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Power–Exergy Trade-Off in Simple and Two-Stage Intercooled Brayton Cycles: An Integrated Thermodynamic and Multi-Objective Assessment

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Abstract

Gas turbine cycles are widely employed for power generation and propulsion, yet their thermodynamic performance is strongly influenced by compressor pressure ratio, turbine inlet temperature, ambient temperature, and internal irreversibilities. Conventional performance assessments predominantly emphasize thermal efficiency and net power output, whereas the simultaneous effects of entropy generation, component-wise exergy destruction, operating feasibility, and multi-objective performance of simple and intercooled Brayton cycles remain insufficiently quantified. In particular, the thermodynamic trade-off associated with intercooling between enhanced specific power and increased irreversibility requires systematic investigation under variable operating conditions. This study addresses this gap through a comprehensive energy, entropy-generation, and exergy-destruction analysis of simple and two-stage intercooled gas turbine cycles. An ideal-gas air model with temperature-dependent specific heat for combustion products is developed, incorporating compressor and turbine isentropic efficiencies, combustor efficiency and heat loss, intercooler effectiveness, and variations in pressure ratio, turbine inlet temperature, and ambient temperature. Physical and chemical exergies of the fuel and combustion products are evaluated, while component-wise entropy generation and exergy destruction are quantified. The computational framework is numerically verified using analytical Brayton-cycle relations, exergy–entropy consistency, component-wise exergy balances, and the overall cycle exergy balance. A total of 1,274 operating cases are evaluated, followed by operating-feasibility, equal-weight multi-objective decision-index, and Pareto analyses. The results show that intercooling substantially increases maximum specific net work, reaching 532.91 kJ/kg air,

compared with 459.00 kJ/kg air for the simple cycle, corresponding to an improvement of approximately 16.1%. However, the maximum thermal efficiency is higher for the simple cycle (41.85%) than for the intercooled cycle (40.59%), while the corresponding maximum second-law efficiencies are 40.37% and 39.16%, respectively. Combustion is identified as the dominant source of exergy destruction, accounting for approximately 76.6% of the total in the simple cycle and 75.0% in the intercooled cycle. Within the investigated operating domain, the intercooled cycle achieves higher maximum specific net work, while its minimum total exergy destruction remains higher than that of the simple cycle. The results therefore indicate a thermodynamic trade-off between the power benefit of intercooling and the additional irreversibility associated with the intercooled configuration. These findings highlight the importance of an integrated multi-objective thermodynamic assessment when selecting gas-turbine cycle configurations.

Keywords: *Gas turbine; Brayton cycle; intercooling; entropy generation; exergy destruction; second-law efficiency; thermodynamic optimization.*

1. Introduction

Gas-turbine systems continue to be important for flexible electricity generation because of their high specific power, compact configuration and ability to respond rapidly to variations in power demand. The thermodynamic performance of an open Brayton cycle is governed by several interacting parameters, among which compressor pressure ratio, turbine inlet temperature (TIT), component efficiencies and compressor inlet conditions are particularly important. Increasing the pressure ratio can improve thermal efficiency up to an optimum operating region; however, further increases also increase compressor work and can eventually reduce the net specific work. Consequently, the pressure ratio corresponding to maximum power output is not necessarily the same as that corresponding to maximum thermal efficiency [1].

One of the established approaches for improving Brayton-cycle performance is intercooling during compression. In a multistage compression process, reducing the temperature of the air between compressor stages decreases the specific compression work and can therefore increase the net work output. However, intercooling also introduces heat-transfer irreversibility and pressure losses. Thus, its benefit cannot be assessed solely from the reduction in compressor work. Sanjay and Prasad [2] demonstrated that an intercooled combustion-turbine-based combined cycle can achieve a substantial increase in plant work output, while the pressure ratio associated with maximum plant

efficiency differs from that associated with maximum plant work. Their results therefore established an important thermodynamic trade-off between power output and efficiency in intercooled systems.

The second-law characteristics of intercooling are equally important. Exergy analysis identifies the portion of the supplied work potential that is destroyed by irreversible processes and allows the contribution of individual components to be quantified. Kumari and Sanjay [3] compared basic and intercooled gas-turbine cycles over different compressor pressure ratios and combustion-related operating parameters. Their results showed that the intercooled cycle could provide higher specific work and thermal efficiency while reducing overall exergy destruction under their investigated conditions. Importantly, the study also demonstrated that improvements in first-law performance and reductions in exergy destruction are not necessarily governed by the same operating parameters.

Entropy-generation analysis provides a complementary approach because entropy generation directly represents thermodynamic irreversibility and can be related to exergy destruction through the Gouy–Stodola relationship. Choudhary and Sanjay [4] applied entropy-generation minimization to an SOFC–intercooled-recuperated gas-turbine hybrid system and investigated the influence of compression ratio and TIT on specific work, efficiency and entropy generation. Their results demonstrated that entropy generation can exhibit non-monotonic behavior with operating conditions and that an operating point that provides high efficiency does not necessarily correspond to minimum entropy generation. Although their study focused on a hybrid SOFC–gas-turbine configuration, it highlights the importance of treating entropy generation as a separate optimization criterion.

Recent research has increasingly combined intercooling with additional technologies such as waste-heat recovery, organic Rankine cycles and fuel cells. Musharavati et al. [5], for example, investigated an intercooled gas-turbine system integrated with waste-heat recovery and applied triple-objective optimization considering thermodynamic and economic criteria. Similarly, Candra et al. [6] investigated an intercooled gas turbine coupled with an organic Rankine cycle and performed parametric and bi-objective optimization using exergy efficiency and thermoeconomic criteria. These studies demonstrate the increasing importance of multi-objective approaches for identifying operating compromises rather than a single optimum.

More recently, dual-stage intercooling has also been investigated in highly integrated gas-turbine configurations. Han et al. [7] proposed a dual-stage intercooled and recuperative gas-turbine system integrated with a transcritical organic Rankine cycle and evaluated its energy and exergy performance. Such work represents an important development in the state of the art because it demonstrates the potential of dual-stage intercooling when combined with recuperation and bottoming-cycle heat recovery. However, the additional subsystems make it difficult to isolate the thermodynamic effects of intercooling itself.

Another important operating variable for open-cycle gas turbines is ambient temperature. Since atmospheric air is directly processed by the compressor, changes in ambient temperature alter the compressor inlet state and consequently affect compressor work, net power, efficiency and exergy performance. Egware and Ighodaro [8] investigated the influence of ambient air temperature on the exergy sustainability of a 153-MW gas-turbine power plant and demonstrated deterioration in exergy-based performance at higher ambient temperatures. A recent plant-based exergy study similarly reported that increasing ambient temperature decreases gas-turbine net power and exergy efficiency while increasing total exergy destruction, with the combustion chamber identified as the dominant source of exergy destruction [9]. These findings demonstrate the importance of considering environmental operating conditions together with internal cycle parameters.

The growing use of multi-objective optimization further emphasizes that Brayton-cycle performance should not be represented by a single indicator. Recent multi-objective investigations of Brayton configurations have considered simultaneous thermodynamic and system-level objectives and have demonstrated that improvements in one performance indicator can involve penalties in another [10]. For an intercooled cycle, this issue is particularly important because reducing compressor work can increase net power while the intercooler introduces an additional source of irreversibility.

Recent studies published in the Journal of the Brazilian Society of Mechanical Sciences and Engineering further reinforce the importance of appropriate thermodynamic modelling, exergy assessment and operating-condition analysis in gas-turbine systems. Araújo et al. [11] demonstrated that the selection of thermodynamic modelling assumptions can influence the predicted performance of gas-turbine systems, particularly when combustion and working-fluid composition are explicitly considered. Mishra et al. [12] applied exergy-based sustainability

analysis to a combined-cycle gas-turbine plant integrated with a double-effect vapour absorption refrigeration system, illustrating the application of exergy assessment to an integrated power-generation system. Multi-criteria thermodynamic assessment has also been applied to power-cycle systems using competing indicators such as power output, exergy destruction and exergetic efficiency [13]. In addition, computational energy–exergy frameworks have been developed for systematic assessment of thermal power systems [14]. More recently, Aksoy and Son [15] demonstrated the applicability of component-wise gas-turbine modelling for evaluating performance under varying operating conditions.

Fawaz et al. [16] developed a simulation-based design framework for radial turbine rotors using reduced-order modelling and CFD, highlighting the role of numerical modelling and verification in turbomachinery analysis. Okirie et al. [17] investigated operational data analytics for gas-turbine power plants, emphasizing the importance of evaluating gas-turbine behavior over varying operating conditions. Ibrahim et al. [18] applied physical and chemical exergy analysis to an industrial thermal system, demonstrating the usefulness of component-level exergy accounting for identifying thermodynamic losses.

The reviewed literature establishes that intercooling can reduce compressor work and increase the specific work output of gas-turbine cycles. Previous investigations have also demonstrated the importance of pressure ratio, turbine inlet temperature, ambient conditions, and component irreversibilities in determining thermodynamic performance. However, a systematic comparison of simple and two-stage intercooled Brayton cycles that jointly considers these operating variables, component-wise entropy generation, component-wise exergy destruction, and competing performance objectives remains valuable. In addition, the operating conditions that maximize specific net work, thermal efficiency, and second-law efficiency may differ from those that minimize exergy destruction. A single optimum operating point may therefore fail to represent the trade-offs between power output and thermodynamic irreversibility.

The present study addresses these issues through a comparative computational assessment of simple and two-stage intercooled Brayton cycles. Rather than claiming intercooling itself as a novel concept, the study focuses on integrating energy analysis, entropy-generation analysis, component-wise exergy accounting, operating-feasibility assessment, and multi-objective evaluation within a common operating domain.

The specific objectives are:

- To evaluate the effects of compressor pressure ratio, turbine inlet temperature, and ambient temperature on the specific net work, thermal efficiency, and second-law efficiency of simple and two-stage intercooled Brayton cycles.
- To quantify entropy generation and exergy destruction in the principal cycle components and identify their contributions to overall thermodynamic irreversibility.
- To compare the operating feasibility and performance limits of the two cycle configurations, distinguishing feasible positive-net-work cases from infeasible operating conditions.
- To identify and compare operating conditions associated with maximum specific net work, maximum thermal efficiency, maximum second-law efficiency, and minimum total exergy destruction.
- To examine the trade-offs among competing thermodynamic objectives using a multi-objective decision index and Pareto analysis.
- The resulting analysis is intended to clarify the conditions under which intercooling improves specific net work and the extent to which that improvement is accompanied by changes in efficiency and exergy destruction. The findings are limited to the assumptions, parameter ranges, and computational model described in the following sections.

The resulting analysis is intended to clarify when intercooling provides a meaningful power benefit, when the associated irreversibility becomes significant, and how the preferred operating region changes depending on the selected thermodynamic objective.

2. Literature Review

2.1 Brayton-cycle pressure ratio and thermodynamic performance

Pressure ratio is one of the principal design variables governing Brayton-cycle performance. For an irreversible cycle, increasing pressure ratio simultaneously increases the expansion work potential and compressor work. Consequently, the pressure ratio associated with maximum net work can differ from that associated with maximum thermal efficiency [1]. This fundamental trade-off is particularly relevant to intercooled cycles because intercooling modifies the compression-work requirement and consequently shifts the pressure-ratio dependence of the cycle.

Sanjay and Prasad [2] investigated an intercooled combustion-turbine-based combined cycle and showed that the pressure ratio for maximum efficiency was lower than that associated with maximum plant work. Their study reported approximately 20% enhancement in plant work output for the intercooled configuration, while the increase in plant efficiency was comparatively modest. The authors also demonstrated that the pressure ratio between compressor stages is an important design parameter. These findings establish that intercooling should be assessed using multiple performance indicators rather than net work or efficiency alone.

Kumari and Sanjay [3] subsequently performed a comparative thermodynamic assessment of basic and intercooled gas-turbine cycles. Their analysis considered compressor pressure ratio together with turbine-rotor inlet temperature and combustion-related parameters. The intercooled configuration showed higher specific work and thermal efficiency than the basic configuration under the investigated conditions, while its overall exergy destruction was reported to be lower. This study provides an important benchmark for the present simple-versus-intercooled comparison.

The selection of thermodynamic modelling assumptions is also important when evaluating gas-turbine performance. Araújo et al. [11] compared alternative thermodynamic representations of a gas-turbine cogeneration system and demonstrated that assumptions concerning combustion and working-fluid modelling can affect predicted thermodynamic performance. This supports the use of an explicit combustion-product formulation in the present study rather than relying exclusively on a simplified air-standard representation.

2.2 Entropy generation and component irreversibility

Although energy analysis determines how much energy is converted into useful work, it does not identify the degradation of energy quality within individual components. Exergy analysis and entropy-generation analysis overcome this limitation by quantifying the irreversible destruction of work potential.

Exergy-based assessment can account for both physical and chemical contributions to quantify the degradation of work potential and identify the contribution of individual components to overall irreversibility [18]. The consideration of both physical and chemical exergy is particularly relevant for combustion-based energy systems in which the chemical composition of the working fluid changes during the process.

Choudhary and Sanjay [4] investigated entropy-generation minimization in an SOFC–intercooled-recuperated gas-turbine hybrid system. Their results showed that compression ratio and TIT have significant effects on entropy generation and cycle efficiency. The study also demonstrated that entropy generation can vary non-monotonically with TIT, emphasizing that increasing TIT does not necessarily lead to a monotonic improvement in second-law performance.

Computational implementations of energy and exergy analysis have also been developed for systematic assessment of thermal power systems. Costa et al. [14] developed and validated an in-house computational framework combining energy, exergy and economic analyses, demonstrating the usefulness of computational thermodynamic tools for comparing alternative thermal-system configurations.

Recent gas-turbine modelling studies have likewise employed component-by-component thermodynamic formulations to evaluate performance under varying operating conditions. Aksoy and Son [15] developed a stage-by-stage model incorporating the compressor, combustion chamber and turbine and evaluated performance under changing operating conditions. Such component-level modelling is consistent with the station-wise formulation adopted in the present study.

For the present study, the principal focus is instead placed on the five thermodynamic components of the two-stage intercooled cycle: LP compressor, intercooler, HP compressor, combustor and turbine. This allows the contribution of intercooling to be separated from the effects of additional hybrid subsystems.

2.3 Intercooled gas turbines and integrated configurations

The recent literature demonstrates a clear movement from conventional intercooled gas turbines toward integrated systems.

Musharavati et al. [5] investigated waste-heat recovery from an intercooled gas turbine using exergo-economic analysis and triple-objective optimization. Their work illustrates the increasing use of multi-objective techniques for balancing thermodynamic and economic performance.

Candra et al. [6] subsequently investigated an intercooled gas turbine with an ORC for recovery of the heat rejected by the intercooler. Their study included parametric analysis and bi-objective optimization using exergy efficiency and thermoeconomic criteria. The study is particularly

relevant because it demonstrates that the heat rejected during intercooling represents a potentially recoverable energy/exergy stream rather than simply a necessary heat loss.

Han et al. [7] extended this concept to a dual-stage intercooled and recuperative gas turbine integrated with a transcritical ORC. Their results demonstrated substantial system-level efficiency benefits from combining dual-stage intercooling, recuperation and bottoming-cycle heat recovery. This represents a recent state-of-the-art direction in intercooled gas-turbine research.

However, these integrated studies also establish a clear opportunity for a more focused investigation. When ORC, recuperation or fuel-cell subsystems are added, improvements in performance result from several simultaneous mechanisms. The present work instead isolates the two-stage intercooled Brayton cycle so that the competing effects of compressor-work reduction and intercooler-induced irreversibility can be examined directly.

2.4 Influence of ambient temperature

Ambient temperature is particularly important for open Brayton cycles because the compressor directly processes atmospheric air. Egware and Ighodaro [8] investigated a 153-MW gas-turbine power plant and found that increasing ambient temperature adversely affected exergy sustainability indicators. Their study provides direct evidence that environmental temperature should be incorporated into gas-turbine thermodynamic assessments.

A more recent exergy-based assessment of an operating gas-turbine power plant also demonstrated that increasing ambient temperature from 0 to 45 °C reduces net power and exergy efficiency while increasing total exergy destruction. The combustion chamber was found to account for the largest fraction of the total exergy destruction [9].

Similar sensitivity of gas-turbine work output and thermodynamic performance to ambient-temperature variation has also been reported in recent exergy-based assessments of gas-turbine power systems [12]. These studies reinforce the need to consider environmental operating conditions together with internal cycle parameters when assessing gas-turbine performance.

The present study extends this perspective by treating ambient temperature not as a separate off-design correction but as one of three simultaneous independent operating variables, together with pressure ratio and TIT.

2.5 Multi-objective assessment and thermodynamic trade-offs

Multi-objective assessment is increasingly used in Brayton-cycle research because no single performance parameter adequately represents all aspects of cycle operation. Musharavati et al. [5] employed triple-objective optimization in an intercooled gas-turbine/waste-heat-recovery system, while Candra et al. [6] used bi-objective optimization to balance exergy performance and thermoeconomic criteria. A recent multi-objective study of open and closed Brayton-cycle configurations further demonstrated that different objectives can produce different preferred configurations and design solutions [10]. Similarly, Gonca and Başhan [13] evaluated a gas–steam combined power system using multiple thermodynamic criteria, including power output, exergy destruction and exergetic efficiency, demonstrating that different performance indicators can lead to different preferred operating conditions. This supports the use of Pareto-based assessment in the present study rather than relying on a single thermodynamic performance indicator.

For the present two-stage intercooled cycle, this issue is particularly important because intercooling can simultaneously:

- decrease compressor work,
- increase net specific work,
- introduce heat-transfer irreversibility,
- modify combustion conditions,
- change exhaust exergy, and
- alter the pressure ratio at which maximum efficiency occurs.

Therefore, the present study treats specific net work, thermal efficiency, second-law efficiency, SFC and exergy destruction as competing rather than interchangeable performance measures.

Existing studies have established the thermodynamic benefits of intercooling [2,3], investigated entropy generation in intercooled/hybrid Brayton systems [4], and applied multi-objective optimization to intercooled gas-turbine systems with waste-heat recovery or other integrated technologies [5–7]. Ambient-temperature studies have independently demonstrated the sensitivity of gas-turbine power and exergy performance to atmospheric conditions [8,9]. However, a unified analysis that simultaneously varies pressure ratio, TIT and ambient temperature while resolving net power, thermal efficiency, entropy generation, component-level exergy destruction and

positive-net-work feasibility for a two-stage intercooled Brayton cycle remains insufficiently addressed. Moreover, the distinction between operating conditions for maximum power, maximum efficiency and minimum irreversibility has not been adequately established for this isolated two-stage intercooled configuration.

