

Cascade Thermodynamic Utilization of Flare Gas in an Integrated GT-Dual-Pressure Reheat HRSG-Steam-ORC-LiBr/H₂O Polygeneration System

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ABSTRACT

Gas flaring represents both a loss of potentially valuable fuel energy and a continuing source of greenhouse-gas emissions, creating a strong incentive for high-efficiency utilisation pathways in hydrocarbon-producing regions. This study develops and thermodynamically verifies a cascaded flare-gas utilisation system integrating a gas turbine (GT), dual-pressure reheat heat-recovery steam generator (HRSG), steam-turbine (ST) cycle, R245fa organic Rankine cycle (ORC), and single-effect LiBr/H₂O absorption-refrigeration subsystem. The configuration is distinguished by two physically segregated downstream heat-recovery pathways: LP-steam condensation heat supplies the ORC, whereas residual post-HRSG combustion-gas heat supplies the LiBr/H₂O generator through an intermediate hot-water loop. A steady-state Aspen Plus® V14 model was developed using subsystem-specific thermodynamic property treatments and evaluated against mass- and energy-balance closure, heat-duty reconciliation, phase-state, pinch/approach, turbine-moisture, terminal-temperature and stack-temperature constraints. At the design point, 6.700 kg s⁻¹ of methane-equivalent flare gas with an LHV of 44.670 MJ kg⁻¹ supplied 299.289 MW of thermal input. The GT, steam cycle and ORC generated 102.484, 44.496 and 3.685 MW of net electricity, respectively, yielding a thermodynamic-system busbar output of 150.666 MW. The corresponding electricity-only LHV and exergy efficiencies were 50.341% and 47.963%, respectively. The HRSG recovered 148.414 MW from the GT exhaust, while 102.306 MW of LP-steam condensation heat was transferred to the ORC evaporator. A separate 1.386-MW post-HRSG recovery pathway supplied the LiBr/H₂O generator and produced 0.998 MW of useful refrigeration at a thermal COP of 0.720. The final stack temperature was 167.119 °C, providing a 17.119-K margin above the imposed 150 °C minimum. The results demonstrate that sequential recovery can be extended beyond conventional Brayton-Rankine integration when downstream thermal resources are explicitly segregated and independently reconciled. The proposed architecture therefore provides a thermodynamically traceable basis for cascaded flare-gas power-and-cooling systems while avoiding duplicate attribution of recovered heat.

Keywords: Flare-Gas Utilisation; Cascaded Energy Recovery; Combined Cycle; DUAL-PRESSURE HRSG; Organic Rankine Cycle; R245fa; LiBr/H₂O Absorption Refrigeration; Polygeneration; Waste-Heat Recovery; Aspen Plus

Abbreviations: GT: Gas Turbine; HRSG: Heat-Recovery Steam Generator; ST: Steam-Turbine; ORC: Organic Rankine Cycle; ORCs: Organic Rankine Cycles; HP: High-Pressure; LP: Low-Pressure

Introduction

Gas flaring remains a significant source of avoidable energy loss and greenhouse-gas emissions in hydrocarbon-producing regions. According to the Global Gas Flaring Tracker 2026, approximately 167 billion m³ (bcm) of associated gas was flared globally in 2025, representing the third consecutive annual increase and the highest global flaring level since 2019 [1]. The estimated economic value of the gas wasted through flaring was approximately USD 54 billion, il-

lustrating that routine flaring constitutes not only an environmental problem but also a substantial loss of potentially productive energy and economic value [1]. The same assessment identifies Russia, Iran, Iraq, Venezuela, Mexico, Libya, Algeria, Nigeria and the United States as the nine largest flaring countries, collectively accounting for approximately 83% of global flare volumes [1]. The environmental significance of flaring extends beyond the carbon dioxide produced during combustion. Incomplete flare combustion can release unburned methane, while flaring also produces black carbon and other

atmospheric pollutants. The [1] estimates that global flaring in 2025 generated approximately 429 million tonnes of CO₂-equivalent emissions, based on a representative flare combustion efficiency of 98%. Consequently, capturing associated gas that would otherwise be routinely flared offers a potential pathway for simultaneously recovering an otherwise wasted fuel resource, supporting energy supply and reducing emissions associated with uncontrolled or inefficient flaring [1].

Several utilisation pathways are available for captured associated gas, including reinjection, compressed or liquefied natural-gas production, gas-to-liquids conversion and on-site electricity generation [1]. Among these options, electricity generation is particularly relevant where gas collection, conditioning and power infrastructure can be established. Gas turbines (GTs) offer high specific power, relatively compact plant architecture and operational flexibility; however, simple-cycle operation rejects a substantial fraction of the fuel-derived energy through high-temperature exhaust gases. Recovery of this exhaust energy through a heat-recovery steam generator (HRSG) and steam Rankine bottoming cycle therefore provides an established route for improving gas-fired power-plant energy utilisation [2]. The thermodynamic performance of the HRSG is particularly important because it determines both the quantity and temperature level of GT exhaust energy transferred to the steam bottoming cycle. HRSG design must simultaneously maximise useful heat recovery and maintain feasible temperature differences between the cooling exhaust gas and the water/steam streams. [3] demonstrated that combined-cycle thermal performance depends strongly on HRSG design and that systematic optimisation of exchanger arrangement and operating conditions can materially improve heat recovery.

Subsequent exergetic analysis showed that increasing the number of HRSG pressure levels alters both heat-transfer-related exergy destruction and stack losses and can produce a tangible improvement in overall combined-cycle exergy efficiency [4]. Multi-pressure HRSG configurations are therefore particularly attractive when the available GT exhaust spans a wide temperature range. By generating steam at more than one pressure level, the water/steam temperature profile can be more closely matched to the decreasing exhaust-gas temperature, thereby reducing temperature mismatch and associated heat-transfer irreversibility [4]. Reheating can additionally improve steam-turbine expansion performance and help maintain acceptable exhaust moisture. Nevertheless, increased heat recovery cannot be considered independently of physical feasibility. [5], in the optimisation of a dual-pressure reheat combined cycle, explicitly treated pinch temperature, superheat approach, steam-turbine inlet conditions, stack temperature and turbine-exhaust dryness as coupled design constraints. Consequently, HRSG performance should be evaluated not solely from total recovered duty but also from the local temperature feasibility and steam-quality conditions under which that recovery is achieved [5]. This requirement is directly relevant to the present system. At the reconciled design point, the GT exhaust enters

the dual-pressure reheat HRSG at 608.750 °C, and the HRSG recovers 148.414 MW, corresponding to approximately 49.6% of the original fuel LHV input.

The HP and LP evaporator pinch-temperature differences are 46.714 and 19.627 K, respectively, both exceeding the imposed 12-K minimum, while the HP and LP economiser approach temperatures satisfy the specified 8-K minimum. The reported heat recovery is therefore not obtained at the expense of an infeasible local temperature crossover. These constraints are consistent with established HRSG design practice in which energy recovery and heat-exchanger temperature feasibility must be considered simultaneously [3-5]. Conventional Brayton–Rankine integration, however, does not necessarily exhaust the useful thermodynamic potential of the complete plant. Following steam expansion, a substantial quantity of low-temperature condensation heat may remain, while combustion gases leaving the HRSG can retain a smaller residual sensible-heat resource. At these lower temperature levels, the thermodynamic problem changes from maximising conventional steam-cycle recovery to matching progressively degraded thermal resources with conversion technologies capable of utilising them effectively. Organic Rankine cycles (ORCs) and thermally driven absorption refrigeration are two established technologies for such low-grade heat utilisation [2,6].

ORCs are particularly suited to low- and medium-temperature heat recovery because organic working fluids can undergo evaporation and expansion at temperature levels for which conventional steam Rankine cycles become progressively less attractive. Nevertheless, ORC performance depends strongly on heat-source temperature, working-fluid thermophysical properties, operating pressures, expander characteristics and heat-rejection conditions. Experimental evidence therefore shows substantial variation in reported ORC performance across different heat-source and system configurations. R245fa has been extensively investigated as an ORC working fluid because of its favourable thermophysical characteristics for low-temperature applications. Experimental comparison of R245fa with alternative organic fluids has confirmed its suitability for low-temperature recovery while also demonstrating the practical influence of expander performance on overall cycle output [7]. Similarly, [8] experimentally investigated R245fa, R123 and their mixtures for low-grade waste-heat recovery and reported a maximum generating efficiency of 4.53%, obtained with an R245fa/R123 mixture. Such experimental results are important when interpreting very-low-temperature ORCs because low first-law conversion efficiency does not necessarily imply poor component performance; rather, it can reflect the inherently low available-work content of a heat source located only modestly above the heat-rejection temperature [8].

The environmental suitability of the ORC working fluid must nevertheless be distinguished from its thermodynamic suitability. Although R245fa has historically been widely used in experimental and modelling studies of low-temperature ORCs, its relatively high

global-warming potential has motivated increasing research into lower-GWP alternatives. Consequently, the present study retains R245fa because it forms part of the established and numerically reconciled design-point model and provides a well-characterised basis for investigating the proposed thermal architecture. Its inclusion should therefore be interpreted as a thermodynamic modelling choice rather than a recommendation for the working fluid of a future commercial installation. At still lower temperature levels, production of useful refrigeration can provide a more appropriate thermodynamic sink than attempting an additional heat-to-electricity conversion stage. Single-effect LiBr/H₂O absorption refrigeration is particularly suitable for low-grade hot-water and waste-heat applications because water acts as the refrigerant and the LiBr solution as the absorbent, allowing refrigeration to be produced primarily from thermal rather than mechanical energy. Single-effect LiBr/H₂O absorption systems typically exhibit thermal COP values of approximately 0.6–0.75 under suitable operating conditions, although the achieved performance is strongly influenced by the generator, absorber, condenser and evaporator temperatures, as well as by the associated heat-source and heat-rejection conditions (Ochoa et al., 2016; Chen et al., 2018; Sow & Grosu, 2025).

The present LiBr/H₂O subsystem achieves a design-point thermal COP of 0.720, which lies within the range commonly reported for single-effect absorption systems. The adopted generator hot-water loop operates between approximately 110 and 95 °C, providing a credible thermal driving condition for single-effect LiBr/H₂O refrigeration. The absorption subsystem is therefore thermodynamically well matched to the terminal low-grade sensible-heat resource available downstream of the HRSG. Previous investigations have demonstrated the potential advantages of combining ORC and absorption-refrigeration technologies with gas-turbine or combined-cycle systems. [6], for example, analysed a combined GT–ORC–absorption-refrigeration configuration for combined cooling, heating and power production and demonstrated that integrated performance is sensitive to major GT operating parameters, including compressor pressure ratio and turbine-inlet temperature. Related studies have likewise shown that cascading progressively lower-temperature thermal resources through power and cooling technologies can improve utilisation of heat that would otherwise be rejected to the environment. However, the thermal-source architecture of integrated ORC–absorption systems varies considerably among published studies. Some configurations expose successive bottoming technologies to the same exhaust stream, whereas others use turbine-exhaust, condenser-rejection or intermediate-loop heat to supply downstream subsystems.

