Anusandhanvallari
Vol 2024, No.1
November 2024
ISSN 2229-3388

Energy and Exergy Analysis of a Parabolic Trough-Assisted Brayton–Rankine Hybrid Power Cycle under Tropical Climatic Conditions

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Abstract: Integrated solar combined-cycle (ISCC) plants offer a low-risk route to cleaner generation in regions with a strong solar resource. This study develops and validates a thermodynamic model of a parabolic trough-assisted hybrid plant that couples a Brayton-cycle gas turbine block (2 × 40 MWe) with a Rankine-cycle steam turbine (80 MWe) through two heat recovery steam generators and a solar steam generator. Site-specific radiation data for Kalaburagi, Karnataka, drive a 183,120 m^2 trough field. Each component was represented by mass and energy balances, and the integrated model was used to evaluate solar-field output, steam-cycle and overall thermal efficiencies, solar-to-electric conversion and a second-law (exergy) measure of performance. The model was validated against hourly net output measured at the Hassi R'mel ISCC plant. Over 24 h the present model reproduced the measured energy within +0.98%, with a mean absolute percentage error of 3.34% and a normalised RMSE of 4.1%, against 6.39% and 6.8% for a reference model. The combined-cycle-mode thermal efficiency reached 67%, corresponding to an exergy efficiency of about 64%, while the solar field operated at an exergy efficiency of 21–28%. The framework provides a robust basis for assessing solar-assisted combined cycles in high-irradiance tropical climates.

Keywords: Integrated solar combined cycle; Brayton–Rankine cycle; Parabolic trough collector; Exergy analysis; Model validation

1. Introduction

Combined-cycle gas turbine plants already convert natural gas to electricity at efficiencies close to 60%, and adding a parabolic trough field that raises steam for the bottoming cycle is one of the cheapest ways of introducing solar heat into a dispatchable plant. In this integrated solar combined cycle (ISCC), solar energy displaces part of the heat that would otherwise be supplied by duct firing or by a larger gas flow, and the steam turbine converts it with the efficiency of a modern Rankine cycle rather than that of a small solar-only plant [1,2]. Parabolic trough fields are preferred for this duty because they are commercially mature and operate in the 290–393 °C range that suits evaporator and superheater integration [1]. The concept was first explored through thermoeconomic optimisation [3], and several plants have since been built in North Africa, the Middle East and the United States.

Dersch et al. [4] compared integrated solar combined-cycle systems with solar-only trough plants of the SEGS type and with conventional combined cycles. Using detailed plant models in GateCycle and IPSEpro, they simulated annual performance and estimated the cost of the solar electricity produced by each configuration. They concluded that integration into a combined cycle was the most favourable route for trough technology among the options studied, because the solar steam benefits from the high-efficiency bottoming cycle and from shared power-block equipment.

Hosseini et al. [5] carried out a technical and economic assessment of six thermal plant options for Iran, including a SEGS-type plant, a simple gas turbine and ISCC plants with 67 MW and 33 MW solar fields. They found that

an ISCC with a 67 MW solar field was the most suitable configuration for the country's first solar plant. Over a 30-year life it would save about US$59 million in fuel and avoid about 2.4 Mt of CO_2. Its levelised energy cost, including environmental costs, was about 10% lower than that of a combined cycle and 33% lower than that of a gas turbine.

Behar et al. [6] modelled the instantaneous performance of the Hassi R'mel ISCC plant in Algeria, the first such plant in the country, using a program written in Visual FORTRAN. At night the plant delivered about 134 MW at 57.5% efficiency. During solar operation the output rose by about 17% to roughly 157 MW, and the net solar electricity ratio reached about 15% at the design point. The same plant layout, with two gas turbines, an 80 MW steam turbine and a 183,120 m^2 trough field, is adopted as the reference configuration in the present study.

