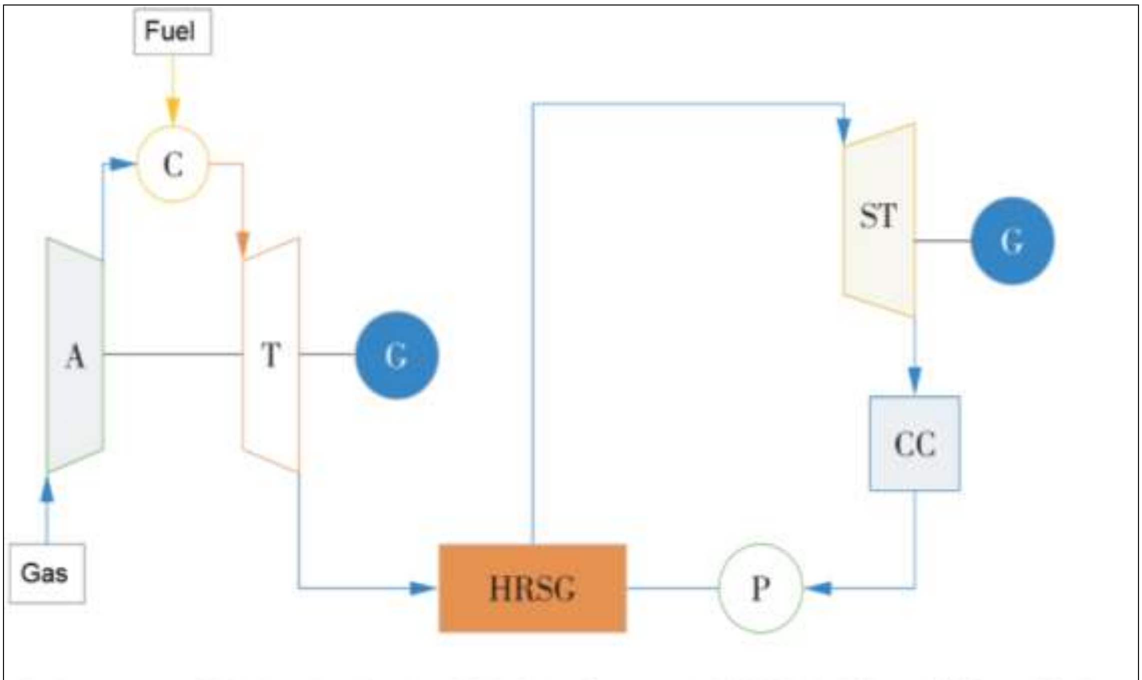
Adnan *et al*. (2022) [19], discussed the thermo-economic analysis of hybrid ISCC systems with concentrated solar power towers, which proved economically and ecologically viable. He *et al*. (2022) [20], investigated combined cooling, heating, and power systems integrated with compressed air energy storage and gas-steam cycles for adding new dimensions of energy storage and multi-use applications. Zhang *et al*. (2022) [21] developed the general performance evaluation method for ISCC systems and gave a holistic framework for optimizing these systems. AlKassem (2021) [22], conducted a performance evaluation for an ISCC power plant integrated with a solar tower in Saudi Arabia. He showed that adding solar to ISCCs is crucial for increasing power output and saving fuel consumption, especially for locations with high solar irradiance. Carapellucci and Giordano (2021) [23], discussed the repowering of combined cycle plants using regenerative gas turbines with steam injection, focusing on improvements in part-load performance and overall efficiency, which are essential for maximizing energy output in variable demand. Abubaker *et al*. (2021) [24], used multi-objective optimization techniques in hybrid solar-gas turbine combined cycles with absorption inlet-air cooling to further enhance system efficiency.

These studies, although important in laying a foundation in the field of ISCC systems, fall short by not considering comprehensive research on the thermodynamic performance comparison of ISCC systems under operational condition variations, such as DNI and ambient temperature. Furthermore, less attention has been given to the economic aspects of integrating solar energy into GTCC systems, especially on savings by fuel and efficiency improvement in the long run. This paper aims to overcome these lacunae through the thermodynamic and economic comparison of the ISCC system with a conventional GTCC system. For that, this work simulates both systems by considering key variables, such as DNI and ambient temperature, within changing environmental conditions by implementing a detailed thermodynamic model that uses the first law of thermodynamics. The analysis evaluates the impact of the integration of solar energy on thermal efficiency, fuel savings, and overall performance. The results give a deep insight into optimizing the complementary use of solar energy in gas-steam combined cycles with thermodynamic and economic advantages.