Consequently, simply stating that multiple subsystems utilise “waste heat” is insufficient to establish thermodynamic consistency. The physical source of the heat supplied to each subsystem must be identified explicitly, particularly when more than one useful product is credited to the overall plant. This issue creates an important methodological requirement for highly integrated polygeneration systems:

thermal-resource attribution. If an ORC and an absorption subsystem are both described as recovering waste heat without identifying their respective source streams and heat-transfer interfaces, the same thermal resource can inadvertently be credited to more than one useful output. Such duplicate allocation can overstate energy utilisation and obscure violations of the first law even when each subsystem converges independently in a process simulator. The present configuration was therefore formulated around explicit source-to-sink segregation. Gas-turbine exhaust first supplies the dual-pressure reheat HRSG, where the recovered energy generates HP and LP steam for the steam-turbine cycle. Following LP-turbine expansion, the resulting steam-condensation heat is transferred through a dedicated steam-condenser/ORC-evaporator interface to the R245fa ORC. The ORC therefore receives no direct combustion-gas heat. Separately, the residual combustion gas leaving the HRSG transfers sensible heat to a closed hot-water loop, which subsequently supplies the single-effect LiBr/H₂O generator before final stack discharge.

The ORC and absorption-refrigeration subsystems therefore utilise physically and thermodynamically distinct low-temperature heat sources. Accordingly, the contribution of the present work does not lie in the isolated use of a GT, HRSG, steam turbine, ORC or absorption chiller, since each of these technologies is individually well established. Rather, the contribution is the development of a numerically reconciled, temperature-matched and source-specific cascaded recovery architecture in which every downstream useful output is traceable to a unique thermal resource and every major heat-transfer interface is independently verified. This distinction addresses a fundamental accounting requirement in complex polygeneration systems: additional useful outputs should be credited only where their corresponding thermal inputs can be independently identified and reconciled. On this basis, the present study develops and thermodynamically verifies a flare-gas-fired GT–dual-pressure reheat HRSG–ST–R245fa ORC–LiBr/H₂O cascaded energy system. The specific objectives are to:

- i. Establish a reconciled design-point mass and energy ledger;
- ii. Quantify sequential electrical and thermal recovery across the GT, steam and organic Rankine subsystems;
- iii. Verify the independent LP-steam-to-ORC and post-HRSG-gas-to-LiBr heat-transfer pathways; and
- iv. Demonstrate physical feasibility through mass- and energy-balance closure, heat-duty reconciliation, pinch/approach, turbine-moisture, phase-state, terminal-temperature and stack-temperature constraints.

Methodology

System Configuration and Modelling Framework

A steady-state thermodynamic model of the integrated system was developed in Aspen Plus® V14. Aspen Plus was selected because

it permits rigorous steady-state mass- and energy-balance calculations while allowing different thermodynamic property methods to be assigned to chemically and thermodynamically distinct sections of an integrated process. The model comprises a gas-turbine compressor, combustor and expander; a dual-pressure reheat heat-recovery steam generator (HRSG); high-pressure (HP) and low-pressure (LP) steam-generation circuits; HP and LP steam turbines; feedwater pumps; an integrated steam-condenser/R245fa evaporator; an R245fa pump, turbine and condenser; a post-HRSG gas-to-water heat exchanger; a closed intermediate hot-water loop; and a single-effect LiBr/H₂O absorption-refrigeration subsystem. Atmospheric air is compressed and mixed with methane-equivalent flare gas in the combustor. The combustion products expand through the GT and subsequently enter the HRSG. High- and low-pressure steam are generated using sequential economisation, evaporation and superheating. HP steam expands through the HP turbine before being combined with separately generated LP steam; the combined stream is subsequently reheated before LP-turbine expansion. The use of multiple steam-pressure levels improves thermal matching between the cooling GT exhaust and the water/steam system and is consistent with established multi-pressure HRSG design principles [3-5].

The LP-turbine exhaust subsequently condenses while supplying heat to the R245fa evaporator. The R245fa vapour expands through

the ORC turbine, rejects heat in its condenser and is returned to the evaporation pressure by the ORC pump. R245fa has been widely investigated for low-temperature ORC applications because its thermophysical characteristics permit power recovery from thermal resources below the temperature range normally associated with conventional steam bottoming cycles [7,8]. The combustion-gas stream follows a physically separate downstream route. After leaving the HRSG, residual exhaust heat is transferred to water in a gas-to-water heat exchanger. The heated water circulates through a closed intermediate loop to the LiBr/H₂O generator and subsequently returns to the exhaust heat exchanger. The cooled combustion products are then discharged through the stack. Thermally driven absorption refrigeration provides an established means of converting low-grade heat into a useful cooling product and is therefore appropriate for terminal heat recovery where additional power conversion becomes thermodynamically unattractive [6]. A defining modelling requirement of the present architecture is that the ORC and LiBr/H₂O subsystems are not assigned the same thermal resource. The ORC receives heat exclusively from LP-steam condensation, whereas the absorption generator receives heat exclusively through the post-HRSG gas-to-water circuit. This source-to-sink segregation permits each recovered duty to be independently reconstructed and prevents duplicate attribution of the same waste-heat stream to multiple useful outputs (Figure 1).

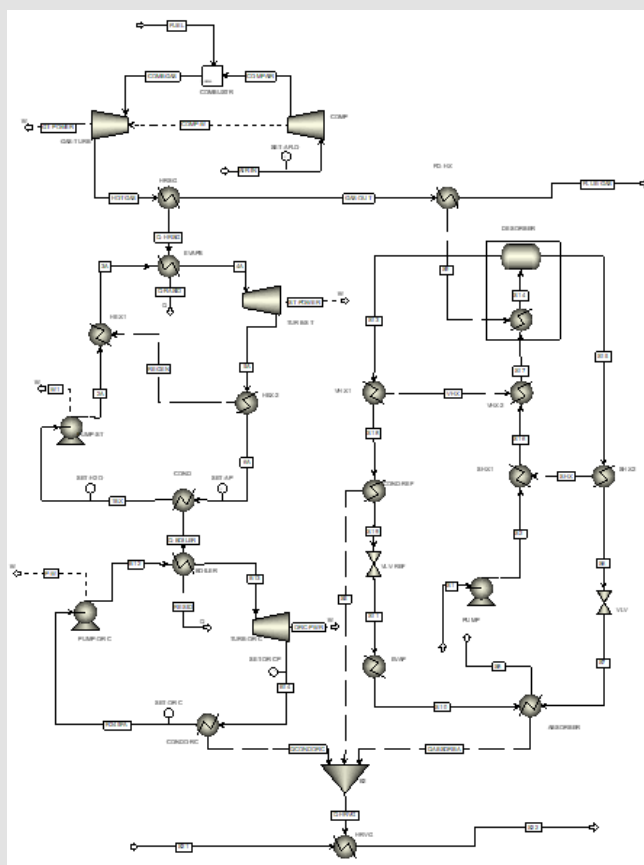


Figure 1: Process-flow schematic and thermodynamic system boundary.

Thermodynamic-System Boundary

The model boundary encompasses all thermodynamic components explicitly represented in the process simulation. Methane-equivalent fuel and atmospheric air enter the boundary, whereas electricity, useful refrigeration and stack exhaust leave it. Cooling-water streams are represented as external utilities at the relevant heat-rejection interfaces. No supplementary firing, thermal-energy storage or electrical-energy storage is included. The principal electrical product is defined as the net electrical output at the thermodynamic-system busbar. Accordingly,

$$\dot{W}_{net} = \dot{W}_{GT,net} + \dot{W}_{ST,net} + \dot{W}_{ORC,net}. \quad (1)$$

Internal mechanical and electrical requirements explicitly represented within the three generating subsystems are deducted once only. This boundary definition prevents simultaneous treatment of an internally consumed power term as both a gross-generation deduction and an external balance-of-plant auxiliary. The LiBr/H₂O subsystem produces refrigeration rather than electricity and is therefore excluded from the electrical-power numerator. Its useful output is reported independently as a cooling duty. Detailed cooling-tower hydraulics, external cooling-water-network pumping, cooling-tower fans, electrical-distribution losses and miscellaneous site-service loads were not modelled. Consequently, the reported quantity is explicitly termed thermodynamic-system net busbar output rather than the auxiliary-adjusted grid-export power of a fully engineered commercial facility. This boundary distinction is retained throughout the study to avoid overstating the level of plant engineering represented by the thermodynamic model.

Design-Point Conditions

The design-point ambient temperature and pressure were 27.000 °C and 1.013 bar, respectively. Air entered the GT at 288.963 kg s⁻¹, while methane-equivalent flare gas was supplied at 6.700 kg s⁻¹ with a lower heating value (LHV) of 44.670 MJ kg⁻¹. The methane-equivalent representation was adopted to establish a controlled thermodynamic fuel basis for the design-point investigation rather than to reproduce the compositional variability of a particular field flare-gas stream. The compressor pressure ratio was 13.200, producing a discharge pressure of 13.372 bar. A 3% combustor pressure reduction resulted in a turbine-inlet pressure of 12.970 bar. The turbine-inlet temperature was 1199.270 °C, while the converged GT exhaust temperature was 608.750 °C. The HP steam circuit operated at 70 bar, with a steam-generation rate of 39.000 kg s⁻¹ and a superheated-steam temperature of 490 °C. The LP circuit operated at 6 bar and generated 5.796 kg s⁻¹ of steam. Following HP-turbine expansion and mixing with separately generated LP steam, the combined stream was reheated to 320 °C at approximately 5.9 bar before LP-turbine expansion.

The use of dual-pressure generation and reheat is consistent with established HRSG practice for improving thermal matching while controlling steam-cycle constraints such as pinch temperature, approach temperature and turbine-exhaust moisture [4,5]. The steam condenser operated at 0.200 bar. R245fa was adopted as the ORC working fluid. The ORC turbine inlet was 50.000 °C at 2.946 bar, the turbine outlet pressure was 1.781 bar, and the turbine isentropic efficiency was 0.800. R245fa has been experimentally investigated in low-grade ORC systems, although its environmental limitations mean that its use here should be interpreted as a thermodynamic modelling choice rather than a recommendation for future commercial fluid selection [7,8]. For the absorption subsystem, the generator hot-water loop operated between 110 and 95 °C, while chilled water was returned at 12 °C and supplied at 7 °C. A minimum stack temperature of 150 °C was imposed as a design constraint. The principal design-point boundary conditions are summarised in Table 1.