The present study addresses this gap by developing a comprehensive operating map of the two-stage intercooled Brayton cycle and comparing it with the corresponding simple Brayton cycle. Particular emphasis is placed on identifying the component responsible for the dominant irreversibility, quantifying the thermodynamic penalty associated with intercooling, establishing the positive-net-work boundary, and identifying Pareto trade-offs between power generation, efficiency and exergy destruction..

3. Mathematical Formulation and Exergy Analysis

The thermodynamic modelling framework for the simple and two-stage intercooled Brayton cycles is presented in this section. The formulation includes the thermodynamic properties of air and combustion products, compressor and turbine models, combustor energy balance, and component-level entropy generation and exergy destruction. Energy-performance indicators and multi-objective assessment criteria are subsequently defined to compare the two cycle configurations over the investigated operating range. The schematic configurations are shown in Fig. 1, and their corresponding temperature–entropy diagrams are presented in Fig. 2.

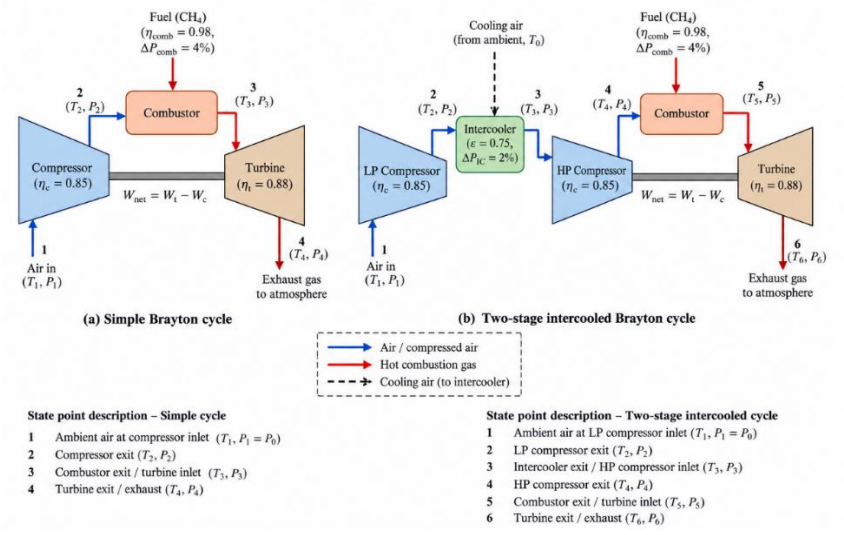
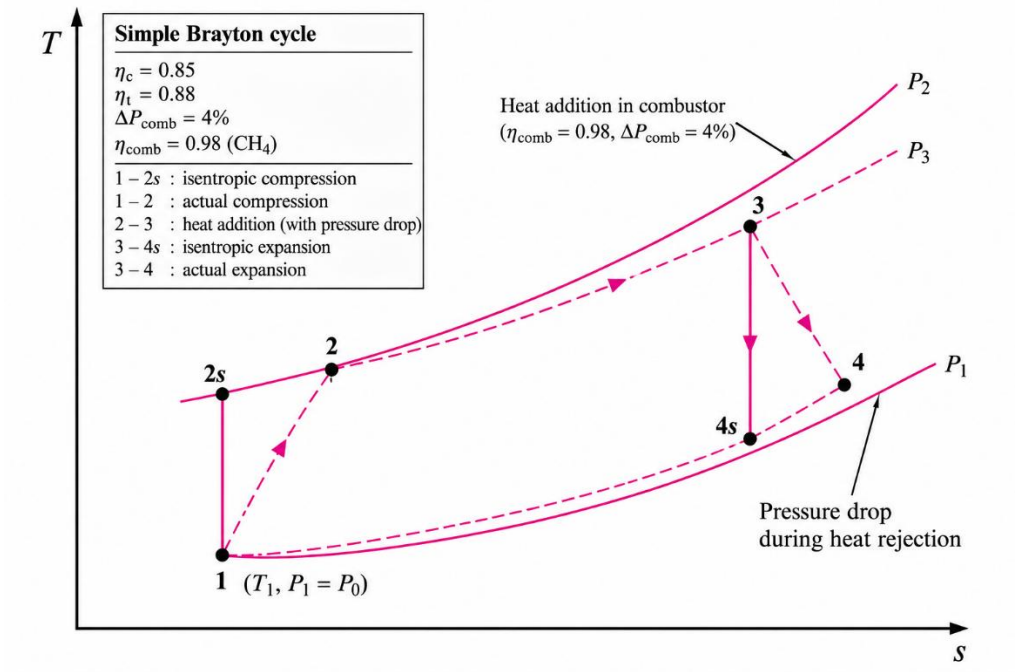
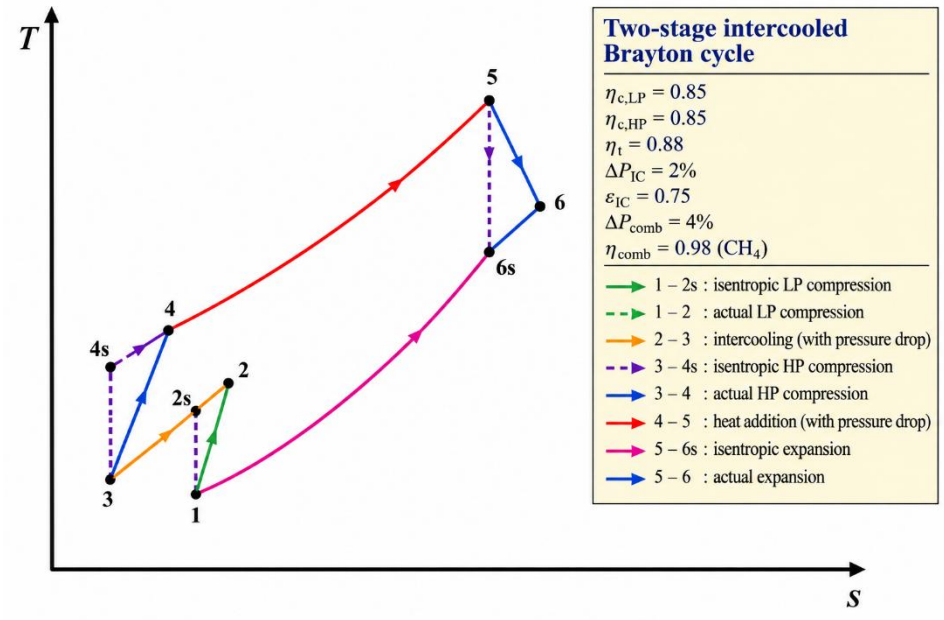


Figure 1. Schematic line diagrams of the simple and two-stage intercooled Brayton gas-turbine cycles.



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318 **(a) Simple Brayton Cycle**



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320 **(b) Two-stage Intercooled Brayton Cycle**

321 **Figure 2. Temperature-entropy (T-s) diagrams of the simple and two-stage intercooled**
322 **Brayton gas-turbine cycles.**

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3.1 Cycle Configuration and Model Assumptions:

Two open Brayton-cycle configurations were investigated: a simple Brayton cycle and a two-stage intercooled Brayton cycle. Both configurations use dry air as the oxidizing medium and methane (CH_4) as the fuel. The analysis is performed on a basis of 1 kg of air entering the cycle.

In the simple cycle, atmospheric air enters the compressor at state 1 and is compressed to state 2. The compressed air then enters the combustor, where methane is supplied and combustion products are generated at the prescribed turbine inlet temperature. The combustion products expand through the turbine and leave the system at the exhaust state.

In the two-stage intercooled cycle, compression is divided into low-pressure (LP) and high-pressure (HP) stages. The air is compressed from state 1 to state 2 in the LP compressor, cooled in the intercooler from state 2 to state 3, and compressed further from state 3 to state 4 in the HP compressor. The compressed air then enters the combustor, and the combustion products expand through the turbine.

The principal independent operating variables are compressor pressure ratio, turbine inlet temperature (TIT), and ambient temperature. The operating ranges are 6–30 for the nominal compressor pressure ratio, 800–1400 °C for TIT, and 15–45 °C for ambient temperature.

The following assumptions are adopted:

- Air and combustion products are modelled as ideal gases.
- Dry air is represented by an oxygen–nitrogen mixture with a molar ratio of 1:3.76.
- Methane undergoes complete combustion.
- Air is modelled with a constant specific heat at constant pressure, whereas the specific heats of the combustion-product species vary with temperature.
- The LP and HP compressor isentropic efficiencies are each 0.85, and the turbine isentropic efficiency is 0.88.
- The intercooler effectiveness is 0.75, with a specified pressure loss of 2%.
- The combustor efficiency is 0.98, and the combustor pressure loss is 4%.
- Changes in kinetic and potential energy are neglected.
- The reference environment is defined by $(T_0 = 298.15\ K)$ and $(P_0 = 101.325\ kPa)$.

- All specific work, heat, and exergy quantities are reported per kilogram of inlet air unless otherwise stated.

The principal model parameters are summarized in Table 1.

Table 1: Model Parameters and Operating Assumptions for the Simple and Intercooled Brayton Cycles

Parameter	Value
Reference temperature, T_0	298.15 K
Reference pressure, P_0	101.325 kPa
Air specific heat, $c_{p,a}$	1.005 kJ kg^{-1} K^{-1}
Air gas constant, R_a	0.28705 kJ kg^{-1} K^{-1}
Methane LHV	50,000 kJ kg^{-1}
Methane chemical exergy	51,830 kJ kg^{-1}
LP compressor efficiency	0.85
HP compressor efficiency	0.85
Turbine efficiency	0.88
Intercooler effectiveness	0.75
Intercooler pressure loss	2%
Combustor pressure loss	4%
Combustion efficiency	0.98

The working fluids are treated as ideal gases, complete methane combustion is assumed, and kinetic and potential energy changes are neglected. Air is modelled using constant specific heat, whereas the specific heats of combustion-product species vary with temperature. The investigated ranges are nominal pressure ratio 6–30, turbine inlet temperature 800–1400°C, and ambient temperature 15–45°C. These values follow the current manuscript's model description.

3.2 Thermodynamic Properties of Air

Air is modelled as an ideal gas with constant specific heat at constant pressure. The adopted values are $c_{p,a} = 1.005$ kJ kg^{-1} K^{-1} and $R_a = 0.28705$ kJ kg^{-1} K^{-1} The specific-heat ratio is obtained

from the relationship between the specific heats at constant pressure and constant volume. The resulting value of γ_a is approximately 1.40. These properties are used to calculate the compressor outlet temperatures, specific compression work and entropy changes.

For an ideal gas, the entropy change of air between two thermodynamic states is given by equation (1)

$$s_2 - s_1 = c_{p,a} \ln\left(\frac{T_2}{T_1}\right) - R_a \ln\left(\frac{P_2}{P_1}\right) \quad (1)$$

Where s is the specific entropy in kJ kg^{-1} K^{-1}, T is temperature in K, and P is pressure in kPa.

The physical specific exergy of air is calculated by the equation (2)

$$e_{ph,a} = c_{p,a}\left[(T - T_0) - T_0 \ln\left(\frac{T}{T_0}\right)\right] + R_a T_0 \ln\left(\frac{P}{P_0}\right) \quad (2)$$

Where, $e_{ph,a}$ is the physical specific exergy of air in kJ kg^{-1}, T_0 is the reference temperature, and P_0 is the reference pressure. The physical specific exergy represents the maximum useful work associated with the departure of the air state from the reference environment, excluding chemical exergy. The reference-state quantities are evaluated using the same property formulation as the corresponding air state.

3.3 Combustion Stoichiometry

Methane is used as the fuel, and complete combustion is assumed. Dry air is represented by an oxygen–nitrogen molar ratio of 1:3.76. On this basis, the methane combustion reaction and the corresponding complete reaction with dry air are expressed as follows.

$$CH_4 + 2(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + 7.52N_2 \quad (3)$$

The stoichiometric air-fuel ratio obtained from the above reaction is $AFR_{st} = 17.12\ kg_{air}/kg_{fuel}$, Where AFR_{st} is the stoichiometric air-fuel ratio. The fuel-air ratio is defined as $f = m_f/m_a$. Where f is the fuel-air ratio, m_f is the methane mass, and m_a is the air mass. Because the calculation basis is 1 kg of inlet air, the calculated fuel mass is numerically equal to the fuel–air ratio when expressed in kilograms of fuel per kilogram of air.

The actual fuel quantity is determined from the combustor energy balance for each combination of pressure ratio, ambient temperature, and turbine inlet temperature as described in section 3.4.

3.4 Combustor Energy Balance

The fuel flow rate is obtained by solving the combustor energy balance. The energy supplied by the fuel together with the sensible energy of the compressed air is equated to the enthalpy rise of the combustion products.

The methane mass required to achieve the specified turbine inlet temperature is determined by solving the combustor energy balance. The balance accounts for the sensible enthalpy of the compressed air and the energy supplied by the fuel, with the combustion efficiency applied according to the implemented model. The resulting combustion-product enthalpy is evaluated using the temperature-dependent property correlations defined in Section 3.9. The governing equation is (4)

$$m_a c_{p,a}(T_2 - T_0) + \eta_{comb} m_f LHV = m_g\left[h_g T_5 - h_g T_0\right] \quad (4)$$

Here, T_2 is compressor outlet temperature, T_5 = turbine inlet temperature, h_g is specific enthalpy of combustion products, kJ kg^{-1}, LHV is lower heating value of methane, kJ kg^{-1}, η_{comb} is combustion efficiency, $m_g\ (m_a + m_f)$ is combustion-product mass, kg, m_a is air mass, kg and m_f is methane mass, kg. The methane mass is adjusted numerically until the implemented energy-balance residual satisfies the specified convergence criterion.

3.5 Simple Brayton-Cycle Compressor

For the simple Brayton cycle, the compressor pressure ratio (PR) is defined as ratio of P_2 (compressor outlet pressure) to P_1 (compressor inlet pressure) $PR = P_2/P_1$.

For the simple Brayton cycle, the compressor raises the pressure of atmospheric air to the combustor inlet pressure. The compressor outlet temperature is first determined for an ideal isentropic process equation (5) and is then corrected using the specified compressor isentropic efficiency by equation (6). The resulting temperature rise is used to calculate the specific compression work (W_c) using equation (7).

$$T_{2s} = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\gamma_a - 1}{\gamma_a}} \quad (5)$$

$$T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_c} \quad (6)$$

$$W_c = c_{p,a}(T_2 - T_1) \quad (7)$$

Where, T_{2s} is the ideal isentropic compressor outlet temperature T_2 is actual compressor exit temperature and η_c is the compressor isentropic efficiency.

The compressor entropy generation is evaluated from the entropy difference between the outlet (s_2) and inlet (s_1) states under the adopted adiabatic-compressor model using equation (8). The corresponding exergy destruction is calculated using the Gouy–Stodola relation equation (9).

$$S_{gen,c} = m_a(s_2 - s_1) \quad (8)$$

$$E_{D,c} = T_0 S_{gen,c} \quad (9)$$

Where, $E_{D,c}$ is compressor exergy destruction in kJ kg^{-1} air and$S_{gen,c}$ is entropy generation in kJ kg^{-1} K^{-1} air.

3.6 Low-Pressure Compressor of the Intercooled Cycle

In the intercooled configuration, the nominal compressor pressure ratio is divided equally between the low pressure (LP) and high pressure (HP) compressor stages. Accordingly nominal stage pressure ratio are $PR_{LP} = \sqrt{PR}$ and $PR_{HP} = \sqrt{PR}$. The LP compressor raises the air pressure from state 1 to state 2, with its outlet pressure determined from the LP stage pressure ratio. The intercooler pressure loss is subsequently applied to determine the pressure at state 3, which becomes the inlet pressure of the HP compressor. The HP compressor then raises the pressure from state 3 to state 4 using the specified HP-stage pressure ratio. Therefore, the actual overall compressor pressure ratio, P_4/P_1, is affected by the intercooler pressure loss and differs from the nominal pressure ratio, (PR). The nominal pressure ratio is retained as the common operating parameter for comparing the simple and intercooled cycle configurations.

$$T_{2s} = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\gamma_a - 1}{\gamma_a}} \quad (10)$$

$$T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_{c,LP}} \quad (11)$$

Where, $\eta_{c,LP}$ is LP compressor isentropic efficiency

$$W_{c,LP} = c_{p,a}(T_2 - T_1) \quad (12)$$

$$S_{gen,LP} = m_a(s_2 - s_1) \quad (13)$$

$$E_{D,LP} = T_0 S_{gen,LP} \quad (14)$$

3.7 Intercooler

The intercooler reduces the temperature of the air leaving the LP compressor before the air enters the HP compressor. The model accounts for the specified intercooler pressure loss ($\Delta P_{IC} = 0.02$) and effectiveness. The outlet temperature and heat rejection are calculated from the implemented effectiveness (ε_{IC}) relation and the air-side energy balance. The pressure loss across the intercooler is represented by equation (15) and the intercooler outlet temperature is calculated using equation (16). Specific heat rejected during the intercooling is evaluated by the equation (17).

$$P_3 = P_2(1 - \Delta P_{IC}) \quad (15)$$

$$T_3 = T_2 - \varepsilon_{IC}(T_2 - T_1) \quad (16)$$

$$Q_{IC} = c_{p,a}(T_2 - T_3) \quad (17)$$

The entropy change of the compressed air across the intercooler is evaluated from the inlet and outlet thermodynamic states using Eq. (18). For the exergy−destruction calculation, the heat rejected by the air is assumed to be transferred to the surroundings at the reference temperature, T_0. The entropy generation is calculated using this adopted boundary assumption, and the corresponding intercooler exergy destruction is determined using the Gouy–Stodola relation, as expressed in Eqs. (19) and (20). Accordingly, the calculated exergy destruction represents the irreversibility evaluated within the scope of the adopted intercooler model and heat−rejection assumption.

$$s_2 - s_1 = c_{p,a} \ln\left(\frac{T_3}{T_2}\right) - R_a \ln\left(\frac{P_3}{P_2}\right) \quad (18)$$

$$S_{gen,IC} = (s_3 - s_2) + \frac{Q_{IC}}{T_0} \quad (19)$$

$$E_{D,IC} = T_0 S_{gen,IC} \quad (20)$$

3.8 High-Pressure Compressor

The HP compressor receives air from the intercooler at state 3 and compresses it to state 4. The outlet temperature is determined using the HP-stage pressure ratio (equation 21) and the specified compressor isentropic efficiency (equation 22). The HP compressor specific work is calculated from the air temperature rise (equation 23). Its entropy generation and exergy destruction are evaluated using the same methodology as that adopted for the LP compressor (equation 24-25).

$$T_{4s} = T_3 \left(\frac{P_4}{P_3}\right)^{\frac{\gamma_a - 1}{\gamma_a}} \quad (21)$$

$$T_4 = T_3 + \frac{T_{4s} - T_3}{\eta_{c,HP}} \quad (22)$$

$$W_{c,HP} = c_{p,a}(T_4 - T_3) \quad (23)$$

$$S_{gen,HP} = (s_4 - s_3) \quad (24)$$

$$E_{D,HP} = T_0 S_{gen,HP} \quad (25)$$

The total compression work for the intercooled cycle is then evaluated by equation 26

$$W_c = W_{c,LP} + W_{c,HP} \quad (26)$$

3.9 Temperature-Dependent Specific Heat of Combustion Products

The specific heat of each combustion-product species is represented using the linear temperature-dependent correlation given in Eq. (27), where the coefficients for the individual species are listed in Table 2.

$$c_{p,i}(T) = a_i + b_i(T - 300) \quad (27)$$

Where, $c_{p,i}$ is the specific heat of species i, a_i & b_i are empirical coefficients, and T is temperature in K. The enthalpy change is obtained by integrating the specific-heat correlation between the selected reference temperature and the state temperature. The mixture enthalpy is then calculated from the contributions of the individual species, using a consistent mass or molar basis.

