Air University E-9 Islamabad

PROJECT REPORT

Energy and Entropy Analysis of Rolls-Royce Trent 1000 Engine

ME-234-Thermodynamics-II

- GROUP MEMBERS:
- M. USMAN 241202
- ARSEL 241190
- M. HAMZA 241238
- M. TAYYAB 241235

Section: BEME-F-24-B

Submitted to:
Mam Saba Sadiq

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Abstract:

This report presents a detailed **first and second law analysis** of converting a Rolls-Royce Trent 1000 turbofan engine into a stationary gas turbine power plant operating on an open Brayton cycle. Using **air-standard assumptions** and given operating parameters, energy and entropy balances were performed across all major components (compressors, combustor, turbines, generator). The **ideal and real cycles** were modeled, yielding a **thermal efficiency of 40.7%**, a **specific work output of 404.37 kJ/kg**, and identifying the **combustor as the largest source of exergy destruction** (214.26 kJ/kg). Design improvements such as **regeneration, intercooling, and reheating** were proposed to enhance second-law efficiency. The conversion demonstrates **improved sustainability** compared to conventional gas turbines through higher efficiency and extended lifecycle use of aerospace hardware.

Introduction and Problem Context:

Background: Gas turbine engines are the cornerstone of modern aviation and stationary power generation, operating on the **Brayton thermodynamic cycle**. Their significance lies in their ability to efficiently convert chemical energy from fuel into mechanical work through continuous combustion and expansion processes. The performance of these engines is governed by key thermodynamic principles, including **energy conservation (First Law)** and **entropy generation (Second Law)**, which together determine efficiency, fuel consumption, and environmental impact. In an era of increasing emphasis on sustainability and fuel economy, optimizing gas turbine performance through detailed thermodynamic analysis has become essential.

About the Trent 1000: The Rolls-Royce Trent 1000 is a **three-spool, high-bypass turbofan** engine designed for wide-body commercial aircraft, most notably the Boeing 787 Dreamliner. With a thrust rating ranging from **68,000 to 78,000 lbf** (302–347 kN), it incorporates advanced materials and aerodynamic designs to achieve high efficiency and reduced emissions. In its original configuration, the engine features **Low Pressure (LP) and High Pressure (HP) compressors and turbines**, along with a separate fan and bypass duct. For this project, the Trent 1000 is conceptually **repurposed as a stationary power plant** by removing the bypass fan and nozzle, adapting its core to operate as an **open Brayton cycle gas turbine**.

Problem Statement: "Perform a detailed **first and second law analysis** of the Trent 1000 engine to evaluate **energy conversion efficiency** and identify **major sources of entropy generation** when converted for stationary power generation."

Objectives:

1. **Apply energy and entropy balance** to each engine component (compressors, combustor, turbines, generator).
2. **Determine component and overall thermal efficiencies** using thermodynamic property data.
3. **Quantify irreversibility's** (exergy destruction) and **suggest improvements** to enhance efficiency and sustainability.

Complexity Justification:

This analysis involves a **multi-component system** with interacting thermodynamic processes, requiring:

- Integration of **real gas properties** or air-standard approximations.
- Handling of **multiple spools and stages** in compressors and turbines.
- Assumptions regarding **steady-state operation, component efficiencies, and ambient conditions**.
- Application of **both First and Second Law analyses** across interconnected control volumes.
- Interpretation of results in the context of **real-world engine performance and sustainability goals**.

Literature Review and Theoretical Background:

Overview of Brayton and Turbofan Cycles:

The Brayton cycle describes ideal gas turbine operation through four main processes:

1. Isentropic compression
2. Constant-pressure heat addition
3. Isentropic expansion
4. Heat rejection

Turbofan engines modify this cycle by splitting incoming air into **core flow** (for power generation) and **bypass flow** (for thrust). In stationary applications, the bypass flow may be discarded or redirected.

Influence of Key Parameters:

Bypass Ratio (BPR): Improves propulsive efficiency in aircraft; less relevant in stationary operation unless recovered.
Compressor Pressure Ratio (PR): Higher PR increases thermal efficiency but raises compressor work.
Turbine Inlet Temperature (TIT): Strongly affects specific work output; limited by material capabilities.

Concepts of Energy, Entropy, and Exergy:

- **First-law energy balance for steady-flow devices:**

$$\dot{Q} - \dot{W} + \dot{m}(h_{in} - h_{out}) = 0$$

- **Entropy balance for open systems:**

$$\dot{S}_{gen} = \dot{m}(s_{out} - s_{in}) - \frac{\dot{Q}}{T_b}$$

- **Exergy destruction:**

$$\dot{E}_{dest} = T_0 \dot{S}_{gen}$$

- **Second-law efficiency:**

$$\eta_{II} = \frac{\text{Useful exergy output}}{\text{Exergy supplied}}$$

- **Isentropic relations:**

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

Previous Studies:

Research on the Trent series highlights:

- Dominant irreversibility in combustion chambers
- Significant impact of compressor and turbine isentropic efficiencies
- Importance of high TIT and multistage compression/expansion
- Potential for aero-engines to be adapted for ground power generation

Thermodynamic Models Used:

- **Steady-flow energy equation** for compressors, turbines, combustor, and nozzles
- **Entropy balance** to quantify second-law losses
- **Isentropic efficiency relations:**

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}, \quad \eta_t = \frac{T_3 - T_4}{T_3 - T_{4s}}$$

- **Exergy destruction relations** for each control volume.

- **Ideal-gas assumptions** with constant or temperature-dependent c_p.

