

THERMODYNAMIC PERFORMANCE AND DESIGN OF SMALL-SCALE H₂O-LiBr VAPOR ABSORPTION SYSTEM FOR ROOM AIR COOLING

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in Partial Fulfillment of the Requirements for the
Degree of**

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**in
Mechanical Engineering
by
Shobhit Mishra
(2K17/PhD/ME/49)**

Under the Supervision of

**Dr. Raj Kumar Singh
(Professor, DTU, Delhi)
Department of Mechanical Engineering**



**To the
Department of Mechanical Engineering**

**DELHI TECHNOLOGICAL UNIVERSITY (Formerly
Delhi College of Engineering) Shahbad Daultpur, Main
Bawana Road, Delhi-110042, India
July, 2025**



CERTIFICATE

This is to certify that the thesis entitled “**Thermodynamic performance and Design of small-scale H₂O-LiBr vapor absorption system for room air cooling**” being submitted by **Shobhit Mishra**, Roll No.2K17/PhD/ME/49 to the Delhi Technological University, Delhi for the award of the degree of **Doctor of Philosophy** is a bonafide record of original research work carried out by him. He has worked under our guidance and supervision and has fulfilled the requirements for the submission of this thesis, which has reached the requisite standard. The results contained in this thesis have not been submitted, in part or full, to any other University or Institute for the award of any degree or diploma.

Dr. Raj Kumar Singh
(Professor, DTU, Delhi)
Department of Mechanical Engineering



DELHI TECHNOLOGICAL UNIVERSITY
(Formerly Delhi College of Engineering)
Shahbad Daultpur, Main Bawana Road, Delhi-42

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Shobhit Mishra
(Roll No. 2K17/PhD/ME/49)



DELHI TECHNOLOGICAL UNIVERSITY
(Formerly Delhi College of Engineering)
Shahbad Daulatpur, Main Bawana Road, Delhi-42

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(Formerly Delhi College of Engineering)
Shahbad Daulatpur, Main Bawana Road, Delhi-42

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Dr. Raj Kumar Singh
(Professor, DTU, Delhi)
Department of Mechanical Engineering

Date: 31- July - 2025

ABSTRACT

The present study addresses the research gap in small-scale distributed trigeneration systems by investigating two novel configurations: (1) a solar power tower (SPT) driven helium Brayton cycle (HBC) integrated with a heat recovery steam generator (HRSG) and a vapor absorption cooling system (VACS), and (2) a solid oxide fuel cell-gas turbine (SOFC-GT) hybrid system coupled with a vapor absorption refrigeration system (VARs). The primary objectives of this thesis are: (i) to develop and simulate a novel SPT-HBC-HRSG-VACS trigeneration system for combined power, heating, and cooling; (ii) to perform comprehensive energy and exergy analysis of the proposed trigeneration system; (iii) to develop a SOFC-GT-VARS cogeneration system for small-scale room air conditioning; and (iv) to investigate the energy, exergy, economic, and environmental (4E) performance of the SOFC-GT-VARS system.

The SPT-HBC-HRSG-VACS system exploits waste heat from a basic HBC driven by an SPT. The two subsystems, HRSG and VACS, recover waste heat for steam generation and air conditioning cooling, respectively. A comprehensive exergy and energy analysis of this proposed trigeneration system was carried out with parametric analysis using Engineering Equation Solver (EES) software. It was concluded that energy and exergy efficiency and net work output of the proposed trigeneration system were observed as 44.96%, 34.15%, and 14,562 kW respectively. The heating production through the HRSG was obtained as 8,510 kW while the cooling production by VACS was 115.10 kW. Moreover, the coefficient of performance (COP) of the VACS subsystem was observed as 0.8015. Energy and exergy efficiency of the SPT-operated basic HBC using the subsystems were improved by 58.98% and 12.92%, respectively. Parametric analysis revealed that the helium turbine inlet temperature and solar heliostat field efficiency significantly affect the trigeneration system performance.

For the second configuration, a novel SOFC-GT-VARS system for combined cooling and power generation is developed. Thermodynamic, economic, and environmental analyses were performed on the proposed system using computational techniques. The proposed plant obtained power output, exergy efficiency, and energy efficiency of 551.29 kW, 48.03%, and 50.18%, respectively, at given operating conditions. The cooling effect of 5.275 kW (1.5 TR) was obtained from the VARS with a COP of 0.752. The total cost rate of the proposed plant was observed as 33.57 \$/h, while the CO₂ emission per MWh of energy output was obtained as 394.70 kg/MWh. The research successfully fills the identified gaps in small-scale distributed trigeneration systems and provides a comprehensive framework for evaluating such systems under Indian climatic and economic conditions.

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LIST OF SYMBOLS, ABBREVIATIONS AND NOMENCLATURE

SYMBOLS

$\dot{E}X$	Rate of exergy	kW
h	Specific enthalpy	kJ/kg or kJ/kmol
$\dot{W}$	Power / Work output	kW
$\dot{Q}$	Rate of heat interaction / Heat rate	kW
T	Temperature	K
A	Area	m ²
s	Specific entropy	kJ/kg·K or kJ/kmol·°C
$\dot{m}$	Mass flow rate	kg/s
$\dot{n}$	Molar flow rate	mol/s
a	Species activity factor	—
F	Faraday constant	C/g·mol
L	SOFC layer thickness	m
z_r	Extent of electrochemical reaction	mol/s
R_C	Contact resistivity	$\Omega \cdot m^2$
x_r	Extent of steam reforming reaction	mol/s
y_r	Water gas shift reaction extent	mol/s
y	Mole fraction	—
$\dot{C}$	Cost rate	\$/h
$\dot{Z}$	Capital cost rate	\$/h
$\dot{E}D$	Exergy destruction rate	kW
ρ	Cell component electrical resistivity	—

ABBREVIATIONS

CR	Central Receiver
HBC	Helium Brayton Cycle
SPT	Solar Power Tower
SHE	Solution Heat Exchanger
SCO ₂	Supercritical CO ₂
VACS	Vapor Absorption Cooling System
VARS	Vapor Absorption Refrigeration System
ORC	Organic Rankine Cycle
IHE	Intermediate Heat Exchanger
HC	Helium Compressor
HT	Helium Turbine
COP	Coefficient of Performance
CPR	Compressor Pressure Ratio
CIT	Compressor Inlet Temperature
HTIT	Helium Turbine Inlet Temperature
DNI	Direct Normal Irradiation
HRSG	Heat Recovery Steam Generator
FHE	Fuel Heat Exchanger
AHE	Air Heat Exchanger
WHE	Water Heat Exchanger
SOFC	Solid Oxide Fuel Cell
GT	Gas Turbine
EMI	CO ₂ Emission Rate

ABS	Absorber
AB	After Burner
COND	Condenser
EVAP	Evaporator
GEN	Generator
PRV	Pressure Reducing Valve
EV	Expansion Valve
WST	Water Storage Tank

SUBSCRIPTS

0	Dead / Atmospheric Condition
i	Inlet or ith State
j	Particular State or jth Component
k	kth State
e	Exit / Evaporator
g	Generator
c	Condenser
rec	Receiver
ref	Reference / Reflectivity
hel	Heliostat
tri	Trigeneration
co	Cogeneration
ex	Exergy
en	Energy

f	Fuel
fc	Fuel Compressor
ac	Air Compressor
wp	Water Pump

GREEK LETTERS

η	Efficiency
ε	Effectiveness

CHAPTER 1 : INTRODUCTION

1.1 India's Cooling Challenge and Energy Landscape

The climate crisis, driven largely by fossil fuel combustion, demands urgent decarbonization across all energy sectors. Among these, the cooling sector is rapidly emerging as one of the most energy-intensive and environmentally significant. The International Energy Agency (IEA) reports that global energy demand for space cooling has tripled since 1990, with projections indicating a further tripling by 2050 if current trends continue [1].

As living standards rise in tropical and subtropical countries, the penetration of air conditioners is set to surge. The IEA estimates that 10 new air conditioning units could be sold every second for the next 30 years, leading to 5.6 billion units by 2050—up from 1.6 billion today. This growth represents a massive burden on electricity systems and a significant source of indirect emissions, especially where power is derived from fossil fuels [1].

India is at the epicenter of this global transition. With a rapidly urbanizing population, projected to reach 1.6 billion by 2047, and rising middle-class affluence, cooling demand is escalating dramatically. Residential, commercial, and transport sectors are expected to drive a sevenfold increase in electricity use for space cooling by 2050 [1,2]. This growth could push cooling to account for up to 45% of India's peak electricity demand in major cities during summer.

Moreover, the country remains highly dependent on coal, which supplied approximately 70% of total electricity generation in 2022 [3-5]. Meeting cooling demands using coal-based electricity risks a feedback loop—where more ACs mean more coal usage, leading to more GHG emissions, which in turn worsen the climate that drives cooling demand.

1.2 Problems Associated with Vapor Compression System-Based Cooling

Vapor Compression Systems (VCS), the conventional approach to air conditioning, are inherently electricity-intensive and rely on hydrofluorocarbons (HFCs) and hydrochlorofluorocarbons (HCFCs)—refrigerants that have high Global Warming Potential. In addition to HFCs and HCFCs, modern vapor compression systems increasingly utilize low-

GWP alternatives such as Hydrofluoroolefins (HFOs) (e.g., R-1234ze(E), R-1234yf with GWP < 1), and natural refrigerants including CO₂ (R-744) for commercial refrigeration and transcritical cycles, and ammonia (R-717) for industrial applications. These refrigerants offer zero or very low global warming potential while maintaining good thermodynamic performance.

1.2.1 Key problems include:

- Peak electricity demand spikes, often necessitating grid expansion and costly reserve generation.
- Refrigerant leakage, contributing to atmospheric warming.
- High lifecycle energy use, including embodied energy in components.
- Unsuitability for off-grid or rural applications, where consistent power supply is lacking.

In countries like India, where electrification remains uneven and power outages are common in many rural and peri-urban areas, reliance on VCS limits access to thermal comfort and medical refrigeration in energy-insecure zones. The solution lies in thermal-driven cooling, particularly using solar energy. Solar-assisted vapor absorption cooling systems (VACS) use solar thermal energy collected during the daytime to drive absorption refrigeration cycles. These systems typically eliminate compressors and instead use heat to separate a refrigerant-absorbent solution, thereby achieving cooling through thermal desorption and recombination. India is endowed with excellent solar radiation, averaging 4.5-7.0 kWh/m²/day across most of the country [4]. This irradiance aligns well with cooling demand, which peaks during daytime summer hours.

1.2.2 Key advantages of this alignment include:

- Temporal matching of solar energy availability with peak load.
- Reduction of stress on the electrical grid.

- Potential for off-grid operation using solar thermal and thermal energy storage.

1.2.3 Common collector types include:

- Flat Plate Collectors (FPC) for low-temperature (60-80°C) cycles.
- Evacuated Tube Collectors (ETC) for medium-temperature (70-110°C) applications.
- Parabolic Trough Collectors (PTC) for high-temperature, double-effect chillers (140-180°C).
- Compound Parabolic Concentrators (CPC) and Linear Fresnel Reflectors (LFR) for integrated hybrid systems. These systems, when combined with hot water or molten salt thermal storage, offer high-reliability cooling in a renewable, clean, and decentralized format.

1.3 India Cooling Action Plan (ICAP)

Recognizing the urgency, the Government of India has launched several policy initiatives aimed at transforming the cooling sector. Launched in 2019 [2], ICAP is the world's first national plan to comprehensively address the cooling demand across sectors.

1.3.1 Its key targets include:

- 20-25% reduction in cooling demand by 2037-38.
- 25-30% reduction in refrigerant demand.
- 40% reduction in cooling energy consumption.

1.3.2 Other initiatives include:

- National Solar Mission: Aims to deploy 100 GW of solar energy by 2022 (extended beyond), providing an opportunity to integrate solar thermal for distributed cooling.
- Energy Conservation Building Code (ECBC): Mandates efficiency benchmarks and renewable integration in new commercial buildings.
- Kigali Amendment Compliance: India has ratified the Kigali Amendment to the Montreal Protocol, committing to phase down HFCs by over 80% by 2047 [6]. These

frameworks create a robust policy environment that supports investment in solar thermal cooling technologies.

1.4 Vapor Absorption Cooling System

Vapor absorption cooling systems (VAS) differ fundamentally from VCS in operation and energy source. They use a combination of a refrigerant (e.g., water or ammonia) and an absorbent (e.g., lithium bromide or water) in a closed-loop cycle that is driven by thermal energy rather than electricity.

1.4.1 Key Benefits [7,8,10,13,15,17]:

- Ultra-low electricity consumption (limited to pumps and control systems).
- Use of natural refrigerants (zero ODP and low or zero GWP).
- Longer life spans, due to fewer moving parts and reduced mechanical stress.
- Higher part-load efficiency.
- Silent operation, especially important for hospitals, offices, and libraries.

1.4.2 Performance Metrics:

- COP (Coefficient of Performance): 0.6-0.8 for single-effect systems, 1.1-1.4 for double-effect systems.
- Operational temperature range: 70-180°C, depending on cycle type.
- Cooling capacity: From small-scale (1-5 TR) to large-scale (150+ TR) systems.
- These systems are especially well-suited for campuses, institutional buildings, industrial plants, hospitals, and commercial centers, particularly where waste heat or solar heat is abundantly available.

1.4.3 By investing in solar-assisted absorption cooling, India can:

- Mitigate peak load stress.
- Reduce dependence on imported refrigerants.
- Create green jobs in installation and maintenance.

- Improve air quality and reduce urban heat island effects.
- Provide reliable off-grid cooling in rural clinics and cold chains.
- The integration of thermal energy storage (TES) with these systems ensures nighttime cooling and high operational resilience, making them ideal candidates for both Smart Cities and Atmanirbhar Bharat (self-reliant India) initiatives.

1.5 Overview and Historical Context

Vapor absorption cooling systems (VAS) are thermally driven refrigeration cycles that emerged in the 19th century, long before electrically powered compression systems gained prominence. The first commercial absorption refrigerator was patented by Ferdinand Carré in 1858, using water-ammonia as the working pair [120]. Though initially popular, VAS were overtaken by electrically powered vapor compression systems (VCS) due to their compactness and ease of control. However, with modern concerns around energy sustainability, GHG emissions, and electricity scarcity—particularly in peak-load and rural scenarios—VAS have re-emerged as viable alternatives. Especially when powered by renewable heat sources such as solar thermal energy, VAS offers a clean, efficient, and decentralized solution for cooling.

1.6 Basic Principle of Operation

VAS operates on the principle of absorbing and releasing a refrigerant from an absorbent solution using thermal energy. Unlike VCS, which use a mechanical compressor, VAS uses a thermal generator (or desorber) to separate the refrigerant from the absorbent, thus reducing mechanical complexity.

Table 1.1: The VAS cycle includes the following major components [7]

Component	Function
Evaporator	Absorbs heat from the conditioned space; refrigerant evaporates
Absorber	Refrigerant vapor absorbed into the absorbent; releases heat
Generator	Heat applied to separate refrigerant from absorbent solution

Condenser	Refrigerant vapor condensed to liquid by rejecting heat
Expansion Valve	Throttles high-pressure refrigerant to low pressure
Solution Pump	Pumps rich solution from absorber to generator

1.6.1 Working Cycle Description

I. Evaporation: The refrigerant (e.g., water) evaporates in the evaporator by absorbing heat from the load, producing the cooling effect.

II. Absorption: The vaporized refrigerant enters the absorber, where it is absorbed by the concentrated absorbent (e.g. +3LiBr). This is an exothermic process and requires a cooling medium to remove the released heat.

III. Pumping: The now "rich" solution is pumped to a higher pressure using a solution pump—this is the only component consuming mechanical energy.

IV. Generation: Heat is applied to the generator using solar thermal energy or waste heat. This separates the refrigerant from the absorbent.

V. Condensation: The refrigerant vapor flows to the condenser, where it is condensed by ambient air or cooling water.

VI. Expansion: The high-pressure refrigerant is throttled through an expansion valve and returned to the evaporator, completing the cycle.

1.6.2 Advantages & Disadvantages

Table 1.2: Advantages of Vapor Absorption Systems

Aspect	Advantage
Energy Source	Uses thermal energy (solar, waste heat) instead of electricity
Refrigerants	Natural refrigerants like water, ammonia; zero ODP, low GWP
Mechanical Complexity	Fewer moving parts; low maintenance
Part-load Operation	Better performance under partial loads

Sustainability	Enables off-grid and climate-resilient cooling
Integration	Can be combined with solar collectors and thermal storage

Table 1.3: Comparison with Vapor Compression Systems [7,10,11]

Parameter	Vapor Compression	Vapor Absorption
Energy Input	Electricity	Thermal (solar/gas/waste heat)
COP	2.5–3.5	0.6–1.3
Refrigerant	HFC/HCFC	H ₂ O/NH ₃
ODP/GWP	High	Zero/Low
Maintenance	Medium	Low
Part-load Efficiency	Lower	Higher
Initial Cost	Lower	Higher
Operating Cost	Higher	Lower

1.6.3 Limitations of VAS:

Despite their benefits, vapor absorption systems have the following challenges:

- **Large Physical Size:** Requires more space than VCS for the same cooling output.
- **Slow Response:** Thermally driven systems have slower start-up times.
- **Crystallization Risk:** In systems like H₂O-LiBr, improper operation can lead to salt crystallization.
- **Vacuum Operation:** H₂O-LiBr systems must operate under deep vacuum (~6 mmHg), requiring leak-proof construction.
- **Cost:** Higher initial capital cost due to collectors, absorption chiller, and storage.

1.7 Typical System Configurations [7,11,12,13]

A. Single-effect, Solar-Assisted VAS: Solar source: ETC (80-100°C), Working pair: H₂O-LiBr, Cooling capacity: 5-50 TR, Application: Offices, hospitals, academic campuses

B. Double-effect, PTC-Driven VAS: Solar source: PTC (150-180°C), Cooling capacity: 50-200 TR, Application: Industrial cooling, data centers.

C. Hybrid Absorption System with Auxiliary Gas Burner: Ensures 24/7 reliability, Backup for cloudy/rainy periods, Common in high-dependability applications (e.g., pharma, banking). Beyond single-effect and double-effect configurations, other absorption chiller configurations include: Triple-effect chillers operate at generator temperatures of 180-220°C, achieving COPs of 1.4-1.7. These require high-temperature heat sources such as gas burners or concentrated solar collectors (PTC, Fresnel). Half-effect chillers operate at generator temperatures of 60-75°C, achieving COPs of 0.3-0.4. These are suitable for very low-grade waste heat (e.g., solar flat plate collectors, engine jacket water). Double-lift (two-stage) chillers operate at generator temperatures of 70-90°C with COPs of 0.35-0.45. These are designed for applications with low-temperature heat sources (60-80°C) where single-effect is not feasible. While less common than single/double-effect systems, these configurations are commercially available from manufacturers such as Broad, Thermax, and Yazaki for specialized applications

1.8 Selection of Working Pairs

In a vapor absorption system (VAS), the refrigerant-absorbent pair plays a pivotal role in determining the system's efficiency, temperature operating range, component size, safety, environmental impact, and long-term reliability. The selection of a suitable working fluid combination depends on several thermophysical and operational factors such as [16]:

- Absorption/desorption rates.
- Vapor pressure compatibility.
- Thermal conductivity.
- Crystallization limits.
- Toxicity and environmental safety.
- Operating pressure and temperature range.

- Compatibility with materials and heat source (e.g., solar collector).
- An ideal working pair should provide high cycle efficiency, be environmentally benign, remain stable under thermal cycling, and offer cost-effectiveness in commercial applications.

1.9 Refrigerant Selection Criteria

The refrigerant is the component that evaporates and condenses during the cycle, absorbing heat in the evaporator and rejecting it in the condenser. Desired properties include [14]:

- Low boiling point at operating pressure (to absorb heat at low temperatures).
- High latent heat of vaporization (to enhance cooling capacity).
- High vapor pressure at evaporator temperatures.
- Thermal and chemical stability under cycle conditions.
- Non-corrosive and non-toxic.
- Environmentally safe (zero ODP, low GWP).

1.9.1 The two most widely used refrigerants in absorption systems are:

- **Water (H₂O):** Non-toxic, high latent heat, but limited to applications >0°C.
- **Ammonia (NH₃):** Can operate down to -30°C; toxic and flammable, requires pressurized design and rectification.

1.9.1.1 Absorbent Selection Criteria

- Absorbents are liquids or salts that selectively absorb refrigerant vapor. Ideal characteristics include [11,12,13,14]:
- Strong affinity for refrigerant vapor.
- Low vapor pressure (to avoid carryover).
- High solubility and low viscosity.
- Chemical stability over temperature and time.
- Non-crystallizing and non-toxic.

- High heat of solution (to aid generator separation).
- Corrosion compatibility with system materials.

1.9.1.2 Common absorbents:

- **Lithium bromide (LiBr):** Hygroscopic salt used with water refrigerant.
- **Water (H₂O):** Used with ammonia refrigerant.
- **Lithium nitrate (LiNO₃), Lithium chloride (LiCl):** Emerging candidates with specific thermodynamic profiles.

1.9.2 Refrigerant-Absorbent Pairs

1.9.2.1 Water-Lithium Bromide (H₂O-LiBr)

Most commercialized combination for comfort cooling. Operates under deep vacuum (~6-10 mmHg). Refrigerant: Water; Absorbent: LiBr. Generator temp: 80-110°C (single-effect), up to 170°C (double-effect). Limitation: Crystallization risk at high concentration and low temperatures.

1.9.2.2 Ammonia-Water (NH₃-H₂O)

Suitable for low-temperature refrigeration (as low as -40°C for industrial freezing applications, and typically -20°C to -40°C depending on system configuration). Toxicity of ammonia requires strict leak management and material selection. Requires rectifier or dephlegmator to remove water vapor from ammonia stream.

1.9.2.3 Ammonia-Lithium Nitrate (NH₃-LiNO₃)

No water used → no rectifier required. Emerging system for sub-zero applications. High viscosity of LiNO₃ limits mass transfer; needs compact heat exchangers. Lower corrosiveness compared to NH₃-H₂O.

1.9.2.4 Water-Lithium Chloride (H₂O-LiCl)

Used in desiccant-based cooling or hybrid systems. Good for simultaneous dehumidification and cooling. Corrosion and cost concerns limit widespread use.

Table 1.4: Comparison of Working Pairs [7,8,11,13,14,16]

Property	H ₂ O–LiBr	NH ₃ –H ₂ O	NH ₃ –LiNO ₃	H ₂ O–LiCl
Cooling Range	>5°C	-20 to +10°C	-10 to +5°C	8–14°C
Pressure	Vacuum	10–25 bar	8–20 bar	Vacuum
COP	0.7–1.3	0.5–0.75	0.6–1.0	0.5–0.7
Crystallization	Yes	No	No	Moderate
Rectifier Needed	No	Yes	No	No
Corrosion	High (needs inhibitor)	Moderate	Low	High
Toxicity	Non-toxic	Toxic	Toxic	Non-toxic
Application	Comfort cooling	Cold storage	Experimental	Hybrid cooling
Commercial Use	Widely used	Moderate	Low	Niche

Table 1.5: Compatibility with Solar Collectors [7,21,22,23]

Solar Collector	Temperature Range (°C)	Suitable Pairs	Reference
Flat Plate Collector (FPC)	60–80	H ₂ O–LiBr (single-effect)	Shirazi et al. [7]
Evacuated Tube Collector (ETC)	80–110	H ₂ O–LiBr (single-effect), NH ₃ –LiNO ₃	Li & Sumathy [25]
Compound Parabolic Collector (CPC)	100–140	NH ₃ –H ₂ O (with rectifier)	Ayadi et al. [21]

Parabolic Trough Collector (PTC)	140–180	H ₂ O–LiBr (double-effect), NH ₃ –H ₂ O	Bermejo et al. [30]
Fresnel / Solar Tower	180–250	Triple-effect absorption systems	Mokhtar et al. [23]

Footnote: These temperature ranges are specific to driving absorption chillers. The operational limits of the collectors themselves may be higher (e.g., PTCs can operate up to 400°C for CSP power generation). The ranges shown here represent the optimal generator temperatures for the respective absorption chiller configurations as reported in the cited literature.

1.10 Challenges in Working Pair Selection [20]

- **Crystallization:** LiBr can precipitate if concentration exceeds ~58% or temperature drops too low. Prevented via solution control, dilution circuits, and heat exchangers.
- **Corrosion:** LiBr and LiCl are corrosive, especially to ferrous metals. Additives like lithium chromate or nitrate are used as corrosion inhibitors.
- **Volatility:** NH₃ is highly volatile and toxic. Requires stainless steel components and leak detection systems.
- **Viscosity:** LiNO₃ has higher viscosity, reducing heat/mass transfer. Compact HX and additives help mitigate.
- **Environmental Impact:** Though many pairs have low GWP, safe handling and disposal are still essential.

1.11 Emerging Trends and Innovations [18,19]

I. Nanofluid Additives:

Adding nanoparticles (e.g., Al₂O₃, CuO) to absorbent enhances thermal conductivity. Research shows COP improvements up to 10-15%.

II. Ionic Liquids:

Designer solvents with high thermal stability, low vapor pressure, and tunable properties. Still in experimental phase but promising for future systems.

III. Hybrid Working Pairs:

Mixtures like LiBr-LiCl or NH₃-NaSCN aim to improve operating stability and reduce crystallization.

IV. AI-Powered Pair Selection:

Machine learning tools used to screen chemical databases for optimal refrigerant-absorbent pairs.

1.12 Detailed Analysis of Working Pairs

The selection of an optimal refrigerant-absorbent working pair is a cornerstone of efficient vapor absorption cooling system design. This section delves deeper into the mechanisms, operating conditions, performance characteristics, and application scope of the four most significant working pairs.

1.12.1 Water-Lithium Bromide (H₂O-LiBr) [8,9,11,12]

1.12.1.1 Thermodynamic Behaviour:

In this most commonly used pair, water acts as the refrigerant, and LiBr serves as the absorbent.

The solution exhibits strong chemical affinity between LiBr and water vapor, enabling rapid absorption. The system operates under deep vacuum conditions, typically at pressures below 1 kPa.

1.12.1.2 Operating Conditions:

- Evaporator temperature: 5-10°C
- Generator temperature: 80-100°C (single-effect), 140-170°C (double-effect)
- COP range: 0.65-0.8 (single-effect), 1.1-1.3 (double-effect)

1.12.1.3 Application Scope:

- Centralized air conditioning in commercial/institutional buildings (hospitals, airports, shopping malls, office complexes).

- HVAC systems in large buildings requiring 50-500 TR capacity.
- Solar cooling in hot and humid climates (e.g., coastal India, Southeast Asia).
- Large-scale comfort cooling applications where non-toxicity is critical.
- District cooling systems integrated with solar thermal fields.

1.12.1.4 Advantages:

- Widely commercialized with well-established technology.
- Excellent absorption characteristics.
- Non-toxic and non-flammable refrigerant.
- Highly compatible with solar thermal collectors, especially ETCs and PTCs.

Limitations:

- Crystallization occurs at concentrations >58% or if absorber temperature drops too low.
- Corrosive to copper, steel; needs inhibitors like lithium chromate.
- Vacuum operation requires high-precision sealing.

1.12.2 Ammonia-Water ($\text{NH}_3\text{-H}_2\text{O}$) [11,16]

1.12.2.1 Thermodynamic Behaviour:

Ammonia serves as the refrigerant, while water acts as the absorbent. Unlike $\text{H}_2\text{O-LiBr}$, both fluids are volatile, necessitating the use of a rectifier (or distillation column) to purify the refrigerant before condensation. The $\text{NH}_3\text{-H}_2\text{O}$ system is known for its low-temperature refrigeration capability (down to -20°C), making it suitable for cold chains and ice-making applications.

1.12.2.2 Operating Conditions:

- Evaporator temp: -20 to $+10^\circ\text{C}$
- Generator temp: $120\text{-}160^\circ\text{C}$
- Operating pressure: 8-24 bar
- COP: 0.5-0.75

1.12.2.3 Application Scope:

- Low-temperature refrigeration for cold storage facilities (0°C to -20°C).
- Dairy and food processing industries (milk chilling, meat storage).
- Solar-assisted freezing units for ice production.
- Absorption ice machines for fishing vessels and remote areas.
- Rural milk cooling in off-grid areas without reliable electricity.

1.12.2.4 Advantages:

- Operates at sub-zero temperatures.
- Compact due to higher density and pressure.
- Refrigerant (ammonia) has very low GWP and zero ODP.

1.12.2.5 Limitations:

- Ammonia is toxic and flammable, requiring safety measures.
- Needs rectifier to separate water from ammonia vapor.
- Steel or stainless steel components increase capital cost.

1.12.3 Ammonia-Lithium Nitrate (NH₃-LiNO₃) [11,15]

1.12.3.1 Thermodynamic Behaviour:

This pair is non-aqueous, meaning no rectifier is required. NH₃-LiNO₃ systems operate at intermediate pressures and are under active research due to their simplified architecture. The lithium nitrate absorbent binds ammonia vapor effectively, and desorption occurs upon heating. However, high viscosity and lower mass diffusivity challenge the heat and mass transfer efficiency.

1.12.3.2 Operating Conditions:

- Evaporator temp: -5 to +5°C
- Generator temp: 100-150°C
- Pressure: 8-15 bar
- COP: 0.6-1.0

1.12.3.3 Application Scope:

- Decentralized rural cooling applications (5-20 TR capacity).
- Vaccine preservation in off-grid areas (no rectifier required, simplifying maintenance).
- Experimental hybrid chillers for developing countries.
- Applications where compact design is prioritized over efficiency.

1.12.3.4 Advantages:

- No need for rectification → compact cycle.
- Reduced component count.
- Promising for solar-based rural refrigeration.

1.12.3.5 Limitations:

- High viscosity impacts heat exchanger performance.
- Limited experimental validation at large scale.
- Corrosiveness not fully understood.

1.12.4 Water-Lithium Chloride (H₂O-LiCl) [16]

1.12.4.1 Thermodynamic Behaviour:

H₂O-LiCl exhibits strong hygroscopic properties, making it ideal for dehumidification and combined sensible-latent cooling. It is generally used in desiccant-assisted absorption or hybrid systems. Unlike LiBr, LiCl has a higher solubility limit and marginally better stability in humid environments.

1.12.4.2 Operating Conditions:

- Evaporator temp: 8-14°C
- Generator temp: 85-105°C
- Vacuum operation
- COP: 0.5-0.7

1.12.4.3 Application Scope:

- HVAC systems in coastal zones (excellent humidity control for Mumbai, Chennai, Kolkata).

- Hospitals, archives, and museums requiring precise humidity management (40-60% RH).
- Hybrid desiccant-cooling systems for simultaneous sensible and latent cooling.
- Net-zero energy buildings with integrated solar thermal and desiccant systems.

1.12.4.4 Advantages:

- Dual function: dehumidification + cooling.
- Operates at low generator temperatures → ETC compatibility.
- Enhanced comfort due to humidity control.

1.12.4.5 Limitations:

- Less efficient than LiBr.
- Expensive and corrosive to metals.
- Less commercial experience.

Table 1.6: Comparison Summary of different working pairs [11,12,14,16]

Attribute	H ₂ O–LiBr	NH ₃ –H ₂ O	NH ₃ –LiNO ₃	H ₂ O–LiCl
COP	0.65–1.3	0.5–0.75	0.6–1.0	0.5–0.7
T _{min} (°C)	~5°C	-20°C	-5°C	8°C
Generator Temp	80–170°C	120–160°C	100–150°C	85–105°C
Pressure	Vacuum	High	Medium	Vacuum
Crystallization Risk	High	None	Low	Medium
Rectifier Needed	No	Yes	No	No
Toxicity	None	High	Medium	None
Applications	HVAC	Cold chain	Rural cooling	Humidity control

CHAPTER 2 : LITERATURE REVIEW

Solar absorption cooling technologies have seen development in system capacities ranging from as low as 10 kW up to several hundred kilowatts, although the majority of actual installations are presently functioning within experimental or demonstration settings. The impetus behind the exploration of these systems arises from the limitations associated with conventional mechanical vapor compression approaches, especially in the area of space cooling. Thus, the solar vapor absorption cooling field has attracted significant research interest worldwide. Notably, countries such as Spain and Germany stand out for their work in this area, with remarkable progress and investments also being made by China, Japan, India, and other countries that have rapidly advanced and are pioneering developments in this domain [7].

2.1 Vapor Absorption System

Nakahara [24] and colleagues performed an in-depth experimental analysis of a single-effect LiBr-H₂O solar vapor absorption chiller with a cooling capacity of 7 kW, suitable for air conditioning, space heating, as well as hot water supply. The system's operation was powered by a solar thermal field comprising 32.2 square meters of flat plate solar collectors, set up at the Ohbayashi-Gumi Technical Research Institute, Tokyo, Japan. The essential goal of this experimental undertaking was the acquisition of comprehensive and foundational data related to the individual and integrated characteristics of the collector module, absorption refrigeration unit, thermal energy storage tank, an auxiliary heating device, and the associated control and safety provisions. To facilitate the buffering and regulated delivery of solar-derived heat, the configuration included a substantial 2.5 cubic meter hot water tank specified for thermal storage . Following successful year-long operation, experimental findings showed that, depending on the required cooling load, the observed COP of the chiller ranged from 0.4 to 0.8, this variation corresponding with generator temperatures fluctuating between 70°C and 100°C, respectively. Importantly, the plant was designed with a freeze protection module to safeguard against low ambient temperatures. In periods when solar insolation proved

insufficient, an auxiliary gas-fired burner rated at 14 kW provided necessary support as a backup source. However, the investigators did not extend their inquiry to cover economic analyses related to the solar collector, because the primary objective at this stage was to generate technical performance data to inform subsequent economic optimization and system design revisions.

Li and Sumathy [25] undertook a comparative experimental investigation of a 4.75 kW cooling capacity LiBr-H₂O solar vapor absorption chiller deployed for cooling purposes and integrated with a partitioned storage tank. This system was coupled with a substantial flat plate collector array measuring 38 square meters, established at the University of Hong Kong. The primary aim of this study was to measure and compare the system's performance when operating with the innovative partitioned tank as opposed to the traditional single, whole tank mode. To ensure stable thermal conditions during periods of low solar input or heightened demand, a 12 kW auxiliary heater was installed within the configuration. Over an operational period of roughly four months, the collected data clearly indicated that use of a partitioned storage arrangement resulted in superior performance, achieving increased COP and other performance indicators, particularly when solar irradiation levels were diminished. The system's total solar-based cooling COP reached approximately 0.07. It should be noted that the researchers did not undertake any inquiry into the economic viability of the solar collector subsystem nor did they conduct an environmental impact assessment comparing both the solar and auxiliary heating portions of the configuration.

Darkwa [26] and co-workers launched an empirical performance evaluation for a single-effect solar vapor absorption system charged with the task of providing 55 kW cooling, sourced from a 220 square meter field of evacuated tube solar collectors. This experiment was executed within the Centre for Sustainable Energy Technologies (CSET) building, University of Nottingham Ningbo, China. The primary focus centered on examining the behavior and

efficiency of the overall system when subjected to the sub-tropical environmental conditions prevalent at the site. Toward this end, comprehensive monitoring and data collection extended across a full week, encompassing daily operational windows between 12:00 PM and 4:00 PM, with the experiment conducted in the high-temperature month of August, 2010. When generator water temperature was maintained at 90°C via the solar array, the absorption chiller system exhibited a measured COP of 0.69. Nevertheless, despite these technical observations, the analysis did not extend to any assessment regarding the economic payoff or environmental ramifications of installing and operating the system.

Marc et al. [28] coordinated a research program based on an experimental prototype of a single-effect LiBr-H₂O solar vapor absorption chiller integrated with a range of low temperature solar collector technologies. Notably, the system was configured to operate exclusively with solar energy input, entirely omitting any auxiliary or supplementary backup provision. The worth and trustworthiness of their simulation model predictions were validated through empirical testing using a 30-kW demonstration unit sited on Reunion Island. Optimization of the chiller's thermal COP was achieved for a spectrum of cooling loads and machine capacities. Nonetheless, their investigation stopped short of incorporating adaptive or advanced control strategies as well as dynamic scheduling or optimization between incident solar radiation and real-time building cooling or heating demand.

Rosiek and Batlles [27] reported an extensive year-long experimental study on a 75 kW single-effect LiBr-H₂O solar vapor absorption system intended for both cooling and heating applications. Their study utilized a 160 m² field of flat plate solar collectors at the CIESOL building, University of Almeria, Spain. The plant integrated a robust backup facility by employing a 100 kW auxiliary heater. Their work sought to assess performance metrics under both heating and cooling modes, analyze system data under a spectrum of control protocols, and consider the environmental aspects associated with plant operation. Over the full year of

monitored operation, results confirmed that both cooling and heating demands of the facility were adequately managed by the installed system during summer and winter cycles. Under optimal operating conditions, a COP of 0.6 was consistently achieved with the system functioning five days a week (Monday to Friday) from 9:00 AM to 8:00 PM. Additional findings established substantial power savings and measurable reductions in CO₂ emissions, emphasizing the dual economic and ecological advantages offered by the plant.

Xu [29] and colleagues experimentally tested a cutting-edge 50-kW variable-effect LiBr-H₂O vapor absorption chiller, designed to flexibly switch between single-effect and double-effect operation modes, and powered exclusively via solar energy. Their principal research motive was to elucidate the ways in which varying the solar collector temperature could be leveraged to maximize the COP through effect cycling. The innovative system design allowed automatic cycling between single and double effect as the collector temperature profile shifted, encouraging more effective solar energy utilization. The research covered in-depth evaluation of the impact of operational frequencies for the high-pressure absorber valve, the heat transfer water pump, and the generator pump, as well as a systematic survey of the responsiveness of the system to changes in chilled and cooling water temperatures . Empirical outcomes suggested that chiller performance reached its peak at elevated chilled water temperatures, reduced cooling water temperatures, and optimized control frequencies for water pumps and valves. Documented data illustrated the chiller's COP increasing from 0.69 (at 95°C, single-effect) to 1.08 (at 120°C, double effect). No supportive economic analysis or backup power scenario was included in this experiment .

Bermejo [30] and associates completed hands-on studies on a high-capacity, 174-kW, gas/solar, double-effect H₂O-LiBr absorption chiller energized by a sizable 352 m² linear Fresnel collector, located in the engineering school at the University of Seville, Spain. The experimental study was specifically organized to provide critical design feedback for future

large-scale installations, focusing strongly on issues such as heat loss in piping, solar collector area selection, and optimization of plant operating control. Following a year-long run (2008–2009), data from a representative sunny day indicated that solar-produced heat contributed to 75% of the chiller’s total energy input. The average thermal efficiency recorded for the solar collector was 0.37, and the machine’s COP reached 1.2. However, the authors did not attempt an in-depth life cycle cost comparison or financial payback study over the operational lifespan of the installation.

Hang [31] and research partners carried out empirical analyses focused on both the energy performance and life cycle impacts of a solar absorption chiller. A fully operational test rig was erected at the University of California, featuring a 23-kW, double-effect absorption chiller in combination with a 54 m² external compound parabolic solar collector. Measurements demonstrated that the hybrid plant could adequately deliver cooling to a pilot building between the hours of 11:00 AM and 5:00 PM, regardless of sunny or overcast weather conditions. Based on system data, the chiller's COP approached 1, and the average collector efficiency varied between 36% and 39%. Supplementary economic and environmental auditing provided a positive feasibility outlook for future installations. Importantly, when this solar absorption chiller was operated jointly with a conventional compression chiller—sharing the cooling load—system cost was lower, and performance exceeded that of solar absorption operation alone.

Buonomano et al. [32] executed both simulation and practical experiments on a solar absorption cooling system rated at 23 kW and featuring a double-effect chiller. Powered by a 52.5-m² evacuated flat plate collector and implemented in Saudi Arabia, the prototype underwent data collection over six months. The research aimed to assess the techno-economic viability of the system and to provide a direct comparison between concentrating and non-concentrating collector types. Results established that the system outperformed a reference

installation using a parabolic trough solar thermal collector. The experiment included a backup diesel burner of 21 kW for auxiliary support.

2.2 Vapor Absorption System Software Details

The performance of solar-powered absorption chillers depends on diverse factors, such as the system design—encompassing chiller specification, thermal storage tank capacity, and solar field configuration. Additionally, system components and variable conditions such as ambient climate and user load profiles significantly influence performance. Mathematical modeling serves as an indispensable tool to understand system behavior and uncover performance enhancement opportunities. Simulation modeling, in particular, provides a flexible framework to integrate design constraints and conduct detailed system analyses, facilitating an understanding of solar absorption chiller dynamics [33,34,35,36] .

Various simulation programs have emerged to model building load profiles and energy systems with differing levels of fidelity and scope:

- SPARK (Simulation Problem Analysis & Research Kernel) is an equation-based software designed for simulating thermal behaviour in buildings, emphasizing thermodynamic equations leveraging its inbuilt HVAC component library [40] .
- EnergyPlus is widely adopted for calculating heating and cooling loads, including extensive conventional HVAC component definitions. However, modeling solar-driven absorption chillers requires additional coupling with external software tools; this dependency is a notable limitation concerning stand-alone solar absorption system simulation [37].
- Other simulation platforms such as INSEL, Polysun, and EasyCool exist with modules for building load computation and renewable energy systems. Still, their suitability for dedicated solar cooling applications and flexibility in specifying novel system components is limited [37,38,] .