Derbal-Mokrane et al. [7] simulated the annual behaviour of the Hassi R'mel ISCC with TRNSYS and the Solar Thermal Electric Components library. They assembled the technical data needed to model the trough field, the heat recovery and solar steam generators and both turbines. The simulations indicated a plant of about 150 MW with a solar contribution of around 30 MW and an overall efficiency near 52%. **Achour et al.** [8] later developed an ISCC model for the hotter conditions of southern Algeria, validated it against operating data from Hassi R'mel and obtained good agreement. This confirmed that a model calibrated on one site can be transferred to another climate once the radiation input is replaced.

Baghernejad and Yaghoubi [9] performed energy, exergy and exergoeconomic analyses of the Yazd ISCC in Iran and optimised it with a genetic algorithm. They reported an exergy efficiency of about 46.8% and showed that optimisation reduced the cost associated with exergy destruction by about 14.8%. **Reddy et al.** [10] applied exergy analysis to a solar-concentrator-aided natural-gas combined cycle and identified the solar collectors and the combustion chamber as the dominant sites of exergy destruction. **Zhu et al.** [11] evaluated solar integration into a natural-gas combined cycle at several integration points and found that the incremental solar-to-electric efficiency could exceed that of a stand-alone solar steam cycle. **Li and Yang** [12] showed that introducing solar heat in two stages rather than one raised the solar-to-electric efficiency by about 1.2 percentage points.

Most of these studies concern arid desert sites in North Africa and the Middle East. Tropical Indian conditions differ in a decisive way, because the south-west monsoon removes much of the beam radiation for three to four months while the dry winter offers DNI above 6 kWh m^{-2} day^{-1}. No validated Brayton–Rankine ISCC model driven by measured radiation for Northern Karnataka has been reported. Nor have the first-law results of such a plant been complemented by a second-law measure that separates solar-field degradation from power-block performance. The present work develops a component-level energy model of an ISCC driven by Kalaburagi radiation data, validates it hour by hour against measured Hassi R'mel output, and extends it with an exergy assessment of the solar field and of the plant in combined-cycle mode.

2. Experimental Methodology

2.1 Plant configuration and input data

The modelled plant (Fig. 1) is a conventional combined cycle with two SGT-800 gas turbines of 40 MWe each and one SST-900 steam turbine of 80 MWe, coupled to two identical single-pressure heat recovery steam generators (HRSGs) and a solar steam generator (SSG) that operates in parallel with the HRSG evaporators [1,6]. During the day, heat from the trough field is merged with heat recovered from the gas-turbine exhaust; at night, or when the field is shut down, the plant runs as a plain combined cycle. The main power-block components are shown in Fig. 2 and the design data are listed in Table 1. Radiation inputs were the IMD monthly mean daily global and diffuse radiation for Kalaburagi (17.35° N, 76.85° E), from which hourly DNI on the collector aperture was obtained with the Collares-Pereira and Rabl and Liu and Jordan disaggregation [13,14]. The annual mean DNI and GHI are 5.28 and 5.79 kWh m^{-2} day^{-1}.

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Vol 2024, No.1
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ISSN 2229-3388

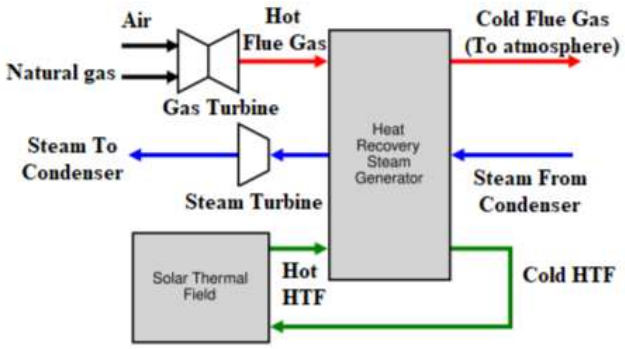


Fig. 1. Schematic of the integrated solar combined cycle with gas turbines, heat recovery steam generators, solar steam generator, steam turbine and parabolic trough solar field.

Table 1. Design data of the modelled ISCC plant.