2. System Description

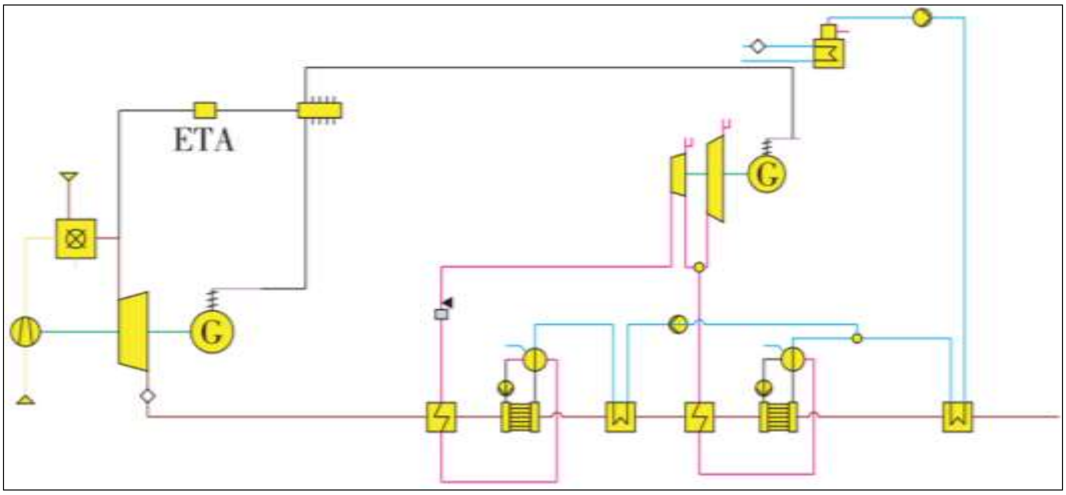
2. 1 Construction of system model

Compared to conventional coal-fired generation, the GTCC system fully combines the benefits of single-gas turbine power generation technology with steam power generating technology and further has more scope for development in the future. The gas turbine and steam power cycle work together, with high-temperature exhaust gases from the gas turbine heating the waste heat boiler to generate higher-temperature steam. This steam is exchanged at both high and low-pressure stages to enhance power generation and cycle thermal efficiency, represented as η. The GTCC system model is shown in Figure 1.



A. Compressor; C. Combustion Chamber; T. Turbine; G. generator; HRSG. Heat Waste; ST. Steam Turbine; CC. Condenser; P. Pump Feedwater

a)



b)

Fig 1: GTCC System Model a) Operational Workflow b) Architectural Framework

Passing through the compressor in the gas turbine raises the air's pressure and temperature to some extent. Then, it mixes with natural gas and undergoes complete combustion in the combustion chamber. Complementing this with the traditional steam power cycle after the turbine's exhaust gas expansion forms a high-intake, low-exhaust temperature GTCC system.

The energy equation for the gas turbine part is as follows:

$$Q_{in} = Q_{gas,in} + Q_{fuel} \tag{1}$$

$$Q_{out} = \frac{E_{gt}+E_{loss}}{E_{tag}} + Q_{gas,out} + Q_{cool} \tag{2}$$

$$\eta_{bd} = \frac{Q_{in} - Q_{out}}{Q_{in}} \tag{3}$$

Where: Q_{in}, Q_{out} are the total input and total output heat of the gas turbine, MW; $Q_{gas,out}$, $Q_{gas,in}$ are the air heat at the outlet and inlet of the gas turbine, MW; Q_{fuel} Is the heat of the natural gas input into the combustion chamber of the gas turbine, MW; E_{gt} Is the power generation power of the gas turbine, MW; E_{tag} is the power loss of the auxiliary components in the gas turbine, MW; Q_{cool} is the cooling loss, MW; η_{bd} Is the efficiency of the gas turbine generator; E_{loss} Is the power generation loss of the gas turbine, MW.

The system cycle thermal efficiency is the ratio of the total output power to the inlet input power.

$$\eta_c = \frac{E_{GTCC}}{q_{m,fuel} \times Q_{LHV} \times 10^3}, \tag{4}$$

$$E_{GTCC} = E_{GTCC,gt} + E_{GTCC,st} \tag{5}$$

Where: η_c is the thermal efficiency of GTCC; $E_{GTCC,gt}$ is the power generation of the gas turbine in the GTCC system; $E_{GTCC,gt}$ is the power generation of the steam turbine in the GTCC system, MW; E_{GTCC} is the total output power of the combined cycle system, MW; $q_{m,fuel}$ is the mass flow rate of natural gas fuel entering the gas turbine, kg/s; Q_{LHV} is the lower calorific value of natural gas, kJ/kg.

The flue gas at the gas turbine outlet enters the economizer, heat exchanger and evaporator of the waste heat boiler for heat exchange, and the high-pressure and low-pressure steam generated enters the steam turbine to generate electricity. If the steam turbine component is set as an isentropic work thermodynamic model, the steam turbine output power is

$$E_{st} = \left[(h_1 - h_2) \cdot q_{m,in}\right] \cdot \eta_{mech} - Q_{loss}, \tag{6}$$

where: $q_{m,in}$ is the steam mass flow rate at the steam turbine inlet, kg/s; h_1 and 和 h_2 are the steam turbine inlet/outlet specific enthalpy, kJ/kg; η_{mech} is the mechanical energy loss efficiency; Q_{loss} is the mechanical energy loss, kW.