Table 2: Temperature-Dependent Specific Heat Coefficients for Combustion Product Species

Species	a_i	b_i
CO_2	0.846	1.0×10^{-4}
H_2O	1.864	2.0×10^{-4}
O_2	0.918	1.0×10^{-4}
N_2	1.040	0.8×10^{-4}

$$\Delta h_i = a_i\left(T - T_{ref}\right) + \frac{b_i}{2}\left[(T - 300)^2 - \left(T_{ref} - 300\right)^2\right] \quad (28)$$

Where, T_{ref} is the reference temperature used for the enthalpy calculation. The total enthalpy of the combustion-product mixture is obtained by summing the contributions of all species (equation 29). Where, n_i represents the amount of each combustion-product species.

$$h_g T = \sum_i n_i \, \Delta h_i \quad (29)$$

3.10 Entropy of Combustion Products

The specific entropy of the combustion-product mixture is evaluated by accounting for the temperature and pressure contributions to the entropy of each species. The temperature-dependent contribution is calculated using the corresponding specific-heat relation, while the pressure contribution is included in the species entropy calculation, as expressed in Eqs. (30) and (31). The entropy contributions of the individual combustion-product species are then combined to obtain the mixture-specific entropy using Eq. (32), with all quantities expressed on a consistent mixture basis.

$$\Delta s_{T,i} = (a_i - 300b_i)\ln\left(\frac{T}{T_{ref}}\right) + b_i(T - T_{ref}) \quad (30)$$

$$\Delta s_{P,i} = -R_i \ln\left(\frac{y_i P}{P_0}\right) \quad (31)$$

$$s_g(T,P) = \sum_i \left[\Delta s_{T,i} - \Delta s_{P,i}\right] \quad (32)$$

3.11 Physical Exergy of Combustion Products

The physical specific exergy of the combustion-product mixture is evaluated from the differences between the product-gas state and its corresponding reference state, as expressed in Eqs. (33) and (34). The specific exergy is expressed per unit mass of the combustion-product mixture, in kJ kg^{-1} of products. For the component exergy balance, this value is converted to a basis of 1 kg of inlet air by accounting for the mass of combustion products generated per kilogram of inlet air. Accordingly, the physical exergy used in the combustor exergy balance is expressed in kJ per kilogram of inlet air.

$$e_{ph,g} = \left(h_g - h_{g,0}\right) - T_0\left(s_g - s_{g,0}\right) \quad (33)$$

Where, $e_{ph,g}$ is physical specific exergy of combustion products, kJ kg^{-1}; h_g is product-gas specific enthalpy, kJ kg^{-1}; $h_{g,0}$ is product-gas specific enthalpy at the reference state; s_g is product-gas specific entropy, kJ kg^{-1} K^{-1}; $s_{g,0}$ is product-gas specific entropy at the reference state.

$$Ex_{ph,g} = m_g e_{ph,g} \quad (34)$$

Where, $Ex_{ph,g}$ is the physical exergy of the combustion products in kJ per kg of inlet air.

3.12 Chemical Exergy of Combustion Products

The chemical exergy of the combustion products is evaluated separately from their physical exergy. The standard chemical exergies of the principal combustion-product species used in the model are: $e^{-0}_{ch,CO_2} = 19.87\ kJ/mol$, $e^{-0}_{ch,H_2O} = 9.49\ kJ/mol$, $e^{-0}_{ch,O_2} = 3.97\ kJ/mol$, $e^{-0}_{ch,N_2} = 0.72\ kJ/mol$. The calculation accounts for the standard chemical exergy of each species and the

contribution associated with the composition of the product mixture, as expressed in Eq. (35). The total exergy of the combustion products is then obtained by combining their physical and chemical exergy contributions, as expressed in Eq. (36). All exergy quantities used in the combustor balance are converted to a consistent basis of one kilogram of inlet air.

$$Ex_{ch,g} = \sum_i n_i \, e_{ch,i}^{-0} + \; n_g R_u T_0 \sum_i y_i \ln y_i \tag{35}$$

Where, $Ex_{ch,g}$ = chemical exergy of combustion products, kJ; n_i = number of moles of species *i*; $e_{ch,i}^{-0}$ = standard chemical exergy of species *i,* kJ mol^{-1}; n_g = total number of moles of combustion products; R_u = universal gas constant; y_i = mole fraction of species *i*; T_0 = reference temperature.

$$Ex_g = m_g e_{ph,g} + Ex_{ch,g} \tag{36}$$

Equation explicitly separates the physical and chemical contributions to the total product-gas exergy.

3.13 Fuel Chemical Exergy

The chemical exergy supplied by methane is calculated using the adopted specific chemical exergy of the fuel and the methane mass determined from the combustor energy balance. The fuel chemical-exergy input is expressed per kilogram of inlet air to maintain a consistent basis for the component exergy balances. The chemical exergy supplied by methane is $Ex_f = 51830\, m_f$. Where Ex_f is the methane chemical-exergy input in kJ per kg of inlet air.

3.14 Combustor Exergy Destruction

The combustor exergy balance accounts for the physical exergy of the compressed air entering the combustor and the chemical exergy supplied by methane. The exergy of the combustion products leaving the combustor is evaluated from their physical and chemical contributions. The difference between the total inlet and outlet exergy terms is used to calculate the combustor exergy destruction (equation 37-38). The associated entropy generation is obtained using the Gouy-Stodola theorem using equation (39)

$$Ex_{comb} = Ex_f + Ex_{air,2} \quad (37)$$

$$E_{D,comb} = Ex_f + Ex_{air,2} - Ex_g \quad (38)$$

$$S_{ge,comb} = \frac{E_{D,comb}}{T_0} \quad (39)$$

The combustor entropy generation is obtained from the calculated exergy destruction using the Gouy–Stodola relation. In the implemented model, the energy loss associated with the specified combustion efficiency is treated as heat rejection at the reference temperature, T_0, and therefore contributes zero exergy transfer across the adopted control boundary.

3.15 Turbine Expansion

The combustion products enter the turbine at the turbine inlet temperature, T_5, and pressure, P_5. The turbine exhaust pressure is determined from the specified cycle pressure conditions. The isentropic outlet state is evaluated using the temperature-dependent thermodynamic properties of the combustion-product mixture. The isentropic outlet temperature, T_{6s}, and corresponding enthalpy are determined from the isentropic expansion relation. The actual outlet enthalpy is then calculated using the specified turbine isentropic efficiency and the actual turbine outlet temperature is obtained from the mixture enthalpy relation. The specific turbine work is calculated using Eq. (40), while the turbine entropy generation and exergy destruction are evaluated using Eqs. (41) and (42), respectively.

$$W_t = m_g(h_6 - h_5) \quad (40)$$

$$S_{gen,t} = m_g(s_6 - s_5) \quad (41)$$

$$E_{D,t} = T_0 S_{gen,t} \quad (42)$$

3.16 Net Work Output

The specific net work output of the gas turbine cycle is determined from the difference between the specific turbine work output and the total specific compressor work input. For the simple

Brayton cycle, the compressor work is subtracted from the turbine work to obtain the net work output. For the two-stage intercooled Brayton cycle, the work inputs of the low-pressure and high-pressure compressor stages are added before being subtracted from the turbine work. The resulting specific net work output is expressed in kJ kg^{-1} of inlet air, providing a common basis for comparing the work-producing performance of the two cycle configurations.

3.17 Thermal Efficiency

The thermal efficiency of the gas turbine cycle is evaluated as the ratio of the specific net work output to the chemical energy supplied by the fuel. The specific net work output is calculated by subtracting the total compressor work input from the turbine work output. The fuel energy input is determined from the mass of methane supplied per kilogram of inlet air and the lower heating value of methane. For the intercooled configuration, the work inputs of both compressor stages are included in the net work calculation. The thermal efficiency is expressed as a percentage and is used to compare the energy-conversion performance of the simple and two-stage intercooled Brayton cycles.

$$\eta_{th} = \frac{W_{net}}{m_f LHV} \tag{43}$$

3.18 Specific Fuel Consumption

The specific fuel consumption (SFC) represents the mass of fuel consumed per unit of net work output. It is calculated as the ratio of the methane fuel consumption rate to the specific net work output of the cycle, using a consistent mass and energy basis. For the intercooled configuration, the net work output accounts for the work inputs of both compressor stages. The SFC is expressed in kg kWh^{-1} and is used to assess the fuel-use characteristics of the two cycle configurations (equation 44). Lower SFC values indicate less fuel consumption for a given net work output.

$$SFC = \frac{m_f}{W_{net}} \times 3600 \tag{44}$$

3.19 Total Entropy Generation

The total entropy generation for the simple Brayton cycle is calculated by summing the entropy generation associated with the relevant cycle components, as expressed in Eq. (45). For the two-stage intercooled Brayton cycle, the total entropy generation includes the contributions from the low-pressure compressor, intercooler, high-pressure compressor, combustor, and turbine, as expressed in Eq. (46). The total exergy destruction is subsequently determined from the Gouy–Stodola relation given in Eq. (47), using the reference environmental temperature, T_0. The total exergy destruction is expressed in kJ per kilogram of inlet air.

$$S_{gen,total} = S_{gen,C} + S_{gen,comb} + S_{gen,t} \quad (45)$$

$$S_{gen,total} = S_{gen,LP} + S_{gen,IC} + S_{gen,HP} + S_{gen,comb} + S_{gen,t} \quad (46)$$

$$E_{D,total} = T_0 S_{gen,total} \quad (47)$$

3.20 Component-Wise Exergy Destruction

For the simple cycle, the total exergy destruction is obtained from the component contributions according to equation (48) and for intercooled cycle according to equation (49). The contribution of an individual component to the total exergy destruction is then evaluated according to equation (50). Where, ϕ_j is the percentage contribution of component *j* to total exergy destruction.

$$E_{D,total} = E_{D,c} + E_{D,comb} + E_{D,t} \quad (48)$$

$$E_{D,total} = E_{D,LP} + E_{D,HP} + E_{D,IC} + E_{D,comb} + E_{D,t} \quad (49)$$

$$\phi_j = \frac{E_{D,j}}{E_{D,total}} \times 100 \quad (50)$$

3.21 Overall Exergy Balance

The overall exergy balance is evaluated using Eq. (51) to assess the consistency between the exergy supplied to the system, the useful work output, the exergy carried by the exhaust stream, and the

total exergy destruction within the adopted system boundary. The resulting balance residual is used to quantify the relative exergy-balance error and assess the numerical consistency of the model.

$$Ex_f + Ex_{air,1} = W_{net} + Ex_{exhaust} + E_{D,total} \quad (51)$$

Where, Ex_f is chemical exergy of fuel; $Ex_{air,1}$ is physical exergy of inlet air; W_{net} is net specific useful work; $Ex_{exhaust}$ is total exergy of exhaust gases; $E_{D,total}$ is total exergy destruction.

The exergy-balance residual and relative exergy-balance error is calculated by the equation (52) and equation (53). Zero numerical exergy-balance error within computational precision is expected.

$$R_{Ex} = Ex_f + Ex_{air,1} - W_{net} - Ex_{exhaust} - E_{D,total} \quad (52)$$

$$\epsilon_{Ex} = \frac{|R_{Ex}|}{Ex_f + Ex_{air,1}} \times 100 \quad (53)$$

3.22 Second-Law Efficiency

The second-law efficiency is defined as the ratio of the specific net work output to the chemical exergy supplied by the methane fuel, as expressed in Eq. (54). The specific net work output accounts for the turbine work output and the total compressor work input; for the intercooled configuration, the work inputs of both compressor stages are included. The fuel exergy input is calculated using the methane fuel-to-air mass ratio and the specific chemical exergy of methane. The resulting second-law efficiency indicates the fraction of the supplied fuel exergy converted into useful net work.

$$\eta_{II} = \frac{W_{net}}{Ex_f + Ex_{air,1}} \quad (54)$$

3.23 Multi-Objective Decision Index

The gas-turbine operating conditions were evaluated using multiple performance indicators. These includes net work, thermal efficiency, second-law efficiency, specific fuel consumption, and total exergy destruction. Because these parameters have different units and optimization directions, they were normalized before combining them into a single decision index. For a benefit-type parameter

X, for which a higher value is desirable, the normalized value is obtained by the equation (55). For a cost-type parameter *X*, for which a lower value is desirable, the normalized value is is obtained by the equation (56).

$$X_{norm} = \frac{X - X_{min}}{X_{max} - X_{min}} \tag{55}$$

$$X_{norm} = \frac{X_{max} - X}{X_{max} - X_{min}} \tag{56}$$

The benefit criteria are specific net work output W_{ne}, thermal efficiency η_{th}, and second-law efficiency η_{II}, for which higher values are preferred. The cost criteria are specific fuel consumption *SFC* and total exergy destruction $E_{D,total}$, for which lower values are preferred. Equal weighting is assigned to all five normalized criteria. The overall decision index *DI* is calculated using Eq. (57). A higher *DI* indicates a more favorable compromise among the selected performance criteria under the adopted normalization and weighting scheme.

$$DI = \frac{I_{wnet} + I_{\eta_{th}} + I_{\eta_{II}} + I_{SFC} + I_{E_D}}{5} \tag{57}$$

3.24 Pareto-Optimal Analysis

A Pareto-based analysis was performed to identify feasible operating points that provide favorable trade-offs among four thermodynamic objectives: maximizing specific net work output W_{net}, maximizing thermal efficiency η_{th}, maximizing second-law efficiency η_{th}, and minimizing total exergy destruction $E_{D,total}$,. An operating point is classified as Pareto-optimal when no other feasible operating point can improve at least one objective without worsening at least one of the other objectives. Unlike the decision-index method, Pareto analysis does not combine the objectives into a single score; instead, it identifies a set of non-dominated operating conditions. This distinction is important because maximum power, maximum efficiency, and minimum exergy destruction do not necessarily occur at the same operating condition.

3.25 Numerical Verification and Thermodynamic Consistency

The computational model was subjected to several numerical verification checks to ensure consistency between the governing equations and their implementation.

3.25.1 Analytical Brayton-Cycle Verification

The analytical thermal efficiency of an ideal Brayton cycle is calculates by equation (58). Where, *PR* is the pressure ratio and γ_a is the specific heat ratio of air. The deviation between the analytical and computational values is calculated by equation (59). The maximum deviation between the analytical and computational thermal efficiencies for the ideal Brayton cycle was approximately *4.19* x 10^{-14} *%* over the tested pressure ratios, indicating agreement within numerical precision.

$$\eta_{ideal} = 1 - PR^{\frac{\gamma_a - 1}{\gamma_a}} \tag{58}$$

$$\delta_{ideal} = \left| \frac{\eta_{model} - \eta_{analytical}}{\eta_{analytical}} \right| \times 100 \tag{59}$$

3.25.2 Gouy-Stodola Consistency

For every component, the calculated exergy destruction was checked against the Gouy-Stodola relation equation (60) and the numerical deviation was evaluated by the equation (61). The maximum absolute deviation between the calculated component exergy destruction and the corresponding $T_0 S_{gen}$ value was approximately *1.14* x 10^{-13} kJ/kg of inlet air, indicating agreement within numerical precision.

$$E_{D,j} = T_0 S_{gen,j} \tag{60}$$

$$\delta_{GS,j} = E_{D,j} - T_0 S_{gen,j} \tag{61}$$

3.25.3 Component Exergy-balance Check

The sum of component exergy destructions was compared with the independently calculated total exergy destruction. Equation (62) represents the verification equation and the component-balance residual is represented by the equation (63). A zero value of R_{comp} confirms consistency between the component-wise and total exergy calculations.

$$E_{D,total} = \sum_j E_{D,j} \tag{62}$$

$$R_{comp} = E_{D,total} - \sum_{j} E_{D,j} \quad (63)$$

3.25.4 Overall Exergy-balance

The overall exergy balance was independently checked using equation (64) and corresponding percentage error is calculated using equation (65). The maximum relative error in the overall exergy balance was approximately *1.16* x *10^{-13} %*, indicating that the calculated exergy balance closed within numerical precision.

$$R_{Ex} = Ex_f + Ex_{air,1} - W_{net} - Ex_{exhaust} - E_{D,total} \quad (64)$$

$$\epsilon_{Ex} = \frac{|R_{Ex}|}{Ex_f + Ex_{air,1}} \times 100 \quad (65)$$

3.25.5 Combustion Feasibility

The calculated air-fuel ratio was checked against the stoichiometric air-fuel ratio: AFR ≥ AFRst The calculated air-to-fuel ratio *(AFR)* was compared with the stoichiometric air-to-fuel ratio *(AFR_{st})* to assess combustion feasibility under the adopted model assumptions. The condition indicates that the calculated mixture is not fuel-rich relative to the stoichiometric requirement. The stoichiometric air-to-fuel ratio for methane was 17.12 kg air per kg fuel. Across the investigated operating conditions, the calculated AFR ranged from 33.22 to 272.15 kg air per kg fuel, and no operating point violated the specified feasibility condition.

3.26 Computational Procedure

The computational procedure was implemented in Python to evaluate the simple and two-stage intercooled Brayton cycles over the specified ranges of ambient temperature, nominal compressor pressure ratio, and turbine inlet temperature. For each operating condition, the compressor states were calculated first. For the intercooled configuration, the low-pressure compression, intercooling, and high-pressure compression processes were evaluated sequentially, accounting for the specified pressure losses. The methane fuel requirement was then determined from the

combustor energy balance, and the combustion-product composition was calculated using the methane combustion reaction. Temperature-dependent thermodynamic properties were used to determine the enthalpy, entropy, and physical and chemical exergy of the combustion products. The turbine outlet state, turbine work, compressor work, specific net work, thermal efficiency, and specific fuel consumption were subsequently evaluated. Entropy generation and exergy destruction were calculated for the individual components and the overall cycle, followed by the determination of second-law efficiency. The calculated results were checked for combustion feasibility and thermodynamic consistency using the ideal Brayton-cycle relation, the Gouy–Stodola relation, component exergy accounting, and the overall exergy balance. All operating cases were retained for the detailed parametric analysis, while cases with non-positive net work were excluded where appropriate from the performance-selection and multi-objective analyses. Finally, the decision-index and Pareto analyses were performed to identify operating conditions that offer different trade-offs among the selected performance objectives.