Parameter	Value
Gas turbines	2 × 40 MWe, SGT-800 type (Brayton cycle)
Steam turbine	1 × 80 MWe, SST-900 type (Rankine cycle)
Heat recovery	2 single-pressure HRSGs + 1 solar steam generator
Steam turbine inlet	83 bar, 560 °C
Extraction to deaerator	4.8 bar, 200 °C; deaerator about 7 bar
Solar field	183,120 m^2; 56 loops × 6 collectors; N–S axis
Collector	Aperture 7.76 m × 99 m (545 m^2); concentration ratio 82; $\eta_o = 0.80$
Heat transfer fluid	Therminol VP-1, 290 → 393 °C
Site radiation (Kalaburagi)	DNI 5.28; GHI 5.79 kWh m^{-2} day^{-1} (annual means)

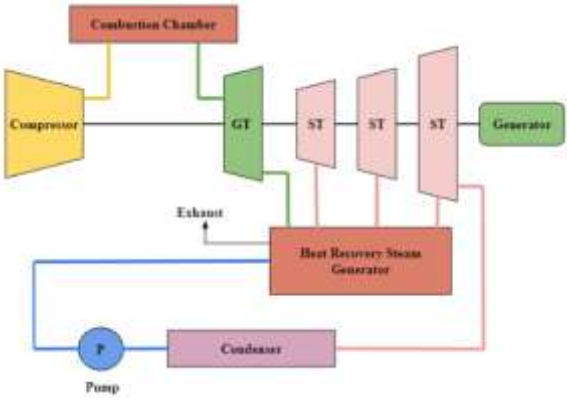


Fig. 2. Power-block arrangement of the hybrid plant showing the gas turbines (GT), heat recovery steam generators and steam turbine (ST).

2.2 Gas-turbine (Brayton) model

Each gas turbine was represented by the air-standard Brayton cycle of Fig. 3a with non-ideal compression and expansion [15,16]. With pressure ratio k_p, the compressor outlet temperature and the specific work of the turbine and compressor are:

$$T_2 = T_1[1 + (k_p^{(\gamma-1)/\gamma} - 1)/\eta_C], \quad w_C = c_{pa}T_1(k_p^{(\gamma-1)/\gamma} - 1)/\eta_C, \quad w_T = c_{pg}T_3\eta_T(1 - k_p^{-(\gamma_\{g\}-1)/\gamma_g\}}) \tag{1}$$

where T_1 is the ambient temperature and T_3 the turbine inlet temperature. The specific heats of air and combustion gas were taken as temperature-dependent [8]:

$$c_{pa} = 1.0189\times10^3 - 0.13784T_a + 1.9843\times10^{-4}T_a^2 + 4.2399\times10^{-7}T_a^3 - 3.7632\times10^{-10}T_a^4 \tag{2}$$

$$c_{pg} = 1.8083 - 2.3127\times10^{-3}T + 4.045\times10^{-6}T^2 - 1.7363\times10^{-9}T^3 \tag{3}$$

with $T_a = (T_1 + T_2)/2$. The fuel–air ratio, net power and efficiency of each gas turbine follow from the combustor energy balance:

$$f = m_f/m_a = (c_{pg}T_3 - c_{pa}T_2)/(LHV - c_{pg}T_3), \quad N_{GT} = m_a(w_T - w_C), \quad \eta_{GT} = N_{GT}/(m_f\,LHV) \tag{4}$$

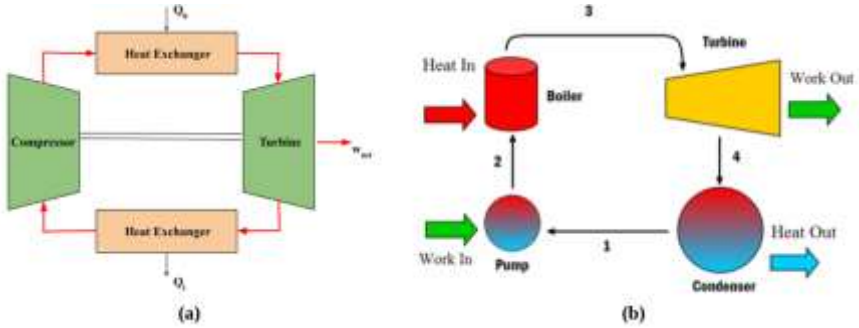


Fig. 3. (a) Brayton-cycle representation of the gas turbine; (b) Rankine-cycle representation of the steam turbine.