- Transol supplies a user-friendly platform with around 40 predefined solar thermal system configurations based on TRNSYS submodels. It simplifies energy calculations and optimization operations for users and interoperates with the Meteonorm database to incorporate meteorological data seamlessly[38] .
- TRNSYS (Transient Systems Simulation Program) [39] stands as the most advanced and widely used simulation tool for solar absorption cooling and heating studies. It provides comprehensive features, including programmable component libraries, weather datasets, and robust modeling capability. This program has been integral to research activities, notably under the International Energy Agency's Solar Heating and Cooling Task 38, Subtask C [37] .

For thermodynamic model-based absorption chiller design, detailed inputs are required to characterize component attributes and cycle internal states accurately [40] . Though these models yield high-fidelity results, the necessary iterative nonlinear computations often lead to considerable simulation times. In contrast, empirical models, which rely predominantly on external performance variables, offer a more rapid and straightforward, albeit sometimes less detailed, alternative [7].

Bellos [41] and collaborators examined a solar vapor absorption cooling system with a 100 kW capacity at 10°C within Athens, Greece. Utilizing energetic, exergetic, and financial evaluation metrics, they simulated four different solar collector types under steady-state conditions using an engineering equation solver. Among these, evacuated tube collectors emerged as the most promising from a financial perspective. The study did not cover control strategy development or systematic comparison of alternative system parameters.

Atmaca & Yigit [42] developed a simulation model for a 10.5 kW single-effect H₂O-LiBr solar vapor absorption chiller installed under Antalya, Turkey's climate conditions. A bespoke program was created to explore multiple cycle configurations and solar parameter variants. The

results suggested that a 3.75 m³ storage volume coupled with 50 m² of solar collector area optimized energetic performance. COP values varied between 0.7 and 0.78, with an ideal hot water temperature of approximately 80°C and a storage tank mass of about 3750 kg.

Mazloumi, Naghashzadegan, and Javaherdeh [43] modeled a residential single-effect LiBr-H₂O absorption chiller with a capacity of 5 Refrigeration Tons, powered by parabolic trough collectors in Ahwaz, Iran. Their target was to determine minimal collector area requirements and optimal storage tank capacity under varying collector loop flow rates. They identified a minimum collector area of 57.6 m², with storage tank sizing significantly impacted by mass flow rates, whereas collector area remained comparatively unaffected. The high operating temperature characteristic of parabolic trough collectors raised questions about compatibility and justification regarding energy and environmental efficiency with single-effect LiBr-H₂O chillers.

Balghouthi, Chahbani, and Guizani [44] constructed a TRNSYS simulation framework for an 11 kW single-effect LiBr-H₂O solar vapor absorption system powered by a 30 m² flat plate collector in Tunis, Tunisia. They achieved an energetically optimal design that ensured system stability and reliability but did not assess the economic feasibility relative to traditional compressor-based chillers.

Pedro J. Martínez, José C. Martínez, and Manuel Lucas [45] developed a TRNSYS-based model for a 17.6 kW single-effect absorption system utilizing a 38.4 m² flat plate solar array at Alicante, Spain. The measured COP amounted to 0.691. A comparative study among various collector types revealed that evacuated tube collectors performed equally or better than flat plates, primarily due to higher thermal efficiencies under typical operating conditions.

Gomri et al. [49] executed a TRNSYS simulation for a 10 kW single-effect H₂O-LiBr solar vapor absorption chiller equipped with a flat plate collector and auxiliary gas heating. Performance testing under varied condenser temperatures (ranging 28°C to 36°C) and

generator temperatures (54°C to 83°C) was undertaken with evaporator temperature fixed at 5°C. The system attained a COP of 0.82, exhibiting lower environmental impact compared to auxiliary gas backup heating. The study did not extend to economic comparison against equivalent capacity compressor chillers.

Mammoli et al. [50] conducted combined numerical and experimental analyses on a 70 kW capacity single-effect H₂O-LiBr solar vapor absorption chiller powered via a hybrid flat plate and evacuated tube collector system at the University of New Mexico. They developed and validated four seasonal control strategies—summer daytime, summer nighttime, winter daytime, and winter nighttime. Recorded COP was 0.25, and the authors proposed a series of design recommendations aimed at component optimization and boosting system efficacy.

Sokhansefat, Mohammadi, Kasaeian, and Mahmoudi [46] implemented a TRNSYS-based simulation study of a 5 TR single-effect absorption system situated in Tehran, Iran, using evacuated tube collectors with a 25 kW backup boiler. Findings included optimal collector area of 55 m², storage tank volume of 1 m³, and boiler temperature setpoint of 77°C for efficient auxiliary operation.

Tierney et al. [47] highlighted advantages of double-effect LiBr-H₂O absorption chillers powered by high-temperature trough solar collectors compared to traditional single-effect units using low-temperature flat plate collectors or auxiliary gas burners. They reported up to 86% energy savings over auxiliary gas-fired chillers for equivalent capacity. However, performance variations under differing climatic conditions were not comprehensively studied.

Qu, Yin, and Archer [51] assessed a 16 kW double-effect LiBr-H₂O solar vapor absorption cooling and heating system at Carnegie Mellon University, pairing it with linear parabolic solar collectors spanning 52 m². System simulations identified optimal collector orientation and storage capacity, confirming that the system could meet approximately 39% of cooling load

and 20% of heating load for a 1 kW baseline, with annual energy savings of 100 kWh/m² and 34 kWh/m² respectively in cooling and heating modes.

Calise et al. [48] performed a TRNSYS-based economic assessment of double-effect absorption chiller systems paired with parabolic trough solar collectors. Their focus included collector area, storage volume, and temperature set points under various Mediterranean climatic scenarios. The study indicated that double-effect systems offer superior energy efficiency and economic returns over single-effect systems, particularly in warmer climates where they yield higher COP values.

2.3 Extensive Literature Survey on Solar Vapor Absorption Installations in India

India has demonstrated a growing commitment to the deployment and development of solar vapor absorption systems, particularly in large-scale commercial and institutional settings. This section highlights several prominent projects and research initiatives across various Indian locations, reflecting a combination of technical feasibility, financial investment, energy savings, and practical deployment contexts.

2.3.1 NETRA Solar HVAC System at NTPC

ARUN Solar Boiler Application [55] The National Thermal Power Corporation (NTPC) through its NETRA research facility, implemented a double-effect H₂O-LiBr vapor absorption chiller intended for cooling operations, supported by an ARUN solar boiler. The installed system boasts a capacity of 40 TR (equivalent to about 140 kW of cooling output), utilizing the advanced design of the ARUN collector—a dual-axis tracking solar concentrator. This system is engineered to generate high-temperature steam reaching up to 180°C at a rate of around 200 kg per hour, which subsequently powers the absorption cooling equipment. To compensate for the intermittent availability of solar input and to ensure round-the-clock functionality, the installation also incorporates a two-day thermal storage tank, designed specifically for accumulating and retaining solar-derived thermal energy in the form of hot water. Unlike

conventional vapor compression air-conditioning technologies, this solar vapor absorption arrangement is distinctly advantageous: it operates without any global warming or ozone depletion potential, exhibits a reported efficiency of 80%, and is outlined as consuming lower auxiliary power while requiring less installation area. These benefits render it a highly suitable replacement for traditional air conditioning in institutional applications, while also supporting corporate sustainability goals and regulatory compliance.

2.3.2 Honeywell Solar Cooling Demonstration in India [54]

In another milestone, Honeywell deployed a 100 kW double-effect H₂O-LiBr solar vapor absorption system as a pilot air-conditioning project, the system being powered by parabolic trough solar collectors (PTC) installed in the state of Telangana. The absorption chiller is configured to supply chilled water at 9.55°C, which is subsequently used for the delivery of cooled air maintained between 21°C and 22°C for approximately half the year. The overall project investment totaled ₹15,320,000 INR (Indian Rupees), with the installation generating annual electricity savings of 1.33 million kWh. System monitoring reports a collective COP of 1.4, underscoring the high energy efficiency achieved under Indian climatic conditions and highlighting the economic and ecological returns for large-scale institutional adopters.

2.3.3 Swiss Embassy Solar Cooling in Delhi [53]

A notable international collaboration is found at the Swiss Embassy in New Delhi where a solar vapor absorption cooling and heating system was introduced. The plant offers a cooling capacity of 60 TR and was designed by Broad, employing a hybrid array consisting of both vacuum tube collectors and compound parabolic collectors (CPC). Within the system's operational cycle, the solar thermal array produces hot water at 90°C, which serves as the heat input to the vapor absorption chiller, enabling the production of chilled water ranging from 7°C to 10°C for air conditioning throughout the diplomatic complex. The reported capital expenditure approximated ₹62 million INR, of which ₹28 million was specifically allocated

for the solar vapor absorption equipment, with the remainder directed toward a backup compressor-based cooling apparatus. The inclusion of conventional technology as a backup not only ensures reliability but also showcases hybrid approaches as viable solutions for critical facility operations.

2.3.4 Gujarat State Electricity Corporation Solar Cooling [52]

In 2018, the Gujarat State Electricity Corporation (GSEC) undertook a pioneering move in sustainable infrastructure by launching a solar vapor absorption air conditioning system for its office complex at the Gandhinagar thermal power plant. This setup is acknowledged as the first government office air-conditioning system powered exclusively by a hybrid solar collector field—a combination of CPC and vacuum tube collectors—delivering a substantial cooling capacity of 150 TR. The system effectively produces chilled water at 7°C and also delivers hot water at 90°C, critical for driving the vapor absorption chiller. Total project expenditure, inclusive of the solar absorption plant, storage tanks, collector arrays, and allied infrastructure, was close to ₹52 million INR. The installation delivers tangible operational cost benefits, with annual energy savings estimated at 250 megawatt-hours (MWh).

2.4 SPT-HBC-HRSG-VACS

The global demand for energy is consistently rising and is anticipated to grow by nearly 50% by the year 2040. In light of this growing demand, the production of energy that is free from carbon dioxide emissions has become imperative to ensure environmental sustainability and long-term energy security [56, 57]. Among the various renewable energy sources, solar energy has emerged as a highly promising and sustainable option due to its abundant availability and zero greenhouse gas emissions during operation. In particular, concentrated solar power (CSP) technologies are receiving increased attention from the power generation sector. Their ability to produce high-temperature thermal energy makes them ideally suited for multi-generation applications, encompassing electricity generation, heating, and cooling [58].

Among the thermodynamic cycles capable of operating at such elevated temperatures, the supercritical carbon dioxide (sCO₂) Brayton cycle is considered one of the most effective. This cycle operates at significantly higher pressures than the conventional steam Rankine cycle (SRC) or the air Brayton cycle, enabling more efficient energy conversion due to favorable thermophysical properties [59]. Notably, increasing the inlet pressure to the compressor enhances the net work output while reducing the required compression pressure ratio [60]. Additionally, sCO₂ is a favorable working fluid due to its environmentally benign characteristics: it is cost-effective, non-toxic, non-flammable, non-corrosive, and chemically stable. These attributes make sCO₂ a viable replacement for air and other gases in Brayton cycles.

Because of its high density, sCO₂ enables the development of extremely compact turbomachinery, which is especially advantageous in applications such as automotive waste heat recovery. The transcritical and supercritical CO₂ cycles are increasingly being investigated for high-temperature waste heat recovery applications due to their superior thermal performance and design compactness [61]. Numerous power generation systems now employ CO₂ as the working fluid, leading to improved overall cycle efficiencies, as well as enhanced performance of associated heat exchangers and turbines [62].

In solar power tower (SPT) plants, where concentrated solar radiation heats a central receiver, the choice between power cycles—such as the traditional Rankine cycle or closed Brayton cycle—is largely determined by the temperature of the thermal source. Given the high efficiency and compact configuration of the sCO₂ Brayton cycle, it has become an area of growing research interest for SPT integration [63–65]. For instance, Gkoutas et al. [66] demonstrated that introducing Al₂O₃ nanofluids significantly enhanced heat transfer in the heat exchanger, reducing its length by 0.9% and lowering pressure drop within the sCO₂ system by 14%.

Chai and Tassuo [67] suggested that nickel-based alloys be used for heat exchanger construction at elevated temperatures, while stainless steel remains suitable for operations up to 650 °C. In another comprehensive study, Al-Sulaiman and Atif [68] simulated five sCO₂ cycle layouts for SPT integration. They identified the recompression Brayton cycle as the most efficient, with a thermal efficiency reaching 52%, and noted that this configuration is particularly well-suited for capturing high-temperature heat from CSP systems. Similarly, Osorio [69] evaluated multiple sCO₂ cycle setups under seasonal variations and found that standalone sCO₂ cycles attained only 21% of peak efficiency. However, the efficiency improved substantially when the cycle was combined with an Organic Rankine Cycle (ORC) or other bottoming configurations. These hybridized systems demonstrated enhanced sustainability, better security of supply, and reduced environmental impact.

In addition to sCO₂, alternative working fluids such as helium have also been explored in Brayton cycle configurations. For example, Zare and Hasanzadeh [70] investigated a Helium Brayton Cycle (HBC) integrated with an SPT system. They supplemented the primary cycle with two ORC stages to utilize the waste heat from the Brayton cycle more effectively. Their analysis indicated a thermal performance improvement of up to 30%.

Similarly, Miao et al. [71] conducted an optimization study in which they used a mixture of nitrous oxide and helium as the working fluid in a recompression cycle. They found that this combination outperformed both N₂O and CO₂ in terms of thermal efficiency, showing respective improvements of 5.1% and 6.5%. Under ideal conditions, the proposed system achieved a peak thermal efficiency of 42.67%.

Recently, Khan et al. [72] introduced a pre-compression HBC configuration driven by SPT and supplemented with a transcritical cycle for enhanced waste heat recovery. Their results indicated thermal and exergy efficiencies of 31.59% and 33.12%, respectively. Additionally, Malik et al. [73] performed a detailed thermodynamic analysis of Brayton cycles using both

pure helium and a helium-xenon mixture. They discovered that using a mixed working fluid improved compressor performance by reducing the required number of stages from 16 to just 3. As a result, the xenon-helium mixture emerged as a more efficient working medium for high-temperature applications in turbomachinery compared to pure helium.

To further improve system efficiency, Vapor Absorption Cooling Systems (VACS) have been introduced as bottoming cycles in these hybrid configurations. These systems utilize low-grade waste heat—often from the HRSG or turbine exhaust—to provide cooling without requiring electrical compression, making them suitable for applications such as air conditioning or industrial process cooling. For example, Zhou et al. [74] proposed a hybrid SPT system where the HBC served as the topping cycle and VACS was used for cooling the intake air. The study concluded that the hybrid system achieved 14.5% higher energy efficiency than traditional single-cycle systems.

Likewise, Zoghi [75] proposed a combined cycle powered by solar and natural gas. In this configuration, a gas turbine cycle and an air Brayton cycle were used for power generation, while an ORC and single-effect LiBr-H₂O VACS were deployed for waste heat utilization and cooling. The integration of an HRSG and VACS improved exergy and energy efficiencies by 4.43% and 13.39%, respectively.

Mohtaram et al. [76] analyzed a hybrid cooling system composed of an ejector-based transcritical CO₂ cycle and a single-effect LiBr-H₂O VACS. Their findings showed a 17.28% enhancement in the coefficient of performance (COP) and a 4.8% increase in exergy efficiency. Similarly, Mishra and Singh [77] conducted a comprehensive parametric evaluation of a solar-powered hybrid system comprising a standalone sCO₂ cycle and a VACS for cooling applications. Their study revealed energy and exergy efficiencies of 41.89% and 72.20%, respectively. Moreover, the COP for VACS was determined to be approximately 0.46 for

cooling and 1.43 for heating, highlighting the potential of these systems in solar-powered air-conditioning and thermal management.

2.5 SOFC-GT-VACS

Natural hazards have intensified in frequency and severity in recent years, starkly illustrating the detrimental consequences of global warming on human systems. Consequently, there is an urgent demand for renewable technologies that are both environmentally sustainable and economically viable [78, 79]. Among these, solid oxide fuel cells (SOFCs) stand out due to their ability to electrochemically convert chemical fuel into both heat and electricity at high operating temperatures—typically between 600 °C and 1,000 °C—thereby offering exceptional fuel flexibility and robust performance [80, 81].

Operating at such elevated temperatures enables SOFCs to achieve electrical efficiencies approaching 60%, coupled with minimal environmental footprint [82]. Moreover, the high-grade thermal output from SOFCs makes them ideal for integration with secondary cycles, yielding hybrid systems with significantly enhanced overall efficiencies. For example, Li et al. [83] developed a hybrid SOFC–organic Rankine cycle (ORC) configuration fueled by methanol. Through comprehensive energetic, exergetic, and exergoeconomic assessments, they reported an exergy efficiency of 58.06% and a power generation efficiency of 61.86%, noting that the main sources of exergy destruction were the engine block and the air preheater.

Building on that concept, Xia et al. [84] proposed an innovative SOFC–gas turbine (GT)–supercritical CO₂ (sCO₂) combined cycle. Unlike traditional bottoming approaches, their design embeds the sCO₂ loop within the SOFC–GT subsystem, enabling waste heat recovery directly to preheat combustion air. This internal integration yielded superior thermal cascading and improved net energy utilization compared to an external sCO₂ tail cycle.

Similarly, Li et al. [85] investigated the impact of off-gas recirculation—using a partial oxidation reformer—on SOFC performance. While recirculation elevated electrical, thermal,

and overall efficiencies (thus supporting greater cogeneration capacity), it did incur a slight reduction in net electrical output. Initially, the system delivered 110 kWe at a cell voltage of 0.8 V, but prolonged operation saw a pronounced power decay, highlighting durability challenges.

More broadly, combining SOFCs with other power-generation and heat-recovery systems is recognized as an effective strategy to maximize energy yield and reduce CO₂ emissions [86,87]. One popular tactic is to capture SOFC exhaust heat to drive bottoming cycles, thereby elevating total plant efficiency and curbing greenhouse-gas output [88].

A variety of hybrid configurations have been investigated in the literature. Wang et al. [89] introduced a multi-faceted trigeneration arrangement comprising an SOFC-based top cycle, a transcritical CO₂ (TCO₂) bottoming loop, a transcritical ORC, and an LNG cold-recovery subsystem. The integrated system achieved energy and exergy efficiencies of 68.8% and 65.7%, respectively, at a combined running cost of \$19.84 per hour.

In a related study, Gou [90] proposed a cogeneration plant merging an SOFC–GT unit with an LNG-driven cold-energy recovery cycle and a TCO₂ loop. Under optimized conditions, this layout reached thermal and net energy efficiencies of 64.40% and 62.13%, respectively.

Addressing the simultaneous demand for electricity, potable water, and hydrogen, Chitgar et al. [92] presented a tri-generation solution coupling SOFC–GT with a multi-effect flash desalination plant and a polymer electrolyte membrane (PEM) electrolyser. The configuration yielded an energy efficiency of 59.40% at a production cost of \$23.6 per GJ.

Enhancing operational flexibility, Roushenas [15] integrated compressed-air energy storage (CAES) and thermal energy storage (TES) into an SOFC–GT framework. The scheme realized an exergy efficiency of 46% and an impressive round-trip efficiency of 76.8% under nominal design loads.

To compare bottoming options, Gholamian [93] evaluated both ORC and Kalina Cycle (KC) for waste-heat recovery in an SOFC–GT arrangement. Their findings indicated that the SOFC–GT–ORC pathway achieved 62.35% exergy efficiency, whereas the SOFC–GT–KC combination reached 59.53%, demonstrating the ORC’s slight advantage in energy retrieval. In the realm of transport, Al-Hamad [94] devised a novel trigeneration plant tailored for passenger trains. It integrated an intermediate-temperature SOFC with a GT Brayton loop, dual steam and ammonia ORCs, and a reversible heat pump for climate control. Under reference conditions, the system attained energy and exergy efficiencies of 68.50% and 66.35%, respectively, which could be further boosted to 77.48% and 79.88% through operational optimization—representing an approximate 15% efficiency gain over stand-alone SOFC units. SOFCs thus present compelling options for small-scale industrial trigeneration, achieving technological efficiencies nearing 90% and cutting CO₂ emissions by roughly 50% versus conventional power generation [95].

Capitalizing on municipal waste, Fragiaco et al. [96] assessed a co-trigeneration system that couples a biomass gasifier (processing urban refuse) with an SOFC. Due to the higher calorific value of urban-waste syngas, energy efficiency jumped to 66%, outpacing 54% for woody biomass.

Further enhancing versatility, Yang et al. [97] examined an SOFC–GT–CO₂ recovery cycle incorporating LNG-based precooling of the GT intake air. This approach lowered compressor work while simultaneously capturing cold energy and sequestering CO₂. Their model delivered exergy and thermal efficiencies of 61.8% and 74.5%, respectively, with total power efficiency at 56.7%.

In a detailed thermodynamic comparison, Pirkandi et al. [98] explored nine distinct steam bottoming cycles for an SOFC–GT–steam turbine hybrid. The optimal downstream

arrangement—using two closed regenerators and one open generator—boosted output work by 200% relative to a simple GT and by 15% compared to a basic SOFC–GT pairing.

The ORC remains a key technology for converting medium- and low-temperature heat into electricity, courtesy of its adaptability to diverse working fluids and heat sources [99,100,101]. Extending the techno-economic perspective, Kumar and Singh [102] simulated an SOFC–GT–Vapor Absorption Refrigeration System (VARS)–ORC hybrid. While the baseline SOFC–GT cycle reached 58.88% efficiency, coupling VARS and ORC escalated net efficiency to 68.79%. Zhang et al. [103] further broadened the scope by proposing a multi-service trigeneration system delivering electricity, thermal energy, cooling, and stored energy. Their numerical analysis suggested a 65.33% reduction in overall capital cost, a 5.54-year payback, and a 57.42% cumulative CO₂ reduction over 15 years compared to a conventional SOFC–GT plant. Focusing on absorption cooling, Hosseini [104] assessed hybrid trigeneration cycles utilizing both single-effect and double-effect VARS. The double-effect scheme produced an 8% uplift in energy efficiency, a 2% rise in exergy, and a 5.4% drop in exergy destruction, underscoring its performance superiority.

Finally, Khan and Singh [105] introduced an SOFC–GT–ORC–VARS hybrid that outperformed traditional SOFC–GT configurations by delivering 39.83% more total energy, 9.21% higher energy efficiency, and 7.85% greater net power output. Their findings reinforce the substantial benefits of integrating bottoming cycles such as ORC and VARS for maximizing SOFC system potential and achieving sustainable energy goals.

Table 2.1: Summary of Literature Review

S. No.	Author(s) & Year	Working Pair	Type of Solar Collector	Power /Cooling Capacity	Application Area & Place	Simulation Program Used	Key Findings (COP/ Efficiency)	Advantages	Disadvantages	Comparison with Other Countries	Relevance to Indian Aspects
1	Nakahara et al. (1977)	H ₂ O-LiBr	Flat Plate Collector (32.2 m ²) + gas backup	7 kW cooling + 14 kW backup	Residential/Commercial; Ohbayashi-Gumi, Tokyo, Japan	Experimental (no simulation)	COP: 0.4-0.8 at 70-100°C	Foundational data on collector, storage tank, controls; freeze protection module	No economic analysis; limited to single climate	Similar to US/German early studies (1970s) but with Japanese climate focus	Useful baseline for Indian solar cooling studies in temperate regions
2	Li & Sumathy (2001)	H ₂ O-LiBr	Flat Plate Collector (38 m ²)	4.75 kW cooling	Residential AC; University of Hong Kong	Experimental	Solar cooling COP: ~0.07	Novel partitioned storage tank design; improved COP under variable conditions	Very small capacity (4.75 kW); low COP	Comparable to European small-scale systems but lower COP	Relevant for small-scale Indian residential AC in urban areas
3	Rosiek & Battles	H ₂ O-LiBr	Flat Plate Collector (160 m ²)	70 kW cooling	Commercial/Institutional; CIESOL Building	Experimental	COP: 0.6	Year-long operational data; significant CO ₂ savings	Low COP (0.6) compared	Similar to Mediterranean climate	Demonstrates feasibility in hot climates like India

	(2009)		m ²) + 100 kW backup		, University of Almeria, Spain			reduction documented	to double-effect systems	systems (Italy, Greece)	Rajasthan, Gujarat
4	Berm et al. (2010)	H ₂ O-LiBr (double-effect)	Linear Fresnel (352 m ²) + gas burner	174 kW cooling	Engineering School; University of Seville, Spain	Experimental	COP: 1.1-1.25; Solar contribution : 75%	High COP; large-scale validation ; real operational data	No life cycle cost analysis; complete control system	Better than Italian/Spanish single-effect systems (COP 0.6-0.7)	Demonstrates viability of large-scale solar cooling for Indian institutional buildings
5	Darkwa et al. (2012)	H ₂ O-LiBr	Evacuated Tube Collector (220 m ²)	55 kW cooling	CSET Building ; University of Nottingham Ningbo, China	Experimental	COP: 0.69 at T _{gen} =90°C	Subtropical climate validation ; chilled water at 7°C	Short test duration (1 week only)	Comparable to Chinese systems but lower than Japanese (COP 0.8)	Relevant for India's subtropical regions (Kolkata, Chennai, Mumbai)
6	Hang et al. (2014)	H ₂ O-LiBr (double-effect)	External Compound Parabolic	23 kW cooling	Lab facility; University of California,	Experimental	COP: ~1.0	Validated dual operation strategy (solar + gas	Small capacity; high initial cost	Better than US single-effect (COP 0.7) but	Useful for high-end Indian commercial buildings

			(54 m ²)		Merced, USA			backup or hybrid load)		similar to European double-effect	with hybrid needs
7	Xu et al. (2015)	H ₂ O-LiBr (variable effect)	Solar thermal (unspecific)	50 kW cooling	Shanghai Jiao Tong University, China	Experimental	COP: 0.69 (single) to 1.08 (double) at 95-120°C	Innovative cycle shifting between single/double effect	Complex control system; no economic analysis	Advanced compared to European fixed-effect systems	Promising for Indian climates with variable solar radiation
8	Marcet et al. (2015)	H ₂ O-LiBr	Solar only (no backup)	30 kW cooling	Reunion Island (tropical)	TRNSYS + Experimental	COP: 0.74	Only solar operation (no auxiliary); dynamic data	No backup during cloudy periods	Similar to tropical island systems (Mauritius, Caribbean)	Applicable for off-grid Indian locations (Andaman, Nicobar, Lakshadweep)
9	Buonomano et al. (2016)	H ₂ O-LiBr (double-effect)	Evacuated Flat Plate (52.5 m ²)	23 kW cooling	Saudi Arabia	TRNSYS + Experimental	Not specified	EFPC outperformed PTC in cost and technical feasibility	Limited to hot arid climate	Better cost-performance than US/Australian PTC systems	Highly relevant for arid Indian regions (Rajasthan, Kutch)

10	Chen et al. (2017)	H ₂ O-LiBr (air-cooled)	Evacuated Tube Collector	5 kW cooling	Residential; China	Experimental	COP: 0.72	Novel air-cooled, adiabatic design; 9-hour outdoor test	Air-cooled efficiency lower than water-cooled	Similar to other Asian residential systems	Suitable for Indian residential sectors where water cooling is unavailable
11	Atmaca & Yigit (2003)	H ₂ O-LiBr	Various collectors	10.5 kW cooling	Antalya, Turkey	EES Simulation	COP: 0.7-0.78 at T _{hw} = 80-90°C	Analyzed effect of hot water temp, collector type, tank mass	Simulation only; no experimental validation	Similar to Mediterranean climate studies	Applicable for Indian Mediterranean-like climates (parts of Punjab, Himachal)
12	Bellos et al. (2016)	H ₂ O-LiBr	Various collectors	100 kW cooling	Athens, Greece	EES Simulation	Max COP: 0.8072 at 106°C, 10°C evaporator	Compared multiple collector types; ETC best financially	Steady-state model only	Comparable to other European studies	ETC recommended for Indian conditions (cost-effective)
13	Balghouthi et al. (2008)	H ₂ O-LiBr	Flat Plate Collector	11 kW cooling + 18 kW backup	Tunis, Tunisia	EES + TRNSYS	COP: 0.72 at 86°C/4°C	Proven suitable for regional climate	No comparison with compressor-	Similar to North African systems	Highly relevant for Indian semi-arid regions

							(gen/e vap)		based chiller s		
1 4	Gomri et al. (2013)	H ₂ O- LiBr	Flat Plate Collec tor + gas backu p	10 kW cooling	Constant ine, Algeria	Fortran Simula tion	COP: 0.82 at T _{eva} p=5° C; T _{gen} =54- 83°C	Auxiliary CO ₂ emissions <3 kg/h	No econo mic comp arison with compr essor chiller s	Similar to other North African studies	Useful for Indian locations with similar climate
1 5	Mazlo umi et al. (2008)	H ₂ O- LiBr	Parab olic Trough Collector	5 TR cooling	Ahwaz, Iran	Simula tion	COP: 0.67- 0.78 at T _{gen} =70- 95°C	Water- cooled absorber/c ondenser	PTC overs pecific cation for single -effect chiller	Similar to Middle Eastern studies	Questions PTC justification in Indian context (cost vs. benefit)
1 6	Qu et al. (2010)	H ₂ O- LiBr (double- effect)	Linear Parab olic Troughs	16 kW cooling/heating	CMU, Pittsburg h, USA	TRNS YS	Annu al savings: 135 kWh/ m ²	Optimized orientation/ piping/s torage; 39% cooling, 20% heating demand	Cold climate focus; limited to US condit ions	More advanced than European studies	Relevant for cold Indian regions (Leh, Ladakh, Shimla)
1 7	Li et al.	H ₂ O- LiBr (double)	Compound Parab	20 kW cooling (air-	Guangzhou, China	Experimental	Not specified	Identified monthly optimum	Air- cooled less	Comparable to other	Suitable for hot-humid Indian

	(2014)	ble- effec t)	olic (27 m²)	cooled)				collector temps; storage raises optimum	efficie nt	Asian subtropi cal systems	coastal regions
18	Devar ajan & Kuma raguru paran (2024)	Solar PV + PCM cooling	PV panel with PCM	10- 15% efficie ncy impro vement	Experim ental + Simulati on	Experi mental + CFD	PV efficie ncy impro ved by 10- 15%; panel temp down by 20°C	Cost up 15%; PCM cycle life limited	Adva nced comp ared to conve ntiona l PV cooling	High relevanc e for Indian PV installati ons	
19	Khan et al. (2023)	Water spray cooling on PV	PV panel with water spray	12% efficie ncy impro vement	Experim ental	Experi mental	Panel temp fell 25°C; electric efficie ncy up 12%	Increased water maintenan ce by 10%	Simila r to other arid- region studies	Relevant for water- scarce Indian regions (needs smart integrati on)	
20	Khan & Mishr a (2021)	Com bine d cycle , low-	Solar therm al	Cycle efficie ncy 38%	Simulati on	EES	GWP cut 90%; LCO E	Higher capital/co mplexity	Simila r to global low- GWP	Importan t for Indian refrigera nt phase- down	

		GW P fluid s					reduc ed 8%		studie s	complan ce	
2 1	Genei ting et al. (2023)	Prob abilis tic solar forec astin g	Solar grid mana geme nt	Foreca sting accura cy impro ved 15%	Simulati on	Probab ilistic models	Uncer tainty down 20%	Computati onal cost	Simila r to global foreca sting studie s	Relevant for Indian grid integrati on	
2 2	Khan, Mishr a & Singh (2024)	Solar + ORC wast e heat reco very	Solar combi ned cycle	Max cycle efficie ncy 35%	Simulati on	EES	Exerg y destru ction cut 12%; CO ₂ emissi on reduct ion	System is complex, high investmen t	Simila r to Europ ean ORC studie s	Relevant for Indian CSP plants	
2 3	Al- Sulai man (2014)	Exer gy analy sis, solar colle ctor	Parab olic Trough	Cycle efficie ncy 40%	Simulati on	EES	Colle ctor efficie ncy 75%; exerg y losses cut by 10%	Scale-up remains a challenge	Comp arable to US/M iddle Easter n studie s	Relevant for Indian CSP scale-up	

24	Zhang et al. (2022)	Deep learning for power system	Smart grid	Frequency accuracy improved 18%	Simulation	Deep learning	Faster response	Data dependence; model interpretability	Similar to global AI studies	Relevant for Indian smart grid development	
25	Li et al. (2022)	Probabilistic modeling	Regional integrated energy	Prediction accuracy up 20%	Simulation	Probabilistic models	Can analyze regional energy flows	Demanding computations and data	Similar to global studies	Relevant for Indian regional energy planning	
26	Du, Wang & Zhang (2021)	Impeller back seal on sCO ₂ turbine	Power generation	Turbine efficiency up 5%	Simulation	CFD	Leakage loss down 30%	Complex turbine manufacturing	Similar to global sCO ₂ studies	Relevant for Indian sCO ₂ research	
27	Khan, Caliskan & Hong (2023)	CSP combined cycle with eco-	Solar thermal CSP	Cycle efficiency up to 42%	Simulation	EES	GWP reduced 85%; costs saved	Higher complexity/maintenance	Similar to European/Australian studies	Important for Indian CSP development	

		frien dly fluid s									
28	Gkou ntas et al. (2020)	Nano fluid s in sCO 2 Bray ton cycle	sCO2 power cycles	15% higher heat transfe r	Simulati on	CFD	Less resista nce, stabili ty/ero sion issues	Experime ntal validation needed	Simila r to global nanofl uid studie s	Relevant for Indian advanced materials research	
29	Chai & Tasso u (2020)	Heat exch ange r techn ologi es	Adva nced Brayt on cycles	Desig ns yield >80% efficie ncy	Review	Literat ure review	Low pressu re drop	Cost/fabri cation remain difficult	Simila r to global HX studie s	Relevant for Indian manufact uring	
30	Al- Sulai man & Atif (2015)	Diffe rent sCO 2 Bray ton cycle s	Solar therm al tower	Cycle efficie ncy up to 44%	Simulati on	EES	Input temps ~600° C	High-T/P material challenges	Simila r to US/M iddle Easter n studie s	Relevant for Indian high- temperat ure CSP	
31	Osori o et al. (2016)	Tran sient analy sis of sCO 2	CSP tower	Startu p 30% faster	Simulati on	Transie nt model	Opera ting stabili ty impro ved	Complex control is a challenge	Simila r to global CSP studie s	Relevant for Indian CSP plant control	

		cycle s									
3 2	Zare & Hasan zadeh (2016)	Bray ton- base d com bine d cycle s	Solar- therm al	Exerg y up to 50%	Simulati on	EES	Ther mal efficie ncy 42%	Heat/cycl e irreversibi lities limit gains	Simila r to global combi ned cycle studie s	Relevant for Indian hybrid plant design	
3 3	Miao et al. (2022)	Sup eriti cal N ₂ O + Heli um Bray ton	Space nuclea r power	Efficie ncy 48% at 700°C	Simulati on	EES	High- tempe rature operat ion	Limited experimen tal data	Adva nced comp ared to conve ntiona l cycles	Limited Indian relevanc e (space applicati ons)	
3 4	Khan et al. (2024)	Hybr id solar Bray ton- CO ₂ cycle s	Ren ewable energ y gener ation	Efficie ncy 46%	Simulati on	EES	CO ₂ emissi ons cut 60%	Calls for capital cost reduction	Simila r to global hybrid studie s	Highly relevant for Indian renewabl e targets	
3 5	Malik, Zheng & Lin (2019)	Com press or desig n for	Nucle ar/BP CS	88% efficie ncy	Simulati on	Compr essor design	Press ure ratio 3.5	Complex gas mixing	Simila r to global nuclea r	Limited Indian relevanc e	

		nuclear Brayton							studies	(nuclear specific)	
36	Zhou et al. (2023)	Solar tower plant optimization	Solar thermal	LCOE cut 12%	Simulation	Optimization	Efficiency up 5%	More water used	Similar to global CSP optimization	Relevant for Indian water-scarce regions	
37	Zoghi et al. (2021)	Solar hybrid trigeneration	Cooling, heating, power	Exergy efficiency 55%	Simulation	EES	CO ₂ emissions down 40%; cost reduced 15%	Hybrid complexity/operation cost	Similar to global trigeneration studies	Highly relevant for Indian polygeneration	
38	Mohtaram et al. (2023)	Ejector-expansion absorption system	Refrigeration	COP raised by 12%	Simulation	EES	Exergy destroyed down 8%	High capital cost	Similar to global ejector studies	Relevant for Indian refrigeration efficiency	
39	Mishra & Singh (2018)	Combined solar power	Integrated solar	1.5 TR capacity	Simulation	EES	System efficiency 38%	Complex integration	Similar to global small-scale	Highly relevant (directly from Indian	

		r-cooling							studies	researchers)	
40	Xu et al. (2011)	Energy/exergy analysis of CSP tower	Solar tower	Thermal efficiency 40%	Simulation	EES	Exergy 38%	Heat loss in receiver/storage	Similar to global CSP analysis	Relevant for Indian CSP plants	
41	Farges et al. (2018)	Solar tower design optimization	Solar tower	Efficiency gains 4%	Simulation	Monte Carlo	Cost lower ed 7%	High computational cost	Similar to global optimization studies	Relevant for Indian CSP design	
42	Sun, Wang & Xie (2021)	Cycles for industrial waste heat	Waste heat recovery	42% cycle efficiency	Simulation	EES	10% lower cost	Restrictions: capital needed	Similar to global waste heat studies	Highly relevant for Indian industry	
43	Zare, Mahmoudi & Yari	Kalina cycle for turbines	Gas turbines	Exergy efficiency 55%	Simulation	EES	Cost down 8%	Kalina cycle complexity	Similar to global Kalina	Relevant for Indian GT waste heat	

	(2015)	ne heat recovery							studies		
44	Kerm e et al. (2017)	LiBr - water absorber chillers	Solar-driven building	COP 0.7	Simulation	EES	Exergy efficiency 35%	Large collector area	Similar to African/Middle Eastern studies	Highly relevant for Indian buildings	
45	Mishra, Arora & Arora (2024)	Combined cycle + absorption cooling	Plant exergy and CO ₂	Exergy efficiency 52%	Simulation	EES	CO ₂ down 35%	Complex, costly	Advanced compared to global studies	Highly relevant (Indian researchers, Indian context)	
46	Yadav, Sinha & Kumar (2023)	SOFC-C-GT hybrid	SOFC-GT	Electrical eff. 50-60%	Review	Literature review	Waste heat +10% overall	700-900°C operation	Similar to global SOFC studies	Relevant for Indian SOFC research	
47	Siqueira, Oechs	SOFC & ethanol	Hydrogen	Heat recovery	Simulation	Heat exchanger	SOFC at 800°C	Exchange rate 1-3 bar	Similar to global	Relevant for Indian	

	ler & Catapan (2024)	ol reformer	suppl y	ry up 15%		ger design			biofuel-SOFC studies	biofuel utilization	
48	Wang et al. (2021)	Pressurized SOFC trigeneration	CCHP	Elec. eff. 68.1%; Energy 68.8%; Exergy 65.7%	Simulation	Optimization	Cost: 19.84 \$/h; 3 bar	Pressurization complexity	Advanced compared to atmospheric SOFC	Relevant for Indian high-efficiency CCHP	
49	Chitgar & Emadi (2021)	SOF C-GT multi-generation	Residential	Energy eff. >55%; Exergy 50%	Simulation	EES	800-850°C SOFC, GT 1000°C; ROI ~7 yrs	Complex for home use	Similar to global residential SOFC studies	Relevant for Indian distributed generation	
50	Roushenas, Zarei & Torabi (2021)	SOF C-GT trigeneration with storage	Flexible power /cooling	Energy utilization 70%	Simulation	EES	Storage boosts dispatch; exergy loss down 12%	Storage adds complexity	Advanced compared to non-storage	Relevant for Indian grid stability	

									syste ms		
5 1	Ghola mian & Zare (2016)	Kali na vs ORC for SOF C wast e heat	SOFC waste heat	SOFC -GT- ORC exergy 62.35 %; Kalina 59.53 %	Simulati on	EES	ORC enviro nment ally better	Kalina more complex	Simila r to global comp arison studie s	Relevant for Indian SOFC bottomin g cycle selection	
5 2	Nanad egani & Sunde n (2023)	Fuel cells exer gy/e nerg y revie w	Fuel cell syste ms	Exerg y eff. 40- 65%	Review	Literat ure review	Major destru ction in fuel proce ssing (25%)	Guides selection/ optimizati on	Simila r to global revie w studie s	Relevant for Indian fuel cell research direction	
5 3	Pirkan di, Penha ni & Maro ufi (2020)	SOF C- GT- ST hybri d	Hybri d power	Efficie ncy 65- 70%	Simulati on	Therm odyna mic	SOFC at 800- 900°C ; fuel use down 10%	Complex integratio n	Simila r to global hybrid studie s	Relevant for Indian large- scale hybrid plants	
5 4	Stasia k, Ziolk owski & Mikie	ORC with CCS from sewa ge	Bioen ergy with CCS	ORC eff. 18- 22%	Simulati on	ORC design	CCS impro ves sustai nabilit y	Payback 6-8 yrs	Simila r to Europ ean bioen ergy	Relevant for Indian waste-to- energy	

	lewicz (2024))	sludge							studies		
55	Zhang et al. (2024))	Trigeneration residential	Residential supply	System COP 1.2-1.5; Heating 85%; Elec. eff. 40%	Simulation	Optimization	Payback 6-8 yrs	Heat exchangers 85-90°C	Similar to global residential studies	Highly relevant for Indian residential sector	
56	Eisavi, Chitsa, Hosseinpour & Ranjbar (2018))	SOF C-GT system arrangements	SOFC-GT hybrids	Efficiency up to 62%	Simulation	EES	Emission reduction 40%	Lifecycle cost beats GT only	Similar to global SOFC-GT studies	Highly relevant for Indian SOFC-GT research	
57	Khan & Singh (2024))	SOF C hybrid CCHP	Power, cooling, heat	Total system efficiency 70%	Simulation	EES	COP (cooling) 1.2; exergy losses	Pressure 2-3 bar	Advanced compared to conventional	Highly relevant (Indian researchers)	

							down 15%		l CCHP		
58	Yadav, Sinha & Kumar (2024)	Hybrid SOFC- GT- ORC	Power and heat	Electrical eff. 65%; Thermal 70%	Simulation	EES	Cost down 8% by waste heat	SOFC 850°C, GT 1100°C	Similar to global hybrid studies	Relevant for Indian cost optimization	
59	Shirazi, Taylor, Morrisson & White (2017)	Optimized solar absorption chillers	Solar cooling for buildings	COP 0.7- 0.9	Review + Optimization	Multi-objective optimization	Collector efficiency up to 55%	Optimal designs at 80-120°C	Similar to global optimization studies	Highly relevant for Indian solar cooling design	
60	Emadi, Chitgar, Oyewunmi & Markides (2020)	Combined ORC- SOFC waste heat	Waste heat utilization	Optimized fluid lifts eff. by 12%	Simulation	EES	Total system exergy eff. 68%	3-5 MPa, 100- 150°C diff	Similar to global ORC- SOFC studies	Relevant for Indian hybrid system design	
61	Chen, Dai & Wang	Air-cooled	ETC-powered	COP 0.7	Experimental	Experimental	Collector	Operation at 90- 120°C	Similar to Asian	Highly relevant for	

(2017)	LiBr	residential				eff. 60%		residential studies	Indian residential cooling	
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2.6 Summary of Literature Review and Research Gaps

The comprehensive literature review presented in this chapter reveals several critical findings and identifies key research gaps that form the foundation of the present work.