4. Results and Discussion

4.1 Overview of Numerical Results

The thermodynamic performance, entropy generation, and exergy destruction of the simple and two-stage intercooled Brayton cycles were evaluated over compressor pressure ratios of 6–30, turbine inlet temperatures of 800–1400°C, and ambient temperatures of 15–45°C. The model incorporates temperature-dependent specific heats for the combustion products and evaluates physical and chemical exergy, component-wise entropy generation, and exergy destruction. A total of 1,274 operating cases was evaluated across the two cycle configurations. Operating feasibility and positive-net-work conditions were assessed separately where required for the subsequent performance-selection and multi-objective analyses.

Numerical verification indicated that the calculated exergy balances and the consistency between exergy destruction and the Gouy–Stodola relation satisfied the prescribed checks within computational precision. The maximum overall exergy-balance error was approximately 1.16×10^{-13} %, while the maximum deviation between component exergy destruction and $T_0 S_{gen}$ was approximately 1.14×10^{-13} kJ/kg of inlet air. No negative component-wise exergy-destruction values or violations of the imposed stoichiometric combustion constraint were detected. These

checks support the numerical consistency of the implemented calculations; however, they should not be interpreted as independent experimental validation of the thermodynamic model.

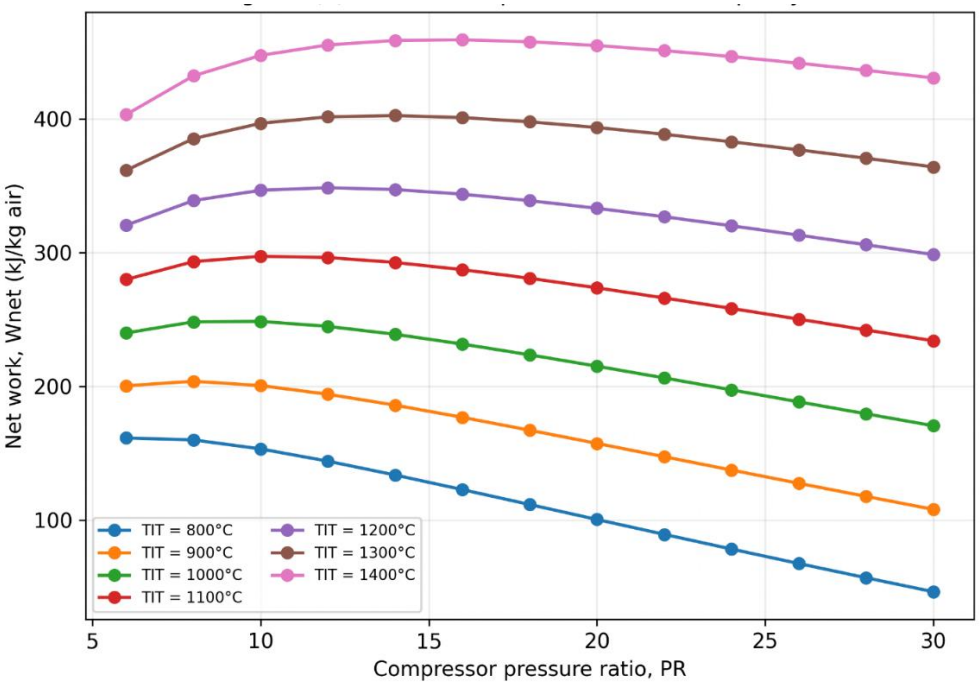
4.2 Effect of Pressure Ratio and Turbine Inlet Temperature on Net Work

Figure 3 shows the variation of specific net work with nominal compressor pressure ratio at different turbine inlet temperatures (TIT) for the simple and two-stage intercooled Brayton cycles. Specific net work is calculated as the turbine work minus the compressor work per unit mass of air. For the simple cycle, the maximum specific net work within the investigated operating range is 459.00 kJ/kg air, obtained at an ambient temperature of 15°C, a nominal pressure ratio of 16, and a TIT of 1400°C.

As shown in Fig. 3(a), increasing TIT generally increases specific net work because the higher turbine inlet temperature increases the enthalpy available for expansion and hence the turbine work. The effect of pressure ratio is non-monotonic. Increasing pressure ratio initially improves specific net work, but the associated increase in compressor work eventually offsets the additional turbine work. Consequently, the maximum specific net work occurs at a pressure ratio of 16 rather than at the upper limit of the investigated pressure-ratio range.

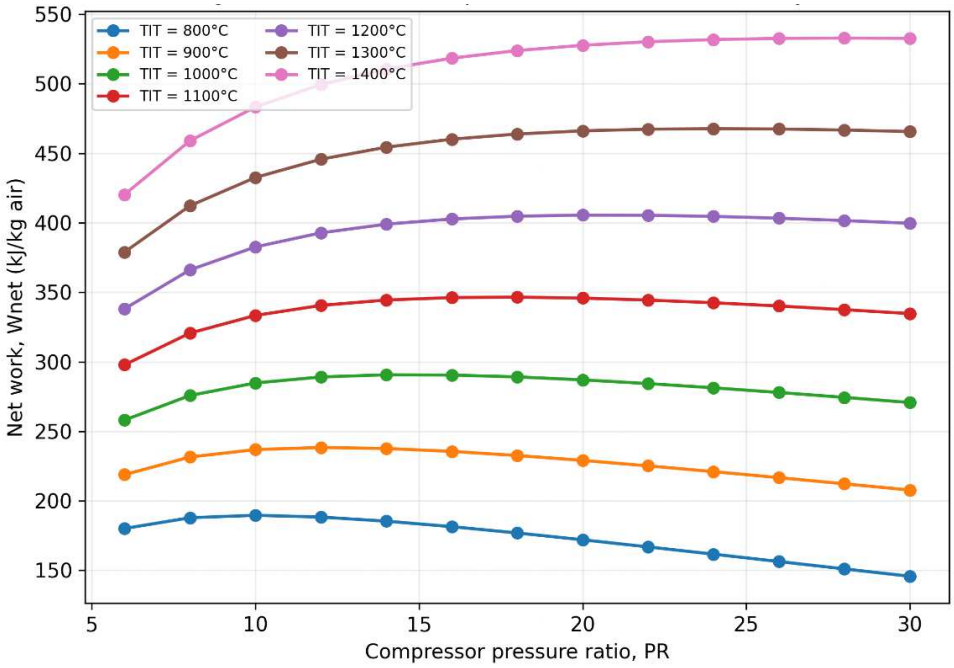
For the two-stage intercooled cycle, Fig. 3(b) shows a maximum specific net work of 532.91 kJ/kg air at an ambient temperature of 15°C, a nominal pressure ratio of 28, and a TIT of 1400°C. This is approximately 16.1% higher than the maximum obtained for the simple cycle. The increase is consistent with the reduction in compression work achieved by cooling the air between the low-pressure and high-pressure compressor stages. Because the high-pressure compressor receives cooler air, its specific work requirement is reduced, allowing the intercooled cycle to retain a net-work advantage at higher pressure ratios.

The shift in the pressure ratio associated with maximum specific net work—from 16 for the simple cycle to 28 for the intercooled cycle—shows that intercooling changes the balance between compressor work and turbine work. However, these values represent the maxima found within the specified ranges of ambient temperature, TIT, and nominal pressure ratio; they should not be interpreted as universal optimum pressure ratios for gas turbines under all operating conditions.



802

803 **(a) Simple Brayton**



804

805 **(b) Two-stage intercooled Brayton cycles**

806 **Figure 3. Variation of specific net work with compressor pressure ratio at different turbine**
807 **inlet temperatures for (a) the simple Brayton cycle and (b) the two-stage intercooled**
808 **Brayton cycle.**

4.3 Effect of Ambient Temperature on Net Work

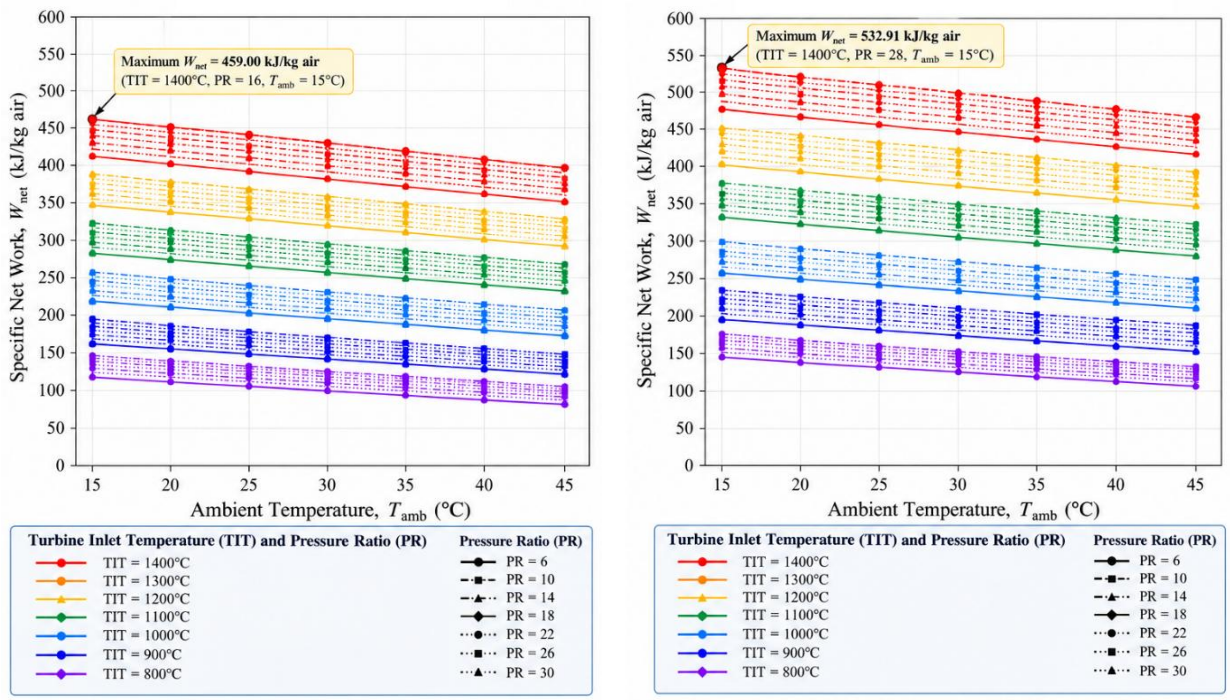
Figure 4 illustrates the variation of specific net work with ambient temperature for different turbine inlet temperatures (TIT) and nominal compressor pressure ratios (PR) for the simple and two-stage intercooled Brayton cycles. The results show how the compressor inlet condition influences the useful work produced by the two configurations over the investigated operating range.

For both cycle configurations, specific net work decreases as ambient temperature increases from 15 to 45°C. At a fixed nominal pressure ratio and TIT, a higher ambient temperature increases the compressor inlet temperature. This raises the temperature at the compressor outlet and increases the specific compression work. Because specific net work is the difference between turbine work and compressor work, the increased compression-work requirement reduces the useful work available from the cycle. The downward trend in Fig. 4 therefore indicates an adverse ambient-temperature effect on gas-turbine power-producing capability.

The effect of TIT is also evident in Fig. 4. At a given ambient temperature and nominal pressure ratio, increasing TIT generally increases specific net work because the higher turbine inlet enthalpy increases the work obtainable during turbine expansion. Thus, the higher-TIT operating conditions generally provide greater specific net work than the lower-TIT conditions. However, the magnitude of the benefit depends on the nominal pressure ratio because the compressor-work requirement also changes with pressure ratio.

The simple-cycle results in Fig. 4(a) show that the combined effects of ambient temperature, TIT, and pressure ratio determine the resulting specific net work. Within the investigated range, the maximum value is 459.00 kJ/kg air at an ambient temperature of 15°C, TIT of 1400°C, and nominal pressure ratio of 16. This result illustrates that the highest specific net work is obtained from a combination of operating parameters rather than from the maximum pressure ratio alone.

Figure 4(b) shows similar ambient-temperature and TIT trends for the two-stage intercooled cycle, but with a different pressure-ratio response. Intercooling reduces the air temperature entering the high-pressure compressor and consequently reduces the work required for the second compression stage. This reduction in compression work contributes to the higher specific net work obtained with the intercooled configuration, particularly at higher nominal pressure ratios. The maximum specific net work is 532.91 kJ/kg air at an ambient temperature of 15°C, TIT of 1400°C, and nominal pressure ratio of 28.



(a) Simple Brayton Cycle **(b) Two-stage intercooled Brayton cycles**

Figure 4. Variation of specific net work with ambient temperature at different turbine inlet temperatures and compressor pressure ratios for (a) the simple Brayton cycle and (b) the two-stage intercooled Brayton cycle..

Although intercooling improves specific net work under the identified high-performance conditions, it does not eliminate the adverse effect of increasing ambient temperature. The compressor inlet temperature still rises with ambient temperature, and the resulting increase in compression work continues to penalize cycle performance. These results indicate that intercooling can mitigate the compression-work penalty associated with high pressure ratios, while ambient-temperature management remains important for maintaining gas-turbine output in hot operating conditions. The reported maxima and trends apply to the operating range and assumptions considered in this numerical study.

4.4 Effect of Pressure Ratio, TIT and ambient temperature on Thermal Efficiency and specific net work output.

Figure 5 presents the variation of thermal efficiency and specific net work with nominal compressor pressure ratio for the simple and two-stage intercooled Brayton cycles at different

855 turbine inlet temperatures (TIT) and ambient temperatures. The calculations cover nominal
856 compressor pressure ratios of 6–30, TIT values of 800–1400°C, and ambient temperatures of 15–
857 45°C. Thermal efficiency and specific net work are considered together because the operating
858 condition that maximizes the conversion of fuel energy into useful work is not necessarily the
859 condition that produces the maximum work per unit mass of air.

860 *Simple Brayton cycle*

861 Figure 5(a) shows that thermal efficiency generally increases with nominal compressor pressure
862 ratio over the lower-to-intermediate range and approaches its maximum at the upper end of the
863 investigated range. This behavior reflects the competing effects of compression and expansion.
864 Increasing the pressure ratio changes both compressor work and turbine work, as well as the heat
865 supplied during combustion. Consequently, thermal efficiency does not necessarily follow the
866 same trend as specific net work.

867 Increasing TIT generally improves thermal efficiency within the investigated operating range
868 because it increases the enthalpy of the gas entering the turbine and the potential work obtainable
869 during expansion. The maximum thermal efficiency calculated for the simple cycle is 41.85%,
870 obtained at an ambient temperature of 15°C, a nominal compressor pressure ratio of 30, and a TIT
871 of 1400°C.

872 The specific-net-work results demonstrate why the two performance indicators should be
873 interpreted separately. The maximum specific net work for the simple cycle is 459.00 kJ/kg of air,
874 obtained at an ambient temperature of 15°C, a nominal pressure ratio of 16, and a TIT of 1400°C.
875 Thus, the pressure ratio associated with maximum net work is lower than that associated with
876 maximum thermal efficiency. Beyond the net-work maximum, the additional compressor-work
877 requirement offsets the benefit of further increasing turbine expansion work, even though thermal
878 efficiency can continue to improve.

879 Ambient temperature adversely affects both performance indicators. At a fixed nominal pressure
880 ratio and TIT, increasing ambient temperature raises the compressor inlet temperature and
881 increases the specific work required for compression. This reduces the difference between turbine
882 work and compressor work and therefore decreases specific net work. The corresponding reduction
883 in thermal efficiency indicates that a smaller fraction of the fuel energy is converted into useful
884 net work under the higher-ambient-temperature conditions considered.

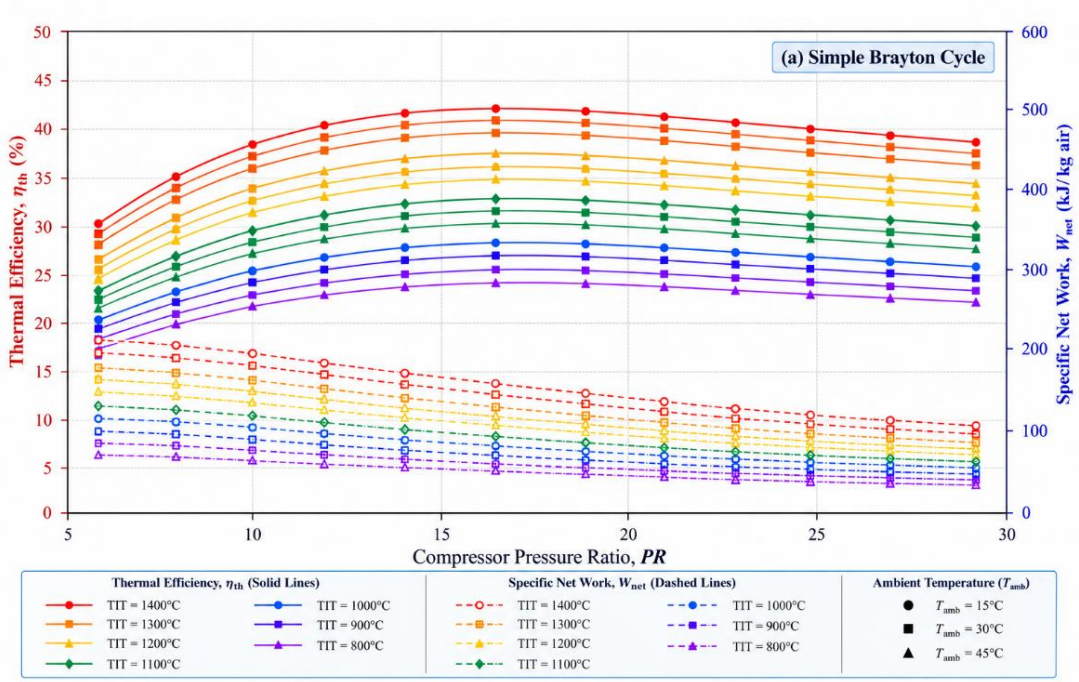
Two-stage intercooled Brayton cycle

Figure 5(b) presents the corresponding results for the two-stage intercooled Brayton cycle. Intercooling reduces the temperature of the air entering the high-pressure compressor, thereby reducing the work required for the second compression stage. This changes the relationship between pressure ratio and net work and allows the cycle to retain a higher net-work output at a higher nominal pressure ratio than the simple cycle.