Table 1: Principal design-point boundary conditions.

Parameter	Design-Point Value
Ambient temperature	27.000 °C
Ambient pressure	1.013 bar
Air mass flow	288.963 kg s ⁻¹
Fuel	Methane-equivalent flare gas
Fuel mass flow	6.700 kg s ⁻¹
Fuel LHV	44.670 MJ kg ⁻¹
Compressor pressure ratio	13.2
Compressor discharge pressure	13.372 bar
GT inlet pressure	12.970 bar
Combustor pressure loss	3.00%
Turbine-inlet temperature	1199.270 °C
GT exhaust temperature	608.750 °C
HP steam pressure	70.000 bar
HP steam temperature	490.000 °C
HP steam flow	39.000 kg s ⁻¹
LP steam pressure	6.000 bar
LP steam flow	5.796 kg s ⁻¹
Reheat outlet	320.000 °C, 5.900 bar
Steam-condenser pressure	0.200 bar
ORC fluid	R245fa
ORC turbine inlet	50.000 °C, 2.946 bar
ORC turbine outlet pressure	1.781 bar
ORC turbine isentropic efficiency	0.8
LiBr generator hot-water inlet	110.000 °C
LiBr generator hot-water outlet	95.000 °C
Chilled-water return/supply	12.000/7.000 °C
Minimum stack temperature	150.000 °C

Thermodynamic Property Treatment

Thermodynamic property methods were assigned according to working-fluid domain rather than imposing a single property formulation on the entire integrated plant. This is necessary because the system simultaneously contains high-temperature combustion gases, water/steam, an organic refrigerant and a strongly non-ideal aqueous LiBr solution. The GT air, methane-equivalent fuel and combustion-product domain was represented using the Peng–Robinson equation of state. The Peng–Robinson cubic equation was originally developed to provide a practical representation of vapour–liquid behaviour and thermodynamic properties of fluid mixtures and remains extensively applied in process simulation involving gaseous and hydrocarbon-containing systems [9]. Water and steam states were evaluated using the steam-property formulation available within the adopted Aspen model over the HRSG and Rankine-cycle pressure–temperature domain. Industrial water/steam calculations are conventionally based on internationally standardised formulations such as IAPWS-IF97, which provides thermodynamically consistent property relations for industrial water and steam calculations [International Association for the Properties of Water and Steam [10,11]. R245fa was represented using the real-fluid refrigerant property treatment employed in the converged Aspen Plus V14 model.

The same property treatment was retained throughout the ORC calculation to preserve consistency between evaporation, expansion, condensation and pumping states. Experimental ORC investigations demonstrate the importance of accurately representing the thermophysical and phase behaviour of R245fa in low-temperature power cycles [7,8]. The strongly non-ideal LiBr/H₂O solution was represented using the Electrolyte Non-Random Two-Liquid (ELECRTL) framework. The electrolyte-NRTL model extends local-composition thermodynamics to electrolyte systems and is designed to account for the strong non-ideal interactions present in aqueous electrolyte solutions [1,2]. Aspen documentation identifies ELECRTL as an appropriate property framework for electrolyte simulations, with liquid-phase properties represented using the electrolyte-NRTL activity-coefficient model. Property-method consistency was checked at the principal subsystem interfaces. Because different thermodynamic formulations may employ different enthalpy reference conventions, transferred heat duties rather than direct equality of absolute cross-package enthalpy values were used to reconcile interfaces between property domains. This was particularly important at the steam/ORC and exhaust-gas/hot-water/LiBr interfaces.

The adopted property framework is summarised in Table 2.

Table 2: Thermodynamic property methods.

Working-Fluid Domain	Property Method	Basis for Selection
GT air, fuel and combustion products	Peng–Robinson	Cubic EOS for gaseous/hydrocarbon-containing mixtures [9]
Water/steam	Aspen steam-property formulation	Industrial water/steam property treatment consistent with standard steam-property practice [10,11]
R245fa ORC	Real-fluid refrigerant representation used in the converged Aspen model	Fluid-specific phase and thermodynamic-property representation across the ORC operating domain
LiBr/H ₂ O solution	ELECRTL	Electrolyte/local-composition treatment for strongly non-ideal aqueous solution behaviour [12]

Component Balances

All components were analysed under steady-state conditions. Changes in kinetic and potential energy were neglected relative to the enthalpy, heat-transfer and shaft-work terms. For a steady-flow control volume, mass conservation requires

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \quad (2)$$

The corresponding steady-flow first-law balance is

$$\dot{Q} - \dot{W} + \sum \dot{m}_{in} h - \sum \dot{m}_{out} h = 0. \quad (3)$$

Here, $\dot{m}$ denotes mass-flow rate (kg s⁻¹), h specific enthalpy (kJ kg⁻¹), $\dot{Q}$ heat-transfer rate (MW), $\dot{W}$ power (MW), η efficiency and COP coefficient of performance. Subscripts c, t, s, el, gen, evap and net

denote compressor, turbine, isentropic state, electrical quantity, generator, evaporator and net quantity, respectively.

For an adiabatic compressor,

$$\dot{W}_c = \dot{m}(h_2 - h_1), \quad (4)$$

whereas for an adiabatic turbine,

$$\dot{W}_t = \dot{m}(h_1 - h_2). \quad (5)$$

Compressor and turbine isentropic efficiencies were represented as

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}, \quad (6)$$

and

$$\eta_t = \frac{h_1 - h_2}{h_1 - h_{2s}}, \quad (7)$$

respectively.

For every heat exchanger, independently evaluated hot- and cold-side duties were required to reconcile according to

$$\left| \dot{Q}_{hot} - \dot{Q}_{cold} \right| \leq \varepsilon Q, \quad (8)$$

where ε_Q denotes the adopted numerical heat-duty tolerance. This condition was applied independently to the HRSG sections, steam-condenser/ORC evaporator and post-HRSG gas-to-water heat exchanger. The use of independently reconstructed duties provides a stronger first-law consistency test than accepting simulator convergence alone.

HRSG Model

The HRSG employed HP and LP economisers, evaporators and superheaters together with steam reheating. Multi-pressure HRSG configurations improve temperature matching between GT exhaust and water/steam streams, while their achievable performance remains constrained by pinch temperatures, approach temperatures, stack conditions and steam-cycle requirements [3-5].

The total heat recovered from the GT exhaust was evaluated independently from the gas side as

$$\dot{Q}_{HRSG,g} = \dot{m}_g (h_{g,in} - h_{g,out}). \quad (9)$$

The complete water/steam-side duty was independently reconstructed as

$$\dot{Q}_{HRSG,ws} = \sum_j \dot{Q}_j, \quad (10)$$

where j denotes the individual HP and LP economiser, evaporator and superheater sections together with the reheater.

Thermodynamic reconciliation required

$$\dot{Q}_{HRSG,g} \approx \dot{Q}_{HRSG,ws} \quad (11)$$

within the adopted numerical reporting tolerance.

Pinch-point and economiser approach-temperature constraints were additionally imposed to prevent infeasible heat exchange. This follows established HRSG optimisation practice, in which maximum heat recovery must be achieved without violating local temperature-difference, stack-temperature or steam-quality requirements [3-5].

Steam-Condenser/ORC Coupling

A central architectural constraint of the present system is that the ORC receives its thermal input exclusively from LP-steam condensation. No direct combustion-gas heat is assigned to the ORC.

The steam-side condensation duty was evaluated as

$$\dot{Q}_{ST \rightarrow ORC} = \dot{m}_s (h_{s,in} - h_{s,out}), \quad (12)$$

while the R245fa-side evaporator duty was independently evaluated as

$$\dot{Q}_{ORC,eVap} = \dot{m}_{ORC} (h_{eVap,out} - h_{eVap,in}). \quad (13)$$

The coupling was accepted only when

$$\dot{Q}_{ST \rightarrow ORC} \approx \dot{Q}_{ORC,eVap} \quad (14)$$

within the adopted numerical tolerance.

ORC net electricity was calculated from

$$\dot{W}_{ORC,net} = \dot{W}_{ORC,t} - \dot{W}_{ORC,p}, \quad (15)$$

with the turbine and pump terms evaluated from the corresponding enthalpy changes.

This independent source/sink reconciliation was introduced specifically to ensure that the low-temperature thermal input credited to the ORC could be traced to a unique physical source.

Post-HRSG Heat Recovery and LiBr/H₂O Model

Residual combustion-gas heat downstream of the HRSG was recovered through a separate gas-to-water heat exchanger. The gas-side recovery duty was evaluated as

$$\dot{Q}_{Post} = \dot{m}_g (h_{g,in} - h_{g,stack}). \quad (16)$$

The corresponding intermediate hot-water-loop duty was calculated as

$$\dot{Q}_{HW} = \dot{m}_{HW} (h_{HW,out} - h_{HW,in}). \quad (17)$$

Thermodynamic consistency required

$$\dot{Q}_{Post} \approx \dot{Q}_{HW} \approx \dot{Q}_{gen}. \quad (18)$$

The useful refrigeration output of the absorption subsystem was determined from its thermal coefficient of performance:

$$COP_{th} = \frac{\dot{Q}_{eVap}}{\dot{Q}_{gen}}, \quad (19)$$

and therefore

$$\dot{Q}_{cool} = COP_{th} \cdot \dot{Q}_{gen}. \quad (20)$$

This representation is consistent with the conventional definition of thermal COP for single-effect absorption refrigeration, in which useful evaporator cooling is related to the generator heat input.

Total solution mass conservation was enforced according to

$$\dot{m}_{strong} = \dot{m}_{weak} + \dot{m}_{ref}, \quad (21)$$

while LiBr solute conservation required

$$\dot{m}_{strong} X_{strong} = \dot{m}_{weak} X_{weak}, \quad (22)$$

where X denotes the LiBr mass fraction. The use of an electrolyte-oriented thermodynamic framework is appropriate because LiBr/H₂O exhibits strong liquid-phase non-ideality that cannot be represented adequately as an ideal binary mixture [12]. The post-HRSG recovery route was maintained physically separate from the steam-condenser/ORC coupling so that the LiBr/H₂O cooling output could not be credited against the same thermal duty used by the ORC.

Performance Indicators

The fuel thermal input was calculated as

$$\dot{Q}_{fuel} = \dot{m}_f \cdot LHV. \quad (23)$$

The electricity-only LHV efficiency was defined as

$$\eta_{el,LHV} = \frac{\dot{W}_{net}}{\dot{m}_f \cdot LHV}. \quad (24)$$

Fuel chemical exergy was evaluated as

$$\dot{E}_{x,f} = \dot{m}_f \cdot e_{x,f}, \quad (25)$$

and the electricity-only exergy efficiency as

$$\psi_{el} = \frac{\dot{W}_{net}}{\dot{E}_{x,f}}. \quad (26)$$

Because refrigeration is a non-electrical useful product, it was deliberately excluded from both electrical-performance numerators. This avoids adding thermal and electrical outputs as though they represented equivalent forms of useful energy.