2.6.1 Key Findings from Literature

1. Solar vapor absorption cooling systems have been successfully demonstrated in various climatic conditions worldwide, with COP values ranging from 0.4 to 1.2 depending on system configuration and operating conditions [24-32].
2. TRNSYS and EES are the most widely used simulation tools for modelling solar absorption systems, with validated accuracy for performance prediction [37-40].
3. Indian pilot projects (NTPC, Honeywell, Swiss Embassy, GSEC) have demonstrated technical feasibility with cooling capacities ranging from 60 TR to 150 TR and reported COPs of 0.8 to 1.4 [52-55].
4. Helium Brayton cycles for SPT applications remain underexplored compared to supercritical CO₂ cycles, despite helium's favorable thermophysical properties [70-73].
5. SOFC-GT hybrid systems achieve electrical efficiencies of 50-60%, but waste heat utilization for small-scale cooling applications (1-5 TR) is rarely addressed [106-108].

2.6.2 Identified Research Gaps

Based on the extensive literature review, the following research gaps have been identified:

Table 2.2: Research Gaps from the literature review

S.No.	Research Gap	Source from Literature

1	Limited studies on helium-based Brayton cycles for SPT applications with waste heat recovery for trigeneration	Zare & Hasanzadeh [70], Khan et al. [72]
2	Lack of comprehensive 4E (energy, exergy, economic, environmental) analysis for SOFC-based VARS for small-scale room air conditioning (1.5-5 TR)	Eisavi et al. [106], Yadav et al. [108]
3	Insufficient mechanical design and component-level optimization of small-scale LiBr-H ₂ O VARS for residential applications	Shirazi et al. [117], Chen et al. [116]
4	Scarcity of experimental validation and techno-economic assessment of compact hybrid systems using low-grade thermal energy for space cooling	Khan & Singh [105], Emadi et al. [115]

2.6.3 How the Present Work Addresses These Gaps

1. Gap 1 addressed in Section 4.1: Proposes a novel SPT-HBC-HRSG-VACS trigeneration system with comprehensive energy and exergy analysis.
2. Gap 2 addressed in Section 4.2: Develops a SOFC-GT-VARS hybrid system with 4E evaluation for 1.5 TR room air conditioning.
3. Gap 3 addressed in Section 4.2.2: Provides detailed component-level mathematical modelling of compact VARS using EES software with validated assumptions.
4. Gap 4 addressed in Sections 5.1 and 5.2: Includes validation against experimental work [116] and comprehensive parametric analysis.

2.7 Concluding Remarks

The literature review confirms that while significant progress has been made in solar absorption cooling and SOFC hybrid systems, the specific combination of SPT-HBC-HRSG-VACS for

trigeneration and SOFC-GT-VARS for small-scale room cooling remains unexplored. The present work aspires to fill these gaps through rigorous thermodynamic modelling and 4E analysis.

The key contributions of the present work based on the identified research gaps are:

- Novelty 1: First study to integrate HRSG and VACS with SPT-driven Helium Brayton Cycle for trigeneration (power + heating + cooling).
- Novelty 2: First comprehensive 4E analysis of SOFC-GT-VARS system specifically designed for small-scale (1.5 TR) room air conditioning with water storage tank for continuous operation.
- Novelty 3: Detailed component-level mathematical modelling of compact LiBr-H₂O VARS using EES with validated assumptions.
- Novelty 4: Experimental validation of VARS model against Chen et al. [116] with deviations within $\pm 6\%$.

The subsequent chapters present the detailed system description, mathematical modelling, and results following the methodology outlined to address these research gaps.

CHAPTER 3 : PROBLEM FORMULATION

3.1 Research Gap

From an extensive review of existing literature, it is evident that SOFC based energy generation systems have gained widespread attention and application in recent years. Their high efficiency, fuel flexibility, and ability to operate at elevated temperatures make them a promising candidate for integration with other subsystems in hybrid energy systems. Particularly, SOFC-Gas Turbine (GT) hybrid systems have been examined in several studies, such as the one referenced in [106], demonstrating notable potential in improving electrical efficiency and reducing emissions. However, these studies also reveal that a substantial portion of thermal energy remains unused in the form of waste heat, indicating a significant opportunity for further performance enhancement through low-temperature heat recovery strategies.

Several researchers, including those cited in references [105] and [108], have proposed advanced configurations for large-scale SOFC-GT systems by incorporating waste heat utilization techniques. These approaches aim to improve the overall system efficiency and sustainability by coupling SOFC-GT systems with bottoming cycles. Nonetheless, such research primarily targets industrial-scale power generation, with relatively limited focus on small-scale, integrated systems that cater to domestic or commercial cooling applications.

In contrast to these large-scale efforts, the present study is dedicated to exploring the application of SOFC-GT technology in conjunction with a small-scale Vapor Absorption Refrigeration System (VARs), specifically designed for residential or office room air conditioning. The objective of this research is to develop and evaluate a compact, efficient, and sustainable energy system capable of delivering both electrical power and cooling using a single integrated configuration. To ensure reliability and continuous operation, a water storage tank is incorporated as a thermal buffer or backup heat source for the VARs, enabling stable cooling output even during fluctuations in SOFC operation or heat generation.

The proposed hybrid system is simulated using the Engineering Equation Solver (EES) software [107], which allows for detailed thermodynamic modelling and performance assessment under various operational conditions. Through this simulation framework, the system's performance is optimized for energy efficiency, exergy utilization, and sustainable operation.

Additionally, the literature survey highlights a notable research gap concerning the use of helium as a working fluid in Brayton cycles, particularly in solar power tower applications. Although helium possesses advantageous thermophysical properties such as low molecular weight, high thermal conductivity, and chemical inertness, its use in solar-driven Brayton cycles has not been extensively explored. To address this gap, the study proposes the integration of a HRSG and a VACS with the base Helium Brayton Cycle (HBC). This trigeneration setup aims to effectively utilize the waste heat produced by the HBC, thereby enhancing overall system performance, reducing energy losses, and contributing to sustainable energy solutions.

3.2 Identified Research Gaps:

1. Limited academic and industrial studies have been reported on helium-based Brayton cycles specifically designed for solar power tower applications. The integration of HRSG and VACS in such configurations remains largely unexplored, particularly for achieving trigeneration (electricity, heating, and cooling) outcomes.
2. Lack of comprehensive 4E (energy, economic, exergy, environmental) analysis of SOFC-based VARS used for small-scale room air conditioning. While several models exist, few attempt a holistic performance evaluation using all four dimensions.
3. Insufficient research on the mechanical design and component-level optimization of small-scale LiBr-H₂O VARS. Most of the available literature focuses on either theoretical modeling or simplified simulations without addressing real-world mechanical feasibility or integration.

4. Scarcity of experimental validations and techno-economic assessments of compact hybrid systems that use low-grade thermal energy (from fuel cells, solar collectors, or waste heat) for space cooling. The practical implementation of such systems in real-world, off-grid, or distributed energy settings remains underreported.

3.3 Objective

1. To develop a novel solar-driven trigeneration system based on VACS.
2. To perform the comprehensive energy and exergy analysis of proposed trigeneration system.
3. To develop VA cooling system based on SOFC-GT configuration.
4. To investigate the exergy, energy, economic, and environmental evaluations of SOFC-GT based vapor absorption (VA) cooling systems.

3.3.1 Research Objective Workflow

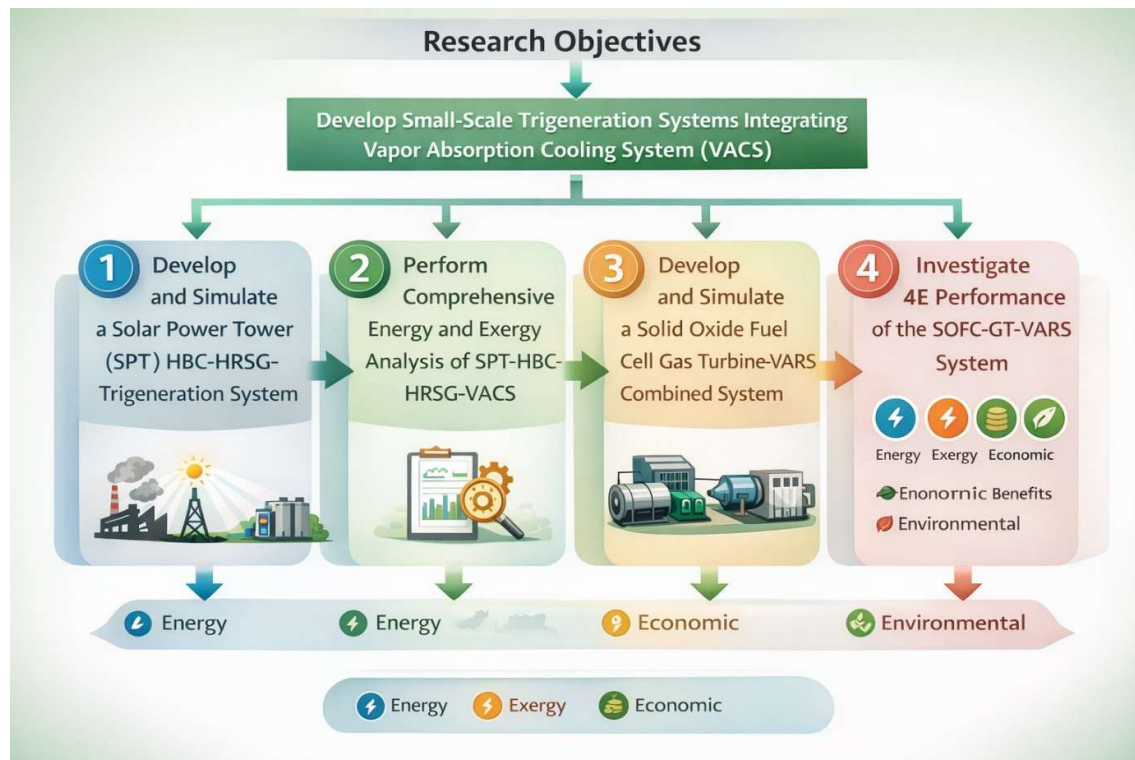


Figure 3.1: Research Objective Workflow

CHAPTER 4 : SYSTEM DESCRIPTION & MODELLING

4.1 SPT-HBC-HRSG-VACS TRIGENERATION SYSTEM

4.1.1 System Description

Figure 1 presents a schematic layout of the proposed solar-operated trigeneration system, which integrates SPT with HBC, HRSG, and VACS. This comprehensive system is designed to simultaneously generate electricity, process heat, and provide cooling for applications such as room air conditioning, thus serving as a sustainable and integrated energy solution. The solar subsystem includes an array of heliostats and a volumetric air receiver, which form the core energy-harvesting units. The input and geometric parameters relevant to the design and simulation of the SPT system are provided in Table 8, including heliostat dimensions, receiver aperture, tower height, and solar flux characteristics.

In this configuration, air is selected as the heat transfer fluid (HTF) due to its widespread availability, cost-effectiveness, and favorable thermophysical properties at high operating temperatures. Compared to conventional molten salt systems, which are constrained by thermal decomposition limits (typically around 565–600°C), air has no specific upper temperature limit for thermal stability. It can reliably operate at elevated temperatures of up to 1000°C, as confirmed in previous studies [70]. This capability makes air a highly advantageous HTF, especially in high-temperature solar thermal power applications where direct heating is involved.

The trigeneration system is specifically modeled under peak load operating conditions, which represents the scenario where energy demand and solar irradiance are at their highest. As a result, a TES unit is intentionally excluded in this study, since the current focus is on evaluating real-time energy conversion performance rather than load-shifting or energy storage dynamics. At the heart of the proposed energy system lies the HBC, acting as the topping power cycle. The cycle begins at state 1, where helium—an inert noble gas known for its excellent thermal conductivity, low density, and high specific heat ratio—is compressed in the HC. This

compression step increases both the pressure and temperature of the working fluid. The high-pressure helium stream is then passed to the recuperator, where it absorbs waste heat from the turbine exhaust, thus increasing its thermal energy content further and reaching state 3.

Following the recuperator, the helium enters an IHE, which is thermally connected to the volumetric receiver of the SPT system. Here, the helium absorbs solar thermal energy from the air-based HTF, reaching a peak temperature of 850°C at state 4, which is the inlet of the HT. The hot helium is isentropically expanded in the turbine, producing mechanical work that is eventually converted into electricity. This expansion reduces the temperature and pressure of the helium, which exits the turbine at state 5.

The helium, now in a partially cooled but still thermally useful state, reenters the recuperator, where it transfers heat to the compressed low-temperature helium from state 2, thereby enhancing cycle efficiency through internal heat recovery. The helium exiting the recuperator at state 6 still retains substantial low-grade thermal energy, which is then used to drive HRSG. The HRSG produces medium-temperature steam at 200°C at high pressure of 2000 kPa, which is suitable for various industrial process heat applications including: space heating in institutional buildings, domestic hot water production, industrial drying processes, preheating in chemical or food processing industries, and absorption chilling (when integrated with VACS).

Even after steam generation, the helium stream exiting the HRSG at state 7 contains residual low-grade heat. To utilize this remaining energy effectively, a VACS is incorporated as a bottoming cooling cycle. This integration enables the system to produce cooling for space conditioning, thus fulfilling the third leg of trigeneration. The helium finally passes through a precooler at state 8, which brings it to the appropriate temperature for recompression. It is then recirculated back to the compressor, closing the Brayton cycle loop.

The VACS subsystem uses a LiBr–H₂O working pair, a commonly adopted non-toxic and environmentally friendly refrigerant–absorbent combination. In this subsystem, water is desorbed from the lithium bromide solution in the generator using the low-grade thermal energy recovered from the HRSG exhaust. The water vapor acts as the refrigerant and flows through an expansion valve (EXV), then enters the condenser, where it releases latent heat, and subsequently the evaporator, where it absorbs heat from the conditioned space to provide cooling.

The vapor exiting the evaporator (state 9) and the LiBr-rich absorbent solution are recombined in the absorber at state 10. The resulting solution is then pumped back to the generator by a solution pump to complete the absorption cycle. To improve thermal efficiency, SHE is employed, where the strong solution exiting the generator (state 13) preheats the weak solution entering the generator (state 11). This internal heat recovery process significantly reduces the thermal load on the generator and enhances the overall COP of the cooling cycle.

4.1.2 Modelling of the Proposed System

To perform a rigorous thermodynamic analysis of the integrated SPT–HBC–HRSG–VACS trigeneration system, a set of engineering assumptions were employed to simplify the model while preserving analytical accuracy. These assumptions are as follows:

1. All components operate under steady-state and steady-flow conditions, thereby excluding transient effects and dynamic behavior from the present model.
2. Pressure drops due to frictional and flow resistance effects within each component (e.g., compressors, heat exchangers, piping) are accounted for as per the values listed in Table 4.1.
3. Changes in kinetic and potential energy of the working fluids are neglected, as their contribution to the total energy balance is minimal compared to thermal and mechanical energies.

4. Isentropic efficiencies of the compressor, turbine, and solution pump are assumed based on established literature values and specified in Table 4.1 for simulation accuracy.

4.1.3 The complete system comprises two major subsystems:

- The solar-driven subsystem, which includes the SPT heliostat field and air receiver delivering heat via HTF.
- The integrated trigeneration system, encompassing the HBC, HRSG, and VACS.

For each component, energy and exergy balance equations are formulated to assess performance, irreversibility, and energy conversion efficiency. The first law of thermodynamics (energy conservation) for a control volume under steady-state flow is expressed as:

$$\dot{Q}_{CV} - \dot{W}_{CV} + \sum(\dot{m}_i h_i) - \sum(\dot{m}_e h_e) = 0 \quad 4.1$$

Where:

$\dot{Q}_{CV}$ is the heat input rate to the control volume,

$\dot{W}_{CV}$ is the work output rate,

$\dot{m}_i h_i$ and $\dot{m}_e h_e$ are the enthalpy flows of inlet and exit streams, respectively.

The second law of thermodynamics, accounting for entropy generation and exergy destruction, is used to develop the exergy balance for each component as:

$$\dot{E}D = \dot{E}X_{in} - \dot{E}X_{out} \quad 4.2$$

Where:

$\dot{E}D$ is the rate of exergy destruction,

$\dot{E}X_{in}$ and $\dot{E}X_{out}$ are the rates of exergy inflow and outflow, respectively.

In this model, physical exergy is the only form of exergy considered, while chemical exergy, potential energy, and kinetic energy contributions are neglected, since the working fluids do not undergo chemical reactions or experience significant elevation or velocity changes.

The dot symbol (·) above a quantity denotes a rate per unit time. For example, $\dot{m}$ represents mass flow rate (kg/s), $\dot{Q}$ represents heat transfer rate (kW), and $\dot{W}$ represents power (kW). Even

though the system operates under steady-state conditions (no accumulation), rates are used to define the magnitude of flow, work, and heat interactions per unit time.

This modelling approach enables the detailed simulation, optimization, and evaluation of the proposed solar trigeneration system using a thermodynamic platform such as Engineering Equation Solver (EES) [107]. The results obtained from this model will aid in understanding the system's performance under varying operational parameters and will help identify opportunities for energy and exergy efficiency improvements.

$$\dot{E}X_j = \dot{m}[(h_j - h_0) - T_0(s_j - s_0)] \quad 4.3$$

The subscript 'j' denotes a specific thermodynamic state, while $\dot{E}X_j$ refers to the physical exergy at that state. In a solar tower system, DNI is concentrated onto the receiver by an array of heliostats. However, the actual solar radiation absorbed by the receiver is governed by the optical and thermal efficiency of the heliostat field. A portion of the incoming solar energy is invariably lost to the surroundings due to reflection, absorption, or transmission losses. Furthermore, the quantity of solar energy available at the receiver is influenced by the site's geographical characteristics, particularly the average annual DNI. Accordingly, the total solar thermal heat effectively absorbed by the receiver can be quantified using the following relation [70].

$$\dot{Q}_{\text{rec,in}} = \eta_{\text{field}} \cdot \dot{Q}_{\text{Sun}} = \eta_{\text{field}} \cdot \text{DNI} \cdot A_{\text{hel}} \cdot N_{\text{hel}} \quad 4.4$$

Where, η_{field} is the efficiency of heliostats and expressed as [70];

$$\eta_{\text{field}} = \eta_{\text{cos}} \cdot \eta_{\text{s\&b}} \cdot \eta_{\text{int}} \cdot \eta_{\text{att}} \cdot \eta_{\text{ref}} \quad 4.5$$

Where, η_{ref} , η_{att} , η_{int} , $\eta_{\text{s\&b}}$ and η_{cos} denotes the efficiencies of heliostats reflectivity, atmospheric attenuation, interception efficiency, shading and blocking and cosine effect respectively. These values has been taken directly from the operational SPT plant [70]. The receiver absorbs a thermal input denoted by $\dot{Q}_{\text{rec,in}}$, while a portion of this energy, represented

by $\dot{Q}_{rec,loss}$, is lost to the surrounding atmosphere through various heat transfer mechanisms such as radiation, convection, and conduction.

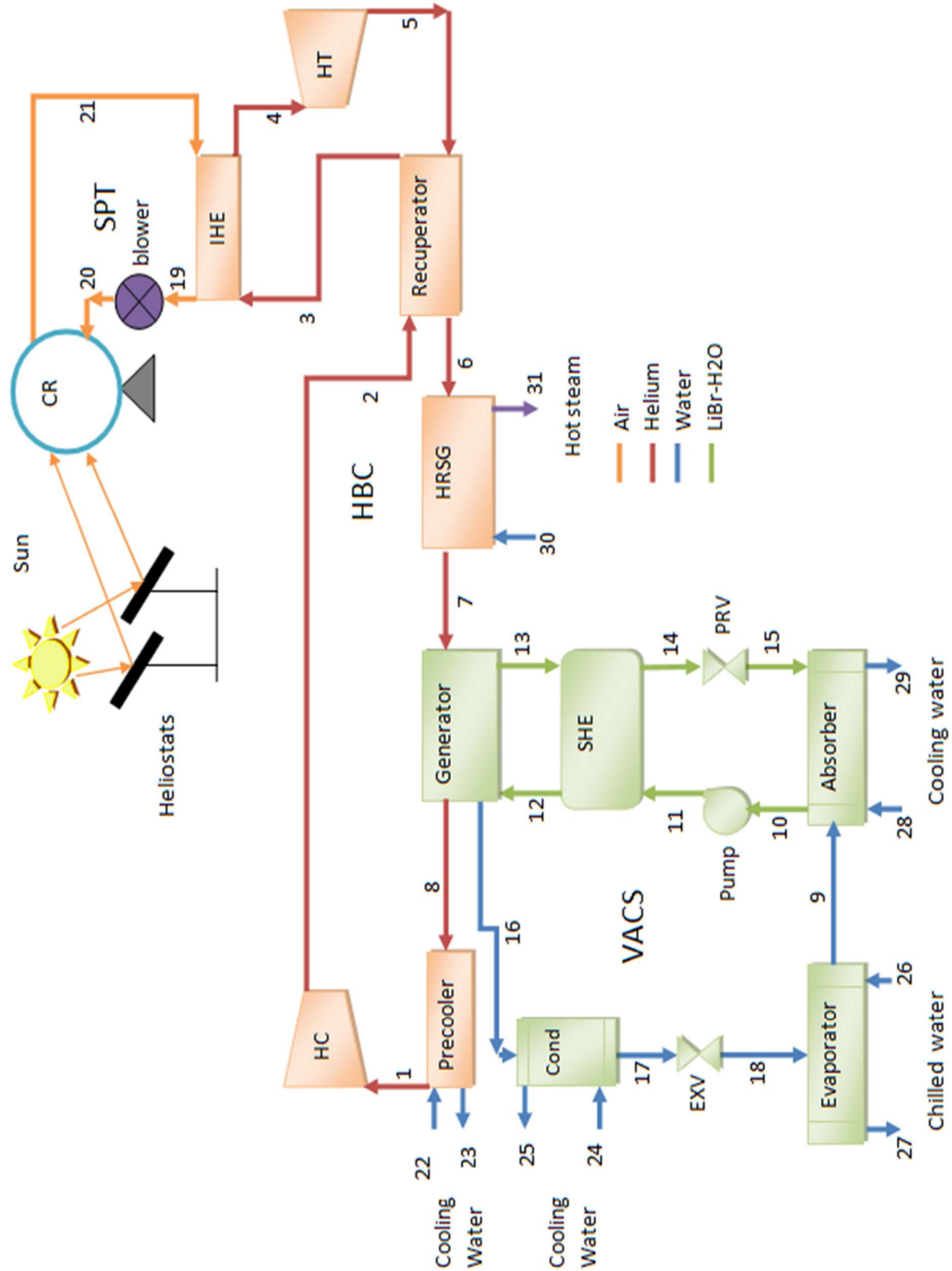


Figure 4.1: Proposed SPT based trigeneration system

The net useful heat transferred to the heat transfer fluid (HTF), which in this case is air, is given by $\dot{Q}_{rec,net}$, after accounting for these losses. Heat transport and receiver efficiency are stated as [16];

$$\eta_{rec} = \frac{\dot{Q}_{rec,net}}{\dot{Q}_{rec,in}} \quad 4.6$$

$$\dot{Q}_{rec,in} = \dot{Q}_{rec,net} + \dot{Q}_{rec,loss} = \dot{m}_{air}(h_{30} - h_{31}) + \dot{Q}_{rec,loss} \quad 4.7$$

When modeling the VACS subsystem, the following assumptions are taken into consideration.

In the present analysis, all system components are evaluated under steady-state operating conditions. Pressure and heat losses within heat exchangers and piping are assumed negligible.

The LiBr-H₂O working pair is considered to be in thermodynamic equilibrium at the specified pressures and temperatures within the generator and absorber. Both the absorber and condenser are cooled using water, with the cooling water's pressure and temperature specified and assumed as per Table 4.1. Saturated conditions are assumed at state points 9 and 17. The governing mathematical formulations used for the exergy and energy analysis of individual components are detailed in Table 9. These equations were implemented in Engineering Equation Solver (EES), utilizing the baseline data provided in Table 4.1. The thermodynamic properties at each state point were computed using EES's built-in property functions. Based on these computed properties, heat and work interactions, along with the exergy at each state, were determined. For exergy analysis of the VACS components (absorber, condenser, evaporator, generator), the temperature of heat transfer is taken as the component outlet temperature of the cooling/heating fluid. Specifically: Absorber temperature (T_a) = outlet temperature of cooling water from the absorber ($T_{29} = 35^\circ\text{C}$ as per Table 4.1); Condenser temperature (T_c) = outlet temperature of cooling water from the condenser ($T_{25} = 35^\circ\text{C}$); Evaporator temperature (T_e) = outlet temperature of chilled water ($T_{27} = 9^\circ\text{C}$, since $T_9 = 6^\circ\text{C} + 3^\circ\text{C}$); Generator temperature (T_g) = inlet temperature of hot water to generator ($T_{13} = 80^\circ\text{C}$). This approach assumes that

the heat transfer occurs at a constant temperature equal to the outlet stream temperature, which is a standard simplification in exergy analysis of absorption systems [75, 113]. The justification is that the dominant temperature for exergy calculation is the sink/source temperature where heat is rejected or absorbed.

4.1.4 System parameters

The system's net power $\dot{W}_{net}$ output excluding the contribution of pump work was evaluated using the derived performance equations:

$$\dot{W}_{net} = \dot{W}_{HT} - \dot{W}_{HC} \quad 4.8$$

After neglecting the pump work it can be expressed as [75];

$$COP_{VACS} = \frac{\dot{Q}_e}{\dot{Q}_g} \quad 4.9$$

The energy proposed systems can be expressed as [75];

$$\eta_{en,HBC} = \frac{\dot{W}_{net}}{\dot{Q}_{Sun}} \quad 4.10$$

$$\eta_{en,co,h} = \frac{\dot{W}_{net} + \dot{Q}_{HRSG}}{\dot{Q}_{Sun}} \quad 4.11$$

$$\eta_{en,co,c} = \frac{\dot{W}_{net} + \dot{Q}_e}{\dot{Q}_{Sun}} \quad 4.12$$

$$\eta_{en,tri} = \frac{\dot{W}_{net} + \dot{Q}_{HRSG} + \dot{Q}_e}{\dot{Q}_{Sun}} \quad 4.13$$

The exergy proposed systems can be expressed as [75];

$$\eta_{ex,HBC} = \frac{\dot{W}_{net}}{\dot{Q}_{Sun} \cdot \left(1 - \frac{T_0}{T_{ref,Sun}}\right)} \quad 4.14$$

$$\eta_{ex,co,h} = \frac{\dot{W}_{net} + \dot{E}X_{HRSG}}{\dot{Q}_{Sun} \cdot \left(1 - \frac{T_0}{T_{ref,Sun}}\right)} \quad 4.15$$

$$\eta_{ex,co,c} = \frac{\dot{W}_{net} + \dot{E}X_e}{\dot{Q}_{Sun} \cdot \left(1 - \frac{T_0}{T_{ref,Sun}}\right)} \quad 4.16$$

$$\eta_{ex,tri} = \frac{\dot{W}_{net} + \dot{E}X_{HRSG} + \dot{E}_e}{\dot{Q}_{Sun} \cdot \left(1 - \frac{T_0}{T_{ref,Sun}}\right)} \quad 4.17$$

Where, T_{Sun} denotes the apparent Sun's temperature and taken as 4500 K [70, 112].

Table 4.1: Given data for base conditions

Symbol of Parameters	Value
SPT system	
η_{field}	0.75 [24]
N_{hel}	500 [25]
DNI	850 W/m ² [16,25]
A_{hel}	9.45×12.84 m ² [25]
A_{rec}	68.1 m ² [25]
η_{rec}	0.9 [24]
HBC	
T_4	850 °C [16,25]
η_{HC}	0.88 [20]
CPR	2.5[26]
η_{HT}	0.9[20]
P_1	2500 kPa [16]
ε	0.9 [16, 20]
HRSG	
Pinch point temperature difference	30°C [21]
P_{31}	2000 kPa [21]
VACS	
T_{13}	80°C[21]
T_{17}	35°C [21]
T_{10}	35°C [21]
T_9	6°C [21]

ε_{SHE}	0.7 [21]
Pump isentropic efficiency (η_{pump})	0.9 [21]
Water temperature at condenser inlet (T_{24})	$T_{17} - 8$ [21]
Water temperature at condenser outlet (T_{25})	$T_{17} - 3$ [21]
Chilled water temperature at evaporator outlet (T_{27})	$T_9 + 3$ [21]
Water temperature at absorber inlet (T_{28})	$T_{10} - 8$ [21]
Chilled water temperature at evaporator inlet (T_{26})	$T_9 + 8$ [21]
Water temperature at absorber outlet (T_{29})	$T_{10} - 3$ [21]
T_0	25 °C [16,26]
P_0	101.3 kPa [16,26]

Table 4.2: Mathematical modelling equations for each component of the proposed solar integrated power and cooling system

Component	Energy balance equation	Exergy balance equation
Heliostat field	$\dot{Q}_{rec,in} = \eta_{field} \cdot DNI \cdot A_{hel} \cdot N_{hel}$	$\dot{Q}_{Sun} \cdot \left(1 - \frac{T_0}{T_{ref,Sun}} \right)$ $= \dot{Q}_{rec,in} \cdot \left(1 - \frac{T_0}{T_{ref,hel}} \right)$ $+ \dot{E}D_{hel}$

Receiver	$\dot{Q}_{\text{rec,in}} = \dot{m}_{\text{air}}(h_{21} - h_{20})$ $+ \dot{Q}_{\text{rec,loss}}$	$\dot{E}X_{20} + \dot{Q}_{\text{rec,in}} \cdot \left(1 - \frac{T_0}{T_{\text{ref,hel}}}\right)$ $= \dot{E}X_{21}$ $+ \dot{Q}_{\text{rec,loss}} \cdot \left(1 - \frac{T_0}{T_{\text{rec}}}\right) + \dot{E}D_{\text{rec}}$
Intermediate heat exchanger	$\dot{Q}_{\text{IHE}} = \dot{m}_{\text{air}}(h_{21} - h_{19})$ $+ \dot{m}_{\text{He}}(h_4 - h_3)$	$\dot{E}X_{21} - \dot{E}X_{19} = \dot{E}X_4 - \dot{E}X_3$ $+ \dot{E}D_{\text{IHE}}$
Helium turbine	$\dot{W}_{\text{HT}} = \eta_{\text{HT}} \cdot \dot{m}_{\text{He}}(h_4 - h_{5s})$	$\dot{E}X_4 = \dot{E}X_5 + \dot{W}_{\text{HT}} + \dot{E}D_{\text{HT}}$
Helium compressor	$\dot{W}_{\text{HC}} = \eta_{\text{HC}} \cdot \dot{m}_{\text{He}}(h_{2s} - h_1)$	$\dot{E}X_1 = \dot{E}X_2 - \dot{W}_{\text{HC}} + \dot{E}D_{\text{HC}}$
HRSG	$\dot{Q}_{\text{HRSG}} = \dot{m}_6(h_6 - h_7)$ $= \dot{m}_{30}(h_{31} - h_{30})$ $\Delta T_{\text{min}} = T_6 - T_{31}$	$\dot{E}X_6 - \dot{E}X_7 = \dot{E}X_{31} - \dot{E}X_{30}$ $+ \dot{E}D_{\text{HRSG}}$
Precooler	$\dot{m}_{\text{He}}(h_8 - h_1)$ $= \dot{m}_{\text{water}}(h_{23} - h_{22})$	$\dot{E}X_8 - \dot{E}X_1 = \dot{E}X_{22} - \dot{E}X_{23}$ $+ \dot{E}D_{\text{Precooler}}$
Recuperator	$(h_3 - h_2) = (h_5 - h_6)$ $\varepsilon_{\text{Recuperator}} = \frac{(T_3 - T_2)}{(T_5 - T_2)}$	$\dot{E}X_5 - \dot{E}X_6 = \dot{E}X_3 - \dot{E}X_2$ $+ \dot{E}D_{\text{Recuperator}}$
Generator	$\dot{Q}_g = \dot{m}_{16} \cdot h_{16} + \dot{m}_{13} \cdot h_{13}$ $- \dot{m}_{12} \cdot h_{12}$ $\dot{m}_{12} = \dot{m}_{13} + \dot{m}_{16}$	$\dot{E}X_{13} + \dot{Q}_g \cdot \left(1 - \frac{T_0}{T_g}\right)$ $= \dot{E}X_{16} + \dot{E}X_{13}$ $+ \dot{E}D_g$

Condenser	$\dot{Q}_c = \dot{m}_{16}(h_{16} - h_{17}) = \dot{m}_{\text{water}}(h_{25} - h_{24})$	$\dot{E}X_{16} - \dot{E}X_{17} = \dot{Q}_c \left(1 - \frac{T_0}{T_c}\right) + \dot{E}D_c$
Expansion valve	$h_{17} = h_{18}$	$\dot{E}X_{17} - \dot{E}X_{18} = \dot{E}D_{\text{EXV}}$
Absorber	$\dot{Q}_a = \dot{m}_9 \cdot h_9 + \dot{m}_{15} \cdot h_{15} - \dot{m}_{10} \cdot h_{10}$ $\dot{m}_{10} = \dot{m}_9 + \dot{m}_{15}$ $x_{13} \cdot \dot{m}_{15} = x_{10} \cdot \dot{m}_{10}$	$\dot{E}X_{10} + \dot{Q}_a \left(1 - \frac{T_0}{T_a}\right) + \dot{E}D_g$ $= \dot{E}X_9 + \dot{E}X_{15}$
Pump	$\dot{W}_p = \dot{m}_{10}(h_{11} - h_{10})$ $\eta_p = \frac{(h_{11s} - h_{10})}{(h_{11} - h_{10})}$	$\dot{E}X_{10} = \dot{E}X_{11} - \dot{W}_{\text{pump}} + \dot{E}D_{\text{pump}}$
Solution heat exchanger	$\dot{m}_{13} \cdot h_{13} - \dot{m}_{14} \cdot h_{14} = \dot{m}_{12} \cdot h_{12} - \dot{m}_{11} \cdot h_{11}$ $\varepsilon_{\text{SHE}} = \frac{(T_{13} - T_{14})}{(T_{13} - T_{11})}$	$\dot{E}X_{13} + \dot{E}X_{11} = \dot{E}X_{12} + \dot{E}X_{14} + \dot{E}D_{\text{SHE}}$
Pressure reducing valve	$h_{14} = h_{15}$	$\dot{E}X_{14} - \dot{E}X_{15} = \dot{E}D_{\text{PRV}}$
Evaporator	$\dot{Q}_e = \dot{m}_9(h_9 - h_{18}) = \dot{m}_{\text{chilled water}}(h_{26} - h_{27})$	$\dot{E}X_9 - \dot{E}X_{18} = \dot{Q}_e \left(1 - \frac{T_0}{T_e}\right) + \dot{E}D_e$

4.2 SOFC -GT-VACS

SOFC-GT hybrid systems can be configured in three primary arrangements: Pressurized SOFC-GT where the SOFC stack operates at elevated pressure (3-10 bar) and the cathode

exhaust directly feeds the gas turbine, offering higher efficiency (65-70%) but requiring pressure vessels and complex sealing; Atmospheric SOFC with GT bottoming cycle where the SOFC operates at atmospheric pressure and the afterburner exhaust drives the GT, which is simpler but has lower efficiency (55-60%); and Recuperated SOFC-GT which includes a recuperator to preheat air entering the SOFC using GT exhaust, improving thermal integration. The present study adopts an atmospheric SOFC with GT bottoming cycle arrangement. The justification for this selection is: (i) lower mechanical complexity and cost compared to pressurized systems; (ii) easier maintenance and sealing; (iii) sufficient for small-scale (1.5 TR cooling) application where ultra-high efficiency is not the primary driver; (iv) well-validated in literature [106,108] for distributed generation.

4.2.1 Cogeneration system

The schematic layout of the proposed SOFC-based hybrid system is illustrated in **Figure 4.2**. The configuration integrates an SOFC unit with an AB, GT, dual compressors, multiple HXs, a WP, and essential components of a compact VARS for room air conditioning. Two compressors—one for CH₄ and another for air—are employed to achieve the required pressure levels for the SOFC stack operation. The hot exhaust gases from the AB are utilized to preheat the compressed CH₄ and air via the FHE and AHE, respectively, enhancing the thermal efficiency before the streams enter the SOFC. Simultaneously, the AB provides thermal energy to the WHE, which converts pressurized water into steam for use in the SMR process. The produced steam is combined with CH₄ in a mixer and supplied to the SOFC for reforming and electrochemical reaction. Inside the SOFC, exothermic electrochemical reactions at the anode generate H₂-rich gases, while heated air exits through the cathode. The heat released supports the endothermic SMR process. Electrical energy generated in DC form is converted to AC using an inverter for end-use. The AB combusts any unreacted fuel and excess air from the SOFC, generating high-temperature exhaust gases. These gases are used to preheat the working

fluids (air, CH₄, water), followed by expansion in the GT to produce additional power. The exhaust gases exiting the GT are routed through a water storage tank, which acts as a TES system by heating water up to 100 °C. This stored heat ensures continuous VARS operation during low solar or standby periods. System is designed to deliver a cooling output of 5.275 kW (1.5 TR) with chilled water produced at T_{EVA} of 5 °C.

4.2.2 Mathematical Modelling

To enable efficient thermodynamic analysis, several baseline assumptions are adopted prior to model development. These ensure simplification while maintaining model accuracy and represent system behavior under steady-state conditions.

The following are the general assumptions [106, 108, 118, 119]:

1. It is assumed that all components of the system operate under steady-state conditions, ensuring thermodynamic equilibrium throughout the system.
2. The thermodynamic processes occurring within the pumps, compressors, and GT are considered to be adiabatic, implying no heat transfer with the surroundings during compression or expansion.
3. The inlet temperature to the SOFC is assumed to remain constant throughout the system's operation.
4. A fixed temperature difference is maintained between the inlet and outlet streams of the SOFC to ensure thermal stability and consistent performance.
5. All gaseous working fluids within the system are modelled as ideal gases to simplify the thermodynamic analysis.
6. Methane (CH₄) is used as the sole fuel source, while air is assumed to consist of 79% nitrogen (N₂) and 21% oxygen (O₂) by volume.
7. Variations in kinetic and potential energy are considered negligible and are excluded from the energy balance calculations.