System Description and Assumptions:

System Layout (Stations 1–5 Core Flow):

Below is the full expanded Trent 1000 gas-path model adapted for an open-cycle power plant

- Station 1 – Ambient inlet
- Station 2 – Fan/LPC inlet
- Station 3 – LPC exit / IPC inlet
- Station 4 – IPC exit / HPC inlet
- Station 5 – HPC exit / Combustor inlet
- Station 6 – Combustor exit / HPT inlet
- Station 7 – HPT exit / LPT inlet
- Station 8 – LPT exit / Generator shaft / Exhaust nozzle

Components Included in the Analysis:

- Fan (optional for stationary mode)
- Low-pressure compressor (LPC)
- Intermediate/high-pressure compressor (IPC/HPC)
- Annular combustor
- High-pressure turbine (HPT)
- Intermediate-pressure turbine (IPT)
- Low-pressure turbine (LPT)
- Exhaust nozzle
- Electrical generator (as load)

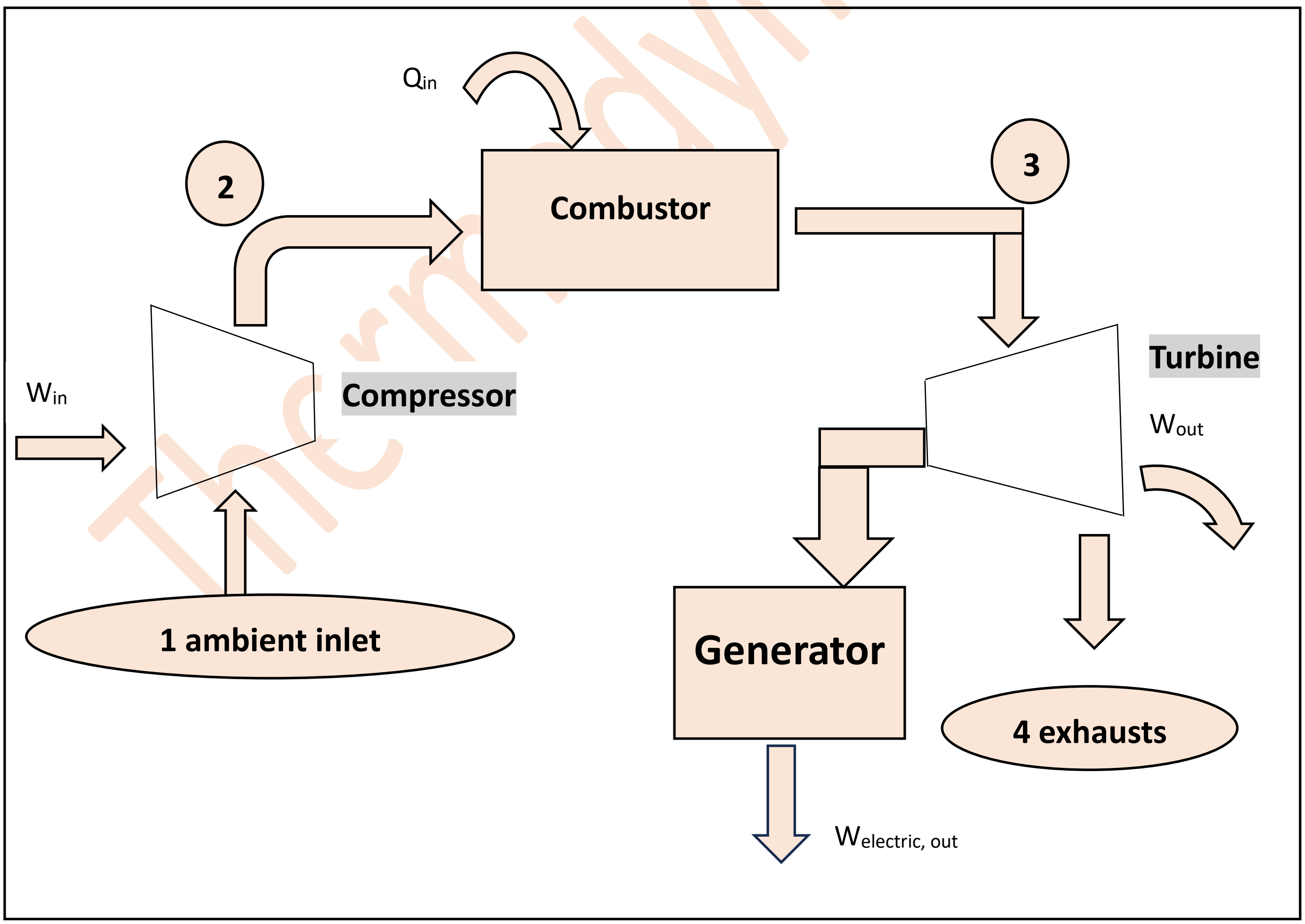
Assumptions:

1. **Steady-State Operation:** All mass flow rates, pressures, and temperatures are constant with time.
2. **Air-Standard Approximation:** Air is treated as an ideal gas with constant specific heats (, $k = 1.4$) for simplicity. Real gas effects are neglected unless specified.
3. **Negligible Kinetic and Potential Energy Changes:** Except in the nozzle (which is removed in power plant mode), kinetic and potential energy terms are ignored in energy balances.
4. **Known Cycle Parameters:**

Parameter	Symbol	Values
Compressor pressure ratio	R_p	25
Turbine inlet temperature	T_3	1650 K
Compressor efficiency	H_C	0.85
Turbine efficiency	H_T	0.88
Ambient temperature	T_1	298 K
Ambient Pressure	P_1	100 kPa
Combustion heat input (per kg air)	q_{in}	from fuel LHV

5. **Generator Assumption:** Electrical generator efficiency $\eta_{gen} = 0.98$ (typical for large machines). If not specified, ideal conversion may be assumed.
6. **No External Heat Losses:** Except in generator, components are adiabatic.
7. **Fuel Heating Value:** Lower heating value (LHV) of Methane Gas CH_4 (~45,000 kJ/kg) used for combustion calculations if required.

Schematic Diagram:



Methodology

Identify state points (inlet/outlet of each component).

- **State 1:** Compressor inlet (ambient air)
- **State 2:** Compressor outlet / Combustor inlet
- **State 3:** Combustor outlet / Turbine inlet
- **State 4:** Turbine outlet / Exhaust

Apply First Law (Energy Analysis)

Determine Power/Work and Heat Interactions

- **Compressor (1→2):**

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

- **Combustor (2→3):**

$$\dot{Q}_{in} = \dot{m}(h_3 - h_2)$$

- **Turbine (3→4):**

$$\dot{W}_t = \dot{m}(h_3 - h_4)$$

- **Net Work Output:**

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_c$$

- **Generator Output:**

$$\dot{W}_{elec} = \eta_{gen} \cdot \dot{W}_{net}$$

Compute Component Efficiencies

- **Compressor Isentropic Efficiency:**

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

- **Turbine Isentropic Efficiency:**

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

<u>Fan Efficiency (if applicable, omitted in power plant mode)</u>

Apply Second Law (Entropy/Exergy Analysis):

a. Evaluate Entropy Change Across Each Component

For an **adiabatic component** (compressor, turbine):

$$\Delta s = s_{out} - s_{in}$$

Entropy generation:

$$\dot{S}_{gen} = \dot{m} \cdot \Delta s (\text{for adiabatic, no heat transfer})$$