2.2 ISCC system model: The ISCC system studied in this paper is a system in which solar energy is coupled to the top cycle. Tower solar thermal collectors are generally suitable for high-temperature heat collection. Therefore, this paper introduces tower solar thermal collectors to complement the top cycle. The ISCC system model is shown in Figure 2 [15, 16]. After the heliostat field reflects solar energy, the heat is concentrated on the heat receiver of the solar thermal collector tower. The molten salt is heated and enters the solar thermal collector tower through the molten salt pump; the heated high-temperature molten salt heats the outlet high-pressure air of the gas turbine compressor through the heat exchanger so that the high-temperature air enters the combustion chamber and is thoroughly mixed with the fuel for combustion. Then, the outlet flue gas enters the steam power cycle for power generation [15-16]. The simulation calculation process of the tower solar thermal collector requires meteorological data such as the direct normal irradiance (DNI) of the sun and the atmospheric ambient temperature. The mathematical model of DNI is:

$$I_{DNI} = I_{GHI} - I_{DHI} \cos A_{SZA} \tag{7}$$

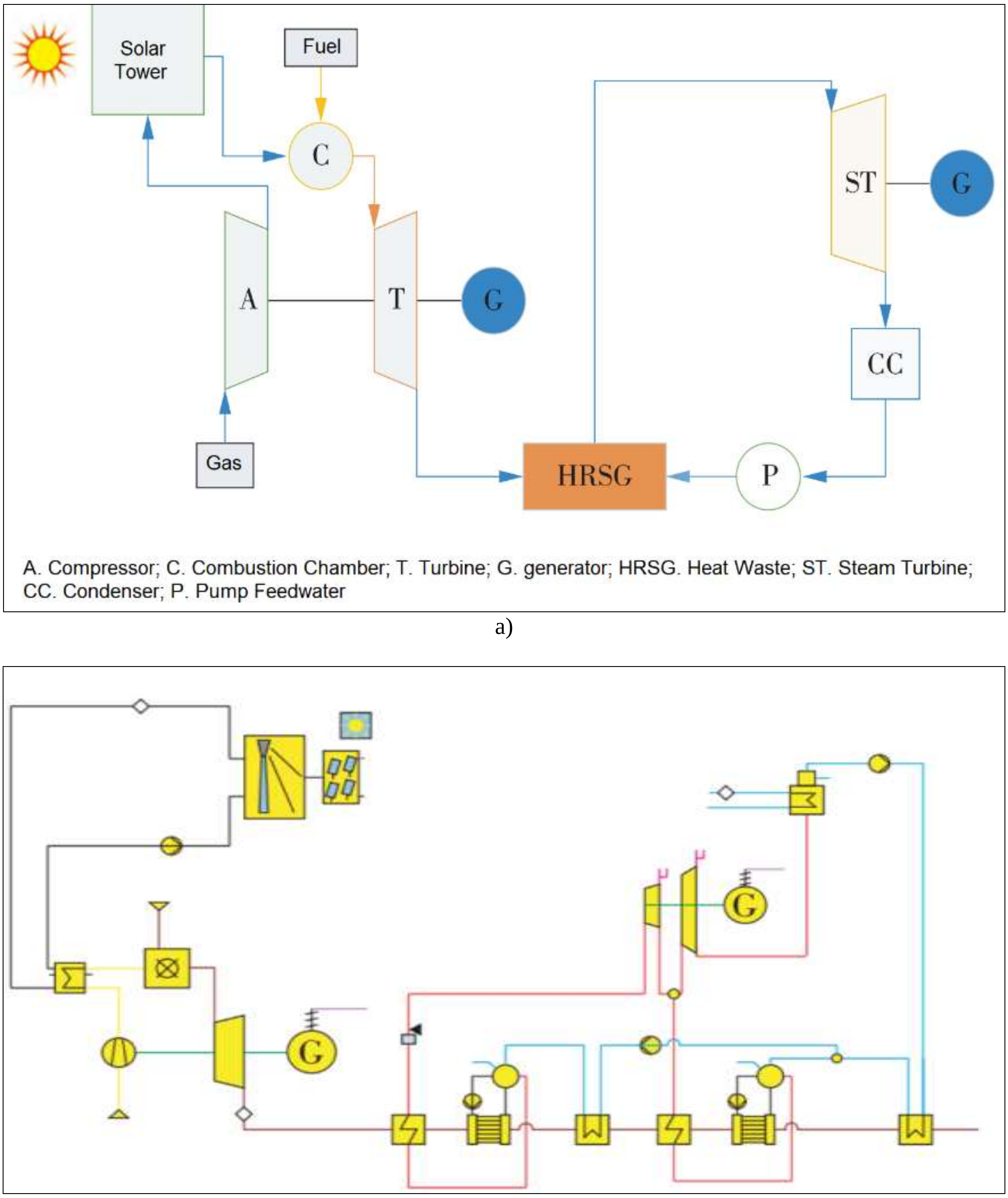
Where: A_{SZA} is the solar zenith angle, (°); I_{DNI} is the solar diffuse radiation, W/m2; I_{DHI} is the solar total radiation, W/m2.
The light-heat conversion process of the tower solar thermal collector is based on the relevant thermodynamic properties.