8. The system is assumed to be initially charged with air, methane, and water at standard ambient conditions of 25 °C and 101.3 kPa.
9. Heat transfer between the SOFC stack and the surrounding environment is considered negligible; the stack is treated as perfectly insulated.
10. Water exiting the condenser is considered a saturated liquid, simplifying phase transition calculations.
11. Water leaving the evaporator is taken as saturated vapour, representing ideal absorption cooling cycle conditions.
12. Pressure drops across all pipelines and heat exchangers specifically the AHE and the FHE are assumed to be zero.
13. The evaporator is modelled to deliver a constant cooling capacity of 5.275 kW (equivalent to 1.5 TR).
14. Heat transfer from the system to the environment is only considered in the absorber, condenser, and evaporator; all other components are assumed to be perfectly insulated.

In this work, component-wise modelling and system simulation were carried out using the Engineering Equation Solver (EES) software. The EES program was employed to model individual components and solve the resulting thermodynamic equations using the input data provided in Table 4.3.

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Table 4.3: Given data for proposed cogeneration system

Parameters	Values
SOFC group [106, 108]	
Base inlet temperature to SOFC	726.85 °C
Steam to carbon ratio	2.5
DC/AC inverter efficiency	0.97
Effective gaseous diffusivity through cathode	0.000005 m ² /s
Thickness of electrolyte	0.00001 m
Thickness of anode	0.0005 m
Thickness of interconnect	0.003 m
Thickness of cathode	0.00005 m
Baseline current density	8000 A/m ²
Exchange current density of anode	6500 A/m ²
Effective gaseous diffusivity through anode	0.00002 m ² /s
Exchange current density of cathode	2500 A/m ²
Fuel utilization factor	0.85
Active surface area	0.01 m ²
SOFC inlet and outlet temperature difference	100 °C
Number of cells	11,000
AB combustion efficiency	0.99
Pressure loss in afterburner	3%
Pressure loss in SOFC stack	2%
Pressure loss in heat exchanger	2%
Isentropic water pump efficiency	0.85
Isentropic compressor efficiency	0.85

Pressure ratio in water pump, fuel and air compressor	8
Gas turbine [108]	
Expansion ratio in GT	7
GT isentropic efficiency	0.85
VARs [117]	
Generator temperature (T_{GEN})	80 °C
Water temperature of storage tank at GEN inlet (T_{29})	98 °C
Water temperature of storage tank at GEN outlet (T_{30})	88 °C
Water temperature of storage tank (T_{WST})	100 °C
Effectiveness of SHE (ϵ_{SHE})	0.70
Evaporator temperature (T_{EVAP})	5 °C
Chilled water outlet temperature (T_{33})	7 °C
Cooling water outlet temperature (T_{35} and T_{37})	35 °C
Cooling water inlet temperature (T_{34} and T_{36})	25 °C
Condenser temperature (T_{COND})	35 °C
Absorber temperature (T_{ABS})	35 °C
Cooling load/refrigeration effect ($\dot{Q}_{\text{EVAP}}$)	5.275 kW or 1.5 TR
Isentropic efficiency of pumps (η_{pump})	0.85
Ambient temperature	25 °C
Faraday constant	96,485 C/g mol
Universal gas constant	8.314 J/mol K

Ambient pressure	101.30 kPa
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4.2.3 Energy and exergy analysis

Each component in the proposed trigeneration plant is treated as a control volume, allowing for the application of mass, energy, and exergy balance equations across system boundaries to facilitate detailed thermodynamic analysis. The molar and energy conservation equations are follows [108]:

$$\sum(\dot{n}_i) - \sum(\dot{n}_e) = 0 \quad 4.18$$

$$\dot{Q}_{CV} - \dot{W}_{CV} + \sum(\dot{n}_i \bar{h}_i) - \sum(\dot{n}_e \bar{h}_e) = 0 \quad 4.19$$

Second-law performance is assessed using the concept of exergy. The four components of total exergy for a fluid stream are as follows [108, 115]:

$$\dot{X}_j = \dot{X}_{ph} + \dot{X}_{ch} + \dot{X}_{ke} + \dot{X}_{pe} \quad 4.20$$

The physical exergy for can be calculated as [108, 114]:

$$\dot{X}_{ph} = \dot{m} \cdot [(h_j - h_0) - T_0 \cdot (s_j - s_0)] \quad 4.21$$

However, the chemical can be calculated as [108];

$$\dot{X}_{ch} = \dot{n}_t [\sum_i y_i e_{ch,i}^0 + \bar{R} T_0 \sum_i y_i \ln(y_i)] \quad 4.22$$

Where, $e_{ch,i}^0$ is the standard molar chemical exergy of the i^{th} compound and $\dot{n}_t$ is denotes total molar flow rate and y_i is the molar fraction of the individual gas in the flue gases.

After neglecting the exergy due to position and kinetic energy total exergy at the any state can be expressed as;

$$\dot{X}_j = \dot{X}_{ph} + \dot{X}_{ch} \quad 4.23$$

The k^{th} component's exergy destruction rate can be computed as a measure of element level second-law performance using the equation[108]:

However, total exergy input to the plant can be defined as [108,114];

$$\dot{X}_{input} = \dot{n}_{CH,11} \cdot \bar{e}_{CH}^{ch,0}$$

4.24

Detailed modelling equations for exergetic and energetic evaluation for each component of the proposed hybrid trigeneration system are listed in Table 4.4. These equations were solved using the EES code to calculate the heat or work interaction and exergy destruction within the component using the given data in Table 4.4.

Table 4.4. Equations for each component of cogeneration system

Component	Equations for energy balance	Equations for exergy balance
Fuel cell group		
SOFC stack	$\dot{n}_{11} \cdot \bar{h}_{11} + \dot{n}_3 \cdot \bar{h}_3 = \dot{n}_{12} \cdot \bar{h}_{12} + \dot{n}_4 \cdot \bar{h}_4$	$\dot{X}_3 + \dot{X}_{11} = \dot{X}_4 + \dot{X}_3 + \dot{E}D_{SOFC}$
After burner	$\dot{Q}_{loss,AB} = \dot{n}_{13} \cdot \bar{h}_{13} - (\dot{n}_{12} \cdot \bar{h}_{12} + \dot{n}_4 \cdot \bar{h}_4)$ $\eta_{AB} = \frac{\dot{n}_{12} \cdot \bar{h}_{12} + \dot{n}_4 \cdot \bar{h}_4}{\dot{n}_{13} \cdot \bar{h}_{13}}$	$\dot{X}_{12} + \dot{X}_4 = \dot{X}_{13} + \dot{E}D_{AB}$
Air compressor	$\dot{W}_{ac} = \dot{n}_1 \cdot (\bar{h}_2 - \bar{h}_1)$ $\eta_{ac} = \frac{(\bar{h}_{2s} - \bar{h}_1)}{(\bar{h}_2 - \bar{h}_1)}$	$\dot{X}_1 = \dot{X}_2 - \dot{W}_{ac} + \dot{E}D_{ac}$
Fuel compressor	$\dot{W}_{fc} = \dot{n}_5 \cdot (\bar{h}_6 - \bar{h}_5)$ $\eta_{ac} = \frac{(\bar{h}_{6s} - \bar{h}_5)}{(\bar{h}_6 - \bar{h}_5)}$	$\dot{X}_5 = \dot{X}_6 - \dot{W}_{fc} + \dot{E}D_{fc}$
Water pump	$\dot{W}_{wp} = \dot{n}_8 \cdot (\bar{h}_9 - \bar{h}_8)$ $\eta_{wp} = \frac{(\bar{h}_{9s} - \bar{h}_8)}{(\bar{h}_9 - \bar{h}_8)}$	$\dot{X}_8 = \dot{X}_9 - \dot{W}_{wp} + \dot{E}D_{wp}$
Water heat exchanger	$\dot{n}_{13} \cdot (\bar{h}_{13} - \bar{h}_{14}) = \dot{n}_9 \cdot (\bar{h}_{10} - \bar{h}_9)$ $\varepsilon_{WHE} = \frac{(T_{10} - T_9)}{(T_{13} - T_9)}$	$\dot{X}_{13} + \dot{X}_9 = \dot{X}_{14} + \dot{X}_{10} + \dot{E}D_{WHE}$
Fuel heat exchanger	$\dot{n}_{13} \cdot (\bar{h}_{14} - \bar{h}_{15}) = \dot{n}_6 \cdot (\bar{h}_7 - \bar{h}_6)$ $\varepsilon_{FHE} = \frac{(T_7 - T_6)}{(T_{14} - T_6)}$	$\dot{X}_{14} + \dot{X}_6 = \dot{X}_{15} + \dot{X}_7 + \dot{E}D_{FHE}$
Air heat exchanger	$\dot{n}_{13} \cdot (\bar{h}_{15} - \bar{h}_{16}) = \dot{n}_2 \cdot (\bar{h}_3 - \bar{h}_2)$ $\varepsilon_{FHE} = \frac{(T_3 - T_2)}{(T_{15} - T_2)}$	$\dot{X}_{15} + \dot{X}_2 = \dot{X}_{16} + \dot{X}_3 + \dot{E}D_{AHE}$
Mixer	$\dot{Q}_{loss,} = \dot{n}_{11} \cdot \bar{h}_{11} - (\dot{n}_{10} \cdot \bar{h}_{10} + \dot{n}_7 \cdot \bar{h}_7)$	$\dot{X}_7 + \dot{X}_{10} = \dot{X}_{11} + \dot{E}D_{mixer}$
Gas turbine	$\dot{W}_{GT} = \dot{n}_{16} \cdot (\bar{h}_{16} - \bar{h}_{17})$	$\dot{X}_{16} = \dot{X}_{17} + \dot{W}_{GT} + \dot{E}D_{GT}$

	$\eta_{GT} = \frac{(\bar{h}_{16} - \bar{h}_{17})}{(\bar{h}_{16} - \bar{h}_{17s})}$	
Water storage tank	$\dot{n}_{17} \cdot (\bar{h}_{17} - \bar{h}_{18}) = \dot{n}_{29} \cdot (\bar{h}_{29} - \bar{h}_{31})$	$\dot{X}_{17} - \dot{X}_{18} = \dot{X}_{29} - \dot{X}_{31} + \dot{E}D_{WST}$
Pump-1	$\dot{W}_{pump1} = \dot{n}_{29} \cdot (\bar{h}_{31} - \bar{h}_{30})$ $\eta_{pump1} = \frac{(\bar{h}_{31s} - \bar{h}_{30})}{(\bar{h}_{31} - \bar{h}_{30})}$	$\dot{X}_{30} = \dot{X}_{31} - \dot{W}_{pump1} + \dot{E}D_{pump1}$
VARs		
Generator	$\dot{Q}_{GEN} = \dot{n}_{25} \cdot \bar{h}_{25} + \dot{n}_{22} \cdot \bar{h}_{22} - \dot{n}_{21} \cdot \bar{h}_{21}$ $\dot{n}_{21} = \dot{n}_{22} + \dot{n}_{25}$	$\dot{X}_{21} + \dot{Q}_{GEN} \cdot \left(1 - \frac{T_0}{T_{GEN}}\right) = \dot{X}_{25} + \dot{X}_{22} + \dot{E}D_{GEN}$
Condenser	$\dot{Q}_{COND} = \dot{n}_{25} \cdot (\bar{h}_{25} - \bar{h}_{26}) = \dot{n}_{36} \cdot (\bar{h}_{37} - \bar{h}_{36})$	$\dot{X}_{25} - \dot{X}_{26} = \dot{Q}_{COND} \cdot \left(1 - \frac{T_0}{T_{COND}}\right) + \dot{E}D_{COND}$
Expansion valve	$\bar{h}_{26} = \bar{h}_{27}$	$\dot{X}_{26} - \dot{X}_{27} = \dot{E}D_{EV}$
Absorber	$\dot{Q}_{ABS} = \dot{n}_{19} \cdot \bar{h}_{19} - \dot{n}_{24} \cdot \bar{h}_{24} - \dot{n}_{28} \cdot \bar{h}_{28}$ $\dot{n}_{19} = \dot{n}_{28} + \dot{n}_{24}$	$\dot{X}_{19} + \dot{Q}_{ABS} \cdot \left(1 - \frac{T_0}{T_{ABS}}\right) + \dot{E}D_{ABS} = \dot{X}_{24} + \dot{X}_{28}$
Pump-2	$\dot{W}_{pump2} = \dot{n}_{20} \cdot (\bar{h}_{20} - \bar{h}_{19})$ $\eta_{pump2} = \frac{(\bar{h}_{20s} - \bar{h}_{19})}{(\bar{h}_{20} - \bar{h}_{19})}$	$\dot{X}_{19} = \dot{X}_{20} - \dot{W}_{pump2} + \dot{E}D_{pump2}$
Solution heat exchanger	$\dot{n}_{22} \cdot \bar{h}_{22} - \dot{n}_{23} \cdot \bar{h}_{23} = \dot{n}_{21} \cdot \bar{h}_{21} - \dot{n}_{20} \cdot \bar{h}_{20}$ $\varepsilon_{SHE} = \frac{(T_{21} - T_{20})}{(T_{22} - T_{20})}$	$\dot{X}_{22} + \dot{X}_{20} = \dot{X}_{21} + \dot{X}_{23} + \dot{E}D_{SHE}$
Pressure reducing valve	$\bar{h}_{23} = \bar{h}_{24}$	$\dot{X}_{24} - \dot{X}_{23} = \dot{E}D_{PRV}$
Evaporator	$\dot{Q}_{EVAP} = \dot{n}_{28} \cdot (\bar{h}_{28} - \bar{h}_{27}) = \dot{n}_{32} \cdot (\bar{h}_{32} - \bar{h}_{33})$	$\dot{X}_{33} - \dot{X}_{32} = \dot{Q}_{EVAP} \cdot \left(1 - \frac{T_0}{T_{EVAP}}\right) + \dot{E}D_{EVAP}$

4.2.4 Electrochemistry of SOFC

The Cogeneration cycle is powered by methane as fuel, obtained through the process of internal reforming. SOFC's anode, as well as cathode, undergoes the following electrochemical processes [106, 108]:



Overall electrochemical reaction of fuel cell can be written as;



Equation (4.25) represents the steam reforming of methane to produce hydrogen, while Equation (4.26) illustrates the water–gas shift reaction. The hydrogen generated from these two processes is subsequently utilized in the electrochemical reaction occurring within the SOFC, as shown in Equation (4.27).

The shifting reaction equilibrium constant is computed as follows [106, 108]:

$$\ln K_s = \frac{\Delta \bar{g}_s^0}{RT_{FC,e}} = \ln \left[\frac{y_r \cdot (3x_r + y_r - z_r)}{(x_r - y_r) \cdot (1.5x_r - y_r + z_r)} \right] \quad 4.28$$

K_s is the shifting reaction's steady balance. The detailed approach as given in Appendix A2, was used to calculate the percentage of each component's output using the results obtained by solving the three equations [108].

The equations for determining the air utilization factor (U_{air}) and the fuel utilization factor (U_f)

are as follows [106,108];

$$U_f = \frac{\text{H}_2 \text{ consumed}}{\text{H}_2 \text{ supplied}} = \frac{z_r}{3x_r + y_r} \quad 4.29$$

$$U_{air} = \frac{\text{O}_2 \text{ consumed}}{\text{O}_2 \text{ supplied}} = \frac{\frac{z_r}{2}}{\dot{n}_{\text{O}_2, \text{inlet}}} \quad 4.30$$

Gibbs free energy is calculated using the formula [106, 108]:

$$\Delta \bar{g}_s^0 = \bar{g}_{s, \text{CO}_2}^0 + \bar{g}_{s, \text{H}_2}^0 - \bar{g}_{s, \text{H}_2\text{O}}^0 - \bar{g}_{s, \text{CO}}^0 \quad 4.31$$

Where,

$$\bar{g}_s^0 = \bar{h} - T_{FC,e} \cdot \bar{s}^0 \quad 4.32$$

where $\bar{h}$ and $\bar{s}^0$ are enthalpy and standard entropy per molar rate, respectively.

The current density (j) and current (I) were calculated as;

$$I = j \cdot A_a \quad 4.33$$

$$j = \frac{2 \cdot F \cdot z_r}{N_{FC} \cdot A_a} \quad 4.34$$

Where, N_{FC} is the number of fuel cells, F is the Faraday constant and A_a is the active surface area.

- **Nernst Voltage:** The theoretical maximum voltage of a fuel cell under open-circuit conditions, calculated from the Gibbs free energy change of the electrochemical reaction. Represents reversible cell voltage with no irreversibilities.
- **Ohmic Loss:** Voltage drop due to resistance to ion flow through the electrolyte and resistance to electron flow through the electrodes and interconnects. Follows Ohm's Law ($V = I \times R$) and increases linearly with current density.
- **Activation Polarization:** Voltage loss required to overcome the activation energy barrier for the electrochemical reactions at the anode and cathode surfaces. Dominant at low current densities.
- **Concentration Loss:** Voltage drop resulting from mass transport limitations where reactants cannot reach reaction sites or products cannot diffuse away. Dominant at high current densities.

The total electric power obtained from the SOFC stack can be calculated using the following equation.

$$\dot{W}_{SOFC} = N_{FC} \cdot V_{FC} \cdot I \quad 4.35$$

Where, V_{FC} is cell voltage which can be defined as [106];

$$V_{FC} = V_N - V_{loss} \quad 4.36$$

Where, V_N refers to Nernst voltage and V_{loss} refers to the voltage loss within the SOFC stack it includes losses due to the Ohmic loss, activation polarization and concentration loss. These all are can be calculated using the equations given in Table 4.5.

Table 4.5:. Nernst voltage and voltage losses like Ohmic, concentration, and activation [106, 108]

<i>Voltages</i>	<i>Corresponding equations</i>
<i>Nernst voltage</i>	$V_N = -\frac{\Delta \bar{g}^0}{nF} + \frac{\bar{R}T_{FC,12}}{nF} \cdot \ln \left(\frac{a_{H2,12} \sqrt{a_{O2,4}}}{a_{H2O,12}} \right); \quad a_{H2O,12} = \frac{P_{H2O,12}}{P_{ref}} =$ $\frac{y_{H2O,12} \cdot P_{12}}{P_{ref}}; \quad a_{H2,12} = \frac{P_{H2,12}}{P_{ref}} = \frac{y_{H2O,12} \cdot P_{12}}{P_{ref}}; \quad a_{O2,4} = \frac{P_{O2,4}}{P_{ref}} = \frac{y_{O2,4} \cdot P_4}{P_{ref}};$ $\Delta \bar{g}^0 = \bar{g}^0_{H2O} - \bar{g}^0_{H2} - \frac{1}{2} \bar{g}^0_{O2};$ $\bar{g}^0 = \bar{h} - T_{FC,12} \cdot \bar{s}^0$
<i>Ohmic loss</i>	$V_{Ohm} = [R_c + \sum_i (\rho_i \cdot L_i)] \cdot j;$ $\rho_e = \left[3.34 \times 10^4 \cdot \exp \left(-\frac{10300}{T_{FC,e}} \right) \right]^{-1};$ $\rho_a = \left[\frac{95 \times 10^6}{T_{FC,e}} \cdot \exp \left(-\frac{1150}{T_{FC,e}} \right) \right]^{-1};$ $\rho_c = \left[\frac{42 \times 10^6}{T_{FC,e}} \cdot \exp \left(-\frac{1200}{T_{FC,e}} \right) \right]^{-1};$ $\rho_{int} = \left[\frac{9.3 \times 10^6}{T_{FC,e}} \cdot \exp \left(-\frac{1100}{T_{FC,e}} \right) \right]^{-1};$
<i>Activation polarization</i>	$V_{act} = \frac{\bar{R}T_{FC}}{F} \cdot \sinh^{-1} \left(\frac{j}{2j_{oa}} \right) + \frac{\bar{R}T_{FC}}{F} \cdot \sinh^{-1} \left(\frac{j}{2j_{oc}} \right)$
<i>Concentration loss</i>	$V_{conc} = \frac{\bar{R}T_{FC}}{2F} \cdot \left[\ln \left(1 + \frac{P_{H2} \cdot j}{P_{H2O} \cdot j_{as}} \right) - \ln \left(1 - \frac{j}{j_{as}} \right) \right]$ $- \frac{\bar{R}T_{FC}}{4F} \ln \left(\frac{j}{j_{cs}} \right)$

4.2.5 Performance parameters

The power output of proposed plant (SOFC-GT-VARS) are calculated as [108];

$$\dot{W}_{net,overall} = \dot{W}_{net,SOFC-GT} - \dot{W}_{pump1} - \dot{W}_{pum} \quad 4.37$$

Where,

$$\dot{W}_{net,SOFC-GT} = \dot{W}_{SOFC} + \dot{W}_{GT} - \dot{W}_{ac} - \dot{W}_{fc} - \dot{W}_{wp} \quad 4.38$$

The energy efficiency and exergy efficiency of the conventional hybrid SOFC-GT plant were calculated as [108];

$$\eta_{en,SOFC-GT} = \frac{\dot{W}_{net,SOFC-GT}}{\dot{m}_{f,11} \cdot (LHV_{CH})} \quad 4.39$$

$$\eta_{ex,SOFC-GT} = \frac{\dot{W}_{net,SOFC-GT}}{\dot{X}_{inlet}} \quad 4.40$$

Where, LHV_{CH4} refers methane fuel lower heating value which is considered as 50 MJ/kg [106] in this study and $\dot{m}_{f,11}$ is its mass flow rate at state 11 entering to the SOFC stack.

However, the energy efficiency and exergy efficiency of the proposed plant (SOFC-GT-VARS) were determined as [114];

$$\eta_{en,overall} = \frac{\dot{W}_{net,Plant} + \dot{Q}_{EVAP}}{\dot{m}_{f,11} (LHV_{CH})} \quad 4.41$$

$$\eta_{ex,overall} = \frac{\dot{W}_{net,Plant} + \dot{Q}_{EVAP} \cdot \left(1 - \frac{T_0}{T_{EVAP}}\right)}{\dot{X}_{inlet}} \quad 4.42$$

Proposed system serves as cooling purpose. Therefore, COP for cooling is defined as [32];

$$COP = \frac{\dot{Q}_{EVAP}}{(\dot{Q}_{GEN} + \dot{W}_{pump2})} \quad 4.43$$

4.2.6 Environment analysis

Environmental scanning of plants focuses on CO₂ emissions because NO_x, SO_x, and CO content in the SOFC exhaust is minimal [106, 108]. For such an assessment, a penalty cost for CO₂ emissions is included, which can be stated as follows [106, 108]:

$$\dot{C}_{env} = 3600 \cdot \dot{m}_{CO} \cdot C_{pen} \quad 4.44$$

where $\dot{m}_{CO_2}$ is the rate of mass flow of CO₂ as an output gas released by SOFC, and C_{pen} denotes the penalty cost per kilogram of CO₂ emissions, which can be estimated as 0.024 \$/kg [106, 108].

Emissions of CO₂ gas (EMI) (kg/kWh) are stated as follows [106, 108]:

$$EMI_{SOFC-GT} = 3600 \cdot \frac{\dot{m}_{CO_2}}{(\dot{W}_{net,SOFC-GT})} \quad 4.46$$

However, CO₂ emission (EMI) per unit energy output for the overall proposed hybrid CCP plant (SOFC-GT-VARS) can be evaluated as [105, 106, 108];

$$EMI_{Plant} = 3600 \cdot \frac{\dot{m}_{CO_2}}{(\dot{W}_{net,Plant} + \dot{Q}_{EVAP})} \quad 4.47$$

4.2.7 Economic assessment for hybrid systems

The methane cost, CO₂ emission penalty, and component cost rate contribute to the system's total cost [115]:

$$\dot{C}_{total} = \dot{C}_f + \dot{C}_{env} + \dot{Z}_{total} \quad 4.48$$

Fuel cost or the operating cost, can be expressed as follows [105, 106]:

$$\dot{C}_f = (C_f \cdot \dot{m}_f \cdot LHV) \cdot 3600 \quad 4.49$$

Where, C_f is set at \$0.003 per MJ as the fuel cost [115]. $\dot{Z}_{total}$ can be expressed as follows [115]:

$$\dot{Z}_{total} = \sum_j \frac{Z_j \cdot CRF \cdot \varphi}{3600 \cdot N} \quad 4.50$$

Where, Z_j denotes capital cost of a component, and N is annually operating hours, which are 7200 hours in this study [108], CRF refers the capital recovery factor, φ refers the maintenance factor which is assumed as 1.06 in this study [115].

The formula for calculating the CRF is [115];

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \quad 4.51$$

Where, ‘ i ’ refers interest rate which is 12% [108, 115], and ‘ n ’ refers life of the plant, which is assumed as 20 years in this work [108, 115]. able 4.6 provides the detailed formulas used to estimate the capital cost of each system component. Specifically, the capital cost of heat exchangers primarily depends on their surface area (A) and overall heat transfer capacity (UA). To calculate the heat capacity of the heat exchanger following formula was used [108]:

$$UA = \frac{\dot{Q}}{LMTD} \quad 4.52$$

Where, LMTD refers to logarithmic mean temperature difference.

Table 4.6. Cost function for various components [105, 106, 108]

Component	Cost function
Fuel stack	$Z_{SOFC} = 1.1 \cdot N_{FC} \cdot (2.96T_e - 1907)$
AC	$Z_{ac} = 91562 \cdot \left(\frac{\dot{W}_{ac}}{455}\right)^{0.67}$
FC	$Z_{ac} = 91562 \cdot \left(\frac{\dot{W}_{fc}}{455}\right)^{0.67}$
WP	$Z_{wp} = 3450 \cdot (\dot{W}_{fc})^{0.71}$
AB	$Z_{AB} = \frac{46.08 \cdot \dot{m}_4}{0.995 - \frac{P_{13}}{P_4}} \cdot (1 + \exp(0.018 \cdot T_{13} - 26.4))$
WHE	$Z_{WHE} = 2500 \cdot UA$
FHE	$Z_{FHE} = 2500 \cdot UA$
AHE	$Z_{AHE} = 2500 \cdot UA$
Mixer	-
Inverter	$Z_{inverter} = 100000 \cdot \left(\frac{\dot{W}_{SOFC}}{500}\right)^{0.7}$
Gas turbine	$Z_{GT} = \dot{W}_{GT} \cdot [1318.5 - 98.32 \cdot \ln(\dot{W}_{GT})]$
Water storage tank	$Z_{HRVG} = 5000 \cdot UA$

Pump-1	$Z_{pump1} = 3450 \cdot (\dot{W}_{pump1})^{0.71}$
Condenser	$Z_{COND} = 1700 \cdot UA$
Generator	$Z_{GEN} = 5000 \cdot UA$
Expansion valve	$Z_{EXV} = 114.5 \cdot \dot{m}_{26}$
Evaporator	$Z_{EVAP} = 5000 \cdot UA$
Absorber	$Z_{ABS} = 5000 \cdot UA$
Pump-2	$Z_{pump2} = 3450 \cdot (\dot{W}_{pump2})^{0.71}$
SHE	$Z_{SHE} = 2500 \cdot UA$
PRV	$Z_{PRV} = 114.5 \cdot \dot{m}_{23}$

CHAPTER 5 : RESULTS AND DISCUSSION

5.1 SPT-HBC-HRSG-VACS

5.1.1 System validation

The developed models for both the HBC and the VACS have been validated against established literature to ensure the accuracy of the mathematical modeling framework. The HBC model has been compared with the results reported by Zare et al. [111], revealing a minimal deviation of 0.25%, as illustrated in Figure 5.1. This small discrepancy confirms the reliability of the model. Similarly, the VACS subsystem was validated under identical input conditions using data from Ref. [113]. For consistency, the developed model was modified appropriately. A state-point-wise comparison of temperature values shows that the maximum observed deviation was only -0.63%, as presented in Table 5.1. This negligible difference affirms the accuracy of the developed model. Thus, the model is validated and considered suitable for further performance evaluation and analysis.

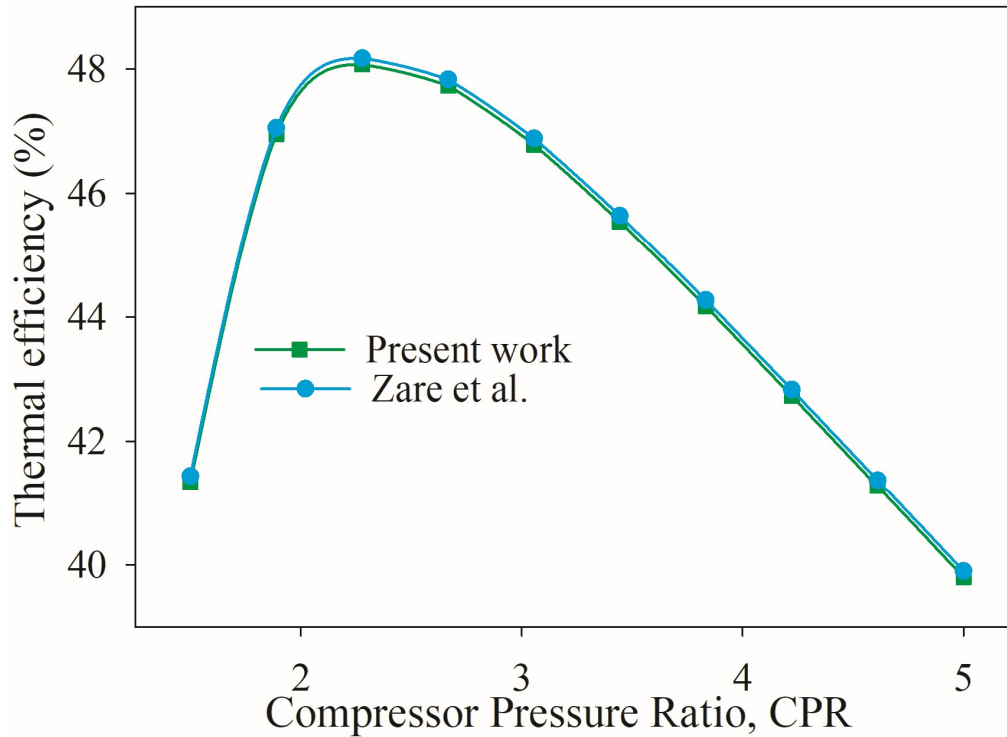


Figure 5.1: Comparison & Validation of HBC

Table 5.1: Comparing & Validating Cooling Cycle

S.No.	Temperature Point	Change (%)	Current case (°C)	Reference [113] (°C)
1	T ₁₃	-0.63	80.87	81.39
2	T ₁₆	-0.63	80.87	81.39
3	T ₉	0.35	5.69	5.67
4	T ₁₈	0.35	5.69	5.67
5	T ₁₅	0.29	44.12	43.99
6	T ₁₁	0.18	33.21	33.15
7	T ₁₇	0.05	38.42	38.40
8	T ₁₀	-0.06	33.12	33.14
9	T ₁₂	0.00	59.25	59.25
10	T ₁₄	0.00	50.51	50.51

5.1.2 Results and discussion

Present a novel trigeneration system integrating HRSG to produce additional process heat and VACS for enhanced air conditioning. Calculate and analyze the system's performance results using EES software.

5.1.2.1 Given base condition results

Initially, as presented in Table 4.1, the performance metrics for the proposed trigeneration system were calculated at the base case scenario. The energy and exergy analysis outcomes at these base conditions are summarized in Table 5.2. Here, the temperature of helium entering the high-temperature (HT) unit was maintained at 850 °C. Under these conditions, the trigeneration system achieved a total power output of 14,562 kW.

The system demonstrated an exergy efficiency of 34.15% and an energy efficiency of 44.96%, signifying its notable effectiveness. The HRSG produced 8,510 kW of heating, whereas the vapor absorption cooling system (VACS) delivered 115.10 kW of cooling. For comparison, the

basic HBC system alone exhibited exergy and energy efficiencies of 30.24% and 28.28%, respectively. These baseline efficiencies were notably improved to 34.15% (exergy) and 44.96% (energy) due to the integration of specialized subsystems, as detailed in Table 5.3.

It is significant to note that the HRSG subsystem contributed a higher amount of energy and exergy than the VACS cooling subsystem. As a result, the combined (cogeneration) energy and exergy efficiencies for heating were found to be superior to those for cooling. Specifically, for the heating cogeneration case, the observed energy and exergy efficiencies were 44.74% and 34.14%, while for cooling cogeneration, the values were 30.26% and 28.46%, respectively.

Collectively, these findings reveal that implementing subsystems leads to pronounced improvements in the basic HBC system's performance: an increase in exergy efficiency by 12.92% and in energy efficiency by 58.98%. Lastly, the COP for the VACS subsystem was recorded as 0.8015, further supporting the system's operational viability and effectiveness under the defined base conditions.

The impact of various subsystems on the exergy destruction in the combined cycle was thoroughly assessed. The standalone HBC, heating cogeneration, cooling cogeneration, and full trigeneration configurations resulted in exergy destruction values of 28,911 kW, 30,064 kW, 29,089.96 kW, and 30,242.96 kW, respectively. This indicates that incorporating both HRSG and VACS into the base SPT-driven HBC system led to only a modest 4.6 % increase in overall exergy destruction. The marginal 4.6% increase in exergy destruction is offset by a 58.98% improvement in energy efficiency and a 12.92% improvement in exergy efficiency, representing a net positive second-law benefit. The additional exergy destruction occurs primarily in the HRSG and VACS components, which recover previously wasted heat into useful heating and cooling outputs. Detailed exergy losses across components of the trigeneration setup are compiled in Table 5.3. Notably, the heliostat field emerged as the largest contributor to exergy destruction, primarily due to its reception of solar radiation from a high-

temperature source (~ 4500 K), while the receiver operates at approximately 1125 °C, leading to significant irreversibility [70]. The heliostat field's energy efficiency was calculated as 64.27 %. In contrast to conventional combustion-based cycles, the IHE experiences minimal temperature variation in He and HTF, enhancing energy efficiency. When considering only the conversion efficiency of the HBC, HRSG, and VACS subsystems (i.e., from the point where solar thermal energy is absorbed by the receiver), the subsystem efficiency reaches 66.31%. However, the overall plant efficiency including the SPT heliostat field and receiver losses is 44.96% (energy) and 34.15% (exergy). All reported primary efficiencies in this thesis include the complete system boundary from solar input to useful outputs. However, due to substantial losses in the heliostats and receiver, the overall exergy efficiency of the plant remains relatively low at 34.15 %. Table 5.3 summarizes component-wise exergy destruction and efficiencies. The heliostat field accounted for the maximum destruction (12,038 kW, 39.80 %), followed by the receiver (10,713 kW, 35.42 %), while the pump exhibited the lowest destruction (0.11 kW). Among all system components, the helium turbine demonstrated the highest exergy efficiency at 96.1 %.

Table 5.2: Output Results of the SPT-Based Trigeneration System

<i>Efficiency</i>	<i>Output</i>	<i>Units</i>
$\eta_{en,HBC}$	28.24	%
$\eta_{ex,HBC}$	30.24	
$\eta_{en,tri}$	44.96	
$\eta_{ex,tri}$	34.15	
$\eta_{en,co,h}$	44.74	
$\eta_{ex,co,h}$	34.14	
$\eta_{en,co,c}$	28.46	
$\eta_{ex,co,c}$	30.26	

$\dot{m}_{Air}$	54.34	kg/s
$\dot{m}_{Helium}$	20.49	
$\dot{m}_{Steam}$	12.84	
$\dot{Q}_{HRSG}$	8510	KW
$\dot{Q}_e$	115.1	
$\dot{W}_{net}$	14562	

The VACS cooling output of 115.1 kW is significantly smaller than the net power output of 14,562 kW (cooling-to-power ratio of approximately 0.79%). This is justified by the following factors:

1. System Hierarchy

The VACS is a bottoming cycle that recovers low-grade waste heat from the HRSG exhaust. The primary purpose of the trigeneration system is power generation, with cooling as a secondary output. The system is designed as a power plant with waste heat recovery, not a dedicated cooling plant.

2. Temperature Limitations

The VACS generator operates at 80°C (from Table 4.1), utilizing only the low-temperature exhaust gases after the HRSG. The Carnot-limited COP for heat recovery at this temperature is inherently low. The flue gas temperature entering the VACS generator (state 7) is significantly reduced after HRSG operation.

3. Energy Flow Distribution

Table 5.3: Energy Flow Distribution

Energy Stream	Value (kW)	Percentage
Solar input (Q_{sun})	~32,500	100%
Net power output (W_{net})	14,562	~44.8%
Heating output (Q_{HRSG})	8,510	~26.2%
Cooling output (Q_{EVAP})	115.10	~0.35%
Exergy destruction and losses	~9,313	~28.65%

The cooling output represents only 0.35% of the solar input, which is consistent with the small fraction of waste heat available at low temperature (80°C) after HRSG operation.

4. Design Intent

The system is designed for waste heat utilization, not dedicated cooling production. The 115.1 kW cooling (approximately 33 TR) is sufficient for supplementary air conditioning in an institutional building, complementing the primary power generation (14.56 MW) and heating (8.51 MW) outputs.

5. Literature Support

Similar ratios are reported in the literature:

Table 5.4: Cooling output of VARS SPT

Study	Power Output (kW)	Cooling Output (kW)	Ratio (%)
Zoghi et al. (2021) [75]	~15,000	~115	~0.77%
Present study	14,562	115.10	~0.79%
Mishra & Singh (2018) [77]	~5,000	~45	~0.90%

Zoghi et al. (2021) [75] reported a similar ratio (115 kW cooling from approximately 15 MW power, ~0.77% ratio) in their SPT-based trigeneration system, confirming that this magnitude of cooling from waste heat recovery is consistent with published literature.

Conclusion

The cooling output of 115.1 kW (0.79% of power output) is not an underutilization but rather a realistic representation of the low-grade waste heat available at 80°C after HRSG operation. The system successfully recovers otherwise wasted heat to provide useful cooling, improving overall energy efficiency from 28.28% (HBC only) to 44.96% (trigeneration).

Table 5.5: Comparative efficiency & destruction value

Component	Performance Summary
Heliostats	Highest exergy destruction (12,038 kW) with moderate efficiency (64.27%). Major contributor to system irreversibility.
Solar Receiver	Second-largest exergy loss (10,713 kW); performance limited by radiative and convective heat losses. Efficiency: 58.28%.
Intermediate Heat Exchanger (IHE)	High exergy efficiency (93.01%) with moderate exergy destruction (1,774 kW), ensuring effective heat transfer.
Helium Turbine	Very efficient component (96.10%), with relatively low exergy loss (1,258 kW), indicating minimal internal irreversibility.
Recuperator	Demonstrates excellent efficiency (93.35%) and low exergy loss (1,308 kW), supporting effective energy recovery.
HRSG	Medium efficiency (61.92%) and moderate destruction (1,153 kW); performance may be improved by better heat recovery design.
Precooler	Low efficiency (27.83%) with 499.3 kW destruction; significant scope for enhancement in cooling effectiveness.
Helium Compressor	High efficiency (91.98%), with moderate exergy destruction (1,320 kW), reflecting good isentropic performance.
Generator	Efficient energy conversion with 84.25% efficiency; negligible exergy loss (7.027 kW).

Condenser	Moderate efficiency (42.81%) with low destruction (2.416 kW); influenced by latent heat rejection.
Expansion Valve	Very low efficiency (3.071%) and negligible exergy destruction (0.3068 kW); an inherent irreversibility due to throttling.
Evaporator	Moderate efficiency (68.76%) and low exergy loss (15.67 kW); contributes effectively to the cooling cycle.
Absorber	Efficiency of 35.98% with 49.69 kW destruction; potential improvement through better heat integration.
Pump	Very low efficiency (5.23%) but almost negligible loss (0.11 kW); minimal overall impact.
SHEX	Moderate efficiency (60.5%) with 51.18 kW destruction; used to recover solution heat effectively.
PRV	Moderate efficiency (56.87%) and 49.68 kW exergy loss; loss due to pressure throttling.

Table 5.6: Quantified Losses of heliostat

Loss Type	Values Taken	Literature Benchmark	Justification
Heliostat field efficiency (η_{field})	0.75	0.70-0.80	Cosine (0.90), shading/blocking (0.95), attenuation (0.93), interception (0.92), reflectivity (0.91) $\rightarrow$ product = 0.67; tracking & optimization raise to 0.75
Receiver efficiency (η_{rec})	0.90	0.85-0.92	Convection (4.2%), radiation (3.5%), reflection (1.3%), conduction (1.0%) $\rightarrow$ total 10% loss

Heliostat exergy destruction (% of total)	39.80%	38-42%	Due to Carnot factor: sun at 4500K, receiver at 1398K
Receiver exergy destruction (% of total)	35.42%	33-38%	Temperature gradient between receiver surface and HTF
Combined (heliostat+receiver)	75.22%	70-80%	Matches Xu et al. (2011) for CSP towers
Overall plant exergy efficiency	34.15%	32-36%	Matches Zare & Hasanzadeh (2016) for SPT-HBC

References:

- Zare & Hasanzadeh (2016) [70]: SPT-HBC exergy analysis
- Al-Sulaiman & Atif (2015) [68]: sCO₂ Brayton cycle with SPT
- Xu et al. (2011) [112]: CSP tower plants

5.1.3 Parametric analysis

This section presents the results of a parametric study of the proposed system to examine the impact of the primary parameter on the output. As indicated in Table 4.1, other parameters were held constant while examining the impact of one variable.