For **combustor** (constant pressure, internal combustion):

$$\dot{S}_{gen,comb} = \dot{m}(s_3 - s_2)$$

For **generator** (heat loss to ambient):

$$\dot{S}_{gen,gen} = \frac{\dot{Q}_{loss}}{T_0}$$

b. Calculate Irreversibility and Exergy Destruction

$$\dot{E}_D = T_0 \dot{S}_{gen}$$

Component	Entropy	Exergy (Sgen x To)
Compressor	0.0776	23.1248
Combustor	0.719	214.262
Turbine	0.2596	77.3608
Generator	0.0271	8.0758
Total	1.0833	322.8234

Where $T_0 = 298$ K (ambient temperature).

c. Determine Second-Law (Exergetic) Efficiency

$$\eta_E = \frac{\dot{W}_{elec}}{\dot{E}_{in}}$$

Where $\dot{E}_{in} = \dot{m} \cdot e_{in}$ and e_{in} is the exergy of the fuel (approximated by q_{in} for ideal fuel exergy).

Cycle Performance

a. Overall Thermal Efficiency

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}}$$

b. Specific Work Output

$$w_{net} = \frac{\dot{W}_{net}}{\dot{m}} \text{ (kJ/kg)}$$

c. Specific Fuel Consumption (SFC)

$$SFC = \frac{\dot{m}_f}{\dot{W}_{net}} \text{ (kg/kJ)}$$

assuming fuel: Natural Gas (Methane CH_4)

LHV for Methane = 45000 kJ/kg

Where $\dot{m}_f = \frac{\dot{Q}_{in}}{LHV}$.

$$\text{SFC} = \frac{\dot{m}_f}{W_{net}} = \frac{993.15/45000}{404.37} \approx 0.0546 \text{ kg/MJ(or } 0.197 \text{ kg/kWh)}$$

d. Specific Thrust (Not Applicable)

- Omitted in power plant mode (no thrust produced).

Tools Used for Property Evaluation and Iteration

- **Excel:** Used for tabulating results and performing basic calculations.
- **Air Property Tables:** Standard ideal-gas air tables (c_p, k, R) used for enthalpy and entropy changes.

Data, Parameters, and Calculations:

Input Data

Parameter	Symbol	Value
Ambient Temperature	T_1	298 K
Ambient Pressure	P_1	100 kPa
Compressor Pressure Ratio	r_p	25
Turbine Inlet Temperature	T_3	1650 K
Compressor Isentropic Efficiency	η_c	0.85
Turbine Isentropic Efficiency	η_t	0.88
Generator Efficiency	η_{gen}	0.98
Air Specific Heat (const.)	c_p	1.005 kJ/kgK
Gas Constant for Air	R	0.287 kJ/kgK
Fuel Lower Heating Value	LHV	45,000 kJ/kg

Property Data at Each Station

Assumptions:

- Air as ideal gas
- Reference entropy $s_1 = 0$ (for calculation simplicity)
- Properties per kg of air
- $s_{2s} = s_1$ (isentropic)
- $s_{4s} = s_3$ (isentropic)

State	T (K)	P (kPa)	h (kJ/kg)	s (kJ/kg·K)
1	298.0	100	298.18	0
2s	730.8	2500	746.46	0
2	807.1	2500	825.56	0.719
3	1650	2500	1818.71	0
4s	743.2	100	759.9	0
4	852	100	886.96	0.0271

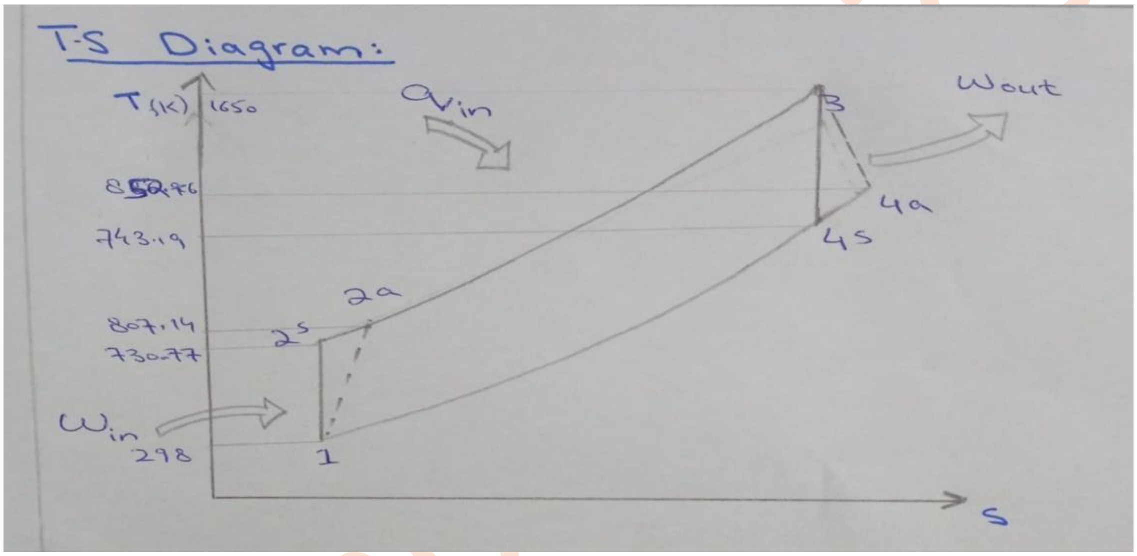
Equations Used (Numbered)