$$Q_s = A_s \times I_{DHI} \tag{8}$$

Where: Q_s is the total solar heat received, MW; A_s is the total reflecting area of the heliostat field, m^2.

$$Q_{tower} = Q_s \cdot \eta_{foc} \cdot \eta_{ref} \cdot \eta_{mar} \cdot \eta_{wind} \tag{9}$$

Where: Q_{tower} do the tower receiver, MW receive the heat; η_{foc} is the lens tracking efficiency; η_{ref} is the mirror reflection efficiency; η_{mar} is the mirror field two-dimensional array efficiency; η_{wind} is the wind speed influencing factor.



a)

b)

Fig 2: ISCC system model a) Operational Workflow b) Architectural Framework.

The thermal calculation process in the tower solar thermal collector is as follows:

$$Q_{solar} = Q_{tower} - Q_{s,loss} \tag{10}$$

Where: Q_{solar} is the available energy of solar radiation heat, MW; $Q_{s,loss}$ is the heat loss of the collector, MW.
In a tower solar collector, the heat absorbed by the heat exchange fluid in the receiver

$$Q_{salt} = q_{m,HTF}(h_{out} - h_{in}) \tag{11}$$

Where: $q_{m,HTF}$ is the mass flow rate of the heat exchange fluid entering the receiver, kg/s; h_{out} is the specific enthalpy of the heat exchange fluid at the receiver outlet, kJ/kg; h_{in} is the specific enthalpy of the heat exchange fluid at the receiver inlet, kJ/kg.

The cycle thermal efficiency of the ISCC system refers to the ratio of the sum of the gas turbine power ($E_{ISCC,g}$)t and the steam turbine power generation part output power ($E_{ISCC,s}$)t in the power generation system to the power of the input part [19]. Among them, the heat of the input part includes the heat absorbed by the solar collector and the calorific value of the natural gas entering the combustion chamber, that is,

$$E_{ISCC} = E_{ISCC,gt} + E_{ISCC,st} \tag{12}$$

$$\eta_{ISCC} = \frac{E_{ISCC}}{Q_{solar} + q_{m,fuel} \times Q_{LHV} \times 10^3} \tag{13}$$

Where: η_{ISCC} is the cycle thermal efficiency of the ISCC system.

3. Results and discussion

3.1 System operation parameter simulation

After the GTCC power generation system model is built, the inlet boundary value is set, and the EBSILON simulation software is used for numerical calculation to analyze the main operating performance parameter values of the gas-steam combined cycle power generation system and the main parameter change characteristics under the design condition and variable condition are explored respectively. The gas turbine inlet working fluid parameter values and the main system operating parameters of the GTCC system are shown in Tables 1-2, respectively.

Table 1: Gas turbine working fluid parameters

Working fluid	Pressure (kPa)	Temperature (℃)	Flow (kg·Sec.$^{-1}$)
Air	100	20.0	500
Natural gas	1000	20.0	10
Flue gas	100	600.9	510

Table 2: Main parameters of the GTCC system

Equipment	Unit
Gas turbine power generation	167.059(MW)
Steam turbine power generation	100.099(MW)
Combined cycle power generation	267.158(MW)
Gas turbine power generation efficiency	24.9(%)
Cycle thermal efficiency	39.8(%)
Exhaust gas temperature	83.374(℃)

After the ISCC power generation system model is built, the EBSILON simulation software is used to perform numerical calculations on it, analyze the impact of the solar energy coupling part on the thermodynamic performance of the traditional combined cycle, and explore the main parameter change characteristics under design conditions and variable conditions. The main parameter values of the ISCC system model are shown in Table 3 [21].