The maximum thermal efficiency of the intercooled cycle is 40.59%, obtained at an ambient temperature of 15°C, a nominal pressure ratio of 30, and a TIT of 1400°C. This is lower than the maximum thermal efficiency of the simple cycle, which is 41.85%. The result shows that reducing compressor work through intercooling does not automatically improve the overall ratio of net work output to fuel energy supplied. The additional intercooling process and the associated heat rejection must also be considered when evaluating the complete cycle.

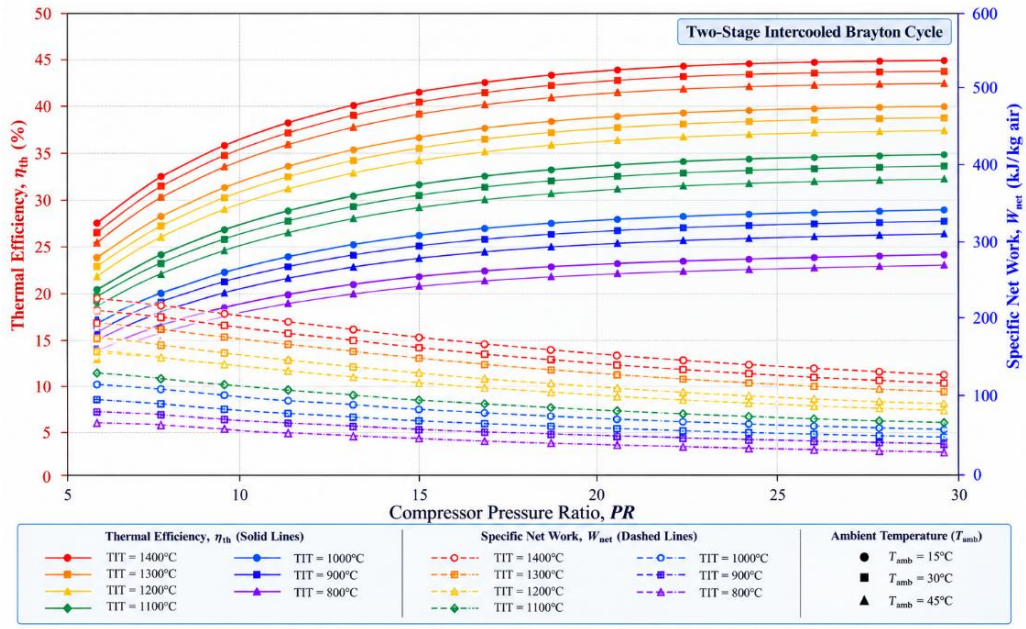
The intercooled cycle achieves a maximum specific net work of 532.91 kJ/kg of air at an ambient temperature of 15°C, a nominal pressure ratio of 28, and a TIT of 1400°C. This maximum is approximately 16.1% higher than the maximum specific net work obtained for the simple cycle. The improvement is consistent with the reduction in compression work achieved by intercooling. The nominal pressure ratio associated with maximum net work therefore shifts from 16 for the simple cycle to 28 for the intercooled cycle.

These results reveal a trade-off between power output and thermal efficiency. Under the reported maximum-performance conditions, the intercooled configuration produces greater specific net work, whereas the simple cycle achieves higher maximum thermal efficiency. Increasing ambient temperature remains detrimental to the performance of both configurations. Accordingly, the preferred cycle configuration depends on whether the principal design objective is to maximize specific net work, maximize thermal efficiency, or balance the two performance measures. The reported maxima apply to the investigated operating range and model assumptions and should not be interpreted as universal optimum operating conditions for all gas turbines.



911

912 **(a) Simple Brayton Cycle**



913

914 **(b) Two-stage intercooled Brayton cycles**

915

916 **Figure 5. Variation of Thermal Efficiency and Specific Net Work with Compressor**

917 **Pressure Ratio at Different Turbine Inlet and Ambient Temperatures**

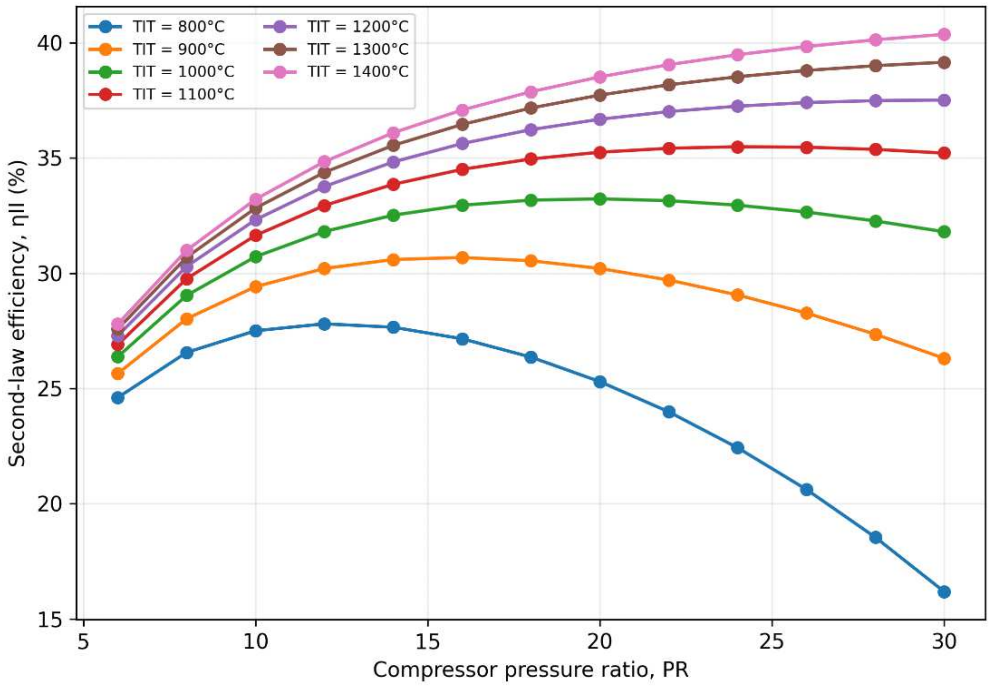
4.5 Effect on Second-Law Efficiency

Figure 6 presents the variation of second-law efficiency with nominal compressor pressure ratio for different turbine inlet temperatures (TIT) for the simple and two-stage intercooled Brayton cycles. Second-law efficiency evaluates cycle performance from an exergy perspective and indicates how effectively the fuel exergy is converted into useful net work. Unlike thermal efficiency, which is based on the energy supplied and useful work produced, second-law efficiency also reflects the thermodynamic quality of the energy conversion processes and the irreversibilities occurring within the cycle.

For the simple Brayton cycle, the maximum second-law efficiency is 40.366%, obtained at an ambient temperature of 15°C, a nominal compressor pressure ratio of 30, and a TIT of 1400°C. For the two-stage intercooled Brayton cycle, the maximum is 39.156%, obtained at the same ambient temperature, nominal pressure ratio, and TIT. Thus, under the investigated conditions, the maximum second-law efficiency of the simple cycle is approximately 1.21 percentage points higher than that of the intercooled cycle.

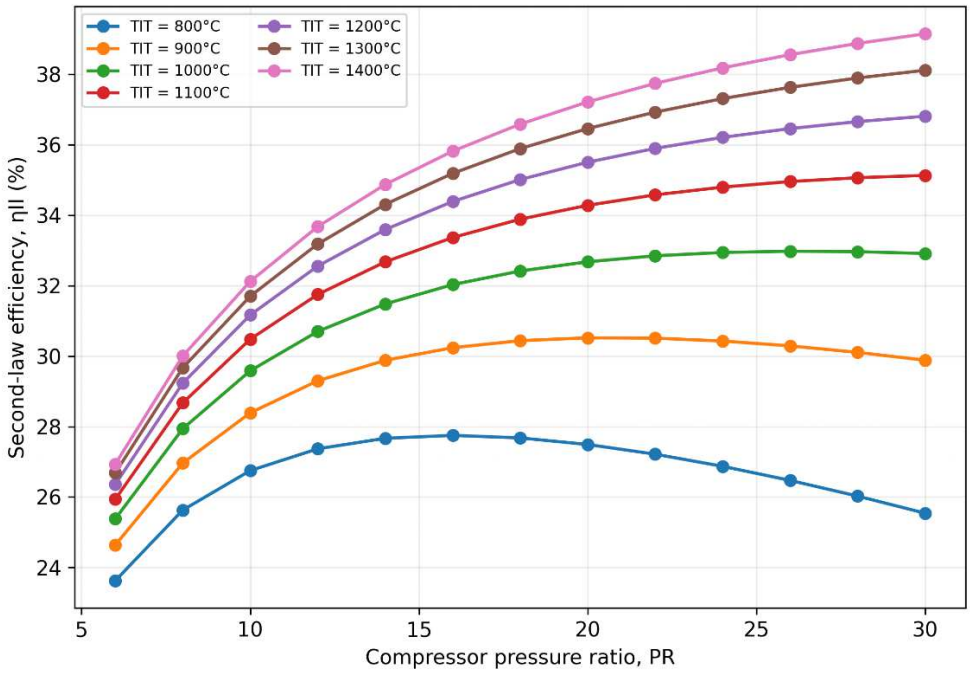
The results indicate that the reduction in compressor work achieved through intercooling does not necessarily produce a corresponding increase in second-law efficiency. Intercooling reduces the temperature of the air entering the high-pressure compressor and lowers the work required for compression. However, the intercooler also involves heat rejection and finite-temperature heat transfer, which contribute to entropy generation and exergy destruction under the assumptions used in the present model. The overall second-law performance therefore depends on the combined effects of compressor work, combustion, turbine expansion, and intercooling rather than on compressor-work reduction alone.

The comparison with thermal efficiency is important. The simple cycle attains a maximum thermal efficiency of 41.85%, whereas the intercooled cycle attains 40.59%. The corresponding second-law-efficiency maxima follow the same ordering. This agreement indicates that the simple cycle performs better according to both maximum-efficiency measures in the investigated range, although the intercooled cycle achieves a higher maximum specific net work. Consequently, selecting a configuration based only on power output may lead to a different choice from selecting it based on energy-conversion efficiency.



947

948 **(a) Simple Brayton**



949

950 **(b) two-stage intercooled Brayton**

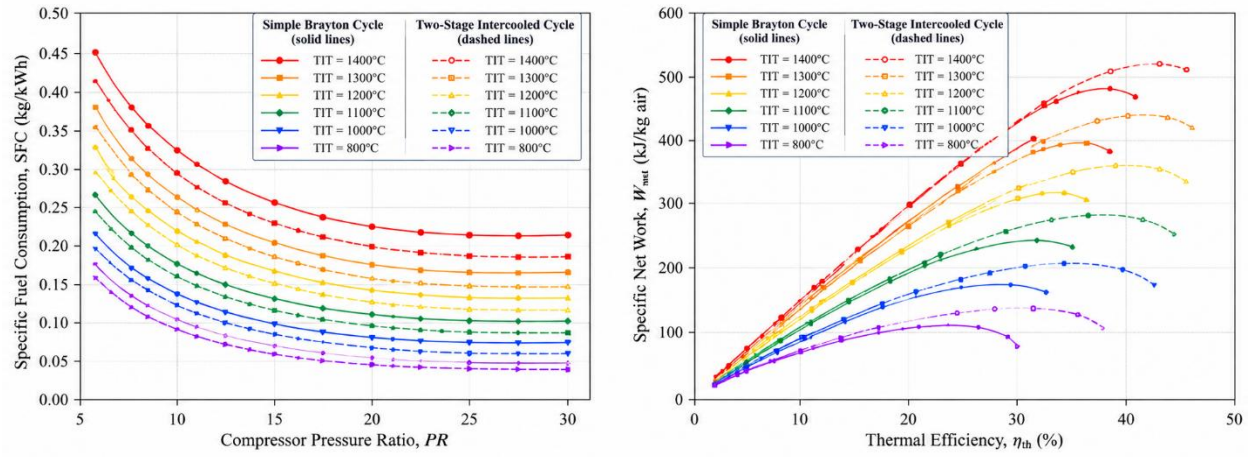
951 **Figure 6. Variation of second-law efficiency with nominal compressor pressure ratio at**
952 **different turbine inlet temperatures for (a) the simple Brayton cycle and (b) the two-stage**
953 **intercooled Brayton cycle.**

These results should be interpreted within the specified operating range and the exergy-accounting assumptions adopted in this study. In particular, the lower maximum second-law efficiency of the intercooled cycle does not, by itself, establish that its total exergy destruction is higher at every matched operating condition. That comparison requires examination of the calculated component-wise and total exergy-destruction values at corresponding operating points.

4.6 Specific Fuel Consumption

Figure 7(a) illustrates the variation of specific fuel consumption (SFC) with compressor pressure ratio for different turbine inlet temperatures (TIT), while Figure 7(b) presents the corresponding relationship between specific net work and thermal efficiency for the simple and two-stage intercooled Brayton cycles.

Figure 7 compares the specific fuel consumption (SFC) and the relationship between specific net work and thermal efficiency for the simple and two-stage intercooled Brayton cycles at an ambient temperature of 15°C. The nominal compressor pressure ratio ranges from 6 to 30, and the turbine inlet temperature (TIT) ranges from 800 to 1400°C. SFC is an important performance indicator because it expresses the fuel mass required to produce a unit of useful energy output. Lower SFC therefore indicates more effective fuel utilization for power generation.



(a) Variation of specific fuel consumption with compressor pressure ratio at different turbine inlet temperatures for simple and two-stage intercooled Brayton cycles.

(b) Variation of specific net work with thermal efficiency at different turbine inlet temperatures for simple and two-stage intercooled Brayton cycles.

Figure 7. Variation of (a) specific fuel consumption with nominal compressor pressure ratio and (b) specific net work with thermal efficiency at different turbine inlet temperatures for the simple and two-stage intercooled Brayton cycles at an ambient temperature of 15°C.

Specific fuel consumption

Figure 7(a) shows that SFC generally decreases as nominal compressor pressure ratio increases over the investigated operating range. The reduction is associated with changes in turbine work, compressor work, and thermal efficiency as the pressure ratio increases. However, the relationship is not governed by pressure ratio alone: the effect depends on TIT and the resulting balance between fuel input and useful net work.

Increasing TIT from 800 to 1400°C shifts the SFC curves toward lower values for both configurations. The higher turbine inlet temperature increases the enthalpy available for expansion and generally increases net work relative to fuel consumption. The resulting improvement in fuel utilization is reflected in the lower calculated SFC at higher TIT.

For the simple Brayton cycle, the minimum SFC decreases from approximately 0.250 kg/kWh at TIT = 800°C to 0.172 kg/kWh at TIT = 1400°C. For the two-stage intercooled cycle, the corresponding minimum decreases from approximately 0.250 to 0.177 kg/kWh. The lowest SFC values obtained in the investigated cases are therefore 0.172 kg/kWh for the simple cycle and 0.177 kg/kWh for the intercooled cycle, both at an ambient temperature of 15°C, TIT = 1400°C, and nominal pressure ratio = 30.

These results indicate that the simple cycle achieves a slightly lower minimum SFC than the intercooled cycle under the reported conditions. Although intercooling reduces compression work and increases the maximum specific net work, these benefits do not necessarily translate into lower fuel consumption per unit of useful output. The fuel required to establish the turbine inlet condition and the resulting cycle efficiency must also be considered.

The nominal pressure ratio associated with minimum SFC changes with TIT. In the simple cycle, the minimum-SFC pressure ratio shifts from approximately 12 at TIT = 800°C to 30 at TIT = 1400°C. In the intercooled cycle, it shifts from approximately 16 to 30 over the same TIT range. This trend indicates that the pressure ratio providing the best fuel utilization depends on the turbine inlet temperature; a single pressure ratio should not be assumed to minimize SFC at every operating condition.

Specific net work versus thermal efficiency

Figure 7(b) complements the SFC analysis by showing the relationship between specific net work and thermal efficiency. For a given TIT, increasing nominal pressure ratio initially improves both indicators over part of the operating range. At higher pressure ratios, however, the additional compressor-work requirement can cause specific net work to decrease even while thermal efficiency continues to increase.

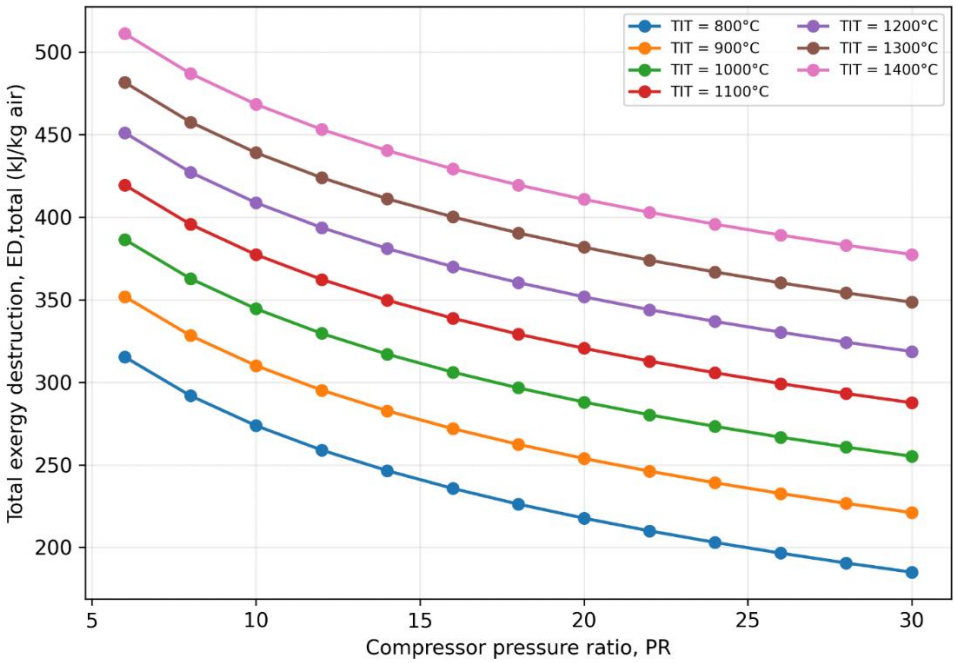
For the simple cycle at TIT = 1400°C and an ambient temperature of 15°C, maximum specific net work is 459.00 kJ/kg of air at a nominal pressure ratio of 16, whereas maximum thermal efficiency is 41.85% at a nominal pressure ratio of 30. For the intercooled cycle, maximum specific net work is 532.91 kJ/kg of air at a nominal pressure ratio of 28, while maximum thermal efficiency is 40.59% at a nominal pressure ratio of 30. These differences demonstrate that the pressure ratio for maximum power output does not necessarily correspond to the pressure ratio for maximum thermal efficiency or minimum SFC.