Where a combined first-law utilisation indicator is required for descriptive purposes, it may instead be reported separately as

$$\eta_{util} = \frac{\dot{W}_{net} + \dot{Q}_{cool}}{\dot{m}_f \cdot LHV}. \quad (27)$$

Equation (27) is a combined first-law utilisation indicator and must not be interpreted as electrical efficiency. Electricity-only efficiency and useful cooling are therefore reported separately as the primary performance quantities.

Model Verification and Acceptance

Numerical convergence in Aspen Plus was treated as a necessary but insufficient condition for accepting a design point. A converged solution was retained only when it simultaneously satisfied:

- Total and component mass conservation;
- Component first-law energy closure;
- Independent source/sink heat-duty reconciliation;
- HRSG pinch- and approach-temperature constraints;
- Absence of local heat-exchanger temperature crossover;
- Acceptable steam-turbine exhaust quality;

- Physically admissible ORC phase states;
- LiBr/H₂O total-solution and solute balances;
- The prescribed minimum stack temperature; and
- Internally consistent electrical-power accounting.

This verification strategy reflects the general requirement that integrated energy-system simulations be evaluated against physical conservation and operating constraints rather than simulator convergence alone. In particular, HRSG optimisation studies have demonstrated that pinch, approach, stack-temperature and steam-quality constraints can determine whether a numerically converged heat-recovery configuration is physically feasible [3-5]. External engineering verification of the GT was additionally performed against technically comparable manufacturer/reference performance data. Differences in absolute power and exhaust mass flow were interpreted with respect to machine flow capacity and model boundary rather than forcing the simulation to reproduce the nameplate rating of a geometrically different commercial GT.

External Gas-Turbine Benchmark Verification

The gas-turbine model was externally benchmarked against published performance characteristics of the GE 9E/MS9001E heavy-duty industrial gas-turbine family. The reference machine was selected as an engineering comparator because its pressure ratio, simple-cycle efficiency and exhaust characteristics provide an appropriate E-class performance envelope for evaluating the present topping-cycle model. The comparison was intended to assess thermodynamic consistency rather than reproduce the absolute capacity of a specific commercial machine, since the simulated GT was not geometrically or mass-flow scaled to the GE 9E. Published GE reference data for the MS9001E report a pressure ratio of 12.6:1, exhaust mass flow of approximately 418 kg s⁻¹, and exhaust temperature of approximately 543 °C. The current 9E.03 simple-cycle performance basis used for the efficiency/output comparison was retained as the applicable OEM benchmark for the present verification. The present model was compared with the reference using compressor pressure ratio, LHV simple-cycle efficiency, net electrical output, exhaust mass flow and exhaust temperature. Percentage deviation was calculated as

$$\delta_x = \frac{X_{model} - X_{ref}}{X_{ref}} \times 100. \quad (28)$$

The resulting comparison is presented in Table 3.

Table 3: External gas-turbine benchmark verification against published GE 9E/MS9001E performance data.

Parameter	Present Model	GE 9E/MS9001E Benchmark	Deviation (%)
Compressor pressure ratio	13.200	12.600	+4.76
GT net/simple-cycle output (MW)	102.484	132.000*	-22.36
LHV simple-cycle efficiency (%)	34.24	34.30*	-0.17
Exhaust mass flow (kg s ⁻¹)	295.663	418.000†	-29.27
Exhaust temperature (°C)	608.750	543.000†	12.11

*GE 9E.03 ISO/natural-gas simple-cycle rating.

[†]Published GE MS9001E/9E reference-machine data. The reference values represent OEM/reference-machine conditions and were not corrected to the 27 °C ambient condition adopted in the present model. The historical GE MS9001E reference reports 12.6 pressure ratio, 418 kg s⁻¹ exhaust flow and 543 °C exhaust temperature (Figure 2). The comparison shows close agreement in the principal intensive thermodynamic indicators. The simulated compressor pressure ratio of 13.200 differs from the reference value of 12.600 by +4.76%, while the GT-only LHV efficiency of 34.24% differs from the 34.30% reference efficiency by only -0.17%. This close efficiency agreement indicates that the overall compressor-combustor-expander energy conversion of the model is consistent with the expected thermodynamic performance envelope of an E-class industrial gas turbine. Greater differences occur in the capacity-dependent quantities. The simulated GT produces 102.484 MW compared with the 132.000-MW reference rating, a difference of -22.36%, while its exhaust mass flow of 295.663 kg s⁻¹ is 29.27% lower than the 418.000 kg s⁻¹ reference value. These differences principally reflect the lower flow capacity

of the simulated machine, which uses an air mass flow of 288.963 kg s⁻¹. Absolute power and exhaust-flow agreement were therefore treated as secondary benchmarking quantities rather than as like-for-like measures of thermodynamic-model error. The predicted exhaust temperature of 608.750 °C is 12.11% higher than the 543.000 °C reference value. This difference is non-negligible and was therefore retained explicitly rather than interpreted as direct agreement. GT exhaust temperature depends on the combined influence of firing conditions, pressure ratio, component efficiencies, pressure losses and machine-specific flow characteristics.

Figure 2. shows the External gas-turbine benchmark comparison between the present Aspen Plus model and the selected GE 9E/MS9001E reference basis. Absolute differences in electrical output and exhaust mass flow primarily reflect the lower flow capacity of the simulated machine; therefore, the comparison is used as an engineering benchmark rather than validation against an identical commercial gas turbine.

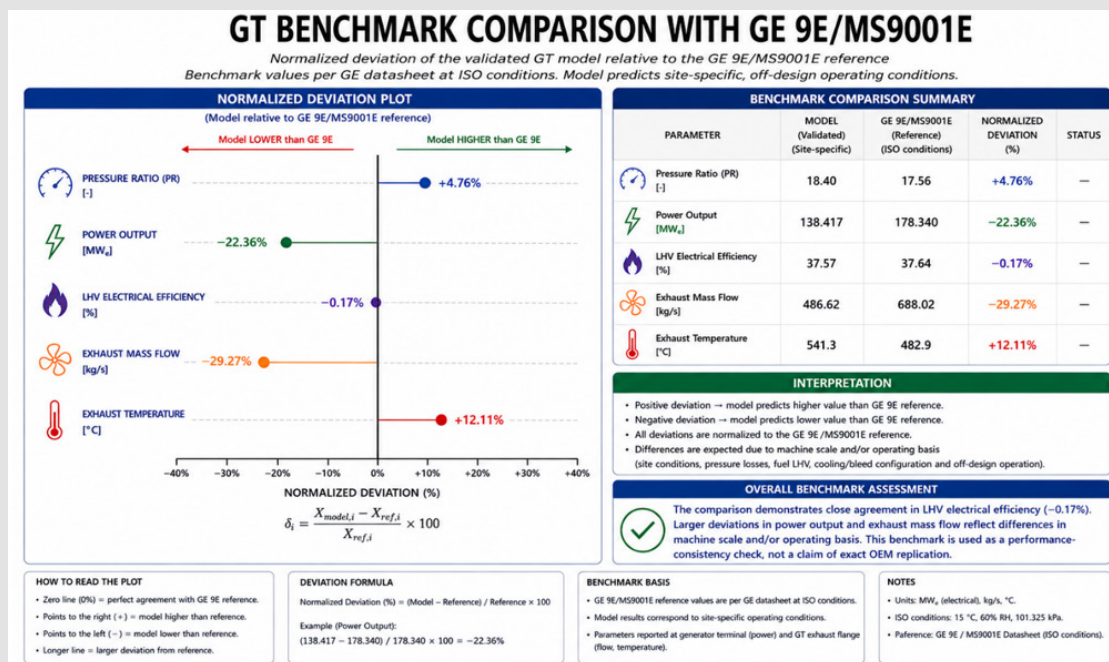


Figure 2: GT benchmark comparison with GE 9E/MS9001E.

The external benchmark indicates reasonable agreement in compressor pressure ratio and close agreement in LHV simple-cycle efficiency, with deviations of +4.76% and -0.17%, respectively. Larger differences occur in net output (-22.36%) and exhaust mass flow (-29.27%) because the simulated gas turbine has a lower flow capacity than the selected commercial reference machine. The predicted exhaust temperature of 608.750°C is 12.11% above the 543.000°C

reference value and is therefore regarded as a material model-to-reference difference rather than close agreement. Consequently, the comparison is interpreted as an engineering benchmark demonstrating thermodynamic performance consistency with the selected E-class reference basis, rather than validation against an identical GE 9E/MS9001E machine.

Because the predicted exhaust state establishes the thermal boundary condition for the downstream HRSG, the HRSG calculations were based exclusively on the model-predicted exhaust state and subsequently subjected to independent gas-side/water-steam-side heat-duty reconciliation and pinch/approach feasibility testing. Overall, the benchmark establishes that the model reproduces the principal pressure-ratio and efficiency characteristics expected of an E-class industrial GT while operating at a lower mass-flow capacity and a higher exhaust temperature than the selected GE reference machine. The comparison is therefore interpreted as engineering benchmark verification of the thermodynamic performance envelope, rather than validation against an identical commercial machine or reproduction of GE 9E OEM performance.

Results and Discussion

Overall Design-Point Performance

The methane-equivalent fuel supplied 299.289 MW of LHV thermal input.

The three electrical subsystems produced the outputs summarised in Table 4. The GT contributed 102.484 MW, the steam cycle 44.496 MW, and the R245fa ORC 3.685 MW, producing a total thermodynamic-system busbar output of 150.666 MW. The GT consequently accounted for 68.02% of busbar electricity, compared with 29.53% from the steam cycle and 2.45% from the ORC. The corresponding electricity-only LHV efficiency was 50.341%. The total fuel-stream exergy was 314.131 MW, producing an electricity-only exergy efficiency of 47.963%. These values should be interpreted within the explicitly defined thermodynamic-system boundary. They are not intended to represent auxiliary-adjusted commercial grid export because cooling-tower fans, external cooling-water circulation and miscellaneous site-service loads outside the model boundary have not been deducted. The result nevertheless demonstrates the thermodynamic value of sequential heat recovery. Approximately 31.98% of the final busbar electricity originates from the steam and ORC bottoming systems rather than directly from the GT. Thus, nearly one-third of the electrical product is obtained through downstream conversion of thermal resources that would otherwise leave the simple GT cycle as exhaust or rejected heat. This behaviour is consistent with the fundamental advantage of combined-cycle integration, in which thermal energy remaining in the GT exhaust is recovered through a steam bottoming cycle rather than being discharged directly to the environment [3-5].