Table 3: Main parameters of the ISCC system model

Condition	Parameters	Unit
Environmental Parameters	DNI/	1000 (W·m^{-2})
	Ambient temperature	20 ℃
Project	Gas turbine power generation	215.205 (MW)
	Steam turbine power generation	138.065(MW)
	Total power generation	353.270(MW)
	Cycle thermal efficiency	44.1 (%)
	Exhaust temperature	65.249 (℃)

3.2 Comparison of the operating performance of ISCC and GTCC systems

In the Integrated Solar Combined Cycle, gas and steam turbines produce output contributing to the total. With the assumed natural gas input rate of 10 kg/s, the GTCC system gives the calculated total output power of 267.158 MW. In this regard, the output power of the ISCC system becomes 353.270 MW, i.e., 86.112 MW more than that of the GTCC system. The significant rise in output power is because solar energy was combined with a gas turbine. Solar energy is used to augment the system's performance by heating the high-pressure air at the outlet compressor with a heat exchanger to increase the global output of the ISCC system. Figure 3 describes in more detail the total outlet power of the two systems compared.

Compared to the traditional GTCC power generation system, the ISCC system incorporates a tower solar thermal collector, which raises the initial temperature of the compressed air through heat exchange. This modification allows each system component to produce more power while the additional input power comes only from the heat exchange between solar energy and compressed air [20]. Under the design conditions, the input power for the traditional GTCC system is 671.458 MW, with a cycle thermal efficiency of 39.7%. In contrast, the ISCC system requires 801.812 MW of input power but achieves a cycle thermal efficiency of 43.1%.These results indicate that the integration of solar thermal energy increases the cycle thermal efficiency of the ISCC system by 4.31%. A comparison of the cycle thermal efficiencies of both systems is shown in Figure 4. Figure 4 shows that the cycle thermal efficiency of the ISCC power generation system surpasses that of the traditional GTCC system under varying natural gas input conditions. The cycle thermal efficiency of the GTCC system ranges from 25% to 40%, whereas the ISCC system can exceed 35%. The findings indicate that, with constant natural gas input, the integration of solar energy positively enhances the system's operational performance.

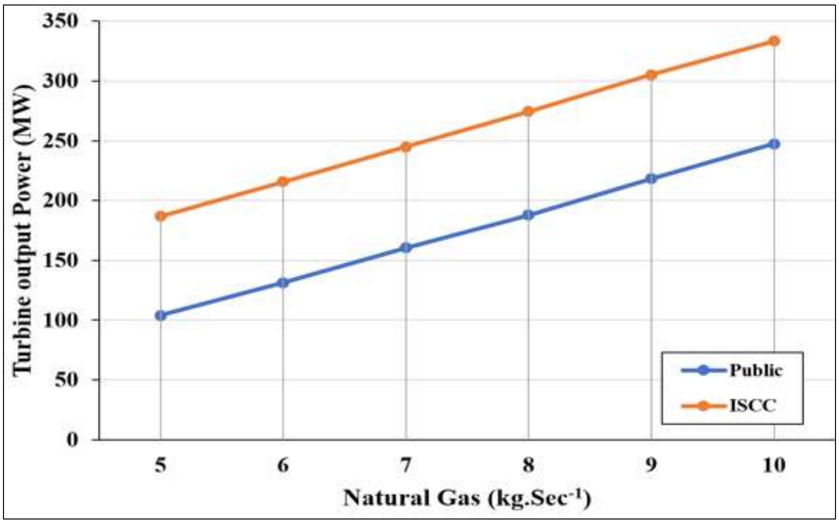


Fig 3: Comparison of total system output power.

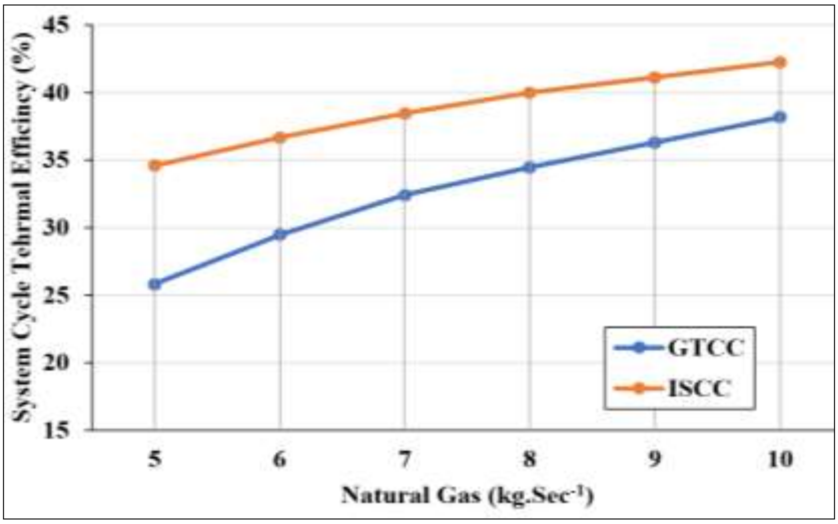


Fig 4: Comparison of system cycle thermal efficiency.