Increasing TIT generally shifts the specific-work–efficiency relationship toward higher performance levels. In the simple cycle, the reported maximum specific net work increases from approximately 161.6 kJ/kg of air at TIT = 800°C to 459.0 kJ/kg of air at TIT = 1400°C. For the intercooled cycle, it increases from approximately 189.6 to 532.9 kJ/kg of air over the same TIT range. Intercooling therefore provides a substantial net-work advantage under the high-TIT conditions investigated, but the results do not establish that it also provides the lowest SFC or the highest thermal efficiency.

Overall, Figure 7 highlights the need to evaluate fuel utilization, net power output, and thermal efficiency together. The simple cycle achieves the lower minimum SFC and higher maximum thermal efficiency, whereas the intercooled cycle achieves the higher maximum specific net work. The preferred configuration consequently depends on the performance objective and the operating conditions rather than on any single indicator.

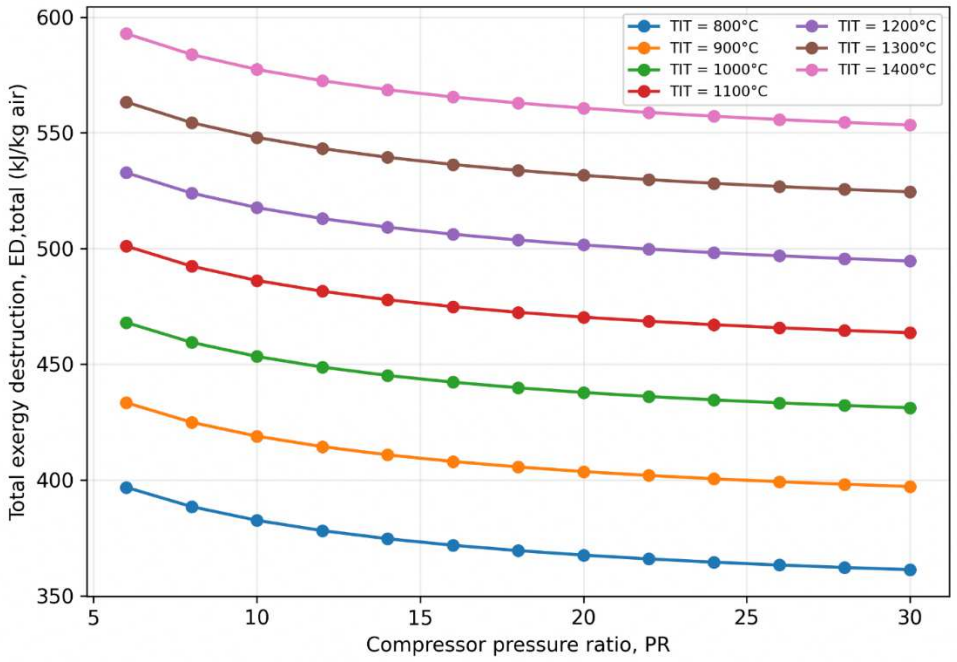
4.7 Effect of Pressure Ratio and TIT on Total Exergy Destruction

Figure 8 presents the variation of total exergy destruction with nominal compressor pressure ratio at different turbine inlet temperatures (TIT) for the simple and two-stage intercooled Brayton cycles. The results are evaluated at an ambient temperature of 15°C, with pressure ratio and TIT. Total exergy destruction quantifies the loss of useful work potential caused by the irreversibilities represented in the thermodynamic model.



1033

1034 **(a) Simple Brayton Cycle.**



1035

1036 **(b)) Two-stage intercooled Brayton cycle.**

1037 **Figure 8. Variation of total exergy destruction with nominal compressor pressure ratio at**

1038 **different turbine inlet temperatures for (a) the simple Brayton cycle and (b) the two-stage**

1039 **intercooled Brayton cycle at an ambient temperature of 15°C.**

Effect of nominal compressor pressure ratio

For the simple Brayton cycle, total exergy destruction decreases with increasing nominal pressure ratio over the investigated range. For example, at TIT = 800°C, total exergy destruction decreases from approximately 315.3 kJ/kg of air at a pressure ratio of 6 to 185.0 kJ/kg at a pressure ratio of 30. At TIT = 1400°C, it decreases from approximately 511.2 to 377.4 kJ/kg over the same pressure-ratio range. These results show that the effect of pressure ratio on exergy destruction differs from its effect on specific net work: the pressure ratio that minimizes exergy destruction is not necessarily the pressure ratio that maximizes net work.

For the two-stage intercooled cycle, total exergy destruction also decreases as nominal pressure ratio increases. At TIT = 800°C, it decreases from approximately 396.9 kJ/kg of air at a pressure ratio of 6 to 361.3 kJ/kg at a pressure ratio of 30. At TIT = 1400°C, the corresponding values are approximately 592.9 and 553.4 kJ/kg. The reduction is smaller than that observed for the simple cycle in the cited cases. This difference reflects the combined contributions of the compressor stages, intercooler, combustor, and turbine, rather than the compression-work reduction alone.

Effect of turbine inlet temperature

Increasing TIT produces a substantial increase in total exergy destruction in both configurations. For the simple cycle, at a nominal pressure ratio of 18, total exergy destruction increases from approximately 226.2 kJ/kg at TIT = 800°C to 419.5 kJ/kg at TIT = 1400°C. The increase is also evident at the pressure-ratio limits: at a pressure ratio of 6, the reported values increase from approximately 315.3 to 511.2 kJ/kg, while at a pressure ratio of 30 they increase from approximately 185.0 to 377.4 kJ/kg.

The same overall trend is observed for the intercooled cycle. At a pressure ratio of 6, total exergy destruction increases from approximately 396.9 kJ/kg at TIT = 800°C to 592.9 kJ/kg at TIT = 1400°C. At a pressure ratio of 30, it increases from approximately 361.3 to 553.4 kJ/kg. These results indicate that the higher-TIT operating conditions are associated with greater absolute exergy destruction in the present model, even though higher TIT also increases the work available from turbine expansion.

The component-wise results identify the combustor as the largest contributor to total exergy destruction. Its average contribution is approximately 249.47 kJ/kg for the simple cycle and 350.42

kJ/kg for the intercooled cycle. The combustor accounts for approximately 76.56% and 75.03% of the respective average total exergy destruction. The importance of combustion irreversibility is therefore evident in both configurations. However, the increase in total exergy destruction with TIT should be interpreted alongside the increased fuel-energy input and power output, rather than being treated as an isolated measure of poor performance.

Comparison of the two configurations

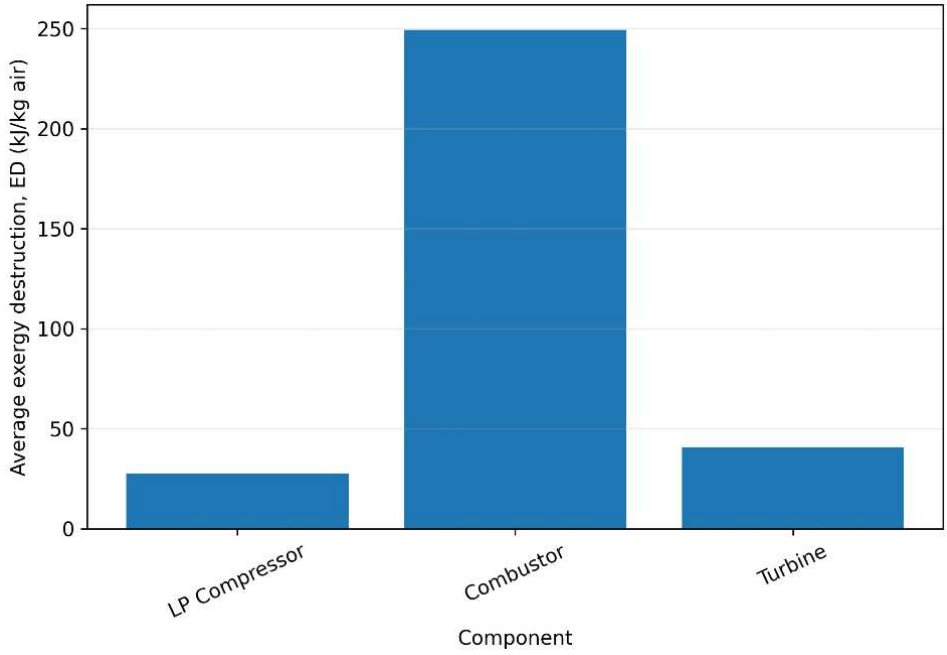
At corresponding TIT and nominal pressure-ratio conditions, the reported results show higher total exergy destruction for the intercooled cycle than for the simple cycle. For example, at TIT = 1200°C and a pressure ratio of 18, the reported values are approximately 360.4 kJ/kg for the simple cycle and 503.7 kJ/kg for the intercooled cycle. At TIT = 1400°C and a pressure ratio of 30, the corresponding values are approximately 377.4 and 553.4 kJ/kg.

The component-wise analysis helps explain this difference. Intercooling reduces the temperature entering the high-pressure compressor and lowers the work required for compression. Nevertheless, the intercooler introduces an additional heat-transfer process and associated exergy destruction. Its average exergy destruction is approximately 38.43 kJ/kg, while the average exergy destruction of each compressor stage is approximately 16.78 kJ/kg. The intercooled cycle also exhibits a higher average combustor exergy destruction in the present calculations. Thus, reduced compression work does not, by itself, ensure lower total exergy destruction.

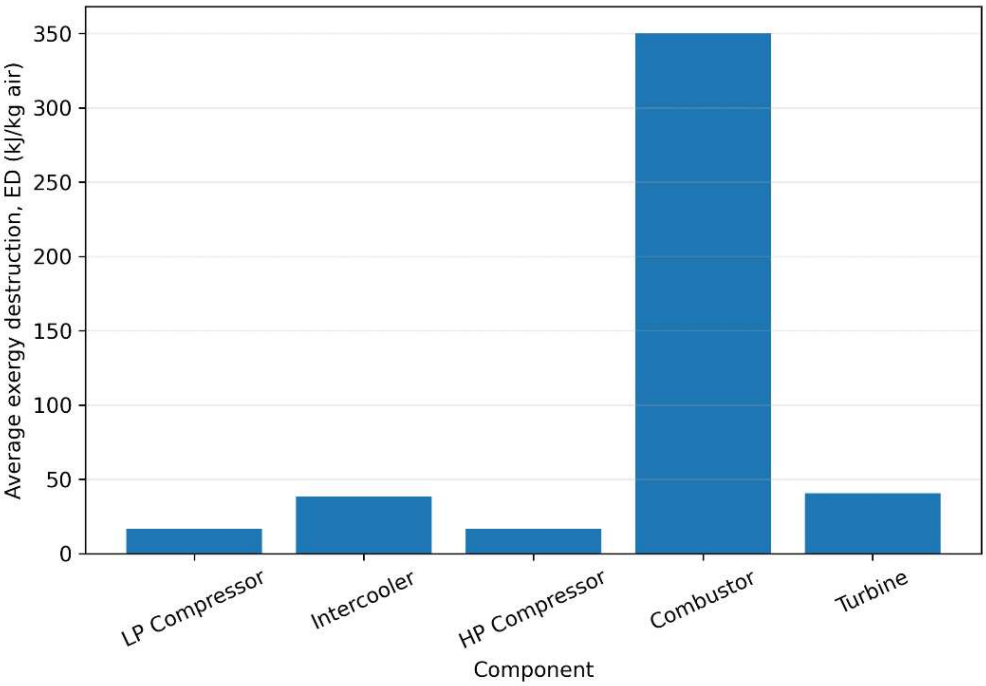
The power-output results provide the complementary perspective. The maximum specific net work is 532.91 kJ/kg of air for the intercooled cycle, compared with 459.00 kJ/kg for the simple cycle. The intercooled configuration therefore provides a higher maximum specific net work, while the reported component and total exergy-destruction results indicate a greater exergy-destruction burden. These findings demonstrate the importance of considering power output and exergy destruction together when comparing cycle configurations.

Overall, within the investigated operating range, increasing nominal pressure ratio reduces total exergy destruction, whereas increasing TIT increases it. The simple cycle exhibits lower total exergy destruction than the intercooled cycle in the reported comparisons, while the intercooled cycle provides higher maximum specific net work. These conclusions apply to the present model, parameter ranges, and exergy-accounting assumptions; they should not be generalized beyond those conditions without further analysis.

4.8 Component-Wise Exergy Destruction



(a) Simple Brayton Cycle



(b) Two-stage intercooled Brayton cycle.

Figure 9. Component-wise exergy destruction for (a) the simple Brayton cycle and (b) the two-stage intercooled Brayton cycle.

Figure 9 compares the component-wise exergy destruction of the simple and two-stage intercooled Brayton cycles. The analysis identifies the combustor as the largest contributor to total exergy destruction in both configurations. The results also show how dividing compression into two stages with intermediate cooling changes the distribution of exergy destruction among the compressors, intercooler, combustor, and turbine.

For the simple Brayton cycle, the average exergy destruction is approximately 249.47 kJ/kg of air in the combustor, 40.79 kJ/kg in the turbine, and 27.47 kJ/kg in the compressor. The combustor therefore represents the dominant source of exergy destruction in the calculated cases. This finding is consistent with the irreversible nature of combustion, which involves chemical reaction and heat transfer across finite temperature differences. The results indicate that combustion-related irreversibility is an important consideration when seeking to improve the thermodynamic performance of the simple cycle.

For the two-stage intercooled Brayton cycle, the average exergy destruction is approximately 350.42 kJ/kg of air in the combustor, 38.43 kJ/kg in the intercooler, 16.78 kJ/kg in the low-pressure compressor, 16.78 kJ/kg in the high-pressure compressor, and 40.65 kJ/kg in the turbine. The combustor remains the largest contributor. The intercooler also makes a distinct contribution because the intermediate cooling process involves heat transfer and associated irreversibility.

Comparison of the compressor results shows that the average exergy destruction calculated for each compressor stage in the intercooled configuration is lower than that of the single compressor in the simple cycle. The simple-cycle compressor contributes approximately 27.47 kJ/kg of air, whereas each compressor stage in the intercooled cycle contributes approximately 16.78 kJ/kg. This comparison is consistent with the reduction in compression work associated with intermediate cooling. However, the two intercooled compressor stages must be considered together when assessing the total compressor-related contribution.

The intercooler introduces an additional average exergy destruction of approximately 38.43 kJ/kg of air. In addition, the average combustor exergy destruction is higher in the intercooled-cycle calculations than in the simple-cycle calculations. Consequently, the reduction in the exergy destruction of each compressor stage does not, by itself, establish a reduction in the total exergy destruction of the complete cycle. The contributions of all components must be included in the comparison.

The component-wise results also help explain the distinction between power output and thermodynamic irreversibility. The intercooled cycle achieves a maximum specific net work of 532.91 kJ/kg of air, compared with 459.00 kJ/kg for the simple cycle. Nevertheless, its calculated component contributions indicate that the additional intercooling process and the higher combustor exergy destruction are important when evaluating its overall exergy performance. A higher net-work output therefore does not necessarily imply lower total exergy destruction.

From a design perspective, the combustor is the primary component to examine when seeking to reduce exergy destruction in either configuration. The intercooler should also be evaluated carefully because its benefit in reducing compressor work must be considered alongside the irreversibility associated with heat rejection. Overall, Figure 9 demonstrates that improving one component does not necessarily minimize the irreversibility of the complete cycle; the effect on total exergy destruction depends on the combined behavior of all components.

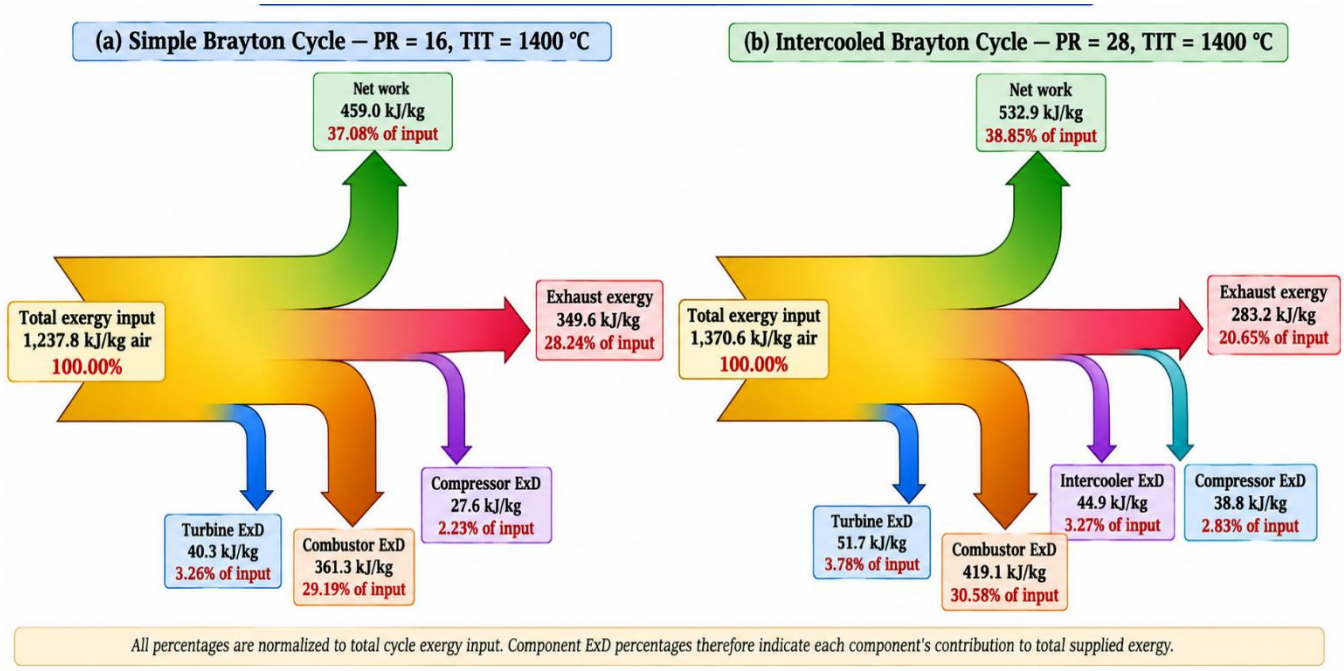


Fig. 10. Proportional exergy-flow (Sankey/Grassmann) diagrams for (a) the simple Brayton cycle at nominal pressure ratio 16 and (b) the two-stage intercooled Brayton cycle at nominal pressure ratio 28, both at a turbine inlet temperature of 1400 °C. All percentages are normalized to the total cycle exergy input.