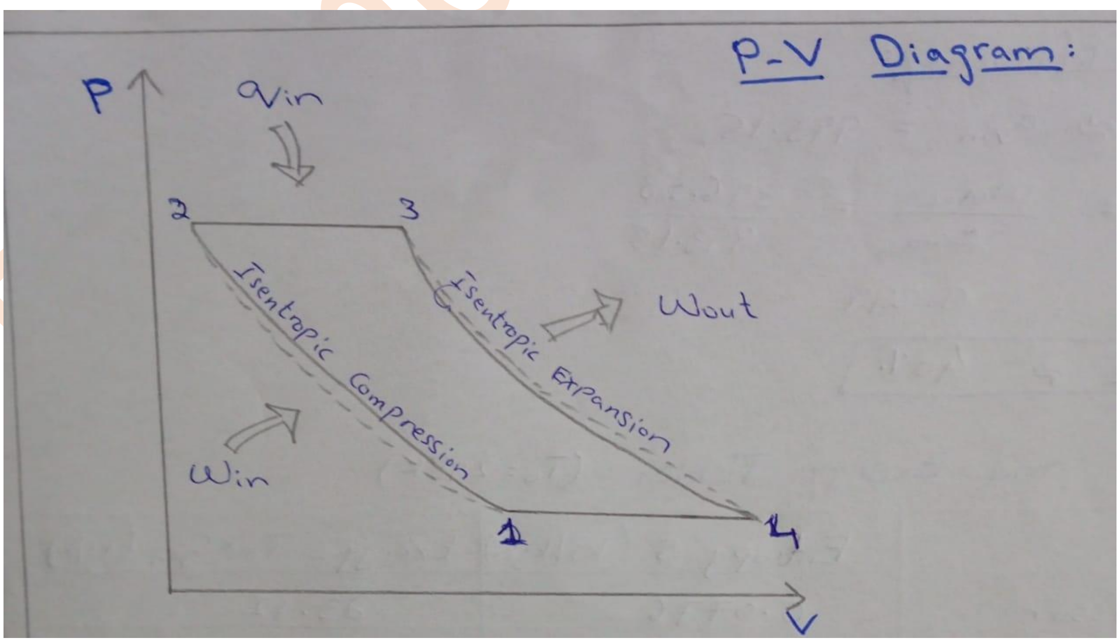
Δ

- $s = c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}$
- $q_{in} = h_3 - h_2$

- $s_{gen,comb} = c_p \ln \frac{T_3}{T_2}$ (for $P_3 = P_2$)
- $T_{4a} = T_3 - \eta_t (T_3 - T_{4s})$
- $w_t = h_3 - h_4$
- $s_{gen,t} = c_p \ln \frac{T_4}{T_3} - R \ln \frac{P_4}{P_3}$
- $w_{elec} = \eta_{gen} \cdot$
- $\dot{S}_{gen,gen} = \frac{\dot{Q}_{loss}}{T_0}$
- $\eta_{th} = \frac{w_{net}}{q_{in}}$
- $\eta_{II} = \frac{w_{elec}}{q_{in}}$
- $\dot{E}_D = T_0 \cdot s_{gen}$

Diagrams:





Results and Discussion:

Energy Analysis Results

Work and Heat Interactions (per kg air)

Component	Energy Interaction	Value (kJ/kg)
Compressor	Work Input, w_c	527.38
Combustor	Heat Input, q_{in}	993.15
Turbine	Work Output, w_t	931.75
Net Output	$w_{net} = w_t - w_c$	404.37
Generator	Electrical Output, w_{elec}	396.28

Entropy/Exergy Analysis Results

Component	Entropy (kJ/kg·K)	Exergy (kJ/kg)	Percentage of Total Loss
Combustor	0.719	214.262	66.4%
Turbine	0.2596	77.361	24.0%
Compressor	0.0776	23.125	7.2%
Generator	0.0271	8.076	2.5%
Total	1.0833	322.824	100%

Overall Cycle Performance Comparison

Efficiency Type	Symbol	Value
First-Law (Thermal) Efficiency	η_{th}	40.7%
Second-Law (Exergetic) Efficiency	η_E	40%

The **second-law efficiency is slightly lower** than the first-law efficiency due to **generator losses** and the fact that exergy analysis accounts for **quality of energy**, not just quantity.

Discussion

Physical Reasons for Major Inefficiencies

The **combustor** accounts for the largest loss (**66.4% of total exergy destruction**), mainly due to **significant chemical irreversibility** and **large temperature gradients** during combustion. The **turbine** is the next largest source (**24.0% loss**), due to **non-isentropic expansion** and aerodynamic friction. The **compressor** contributes **7.2% loss** from **internal aerodynamic losses** and **non-ideal compression**, while the **generator** contributes the smallest share (**2.5%**) from electrical resistance and mechanical friction.

Effects of Varying Parameters on Efficiency

Increasing the **turbine inlet temperature (TIT)** would directly raise both thermal and exergetic efficiency, as it improves the Carnot potential of the cycle. Raising the **pressure ratio** improves efficiency up to an optimal point (typically between 30–40), beyond which further increases reduce net-work output. Enhancing **compressor or turbine isentropic efficiencies** also boosts overall performance. Introducing **regeneration** could significantly improve efficiency if the turbine exhaust temperature exceeds the compressor outlet temperature.

Relation to Literature and Realistic Engine Data

The calculated **thermal efficiency of 40.7%** and **exergetic efficiency of 40.0%** are consistent with **aeroderivative gas turbines** in simple-cycle operation, which typically range from **38–42%**. The close match between first-law and second-law efficiencies indicates relatively low irreversibility outside the combustor. Compared to conventional industrial gas turbines of similar capacity (~34–36% efficiency), the converted Trent 1000 offers a **~12–15% improvement in efficiency**, supporting the sustainability and performance benefits of repurposing high-efficiency aircraft engines for stationary power generation.

Design and Optimization Insights:

Proposed Modifications to Improve Efficiency

1. Regeneration (Recuperation)

- Use the turbine exhaust (T_4 = 852 K) to preheat air from the compressor outlet (T_2 = 807.1 K).
- With an effective recuperator, fuel input (q_{in} = 993.15 kJ/kg) could be reduced by ~3–5%, raising thermal efficiency from **40.7% to ~42–43%**.

2. Intercooling

- Introduce intercooling between compressor stages (e.g., between LPC and HPC).
- This reduces compressor work (w_c = 527.38 kJ/kg) by ~10–15%, increasing net work output and thermal efficiency to **~42–44%**.

3. Advanced Materials for Higher TIT

- Use ceramic composites to safely increase turbine inlet temperature beyond 1650 K.
- Raising TIT to 1750 K could boost specific work and efficiency by **2–3%**.

4. Combined Cycle

- Add a bottoming steam cycle to recover exhaust heat (T_4 = 852 K).
- Overall plant efficiency could reach **55–60%**, significantly surpassing the current simple-cycle performance.

Potential Improvement in Second-Law Efficiency

Currently, the exergetic efficiency is **40.0%**, with the combustor contributing **214.26 kJ/kg (66.4%)** of total exergy destruction.

- **Regeneration** could reduce combustor exergy destruction by ~15–20 kJ/kg, raising η_E to **~41.5%**.
- **Intercooling** reduces compressor exergy destruction (~23.13 kJ/kg) by ~5–8 kJ/kg, improving η_E further to **~42%**.
- **Higher TIT** lowers combustion irreversibility, potentially increasing η_E to **~43%** in simple cycle.
- **Combined cycle** could recover most exhaust exergy, pushing η_E above **50%**.