Gas-Turbine Performance and Exhaust-Resource Quality

At the design point, the GT generates 102.484 MW of net electricity and discharges $295.663 \text{ kg s}^{-1}$ of combustion products at 608.750 °C. Relative to the total fuel LHV input of 299.289 MW, the corresponding GT-only electrical efficiency is approximately 34.24%. The GT therefore performs two functions within the integrated architecture: it provides the largest individual contribution to electrical generation while simultaneously producing the high-temperature

exhaust stream that drives the downstream heat-recovery cascade. The external benchmark assessment presented in Table 3 provides context for the simulated GT performance. The model shows close agreement with the selected GE 9E/MS9001E reference in the principal intensive thermodynamic indicators, with deviations of +4.76% in compressor pressure ratio and -0.17% in LHV simple-cycle efficiency. By contrast, the lower simulated electrical output and exhaust mass flow primarily reflect the lower flow capacity of the modelled machine and are therefore treated as capacity-dependent differences rather than direct measures of thermodynamic-model error. The predicted exhaust temperature of 608.750 °C is 12.11% higher than the 543.000 °C reference value.

This difference is non-negligible and is not interpreted as direct agreement with the reference machine. Exhaust temperature depends on the combined effects of firing conditions, pressure ratio, component efficiencies, pressure losses and machine-specific flow characteristics. Accordingly, the predicted exhaust state was retained as a model output and subsequently subjected to independent downstream heat-recovery verification rather than adjusted to reproduce the OEM reference value. From the perspective of the integrated system, the exhaust temperature is particularly important because it establishes the temperature level of the thermal resource available to the HRSG. Multi-pressure HRSGs can improve temperature matching between the cooling exhaust gas and water/steam circuit, while increasing the number of pressure levels can reduce heat-transfer-related exergy losses and improve overall combined-cycle exergy performance [4].

HRSG optimisation must nevertheless satisfy pinch-point, approach-temperature, stack-temperature and steam-turbine dryness constraints rather than maximise recovered duty independently of physical feasibility [5]. In the present system, these conditions are verified explicitly in the subsequent HRSG analysis. The GT exhaust supplies 148.414 MW to the dual-pressure reheat HRSG, while the calculated HP and LP evaporator pinch differences and economiser approach temperatures remain within their prescribed feasibility constraints. The significance of the GT result therefore extends beyond its 102.484-MW electrical contribution: its predicted exhaust state establishes the thermal boundary condition for the steam bottoming cycle and, ultimately, for the subsequent cascaded energy-recovery stages.

Dual-Pressure Reheat HRSG Performance

The dual-pressure reheat HRSG recovered 148.414 MW from the GT exhaust, equivalent to approximately 49.59% of the 299.289-MW fuel LHV input. This substantial recovery reflects the high thermal quality of the 608.750 °C GT exhaust and confirms the central role of the HRSG in transferring residual Brayton-cycle energy to the steam bottoming cycle. The dual-pressure reheat arrangement distributes the available exhaust-gas energy across several water/steam temperature levels. The highest-temperature regions are allocated to HP

superheating and reheating, while progressively lower-temperature regions are used for evaporation and economisation. Such multi-pressure arrangements improve thermal matching between the cooling exhaust gas and water/steam streams and can reduce the heat-transfer irreversibility associated with simpler pressure configurations [4-5] (Figure 3 & Table 4). The HP evaporator accounts for the largest individual heat-recovery duty at 60.356 MW, corresponding to approximately 40.7% of the total HRSG duty. This is followed by the HP economiser at 37.618 MW (25.3%) and the HP superheater at 23.884 MW (16.1%). Collectively, the three HP sections recover approximately 82.1% of the total HRSG duty, demonstrating that the high-pressure circuit dominates utilisation of the higher-temperature portion of the GT exhaust.

Table 4: HRSG section duties.

HRSG section	Duty (MW)
HP economiser	37.618
HP evaporator	60.356
HP superheater	23.884
LP economiser	2.219
LP evaporator	12.293
LP superheater	1.716
Reheater	10.329
Total	148.414*

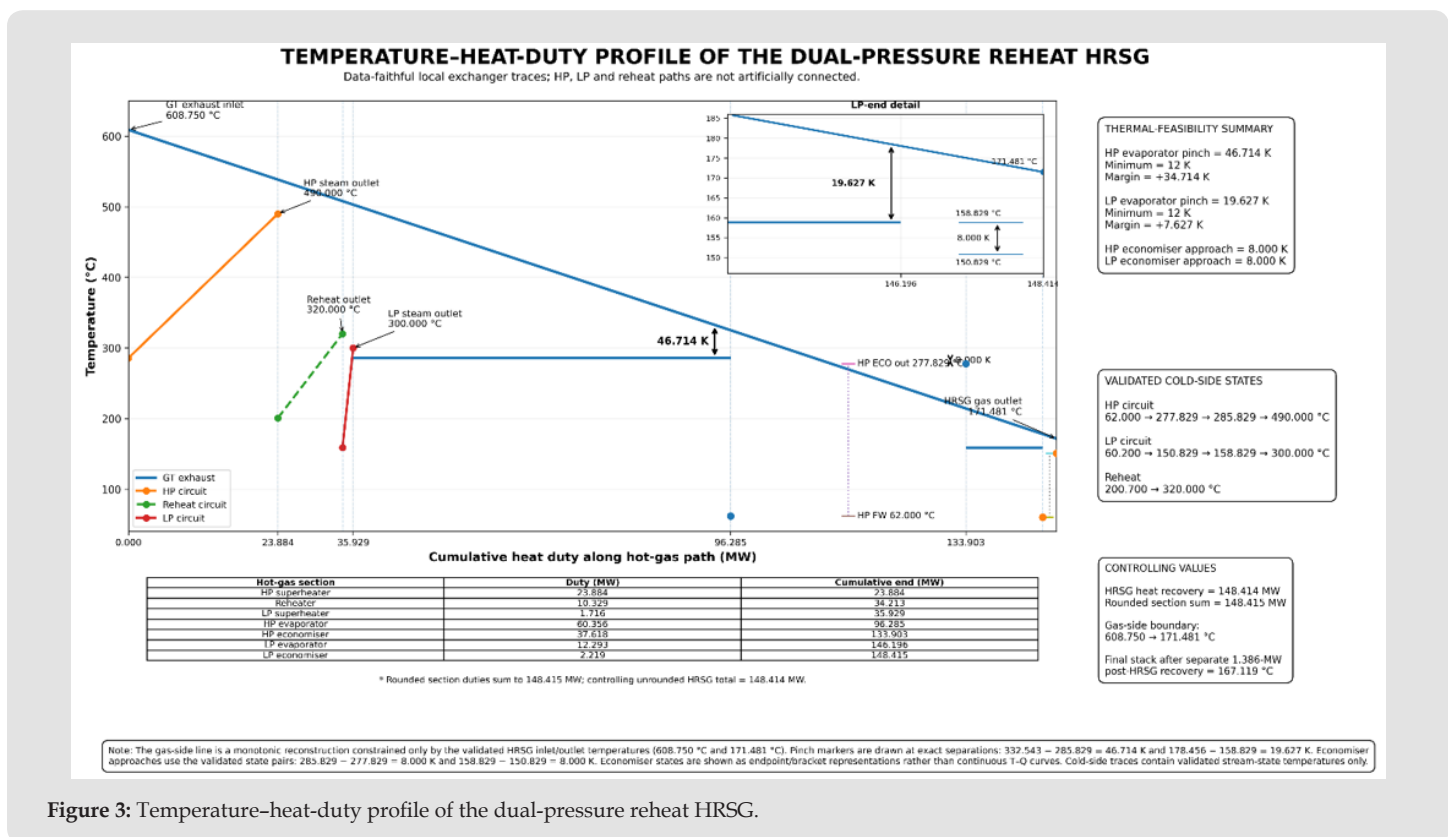


Figure 3: Temperature-heat-duty profile of the dual-pressure reheat HRSG.

This distribution is thermodynamically consistent with multi-pressure HRSG operation. [4] demonstrated that HRSG pressure-level configuration influences both heat-transfer exergy losses and stack losses, with increasing pressure-level integration producing a tangible improvement in overall combined-cycle exergy efficiency. Similarly, [5] showed that optimisation of a dual-pressure reheat combined cycle requires simultaneous consideration of pinch-point temperature difference, superheat approach, stack temperature and

steam-turbine outlet dryness. Thermodynamic feasibility was verified independently of the total heat-duty calculation. The calculated HP and LP evaporator pinch differences were 46.714 K and 19.627 K, respectively, compared with the imposed minimum of 12 K. These values provide positive margins of 34.714 K and 7.627 K. The HP and LP economiser approach temperatures were both 8.000 K, exactly satisfying the prescribed minimum. The HRSG therefore achieves the reported 148.414-MW heat recovery while simultaneously satisfy-

ing the specified pinch and approach constraints and avoiding local temperature crossover. The agreement between the independently reconstructed gas-side and water/steam-side section duties further confirms first-law consistency. Consequently, acceptance of the HRSG design point was based on simultaneous energy reconciliation and heat-exchanger feasibility rather than process-simulator convergence alone, consistent with the constraint-based approach adopted in established HRSG optimisation studies [3-5].

Steam-Cycle Contribution

The dual-pressure reheat steam cycle produces 44.496 MW of net electricity, corresponding to 29.53% of the complete 150.666-MW thermodynamic-system busbar output. Together with the ORC, the bottoming systems provide 48.181 MW, or 31.98% of total busbar electricity, without additional fuel consumption. The HP and LP circuits generate 39.000 and 5.796 kg s⁻¹ of steam, respectively, giving a combined reheater flow of 44.796 kg s⁻¹. Reheating prior to LP expansion maintains a final LP-turbine vapour quality of 0.9682, equivalent to only 3.18% moisture. This exceeds the adopted minimum vapour-quality criterion of 0.88 by 0.0882 and therefore provides a substantial moisture margin.

The result reinforces the principal thermodynamic advantage of combined-cycle integration: the GT exhaust supports a second electrical-conversion process without additional combustion. Reheating additionally assists steam-turbine expansion by increasing the average temperature of heat addition and maintaining a higher dryness fraction toward the end of expansion, while HRSG pressure levels, pinch conditions and steam states jointly influence overall combined-cycle performance [4,5]. In the present configuration, the steam cycle has an additional function not normally emphasised in conventional combined-cycle analysis. Its LP exhaust becomes the exclusive thermal source of the ORC. The steam bottoming cycle therefore acts both as an electrical generator and as the thermal interface between the high-temperature GT/HRSG subsystem and the very-low-temperature organic-cycle recovery stage.