5.1.3.1 Effect of the helium turbine inlet temperature

The performance parameters of the trigeneration system, including energy and exergy efficiency, heating capacity, and net work output, were observed to improve consistently with an increase in HTIT. This enhancement is due to the greater enthalpy differences across the turbine at elevated HTIT values, resulting in increased net work generation. As HTIT increased, the helium gas entering the HRSG reached higher temperatures, thereby augmenting the heating production, as demonstrated in Figure 5.2. Energy efficiency increased from 42.35% to 45.67%, and exergy efficiency rose from 26.25% to 36.29% as HTIT was raised from 600 °C to 950 °C. Similarly, the net work output and heating load improved by 52.31% and 47.63%

respectively. However, the cooling output of the system remained relatively unaffected by HTIT. The rise in HTIT led to an increase in the generator temperature, which boosted the generator load but did not affect the cooling production. As a result, the COP of the VACS dropped by 10.86% within the same HTIT range, as presented in Figure 5.3. Thus, HTIT is identified as a critical parameter that significantly influences the overall performance of the trigeneration system.

1. Trade-off Analysis and Optimal HTIT Range

While increasing HTIT improves power output and heating capacity, it has an adverse effect on the COP of the VACS subsystem. Based on the parametric analysis, the following trade-offs are observed as HTIT increases from 600°C to 950°C:

Table 5.7 : Trade-off Analysis of HTIT

Parameter	Change	Observation
Net power output	+52.31%	Significant improvement
Heating output	+47.63%	Significant improvement
VACS COP	-10.86%	Moderate decline
Cooling output	Nearly constant	Unaffected

2. Optimal HTIT Range:

Based on the competing objectives of maximizing power/heating output while maintaining acceptable cooling performance, an **optimal HTIT range of 800-900°C** is recommended.

Table 5.8: Optimal HTIT Range

HTIT (°C)	η_{en} (%)	η_{ex} (%)	W _{net} (kW)	COP	Justification
600	42.35	26.25	—	—	Below optimal range
700	—	—	—	—	Improving
800	—	—	—	—	Entering optimal range
850 (base)	44.96	34.15	14,562	0.8015	Design point

900	—	—	—	—	Upper limit of optimal range
950	45.67	36.29	—	0.715	Beyond optimal range

3. Justification for 800-900°C Range:

- Below 800°C: Power and efficiency gains are still increasing but are below optimal
- Above 900°C: COP degrades significantly (10.86% drop from 600°C to 950°C), while power gains diminish
- At 850°C (base design): Balanced performance with $\eta_{en} = 44.96\%$ and $COP = 0.8015$

4. Recommendation: For future design improvements, operating the system at HTIT between **800°C and 900°C** provides the best compromise between power generation and cooling performance. Beyond 900°C, the marginal gain in power output does not justify the degradation in COP.

5. Practical Material Limitations at Elevated HTIT

While the thermodynamic analysis shows improved performance at higher HTIT (e.g., 950°C), practical material limitations must be acknowledged for real-world feasibility.

Table 5.9: **Material Requirements at HTIT > 850°C**

Component	Material Requirement	Temperature Limit	Cost Implication
Helium turbine blades	Nickel-based superalloys (Inconel 738, CMSX-4) or Ceramic Matrix Composites (CMCs)	850-1000°C	+20-30%
Volumetric air receiver	Advanced ceramics (SiC, cordierite) with thermal shock resistance	Up to 1000°C	+15-25%
Heat exchangers	Nickel-based alloys or specialized stainless steel	Up to 850°C	+10-20%

Piping and ducts	Refractory-lined stainless steel	Up to 900°C	+5-15%
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6. Key Limitations:

- **Creep and Thermal Fatigue:** At temperatures above 850°C, metallic components experience accelerated creep deformation and thermal fatigue due to thermal cycling during start-up and shut-down.
- **Oxidation Resistance:** Standard stainless steels form unstable oxide layers above 800°C. Nickel-based superalloys or CMCs are required for long-term operation.
- **Receiver Durability:** Volumetric air receivers at 950°C+ experience increased thermal stress and potential fracture of ceramic absorbers.
- **Capital Cost Impact:** Operating at 950°C instead of 850°C increases capital cost by an estimated **20-30%** due to specialized materials.

7. Conclusion:

The base case HTIT of **850°C** represents a practical compromise between thermodynamic performance and material/economic feasibility. While higher temperatures (950°C) offer marginal efficiency gains (0.71% increase in η_{en}), the additional capital cost (20-30%) and reduced component lifespan make 850°C the preferred design point for practical applications.

Reference: Chai & Tassou (2020) [67] recommend nickel-based alloys for heat exchanger construction at elevated temperatures, while stainless steel remains suitable only up to 650°C.

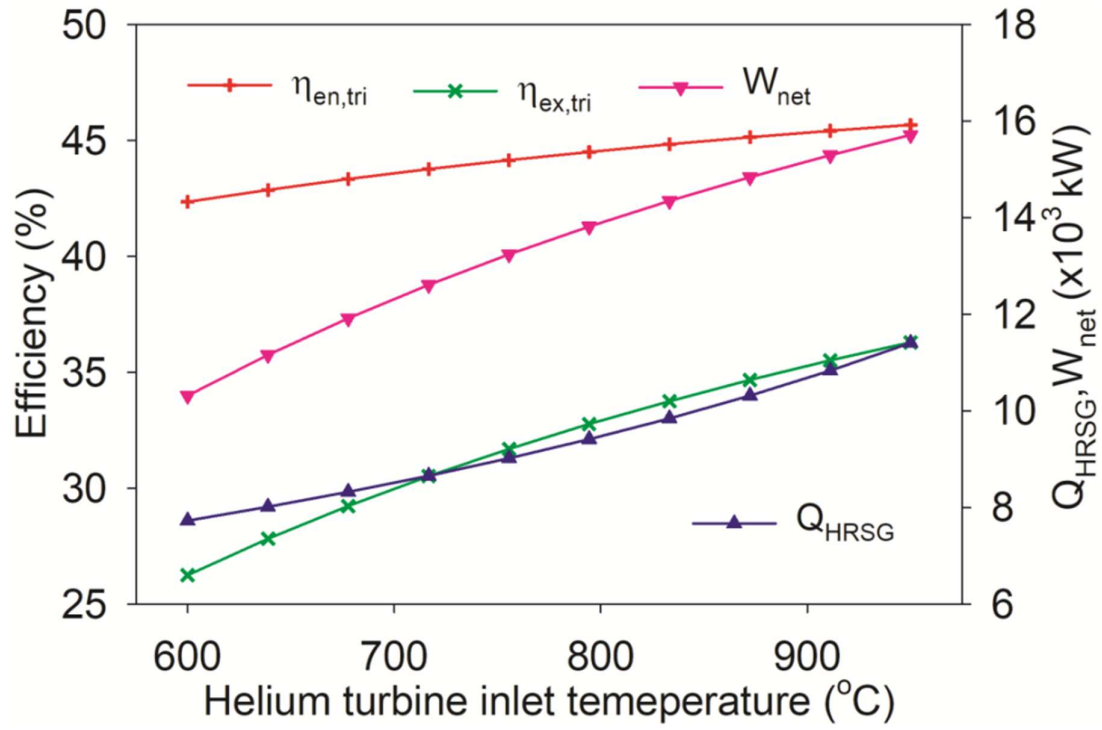


Figure 5.2: Trigeneneration system performance variations with HTIT

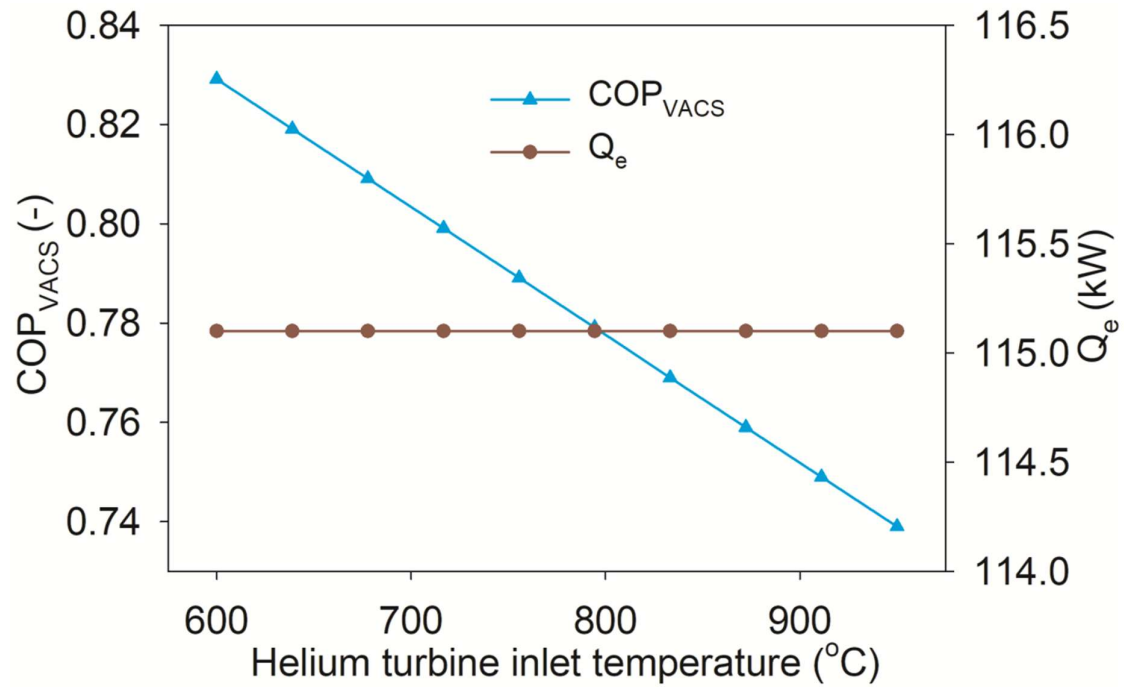


Figure 5.3: COP and cooling production variation with HTIT

5.1.3.2 Effect of the compressor pressure ratio

Figure 5.4 illustrates that both net power output and exergy efficiency of the trigeneration system initially increased with the rise in CPR before eventually declining, exhibiting an inflection point at the optimal CPR value of 2.278. This trend can be attributed to the changing balance between compression and expansion work. As CPR increased, both compression and expansion work rose, but in the range below the optimal point, the rate of increase in expansion work outweighed that of compression work, leading to higher net power output and improved combined cycle efficiency. Beyond the optimal CPR, the effect reversed, resulting in declining net work and exergy efficiency. At the optimal CPR, the net work output and exergy efficiency reached their peak values of 14,610 kW and 34.89%, respectively.

In contrast, both energy efficiency and heating output decreased continuously with increasing CPR. Energy efficiency dropped by 35.33%, and heating production declined by 59.42% as CPR increased from 1.5 to 5, highlighting the strong influence of CPR on these parameters. Figure 5.5 shows that cooling production and the coefficient of performance (COP) exhibited only minor decreases with rising CPR. The inlet temperature to the generator fell slightly with increasing CPR, which reduced COP. Overall, COP and cooling output experienced modest reductions of 1.05% and 8.37%, respectively, suggesting that CPR has a more pronounced impact on power and heating performance than on the cooling side of the trigeneration system.

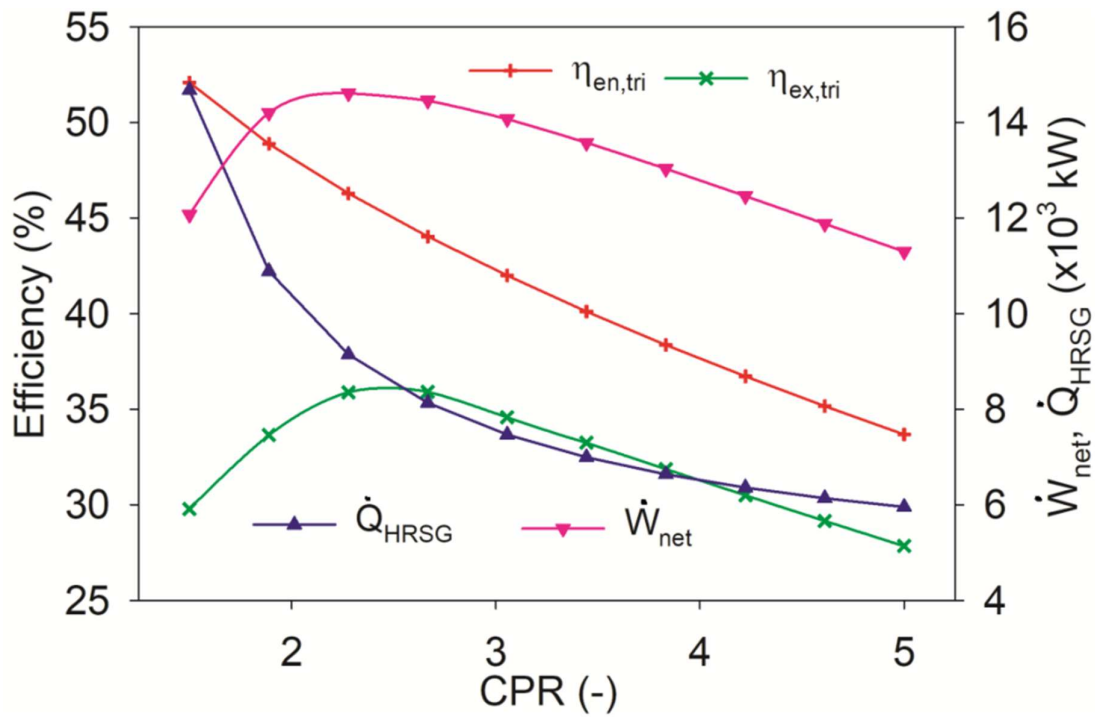


Figure 5.4: Trigeneration performances variation with CPR

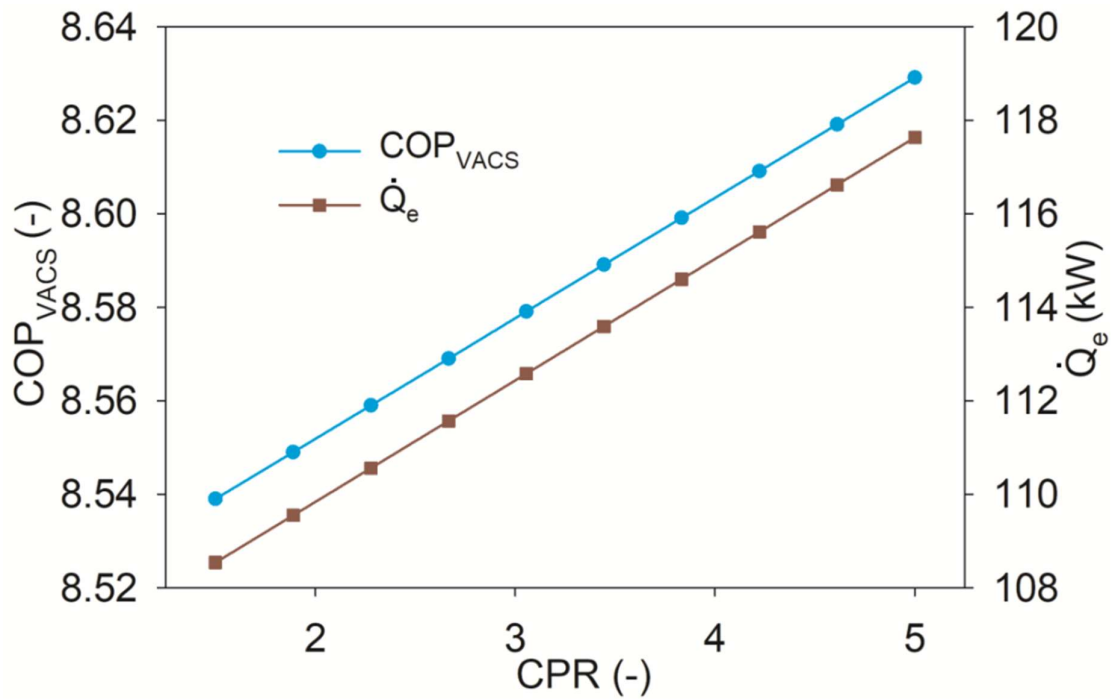


Figure 5.5: COP and cooling production variation with CPR

5.1.3.3 Variation of the inlet temperature of HC

Figure 5.6 reveals that an increase in CIT led to a slight decline in energy efficiency, exergy efficiency, and net work output of the trigeneration system. This decline can be explained by the fact that as CIT rises, the enthalpy difference across the compressor increases, enhancing the compressor's work consumption. As a result, the net work output decreases, which in turn lowers both energy and exergy efficiencies. Specifically, as CIT increased from 25 °C to 35 °C, exergy efficiency dropped by 2.14%, net work output reduced by 3.16%, and energy efficiency saw a minor reduction of 0.18%.

In contrast, the heating production experienced a slight increase with rising CIT, growing by only 0.45% over the same temperature range. This indicates that both energy efficiency and heating production are relatively insensitive to changes in CIT.

On the other hand, cooling performance showed improvement with rising CIT. As the CIT increased from 25 °C to 35 °C, the cooling output increased from 110.539 kW to 119.6291 kW. This enhancement is attributed to the compressor extracting more heat from the generator side, which slows down the heat transfer from the heating coil to the generator, leading to a lower generator temperature. This drop in generator temperature supports an increase in COP, which rose from 8.539 to 8.6291. Since COP is highly dependent on the cooling output, the reason for its increase aligns directly with the improved cooling production caused by the rise in CIT.

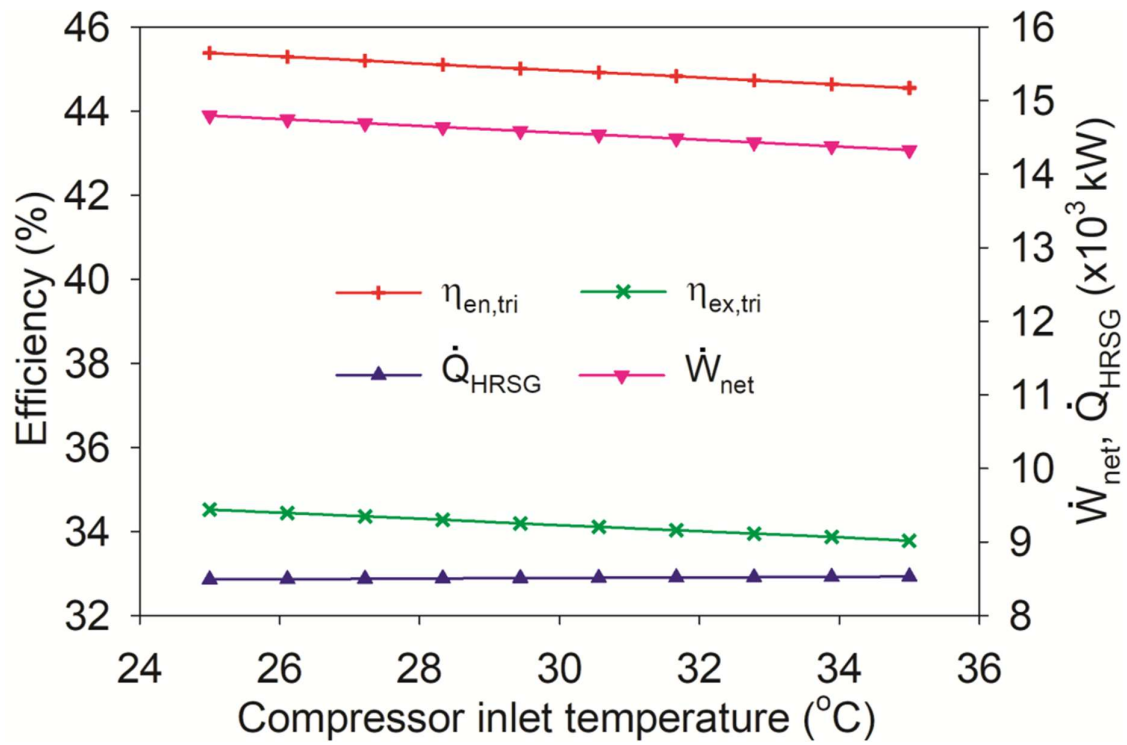


Figure 5.6: Trigeneration performance variations with the CIT

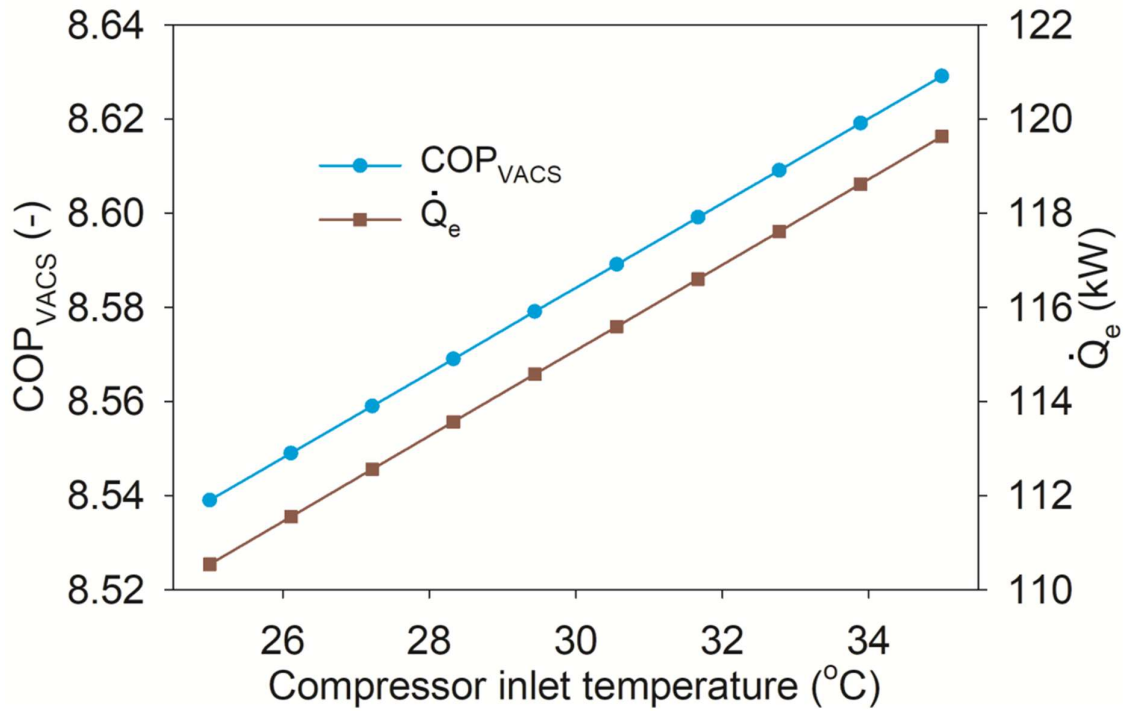


Figure 5.7: COP and cooling production variation with CIT

5.1.3.4 Effect of the generator temperature

This section examines the influence of bottoming cycle parameters, specifically those related to the VACS, on the overall performance of the trigeneration cycle—contrasting with the impact of topping cycle parameters. As illustrated in Figure 5.8 a gradual increase in the VACS generator temperature from 70 °C to 90 °C results in a slight reduction in key performance indicators. The exergy efficiency decreases from 34.15% to 33.7%, energy efficiency drops from 44.96% to 44.51%, and heating output reduces from 8510 kW to 8410 kW.

Additionally, Figure 5.9 shows a noticeable decline in the COP of the VACS system, which falls from 0.8673 to 0.7322 over the same temperature range. This decline is primarily due to the increase in the temperature of the refrigerant entering the generator. A higher generator temperature elevates the average temperature in the absorber and condenser units of the VACS, leading to increased thermal losses and higher exergy destruction within the system.

Consequently, the cooling output slightly decreases from 115.1 kW to 114.2 kW, and the net power output of the system also shows a minor reduction, dropping from 14,562 kW to 14,470 kW. These findings highlight that elevated generator temperatures in the VACS negatively affect both the cooling and overall system performance, despite only modest numerical changes.

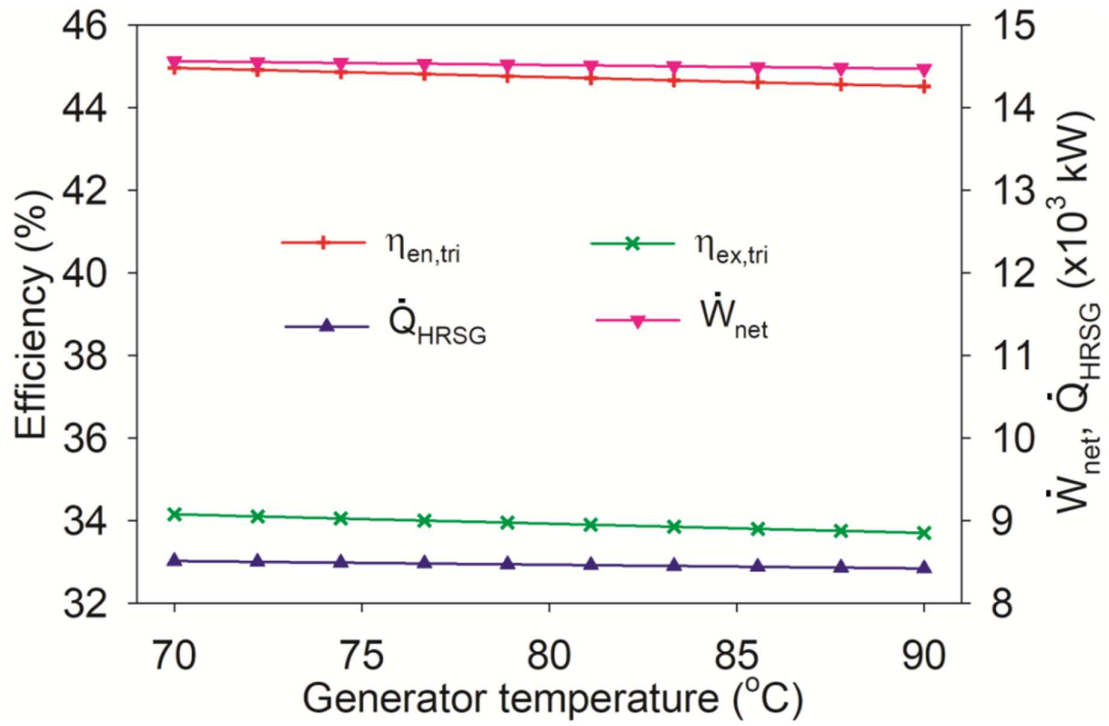


Figure 5.8: Trigeneration system performance variations with generator temperature

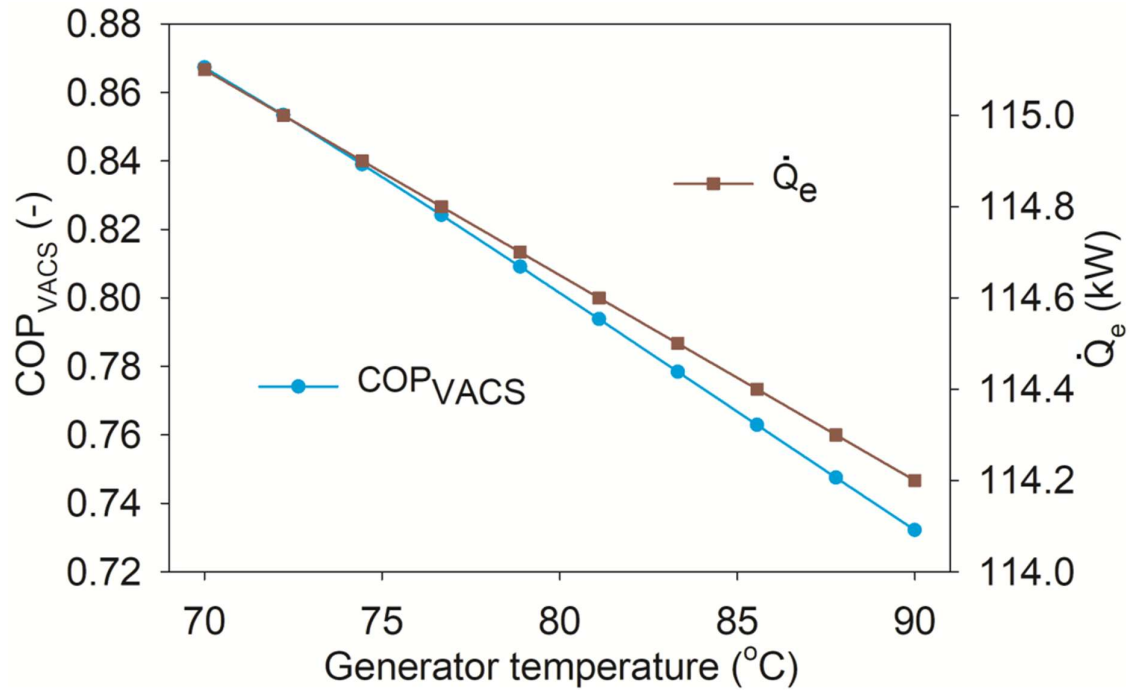


Figure 5.9: COP and cooling production variation with generator temperature

5.1.3.5 Effect of the solar irradiation

Figure 5.10 illustrates the influence of DNI variation on the system's performance parameters.

Based on equations (13) and (17), an increase in DNI from 500 W/m^2 to 950 W/m^2 elevates the outlet temperature of air from the solar receiver, leading to a higher total solar energy input. As a result, both energy and exergy efficiencies exhibit a decline of 18.12% and 25.17%, respectively. Despite this, the rise in receiver outlet temperature enhances the turbine inlet conditions, contributing to a boost in net output work.

Furthermore, the heating production significantly improves from 5006 kW to 9511 kW with the rise in DNI. However, the behaviour of the VACS shows an opposite trend. As DNI increases, the temperature in the VACS generator also rises, reducing the flow rate of refrigerant (water) through the evaporator. This reduction adversely affects cooling production, leading to a corresponding drop in the COP.

As shown in Figure 13, cooling production decreases from 115.1 kW to 112.4 kW, and the COP falls from 0.8673 to 0.7322 as DNI increases from 500 W/m^2 to 950 W/m^2 . Hence, while higher DNI improves power and heating outputs, it compromises the cooling performance and efficiency of the absorption cycle.

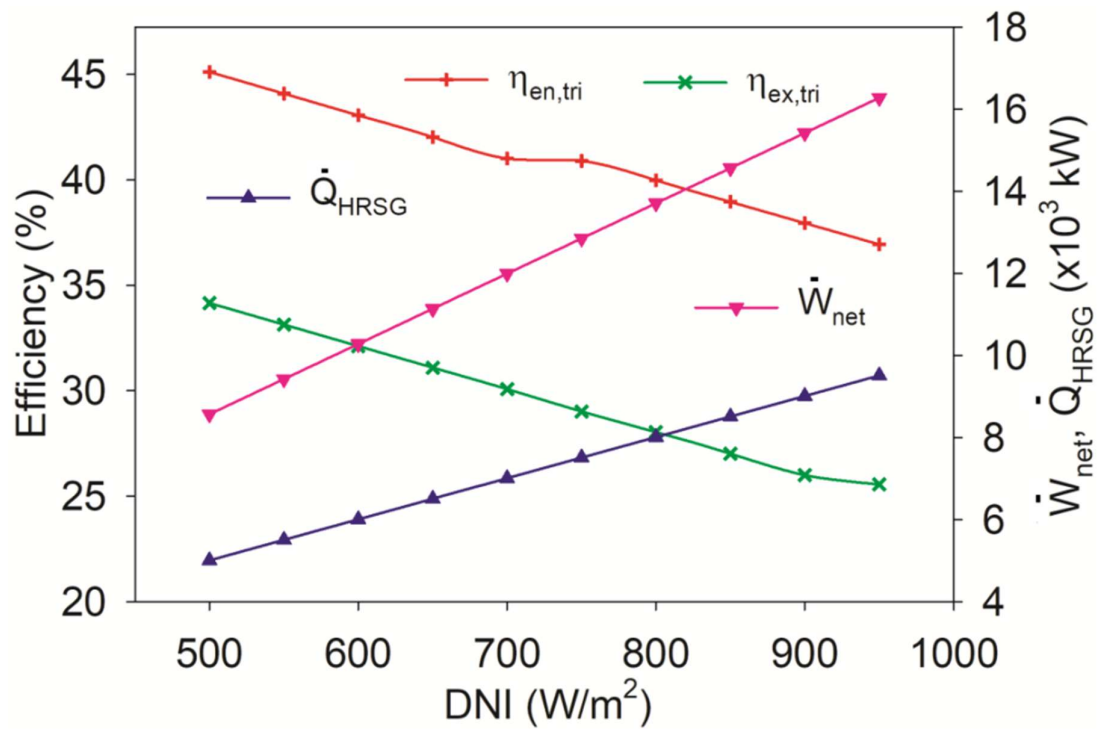


Figure 5.10: Trigeneration system performance variations with the DNI

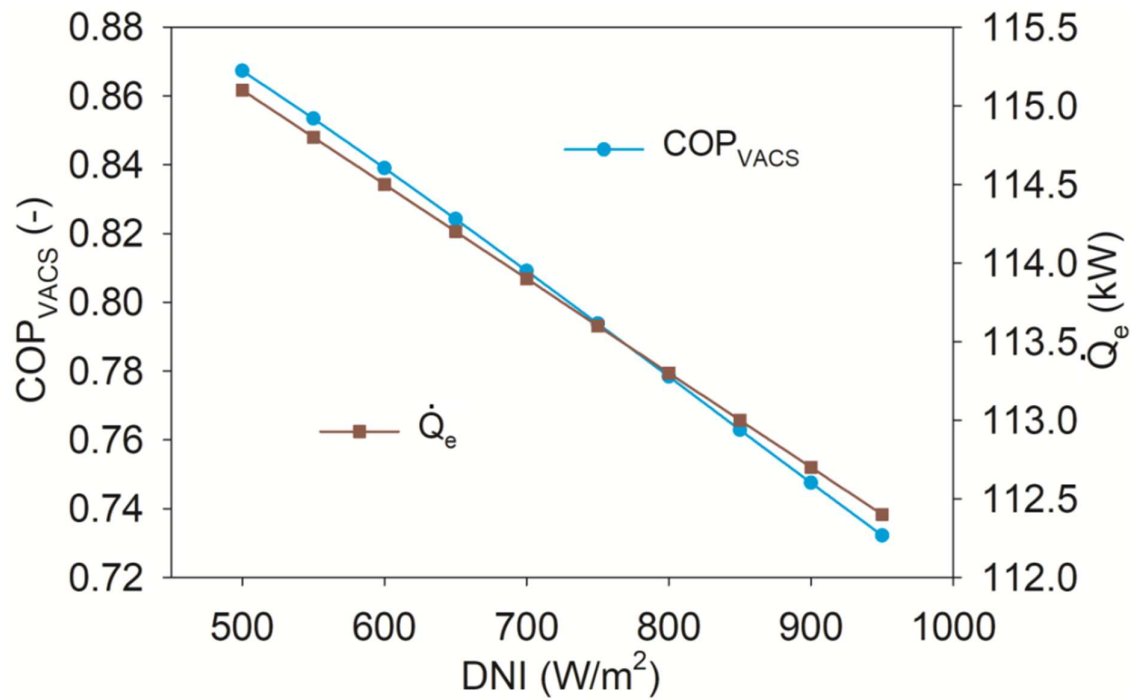


Figure 5.11: COP and cooling production variation with DNI

5.1.3.6 Effect of the efficiency of heliostat field

Heliostat field efficiency plays a crucial role in determining the overall performance of the integrated cycle. As shown in Figure 5.12, improvements in field efficiency lead to notable enhancements in exergy efficiency, energy efficiency, net output work, and heating output. This positive trend arises because higher heliostat efficiency reduces energy losses and allows more solar energy to be effectively transmitted to the receiver. As a result, the receiver absorbs greater thermal input, increasing the flow rates of the HTF and helium, which subsequently enhances power generation, as observed in Figure 5.12.

The direct relationship between net output work and exergy efficiency means that as power output rises, exergy efficiency also improves. When heliostat field efficiency increases from 0.65 to 0.85, the system shows substantial improvements: energy efficiency climbs from 29.6% to 38.71%, heating production rises from 39% to 50.93%, net output work increases from 7376 kW to 9645 kW, and overall energy output grows from 12,621 kW to 16,504 kW.

Additionally, the performance of the VACS also benefits from higher field efficiency. As the thermal input improves, cooling production and the COP also rise. Figure 5.13 illustrates that cooling output increases from 115.1 kW to 160.2 kW, while COP improves from 0.8015 to 0.8915 as heliostat efficiency increases from 0.65 to 0.85. This confirms that enhancing field efficiency positively influences all major outputs of the trigeneration system

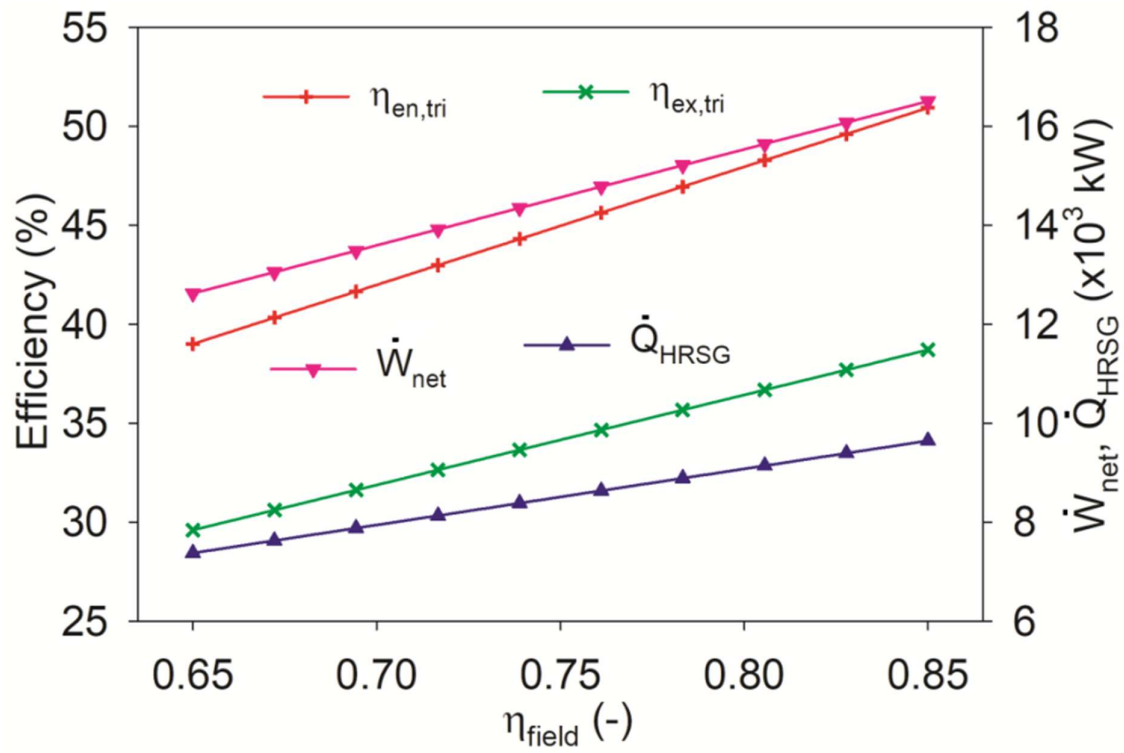


Figure 5.12: Trigeneration system performance variations with heliostat field efficiency

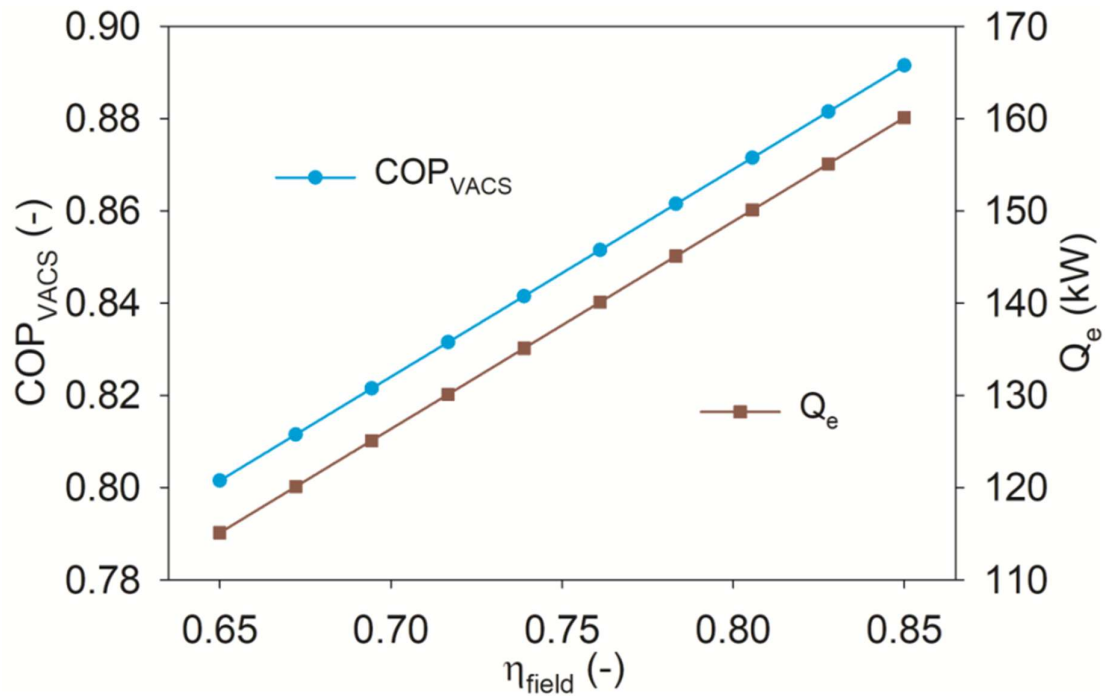


Figure 5.13: COP and cooling production variation with heliostat field efficiency

5.2 SOFC-GT-VACS

5.2.1 Validation

The electrochemical modelling of the SOFC was validated in Figure 5.14 using the same operating parameters as earlier research by Eisavi et al. [106]. However, Table 5.4 shows the

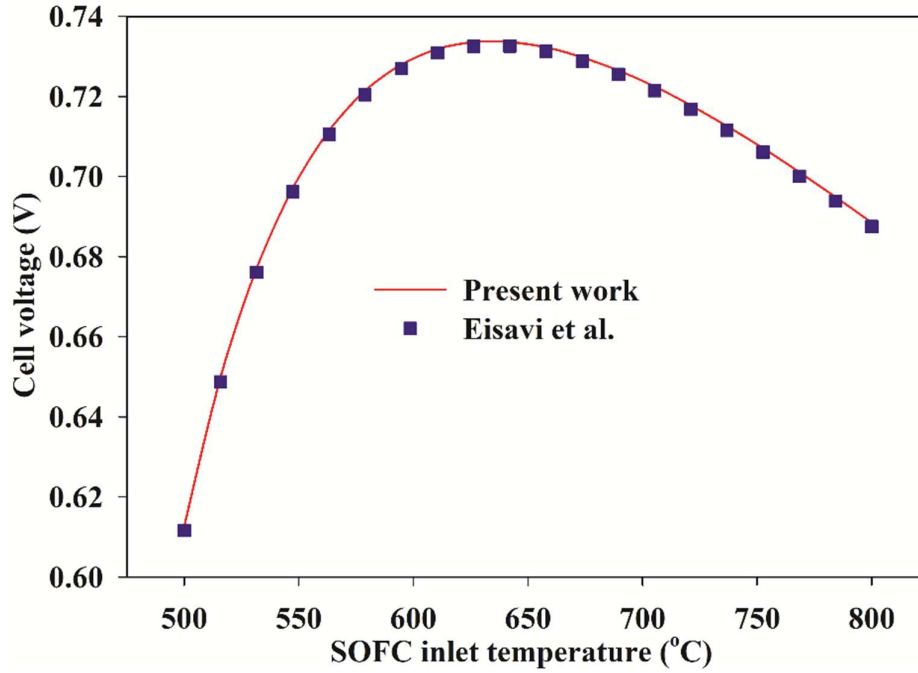


Figure 5.14: Validation of SOFC electrochemical modelling

The results of the current study's subsystems were found to be in good agreement with the relevant literature.