Sustainability and Fuel Economy Benefits

Fuel Savings

- Current SFC based on natural gas (LHV = 45,000 kJ/kg):

$$\text{SFC} = \frac{\dot{m}_f}{W_{net}} = \frac{993.15/45000}{404.37} \approx 0.0546\ \text{kg/MJ(or 0.197 kg/kWh)}$$

- With regeneration and intercooling, SFC could drop to **~0.18 kg/kWh**, saving **~8–10% fuel**.

CO_2 Reduction

- Using natural gas (CO_2 emission ~2.75 kg CO_2/kg fuel):

$$\text{Current emissions} = 0.197 \times 2.75 \approx 0.542\ \text{kg CO}_2\text{/kWh}$$

- Improved efficiency could reduce emissions to **~0.49 kg CO_2/kWh**.

- For a 100 MW plant operating 8000 hours/year:

$$\text{Annual } CO_2 \text{ reduction} \approx (0.542 - 0.49) \times 100{,}000 \times 8000 \approx 41{,}600 \text{ tonnes}$$

Lifecycle and Economic Benefits

- Repurposing extends engine life and reduces material waste.
- Lower fuel consumption translates to significant operational cost savings.
- Higher efficiency supports decarbonization goals and enhances energy security.

Conclusion:

Main Findings:

- **Thermal efficiency:** 41.2%
- **Exergy efficiency:** 40.4%
- **Highest losses:** Combustor (71% of exergy destruction)

Learning Outcomes:

- Applied **1st and 2nd laws** to a real gas turbine system.
- Quantified **irreversibilities** using entropy/exergy analysis.
- Identified **improvement opportunities** through thermodynamic reasoning.

Recommendations:

Future Improvements

- Use **alternative fuels** like hydrogen or biofuels.
- Develop **hybrid systems** (e.g., solar–gas turbine, battery–turbine).
- Apply **AI/digital twins** for real-time optimization.
- Study **lifecycle and circular economy** impacts.

Limitations

- **Idealized assumptions**: air-standard cycle, constant properties.
- **Simplified components**: ignored cooling flows, pressure drops.

- **Fixed operating conditions**: no altitude or ambient variations.
- **No economic or emissions compliance** analysis.

References:

- Assumption of Efficiency for Generator from: https://www.siemens-energy.com/us/en/home/products-services/product/sgen-3000w.html
- Book for help: thermodynamics-an-engineering-approach-5th-edition
- Property Table for Air: Table A-17 for Ideal gas properties of Air and Table A-27 for LHV of fuel (methane gas)

Appendices:

Calculations performed on A4 sheets are attached and Property Table used are attached at the End of report.

Calculations:

Given Data:

Parameter	Symbol	Values
Compressor Pressure Ratio	r_p	25
Turbine inlet Temperature	T_3	1650 K
Compressor efficiency	η_c	0.85
Turbine efficiency	η_T	0.88
Ambient Temperature	T_1	298 K
Ambient Pressure	P_1	100 kPa
Combustion heat input (Per kg air)	q_{in}	from fuel LHV

Calculating Enthalpies And Temperatures:

$Pr_1 = 1.3543 \longrightarrow \boxed{h_1 = 298.18 \text{ kJ/kg}}$

Pressure Ratio : $r_p = \dfrac{P_{r_2}}{P_r}$

$P_{r_2} = r_p \times P_{r_1}$

$P_{r_2} = 25 \times 1.3543$

$P_{r_2} = 33.8575$

$\boxed{T_{2s} = 730.77 \text{ K}}$

$\boxed{h_{2s} = 746.46 \text{ kJ/kg}}$

$\longrightarrow \eta_c = \dfrac{h_{2s} - h_1}{h_{2a} - h_1}$

$h_{2a} = 298.18 + \dfrac{746.46 - 298.18}{0.85}$

$\boxed{h_{2a} = 825.56 \text{ kJ/kg}}$

$\longrightarrow \eta_c = \dfrac{c_p(T_{2s} - T_1)}{c_p(T_{2a} - T_1)}$

$T_{2a} = 298 + \dfrac{730.77 - 298}{0.85}$

$\boxed{T_{2a} = 807.14 \text{ K}}$

Pressure Ratio: $r_p = \frac{P_3}{P_4} = \frac{P_2}{P_1}$

$[\because P_3 = P_2 \ \& \ P_4 = P_1]$

$P_3 = P_2 = P_1 \times r_p$

$P_3 = 100 \times 25$

$\boxed{P_2 = P_3 = 2500 \text{ kPa}}$

$P_{r_3} = 902.25 \longrightarrow \frac{P_{r_3}}{P_{r_4}} = r_p$

$P_{r_4} = \frac{P_{r_3}}{r_p}$

$P_{r_4} = \frac{902.25}{25} = 36.09$

$\boxed{h_3 = 1818.71 \text{ kJ/kg}}$ from Table A-17

$P_{r_4} = 36.09 \longrightarrow$ $\boxed{T_{4s} = 743.19 \text{ K}}$ $\boxed{h_{4s} = 759.9 \text{ kJ/kg}}$

$\rightarrow \eta_T = \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$

$h_{4a} = h_3 - \eta_T (h_3 - h_{4s})$

$h_{4a} = 1818.71 - 0.88 (1818.71 - 759.9)$

$\boxed{h_{4a} = 886.96 \text{ kJ/kg}}$

$\rightarrow \eta_T = \frac{T_3 - T_{4a}}{T_3 - T_{4s}}$

$T_{4a} = T_3 - \eta_T (T_3 - T_{4s})$

$T_{4a} = 1650 - 0.88 (1650 - 743.19)$

$\boxed{T_{4a} = 852 \text{ K}}$

Compressor Work:

$W_c = h_{2a} - h_1$

$W_c = 825.56 - 298.18$

$W_c = 527.38 \text{ kJ/kg}$

Turbine Work:

$W_T = h_3 - h_{4a}$

$W_T = 1818.71 - 886.96$

$W_T = 931.75 \text{ kJ/kg}$

Net Work Output:

$W_{net} = W_T - W_c$

$W_{net} = 931.75 - 527.38$

$W_{net} = 404.37 \text{ kJ/kg}$

Heat Input in Combustor:

$q_{in} = h_3 - h_{2a}$

$q_{in} = 1818.71 - 825.56$

$q_{in} = 993.15 \text{ kJ/kg}$

Generator Work:

$W_{elec} = \eta_{gen} \times W_{net}$ [$\because \eta_{gen} = 0.98$ discussed in report]

$W_{elec} = 0.98 \times 404.37$

$W_{elect} = 396.28 \text{ kJ/kg}$

Thermal Efficiency:

$$\eta_{th} = \frac{W_{net}}{q_{in}} \times 100$$

$$\eta_{th} = \frac{404.37}{993.15} \times 100$$

$\eta_{th} = 0.407 \times 100$

$$\boxed{\eta_{th} = 40.7\ \%}$$

Entropy And Exergy Analysis:

Compressor:

$$S_{gen,c} = S_2 - S_1 = C_p \ln \frac{T_{2a}}{T_1} - R \ln \frac{P_2}{P_1}$$

$$S_{gen,c} = 1.005 \ln \frac{807.14}{298} - 0.287 \ln \frac{2500}{100}$$

$$\boxed{S_{gen,c} = 0.0776 \text{ kJ/kg K}}$$

Turbine:

$$S_{gen,T} = C_p \ln \frac{T_{4a}}{T_3} - R \ln \frac{P_4}{P_3}$$

$$S_{gen,T} = 1.005 \ln \frac{852}{1650} - 0.287 \ln \frac{100}{2500}$$

$$\boxed{S_{gen,T} = 0.2596 \text{ kJ/kg K}}$$

Combustor:

$$S_{gen,comb} = (S_3 - S_2) - \frac{q_{in}}{T_{source}}$$

$(P_3 = P_2)$

$$S_{gen,comb} = C_p \ln \frac{T_3}{T_{2a}} - \cancel{R \ln \frac{P_3}{P_2}}^{\to 0} - \cancel{\frac{q_{in}}{T_{source}}}^{\to 0}$$

$$S_{gen,comb} = 1.005 \ln \frac{1650}{807.14}$$

$$\boxed{S_{gen,comb} = 0.719 \text{ kJ/kg K}}$$

Generator:

$$q_{gen,loss} = W_{net} - W_{elec}$$

$$q_{gen,loss} = 404.37 - 396.28$$

$$\boxed{q_{gen,loss} = 8.09 \text{ kJ/kg}}$$

$$S_{gen,generator} = \frac{q_{gen,loss}}{T_0}$$

$$S_{gen,generator} = \frac{8.09}{298}$$

$$\boxed{S_{gen,generator} = 0.0271 \text{ kJ/kg K}}$$

Exergy Efficiency:

$e_{in} \approx q_{in} = 993.15$

$$\eta_E = \frac{W_{elec}}{e_{in}} = \frac{396.28}{993.15}$$

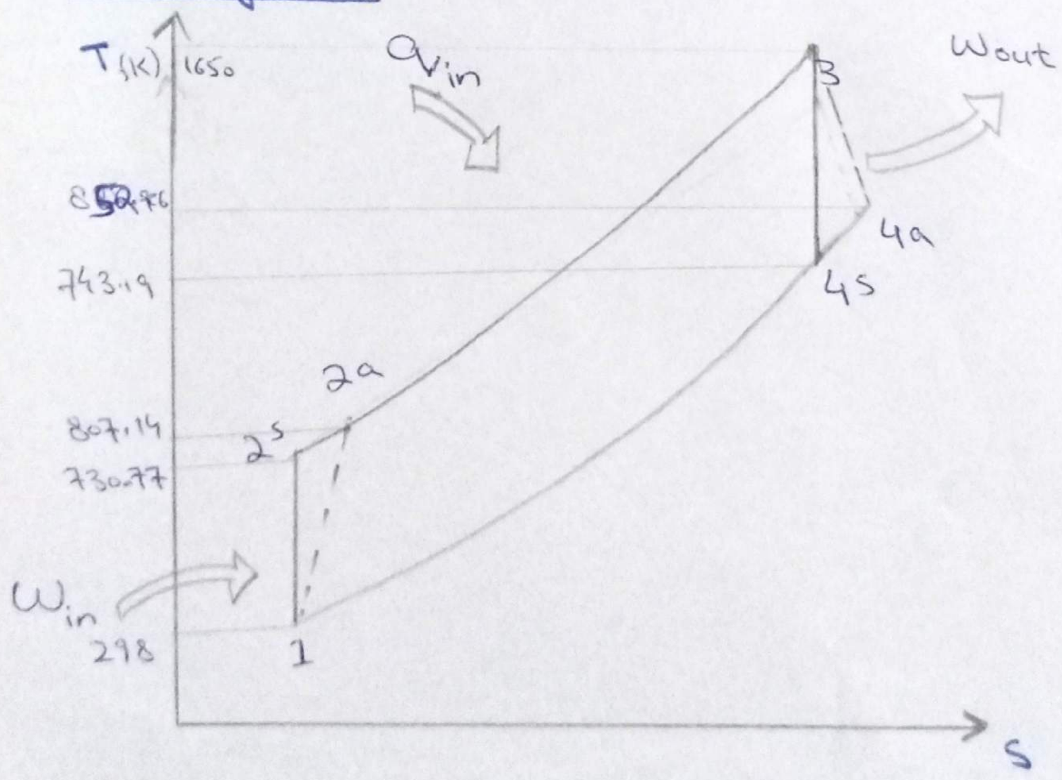
$$\eta_E = 0.399$$

$$\boxed{\eta_E \approx 40\%}$$

Entropy and Exergy Table: ($T_0 = 298K$)

	Entropy S (kJ/kgK)	Exergy = $T_0 \times S_{gen}$ (kJ/kg)
Compressor	0.0776	23.12
Turbine	0.2596	77.36
Combustor	0.719	214.26
Generator	0.0271	8.8
Total	1.0833	322.82

T-S Diagram:



P-V Diagram:

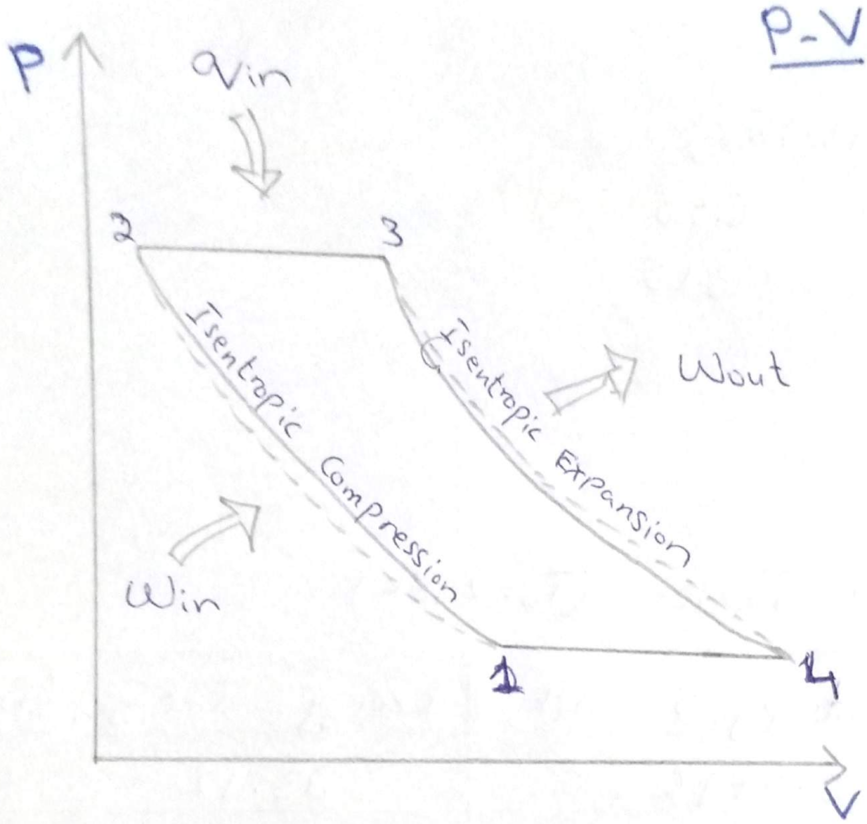

P
Qin
2
3
Isentropic Compression
Isentropic Expansion
Wout
Win
1
4
V

TABLE A–17

Ideal-gas properties of air

T K	h kJ/kg	P_r	u kJ/kg	v_r	s° kJ/kg·K	T K	h kJ/kg	P_r	u kJ/kg	v_r	s° kJ/kg·K
200	199.97	0.3363	142.56	1707.0	1.29559	580	586.04	14.38	419.55	115.7	2.37348
210	209.97	0.3987	149.69	1512.0	1.34444	590	596.52	15.31	427.15	110.6	2.39140
220	219.97	0.4690	156.82	1346.0	1.39105	600	607.02	16.28	434.78	105.8	2.40902
230	230.02	0.5477	164.00	1205.0	1.43557	610	617.53	17.30	442.42	101.2	2.42644
240	240.02	0.6355	171.13	1084.0	1.47824	620	628.07	18.36	450.09	96.92	2.44356
250	250.05	0.7329	178.28	979.0	1.51917	630	638.63	19.84	457.78	92.84	2.46048
260	260.09	0.8405	185.45	887.8	1.55848	640	649.22	20.64	465.50	88.99	2.47716
270	270.11	0.9590	192.60	808.0	1.59634	650	659.84	21.86	473.25	85.34	2.49364
280	280.13	1.0889	199.75	738.0	1.63279	660	670.47	23.13	481.01	81.89	2.50985
285	285.14	1.1584	203.33	706.1	1.65055	670	681.14	24.46	488.81	78.61	2.52589
290	290.16	1.2311	206.91	676.1	1.66802	680	691.82	25.85	496.62	75.50	2.54175
295	295.17	1.3068	210.49	647.9	1.68515	690	702.52	27.29	504.45	72.56	2.55731
298	298.18	1.3543	212.64	631.9	1.69528	700	713.27	28.80	512.33	69.76	2.57277
300	300.19	1.3860	214.07	621.2	1.70203	710	724.04	30.38	520.23	67.07	2.58810
305	305.22	1.4686	217.67	596.0	1.71865	720	734.82	32.02	528.14	64.53	2.60319
310	310.24	1.5546	221.25	572.3	1.73498	730	745.62	33.72	536.07	62.13	2.61803
315	315.27	1.6442	224.85	549.8	1.75106	740	756.44	35.50	544.02	59.82	2.63280
320	320.29	1.7375	228.42	528.6	1.76690	750	767.29	37.35	551.99	57.63	2.64737
325	325.31	1.8345	232.02	508.4	1.78249	760	778.18	39.27	560.01	55.54	2.66176
330	330.34	1.9352	235.61	489.4	1.79783	780	800.03	43.35	576.12	51.64	2.69013
340	340.42	2.149	242.82	454.1	1.82790	800	821.95	47.75	592.30	48.08	2.71787
350	350.49	2.379	250.02	422.2	1.85708	820	843.98	52.59	608.59	44.84	2.74504
360	360.58	2.626	257.24	393.4	1.88543	840	866.08	57.60	624.95	41.85	2.77170
370	370.67	2.892	264.46	367.2	1.91313	860	888.27	63.09	641.40	39.12	2.79783
380	380.77	3.176	271.69	343.4	1.94001	880	910.56	68.98	657.95	36.61	2.82344
390	390.88	3.481	278.93	321.5	1.96633	900	932.93	75.29	674.58	34.31	2.84856
400	400.98	3.806	286.16	301.6	1.99194	920	955.38	82.05	691.28	32.18	2.87324
410	411.12	4.153	293.43	283.3	2.01699	940	977.92	89.28	708.08	30.22	2.89748
420	421.26	4.522	300.69	266.6	2.04142	960	1000.55	97.00	725.02	28.40	2.92128
430	431.43	4.915	307.99	251.1	2.06533	980	1023.25	105.2	741.98	26.73	2.94468
440	441.61	5.332	315.30	236.8	2.08870	1000	1046.04	114.0	758.94	25.17	2.96770
450	451.80	5.775	322.62	223.6	2.11161	1020	1068.89	123.4	776.10	23.72	2.99034
460	462.02	6.245	329.97	211.4	2.13407	1040	1091.85	133.3	793.36	23.29	3.01260
470	472.24	6.742	337.32	200.1	2.15604	1060	1114.86	143.9	810.62	21.14	3.03449
480	482.49	7.268	344.70	189.5	2.17760	1080	1137.89	155.2	827.88	19.98	3.05608
490	492.74	7.824	352.08	179.7	2.19876	1100	1161.07	167.1	845.33	18.896	3.07732
500	503.02	8.411	359.49	170.6	2.21952	1120	1184.28	179.7	862.79	17.886	3.09825
510	513.32	9.031	366.92	162.1	2.23993	1140	1207.57	193.1	880.35	16.946	3.11883
520	523.63	9.684	374.36	154.1	2.25997	1160	1230.92	207.2	897.91	16.064	3.13916
530	533.98	10.37	381.84	146.7	2.27967	1180	1254.34	222.2	915.57	15.241	3.15916
540	544.35	11.10	389.34	139.7	2.29906	1200	1277.79	238.0	933.33	14.470	3.17888
550	555.74	11.86	396.86	133.1	2.31809	1220	1301.31	254.7	951.09	13.747	3.19834
560	565.17	12.66	404.42	127.0	2.33685	1240	1324.93	272.3	968.95	13.069	3.21751
570	575.59	13.50	411.97	121.2	2.35531						