2.3 Steam-turbine (Rankine) model

The bottoming cycle (Fig. 3b) was modelled with the steam-turbine inlet at 83 bar and 560 °C and a single extraction to the deaerator at 4.8 bar and 200 °C. With state 1 at the turbine inlet, e at the extraction, 2 at the turbine exhaust, 3 at the condensate pump inlet, 4 at the feed-pump outlet and 5 at the deaerator outlet, the extraction fraction Y and the steam-turbine work are:

$$Y = (h_5 - h_4)/(h_e - h_4), \quad W_{ST} = m_s[(h_1 - h_e) + (1 - Y)(h_e - h_2)] \tag{5}$$

$$Q_{cond} = m_s(1 - Y)(h_2 - h_3), \quad N_{ST} = \eta_m W_{ST} - W_{FP}/\eta_p, \quad \eta_{ST} = N_{ST}/(Q_{sol,field} + Q_{ex}) \tag{6}$$

where Q_{ex} is the heat recovered from the gas-turbine exhaust in the HRSGs. The steam mass flow through the turbine was obtained from Stodola's ellipse law, and the condenser pressure from the cooling-water inlet temperature and a fixed terminal temperature difference. The solar field supplies $Q_{sol,field}$ to the SSG; its model, with optical efficiency, incidence-angle modifier and PTR70 receiver heat loss, is that of the companion study on the same site and follows standard trough practice [1].

2.4 Integrated plant, exergy and validation

The ISCC net power and the overall thermal, solar-to-electric and exergy efficiencies were defined as:

$$N_{ISCC} = 2N_{GT} + N_{ST} - N_{aux}, \quad \eta_{ISCC} = N_{ISCC}/(m_f\,LHV + Q_{sol,field}) \tag{7}$$

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ISSN 2229-3388

$$N_{sol,el} = N_{ISCC} - \eta_{CC}\, m_f\, LHV, \quad \eta_{sol,el} = N_{sol,el}/(DNI\, A_p) \qquad (8)$$

$$Ex_u = m\, c_p[(T_{out} - T_{in}) - T_0 \ln(T_{out}/T_{in})], \quad \eta_{ex,field} = Ex_u/(\psi\, DNI\, A_p) \qquad (9)$$

$$\psi = 1 - (4/3)(T_0/T_s) + (1/3)(T_0/T_s)^4, \quad \eta_{ex,ISCC} = N_{ISCC}/(\varphi\, m_f\, LHV + \psi\, DNI\, A_p) \qquad (10)$$

Here η_{CC} is the efficiency of the plant without solar input, $\psi = 0.930$ is Petela's exergy factor for solar radiation with $T_s = 5770$ K and $T_0 = 303.15$ K [17], and $\varphi = 1.04$ is the ratio of chemical exergy to lower heating value of natural gas [18]. The model was implemented in MATLAB/Simulink (Fig. 4), with separate subsystems for the compressor, combustion chamber, turbine, economiser, evaporator, superheater, feed-water heater, deaerator and condenser. A reference model of the Hassi R'mel plant (Fig. 4a) and the present model (Fig. 4b) were run for four representative days (Day 1–Day 4) spanning high- and low-irradiance conditions. For validation, both models were compared with the hourly net output measured at Hassi R'mel [6] using the mean bias error (MBE), mean absolute error (MAE), root-mean-square error (RMSE), RMSE normalised by the measured mean (nRMSE) and mean absolute percentage error (MAPE).