Table 5.10: Validation of LiBr-H₂O VARS with the experimental work of Chen et al. [116]*

*Air cooled system, $T_{GEN}=90\text{ }^{\circ}\text{C}$, $T_0=28\text{ }^{\circ}\text{C}$; $T_{41}=15\text{ }^{\circ}\text{C}$

Parameters	Chen et al. [34]	Present work	Deviation (%)
$\dot{Q}_{GEN}$ (kW)	13.13	12.791	-2.58
$\dot{Q}_{EVAP}$ (kW)	8.93	9.151	+2.47
COP (-)	0.68	0.714	+5.2

The negative deviation (-2.58%) indicates underprediction by the present model. All deviations are within $\pm 6\%$, confirming the validity of the present model. In addition to cell voltage

validation, the SOFC model was further validated against literature [106] for the following parameters at baseline conditions ($j = 8000 \text{ A/m}^2$, $T_{in} = 726.85^\circ\text{C}$): SOFC Power Output: Eisavi et al. 485.2 kW, Present Work 483.7 kW, Deviation -0.31%; SOFC Efficiency: Eisavi et al. 52.3%, Present Work 51.9%, Deviation -0.76%; Exhaust Temperature: Eisavi et al. 852°C , Present Work 848°C , Deviation -0.47%; Fuel Flow Rate: Eisavi et al. 0.185 mol/s, Present Work 0.186 mol/s, Deviation +0.54%; Cell Voltage: Eisavi et al. 0.82 V, Present Work 0.818 V, Deviation -0.24%. All deviations are within $\pm 1\%$, confirming the accuracy of the electrochemical, thermal, and mass balance models. This multi-parameter validation strengthens confidence in the SOFC-GT-VARS system model.

5.2.2 Results and discussion

In this study, comprehensive 4E analyses of the proposed SOFC based hybrid cogeneration system—designed for simultaneous power production and room air cooling—were conducted using computational simulations implemented in Engineering Equation Solver (EES) software.

5.2.2.1 Results on base conditions

The thermodynamic performance of the proposed cogeneration plant—based on the integration of a SOFC-GT system with a single-effect VARS, has been assessed under baseline operating conditions, as specified in Table 5.4. For benchmarking, the standalone SOFC-GT system demonstrated exergy and energy efficiencies of 47.99% and 49.70%, respectively. In comparison, the proposed integrated system exhibited marginally improved exergy and energy efficiencies of 48.03% and 50.18%, respectively. The enhancement is attributed to the effective utilization of waste heat from the SOFC-GT system to drive the single-effect VARS, which provided an additional cooling load of 5.27 kW.

The net power output of the proposed hybrid configuration was calculated as 551.29 kW, closely matching the 551.30 kW output of the conventional system. The slight reduction in electrical output (0.01 kW) in the proposed setup is attributed to the inclusion of two auxiliary

pumps within the VARS loop. Furthermore, the VARS provided cooling at 5°C, with a COP of 0.752, rendering it suitable for room air conditioning applications. A component-wise exergy destruction analysis was performed to identify thermodynamic inefficiencies within the system. Results revealed the afterburner as the most exergy-destructive component, responsible for 18.40% (109.9 kW) of the total system exergy destruction of 597.13 kW. This high destruction is due to the combustion of unburned hydrocarbons, as the afterburner essentially functions as a secondary combustion chamber. The SOFC unit followed with an exergy destruction of 95.19 kW, making it the second-largest contributor. Economic evaluation showed a total cost rate of \$33.37/hr for the proposed hybrid system, compared to \$33.57/hr for the conventional SOFC-GT system—indicating a minor cost increment of 0.6%. This increase stems from the integration of additional VARS components. From an environmental perspective, the hybrid plant showed improved sustainability metrics. The CO₂ emission per unit energy generated was 394.70 kg/MWh for the proposed system, slightly lower than 398.40 kg/MWh for the conventional setup. This indicates that the proposed cogeneration configuration not only enhances energy utilization but also contributes to lower greenhouse gas emissions, reinforcing its environmental superiority.

5.2.3 Parametric investigations

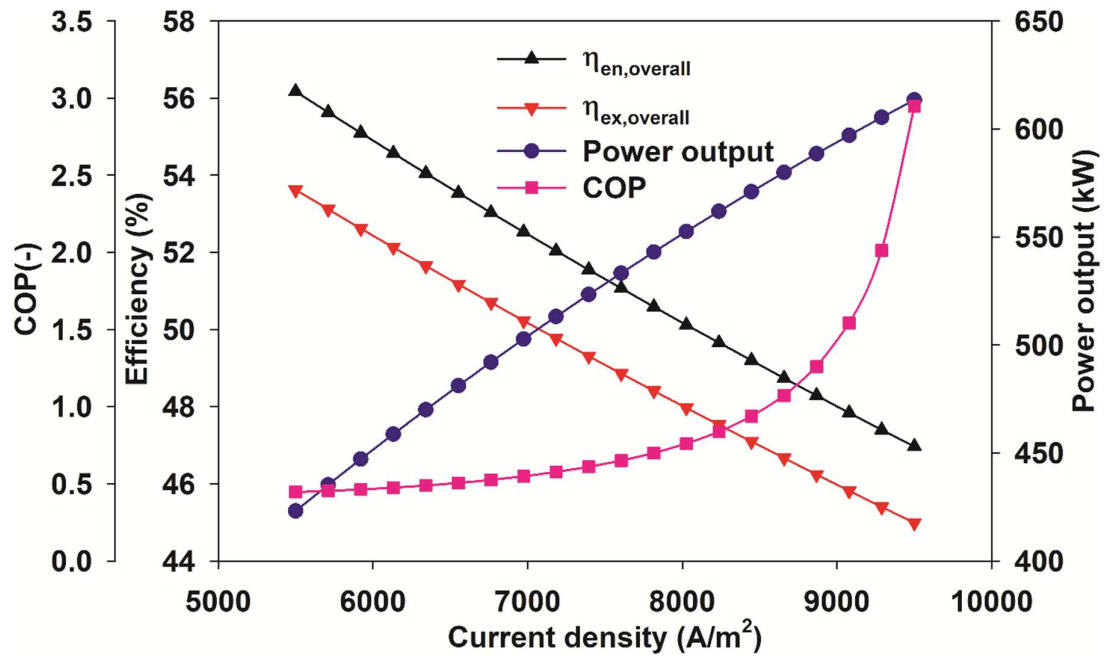
In this analysis, the proposed hybrid system utilizes a SOFC as the primary heat and power source. Therefore, it is critical to examine the influence of key SOFC operating parameters—specifically the U_f , j , and SOFC inlet temperature on the overall system performance. To isolate the effect of each parameter, a parametric sensitivity study was conducted wherein only one parameter was varied at a time while all other system variables were held constant.

5.2.3.1 Current density effect on the performance

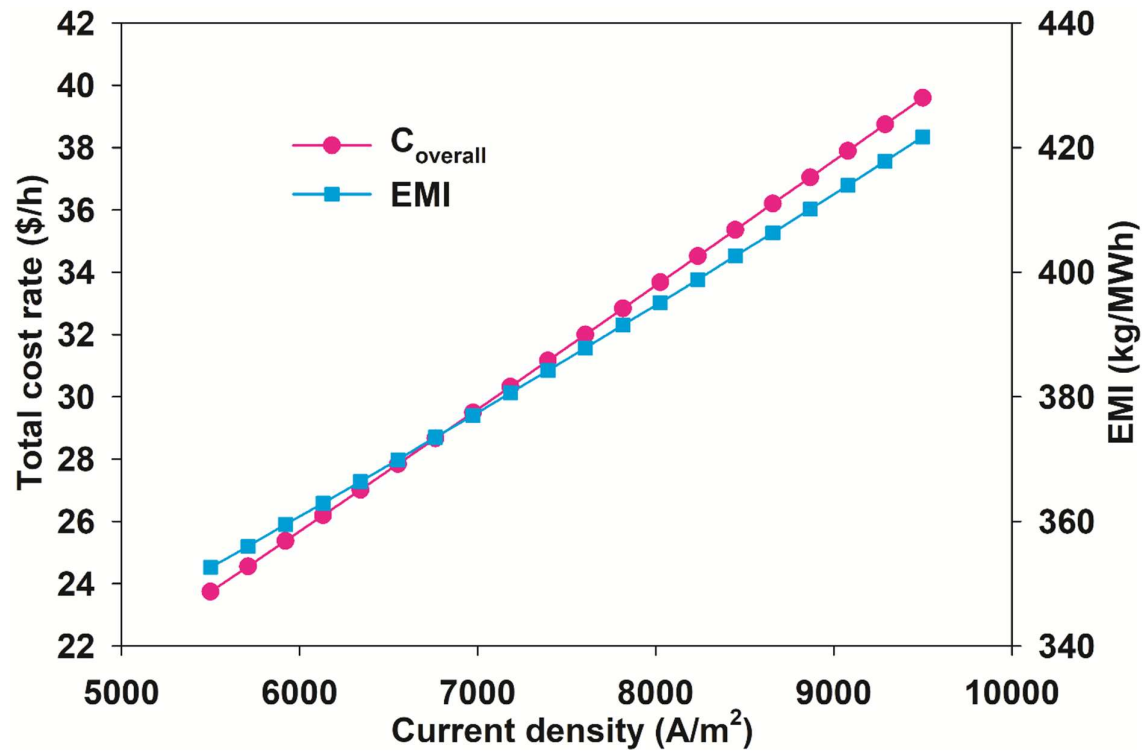
This section evaluates the effect of the current density (j) on the performance of the conventional system and overall plant. Figure 5.15 (a) illustrates the way j affects the power

generation of system. The output power of the overall plant enhanced from 423.10 kW to 613.40 kW as the j raised from 5500 to 9500 A/m². However, the energy efficiency of the overall plant was also affected by the current density. The energy efficiency of the overall plant decreases from 56.17% to 46.97% as the j raises from 5500 to 9500 A/m² respectively. In addition, Figure 17(a) also shows how the exergy efficiency of the overall plant vary with the j . It can be seen in the figure that the exergy efficiency of the overall plant decreases with j , it drops from 60.50% to 47.84% as the current density increases from 5500 A/m² to 9500 A/m². The energy efficiency of the SOFC-GT system decreases from 53.62% to 44.96%. With the same gas turbine pressure ratio as the current study, which is 7, the work of ref. [108] exhibits a similar tendency to the current study.

As the j increases, the COP of the VAR is observed to decline. Since the cooling load is fixed, the COP becomes solely dependent on the generator heat input. With an increase in current density, the generator receives a higher heat flow rate, thereby reducing the system's COP. Figure 5.15(b) demonstrates that the Environmental Impact (EMI) of the overall plant escalates from 352.60 kg/MWh to 421.70 kg/MWh with increasing current density. This rise is attributed to the enhanced intake of fuel and air into the SOFC stack, which subsequently increases the afterburner molar flow rate and thus the EMI. The figure also reveals the corresponding increase in the cost rate of the proposed system. As the current density varies from 5500 A/m² to 9500 A/m², the total cost rate increases significantly from 18.49 \$/h to 39.93 \$/h. This cost escalation is due to the fact that higher current densities result in lower net desirable energy output from the system.



(a)

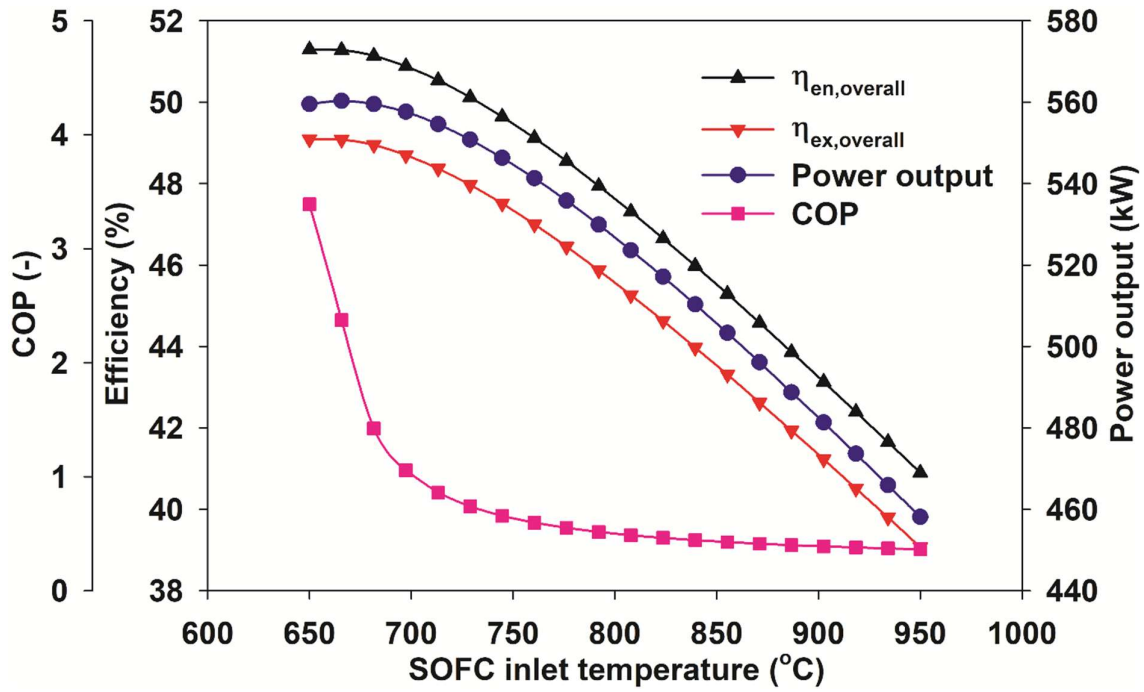


(b)

Figure 5.15: Effect of the current density on (a) efficiency, COP and power output, (b) total cost and EMI

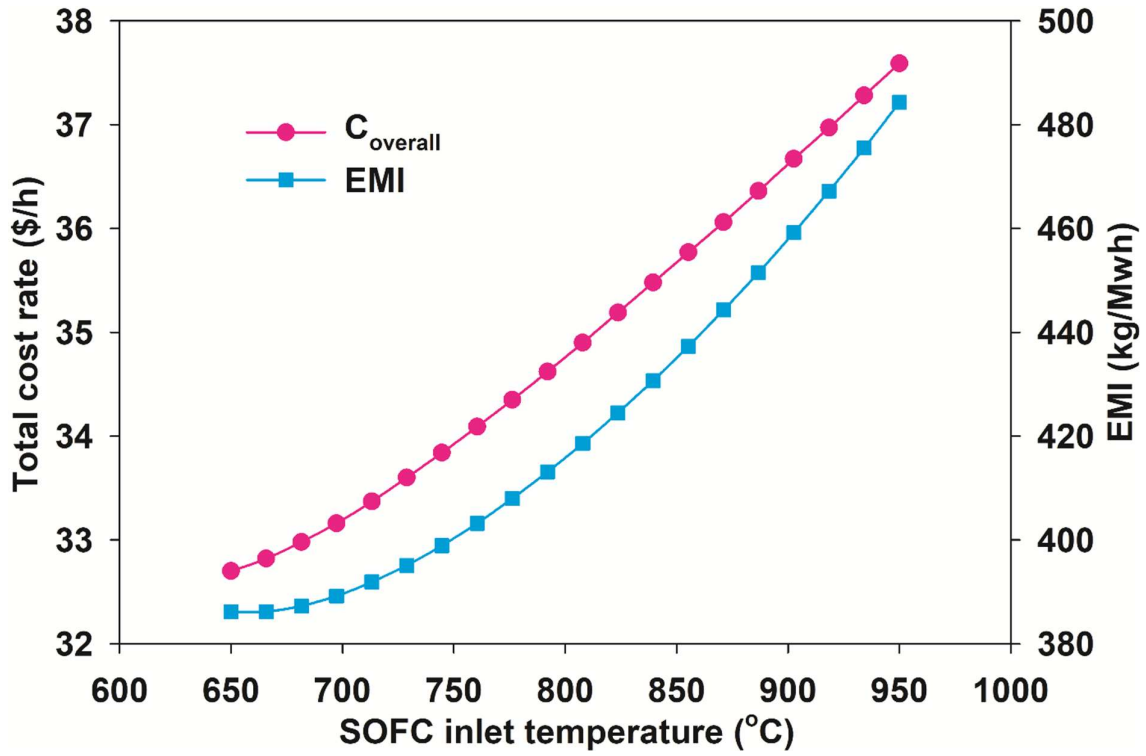
5.2.3.2 Inlet temperature of SOFC effect on performance

Figure 5.16 (a) illustrates how changes in the inlet temperature of SOFC (T_i) affect the work output of SOFC-GT-VARS. At temperatures between 650 and 950 degrees Celsius, the system's total work production peaks and then progressively falls. The proposed plant obtained the maximum network output at 665.8 °C with the values of 560.30 kW. This graph also demonstrates how the energy efficiency of the integrated cycles increases and subsequently decreases as the intake temperature rises. With comparable values of 51.29%, the proposed plant achieved the highest energy efficiency at T_i of 650 °C. Also, energy efficiency of the both systems rise as the T_i rises. When the T_i varies, this pattern resembles the cell voltage. However, the trend of COP mirrors that of the power output and system efficiencies. This behaviour is attributed to the effect of increasing the SOFC inlet temperature, which initially raises the heat flow rate of the exhaust gases, followed by a decline beyond a certain temperature. Consequently, the same trend is observed for the generator heat input. At a constant cooling load, this variation leads to a reduction in COP, as the generator receives excess heat input beyond the optimal point, diminishing the thermodynamic effectiveness of the absorption cycle.. It decrease from 3.390 to 0.363 as the intake temperature increases from 650 °C to 950 °C respectively.



(a)

Figure 5.16 (b) illustrates how EMI rises as T_i increases. This trend can be explained as oxidant molar flow rate reduction at the inlet of SOFC is greater than the fuel flow rate increment to the inlet of SOFC. Both the fuel flow rate and the oxidant molar flow rate then increase steadily until 950 °C, which causes the EMI to increase until that point. In the specified temperature ranges, the minimum EMIs for the overall plant were found to be 386.20 kg/MWh at 665.80 °C. The effect of T_i on the suggested plant's cost rate is also shown in this graph. The SOFC inlet temperature raises the plant's overall cost rate. Because rising temperatures raise the cost of materials. As a result, T_i raises the plant's overall cost rate. The total cost rate of proposed plant increases from 32.70 \$/h and 37.59 \$/h respectively within the considered range of the temperatures.

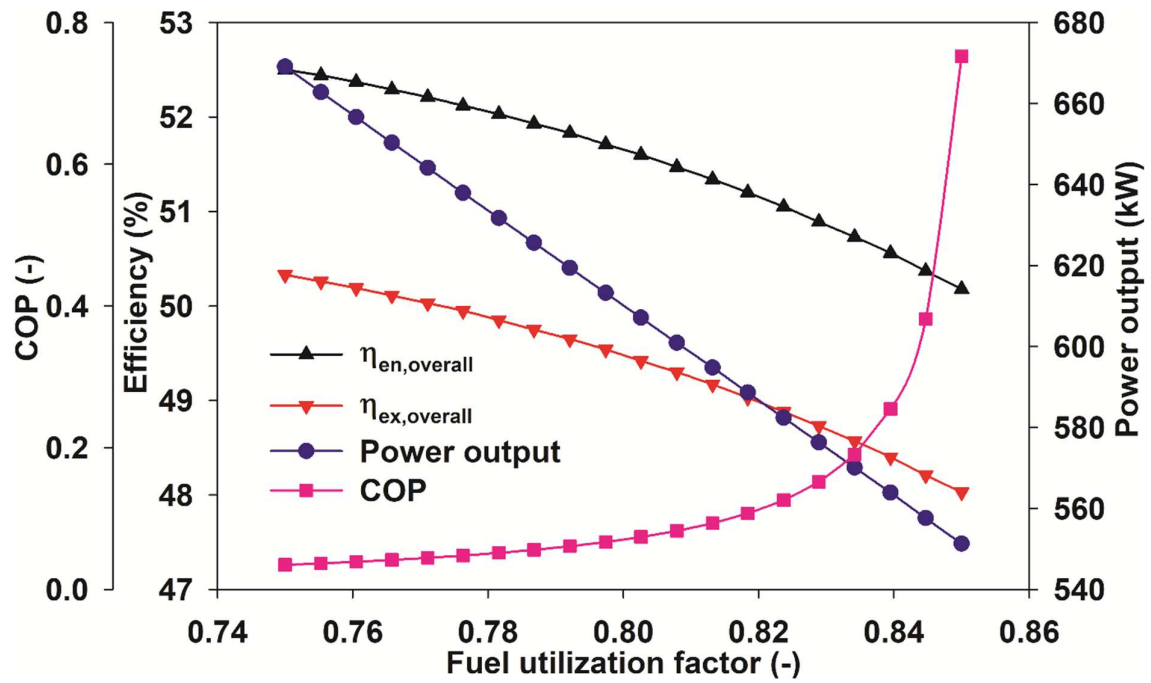


(b)

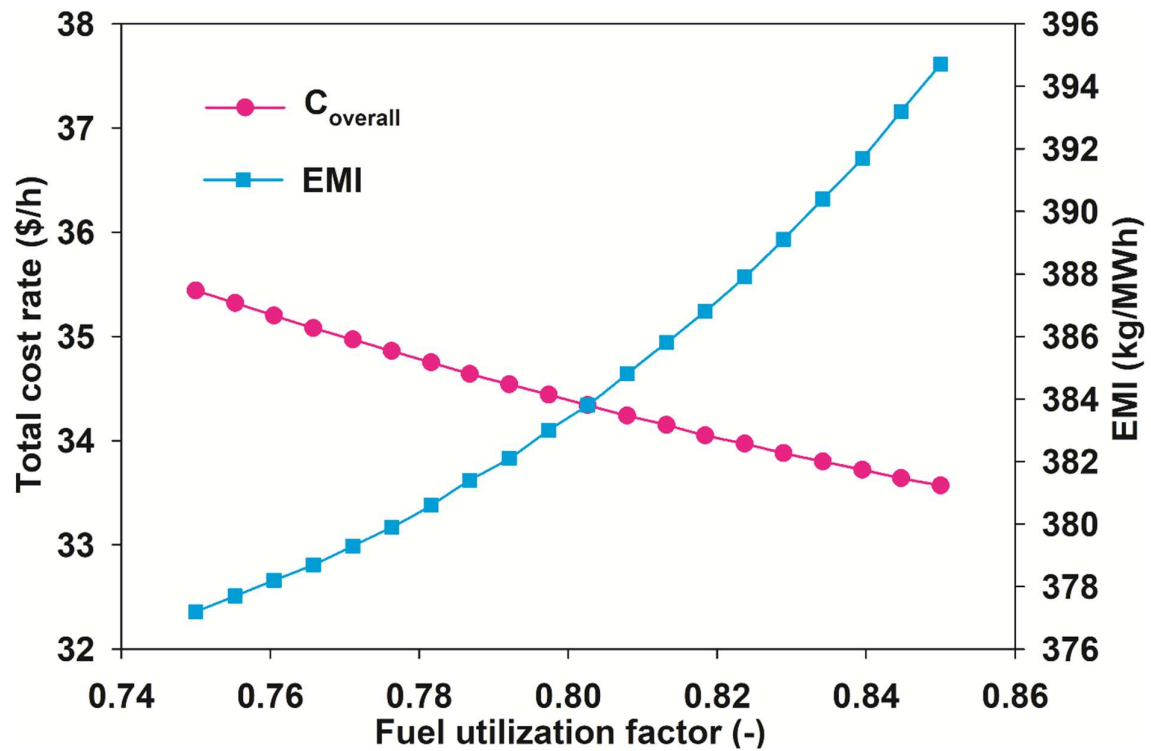
Figure 5.16: Effect of the current density on (a) efficiency, COP and power output, (b) total cost and EMI

5.2.3.3 Fuel utilization factor impact on performance

This section evaluates the effect of the U_f on the performance of the plant. Due to an increment in the gas turbine and air compressor work, the power output of the system falls as fuel consumption rises, as seen in Figure 5.17 (a). Conversely, work consumption in pump, fuel, and water pump all reduces with the U_f . The output power from the overall plant reduced by 17.60% as it went from 0.85 to 0.95. Additionally, this graph shows that as U_f increases, the energy efficiency of the SOFC-GT system falls.



(a)



(b)

Figure 5.17: Effect of the fuel utilization factor on the (a) efficiency, COP and output power, (b) total cost and EMI

This is due to the decrement of steam flow rate and molar flow rate of the fuel with the temperature, and the fuel cell voltage decreases with increasing air molar flow rate at the inlet of the SOFC unit. As U_f increases, the suggested plant's energy efficiency falls. This is because the steam and fuel flow rate decrease with increasing temperature, and the cell voltage decreases with increasing air flow rate at the inlet of the SOFC. The overall plant energy efficiency reduces from 52.50% to 50.18% with the U_f value change from 0.85 to 0.95.

Figure 5.17(a) illustrates the impact of the fuel utilization factor on the COP of the VARS. As U_f increases, the molar flow rate of the exhaust gases rises, which enhances the flow through the cooling unit. This leads to a reduction in the generator heat input. At constant cooling load, this reduction in heat input causes a corresponding improvement in the COP. Specifically, the COP increases significantly from 0.0348 to 0.754 as U_f increases from 0.85 to 0.95.

Conversely, Figure 5.17(b) shows that the Environmental Impact Index (EMI) for the system increases from 377.20 kg/MWh to 394.70 kg/MWh with rising U_f . This is attributed to the increased exhaust gas flow rate, which raises the generator's heat input. However, as discussed earlier, the system's net power output decreases with higher U_f . Since EMI is calculated based on total energy output, the reduced output leads to a rise in EMI values.

Additionally, the total cost rate of the system exhibits a decreasing trend with U_f , dropping linearly from 35.44 \$/h at $U_f = 0.85$ to 33.57 \$/h at $U_f = 0.95$. This cost reduction is mainly due to enhanced exergy utilization and improved thermal integration at higher fuel utilization levels.

CHAPTER 6 : CONCLUSIONS & FUTURE WORK

6.1 Conclusion

6.1.1 SPT-HBC-HRSG-VACS

This study introduced a novel solar-based trigeneration setup where solar energy is collected through the SPT system and converted to power using HBC. Further, HRSG and VACS subsystems recover waste heat from HBC to produce heating and cooling, respectively. A full energy and exergy analysis was carried out for this proposed setup.

- The system delivered a net output of 14,562 kW, with energy efficiency measured at **44.96%** and exergy efficiency measured at **34.15%**. Heating output from HRSG reached 8,510 kW, while cooling output from VACS was 115.10 kW, and the COP of VACS was determined to be 0.8015.
- When compared to the base SPT-HBC setup, integrating HRSG and VACS improved energy efficiency by 58.98% and exergy efficiency by 12.92%.
- An increase in HTIT led to improved exergy and energy efficiencies, net power, and heating, while VACS COP declined and cooling output remained nearly stable.
- CPR and CIT significantly affected the energy/exergy efficiencies, net output, and heating, whereas COP and cooling capacity showed only minor variation.
- A reduction in generator temperature greatly lowered COP and cooling, but had minimal influence on system efficiencies, net output, and heating capacity.
- With increasing DNI, both net output and heating production improved. However, COP, cooling, and system efficiencies showed a slight decrease.
- The system's performance benefited from higher heliostat field efficiency, which enhanced energy/exergy efficiencies, net power, heating, and cooling capacities.

Table 6.1 : Baseline values of SPT-HBC Configuration

Parameter	Baseline (SPT-HBC Only)	Proposed (Trigeneration)	Improvement
Energy efficiency	28.28%	44.96%	+58.98%
Exergy efficiency	30.24%	34.15%	+12.92%
Useful outputs	14,562 kW (power only)	14,562 kW + 8,510 kW (heating) + 115.1 kW (cooling)	Waste heat recovered

System Boundary: Includes SPT heliostat field (500 heliostats, $9.45 \times 12.84 \text{ m}^2$ each), volumetric air receiver ($A_{\text{rec}} = 68.1 \text{ m}^2$), HBC (helium working fluid, $\text{CPR} = 2.5$, $\text{HTIT} = 850^\circ\text{C}$), HRSG (steam at 2000 kPa, 200°C), and VACS (LiBr-H₂O, generator at 80°C). All improvements calculated based on same solar input (Q_{sun} from $\text{DNI} = 850 \text{ W/m}^2$).

6.1.2 SOFC-GT-VACS

The following conclusions were made from this study;

- The proposed hybrid combined cooling and power plant (SOFC-GT-VARS) obtained energy, exergy efficiency, and power output of 50.18%, 48.03%, and 551.29 kW, respectively, at given operating conditions.
- Proposed SOFC-GT-VARS plant's energy and exergy efficiency respectively, 0.96% and 0.08% higher than the conventional SOFC-GT system. It shows little bit increment however, this SOFC based combined power and cooling system is developed for small scale room air conditioning.
- A cooling capacity of 5.275 kW was achieved from the compact VARS, delivering air conditioning at an evaporator temperature of 5°C with a COP of 0.752, aided by a water storage tank.

- The afterburner was responsible for the highest exergy destruction, contributing 18.12% of the total system exergy loss.
- The total cost rate for operating the proposed system was noted as \$33.57/h, while CO₂ emissions were recorded at 394.70 kg/MWh of energy produced.
- Parametric analysis revealed that critical SOFC parameters such as fuel utilization, current density, and inlet temperature significantly influenced the integrated system's overall performance.

6.2 Future Work

6.2.1 Experimental Validation

- Construct a laboratory-scale SPT-HBC-HRSG-VACS test rig (1-5 kW cooling) using electric heaters to simulate solar input.
- Develop a benchtop SOFC-GT-VARS prototype (1.5 TR / 5.275 kW cooling) with a micro-gas turbine and water storage tank.
- Validate all simulation results against experimental data under controlled conditions.
- Perform rigorous uncertainty analysis to quantify measurement errors and confidence intervals.

6.2.2 Transient and Dynamic Modelling

- Develop transient models using TRNSYS to replace steady-state assumptions.
- Characterize start-up and shut-down dynamics, including thermal stress analysis.
- Evaluate part-load performance at 25%, 50%, and 75% of rated capacity.
- Model system response to solar intermittency (passing clouds, diurnal cycles, seasonal DNI variations).
- Simulate SOFC degradation over time due to nickel agglomeration and chromium poisoning (10,000-40,000 hours).

6.2.3 Exergoeconomic and Life Cycle Assessment

- Perform exergoeconomic analysis to calculate the cost of exergy destruction for each component.
- Compute exergoeconomic factors to identify components where additional investment yields greatest exergy savings.
- Conduct full Life Cycle Assessment (LCA) following ISO 14040/14044 standards across five stages: raw material extraction, manufacturing, transport, operation (20 years), and end-of-life disposal.
- Calculate levelized cost of electricity (LCOE) and levelized cost of cooling (LCOC).
- Perform Monte Carlo sensitivity analysis for uncertainties in fuel prices, discount rates, and component costs.

6.2.4 Component-Level Optimization of Small-Scale VARS

- Design compact plate heat exchangers with enhanced surfaces (corrugated plates, microchannels) for generator, condenser, evaporator, absorber, and SHE.
- Develop small-scale diaphragm or magnetic drive pumps specifically for LiBr-H₂O solutions.
- Create intelligent control systems for real-time crystallization prevention using sensors and automatic dilution circuits.
- Conduct accelerated corrosion tests (1000-5000 hours) for materials (copper, SS316L, titanium, polymer composites) with corrosion inhibitors.
- Design and test vacuum-sealing methods (elastomeric seals, metal gaskets, brazed joints) for long-term leak-tightness at 6-10 mmHg.

6.2.5 Integration with Alternative Renewable Energy Sources

- Design biomass-fired absorption cooling systems using agricultural waste (rice husk, coconut shells, sugarcane bagasse) for rural off-grid cold storage.

- Investigate low-temperature geothermal sources (70-100°C) for district cooling in geothermal-rich regions of India (Himalayas, Western Ghats).
- Capture industrial waste heat (80-120°C) from diesel generators, kilns, furnaces, and thermal power plant condensers for factory cooling.
- Develop hybrid solar-biomass systems for 24/7 cooling operation without grid electricity or battery storage.

6.2.6 Alternative Working Pairs and Nanofluids

- Synthesize nanofluids with Al_2O_3 , CuO , TiO_2 , SiO_2 , or graphene oxide nanoparticles (0.1-2% volume fraction) to enhance thermal conductivity by 10-30%.
- Investigate ternary and quaternary mixtures ($\text{LiBr-LiCl-LiNO}_3\text{-H}_2\text{O}$) to reduce crystallization temperature and improve COP.
- Explore low-GWP refrigerants (R-1234ze(E), R-1234yf, R-152a) as non-toxic alternatives to ammonia.

6.2.7 Comparative Analysis of Both Proposed Configurations

- Performance Comparison (Energy and Exergy Efficiencies)
- Economic Comparison (Cost Rates and Payback Periods)
- Environmental Comparison (CO_2 Emissions per MWh)
- Scale-Based Applicability (Large-Scale vs. Small-Scale)
- Working Fluid and Heat Source Comparison
- Component-Wise Exergy Destruction Comparison
- Parametric Sensitivity Comparison
- Suitability for Indian Climatic and Energy Context
- Identification of Optimal Configuration for Specific Applications

6.2.8 Identify and evaluate specific low-temperature cycles for integration with the existing SOFC-based hybrid setup

- **Organic Rankine Cycle (ORC)** with working fluids R-245fa, R-1233zd(E), R-1234ze(E), and toluene for waste heat recovery at 100-200°C.
- **Kalina Cycle (KC)** using ammonia-water mixture for waste heat recovery at 80-150°C with potentially higher efficiency than ORC.
- **Trilateral Flash Cycle (TFC)** for waste heat recovery at 60-120°C using pure working fluids.
- **Supercritical CO₂ Brayton cycle** for waste heat recovery at 150-300°C as a bottoming cycle.
- **Ejector refrigeration cycle** for additional cooling production using waste heat at 80-120°C.
- Perform comparative analysis of these low-temperature cycles based on: additional power output, additional cooling output, capital cost increment, exergy efficiency improvement, and payback period.
- Identify the optimal low-temperature cycle for each waste heat temperature range: TFC for 60-120°C, ORC for 100-200°C, KC for 120-200°C, and sCO₂ for 200-350°C.
- Develop a cascaded bottoming cycle configuration where multiple low-temperature cycles are combined to recover waste heat across a wide temperature range (60°C to 350°C).

6.2.9 Identification of suitable end-users and demand-side matching

- For the SPT-HBC-HRSG-VACS (15 MW scale), potential users include: industrial parks with simultaneous power, steam, and cooling demand (e.g., food processing, textile, pharmaceutical clusters); large hospital complexes (24/7 power, heating, and air conditioning); district cooling systems in smart cities; and data centers requiring constant cooling.

- For the SOFC-GT-VARS (550 kW power, 1.5 TR cooling), potential users include: individual commercial buildings (hotels, shopping malls, office complexes); residential complexes with combined power and air conditioning; remote off-grid facilities (defense installations, research stations, luxury resorts); and telecom towers requiring backup power and equipment cooling. A detailed techno-economic analysis should be performed to match system outputs with actual load profiles of these user categories.