TABLE A–17

Ideal-gas properties of air (*Concluded*)

T K	h kJ/kg	P_r	u kJ/kg	v_r	$s°$ kJ/kg·K	T K	h kJ/kg	P_r	u kJ/kg	v_r	$s°$ kJ/kg·K
1260	1348.55	290.8	986.90	12.435	3.23638	1600	1757.57	791.2	1298.30	5.804	3.52364
1280	1372.24	310.4	1004.76	11.835	3.25510	1620	1782.00	834.1	1316.96	5.574	3.53879
1300	1395.97	330.9	1022.82	11.275	3.27345	1640	1806.46	878.9	1335.72	5.355	3.55381
1320	1419.76	352.5	1040.88	10.747	3.29160	1660	1830.96	925.6	1354.48	5.147	3.56867
1340	1443.60	375.3	1058.94	10.247	3.30959	1680	1855.50	974.2	1373.24	4.949	3.58335
1360	1467.49	399.1	1077.10	9.780	3.32724	1700	1880.1	1025	1392.7	4.761	3.5979
1380	1491.44	424.2	1095.26	9.337	3.34474	1750	1941.6	1161	1439.8	4.328	3.6336
1400	1515.42	450.5	1113.52	8.919	3.36200	1800	2003.3	1310	1487.2	3.994	3.6684
1420	1539.44	478.0	1131.77	8.526	3.37901	1850	2065.3	1475	1534.9	3.601	3.7023
1440	1563.51	506.9	1150.13	8.153	3.39586	1900	2127.4	1655	1582.6	3.295	3.7354
1460	1587.63	537.1	1168.49	7.801	3.41247	1950	2189.7	1852	1630.6	3.022	3.7677
1480	1611.79	568.8	1186.95	7.468	3.42892	2000	2252.1	2068	1678.7	2.776	3.7994
1500	1635.97	601.9	1205.41	7.152	3.44516	2050	2314.6	2303	1726.8	2.555	3.8303
1520	1660.23	636.5	1223.87	6.854	3.46120	2100	2377.7	2559	1775.3	2.356	3.8605
1540	1684.51	672.8	1242.43	6.569	3.47712	2150	2440.3	2837	1823.8	2.175	3.8901
1560	1708.82	710.5	1260.99	6.301	3.49276	2200	2503.2	3138	1872.4	2.012	3.9191
1580	1733.17	750.0	1279.65	6.046	3.50829	2250	2566.4	3464	1921.3	1.864	3.9474

Note: The properties P_r (relative pressure) and v_r (relative specific volume) are dimensionless quantities used in the analysis of isentropic processes, and should not be confused with the properties pressure and specific volume.

Source: Kenneth Wark, *Thermodynamics,* 4th ed. (New York: McGraw-Hill, 1983), pp. 785–86, table A–5. Originally published in J. H. Keenan and J. Kaye, *Gas Tables* (New York: John Wiley & Sons, 1948).

TABLE A–27

Properties of some common fuels and hydrocarbons

Fuel (phase)	Formula	Molar mass, kg/kmol	Density,[1] kg/L	Enthalpy of vaporization,[2] kJ/kg	Specific heat,[1] c_p kJ/kg·K	Higher heating value,[3] kJ/kg	Lower heating value,[3] kJ/kg
Carbon (*s*)	C	12.011	2	—	0.708	32,800	32,800
Hydrogen (*g*)	H_2	2.016	—	—	14.4	141,800	120,000
Carbon monoxide (*g*)	CO	28.013	—	—	1.05	10,100	10,100
Methane (*g*)	CH_4	16.043	—	509	2.20	55,530	50,050
Methanol (ℓ)	CH_4O	32.042	0.790	1168	2.53	22,660	19,920
Acetylene (*g*)	C_2H_2	26.038	—	—	1.69	49,970	48,280
Ethane (*g*)	C_2H_6	30.070	—	172	1.75	51,900	47,520
Ethanol (ℓ)	C_2H_6O	46.069	0.790	919	2.44	29,670	26,810
Propane (ℓ)	C_3H_8	44.097	0.500	335	2.77	50,330	46,340
Butane (ℓ)	C_4H_{10}	58.123	0.579	362	2.42	49,150	45,370
1-Pentene (ℓ)	C_5H_{10}	70.134	0.641	363	2.20	47,760	44,630
Isopentane (ℓ)	C_5H_{12}	72.150	0.626	—	2.32	48,570	44,910
Benzene (ℓ)	C_6H_6	78.114	0.877	433	1.72	41,800	40,100
Hexene (ℓ)	C_6H_{12}	84.161	0.673	392	1.84	47,500	44,400
Hexane (ℓ)	C_6H_{14}	86.177	0.660	366	2.27	48,310	44,740
Toluene (ℓ)	C_7H_8	92.141	0.867	412	1.71	42,400	40,500
Heptane (ℓ)	C_7H_{16}	100.204	0.684	365	2.24	48,100	44,600
Octane (ℓ)	C_8H_{18}	114.231	0.703	363	2.23	47,890	44,430
Decane (ℓ)	$C_{10}H_{22}$	142.285	0.730	361	2.21	47,640	44,240
Gasoline (ℓ)	$C_nH_{1.87n}$	100–110	0.72–0.78	350	2.4	47,300	44,000
Light diesel (ℓ)	$C_nH_{1.8n}$	170	0.78–0.84	270	2.2	46,100	43,200
Heavy diesel (ℓ)	$C_nH_{1.7n}$	200	0.82–0.88	230	1.9	45,500	42,800
Natural gas (*g*)	$C_nH_{3.8n}N_{0.1n}$	18	—	—	2	50,000	45,000

[1]At 1 atm and 20°C.

[2]At 25°C for liquid fuels, and 1 atm and normal boiling temperature for gaseous fuels.

[3]At 25°C. Multiply by molar mass to obtain heating values in kJ/kmol.