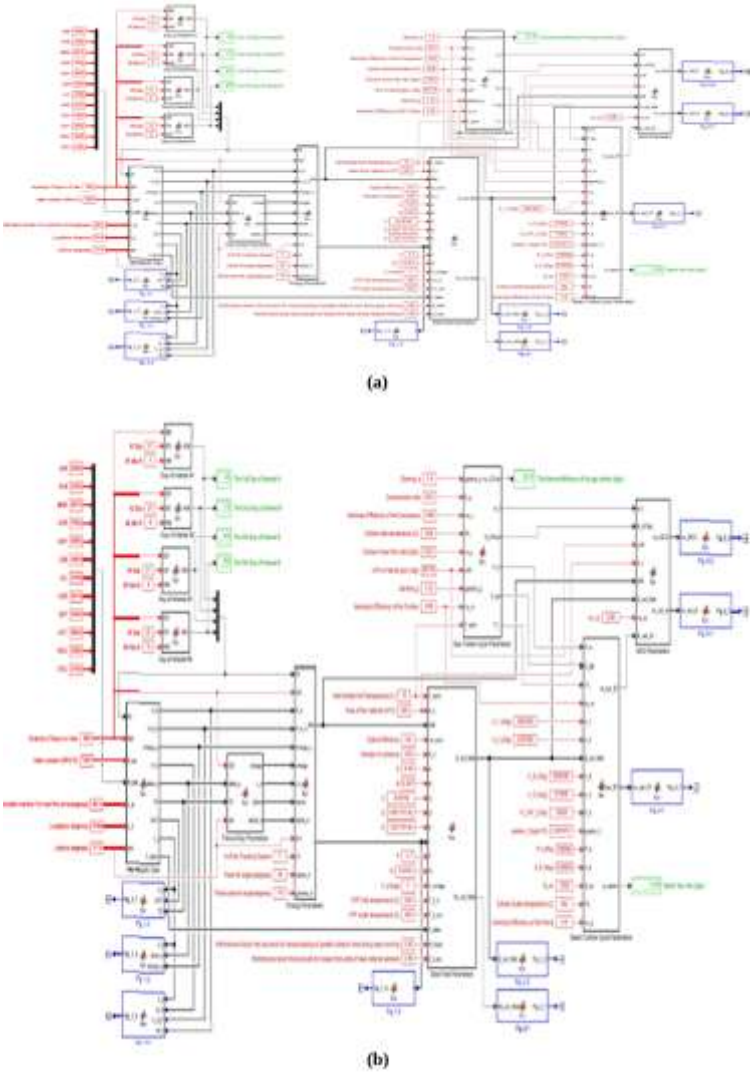


Fig. 4. MATLAB/Simulink implementation: (a) Brayton-cycle gas-turbine model of the reference plant; (b) integrated model with the Rankine-cycle steam turbine driven by Kalaburagi radiation data.

3. Results and Discussion

3.1 Solar input, steam cycle and solar-to-electric conversion

With Kalaburagi radiation, the trough field delivered between about 7 MW on a low-irradiance day and 45 MW on a clear day at solar noon, and its thermal efficiency stabilised at 38–51% during operation. This heat enters the bottoming cycle in parallel with the HRSG duty. The steam-cycle efficiency defined by Eq. (6) remained in the range of about 33–48% during solar operation. On that basis, the solar field adds roughly 2–22 MWe of incremental electricity, the upper value being equivalent to about 14% of the plant output. This is close to the net solar electricity ratio of about 15% reported for Hassi R'mel [6] and to the 20–30 MW solar contribution estimated by Derbal-Mokrane et al. [7]. The steam turbine thus operates well below its 80 MWe rating at night and is loaded progressively by solar steam during the day. This part-load behaviour is characteristic of ISCC plants [1,11], and it caps the attainable solar share unless the steam turbine is oversized.

3.2 Overall thermal efficiency of the ISCC

Fig. 5 compares the hourly overall thermal efficiency, Eq. (7), predicted by the reference and present models. Both curves show a flat night-time plateau, which corresponds to combined-cycle operation, and a trough during solar hours. The present model gives a combined-cycle plateau of about 67%, against about 60.5% for the reference model. The difference is attributed mainly to the temperature-dependent specific heats (Eqs. 2–3) and to the explicit deaerator and feed-water-heater balances of the present formulation. The present value lies at the upper end of the 57.5–67% range reported for Hassi R'mel [6], so it should be regarded as an upper bound: auxiliary and part-load losses are only partly represented.