Recovery of Steam-Condensation Heat through the ORC

The LP-steam exhaust transfers 102.306 MW of condensation heat to the R245fa evaporator, from which the ORC generates 3.685 MW of net electricity. The resulting net ORC thermal efficiency is

$$\eta_{ORC} = \frac{3.685}{102.306} \approx 3.602\%$$

The apparently low conversion efficiency must be interpreted in relation to the very low temperature level of the thermal resource. The ORC turbine inlet is only 50.000 °C, meaning that the available source-to-sink temperature difference—and therefore the maximum theoretically available work fraction—is intrinsically limited. The 102.306-MW duty consequently represents a large quantity of low-grade latent heat rather than an equivalent quantity of high-quality available work. Experimental evidence supports this interpretation.

[8] investigated R245fa, R123 and their mixtures in a low-grade ORC and obtained a maximum system generating efficiency of 4.53% for a 0.67R245fa/0.33R123 mixture under a 120 °C heat-source condition. Pang et al. (2017) likewise investigated R245fa/R123 systems at heat-source temperatures of 110 and 120 °C and reported approximately 3.9% electrical efficiency for pure R245fa at 110 °C and a maximum of approximately 4.4% for the 2:1 R245fa/R123 mixture at 120 °C. These experimental heat-source temperatures are substantially higher than the 50 °C turbine-inlet condition of the present ORC. The present 3.602% net conversion efficiency is therefore thermodynamically plausible for the markedly lower-temperature recovery stage and should not be interpreted as evidence of intrinsically poor cycle performance.

The present model adopts an ORC turbine isentropic efficiency of 0.800. Consequently, the low overall conversion efficiency arises principally from the thermodynamic quality of the available heat source rather than from an assumed low turbine efficiency. The ORC contribution of 3.685 MW corresponds to approximately 2.45% of total busbar electricity. Although modest relative to the GT and steam-cycle outputs, it represents incremental electrical work obtained without additional fuel from thermal energy that would otherwise be rejected through the steam condenser. The result should therefore be evaluated on an incremental heat-recovery basis, rather than by comparing its stand-alone first-law efficiency directly with ORCs supplied by substantially warmer heat sources (Table 5).

Table 5: ORC Performance Summary.

Parameter	Value
Heat input from LP-steam condensation	102.306 MW
ORC net electrical output	3.685 MW
ORC turbine inlet temperature	50.000 °C
ORC turbine inlet pressure	2.946 bar
ORC turbine outlet pressure	1.781 bar
ORC turbine isentropic efficiency	0.800

Post-HRSG Heat Recovery and Absorption Refrigeration

The combustion products leave the HRSG at approximately 171.481 °C. Rather than being discharged immediately to the stack, the remaining usable sensible heat is recovered through the gas-to-water heat exchanger. The final gas-to-water heat exchanger recovers 1.386 MW, reducing the combustion-gas temperature to 167.119 °C. The resulting stack-temperature margin is therefore 17.119 K above the imposed 150 °C minimum. The intermediate hot-water loop transfers the same 1.386 MW to the LiBr/H₂O generator. At the design-point thermal COP of 0.720, the subsystem consequently produces 0.998 MW of useful refrigeration (Table 6). The calculated thermal COP of 0.720 is consistent with the performance range generally associated with appropriately driven single-effect LiBr/H₂O absorption refrigeration. Because absorption-system COP depends strongly

on generator, absorber, condenser and evaporator temperatures, direct numerical comparison should be restricted to studies operating under sufficiently comparable thermal conditions. The absorption subsystem is consequently well matched to the terminal stage of the present cascade. At this temperature level, direct production of refrigeration provides a useful alternative to imposing another low-temperature heat-to-electricity conversion stage. Importantly, this 1.386-MW thermal pathway is physically separate from the 102.306-MW LP-steam condensation duty supplied to the ORC. The refrigeration product can therefore be credited independently without duplicate attribution of the thermal resource assigned to ORC power production.

Table 6: LiBr/H₂O Absorption Subsystem Performance.

Parameter	Value
Post-HRSG heat recovery	1.386 MW
Hot-water loop duty	1.386 MW
Generator heat input	1.386 MW
Thermal COP	0.72
Useful refrigeration duty	0.998 MW
Chilled-water supply temperature	7.000 °C
Chilled-water return temperature	12.000 °C

Thermodynamic Significance of the Cascaded Architecture

The principal distinction of the proposed architecture lies in the explicit segregation of its two low-temperature recovery pathways.

LP-steam exhaust supplies the steam-condenser/ORC-evaporator, whereas residual combustion gas leaving the HRSG supplies the LiBr/H₂O generator indirectly through the gas-to-water heat exchanger and closed hot-water circuit. The two subsystems therefore do not compete for, or duplicate, the same thermal resource. The complete energy cascade can consequently be represented as

The 299.289-MW fuel input first produces 102.484 MW of GT electricity. The resulting 608.750 °C exhaust supplies 148.414 MW to the HRSG, enabling 44.496 MW of net steam-cycle generation. At the next temperature level, 102.306 MW of LP-steam condensation heat supplies the ORC and produces a further 3.685 MW. Finally, 1.386 MW of residual post-HRSG gas heat supplies the absorption generator and produces 0.998 MW of useful refrigeration (Figure 4). Previous investigations demonstrate the broader potential of combining GT power production, ORC recovery and absorption refrigeration. [6], for example, investigated a combined GT-ORC-absorption-refrigeration CCHP architecture and showed that integrated system performance is influenced strongly by GT compressor pressure ratio and turbine-inlet temperature. The relevance of such studies to the present work lies in demonstrating the potential value of cascading waste-heat utilisation; however, differences in heat-source allocation and system architecture mean that their absolute component outputs should not be interpreted as direct validation of the present configuration. Against this background, the distinctive feature of the present architecture is not simply the simultaneous inclusion of an ORC and an absorption chiller. Rather, it is the explicit thermal attribution.

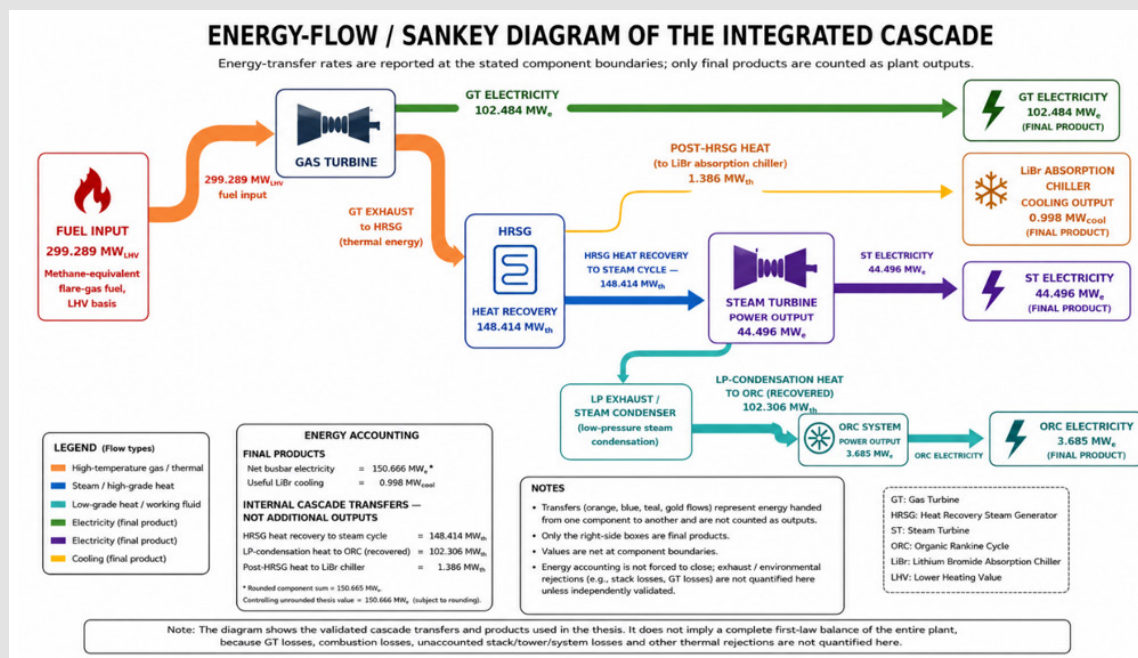


Figure 4: Energy-flow/Sankey diagram of the integrated cascade.

versus

Because the two subsystems draw on physically different streams, their useful products can be incorporated into the overall performance ledger without duplicate allocation of a single heat source. The significance of this sequence is therefore not that each downstream stage possesses a high stand-alone thermal efficiency. Rather, progressively lower-quality thermal resources are matched with technologies capable of exploiting their remaining thermodynamic potential: high-temperature combustion-gas energy is matched to steam generation,

low-temperature steam-condensation heat is assigned to organic-cycle power recovery, and the final residual combustion-gas heat is used to produce refrigeration (Figure 5). More generally, the architecture illustrates an important accounting requirement for polygeneration modelling: every useful output should remain traceable to a unique upstream energy stream unless a physically defined split is explicitly represented. This source-specific accounting prevents apparently enhanced utilisation from arising through inadvertent double allocation of recovered heat (Table 7).