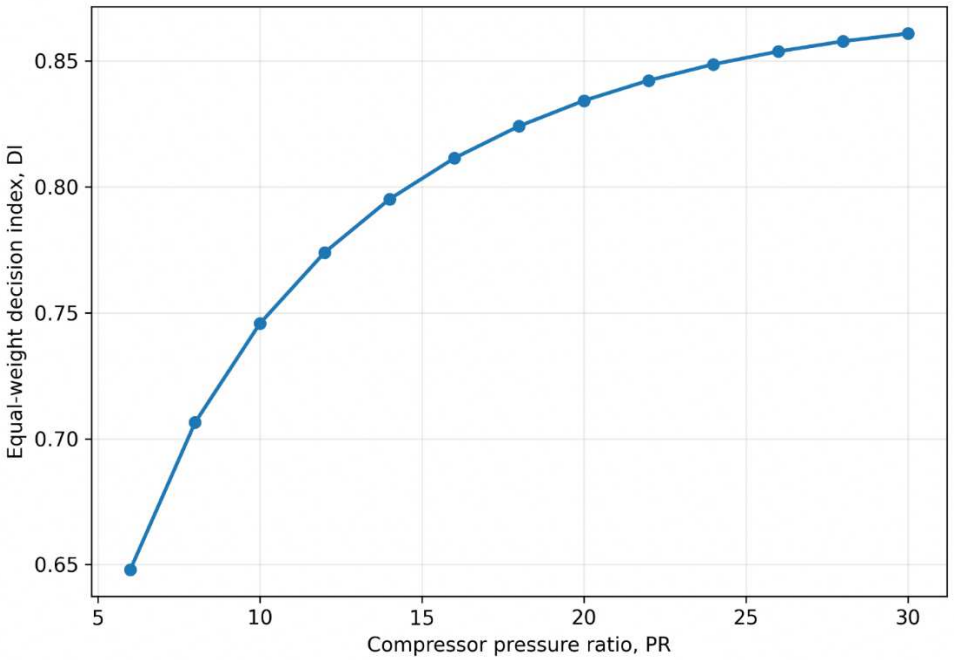
To complement the component-wise exergy destruction analysis, a proportional exergy-flow representation is presented in Fig. 10. The diagram illustrates the distribution of the total exergy input among net work, exhaust exergy, and exergy destruction in the individual components for the selected operating conditions. The simple Brayton cycle is evaluated at a nominal compressor pressure ratio of 16, whereas the intercooled Brayton cycle is evaluated at a nominal compressor pressure ratio of 28; the turbine inlet temperature is 1400 °C for both cases. These operating points correspond to the respective maximum-net-work cases identified in the parametric analysis. Therefore, the figure compares the selected high-work operating points rather than configurations operating at an identical pressure ratio.

As shown in Fig. 10, the simple Brayton cycle produces a net work output of 459.0 kJ/kg, corresponding to 37.08% of the total exergy input of 1237.8 kJ/kg of air. The exhaust stream carries 349.6 kJ/kg of exergy, while the combustor, turbine, and compressor account for the remaining exergy destruction. For the intercooled Brayton cycle, the net work output increases to 532.9 kJ/kg, representing 38.85% of the total exergy input of 1370.6 kJ/kg of air. Its exhaust exergy is 283.2 kJ/kg, or 20.65% of the input. The diagram also identifies the additional exergy destruction associated with the intercooler.

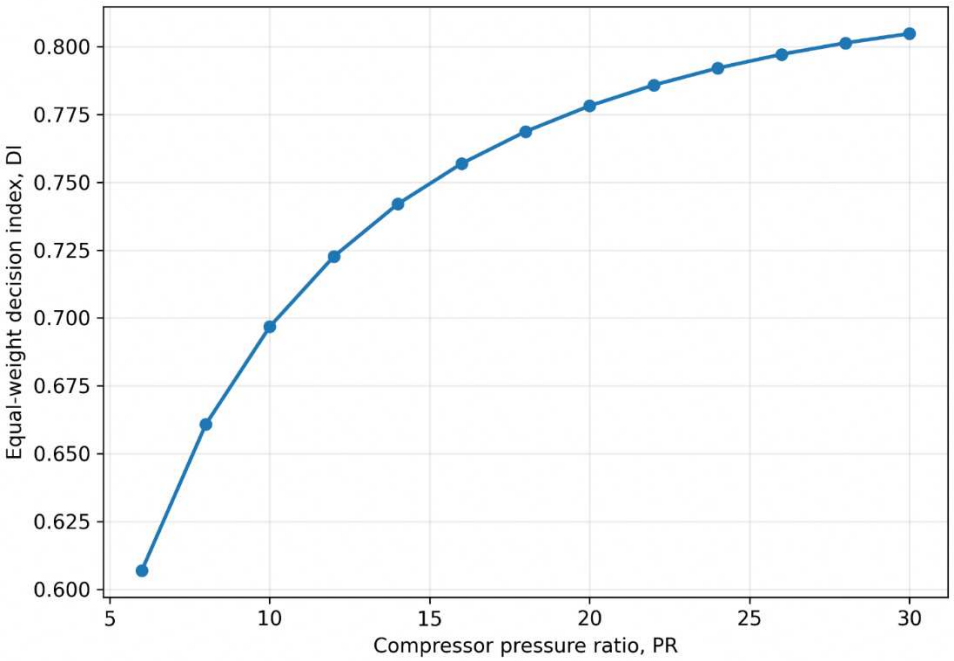
These results illustrate the distribution of exergy at the two selected operating points. Although the intercooled configuration delivers greater net work and a lower exhaust-exergy fraction, it also has a higher total exergy input and additional component-level irreversibility associated with intercooling. Consequently, the diagram should be interpreted together with the thermal-efficiency, second-law-efficiency, and total-exergy-destruction results rather than as evidence that intercooling reduces total irreversibility under all operating conditions.

4.9 Multi-Objective Decision Index

Because maximum specific net work, maximum thermal efficiency, maximum second-law efficiency, minimum SFC, and minimum exergy destruction occur at different operating conditions, a multi-objective decision index was used. The resulting decision-index comparison is shown in Figure 11.



(a) Simple Brayton Cycle



(b) Two-stage intercooled Brayton cycle.

Figure 11. Multi-objective decision index for the simple and two-stage intercooled Brayton cycles.

The multi-objective decision index was used to identify an operating condition that provides a balanced compromise among five performance criteria: net work output, thermal efficiency, second-law efficiency, specific fuel consumption (SFC), and total exergy destruction. Because these criteria have different units and optimization directions, their values were normalized before aggregation. Higher net work, thermal efficiency, and second-law efficiency were treated as beneficial, whereas lower SFC and total exergy destruction were treated as desirable. Equal weighting was assigned to all five criteria; consequently, the resulting index reflects the adopted normalization and weighting scheme rather than a universal ranking independent of decision preferences.

Figure 11 shows the variation of the decision index with nominal compressor pressure ratio at an ambient temperature of 15 °C and a turbine inlet temperature of 1400 °C. For the simple Brayton cycle, the maximum decision index is 0.861 at a nominal pressure ratio of 30. At this condition, the net work output is 430.55 kJ/kg of inlet air, the thermal efficiency is 41.85%, the second-law efficiency is 40.37%, and the SFC is 0.172 kg/kWh. The corresponding total exergy destruction is 377.37 kJ/kg of inlet air. This operating point therefore represents the best compromise according to the five criteria and the equal-weight formulation adopted in this study.

For the two-stage intercooled Brayton cycle, the maximum decision index is 0.805, also at a nominal pressure ratio of 30, an ambient temperature of 15 °C, and a turbine inlet temperature of 1400 °C. At this operating point, the cycle produces 532.68 kJ/kg of inlet air as net work, with a thermal efficiency of 40.59%, a second-law efficiency of 39.16%, and an SFC of 0.177 kg/kWh. Its total exergy destruction is 553.39 kJ/kg of inlet air. Thus, at the respective decision-index optima, the intercooled cycle produces approximately 23.7% more net work than the simple cycle, but its total exergy destruction is approximately 46.6% higher. Its thermal efficiency and second-law efficiency are also lower, while its SFC is slightly higher.

The comparison highlights the trade-off between specific power output and the other thermodynamic performance criteria. Although the intercooled cycle delivers greater net work, the simple cycle achieves the higher decision index because of its more favorable combined performance across efficiency, fuel consumption, and exergy destruction under the adopted formulation. This result does not imply that the simple cycle is preferable for every application: where maximum specific power is the dominant requirement, the intercooled configuration may

be more attractive. The decision-index result should therefore be interpreted as a balanced selection under the specified criteria and equal weighting, and considered alongside the Pareto-optimal analysis presented in the following section.

4.10 Pareto-Optimal Analysis

Pareto analysis was performed to identify feasible operating conditions that provide different compromises among four thermodynamic objectives: maximizing net work output, maximizing thermal efficiency, maximizing second-law efficiency, and minimizing total exergy destruction. Unlike the equal-weight decision index, Pareto analysis does not combine these objectives into a single score. An operating point is Pareto-optimal if no other feasible operating point can improve at least one objective without worsening at least one of the other objectives. The analysis therefore identifies a set of non-dominated operating conditions rather than a single universally optimal solution.

Figure 12 presents the Pareto-optimal operating points for the simple and two-stage intercooled Brayton cycles. The distribution of the solutions illustrates the trade-off between useful work output and thermodynamic irreversibility. Operating conditions that deliver high net work do not necessarily minimize total exergy destruction. Similarly, conditions favorable for thermal or second-law efficiency may not provide the maximum net work. The Pareto front is therefore useful for distinguishing operating conditions according to their relative thermodynamic advantages.

For the simple Brayton cycle, the maximum net work within the investigated operating domain is 459.00 kJ/kg of inlet air, obtained at an ambient temperature of 15 °C, a turbine inlet temperature of 1400 °C, and a nominal compressor pressure ratio of 16. In contrast, the minimum total exergy destruction is approximately 160.94 kJ/kg of inlet air and occurs at a different operating condition: an ambient temperature of 40 °C, a turbine inlet temperature of 800 °C, and a nominal pressure ratio of 28. The difference between these operating points demonstrates that maximizing power output and minimizing thermodynamic irreversibility represent distinct optimization objectives.

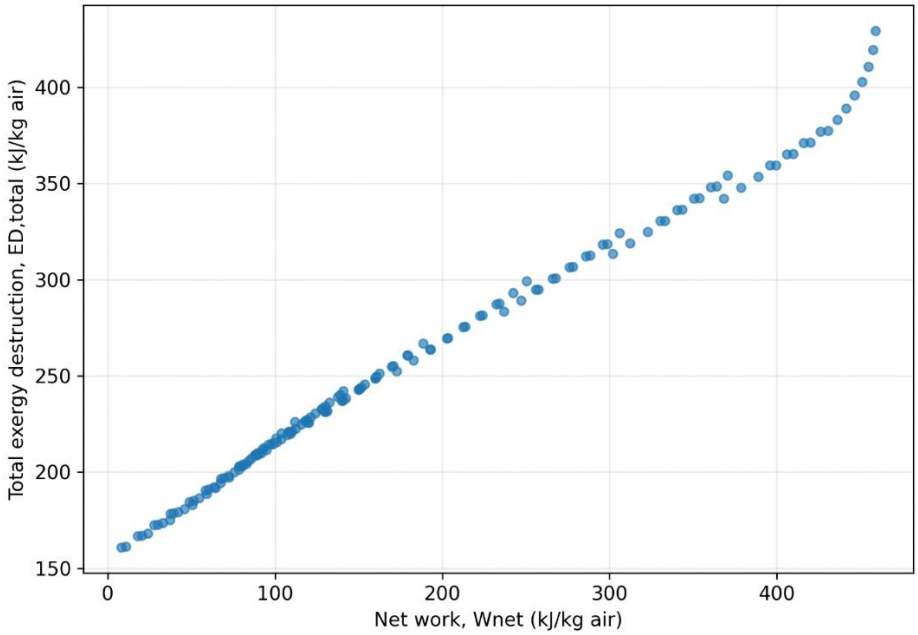
For the two-stage intercooled Brayton cycle, the maximum net work reaches 532.91 kJ/kg of inlet air at an ambient temperature of 15 °C, a turbine inlet temperature of 1400 °C, and a nominal pressure ratio of 28. This is approximately 16.1% higher than the maximum net work obtained with the simple cycle. However, the minimum total exergy destruction for the intercooled configuration is approximately 344.70 kJ/kg of inlet air, occurring at an ambient temperature of

45 °C, a turbine inlet temperature of 800 °C, and a nominal pressure ratio of 30. Thus, the intercooled cycle offers a higher maximum specific work output within the investigated domain, but it does not provide a corresponding reduction in minimum total exergy destruction.

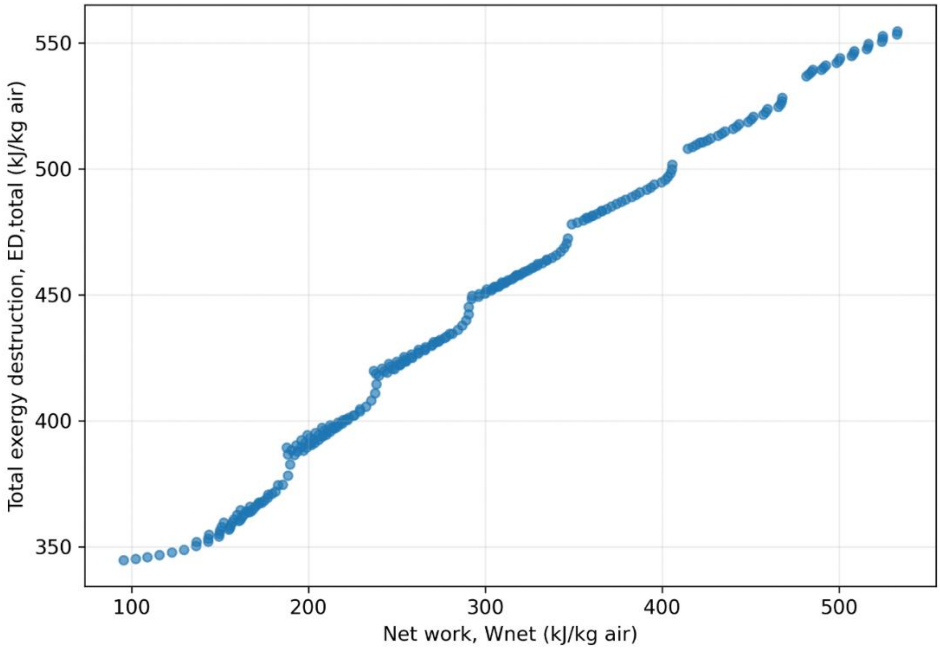
The Pareto results also emphasize the distinction between efficiency and absolute exergy destruction. Thermal and second-law efficiencies describe performance relative to the relevant energy or exergy input, whereas total exergy destruction quantifies the magnitude of the work potential destroyed by irreversible processes. Consequently, a condition with relatively high efficiency can still exhibit substantial absolute exergy destruction. The preferred operating point depends on which performance measure is prioritized.

The component-wise analysis provides further context for these trends. Intercooling reduces the compression work and the exergy destruction associated with each compressor stage, but it also introduces an additional irreversible heat-transfer process in the intercooler. The combustor remains the dominant contributor to component-wise exergy destruction in both configurations. The overall effect of intercooling must therefore be assessed at the cycle level rather than inferred solely from the reduction in compressor work or from the performance of an individual component.

Overall, the Pareto analysis shows that no single operating condition simultaneously optimizes all four objectives considered in this study. For applications prioritizing high specific power output, the high-net-work region of the intercooled cycle is attractive. Where lower total exergy destruction and higher efficiency are more important, other operating points may be preferable. The Pareto front provides a set of candidate operating conditions from which a final operating point can be selected according to the requirements of the intended application and the relative importance assigned to the competing thermodynamic objectives.



(a) Simple Brayton Cycle



(b) Two-stage intercooled Brayton cycle.

Figure 12. Pareto-optimal operating points for (a) the simple Brayton cycle and (b) the two-stage intercooled Brayton cycle, considering net work output, thermal efficiency, second-law efficiency, and total exergy destruction.

4.11 Comparison of Simple and Intercooled Cycles

Table 3. Comparison of principal thermodynamic performance indicators of the simple and two-stage intercooled Brayton cycles over the investigated operating range.

Performance indicator	Simple Brayton cycle	Two-stage intercooled Brayton cycle
Maximum specific net work (kJ/kg air)	459.004	532.909
Pressure ratio at maximum specific net work	16	28
TIT at maximum specific net work (°C)	1400	1400
Ambient temperature at maximum network (°C)	15	15
Maximum thermal efficiency (%)	41.850	40.594
Maximum second-law efficiency (%)	40.366	39.156
Minimum SFC (kg/kWh)	0.17204	0.17737
Minimum total exergy destruction (kJ/kg air)	160.936	344.700
Minimum total entropy generation (kJ/kg K)	0.539784	1.156130
Maximum decision index	0.860916	0.804842

The intercooled Brayton cycle achieves a maximum net work output of 532.91 kJ/kg of inlet air, compared with 459.00 kJ/kg of inlet air for the simple cycle. This corresponds to an increase of approximately 16.1%. The pressure ratio associated with maximum net work shifts from 16 for the simple cycle to 28 for the intercooled cycle, with both maxima occurring at an ambient temperature of 15 °C and a turbine inlet temperature of 1400 °C. This result demonstrates the ability of intercooling to reduce the compression-work requirement and increase the net work available from the cycle under the investigated conditions.

Higher net work output does not, however, imply higher efficiency. The maximum thermal efficiencies are 41.85% for the simple cycle and 40.59% for the intercooled cycle. Similarly, the maximum second-law efficiencies are 40.37% and 39.16%, respectively. The simple cycle also achieves a lower minimum specific fuel consumption, 0.17204 kg/kWh, compared with 0.17737

kg/kWh for the intercooled cycle. These results indicate that the specific-power advantage of intercooling is not accompanied by an improvement in the maximum thermal efficiency, maximum second-law efficiency, or minimum specific fuel consumption within the investigated operating domain.

The exergy results reveal a further difference between the configurations. The minimum total exergy destruction is 160.936 kJ/kg of inlet air for the simple cycle and 344.700 kJ/kg of inlet air for the intercooled cycle. These values represent the respective minima across the investigated operating conditions and should not be interpreted as exergy destruction at the maximum-net-work points. The component-wise analysis shows that the combustor is the dominant source of exergy destruction in both configurations. In the intercooled cycle, the additional irreversibility associated with the intercooler contributes to the overall thermodynamic trade-off, despite the lower exergy destruction associated with each compressor stage.

The equal-weight decision-index analysis provides an integrated comparison of net work, thermal efficiency, second-law efficiency, specific fuel consumption, and total exergy destruction. The maximum decision index is 0.861 for the simple cycle and 0.805 for the intercooled cycle. These values indicate that the simple cycle provides the more favorable compromise under the selected criteria and equal-weight formulation. They do not imply that the simple cycle is superior for every application, because the intercooled cycle retains a clear advantage when maximum specific net work is the primary objective.