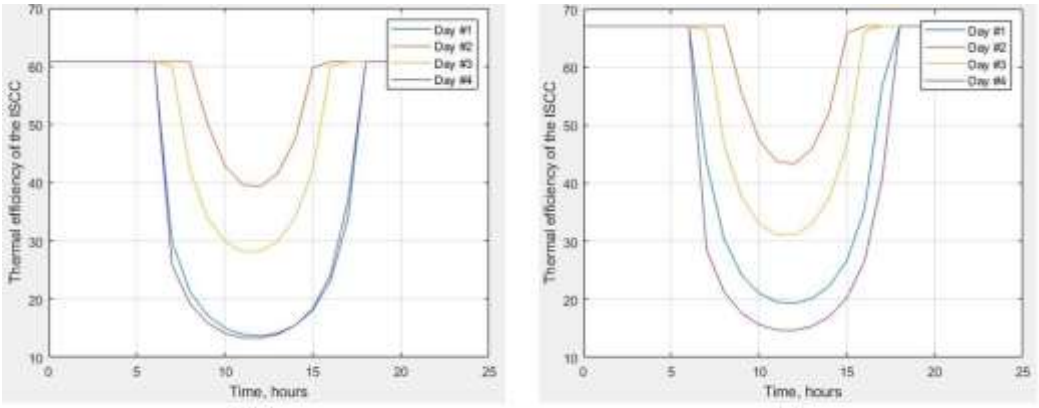


Fig. 5. Hourly overall thermal efficiency of the ISCC on four representative days: (a) reference model; (b) present model driven by Kalaburagi radiation.

During solar hours the efficiency falls, to minima of about 43%, 31%, 19% and 15% on Days 2, 3, 1 and 4 in the present model and to 39%, 28% and about 14% in the reference model. The depth of the trough scales with the solar input, being shallowest on the low-irradiance Day 2 and deepest on the clear Days 1 and 4. This is a direct consequence of the definition in Eq. (7), which charges solar heat to the denominator at full value while converting it only at steam-cycle efficiency. A fuel-based efficiency, $\eta = N_{ISCC}/(m_f\,LHV)$, would instead rise during the day. The two indicators should therefore be reported together, as recommended in earlier ISCC assessments [4,12]. The magnitude of the midday dip on the clear days is larger than a simple heat-input ratio suggests, and it should be interpreted with care until the hourly fuel flow of the model is reported alongside it.

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Vol 2024, No.1
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ISSN 2229-3388

3.3 Validation against Hassi R'mel operating data

The hourly net output of both models is compared with the measured Hassi R'mel data in Fig. 6, the hourly deviations are plotted in Fig. 7, and the error statistics are summarised in Table 2. The measured output is nearly constant at 151–152.5 MWh h^{-1}, indicating that the plant was dispatched at a steady load with solar heat used largely to offset fuel. The reference model under-predicts by about 7.5% at night and over-predicts by up to 9.4% around 13:00. Its daily energy is 2.44% below the measured 3645.3 MWh.