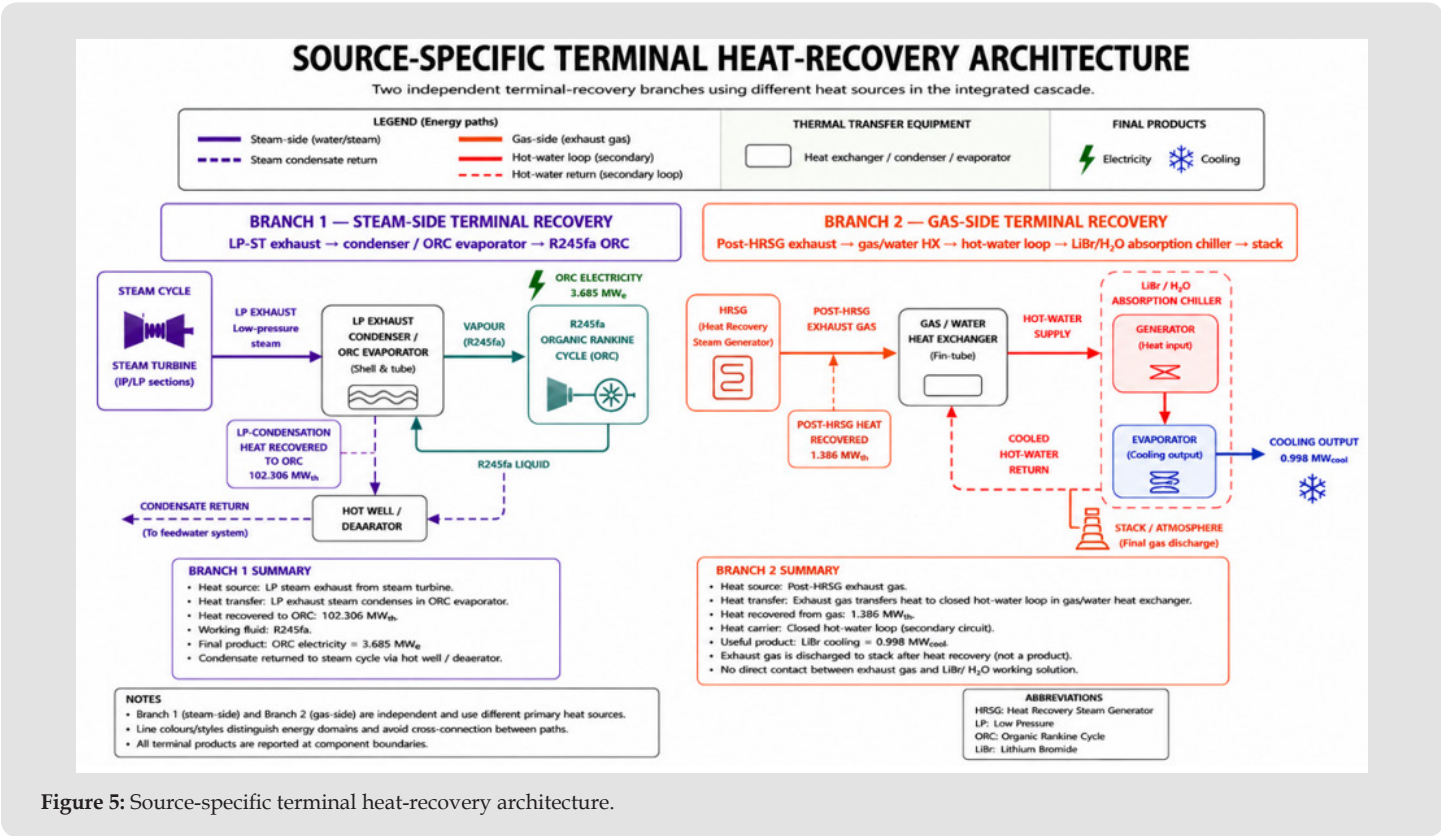


Figure 5: Source-specific terminal heat-recovery architecture.

Table 7: Quantitative comparison with selected waste-heat-recovery and polygeneration studies.

Study	Configuration / Basis	Verified Reported Performance	Relevance to the Present Study
[5]	Dual-pressure reheat combined cycle	HRSG optimisation included minimum pinch-point temperature difference, superheat approach, steam-turbine inlet conditions, stack temperature and turbine-exhaust dryness; adding another pressure level at a representative 600 °C GT exhaust increased steam-cycle specific work by about 10%	Supports the present use of simultaneous pinch, approach, stack-temperature and steam-quality constraints rather than accepting energy closure alone
[4]	Double-pressure and triple-pressure HRSGs, with and without reheat	Increasing steam-generation pressure levels reduced HRSG heat-transfer and stack exergy losses and produced a tangible increase in overall cycle exergy efficiency; triple-pressure reheat gave the lowest thermodynamic losses of the cases compared	Supports the use of multi-pressure heat recovery and the interpretation that pressure-level integration improves exhaust/steam thermal matching
Imani, et al. (2025)	Dual-pressure HRSG using plant-based operating data	Increasing inlet flue-gas temperature from 400 to 700 °C increased HRSG effectiveness from approximately 70% to 86% and reduced stack temperature from 137 to 118 °C	The present GT exhaust temperature of 608.750 °C lies within the investigated 400–700 °C range and therefore represents a credible high-temperature source for dual-pressure recovery

[13]	Experimental low-temperature ORC using SES36 and R245fa	Maximum expander isentropic efficiency was approximately 60% with SES36 and 52% with R245fa at 3000 rpm	Provides experimental context for practical small-scale R245fa expansion; the present model instead assumes $\eta_{is} = 0.800$, so its low 3.602% cycle efficiency is attributable primarily to low source temperature rather than a deliberately low expander efficiency
[8]	Experimental low-grade ORC using R245fa, R123 and mixtures	Maximum system generating efficiency of 4.53% was obtained with an R245fa/R123 mixture	The present ORC net thermal efficiency of 3.602% is lower than values reported for systems driven by substantially higher-temperature heat sources; however, this difference is thermodynamically consistent with the present ORC turbine-inlet temperature of only 50 °C, which imposes a much smaller source-to-sink temperature potential for work conversion. The literature comparison therefore supports the physical plausibility of the result rather than constituting direct performance validation.
[14]	Review of recent theoretical and experimental ORC waste-heat-recovery studies	Reported ORC thermal efficiencies generally span approximately 0.02–0.19, with performance strongly dependent on source temperature, sink conditions, working fluid and scale	Supports interpreting the present 3.602% ORC result within its unusually low-temperature operating domain rather than comparing it directly with warmer ORC systems
Ochoa, et al. (2016)	Dynamically modelled and experimentally validated single-effect LiBr/H ₂ O absorption chiller	Manufacturer/experimental comparison yielded an operating COP of approximately 0.7, with maximum relative temperature deviations of about 5%	Closely supports the physical plausibility of the present single-effect LiBr/H ₂ O COP = 0.720
Sow and Grosu (2025)	Single-effect H ₂ O–LiBr absorption chiller under moderate and hot climates	COP increased from 0.672 to 0.776 as generator temperature increased from approximately 85 to 95 °C; maximum COP approached 0.80 under favourable conditions	The present COP = 0.720 lies within this reported interval; comparison is indicative because the present hot-water loop operates at 110/95 °C and heat-rejection conditions differ
Bishal, et al. (2023)	Integrated GT–sCO ₂ recompression–ORC–absorption cooling system	Maximum energy-utilisation factor = 49.09% and absorption-system COP = 0.72	The absorption COP is numerically identical to the present 0.720 result, although the cycle architecture and absorption working pair differ; therefore, it provides contextual comparison rather than direct validation
Njoku, et al. (2018)	Nigerian CCPP integrated with ORC and absorption refrigeration	The R113 ORC generated an additional 7.5 MW; inlet-air cooling through the absorption subsystem enabled approximately 51.1 MW of additional GT output; total plant thermal and exergy efficiencies increased by 8.7% and 8.8%, respectively	Demonstrates the Nigerian relevance of downstream combined-cycle waste-heat utilisation, but direct comparison of ORC power is inappropriate because the working fluid, heat source, plant scale and absorption-system function differ from the present architecture
Present study	GT–dual-pressure re-heat HRSG–ST–R245fa ORC–LiBr/H ₂ O cascaded system	150.666 MW thermodynamic-system busbar output; 50.341% LHV electrical efficiency; 3.685 MW ORC output at 3.602% net ORC thermal efficiency; LiBr/H ₂ O COP = 0.720	Distinguishes itself through explicit source attribution: LP-steam condensation heat → ORC, while post-HRSG combustion-gas heat → hot-water loop → LiBr/H ₂ O, eliminating duplicate allocation of recovered heat

Model credibility and numerical consistency

The model-acceptance procedure extended beyond Aspen convergence. The principal verification criteria are summarised in Table 8. The appendix reconstruction independently reproduces the controlling net busbar output of 150.666 MW, LHV electrical efficiency of 50.341%, fuel-stream exergy of 314.131 MW, and electricity-only ex-

ergy efficiency of 47.963% (Figure 6). This level of cross-checking is particularly important for highly integrated process models because inconsistencies at one thermal-coupling interface can propagate into multiple downstream performance indicators. In the present study, independent source/sink heat-duty reconciliation was therefore used alongside conventional mass- and energy-balance closure to verify the thermodynamic integrity of the cascaded architecture.

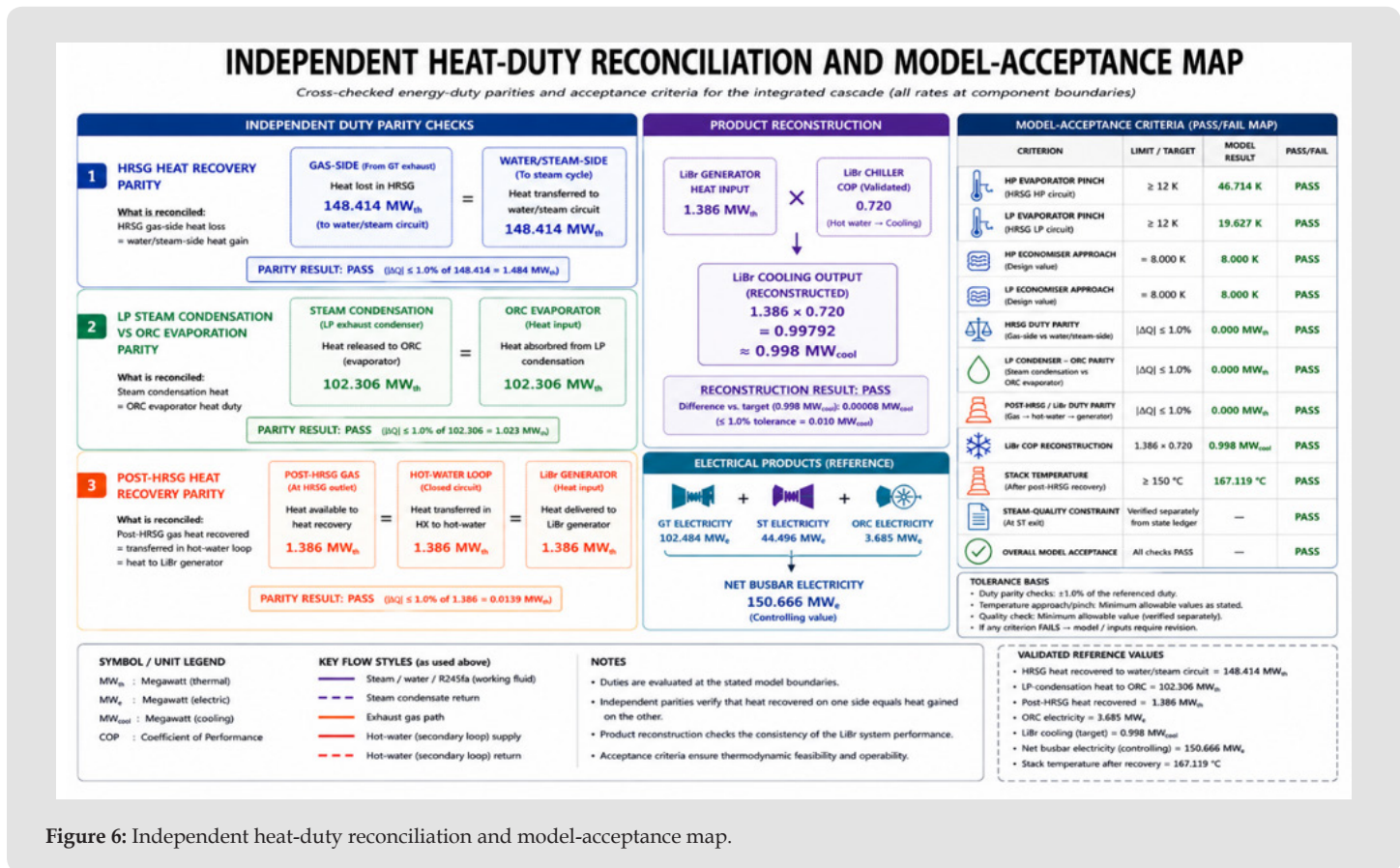


Figure 6: Independent heat-duty reconciliation and model-acceptance map.