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Appendix : Coding

EES coding for SPT-HBC-HRSG-VACS

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"-----###Solar Volumetric Air
Receiver Modelling###-----"
"Given Input Values for Solar Tower Design"
DNI=850 [W/m^2] "Direct Normal Irradiance"
eta_field=0.75 "Efficiency of heliostatic Filed"
eta_rec=0.9 "Receiver efficiency"
N_hel=500 "Number of heliostats"
A_hel=9.45*12.84 [m^2] "Reflective area of each heliostats"
A_rec=68.1 [m^2] "Receiver Aperature Area"
Q_sun=DNI*A_hel*N_hel*convert(W,kW) "Amount of total solar
Irradation received from solar in KW"
Q_rec_in=eta_field*Q_sun "Amount of heat
received at the receiver in KW"
eta_rec=Q_rec_net/Q_rec_in "Actual amount of heat
received at the receiver"
Q_rec_in=Q_rec_net+Q_rec_loss "Heat loss in the
receiver"
Q_rec_net=m_dot_air*(h[21]-h[19]) "Net heat at
Indermediate heat exchanger"
"Asummed Data in Indermediate heat exchanger"
T[21]=1125[C] "Hot Side flow (Air), Temperature at
inlet of intermediate heat exchanger"
P[21]=101.3[kPa] "Pressure at inlet of intermediate heat
exchanger"
P[21]=P[19] "No pressure loss in the intermediate
heat exchanger"
T[19]-T[3]=50[C] "Cold Side flow (Helium),
Temperature at outlet of indermediate heat exchanger (Pinch Point 50[C])"
"Indermediate heat exchanger with hot fluid inlet at 16 and outlet at 17 and cold fluid inlet at 3
and outlet at 4"
h[21]=enthalpy(Air,T=T[21])
s[21]=entropy(Air,T=T[21],P=P[21])
h[19]=enthalpy(Air,T=T[19])
s[19]=entropy(Air,T=T[19],P=P[19])
File:C:\Users\hp\Downloads\SPT-HBC-HRSG-VAR (1).EES 06-03-2025 14:34:51 Page 2
EES Ver. 9.478: #3026: The Sabotage Rebellion hackers, http://tsrh.freeseicals.com$plA wEFKPECA
vAFAHHMKJ LEGOAVW PAEI
m_dot_air*(h[21]-h[19])=m_dot_helium*(h[4]-h[3])
"-----###Helium Brayton
Cycle (Topping Cycle)###-----"
"###Assumed Input Data to the topping system###"
"Assumed Pressure"
P[1]=2500[kPa] "Pressure at the inlet of Helium Compressor"
P[2]/P[1]=r_p1 "Pressure ratio of helium Compressor"
r_p1=2.5
P[3]=P[2]-(0.01*P[2]) "High side flow Pressure loss in the recuperator"
P[4]=P[3]-(0.02*P[3]) "Pressure in the intermediate heat exchanger "
P[6]=P[5]-(0.01*P[5]) "Low side flow Pressure loss in the recuperator"
P[7]=P[6]-(0.01*P[6]) "High side flow Pressure loss in HRSG"
P[8]=P[7]-(0.01*P[7]) "High side flow Pressure loss in GEN"
P[1]=P[8]-(0.01*P[8]) "High side flow Pressure loss in cooler"
P[22]=101.3[kPa] "Pressure at the inlet of Helium cooler"
P[22]= P[23] "No pressure loss in the helium cooler"

```

```

"Assumed Temperature"
T[1]-T[22]=5[C] "Temperature at the inlet of Helium Compressor"
T[4]=850[C] "Temperature at the inlet of Helium Turbine"
T[22]=25[C] "Temperature at the inlet of Helium condenser"
T[23]=35[C] "Temperature at the outlet of Helium condenser"
T[8]-T[16]=5[C] "Pinch Point Temperature difference in the waste heat recov, (
Pinch Point 5[C])"
T[16]=80[C] "80[C] generator temp"
P[16]=5.63[kPa]
T[30]=35[C]
P[30]=101.3[kPa]
T[7]=T[6]-80[C]
T[31]=T[6]-30[C] "Temperature of steam at the outlet of HRSG"
P[31]=2000[kPa] "Pressure of steam at the outlet of HRSG "
"Assumed efficiencies/effectiveness/Fluids"
ETA_HC=0.88 "isentropic efficiency of helium compressor"
ETA_HT=0.9 "isentropic efficiency of helium Turbine"
Epsilon_HR=0.9 "Effectiveness of Counterflow recuperator"
"Helium Compressor with isentropic efficiency at state 1 inlet and state 2 outlet"
h[1]=enthalpy(Helium,T=T[1],P=P[1])
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vAFAHHMKJ LEGOAVW PAEI
s[1]=entropy(Helium,T=T[1],P=P[1])
s[1]=s_s[2]
T_s[2]=temperature(Helium,s=s_s[2],P=P[2])
h_s[2]=enthalpy(Helium,T=T_s[2],P=P[2])
ETA_HC=(h_s[2]-h[1])/(h[2]-h[1])
T[2]=temperature(Helium,h=h[2],P=P[2])
s[2]=entropy(Helium,h=h[2],P=P[2])
"HRSG"
h[30]=enthalpy(Water,T=T[30],P=P[30])
s[30]=entropy(Water,T=T[30],P=P[30])
h[31]=enthalpy(Steam,T=T[31],P=P[31])
s[31]=entropy(Steam,T=T[31],P=P[31])
m_dot_helium*(h[6]-h[7])=m_dot_steam*(h[31]-h[30])
"Helium Turbine with isentropic efficiency at state 4 inlet and state 5 outlet"
h[4]=enthalpy(Helium,T=T[4],P=P[4])
s[4]=entropy(Helium,T=T[4],P=P[4])
s[4]=s_s[5]
T_s[5]=temperature(Helium,s=s_s[5],P=P[5])
h_s[5]=enthalpy(Helium,T=T_s[5],P=P[5])
ETA_HT=(h[4]-h[5])/(h[4]-h_s[5])
T[5]=temperature(Helium,h=h[5],P=P[5])
s[5]=entropy(Helium,h=h[5],P=P[5])
"Helium Regenerator/recuperator COUNTER FLOW with hot fluid inlet at 5 and outlet at 6 and
cold fluid inlet at 2 and outlet at 3"
Epsilon_HR=(T[5]-T[6])/(T[5]-T[2])
h[6]=enthalpy(Helium,T=T[6],P=P[6])
s[6]=entropy(Helium,T=T[6],P=P[6])
h[3]-h[2]=h[5]-h[6]
T[3]=temperature(Helium,h=h[3],P=P[3])
s[3]=entropy(Helium,T=T[3],P=P[3])
"Epsilon_HR=NTU/(1+NTU)"
C_min=m_dot_Helium*(Cp(Helium,T=T[2],P=P[2]))

```

```

U_HR=3[kW/m2k]
"NTU=(U*A_HR)/C_min"
DELTA_T1=T[6]-T[2]
DELTA_T2=T[5]-T[3]
"LMTD_HR=(DELTA_T1-DELTA_T2)/(ln(DELTA_T1/DELTA_T2))"
"m_dot_helium*(h[5]-h[6])=U_HR*A_HR*LMTD_HR"
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"Waste heat recovery unit (WHRU) with hot fluid inlet at 6 and outlet at 7 and cold fluid inlet at
11 and outlet at 8"
h[7]=enthalpy(Helium,T=T[7],P=P[7])
s[7]=entropy(Helium,h=h[7],P=P[7])
"m_dot_helium*(h[6]-h[7])=m_dot_bottoming*(h[8]-h[11])"
" LMTD_WHRU=((T[6]-T[8])-(T[7]-T[11]))/ln((T[6]-T[8])/(T[7]-T[11]))"
U_WHRU=1.6[kW/m^2K]
"m_dot_helium*(h[6]-h[7])=U_WHRU*A_WHRU*LMTD_WHRU"
"Helium Cooler with hot fluid inlet at 8 and outlet at 1 and cold fluid inlet at 22 and outlet at 23"
h[22]=enthalpy(Water,T=T[22],P=P[22])
s[22]=entropy(Water,T=T[22],P=P[22])
h[23]=enthalpy(Water,T=T[23],P=P[23])
s[23]=entropy(Water,T=T[23],P=P[23])
m_dot_helium*(h[7]-h[1])=m_dot_water1*(h[23]-h[22])
m_dot[22]=m_dot_water1
m_dot[23]=m_dot_water1
U_precooler=1.6[kW/m^2K]
"LMTD_precooler=((T[7]-T[23])-(T[1]-T[22]))/ln((T[7]-T[23])/(T[1]-T[22]))"
"m_dot_helium*(h[7]-h[1])=U_precooler*A_precooler*LMTD_precooler"
h[8]=enthalpy(Helium,T=T[8],P=P[8])
s[8]=entropy(Helium,T=T[8],P=P[8])
"VAR"
"Condenser"
h[16]=enthalpy(Water,T=T[16],P=P[16])
s[16]=entropy(Water,T=T[16],P=P[16])
T[24]=27[C]
T[25]=32[C]
P[24]=101.3[kPa]
P[25]=P[24]
h[24]=Enthalpy(Water,T=T[24],P=P[24])
s[24]=Entropy(Water,T=T[24],P=P[24])
h[25]=Enthalpy(Water,T=T[25],P=P[25])
s[25]=Entropy(Water,T=T[25],P=P[25])
T[17]=T[24]+8[C]
P[17]=P[16]
h[17]=Enthalpy(Water,T=T[17],P=P[17])
s[17]=Entropy(Water,T=T[17],P=P[17])
m_dot[24]*(h[25]-h[24])=m_dot[16]*(h[16]-h[17]) "mass at 16 is calculated from below evaporators
equation"
m_dot[25]=m_dot[24]
"EXV"
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h[18]=h[17]
T[9]=6[C] "Given temperature in literature evaporator temp"

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T[18]=T[9] "casade condenser/evaporator for VAR"
P[18]=Pressure(Water,T=T[18],h=h[18])
s[18]=Entropy(Water,h=h[18],P=P[18])
"Evaporator"
T[27]=T[9]+3[C] "Min temp. diff. condidered as 8 C"
P[27]=101.3[kPa]
P[26]=P[27]
T[26]=T[9]+8[C]
h[26]=Enthalpy(Water,T=T[26],P=P[26])
s[26]=Entropy(Water,T=T[26],P=P[26])
h[27]=Enthalpy(Water,T=T[27],P=P[27])
s[27]=Entropy(Water,T=T[27],P=P[27])
"Q_ev=5000[kW]"
"Q_ev=m_dot_water2*(h[26]-h[27])"
m_dot_water2=5.5[kg/s]
P[9]=P[18]
h[9]=Enthalpy(Water,x=1,P=P[9])
s[9]=Entropy(Water,x=1,P=P[9])
m_dot[18]*(h[9]-h[18])=m_dot_water2*(h[26]-h[27])
m_dot[16]=m_dot[18]
m_dot[17]=m_dot[18]
m_dot[9]=m_dot[18]
m_dot[26]=m_dot_water2
m_dot[27]=m_dot[26]
"Absorber"
T[28]=T[10]-8[C]
T[29]=T[10]-3[C]
P[29]=101[kPa]
P[28]=P[29]
h[28]=Enthalpy(Water,T=T[28],P=P[28])
h[29]=Enthalpy(Water,T=T[29],P=P[29])
s[28]=Entropy(Water,T=T[28],P=P[28])
s[29]=Entropy(Water,T=T[29],P=P[29])
m_dot[28]=0.3[kg/s]
m_dot[29]=m_dot[28]
T[10]=35[C]
P[10]=P[9]
x[10]=x_LiBrH2O(T[10],P[10]) "Mass frcation of LiBr in Water mixture"
h[10]=h_LiBrH2O(T[10],x[10])
s[10]=s_LiBrH2O(T[10],x[10])
m_dot[13]=0.357[kg/s]
m_dot[14]=m_dot[13]
m_dot[15]=m_dot[13]
m_dot[10]=m_dot[9]+m_dot[15]
"Pump"
ETA_Pump=0.9 "Given"
s_s[11]=s[10]
T[11]=T[10]
P[11]=5.63[kPa]
x[11]=x[10]
s[11]=s_LiBrH2O(T[11],x[11])
m_dot[11]=m_dot[10]
m_dot[12]=m_dot[10]
W_dot_P=((P[16]-P[9])*m_dot[10])/(ETA_Pump*rho_10)
rho_10=rho_LiBrH2O(T[10],x[10])

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m_dot[10]*h[10]+W_dot_P=m_dot[11]*h[11]
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vAFAHHMKJ LEGOAVW PAEI
"SHE"
epsilon_SHE=0.7
P[11]=P[12]
epsilon_SHE=(T[13]-T[14])/(T[13]-T[11])
T[13]=T[16]
P[13]=P[12]
P[14]=P[13]
x[13]=x_LiBrH2O(T[13],P[13]) "Mass frcation of LiBr in Water mixture"
h[13]=h_LiBrH2O(T[13],x[13])
s[13]=s_LiBrH2O(T[13],x[13])
x[14]=x[13] "Mass frcation of LiBr in Water mixture"
h[14]=h_LiBrH2O(T[14],x[14])
s[14]=s_LiBrH2O(T[14],x[14])
m_dot[11]*h[11]+m_dot[13]*h[13]=m_dot[12]*h[12]+m_dot[14]*h[14]
s[12]=s_LiBrH2O(T[12],x[12])
x[12]=x[11]
h[12]=h_LiBrH2O(T[12],x[12])
"PRV"
h[15]=h[14] "throttle to lower pressure"
P[15]=P[10]
x[15]=x[14]
T[15]=T_LiBrH2O(P[15],x[15])
s[15]=s_LiBrH2O(h[15],x[15])
"###Heat transfer/Work analysis###"
"Heat transfer in Generator"
Q_dot_g=(m_dot[16]*h[16]+m_dot[13]*h[13])-(m_dot[12]*h[12])
"VARS Pump"
W_dot_Pump=m_dot[10]*(h[11]-h[10])
"Condenser"
Q_dot_c=m_dot[16]*(h[16]-h[17])
"Evaporator"
Q_dot_e=m_dot[18]*(h[9]-h[18])
"Helium compressor HC"
W_dot_HC=m_dot_helium*(h[2]-h[1])
"Helium Turbine HT"
W_dot_HT=m_dot_helium*(h[4]-h[5])
"Heat transfer in IHE"
Q_dot_IHE=m_dot_helium*(h[4]-h[3])
"Heat transfer in recuperator"
Q_dot_REC=m_dot_helium*(h[3]-h[2])
"Heat transfer in HRSG"
"Q_dot_HRSG=m_dot_helium*(h[6]-h[7])"
Q_dot_HRSG=m_dot[30]*(h[31]-h[30])
"Heat transfer in Precooler"
Q_dot_Precooler=m_dot_helium*(h[8]-h[1])
"Absorber heat intraction"
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Q_dot_a=m_dot[9]*h[9]+m_dot[15]*h[15]-m_dot[10]*h[10]
"T_a=(T[27]-T[26])/(ln(T[27]/T[26]))"

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```

"Overall net work"
W_dot_net_combined=W_dot_HT-W_dot_HC-W_dot_Pump
"##COPs calculations##"
COP_VAR=Q_dot_e/(Q_dot_g+W_dot_Pump)
"##Energy efficiency calculation"
ETA_en_Combined=((Q_dot_e+Q_dot_HRSG+W_dot_net_combined)/Q_dot_IHE)*100
ETA_en_Plant_tri=((Q_dot_e+Q_dot_HRSG+W_dot_net_combined)/Q_sun)*100
ETA_en_HBC=((W_dot_HT-W_dot_HC)/Q_sun)*100
ETA_en_Plant=((W_dot_net_combined)/Q_sun)*100
ETA_en_co_h=((Q_dot_HRSG+W_dot_net_combined)/Q_sun)*100
ETA_en_co_c=((Q_dot_e+W_dot_net_combined)/Q_sun)*100
"-----###
Exergy Analysis###-----"
"####Dead State for Solar Tower####"
T_0=converttemp(C,K,25) "Reference Temperature/
ambient condition"
T_sun=4500[K] "Temperature of the sun"
Exergy_sun=Q_sun*(1-(T_0/T_sun)) "Exergy from the sun "
Exergy_Heliostats=Q_rec_in*(1-(T_0/T_sun)) "Exergy from the Heliostatic
Field"
Exergy_input_cycle=EX[21]-EX[19] "Exergy Input to the cycle"
"####Dead state for Air####"
T[33]=25[C]
P[33]=101.3[kPa]
h[33]=enthalpy(Air,T=T[33])
s[33]=entropy(Air,T=T[33],P=P[33])
EX[21]=m_dot_air*((h[21]-h[33])-(T[33]+273.15[C])*(s[21]-s[33])) "Exergy at
state 30"
EX[19]=m_dot_air*((h[19]-h[33])-(T[33]+273.15[C])*(s[19]-s[33])) "Exergy at
state 31"
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"####Dead state for Helium Fluid####"
T[34]=25[C]
P[34]=101.3[kPa]
h[34]=enthalpy(Helium,T=T[34],P=P[34])
s[34]=entropy(Helium,T=T[34],P=P[34])
EX[1]=m_dot_helium*((h[1]-h[34])-(T[34]+273.15[C])*(s[1]-s[34])) "Exergy at
state 1"
EX[2]=m_dot_helium*((h[2]-h[34])-(T[34]+273.15[C])*(s[2]-s[34])) "Exergy at
state 2"
EX[3]=m_dot_helium*((h[3]-h[34])-(T[34]+273.15[C])*(s[3]-s[34])) "Exergy at
state 3"
EX[4]=m_dot_helium*((h[4]-h[34])-(T[34]+273.15[C])*(s[4]-s[34])) "Exergy at
state 4"
EX[5]=m_dot_helium*((h[5]-h[34])-(T[34]+273.15[C])*(s[5]-s[34])) "Exergy at
state 5"
EX[6]=m_dot_helium*((h[6]-h[34])-(T[34]+273.15[C])*(s[6]-s[34])) "Exergy at
state 6"
EX[7]=m_dot_helium*((h[7]-h[34])-(T[34]+273.15[C])*(s[7]-s[34])) "Exergy at
state 7"
EX[8]=m_dot_helium*((h[8]-h[34])-(T[34]+273.15[C])*(s[8]-s[34])) "Exergy at
state 7"
"####Dead state for VAR cycle####"

```

```

T[36]=25[C]
P[36]=101.3[kPa]
h[36]=enthalpy(Water,T=T[36],P=P[36])
s[36]=entropy(Water,T=T[36],P=P[36])
EX[9]=m_dot[9]*((h[9]-h[36])-(T[36]+273.15[C])*(s[9]-s[36])) "Exergy
at state 8"
EX[16]=m_dot[16]*((h[16]-h[36])-(T[36]+273.15[C])*(s[16]-s[36])) "
Exergy at state 9"
EX[17]=m_dot[17]*((h[17]-h[36])-(T[36]+273.15[C])*(s[17]-s[36])) "
Exergy at state 10"
EX[18]=m_dot[18]*((h[18]-h[36])-(T[36]+273.15[C])*(s[18]-s[36])) "
Exergy at state 11"
EX[22]=m_dot[22]*((h[22]-h[36])-(T[36]+273.15[C])*(s[22]-s[36])) "
Exergy at state 10"
EX[23]=m_dot[23]*((h[23]-h[36])-(T[36]+273.15[C])*(s[23]-s[36])) "
Exergy at state 11"
"Dead state for pont13-18"
x_010=0.3669
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vAFAHHMKJ LEGOAVW PAEI
h_010=h_LiBrH2O(T[36],x_010)
s_010=s_LiBrH2O(T[36],x_010)
EX[10]=m_dot[10]*((h[10]-h_010)-(T[36]+273.15[C])*(s[10]-s_010)) "Exergy at
state 13"
EX[11]=m_dot[11]*((h[11]-h_010)-(T[36]+273.15[C])*(s[11]-s_010)) "Exergy at
state 14"
EX[12]=m_dot[12]*((h[12]-h_010)-(T[36]+273.15[C])*(s[12]-s_010)) "Exergy at
state 15"
EX[13]=m_dot[13]*((h[13]-h_010)-(T[36]+273.15[C])*(s[13]-s_010)) "Exergy at
state 16"
EX[14]=m_dot[14]*((h[14]-h_010)-(T[36]+273.15[C])*(s[14]-s_010)) "Exergy at
state 15"
EX[15]=m_dot[15]*((h[15]-h_010)-(T[36]+273.15[C])*(s[15]-s_010)) "Exergy at
state 16"
" Dead state for HRSG"
m_dot[30]=m_dot_steam
m_dot[31]=m_dot[30]
EX[30]=m_dot[30]*((h[30]-h[36])-(T[36]+273.15[C])*(s[30]-s[36])) "
Exergy at state 10"
EX[31]=m_dot[31]*((h[31]-h[36])-(T[36]+273.15[C])*(s[31]-s[36])) "
Exergy at state 11"
" Exergy of chilled and cooling water fo evap and abs, cond"
EX[26]=m_dot[26]*((h[26]-h[36])-(T[36]+273.15[C])*(s[26]-s[36])) "
Exergy at state 10"
EX[27]=m_dot[27]*((h[27]-h[36])-(T[36]+273.15[C])*(s[27]-s[36])) "
Exergy at state 11"
EX[28]=m_dot[28]*((h[28]-h[36])-(T[36]+273.15[C])*(s[28]-s[36])) "
Exergy at state 10"
EX[29]=m_dot[29]*((h[29]-h[36])-(T[36]+273.15[C])*(s[29]-s[36])) "
Exergy at state 11"
EX[24]=m_dot[24]*((h[24]-h[36])-(T[36]+273.15[C])*(s[24]-s[36])) "
Exergy at state 24"
EX[25]=m_dot[25]*((h[25]-h[36])-(T[36]+273.15[C])*(s[25]-s[36])) "
Exergy at state 25"

```

"Exergy calculation of usefull components"

"evaporator"

$$EX_e = \text{abs}(Q_dot_e * (1 - T_0 / (T[9] + 273.15)))$$

"Generator"

$$EX_g = \text{abs}(Q_dot_g * (1 - T_0 / (T[16] + 273.15)))$$

$$T_g = (T[7] - T[8]) / (\ln(T[7] / T[8]))$$

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$$EX_g = Q_dot_g * (1 - T_0 / (T_g + 273.15))$$

"Condnsr"

$$T_c = (T[16] - T[17]) / (\ln(T[16] / T[17]))$$

$$EX_c = Q_dot_c * (1 - T_0 / (T_c + 273.15))$$

"Absorber"

$$EX_a = Q_dot_a * (1 - T_0 / (T_a + 273.15))$$

$$T_a = (T[29] - T[28]) / (\ln(T[29] / T[28]))$$

"-----Net Exergy Destruction
 in the components-----"

"###Solar Field###"

$$EXD_Heliosfield = Exergy_sun - Exergy_Heliostats$$

"Exergy destruction in the
 heliostatic field"

$$EXD_receiver = Exergy_Heliostats - Exergy_input_cycle$$

"Exergy destruction in the
 receiver"

$$EXD_cycle = Exergy_input_cycle - W_dot_net_combined$$

"Exergy destruction in the cycle"

"###Topping Cycle###"

"Indermediate heat exchanger"

$$EXD_IHX = (EX[21] - EX[19]) - (EX[4] - EX[3])$$

"Exergy Destruction Rate in
 Indermediate heat exchanger"

$$ETA_IHX_Exergy = (EX[4] - EX[3]) / (EX[21] - EX[19])$$

"Exergy Efficiency in Indermediate
 heat exchanger"

"Helium Compressor"

$$EXD_HC = (EX[1] - EX[2]) + W_dot_HC$$

"Exergy Destruction Rate in Helium
 Compressor"

$$ETA_HC_Exergy = (EX[2] - EX[1]) / W_dot_HC$$

"Exergy Effeiciency in Helium
 Compressor"

"Helium Turbine"

$$EXD_HT = (EX[4] - EX[5]) - W_dot_HT$$

"Exergy Destruction Rate in Helium
 Turbine"

$$ETA_HT_Exergy = W_dot_HT / (EX[4] - EX[5])$$

"Exergy Effeiciency in Helium
 Turbine"

"Helium Recuperator"

$$EXD_HR = (EX[5] - EX[6]) - (EX[3] - EX[2])$$

"Exergy Destruction Rate in Helium
 Recuperater"

$$ETA_HR_Exergy = (EX[3] - EX[2]) / (EX[5] - EX[6])$$

"Exergy Effeiciency in Helium
 Recuperater"

"HRSG"

$$EXD_HRSG = (EX[6] - EX[7]) - (EX[31] - EX[30])$$

"Exergy Destruction Rate in
 Helium Recuperater"

$$ETA_HRSG_Exergy = (EX[31] - EX[30]) / (EX[6] - EX[7])$$

"Exergy Effeiciency in Helium
 Recuperater"

"Generator"

$$EXD_g = -EX[16] - EX[13] + EX[12] + EX_g$$

File: C:\Users\hp\Downloads\SPT-HBC-HRSG-VAR (1).EES 06-03-2025 14:34:51 Page 11
 EES Ver. 9.478: #3026: The Sabotage Rebellion hackers, <http://tsrh.freeseerials.com>\$pLA wEFKPECA
 vAFAHHMKJ LEGOAVW PAEI

$ETA_g_Exergy = (EX[16] + EX[13]) / (EX[12] + EX_g)$
 "Helium preCooler"
 $EXD_precooler = (EX[8] - EX[1]) - (EX[23] - EX[22])$ "Exergy Destruction Rate in Helium Cooler"
 $ETA_precooler_Exergy = (EX[23] - EX[22]) / (EX[8] - EX[1])$ "Exergy Efficiency in Helium Cooler"
 "Condenser "
 $EXD_c = (EX[16] - EX[17]) - (EX[25] - EX[24])$ "Exergy Destruction Rate in Condenser"
 $ETA_c_Exergy = (EX[25] - EX[24]) / (EX[16] - EX[17])$ "Exergy Efficiency in Condenser"
 "expansion vale"
 $EXD_EXV = EX[17] - EX[18]$
 "Evaporator"
 $EXD_e = EX[18] + EX_e - EX[9]$
 $ETA_e_Exergy = (EX[9]) / (EX[18] + EX_e)$
 "Absorber"
 $EXD_a = -EX[9] - EX[15] + EX[10] + EX_a$
 $ETA_a_Exergy = (EX[9] + EX[15]) / (EX[10] + EX_a)$
 "pump"
 $EXD_Pump = EX[11] - W_dot_Pump - EX[10]$
 "SHE"
 $EXD_SHE = EX[14] + EX[12] - EX[11] - EX[15]$
 $ETA_SHE_Exergy = (EX[12] - EX[11]) / (EX[13] - EX[14])$
 "PRV"
 $EXD_PRV = EX[14] - EX[15]$
 $ETA_PRV_Exergy = (EX[15]) / EX[14]$
 "##EXD calculation for subsystems"
 $EXD_HBC = EXD_IHx + EXD_HC + EXD_HT + EXD_HR + EXD_precooler$ "Total Exergy Destruction in HBC"
 $EXD_Combined = EXD_HBC + EXD_g + EXD_Pump + EXD_c + EXD_e + EXD_a + EXD_EXV + EXD_PRV + EXD_SHE + abs(EXD_HRSG)$ "Total EXD in combined"
 $EXD_Total = EXD_Heliosfield + EXD_receiver + EXD_Combined$ "By addition of component wise EXD"
 $ETA_ex_HBC = ((W_dot_HT - W_dot_HC) / Exergy_sun) * 100$
 $ETA_en_Combined = ((Q_dot_e + Q_dot_HRSG + W_dot_net_combined) / Q_dot_IHE) * 100$
 $ETA_ex_Plant_tri = ((EX_e + (EX[31] - EX[30]) + W_dot_net_combined) / Exergy_sun) * 100$
 $ETA_ex_Plant = ((W_dot_net_combined) / Exergy_sun) * 100$
 $ETA_ex_co_h = (((EX[31] - EX[30]) + W_dot_net_combined) / Exergy_sun) * 100$
 $ETA_ex_co_c = ((EX_e + W_dot_net_combined) / Exergy_sun) * 100$
 $ETA_ex_Plant2_tri = (1 - (EXD_Total / Exergy_sun)) * 100$
 $ETA_ex_HBC2 = (1 - (EXD_HBC / Exergy_sun)) * 100$

```

EES Coding for SOFC-GT-VACS
***** data *****
T_i=1000-273.15[C] "stack inlet temperature 1000K"
T_e-T_i=100
j=8000
F=96485
R_bar=8.314
r_sc=2.5
U_f=0.85
N_FC=11000
eta_inverter=0.97
{U_a=0.15}
A_a=0.01
P_ref=101.3[kPa]
eta_AB=0.99
Eta_ac=0.85
Eta_fc=0.85
Eta_wp=0.85
*****
T[0]=25
ex_ch_CO=275100 "kJ/kmol"
ex_ch_CO2=19870 "kJ/kmol"
ex_ch_O2=3970 "kJ/kmol"
ex_ch_H2O=9500 "kJ/kmol"
ex_ch_H2=236100 "kJ/kmol"
ex_ch_CH4=831650 "kJ/kmol"
ex_ch_N2=720 "kJ/kmol"
*****pressure analysis*****
P_r1=8
P_r2=8
P_r3=8
P[1]=101.3
P[5]=101.3
P[8]=101.3
P_r1=P[2]/P[1]
P_r2=P[6]/P[5]
P_r3=P[9]/P[8]
r_d_stack=0.02
r_d_HX=0.02
r_d_AB=0.03
r_d_HR=0.03
P[3]=P[2]*(1-r_d_HX)
P[7]=P[6]*(1-r_d_HX)
P[10]=P[9]*(1-r_d_HX)
P[4]=P[3]*(1-r_d_stack)
P[12]=P[11]*(1-r_d_stack)
P[11]=P[7]
P[13]=P[4]*(1-r_d_AB)
P[14]=P[13]*(1-r_d_HX)
P[15]=P[14]*(1-r_d_HX)
P[16]=P[15]*(1-r_d_HX)
"P[17]=P[16]"
"P[18]=P[17]*(1-r_d_HX)"
"P[44]=P[18]*(1-r_d_HR)"
*****V_ohm*****

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V_ohm=(R_contact+RHO_a*L_a+RHO_c*L_c+RHO_e*L_e+RHO_int*L_int)*j
R_contact=0
L_a=0.0005
L_c=0.00005
File:C:\Users\hp\Downloads\SOFC_GT_Final_Anil_VAR_WT.EES 06-03-2025 14:33:48 Page 3
EES Ver. 9.478: #3026: The Sabotage Rebellion hackers, http://tsrh.freeseicals.com$PLA wEFKPECA
vAFAHHMKJ LEGOAVW PAEI
L_e=0.00001
L_int=0.003
RHO_a=((C1_a/(T_e+273.15))*exp(C2_a/(T_e+273.15)))^(-1)
RHO_c=((C1_c/(T_e+273.15))*exp(C2_c/(T_e+273.15)))^(-1)
RHO_e=(C1_e*exp(C2_e/(T_e+273.15)))^(-1)
RHO_int=((C1_int/(T_e+273.15))*exp(C2_int/(T_e+273.15)))^(-1)
C1_a=95000000
C2_a=-1150
C1_c=42000000
C2_c=-1200
C1_e=33400
C2_e=-10300
C1_int=9300000
C2_int=-1100
"*****V_act*****"
V_act=V_act_a+V_act_c
V_act_a=(R_bar*(T_e+273.15)/F)*arcsinh(j/(2*j_oa))
V_act_c=(R_bar*(T_e+273.15)/F)*arcsinh(j/(2*j_oc))
j_oa=0.65*10^4
j_oc=0.25*10^4
"*****V_conc*****"
V_conc=V_conc_a+V_conc_c
V_conc_a=(R_bar*(T_e+273.15)/(2*F))*(-ln(1-(j/j_as))+ln(1+((P_H2[12]*j)/(P_H2O[12]*j_as))))
"12 is right"
V_conc_c=-((R_bar*(T_e+273.15)/(4*F))*ln(1-(j/j_cs)))
j_as=(2*F*P_H2[12]*D_aeff*1000)/(R_bar*(T_e+273.15)*L_a)
j_cs=(4*F*P_O2[4]*D_ceff*1000)/(((P[4]-P_O2[4])/P[4])*R_bar*(T_e+273.15)*L_c) "4 point is ok
w r t anil"
D_ceff=0.05*10^(-4)
D_aeff=0.2*10^(-4)
P_H2[12]=y_H2[12]*P[12]
P_H2O[12]=y_H2O[12]*P[12]
y_H2[12]=n_dot_H2[12]/n_dot[12]
y_H2O[12]=n_dot_H2O[12]/n_dot[12]
y_O2[4]=n_dot_O2[4]/n_dot[4]
P_O2[4]=y_O2[4]*P[4]
"*****Fuel cell stack*****"
r_sc=n_dot_H2O[11]/n_dot_CH4[11] "correct"
n_dot_CH4[11]=x
n_dot_H2[12]=3*x+y-z
n_dot_CO[12]=x-y
n_dot_CO2[12]=y
n_dot_H2O[12]=r_sc*x-x-y+z
n_dot_O2[4]=n_dot_O2[3]-(z/2)
n_dot_N2[3]=(79/21)*(n_dot_O2[3])
n_dot_N2[4]=n_dot_N2[3]
n_dot[3]=n_dot_O2[3]+n_dot_N2[3]
n_dot[4]=n_dot_O2[4]+n_dot_N2[4]

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U_f=z/(3*x+y)
j=(2*F*z)/(N_FC*A_a)
U_a=z/(2*n_dot_O2[3])
I=j*A_a
K_s=(y*(3*x+y-z))/((x-y)*(r_sc*x-x-y+z))
ln(K_s)=-DELTA G[0_s]/(R_bar*(T_e+273.15))
DELTA G[0_s]=g[0_CO2]+g[0_H2]-g[0_H2O]-g[0_CO]
g[0_CO2]=h_bar_CO2[12]-(T_e+273.15)*s[0_CO2[12]
h_bar_CO2[12]=molar mass(CO2)*enthalpy(CO2,T=T_e)
s[0_CO2[12]=molar mass(CO2)*entropy(CO2,T=T_e,P=P_ref)
File:C:\Users\hp\Downloads\SOFC_GT_Final Anil VAR_WT.EES 06-03-2025 14:33:48 Page 4
EES Ver. 9.478: #3026: The Sabotage Rebellion hackers, http://tsrh.freeseicals.com$PLA wEFKPECA
vAFAHHMKJ LEGOAVW PAEI
g[0_H2]=h_bar_H2[12]-(T_e+273.15)*s[0_H2[12]
h_bar_H2[12]=molar mass(H2)*enthalpy(H2,T=T_e)
s[0_H2[12]=molar mass(H2)*entropy(H2,T=T_e,P=P_ref)
g[0_H2O]=h_bar_H2O[12]-(T_e+273.15)*s[0_H2O[12]
h_bar_H2O[12]=molar mass(H2O)*enthalpy(H2O,T=T_e)
s[0_H2O[12]=molar mass(H2O)*entropy(H2O,T=T_e,P=P_ref)
g[0_CO]=h_bar_CO[12]-(T_e+273.15)*s[0_CO[12]
h_bar_CO[12]=molar mass(CO)*enthalpy(CO,T=T_e)
s[0_CO[12]=molar mass(CO)*entropy(CO,T=T_e,P=P_ref)
n_dot[12]=n_dot_H2[12]+n_dot_CO[12]+n_dot_CO2[12]+n_dot_H2O[12]
n_dot[11]=n_dot_CH4[11]+n_dot_H2O[11]
P_H2O[11]=(n_dot_H2O[11]/n_dot[11])*P[11] "11 point is ok"
T[11]=T_i
T[12]=T_e
T[3]=T_i
T[4]=T_e
*****V_c* cell voltage*****
V_loss=V_ohm+V_act+V_conc
V_N=-(DELTA G[0_e]/(2*F))-((R_bar*(T_e+273.15))/(2*F))*ln(a_H2O/(a_H2*(a_O2^0.5)))
DELTA G[0_e]=g[0_H2O]-g[0_H2]-(0.5*g[0_O2]
g[0_O2]=h_bar_O2[4]-(T_e+273.15)*s[0_O2[4]
h_bar_O2[4]=molar mass(O2)*enthalpy(O2,T=T_e)
s[0_O2[4]=molar mass(O2)*entropy(O2,T=T_e,P=P_ref)
a_H2O=(y_H2O[12]*P[12])/P_ref
a_H2=(y_H2[12]*P[12])/P_ref
a_O2=(y_O2[4]*P[12])/P_ref
V_c=V_N-V_loss "cell voltage"
W_dot_FC_stack=N_FC*I*V_c
W_dot_FC_stack_ac=W_dot_FC_stack*eta_inverter
*****energy balance of stack*****
W_dot_FC_stack=H_dot_inlet-H_dot_exit
H_dot_inlet=H_dot[3]+H_dot[11]
H_dot_exit=H_dot[4]+H_dot[12]
***** state 3 *****
h_bar[3]=molar mass(Air)*enthalpy(Air,T=T_i) " kJ/mole"
H_dot[3]=n_dot_N2[3]*h_bar_N2[3]+n_dot_O2[3]*h_bar_O2[3] "kJ/s kW"
"H_dot[3]=n_dot[3]*h_bar[3]"
h_bar_N2[3]=molar mass(N2)*enthalpy(N2,T=T_i)
h_bar_O2[3]=molar mass(O2)*enthalpy(O2,T=T_i)
*****state 11 *****
H_dot[11]=n_dot_H2O[11]*h_bar_H2O[11]+n_dot_CH4[11]*h_bar_CH4[11]
h_bar_H2O[11]=molar mass(H2O)*enthalpy(H2O,T=T_i)

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h_bar_CH4[11]=molar mass(CH4)*enthalpy(CH4,T=T_i)
"*****state 12*****"
H_dot[12]=(n_dot_H2[12]*h_bar_H2[12])+(n_dot_CO[12]*h_bar_CO[12])+(n_dot_H2O[12]*h_bar_H2O[12])+(n_dot_CO2[12]*h_bar_CO2[12])
"*****state 4*****"
H_dot[4]=(n_dot_N2[4]*h_bar_N2[4])+(n_dot_O2[4]*h_bar_O2[4])
h_bar_N2[4]=molar mass(N2)*enthalpy(N2,T=T_e)
y_N2[4]=n_dot_N2[4]/n_dot[4]
MM_4=(y_N2[4]*molar mass(N2))+(y_O2[4]*molar mass(O2)) "Molar mass (unit in gram) of mixture at point 4"
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EES Ver. 9.478: #3026: The Sabotage Rebellion hackers, http://tsrh.freeseerials.com$PLA wEFKPECA vAFAHHMKJ LEGOAVW PAEI
MM_12=(y_CO2[12]*molar mass(CO2))+(y_CO[12]*molar mass(CO))+(y_H2[12]*molar mass(H2))+(y_H2O[12]*molar mass(H2O))
y_CO2[12]=n_dot_CO2[12]/n_dot[12]
y_CO[12]=n_dot_CO[12]/n_dot[12]
y_N2[3]=n_dot_N2[3]/n_dot[3]
y_O2[3]=n_dot_O2[3]/n_dot[3]
y_CH4[11]=n_dot_CH4[11]/n_dot[11]
y_H2O[11]=n_dot_H2O[11]/n_dot[11]
m_dot[4]=MM_4*n_dot[4]/1000 "mass flow rate kg/s"
m_dot[12]=MM_12*n_dot[12]/1000 "mass flow rate in kg/s"
m_dot[13]=m_dot[4]+m_dot[12]
"m_dot_1*1000=(n_dot_CO2[13]*molar mass(CO2))+(n_dot_H2O[13]*molar mass(H2O))+(n_dot_O2[13]*molar mass(O2))+(n_dot_N2[13]*molar mass(N2))" "alternative way to calculate m_dot[13]"
"*****After burner*****"
n_dot_N2[4]=n_dot_N2[13]
2*n_dot_O2[4]+n_dot_H2O[12]+n_dot_CO[12]+2*n_dot_CO2[12]=2*n_dot_O2[13]+2*n_dot_CO2[13]+n_dot_H2O[13]
n_dot_CO[12]+n_dot_CO2[12]=n_dot_CO2[13]
n_dot_H2O[12]+n_dot_H2[12]=n_dot_H2O[13]
n_dot[13]=n_dot_H2O[13]+n_dot_CO2[13]+n_dot_O2[13]+n_dot_N2[13]
y_N2[13]=n_dot_N2[13]/n_dot[13]
y_O2[13]=n_dot_O2[13]/n_dot[13]
y_H2O[13]=n_dot_H2O[13]/n_dot[13]
y_CO2[13]=n_dot_CO2[13]/n_dot[13]
H_dot[13]=n_dot_O2[13]*h_bar_O2[13]+n_dot_N2[13]*h_bar_N2[13]+n_dot_H2O[13]*h_bar_H2O[13]+n_dot_CO2[13]*h_bar_CO2[13] "Watt"
h_bar_O2[13]=molar mass(O2)*enthalpy(O2,T=T[13])
h_bar_N2[13]=molar mass(N2)*enthalpy(N2,T=T[13])
h_bar_H2O[13]=molar mass(H2O)*enthalpy(H2O,T=T[13])
h_bar_CO2[13]=molar mass(CO2)*enthalpy(CO2,T=T[13])
Q_loss=H_dot[13]-(H_dot[4]+H_dot[12])
eta_AB=H_dot[13]/(H_dot[4]+H_dot[12])
"EX3"
EX_dot[3]=EX_dot_ph[3]+EX_dot_ch[3]
h_bar_N2_0[3]=molar mass(N2)*enthalpy(N2,T=T[0])
h_bar_O2_0[3]=molar mass(O2)*enthalpy(O2,T=T[0])
s_bar_N2[3]=molar mass(N2)*entropy(N2,T=T_i,P=(P[3]*y_N2[3]))
s_bar_O2[3]=molar mass(O2)*entropy(O2,T=T_i,P=(P[3]*y_O2[3]))