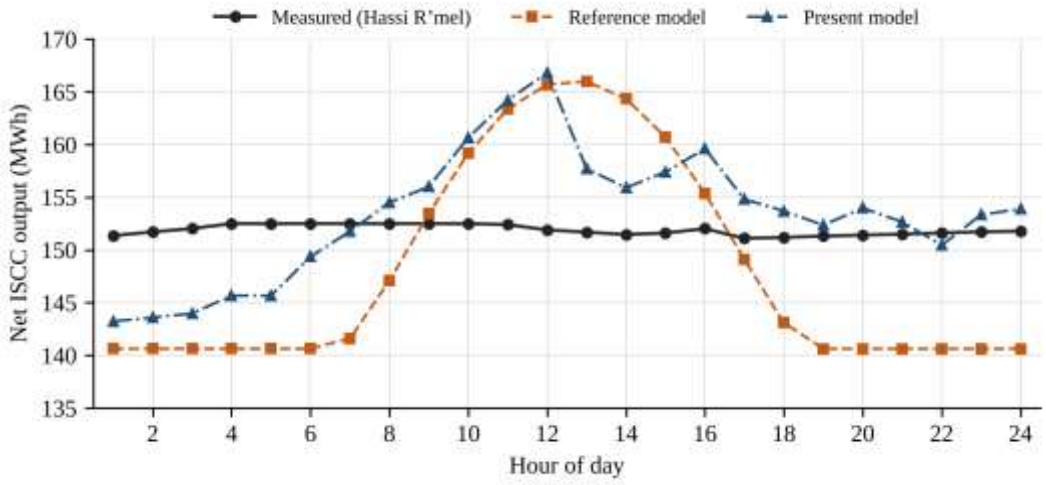


Fig. 6. Hourly net ISCC output: measured data from Hassi R'mel compared with the reference and present models.

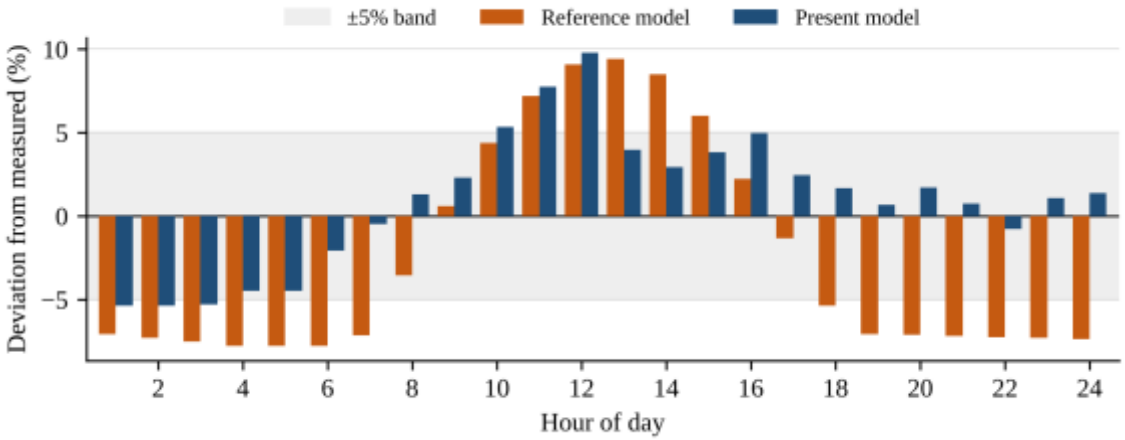


Fig. 7. Hourly percentage deviation of the reference and present models from the measured net output; the shaded band marks ±5%.

Table 2. Validation statistics of the reference and present models against measured hourly net output (24 h).

Indicator	Reference model	Present model
Daily energy (MWh) / deviation (%)	3556.5 / −2.44	3681.2 / +0.98
MBE (MWh h^{-1})	−3.70	+1.49
MAE (MWh h^{-1})	9.70	5.08
RMSE (MWh h^{-1}) / nRMSE (%)	10.30 / 6.78	6.22 / 4.10
MAPE, 24 h (%)	6.39	3.34
MAPE, hours 1–6 / 7–18 / 19–24 (%)	7.54 / 5.40 / 7.21	4.51 / 3.90 / 1.06
Maximum hourly deviation (%)	9.43	9.81

The present model halves the overall error: MAPE falls from 6.39% to 3.34%, RMSE from 10.3 to 6.2 MWh h^{-1}, and the daily energy is reproduced within +0.98%. The improvement is largest in the evening (MAPE 1.06% against 7.21%), when the reference model drops abruptly to its night-time value while the present model retains the thermal inertia of the HRSG and steam cycle. Both models over-predict near solar noon by up to about 10%. This is expected, because the simulations add solar steam to the output whereas the real plant, held at constant load, converts it into fuel savings. The validation errors are comparable with those reported for TRNSYS and FORTRAN models of the same plant [7,8], and an nRMSE of 4.1% is within the range normally accepted for system-level power plant models.