Figure 5 presents the temperature of exhaust gases from the waste heat boiler in both schemes. Under operating conditions with variable natural gas input, the average temperature values of the exhaust gases produced in the heat recovery boiler generated in traditional GTCC were 83.725 °C. In comparison, those from ISCC were considerably lower at 68.275 °C. That indicates that the heat exchange performance in the ISCC system is superior to that of the conventional GTCC system in its efficient operation, with high-temperature flue gas entering and low-temperature exhaust gas exiting.

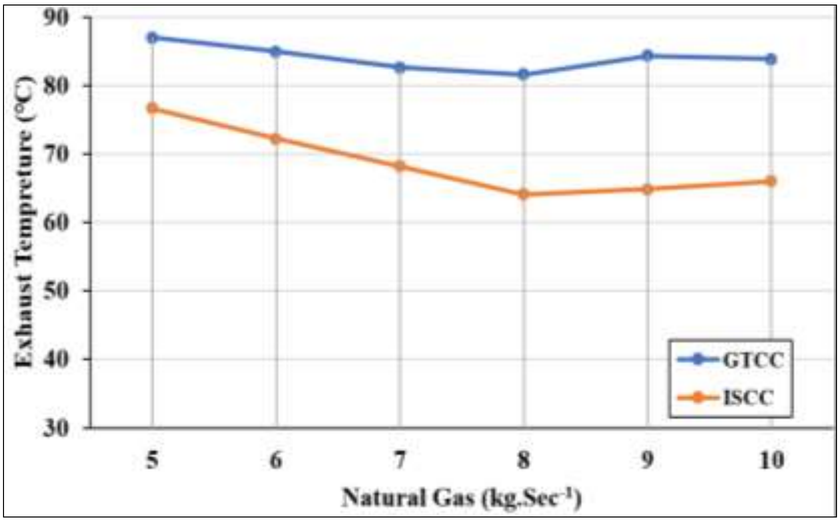


Fig 5: Comparison of exhaust gas temperature of waste heat boiler in the system.

3.3 Analysis of the complementary performance of ISCC

In the current study, main simulations and analyses were done for the trends of change in operational characteristics concerning the DNI variation range from 400 to 1000 W/m² with steps of 100 W/m². These were simulation tests for the performance of the ISCC system, essentially the complementary interaction between the solar energy and the gas turbine after integration. It is then that solar energy becomes necessary for performance gain in the Brayton cycle. More precisely, the heat from the solar collector raises the compressed air temperature at high pressure in the compressor outlet, which is utilized to enhance the overall output power of the gas turbine. Due to the increased DNI, there will be a gradual rise in both the temperature of the compressed air and power from the gas turbine. This rise in DNI provides increased heat transfer between solar energy and compressed air, further magnifying the gas turbine's output. With the increase in the share of solar energy input, the operational parameters of the gas turbine are directly affected and optimized for better thermal efficiency in the cycle with improved fuel saving. Figure 6 details the trend of the key operating parameters of the gas turbine as DNI increases, thus highlighting the positive contribution of solar energy to the performance of the ISCC system.

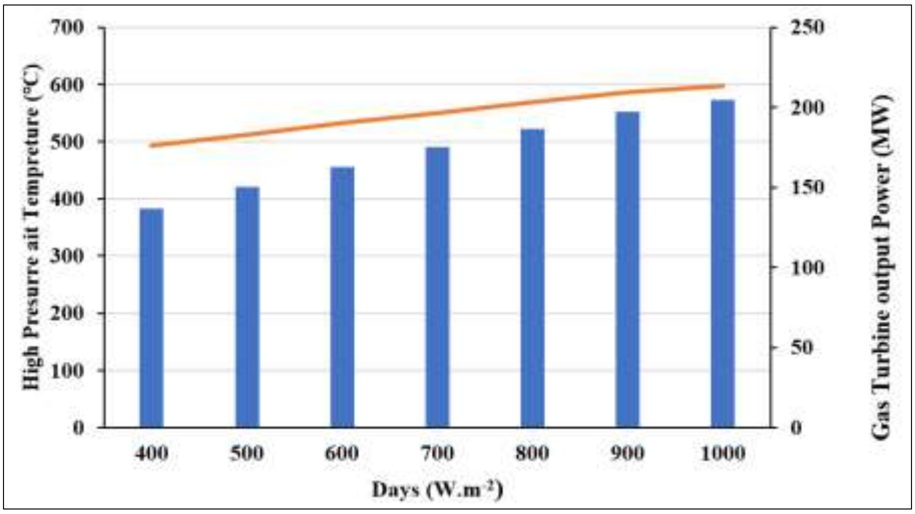


Fig 6: Impact of DNI on High-pressure air temperature and Turbine output Power Variation.