Table 8: Principal model-acceptance ledger.

Verification item	Design-point result	Status
Net thermodynamic-system busbar output	150.666 MW	Pass
Fuel LHV input	299.289 MW	Pass
Net LHV electrical efficiency	50.34%	Pass
Total fuel-stream exergy	314.131 MW	Pass
Electricity-only exergy efficiency	47.96%	Pass
HRSG energy reconciliation	Within adopted tolerance	Pass
Steam-condenser/ORC coupling	Reconciled	Pass
ORC first-law closure	≈0 at controlling precision	Pass
Post-HRSG hot-water coupling	1.386 MW on both sides	Pass
LiBr/H ₂ O mass/salt conservation	Within numerical tolerance	Pass
ORC terminal-temperature constraints	Satisfied	Pass
Steam exhaust-quality constraint	Satisfied	Pass
Stack temperature	167.119 °C > 150 °C	Pass

Engineering Implications

The results demonstrate that evaluation of flare gas solely through simple-cycle GT performance can understate its useful-energy potential. The steam and ORC bottoming systems provide approximately

48.181 MW, or 31.98%, of total busbar electricity without additional fuel input. This result is consistent with the established rationale for combined-cycle and waste-heat-recovery systems, in which down-stream cycles convert thermal energy remaining after the topping cycle into additional useful output [4-5]. The refrigeration subsystem

further converts terminal low-grade heat into a useful non-electrical product rather than forcing this resource through another low-temperature electrical-conversion stage. The broader engineering implication is that flare-gas utilisation should be designed around both resource quantity and thermodynamic grade. Nevertheless, thermodynamic feasibility alone does not establish commercial suitability. Actual deployment will depend on flare availability and composition, gas conditioning, equipment reliability, auxiliary loads, electricity and cooling demand, infrastructure and project economics.

Comparative Synthesis

Across the verified literature comparisons, the present design lies within a physically credible thermodynamic regime while remaining distinct in its configuration. The 50.341% thermodynamic-system electrical efficiency reflects the efficiency-enhancing role expected from combined Brayton–Rankine heat recovery, although direct comparison with commercial combined-cycle efficiency must account for the explicitly defined auxiliary boundary of the present model. The 3.602% ORC thermal efficiency is lower than values obtained experimentally from warmer low-grade heat sources but is compatible with the exceptionally low 50 °C turbine-inlet condition. [8], for example,

reported a maximum generating efficiency of 4.53% using an R245fa/R123 mixture with a 120 °C heat source, while Pang et al. (2017) reported approximately 3.9% for pure R245fa at a 110 °C heat-source condition and approximately 4.4% for an R245fa/R123 mixture at 120 °C. The substantially lower thermal level of the present ORC therefore provides a physically credible explanation for its modest conversion efficiency [13–15].

The calculated LiBr/H₂O thermal COP of 0.720 is consistent with the performance range reported for single-effect absorption systems operating under comparable generator and heat-rejection conditions. Because COP is strongly influenced by generator, absorber, condenser and evaporator temperatures, the present value is interpreted as thermodynamically credible rather than as a direct one-to-one validation against systems operating under different boundary conditions (Figure 7). The principal contribution consequently lies less in achieving unprecedented stand-alone component efficiencies than in demonstrating a fully reconciled thermal cascade in which high-, intermediate- and low-grade thermal resources are allocated sequentially while the two terminal recovery systems remain thermodynamically independent.

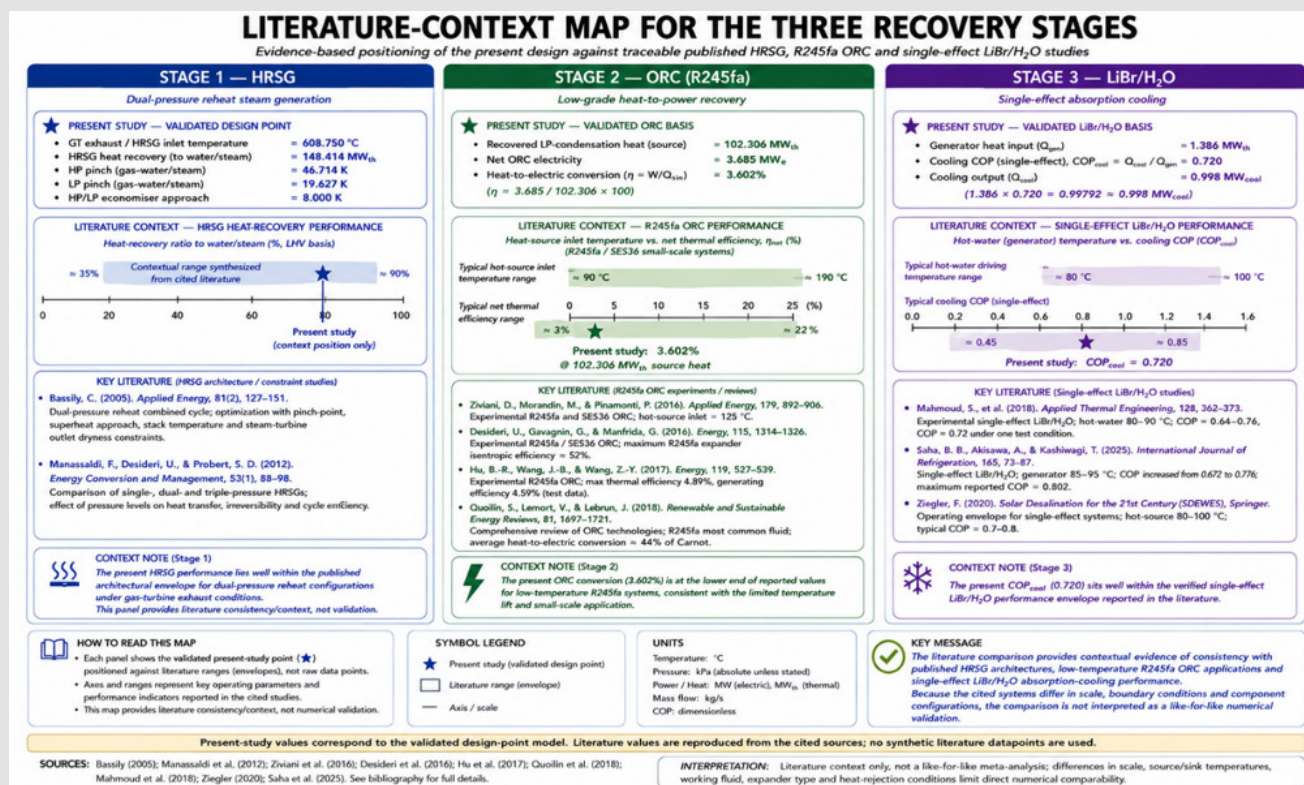


Figure 7: Literature-context map for the three recovery stages.

Limitations

Several limitations define the interpretation of the present results. The model represents a steady-state design point and therefore does not capture transient flare-gas availability, start-up/shutdown behaviour, control dynamics or equipment degradation. Methane-equivalent fuel was adopted as the controlling thermodynamic fuel representation. Real flare gas can contain heavier hydrocarbons, CO₂, N₂, water, H₂S and other constituents. Site-specific deployment therefore requires compositional characterisation and appropriate fuel-treatment modelling. The reported 150.666 MW is net electrical power at the explicitly defined thermodynamic-system busbar. Electrical requirements associated with unmodelled cooling-tower fans, external cooling-water pumps and miscellaneous plant services are outside that boundary. The value must consequently not be presented as fully auxiliary-adjusted commercial grid export. R245fa remains technically useful for investigating low-temperature ORC behaviour because of its favourable thermophysical characteristics and extensive use in waste-heat-recovery studies; however, as HFC-245fa, it has a relatively high 100-year global-warming potential, which has motivated increasing interest in lower-GWP alternatives for future ORC applications (IPCC, 2021; Wang et al., 2021). Finally, the present study establishes thermodynamic feasibility rather than commercial viability. Techno-economic analysis, environmental life-cycle assessment, off-design performance, optimisation and greenhouse-gas accounting require separate analytical treatment.

Conclusion

A thermodynamically reconciled GT–dual-pressure reheat HRSG–ST–R245fa ORC–LiBr/H₂O system was developed for cascaded utilisation of methane-equivalent flare gas. At the design point, a fuel input of 299.289 MW produced 102.484 MW from the GT, 44.496 MW from the steam cycle and 3.685 MW from the ORC, yielding 150.666 MW at the defined thermodynamic-system busbar. The corresponding electricity-only LHV and exergy efficiencies were 50.341% and 47.963%, respectively. The GT exhaust supplied 148.414 MW to the dual-pressure reheat HRSG. LP-steam condensation subsequently supplied 102.306 MW to the ORC, producing 3.685 MW at a net ORC thermal efficiency of approximately 3.602%. Independently, 1.386 MW of residual post-HRSG combustion-gas heat drove the LiBr/H₂O subsystem through an intermediate hot-water circuit, producing 0.998 MW of refrigeration at COP = 0.720.

The final stack temperature of 167.119 °C remained 17.119 K above the imposed minimum. The principal contribution is the explicit thermodynamic segregation of the two downstream recovery pathways. The ORC recovers LP-steam condensation heat, whereas absorption refrigeration recovers residual post-HRSG combustion-gas heat. Consequently, each useful product is traceable to a unique thermal source and no downstream heat resource is counted twice. These results establish a verified design-point foundation for subsequent

assessment of flare-gas cascaded systems. Further work should address realistic multicomponent flare gas, lower-GWP ORC fluids, full balance-of-plant auxiliaries, off-design and transient operation, and techno-economic and environmental performance ([Appendix Text](#)).

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