3.4 Exergy assessment

For the HTF heated from 290 to 393 °C with T_0 = 30 °C, the bracket in Eq. (9) shows that only 50.6% of the absorbed heat is exergy. Dividing by ψ = 0.930 gives $\eta_{ex,field}$ = 0.544 η_{th}. The resulting solar-field exergy efficiencies are listed in Table 3. They range from 21% on the low-irradiance days to 28% on clear days, which confirms that the field, rather than the power block, is the principal site of exergy destruction. This agrees with the findings of Reddy et al. [10] and Baghernejad and Yaghoubi [9]. In combined-cycle mode, Eq. (10) gives plant exergy efficiencies of about 58% for the reference model and 64% for the present model. These values exceed the 46.8% reported for Yazd [9] mainly because they refer to night-time operation without solar input, and they again represent an upper bound.

Table 3. Exergy efficiency of the solar field and of the plant in combined-cycle mode (T_0 = 303.15 K).

Case	η_{th} (%)	η_{ex} (%)	Ratio η_{ex}/η_{th}
Solar field, Day 1 / Day 4 (clear)	50.5–51.0	27.5–27.7	0.544
Solar field, Day 3	40.5	22.0	0.544
Solar field, Day 2 (low irradiance)	38.5	20.9	0.544
Plant, combined-cycle mode – reference	60.5	58.2	0.962
Plant, combined-cycle mode – present	67.0	64.4	0.962

The exergy results point to a clear priority for tropical ISCC plants. Raising the field outlet temperature would raise the exergy factor of the delivered heat, since every 10 K gained at the outlet improves the bracket in Eq. (9). So would reducing the temperature-driven receiver loss. Either change would yield more second-law benefit than further refinement of the power block. Higher outlet temperatures, however, require HTFs or molten salts that tolerate them [1,19].

3.5 Novelty, implications and limitations

The novelty of this work lies in a validated Brayton–Rankine ISCC model driven by measured radiation for a tropical Indian site, combined with hour-by-hour error statistics and a second-law assessment that separates field and power-block performance. For Northern Karnataka, the results suggest that an ISCC of the Hassi R'mel type could add up to about 22 MWe of solar electricity on clear days without changing its combined-cycle core. Designers should, however, expect the steam turbine to run at part load during the monsoon months. The main limitations are the use of four representative days rather than annual simulation, a validation dataset from a single Algerian plant on one day, and an exergy evaluation limited to the field and to combined-cycle mode. Future work should extend the model to annual hourly simulation, component-wise exergy destruction, fuel-saving dispatch and thermal storage.

4. Conclusions

A parabolic trough-assisted Brayton–Rankine ISCC was modelled for Kalaburagi, Karnataka, and validated against Hassi R'mel operating data. The principal conclusions are:

- With an annual mean DNI of 5.28 kWh m^{-2} day^{-1}, the 183,120 m^2 trough field delivered 7–45 MW of heat at 38–51% thermal efficiency on the representative days.
- At a steam-cycle efficiency of about 33–48%, the solar heat added roughly 2–22 MWe, up to about 14% of plant output, consistent with the 15% solar share reported for Hassi R'mel.
- The combined-cycle-mode thermal efficiency was about 67% for the present model and 60.5% for the reference model; the solar-hour efficiency defined on total heat input fell to 15–43%, depending on the solar input.
- Against measured hourly output, the present model reproduced daily energy within +0.98% (3681.2 against 3645.3 MWh), with MAPE 3.34%, RMSE 6.22 MWh h^{-1} and nRMSE 4.10%.

- The reference model gave MAPE 6.39%, RMSE 10.30 MWh h^{-1} and a −2.44% daily deviation; the largest improvement was in the evening hours (MAPE 1.06% against 7.21%).
- Only 50.6% of the heat absorbed by the HTF (290–393 °C) is exergy, giving solar-field exergy efficiencies of 20.9–27.7%.
- Plant exergy efficiency in combined-cycle mode was about 58% (reference) and 64% (present), confirming that the solar field is the dominant source of exergy destruction in the hybrid plant.

Acknowledgement:

Funding Information:

This research was no support from any funding agencies.

Conflict of Interest:

Authors here by declared that, there is no conflict of interest.

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