Figure 7 presents how the system cycle thermal efficiency varies for DNI. When DNI increases, more heat is absorbed by the solar thermal collector through the receiver, thus increasing the thermal efficiency of the power generation system under ISCC. This behavior is because increased DNI implies increased magnitudes of solar energy input, which is transformed into thermal energy and transferred to the air at high pressure in the Brayton cycle. The increased DNI can capture more amounts of solar energy; thus, this leads to improved heat exchange, hence efficiency in power generation. When DNI is at the peak value of 1,000 W/m², heat absorption from the solar thermal collector at the tower is maximum, giving a significant rise in the cycle thermal efficiency of the system. Correspondingly, for this optimum value, the efficiency has reached its peak value of 44.1%, which means the ISCC system is at its best performance. This peak in DNI falls at the time of the day when the solar intensity is maximum; hence, the system could utilize the available solar energy to its fullest. The increased heat extracted from the solar collector is used effectively to raise the performance of the gas turbine, reducing fuel consumption and increasing energy output. The efficiency that keeps increasing with the rise in DNI underlines the importance of solar energy in supplementing the gas turbine cycle for overall system improvement.

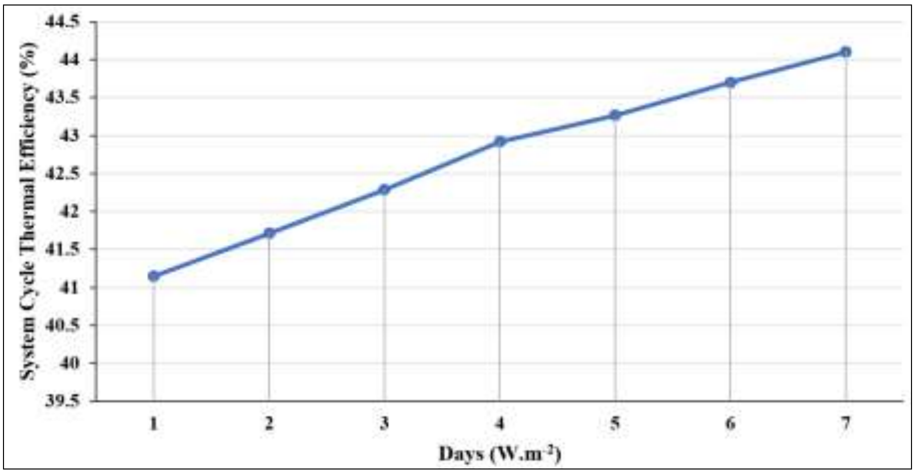


Fig 7: Influence of system cycle thermal efficiency with DNI

The complementary nature of the cyclic characteristics of the solar energy input and Brayton cycle performance versus atmospheric ambient temperatures was studied for an ambient temperature of -5 to 35 °C in 5 °C increments. As the simulation results demonstrate, atmospheric ambient temperature variation tends to affect the performances of some gas turbine key components, such as the compressor and combustion chamber [23]. Either too high or too low, extreme ambient temperatures adversely affect these components' operational efficiency. The main operating parameters of the gas turbine are highlighted in Figure 8. The influence of these changes was explored by simulating different atmospheric ambient temperatures on the gas turbine's main operating parameters. As shown in Table 4, the thermal efficiency of the cycle of the system has an optimal value of 44.1% within the ambient temperature range between 20 °C and 30 °C. When the ambient temperature is above 30 °C or below 0 °C, the ISCC system's thermal efficiency will gradually decrease. This phenomenon occurs because extremely high or low temperatures will have a negative effect on the gas turbine. Results of the study show that high summer temperatures and low winter temperatures greatly impact operating efficiency throughout the year.

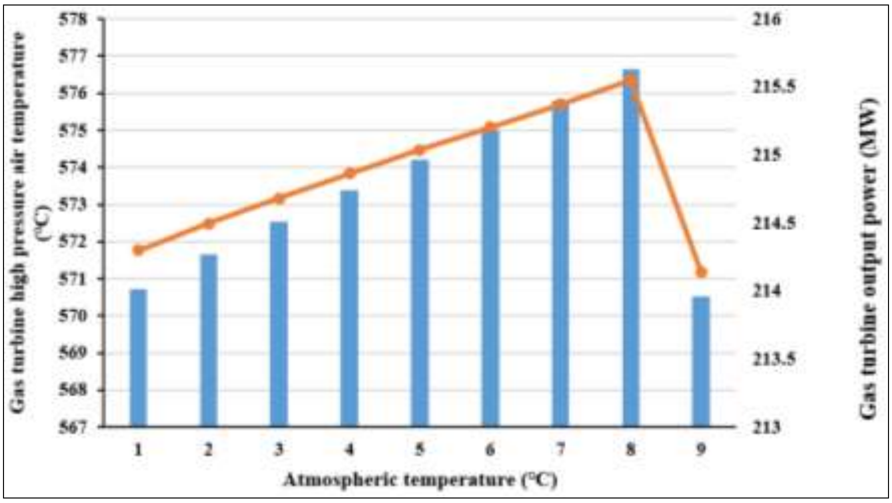


Fig 8: Main parameters of gas turbine under variable ambient temperature conditions