Overall, the comparison identifies two distinct thermodynamic design strategies. The intercooled cycle is attractive when higher specific net work is prioritized, whereas the simple cycle is more favorable under the present modelling assumptions when higher maximum efficiencies, lower minimum specific fuel consumption, and lower minimum total exergy destruction are emphasized. The final selection should therefore be based on the intended application and the relative importance assigned to power output, fuel utilization, efficiency, and thermodynamic irreversibility.

4.12 Integrated Thermodynamic Interpretation

The integrated energy, entropy-generation, exergy-destruction, and multi-objective analyses demonstrate that the simple and two-stage intercooled Brayton cycles offer different thermodynamic advantages within the investigated operating domain. The principal benefit of

intercooling is the increase in specific net work: the maximum net work rises from 459.00 kJ/kg of inlet air for the simple cycle to 532.91 kJ/kg of inlet air for the intercooled cycle, an improvement of approximately 16.1%. The corresponding nominal pressure ratio shifts from 16 to 28, while the ambient temperature and turbine inlet temperature remain at 15 °C and 1400 °C, respectively. This result is consistent with the reduction in compression work achieved by cooling the air between compressor stages.

The increase in specific net work is not accompanied by an improvement in every performance indicator. The maximum thermal efficiency decreases from 41.85% for the simple cycle to 40.59% for the intercooled cycle, while the maximum second-law efficiency decreases from 40.37% to 39.16%. The minimum specific fuel consumption is also lower for the simple cycle. These findings indicate that the reduction in compressor work primarily benefits specific power output under the assumptions adopted in this model; it does not establish a universal efficiency advantage for intercooling.

The component-wise exergy analysis identifies the combustor as the dominant contributor to average component exergy destruction in both configurations. Its reported share is approximately 76.56% for the simple cycle and 75.03% for the intercooled cycle. In the intercooled configuration, each compressor stage has a lower average exergy destruction than the compressor in the simple cycle. However, the intercooler introduces an additional source of irreversibility, and the average combustor exergy destruction is also higher in the intercooled results. The component analysis therefore shows that intercooling redistributes irreversibility among the components rather than eliminating it.

The operating-condition analysis further demonstrates that maximum net work, maximum efficiency, and minimum total exergy destruction do not generally occur at the same operating point. The Pareto analysis represents these competing objectives without combining them into a single score, whereas the equal-weight decision index identifies a balanced operating condition according to the five criteria adopted in this study. The maximum decision index is 0.861 for the simple cycle and 0.805 for the intercooled cycle. This difference reflects the selected criteria and equal weighting; it should not be interpreted as evidence that the simple cycle is superior for every application.

Taken together, the results indicate that cycle selection should depend on the intended operating objective. The intercooled configuration is a candidate when higher specific net work is prioritized, whereas the simple cycle is more favorable within the present model when maximum thermal efficiency, maximum second-law efficiency, lower minimum specific fuel consumption, and lower minimum total exergy destruction are emphasized. These conclusions apply to the specified component assumptions and investigated ranges of nominal pressure ratio, turbine inlet temperature, and ambient temperature. Practical selection would additionally require consideration of intercooler auxiliary power, cooling requirements, equipment cost, and performance under actual plant operating conditions.

4.13 Verification of the Computational Model

The computational model for the simple and two-stage intercooled Brayton cycles was subjected to systematic numerical verification to assess the internal consistency of the calculated thermodynamic results. The verification was performed through independent analytical verification, exergy–entropy consistency checks, component-wise exergy balance checks, overall cycle exergy balance, and physical feasibility checks. These checks were applied to the complete parametric dataset rather than to a single operating condition. First, the ideal Brayton-cycle formulation was verified against the analytical expression for the ideal-gas cycle thermal efficiency,

$$\eta_{th,ideal} = 1 - \frac{1}{PR^{\frac{\gamma-1}{\gamma}}}$$

Where, PR is the compressor pressure ratio and γ is the specific-heat ratio of air. The computational implementation reproduced the analytical ideal Brayton-cycle efficiency with a maximum deviation of approximately 4.19×10^{-14} %, which is effectively zero at the reported numerical precision. This agreement supports the implementation of the ideal-cycle efficiency relation and associated pressure-ratio formulation; however, it does not independently validate the complete combustion, intercooling, or exergy-analysis model.

The consistency of the entropy-generation and exergy-destruction calculations was subsequently verified using the Gouy–Stodola theorem,

$$Ex_D = T_0 S_{gen}$$

Where $T_0 = 298.15\ K$ is the reference-environment temperature. For every investigated operating condition, the component exergy destruction calculated from the exergy balance was consistent with the Gouy–Stodola relation, $T_0 S_{gen}$,. The maximum numerical difference between these two calculations was approximately $1.14 \times 10^{-13}\ kJ/kg$ of inlet air. This negligible difference indicates numerical consistency between the entropy-generation and exergy-destruction calculations within computational precision.

A further consistency check compared the sum of the individual component exergy destructions with the total cycle exergy destruction. For the simple cycle, the sum included the compressor, combustor, and turbine contributions; for the intercooled cycle, it additionally included the low-pressure compressor, intercooler, and high-pressure compressor contributions. The component-wise summation agreed with the reported total exergy destruction, with zero numerical summation error at the reported precision. This check confirms the internal consistency of the component accounting but is not an independent validation of the thermodynamic model.

The complete cycle was also checked using the overall exergy balance,

$$Ex_{fuel} + Ex_{air,in} = W_{net} + Ex_{exhaust} + Ex_{D,total}$$

Where Ex_{fuel} is the chemical exergy supplied with the methane fuel, $Ex_{air,in}$ is the physical exergy of the incoming air, W_{net} is the net cycle work, and $Ex_{exhaust}$ represents the physical and chemical exergy remaining in the exhaust products. The maximum relative exergy-balance error across the entire simulation dataset was approximately $1.16 \times 10^{-13}\ \%$, indicating that the overall exergy balance closed within computational precision.

Physical feasibility checks were also incorporated into the computational procedure. The results showed no negative component exergy-destruction values, confirming that the calculated irreversibilities remain physically meaningful. The computed air-to-fuel ratios were also checked against the stoichiometric requirement for methane combustion. The predicted AFR remained in the range of approximately 33.22–272.15, with no cases below the stoichiometric AFR of 17.12. Thus, all calculated combustion states maintained a non-negative residual oxygen condition.

The complete parametric analysis consisted of 1,274 operating cases covering the investigated ranges of ambient temperature, compressor pressure ratio, and turbine inlet temperature. Only three cases produced negative specific net work, and these occurred in the simple cycle at the low-

TIT/high-ambient-temperature operating region. These cases were retained as physically informative infeasible operating conditions rather than artificially removed from the dataset. The two-stage intercooled configuration remained at positive specific net work throughout the investigated range. Table 4 presents the verification summary for the developed computational model.

Table 4: Verification Summary

Verification criterion	**Result**
Number of simulated cases	1,274
Ideal Brayton analytical verification deviation	4.19×10^{-14} %
Maximum deviation between component ExD and $T_0 S_{gen}$	1.14×10^{-13} kJ/kg air
Maximum component-exergy summation error	0 kJ/kg air
Maximum overall exergy-balance error	1.16×10^{-13} %
Negative component Ex_D cases	0
AFR below stoichiometric value	0
Negative-net-work cases	3
Stoichiometric AFR	17.12
AFR range	33.22–272.15

4.14 Study limitations and future research areas

The present study is based on a steady-state thermodynamic and numerical modelling framework, and the results should therefore be interpreted within the assumptions and operating ranges adopted in the analysis. First, the working fluids are modelled using ideal-gas assumptions, while temperature-dependent specific heat is explicitly considered for the combustion products. Real-gas effects and more detailed thermophysical property variations may influence the predicted performance, particularly under operating conditions outside the investigated range.

Second, the compressor and turbine isentropic efficiencies, combustion efficiency, intercooler effectiveness, and pressure losses are prescribed model parameters. In an actual gas-turbine system, these quantities may vary with load, rotational speed, pressure ratio, temperature, and operating condition. Consequently, the calculated performance represents the response of the

system under the specified component-performance assumptions rather than a direct prediction of a particular commercial gas turbine.

Third, the two-stage intercooled configuration uses a prescribed intercooler effectiveness and pressure loss, and the nominal compressor pressure ratio is divided equally between the low-pressure (LP) and high-pressure (HP) compressor stages. The prescribed intercooler pressure loss affects the actual overall compressor pressure ratio. Alternative stage-pressure-ratio distributions, intercooler designs, and pressure-loss characteristics may therefore produce different optimum operating conditions.

Fourth, the present analysis focuses on thermodynamic performance and does not explicitly include detailed economic analysis, equipment sizing, intercooler auxiliary-power consumption, cooling-water requirements, emissions, mechanical limitations, or transient operating behaviour. These factors may influence the practical selection and implementation of a gas-turbine configuration.

Fifth, the computational model was checked using analytical Brayton-cycle relations, exergy–entropy consistency, component-wise exergy accounting, the overall exergy balance, and combustion-feasibility checks. These procedures establish numerical and thermodynamic consistency within the implemented model; however, they do not constitute experimental validation against measurements from a physical gas-turbine system.

Finally, the conclusions are applicable to the investigated ranges of nominal compressor pressure ratio, turbine inlet temperature, and ambient temperature. The reported performance trends and optimum operating conditions should not be extrapolated directly beyond these ranges without additional analysis and validation.

Future studies should extend the present work by incorporating experimentally measured or plant-validated component performance maps and, where appropriate, non-ideal thermophysical property models. Further investigations should examine unequal compressor-stage pressure-ratio distributions, detailed intercooler design and pressure losses, auxiliary power and cooling-water requirements, and the effects of variable operating conditions. Integrating thermodynamic analysis with economic assessment, environmental impact evaluation, and multi-objective optimization that includes practical design constraints would provide a more comprehensive basis for selecting gas-turbine operating conditions. Experimental or plant-data validation should also be undertaken

to assess the predictive accuracy of the proposed model and strengthen its applicability to practical gas-turbine systems.

Conclusions:

This study demonstrates that the thermodynamic superiority of a gas turbine cycle cannot be assessed from net power output or thermal efficiency alone. By integrating energy analysis, entropy generation, component-wise exergy destruction, operating-feasibility assessment, Pareto analysis, and multi-objective decision evaluation, the study provides a comprehensive basis for comparing simple and two-stage intercooled Brayton cycles. The principal conclusions are as follows:

- Intercooling provides a significant specific-power advantage. The two-stage intercooled cycle achieved a maximum specific net work of 532.91 kJ/kg air, compared with 459.00 kJ/kg air for the simple cycle, corresponding to an improvement of approximately 16.1%. The pressure ratio corresponding to maximum specific net work also shifted toward a higher value with intercooling, from PR=16 for the simple cycle to PR=28 for the intercooled cycle at T_{amb} = 15°C and TIT = 1400°C,.
- Higher specific power does not necessarily result in higher thermal efficiency. The simple Brayton cycle achieved a maximum thermal efficiency of 41.85%, whereas the two-stage intercooled cycle achieved 40.59%. Although intercooling reduces the work required for compression and increases the turbine-cycle specific power, the additional irreversibility associated with the intercooler and the resulting changes in cycle energy interactions prevent a corresponding improvement in maximum thermal efficiency.
- Second-law analysis confirms the power–efficiency trade-off. The maximum second-law efficiencies were 40.37% for the simple cycle and 39.16% for the intercooled cycle. Thus, the configuration delivering the highest specific power does not necessarily provide the highest utilization of the available energy from the thermodynamic viewpoint.
- The combustor is the dominant source of thermodynamic irreversibility. The combustor accounts for approximately 76.6% of the average total exergy destruction in the simple cycle and 75.0% in the intercooled cycle. Within the assumptions and operating conditions considered, the combustor represents the dominant contributor to total exergy destruction.

Consequently, reducing combustion-related irreversibility appears to offer the greatest potential for improving the calculated overall exergy performance.

- Intercooling redistributes rather than eliminates irreversibility. The intercooled configuration reduces the exergy destruction associated with individual compression stages; however, it introduces additional exergy destruction in the intercooler. The average exergy destruction associated with the intercooler is approximately 38.43 kJ/kg air, while the LP and HP compressors each contribute approximately 16.78 kJ/kg air. Consequently, the reduction in compressor irreversibility does not compensate for the additional irreversibility introduced elsewhere in the cycle, resulting in higher total exergy destruction for the intercooled configuration.
- The operating parameters strongly influence the preferred cycle configuration. Compressor pressure ratio, turbine inlet temperature, and ambient temperature jointly affect specific net work, thermal efficiency, entropy generation, and exergy destruction. The maximum-specific net work condition, maximum-efficiency condition, and minimum-exergy-destruction condition occur at different operating points. Hence, no single operating condition can be regarded as universally optimal for all thermodynamic objectives.
- Feasibility analysis identifies an important operating limitation of the simple cycle. Negative-net-work conditions were obtained for selected combinations of high ambient temperature and low turbine inlet temperature in the simple Brayton cycle. In contrast, the two-stage intercooled cycle maintained positive specific net work throughout the investigated operating domain. This suggests a potential operating-range advantage of intercooling under the investigated thermodynamic conditions, although this conclusion should be assessed further using component-level and plant-level constraints.
- Pareto analysis confirms the competing nature of the thermodynamic objectives. The Pareto-optimal solutions demonstrate that maximizing specific net work, maximizing thermal and second-law efficiencies, and minimizing total exergy destruction cannot generally be achieved simultaneously. For example, the simple cycle reaches a maximum specific net work of 459.00 kJ/kg air, whereas its minimum total exergy destruction is only 160.94 kJ/kg air at a different operating condition. Similarly, the intercooled cycle reaches a maximum specific net work of 532.91 kJ/kg air, while its minimum total exergy destruction is 344.70 kJ/kg air at a different condition. Therefore, the Pareto front provides

a more appropriate representation of the competing design objectives than optimization based on a single performance parameter.

- The multi-objective decision index identifies a balanced operating condition. Under the adopted equal-weight formulation incorporating specific net work, thermal efficiency, second-law efficiency, SFC, and total exergy destruction, the maximum decision index was 0.861 for the simple cycle and 0.805 for the intercooled cycle, both occurring at T_{amb} = 15°C, TIT = 1400°C, and PR=30. The higher decision index of the simple cycle reflects its better balance of efficiency, fuel consumption, and exergy destruction, whereas the intercooled cycle provides a substantially higher specific power output. Thus, the preferred configuration depends on the relative importance assigned to power generation and thermodynamic performance.
- The computational model demonstrated numerical and thermodynamic consistency within the adopted formulation. Across the evaluated operating cases, the maximum deviation in the ideal Brayton-cycle analytical check was approximately 4.19×10^{-14} %. The maximum absolute deviation between component exergy destruction and the corresponding $T_0 S_{gen}$ value was approximately 1.14×10^{-13} kJ/kg of inlet air, while the maximum overall exergy-balance error was approximately 1.16×10^{-13} %. The component-wise exergy-summation residual was zero, and no negative component exergy-destruction values or violations of the imposed stoichiometric air–fuel constraint were detected. These results indicate numerical consistency within computational precision; they do not constitute independent experimental validation of the model.

Overall, the results indicate that intercooling provides a significant specific-power benefit within the investigated operating range, but this benefit is accompanied by additional thermodynamic irreversibility and lower maximum thermal and second-law efficiencies under the adopted modelling assumptions. The intercooled configuration is therefore more attractive when higher specific power is prioritized, whereas the simple Brayton cycle is more favorable when thermal efficiency, fuel utilization, and lower total exergy destruction are given greater importance.

The integrated energy–entropy–exergy–feasibility–Pareto framework provides a systematic basis for comparing the two Brayton-cycle configurations and identifying operating regions associated with different thermodynamic objectives. The preferred configuration ultimately depends on the

relative importance assigned to power output, efficiency, fuel consumption, and exergy destruction.

Data Availability Statement

All data used in this study will be made available from the corresponding author upon reasonable request.

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Abbreviations

Symbol	Description	Unit
AFR	Air-fuel ratio	kg/kg
AFR_{st}	Stoichiometric air-fuel ratio	kg/kg
a_i and b_i	Product-gas specific-heat coefficients	—
c_p	Specific heat at constant pressure	kJ kg^{-1} K^{-1}
DI	Multi-objective decision index	—
E_D	Exergy destruction	kJ kg^{-1}
e_{ch}	Specific chemical exergy	kJ kg^{-1}
e_{ph}	Specific physical exergy	kJ kg^{-1}
Ex	Exergy	kJ kg^{-1}
Ex_f	Fuel chemical exergy	kJ kg^{-1}
Ex_g	Total product-gas exergy	kJ kg^{-1}
f	Fuel-air ratio	kg/kg
h	Specific enthalpy	kJ kg^{-1}
LHV	Lower heating value	kJ kg^{-1}
m_a	Air mass	kg
m_f	Fuel mass	kg
m_g	Combustion-product mass	kg
n_i	Moles of species ii	mol
P	Pressure	kPa
P_0	Reference pressure	kPa
PR	Overall compressor pressure ratio	—

PR_{LP}	LP compressor pressure ratio	—
PR_{HP}	HP compressor pressure ratio	—
Q_{IC}	Intercooler heat rejection	kJ
R_a	Specific gas constant of air	kJ kg^{-1} K^{-1}
R_u	Universal gas constant	kJ mol^{-1} K^{-1}
R_{Ex}	Overall exergy-balance residual	kJ kg^{-1}
S_{gen}	Entropy generation	kJ kg^{-1} K^{-1}
SFC	Specific fuel consumption	kg kWh^{-1}
s	Specific entropy	kJ kg^{-1} K^{-1}
T	Temperature	K
T_0	Reference/dead-state temperature	K
T_{amb}	Ambient temperature	°C
TIT	Turbine inlet temperature	°C
W_c	Compressor work	kJ kg^{-1}
W_t	Turbine work	kJ kg^{-1}
W_{net}	Net work output	kJ kg^{-1}
y_i	Mole fraction of species ii	—
γ	Specific heat ratio	—
η_c	Compressor isentropic efficiency	—
η_t	Turbine isentropic efficiency	—
η_{comb}	Combustion efficiency	—
η_{th}	Thermal efficiency	%
η_{II}	Second-law efficiency	%
ε_{IC}	Intercooler effectiveness	—
ΔP_{IC}	Fractional intercooler pressure loss	—
ϕ_j	Component contribution to total exergy destruction	%
ϵ_{Ex}	Relative exergy-balance error	%
δ_{ideal}	Ideal-cycle validation deviation	%
i	Combustion-product species index	—

j	Component index	—
u	Universal-gas-property quantity, where applicable	—

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