Table 4: Impact of atmospheric ambient temperature on System Cycle Thermal Efficiency (%)

Atmospheric temperature (°C)	System Cycle Thermal Efficiency (%)
-5	43.89639
0	44.00241
5	44.00241
10	44.00241
15	44.00241
20	44.10602
25	44.10602
30	44.10602
35	43.9012

3.3 Energy Efficiency Enhancement in ISCC Systems

In the ISCC system, solar energy is concentrated by the heliostat field onto the tower solar receiver, where heat is generated. The heated molten salt is circulated to a heat exchanger, which heats the high-pressure compressed air exiting the compressor before it enters the combustion chamber. Integrating solar energy within the traditional GTCC increases the system cycle thermal efficiency and decreases the natural gas required for power generation. When the input of natural gas is at 10 kg/s, the total output power from the GTCC system becomes 267.158 MW with a cycle thermal efficiency of 39.8%. In the same case, the ISCC system only needs 7.362 kg/s of natural gas to provide the same output, saving 2.638 kg/s. The tower solar collector provides this saved energy input to the gas turbine through the heat exchanger to realize a more energy-saving operation of the ISCC. Figure 9 presents the thermodynamic cycle efficiency of the ISCC system versus the turbine exhaust temperature and natural gas input.

Along with the increase in the input of natural gas, the turbine exhaust temperature increases, and the cycle thermal efficiency of the system improves. In cases where natural gas input is ≤ 6 kg/s, the thermal efficiency growth rate surpasses the exhaust temperature growth rate. At a natural gas input rate of 10 kg/s, the thermal efficiency peaks to 44.1%. The temperature at the turbine exhaust reaches 734.611 °C, thus demonstrating the best performance under design conditions for the ISCC system.

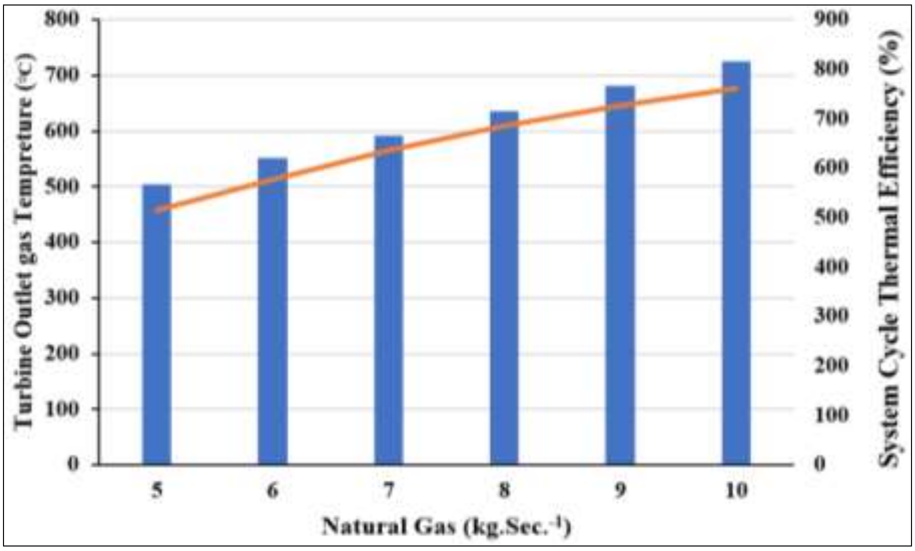


Fig 9: System cycle thermal efficiency changes with turbine outlet temperature

4. Conclusion

The present work compares the performances of a new Integrated Solar Combined Cycle with the traditional Gas-Steam Combined Cycle under different operative scenarios. The key results that have come out are the following:

1. The ISCC system realized a cycle thermal efficiency of 4.3% higher and output power 86.112 MW more than the GTCC system with constant natural gas input.
2. Under optimum conditions, with a DNI of 1,000 W/m², the ISCC system reached its best performance: the cycle thermal efficiency was 44.1%, and the compressed air temperature was 575°C.
3. ISCC performed at its best at ambient temperatures between 20 and 30 °C. Otherwise, this system would lose efficiency when it reached below 0 °C or above 35 °C. Additionally, the system would save 2.638 kg/s of natural gas, benefiting the environment.

Further studies shall be conducted on integrating appropriate energy storage systems to help stabilize performance fluctuations due to environmental conditions. Moreover, scalability research of ISCC systems at variable geographical locations and turbine design optimization is expected to offer increased flexibility and efficiency at extreme temperature fluctuations, thereby contributing to the energy solution roadmap.

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