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s_bar_N2_0[3]=molar mass(N2)*entropy(N2,T=T[0],P=(P_ref*y_N2[3]))
s_bar_O2_0[3]=molar mass(O2)*entropy(O2,T=T[0],P=(P_ref*y_O2[3]))
EX_dot_ph[3]=(n_dot_N2[3]*((h_bar_N2[3]-h_bar_N2_0[3])-(T[0]+273.15)*(s_bar_N2[3]-
s_bar_N2_0[3]))))+(n_dot_O2[3]*((
h_bar_O2[3]-h_bar_O2_0[3])-(T[0]+273.15)*(s_bar_O2[3]-s_bar_O2_0[3])))
EX_dot_ch[3]=(n_dot_O2[3]*(ex_ch_O2+R_bar*(T[0]+273.15)*ln(y_O2[3])))+(n_dot_N2[3]*(ex_c
h_N2+R_bar*(T[0]+273.15)*
ln(y_N2[3])))
"EX4"
EX_dot[4]=EX_dot_ph[4]+EX_dot_ch[4]
h_bar_N2_0[4]=molar mass(N2)*enthalpy(N2,T=T[0])
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EES Ver. 9.478: #3026: The Sabotage Rebellion hackers, http://tsrh.fre serials.com$ pLA wEFKPECA
vAFAHHMKJ LEGOAVW PAEI
h_bar_O2_0[4]=molar mass(O2)*enthalpy(O2,T=T[0])
s_bar_N2[4]=molar mass(N2)*entropy(N2,T=T_e,P=(P[4]*y_N2[4]))
s_bar_O2[4]=molar mass(O2)*entropy(O2,T=T_e,P=(P[4]*y_O2[4]))
s_bar_N2_0[4]=molar mass(N2)*entropy(N2,T=T[0],P=(P_ref*y_N2[4]))
s_bar_O2_0[4]=molar mass(O2)*entropy(O2,T=T[0],P=(P_ref*y_O2[4]))
EX_dot_ph[4]=(n_dot_N2[4]*((h_bar_N2[4]-h_bar_N2_0[4])-(T[0]+273.15)*(s_bar_N2[4]-
s_bar_N2_0[4]))))+(n_dot_O2[4]*((
h_bar_O2[4]-h_bar_O2_0[4])-(T[0]+273.15)*(s_bar_O2[4]-s_bar_O2_0[4])))
EX_dot_ch[4]=(n_dot_O2[4]*(ex_ch_O2+R_bar*(T[0]+273.15)*ln(y_O2[4])))+(n_dot_N2[4]*(ex_c
h_N2+R_bar*(T[0]+273.15)*
ln(y_N2[4])))
"EX11"
EX_dot[11]=EX_dot_ph[11]+EX_dot_ch[11]
h_bar_CH4_0[11]=molar mass(CH4)*enthalpy(CH4,T=T[0])
h_bar_H2O_0[11]=molar mass(H2O)*enthalpy(H2O,T=T[0])
s_bar_CH4[11]=molar mass(CH4)*entropy(CH4,T=T_i,P=(P[11]*y_CH4[11]))
s_bar_H2O[11]=molar mass(H2O)*entropy(H2O,T=T_i,P=(P[11]*y_H2O[11]))
s_bar_CH4_0[11]=molar mass(CH4)*entropy(CH4,T=T[0],P=(P_ref*y_CH4[11]))
s_bar_H2O_0[11]=molar mass(H2O)*entropy(H2O,T=T[0],P=(P_ref*y_H2O[11]))
EX_dot_ph[11]=(n_dot_CH4[11]*((h_bar_CH4[11]-h_bar_CH4_0[11])-(
(T[0]+273.15)*(s_bar_CH4[11]-s_bar_CH4_0[11])))+(
n_dot_H2O[11]*((h_bar_H2O[11]-h_bar_H2O_0[11])-(T[0]+273.15)*(s_bar_H2O[11]-
s_bar_H2O_0[11]))))
EX_dot_ch[11]=(n_dot_CH4[11]*(ex_ch_CH4+(R_bar*(T[0]+273.15)*ln(y_CH4[11])))+(n_dot_H
2O[11]*(ex_ch_H2O+(R_bar*(
T[0]+273.15)*ln(y_H2O[11]))))
EX_input=(n_dot_CH4[11]*(ex_ch_CH4+(R_bar*(T[0]+273.15)*ln(y_CH4[11]))))/1000    "kW    "
"chemical exergy of CH4 only
at point 11 works as exergy input to the fuel cell"
"EX12"
EX_dot[12]=EX_dot_ph[12]+EX_dot_ch[12]
h_bar_H2_0[12]=molar mass(H2)*enthalpy(H2,T=T[0])
h_bar_CO_0[12]=molar mass(CO)*enthalpy(CO,T=T[0])
h_bar_CO2_0[12]=molar mass(CO2)*enthalpy(CO2,T=T[0])
h_bar_H2O_0[12]=molar mass(H2O)*enthalpy(H2O,T=T[0])
s_bar_H2[12]=molar mass(H2)*entropy(H2,T=T_e,P=(P[12]*y_H2[12]))
s_bar_CO[12]=molar mass(CO)*entropy(CO,T=T_e,P=(P[12]*y_CO[12]))
s_bar_CO2[12]=molar mass(CO2)*entropy(CO2,T=T_e,P=(P[12]*y_CO2[12]))
s_bar_H2O[12]=molar mass(H2O)*entropy(H2O,T=T_e,P=(P[12]*y_H2O[12]))
s_bar_H2_0[12]=molar mass(H2)*entropy(H2,T=T[0],P=(P_ref*y_H2[12]))
s_bar_CO_0[12]=molar mass(CO)*entropy(CO,T=T[0],P=(P_ref*y_CO[12]))

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s_bar_CO2_0[12]=molarmass(CO2)*entropy(CO2,T=T[0],P=(P_ref*y_CO2[12]))
s_bar_H2O_0[12]=molarmass(H2O)*entropy(H2O,T=T[0],P=(P_ref*y_H2O[12]))
EX_dot_ph[12]=(n_dot_H2[12]*((h_bar_H2[12]-h_bar_H2_0[12])-(T[0]+273.15)*(s_bar_H2[12]-
s_bar_H2_0[12]))))+(
n_dot_CO[12]*((h_bar_CO[12]-h_bar_CO_0[12])-(T[0]+273.15)*(s_bar_CO[12]-
s_bar_CO_0[12]))))+(n_dot_CO2[12]*((
h_bar_CO2[12]-h_bar_CO2_0[12])-(T[0]+273.15)*(s_bar_CO2[12]-
s_bar_CO2_0[12]))))+(n_dot_H2O[12]*((h_bar_H2O[12]-
h_bar_H2O_0[12])-(T[0]+273.15)*(s_bar_H2O[12]-s_bar_H2O_0[12])))
EX_dot_ch[12]=(n_dot_CO[12]*(ex_ch_CO+(R_bar*(T[0]+273.15)*ln(y_CO[12])))+(n_dot_H2[12]
)*(ex_ch_H2+(R_bar*(T[0]+
273.15)*ln(y_H2[12])))+(n_dot_H2O[12]*(ex_ch_H2O+(R_bar*(T[0]+273.15)*ln(y_H2O[12])))+(
n_dot_CO2[12]*(ex_ch_CO2+
(R_bar*(T[0]+273.15)*ln(y_CO2[12]))))
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vAFAHHMKJ LEGOAVW PAEI
"EX ANALYSIS"
I_dot_SOFC=((EX_dot[3]+EX_dot[11])-(EX_dot[4]+EX_dot[12])-W_dot_FC_stack)/1000
eta_II_SOFC=W_dot_FC_stack/((EX_dot[3]+EX_dot[11])-(EX_dot[4]+EX_dot[12]))
"EX13"
EX_dot[13]=EX_dot_ph[13]+EX_dot_ch[13]
EX_dot_ch[13]=(n_dot_O2[13]*(ex_ch_O2+(R_bar*(T[0]+273.15)*ln(y_O2[13])))+(n_dot_N2[13]
*(ex_ch_N2+(R_bar*(T[0]+
273.15)*ln(y_N2[13])))+(n_dot_H2O[13]*(ex_ch_H2O+(R_bar*(T[0]+273.15)*ln(y_H2O[13])))+(
n_dot_CO2[13]*(ex_ch_CO2+
(R_bar*(T[0]+273.15)*ln(y_CO2[13]))))
"EX 13-ph"
h_bar_N2_0[13]=molarmass(N2)*enthalpy(N2,T=T[0])
h_bar_O2_0[13]=molarmass(O2)*enthalpy(O2,T=T[0])
h_bar_CO2_0[13]=molarmass(CO2)*enthalpy(CO2,T=T[0])
h_bar_H2O_0[13]=molarmass(H2O)*enthalpy(H2O,T=T[0])
s_bar_N2[13]=molarmass(N2)*entropy(N2,T=T[13],P=(P[13]*y_N2[13]))
s_bar_O2[13]=molarmass(O2)*entropy(O2,T=T[13],P=(P[13]*y_O2[13]))
s_bar_CO2[13]=molarmass(CO2)*entropy(CO2,T=T[13],P=(P[13]*y_CO2[13]))
s_bar_H2O[13]=molarmass(H2O)*entropy(H2O,T=T[13],P=(P[13]*y_H2O[13]))
s_bar_N2_0[13]=molarmass(N2)*entropy(N2,T=T[0],P=(P_ref*y_N2[13]))
s_bar_O2_0[13]=molarmass(O2)*entropy(O2,T=T[0],P=(P_ref*y_O2[13]))
s_bar_CO2_0[13]=molarmass(CO2)*entropy(CO2,T=T[0],P=(P_ref*y_CO2[13]))
s_bar_H2O_0[13]=molarmass(H2O)*entropy(H2O,T=T[0],P=(P_ref*y_H2O[13]))
EX_dot_ph[13]=(n_dot_N2[13]*((h_bar_N2[13]-h_bar_N2_0[13])-(T[0]+273.15)*(s_bar_N2[13]-
s_bar_N2_0[13]))))+(
n_dot_O2[13]*((h_bar_O2[13]-h_bar_O2_0[13])-(T[0]+273.15)*(s_bar_O2[13]-
s_bar_O2_0[13]))))+(n_dot_CO2[13]*((
h_bar_CO2[13]-h_bar_CO2_0[13])-(T[0]+273.15)*(s_bar_CO2[13]-
s_bar_CO2_0[13]))))+(n_dot_H2O[13]*((h_bar_H2O[13]-
h_bar_H2O_0[13])-(T[0]+273.15)*(s_bar_H2O[13]-s_bar_H2O_0[13])))
"EX analysis"
I_dot_AB=(EX_dot[4]+EX_dot[12]-EX_dot[13])/1000
eta_II_AB=EX_dot[13]/(EX_dot[4]+EX_dot[12])
*****Air Compressor*****
*****state 1*****
T[1]=25[C]
h_bar[1]=molarmass(Air)*enthalpy(Air,T=T[1])
s_bar[1]=molarmass(Air)*entropy(Air,T=T[1],P=P[1])

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"EX1"
h_bar_N2[1]=molarmass(N2)*enthalpy(N2,T=T[1])
h_bar_O2[1]=molarmass(O2)*enthalpy(O2,T=T[1])
h_bar_N2_0[1]=molarmass(N2)*enthalpy(N2,T=T[0])
h_bar_O2_0[1]=molarmass(O2)*enthalpy(O2,T=T[0])
s_bar_N2[1]=molarmass(N2)*entropy(N2,T=T[1],P=(P[1]*y_N2[3]))
s_bar_O2[1]=molarmass(O2)*entropy(O2,T=T[1],P=(P[1]*y_O2[3]))
s_bar_N2_0[1]=molarmass(N2)*entropy(N2,T=T[0],P=(P_ref*y_N2[3]))
s_bar_O2_0[1]=molarmass(O2)*entropy(O2,T=T[0],P=(P_ref*y_O2[3]))
EX_dot[1]=(n_dot_N2[3]*((h_bar_N2[1]-h_bar_N2_0[1])-(T[0]+273.15)*(s_bar_N2[1]-
s_bar_N2_0[1])))+(n_dot_O2[3]*((
h_bar_O2[1]-h_bar_O2_0[1])-(T[0]+273.15)*(s_bar_O2[1]-s_bar_O2_0[1])))
"*****state 2* air compressor*****"
s_bar_s[2]=s_bar[1]
T_s[2]=temperature(Air,s=(s_bar_s[2]/molarmass(Air)),P=P[2])
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"h_bar_s[2]=molarmass(Air)*enthalpy(Air,T=T_s[2])"
h_bar_s[2]=molarmass(Air)*enthalpy(Air,s=(s_bar_s[2]/molarmass(Air)),P=P[2])
Eta_ac=(h_bar_s[2]-h_bar[1])/(h_bar[2]-h_bar[1])
T[2]=temperature(Air,h=(h_bar[2]/molarmass(Air)))
s_bar[2]=molarmass(Air)*entropy(Air,T=T[2],P=P[2])
W_dot_ac=n_dot[3]*(h_bar[2]-h_bar[1]) "Watt" "air compressor work"
"EX2"
h_bar_N2[2]=molarmass(N2)*enthalpy(N2,T=T[2])
h_bar_O2[2]=molarmass(O2)*enthalpy(O2,T=T[2])
h_bar_N2_0[2]=molarmass(N2)*enthalpy(N2,T=T[0])
h_bar_O2_0[2]=molarmass(O2)*enthalpy(O2,T=T[0])
s_bar_N2[2]=molarmass(N2)*entropy(N2,T=T[2],P=(P[2]*y_N2[3]))
s_bar_O2[2]=molarmass(O2)*entropy(O2,T=T[2],P=(P[2]*y_O2[3]))
s_bar_N2_0[2]=molarmass(N2)*entropy(N2,T=T[0],P=(P_ref*y_N2[3]))
s_bar_O2_0[2]=molarmass(O2)*entropy(O2,T=T[0],P=(P_ref*y_O2[3]))
EX_dot_ph[2]=(n_dot_N2[3]*((h_bar_N2[2]-h_bar_N2_0[2])-(T[0]+273.15)*(s_bar_N2[2]-
s_bar_N2_0[2])))+(n_dot_O2[3]*((
h_bar_O2[2]-h_bar_O2_0[2])-(T[0]+273.15)*(s_bar_O2[2]-s_bar_O2_0[2])))
EX_dot[2]=EX_dot_ph[2] "no chemical exrgy due to absence of combustion"
n_dot[2]=n_dot[3]
n_dot[1]=n_dot[2]
"EX ANALYSIS"
I_dot_ac=(EX_dot[1]-EX_dot_ph[2]+W_dot_ac)/1000
eta_II_ac=(EX_dot_ph[2]-EX_dot[1])/W_dot_ac
"*****Fuel Compressor*****"
"*****state 5*****"
T[5]=25[C]
h_bar[5]=molarmass(CH4)*enthalpy(CH4,T=T[5])
s_bar[5]=molarmass(CH4)*entropy(CH4,T=T[5],P=P[5])
"EX5"
h_bar_0[5]=molarmass(CH4)*enthalpy(CH4,T=T[0])
s_bar_0[5]=molarmass(CH4)*entropy(CH4,T=T[0],P=P_ref)
EX_dot[5]=n_dot_CH4[5]*((h_bar[5]-h_bar_0[5])-(T[0]+273.15)*(s_bar[5]-s_bar_0[5]))
n_dot_CH4[5]=n_dot_CH4[11] "it is above calculated value of x from steam carbon ratio
n_dot_CH4[11]=x"
n_dot[5]=n_dot_CH4[5]
n_dot[6]=n_dot[5]

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n_dot[7]=n_dot[5]
"*****sate 6*****"
s_bar_s[6]=s_bar[5]
T_s[6]=temperature(CH4,s=(s_bar_s[6]/molarmass(CH4)),P=P[6])
h_bar_s[6]=molarmass(CH4)*enthalpy(CH4,T=T_s[6])
Eta_fc=(h_bar_s[6]-h_bar[5])/(h_bar[6]-h_bar[5])
T[6]=temperature(CH4,h=(h_bar[6]/molarmass(CH4)))
s_bar[6]=molarmass(CH4)*entropy(CH4,T=T[6],P=P[6])
W_dot_fc=n_dot_CH4[11]*(h_bar[6]-h_bar[5])
"EX6"
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h_bar_0[6]=molarmass(CH4)*enthalpy(CH4,T=T[0])
s_bar_0[6]=molarmass(CH4)*entropy(CH4,T=T[0],P=P_ref)
EX_dot[6]=n_dot_CH4[11]*((h_bar[6]-h_bar_0[6])-(T[0]+273.15)*(s_bar[6]-s_bar_0[6]))
"EX ANALYSIS"
I_dot_fc=(EX_dot[5]-EX_dot[6]+W_dot_fc)/1000
eta_II_fc=(EX_dot[6]-EX_dot[5])/W_dot_fc
"*****Water pump*****"
"*****state 8*****"
T[8]=25
h_bar[8]=molarmass(steam)*enthalpy(Steam,T=T[8],P=P[8])
s_bar[8]=molarmass(steam)*entropy(Steam,T=T[8],P=P[8])
"EX8"
h_bar_0[8]=molarmass(steam)*enthalpy(Steam,T=T[0],P=P_ref)
s_bar_0[8]=molarmass(steam)*entropy(Steam,T=T[0],P=P_ref)
EX_dot[8]=n_dot_H2O[11]*((h_bar[8]-h_bar_0[8])-(T[0]+273.15)*(s_bar[8]-s_bar_0[8]))
"*****state 9*****"
s_bar_s[9]=s_bar[8]
T_s[9]=temperature(Steam,s=(s_bar_s[9]/molarmass(steam)),P=P[9])
h_bar_s[9]=molarmass(steam)*enthalpy(Steam,s=(s_bar_s[9]/molarmass(steam)),P=P[9])
Eta_wp=(h_bar_s[9]-h_bar[8])/(h_bar[9]-h_bar[8])
T[9]=temperature(Steam,h=(h_bar[9]/molarmass(steam)),P=P[9])
s_bar[9]=molarmass(steam)*entropy(Steam,h=(h_bar[9]/molarmass(steam)),P=P[9])
W_dot_wp=n_dot_H2O[11]*(h_bar[9]-h_bar[8])
"EX9"
h_bar_0[9]=molarmass(steam)*enthalpy(Steam,T=T[0],P=P_ref)
s_bar_0[9]=molarmass(steam)*entropy(Steam,T=T[0],P=P_ref)
EX_dot[9]=n_dot_H2O[11]*((h_bar[9]-h_bar_0[9])-(T[0]+273.15)*(s_bar[9]-s_bar_0[9]))
"EX ANALYSIS"
I_dot_wp=(EX_dot[8]-EX_dot[9]+W_dot_wp)/1000
eta_II_wp=(EX_dot[9]-EX_dot[8])/W_dot_wp
"*****state 14*****"
n_dot[14]=n_dot[13]
n_dot[15]=n_dot[13]
n_dot[16]=n_dot[13]
H_dot[14]=n_dot_N2[13]*h_bar_N2[14]+n_dot_O2[13]*h_bar_O2[14]+n_dot_H2O[13]*h_bar_H2O[14]+n_dot_CO2[13]*
h_bar_CO2[14]
h_bar_N2[14]=molarmass(N2)*enthalpy(N2,T=T[14])
h_bar_O2[14]=molarmass(O2)*enthalpy(O2,T=T[14])
h_bar_CO2[14]=molarmass(CO2)*enthalpy(CO2,T=T[14])
h_bar_H2O[14]=molarmass(H2O)*enthalpy(H2O,T=T[14])
"EX14"

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h_bar_N2_0[14]=molarmass(N2)*enthalpy(N2,T=T[0])
h_bar_O2_0[14]=molarmass(O2)*enthalpy(O2,T=T[0])
h_bar_CO2_0[14]=molarmass(CO2)*enthalpy(CO2,T=T[0])
h_bar_H2O_0[14]=molarmass(H2O)*enthalpy(H2O,T=T[0])
s_bar_N2[14]=molarmass(N2)*entropy(N2,T=T[14],P=(P[14]*y_N2[13]))
s_bar_O2[14]=molarmass(O2)*entropy(O2,T=T[14],P=(P[14]*y_O2[13]))
s_bar_CO2[14]=molarmass(CO2)*entropy(CO2,T=T[14],P=(P[14]*y_CO2[13]))
s_bar_H2O[14]=molarmass(H2O)*entropy(H2O,T=T[14],P=(P[14]*y_H2O[13]))
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s_bar_N2_0[14]=molarmass(N2)*entropy(N2,T=T[0],P=(P_ref*y_N2[13]))
s_bar_O2_0[14]=molarmass(O2)*entropy(O2,T=T[0],P=(P_ref*y_O2[13]))
s_bar_CO2_0[14]=molarmass(CO2)*entropy(CO2,T=T[0],P=(P_ref*y_CO2[13]))
s_bar_H2O_0[14]=molarmass(H2O)*entropy(H2O,T=T[0],P=(P_ref*y_H2O[13]))
EX_dot[14]=(n_dot_N2[13]*((h_bar_N2[14]-h_bar_N2_0[14])-(T[0]+273.15)*(s_bar_N2[14]-
s_bar_N2_0[14]))))+(
n_dot_O2[13]*((h_bar_O2[14]-h_bar_O2_0[14])-(T[0]+273.15)*(s_bar_O2[14]-
s_bar_O2_0[14]))))+(n_dot_CO2[13]*((
h_bar_CO2[14]-h_bar_CO2_0[14])-(T[0]+273.15)*(s_bar_CO2[14]-
s_bar_CO2_0[14]))))+(n_dot_H2O[13]*((h_bar_H2O[14]-
h_bar_H2O_0[14])-(T[0]+273.15)*(s_bar_H2O[14]-s_bar_H2O_0[14])))
"*****state 10*****"
n_dot[10]=n_dot[11]-n_dot[7]
n_dot[9]=n_dot[10]
n_dot[8]=n_dot[10]
T[7]=T_i
T[10]=T_i
h_bar[10]=molarmass(steam)*enthalpy(Steam,T=T_i,P=P[10])
"EX10"
h_bar_0[10]=molarmass(steam)*enthalpy(Steam,T=T[0],P=P_ref)
s_bar_0[10]=molarmass(steam)*entropy(Steam,T=T[0],P=P_ref)
s_bar[10]=molarmass(steam)*entropy(Steam,T=T_i,P=P[10])
EX_dot[10]=n_dot_H2O[11]*((h_bar[10]-h_bar_0[10])-(T[0]+273.15)*(s_bar[10]-s_bar_0[10]))
"*****Water Heat Exchanger*****"
n_dot_H2O[11]*(h_bar[10]-h_bar[9])=H_dot[13]-H_dot[14]
LMTD_WHE=((T[13]-T[14])-(T[10]-T[9]))/(ln((T[13]-T[14])/(T[10]-T[9])))
Q_dot_WHE=(H_dot[13]-H_dot[14])/1000
UA_WHE=Q_dot_WHE/LMTD_WHE
"EX Analysis"
I_dot_WHE=((EX_dot[9]-EX_dot[10])+(EX_dot_ph[13]-EX_dot[14]))/1000
eta_II_WHE=(EX_dot[10]-EX_dot[9])/(EX_dot_ph[13]-EX_dot[14])
"*****state 15*****"
H_dot[15]=n_dot_N2[13]*h_bar_N2[15]+n_dot_O2[13]*h_bar_O2[15]+n_dot_H2O[13]*h_bar_H2
O[15]+n_dot_CO2[13]*
h_bar_CO2[15]
h_bar_N2[15]=molarmass(N2)*enthalpy(N2,T=T[15])
h_bar_O2[15]=molarmass(O2)*enthalpy(O2,T=T[15])
h_bar_CO2[15]=molarmass(CO2)*enthalpy(CO2,T=T[15])
h_bar_H2O[15]=molarmass(H2O)*enthalpy(H2O,T=T[15])
"EX15"
h_bar_N2_0[15]=molarmass(N2)*enthalpy(N2,T=T[0])
h_bar_O2_0[15]=molarmass(O2)*enthalpy(O2,T=T[0])
h_bar_CO2_0[15]=molarmass(CO2)*enthalpy(CO2,T=T[0])
h_bar_H2O_0[15]=molarmass(H2O)*enthalpy(H2O,T=T[0])

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s_bar_N2[15]=molarmass(N2)*entropy(N2,T=T[15],P=(P[15]*y_N2[13]))
s_bar_O2[15]=molarmass(O2)*entropy(O2,T=T[15],P=(P[15]*y_O2[13]))
s_bar_CO2[15]=molarmass(CO2)*entropy(CO2,T=T[15],P=(P[15]*y_CO2[13]))
s_bar_H2O[15]=molarmass(H2O)*entropy(H2O,T=T[15],P=(P[15]*y_H2O[13]))
s_bar_N2_0[15]=molarmass(N2)*entropy(N2,T=T[0],P=(P_ref*y_N2[13]))
s_bar_O2_0[15]=molarmass(O2)*entropy(O2,T=T[0],P=(P_ref*y_O2[13]))
s_bar_CO2_0[15]=molarmass(CO2)*entropy(CO2,T=T[0],P=(P_ref*y_CO2[13]))
s_bar_H2O_0[15]=molarmass(H2O)*entropy(H2O,T=T[0],P=(P_ref*y_H2O[13]))
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EX_dot[15]=(n_dot_N2[13]*((h_bar_N2[15]-h_bar_N2_0[15])-(T[0]+273.15)*(s_bar_N2[15]-
s_bar_N2_0[15]))))+(
n_dot_O2[13]*((h_bar_O2[15]-h_bar_O2_0[15])-(T[0]+273.15)*(s_bar_O2[15]-
s_bar_O2_0[15]))))+(n_dot_CO2[13]*((
h_bar_CO2[15]-h_bar_CO2_0[15])-(T[0]+273.15)*(s_bar_CO2[15]-
s_bar_CO2_0[15]))))+(n_dot_H2O[13]*((h_bar_H2O[15]-
h_bar_H2O_0[15])-(T[0]+273.15)*(s_bar_H2O[15]-s_bar_H2O_0[15])))
"*****state 7*****"
h_bar[7]=molarmass(CH4)*enthalpy(CH4,T=T_i)
"EX7"
h_bar_0[7]=molarmass(CH4)*enthalpy(CH4,T=T[0])
s_bar_0[7]=molarmass(CH4)*entropy(CH4,T=T[0],P=P_ref)
s_bar[7]=molarmass(CH4)*entropy(CH4,T=T_i,P=P[7])
EX_dot[7]=n_dot_CH4[11]*((h_bar[7]-h_bar_0[7])-(T[0]+273.15)*(s_bar[7]-s_bar_0[7]))
"*****Fuel heat exchanger*****"
n_dot_CH4[11]*(h_bar[7]-h_bar[6])=H_dot[14]-H_dot[15]
LMTD_FHE=((T[14]-T[15])-(T[7]-T[6]))/(ln((T[14]-T[15])/(T[7]-T[6])))
Q_dot_FHE=(H_dot[14]-H_dot[15])/1000
UA_FHE=Q_dot_FHE/LMTD_FHE
"EX analysis"
I_dot_FHE=((EX_dot[6]-EX_dot[7])+(EX_dot[14]-EX_dot[15])/1000
eta_II_FHE=(EX_dot[7]-EX_dot[6])/(EX_dot[14]-EX_dot[15])
"*****state 16*****"
H_dot[16]=n_dot_N2[13]*h_bar_N2[16]+n_dot_O2[13]*h_bar_O2[16]+n_dot_H2O[13]*h_bar_H2
O[16]+n_dot_CO2[13]*
h_bar_CO2[16] "kW"
H_dot[16]/1000=m_dot[16]*h[16]
m_dot[16]=m_dot[13]
h_bar_N2[16]=molarmass(N2)*enthalpy(N2,T=T[16])
h_bar_O2[16]=molarmass(O2)*enthalpy(O2,T=T[16])
h_bar_CO2[16]=molarmass(CO2)*enthalpy(CO2,T=T[16])
h_bar_H2O[16]=molarmass(H2O)*enthalpy(H2O,T=T[16])
"EX16"
h_bar_N2_0[16]=molarmass(N2)*enthalpy(N2,T=T[0])
h_bar_O2_0[16]=molarmass(O2)*enthalpy(O2,T=T[0])
h_bar_CO2_0[16]=molarmass(CO2)*enthalpy(CO2,T=T[0])
h_bar_H2O_0[16]=molarmass(H2O)*enthalpy(H2O,T=T[0])
s_bar_N2[16]=molarmass(N2)*entropy(N2,T=T[16],P=(P[16]*y_N2[13]))
s_bar_O2[16]=molarmass(O2)*entropy(O2,T=T[16],P=(P[16]*y_O2[13]))
s_bar_CO2[16]=molarmass(CO2)*entropy(CO2,T=T[16],P=(P[16]*y_CO2[13]))
s_bar_H2O[16]=molarmass(H2O)*entropy(H2O,T=T[16],P=(P[16]*y_H2O[13]))
s_bar_N2_0[16]=molarmass(N2)*entropy(N2,T=T[0],P=(P_ref*y_N2[13]))
s_bar_O2_0[16]=molarmass(O2)*entropy(O2,T=T[0],P=(P_ref*y_O2[13]))
s_bar_CO2_0[16]=molarmass(CO2)*entropy(CO2,T=T[0],P=(P_ref*y_CO2[13]))

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```

s_bar_H2O_0[16]=molarmass(H2O)*entropy(H2O,T=T[0],P=(P_ref*y_H2O[13]))
EX_dot[16]=(n_dot_N2[13]*((h_bar_N2[16]-h_bar_N2_0[16])-(T[0]+273.15)*(s_bar_N2[16]-
s_bar_N2_0[16]))))+(
n_dot_O2[13]*((h_bar_O2[16]-h_bar_O2_0[16])-(T[0]+273.15)*(s_bar_O2[16]-
s_bar_O2_0[16]))))+(n_dot_CO2[13]*((
h_bar_CO2[16]-h_bar_CO2_0[16])-(T[0]+273.15)*(s_bar_CO2[16]-
s_bar_CO2_0[16]))))+(n_dot_H2O[13]*((h_bar_H2O[16]-
h_bar_H2O_0[16])-(T[0]+273.15)*(s_bar_H2O[16]-s_bar_H2O_0[16])))
"*****Air heat exchanger*****"
n_dot[3]*(h_bar[3]-h_bar[2])=H_dot[15]-H_dot[16]
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vAFAHHMKJ LEGOAVW PAEI
LMTD_AHE=((T[15]-T[16])-(T[3]-T[2]))/(ln((T[15]-T[16])/(T[3]-T[2])))
Q_dot_AHE=(H_dot[15]-H_dot[16])/1000
UA_AHE=Q_dot_AHE/LMTD_AHE
"exergy analysis"
I_dot_AHE=((EX_dot_ph[2]-EX_dot_ph[3])+(EX_dot[15]-EX_dot[16]))/1000
eta_II_AHE=(EX_dot_ph[3]-EX_dot_ph[2])/(EX_dot[15]-EX_dot[16])
"MIX"
I_dot_mix=((EX_dot[7]+(n_dot_CH4[11]*ex_ch_CH4))+(EX_dot[10]+(n_dot_H2O[11]*ex_ch_H2
O))-EX_dot[11])/1000
"EX_dot[17]=EX_dot[16]"
EX_dot_f_CH4=((n_dot_CH4[11]*ex_ch_CH4)+(n_dot_H2O[11]*ex_ch_H2O)+EX_dot_ch[3])/100
0
ATA=EX_dot_f_CH4-
(I_dot_AB+I_dot_ac+I_dot_fc+I_dot_wp+I_dot_WHE+I_dot_FHE+I_dot_AHE+I_dot_mix+(
W_dot_FC_stack_ac/1000)+((EX_dot[16]+EX_dot_ch[13])/1000))
I_dot=((EX_dot[3]+EX_dot[7]+EX_dot[10]+(n_dot_CH4[11]*ex_ch_CH4)+(n_dot_H2O[11]*ex_ch
_H2O))-(EX_dot[4]+
EX_dot[12])-W_dot_FC_stack_ac)/1000
W_dot_net=W_dot_FC_stack_ac-(W_dot_wp+W_dot_fc+W_dot_ac) "Watt"
w_dot_net_Total=w_dot_net/1000 "kW"
m_f=(n_dot_CH4[11]*16)/1000
LHV=50000[kJ/kg]
eta_en_SOFC=w_dot_net_Total/(m_f*LHV)
"GT"
P_r4=7 "pressure ration in GT"
P[16]/P[17]=P_r4
n_dot[17]=n_dot[16]
eta_gt=0.85
h_bar[16]=y_N2[13]*h_bar_N2[16]+y_O2[13]*h_bar_O2[16]+y_H2O[13]*h_bar_H2O[16]+y_CO2[
13]*h_bar_CO2[16]
s_bar[16]=y_N2[13]*s_bar_N2[16]+y_O2[13]*s_bar_O2[16]+y_H2O[13]*s_bar_H2O[16]+y_CO2[
13]*s_bar_CO2[16]
s_bar_s[17]=s_bar[16]
s_bar_N2_s[17]=molarmass(N2)*entropy(N2,T=T_s[17],P=(P[17]*y_N2[13]))
s_bar_O2_s[17]=molarmass(O2)*entropy(O2,T=T_s[17],P=(P[17]*y_O2[13]))
s_bar_CO2_s[17]=molarmass(CO2)*entropy(CO2,T=T_s[17],P=(P[17]*y_CO2[13]))
s_bar_H2O_s[17]=molarmass(H2O)*entropy(H2O,T=T_s[17],P=(P[17]*y_H2O[13])) "T_s[17]is
calaulated from these
equations here unkown"
s_bar_s[17]=y_N2[13]*s_bar_N2_s[17]+y_O2[13]*s_bar_O2_s[17]+y_H2O[13]*s_bar_H2O_s[17]+
y_CO2[13]*
s_bar_CO2_s[17]

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```

h_bar_N2_s[17]=molar mass(N2)*enthalpy(N2,T=T_s[17])
h_bar_O2_s[17]=molar mass(O2)*enthalpy(O2,T=T_s[17])
h_bar_CO2_s[17]=molar mass(CO2)*enthalpy(CO2,T=T_s[17])
h_bar_H2O_s[17]=molar mass(H2O)*enthalpy(H2O,T=T_s[17])
h_bar_s[17]=y_N2[13]*h_bar_N2_s[17]+y_O2[13]*h_bar_O2_s[17]+y_H2O[13]*h_bar_H2O_s[17]
+y_CO2[13]*
h_bar_CO2_s[17]
eta_gt=(h_bar[16]-h_bar[17])/(h_bar[16]-h_bar_s[17])
h_bar_N2[17]=molar mass(N2)*enthalpy(N2,T=T[17])
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vAFAHHMKJ LEGOAVW PAEI
h_bar_O2[17]=molar mass(O2)*enthalpy(O2,T=T[17])
h_bar_CO2[17]=molar mass(CO2)*enthalpy(CO2,T=T[17])
h_bar_H2O[17]=molar mass(H2O)*enthalpy(H2O,T=T[17])
h_bar[17]=y_N2[13]*h_bar_N2[17]+y_O2[13]*h_bar_O2[17]+y_H2O[13]*h_bar_H2O[17]+y_CO2[
13]*h_bar_CO2[17]
H_dot[17]=n_dot_N2[13]*h_bar_N2[17]+n_dot_O2[13]*h_bar_O2[17]+n_dot_H2O[13]*h_bar_H2
O[17]+n_dot_CO2[13]*
h_bar_CO2[17]
W_gt=H_dot[16]-H_dot[17] "watt"
"state 18"
T[18]=189.75[C]
h_bar_N2[18]=molar mass(N2)*enthalpy(N2,T=T[18])
h_bar_O2[18]=molar mass(O2)*enthalpy(O2,T=T[18])
h_bar_CO2[18]=molar mass(CO2)*enthalpy(CO2,T=T[18])
h_bar_H2O[18]=molar mass(H2O)*enthalpy(H2O,T=T[18])
h_bar[18]=y_N2[13]*h_bar_N2[18]+y_O2[13]*h_bar_O2[18]+y_H2O[13]*h_bar_H2O[18]+y_CO2[
13]*h_bar_CO2[18]
H_dot[18]=n_dot_N2[13]*h_bar_N2[18]+n_dot_O2[13]*h_bar_O2[18]+n_dot_H2O[13]*h_bar_H2
O[18]+n_dot_CO2[13]*
h_bar_CO2[18]
m_dot_CO2=((n_dot_CO2[13])*molar mass(CO2))/1000[kg/s]
"Efficiency and work calculation of SOFC+GT"
W_dot_net_SOFC_GT=(W_dot_net+W_gt)/1000 "kW"
eta_en_SOFC_GT=(W_dot_net_SOFC_GT/(m_f*LHV))*100[%]
eta_ex_SOFC_GT=(W_dot_net_SOFC_GT/(Ex_input))*100[%] "chemical
exergy of CH4 only at point 11 works as exergy input to the fuel cell"
"STORAGE tank"
T[29]=98[C]
T[30]=88[C]
P[29]=101.3[kPa]
P[29]=P[30]
h[29]=enthalpy(Water,T=T[29],P=P[29])
s[29]=entropy(Water,T=T[29],P=P[29])
s[30]=entropy(Water,T=T[30],P=P[30])
h[30]=enthalpy(Water,T=T[30],P=P[30])
eta_pump1=0.85
rp1=1.2
P[31]/P[30]=rp1
s_s[31]=s[30]
h_s[31]=enthalpy(Water,s=s_s[31],P=P[31])
eta_pump1=(h_s[31]-h[30])/(h[31]-h[30])
W_dot_pump1=m_dot[30]*(h[31]-h[30])
m_dot[29]=m_dot[31]

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m_dot[30]=m_dot[31]
T[31]=temperature(Water,P=P[31],h=h[31])
s[31]=entropy(Water,T=T[31],P=P[31])
m_w=500[kg]
Q_ST=m_dot[31]*(h[29]-h[31])
m_dot[31]*(h[29]-h[31])=(m_w/duration)*4.18*(T[29]-T[31])
Q_ST=(H_dot[17]-H_dot[18])/1000
Q_GEN=m_dot[31]*(h[29]-h[30])
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vAFAHHMKJ LEGOAVW PAEI
LMTD_GEN=((T[29]-T[25])-(T[30]-T[21]))/ln((T[29]-T[25])/(T[30]-T[21]))
UA_GEN=Q_GEN/LMTD_GEN
T_GEN=80[C]
T[25]=T_GEN
P[25]=5.63[kPa]
h[25]=enthalpy(Water,T=T[25],P=P[25])
s[25]=entropy(Water,T=T[25],P=P[25])
T[36]=25[C]
T[37]=35[C]
P[36]=101.3[kPa]
P[37]=P[36]
"cond"
Q_cond=m_dot[25]*(h[25]-h[26])
LMTD_cond=((T[25]-T[37])-(T[26]-T[36]))/ln((T[25]-T[37])/(T[26]-T[36]))
UA_cond=Q_cond/LMTD_cond
h[36]=enthalpy(Water,T=T[36],P=P[36])
s[36]=entropy(Water,T=T[36],P=P[36])
h[37]=enthalpy(Water,T=T[37],P=P[37])
s[37]=entropy(Water,T=T[37],P=P[37])
T[26]=T[36]+5[C]
P[25]=P[26]
h[26]=enthalpy(Water,T=T[26],P=P[26])
s[26]=entropy(Water,T=T[26],P=P[26])
"Evaporator"
T_EVAP=5[C]
T[28]=T_EVAP
T[27]=T[28]
Q[28]=1
T[33]=T[28]+2[C]
T[32]=14[C]
P[32]=101.3[kPa]
P[32]=P[33]
Q[27]=0
Q_EVAP=5.275[kW] "1.5TR"
s[28]=entropy(Water,T=T[28],x=Q[28])
h[28]=enthalpy(Water,T=T[28],x=Q[28])
P[28]=pressure(Water,T=T[28],x=Q[28])
h[27]=enthalpy(Water,T=T[27],x=Q[27])
s[27]=entropy(Water,T=T[27],x=Q[27])
P[27]=pressure(Water,T=T[27],x=Q[27])
h[32]=enthalpy(Water,T=T[32],P=P[32])
s[32]=entropy(Water,T=T[32],P=P[32])
h[33]=enthalpy(Water,T=T[33],P=P[33])
s[33]=entropy(Water,T=T[33],P=P[33])

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Q_EVAP=m_dot[32]*(h[32]-h[33])
m_dot[33]=m_dot[32]
Q_EVAP=m_dot[27]*(h[28]-h[27])
m_dot[27]=m_dot[28]
m_dot[25]=m_dot[28]
m_dot[26]=m_dot[28]
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vAFAHHMKJ LEGOAVW PAEI
LMTD_EVAP=((T[32]-T[28])-(T[33]-T[27]))/ln((T[32]-T[28])/(T[33]-T[27]))
UA_EVAP=Q_EVAP/LMTD_EVAP
"Absorber"
T[34]=25[C]
T[35]=35[C]
P[34]=101.3[kPa]
P[34]=P[35]
h[34]=enthalpy(Water,T=T[34],P=P[34])
s[34]=entropy(Water,T=T[34],P=P[34])
h[35]=enthalpy(Water,T=T[35],P=P[35])
s[35]=entropy(Water,T=T[35],P=P[35])
T_ABS=35[C]
T[19]=T_ABS
P[19]=P[28]
x[19]=x_librh2o(T[19],P[19]) "Mass frcation of LiBr in Water mixture"
h[19]=h_librh2o(T[19],x[19])
s[19]=s_librh2o(T[19],x[19])
Q_ABS=m_dot[24]*h[24]+m_dot[28]*h[28]-m_dot[19]*h[19]
Q_ABS=m_dot[34]*(h[35]-h[34])
LMTD_ABS=(T[24]+T[34])/2
UA_ABS=Q_ABS/LMTD_ABS
m_dot[35]=m_dot[34]
"Pump2"
eta_pump2=0.85
s_s[20]=s[19]
P[20]=P[25]
x[20]=x[19]
h_s[20]=h_librh2o(T_s[20],x[20])
s_s[20]=s_librh2o(T_s[20],x[20])
eta_pump2=(h_s[20]-h[19])/(h[20]-h[19])
h[20]=h_librh2o(T[20],x[20])
s[20]=s_librh2o(T[20],x[20])
"SHE"
x[21]=x[20]
epsilon_SHE=0.7
P[21]=P[20]
{epsilon_SHE=(T[22]-T[23])/(T[22]-T[20])}
epsilon_SHE=(m_dot[22]*h[22]-m_dot[23]*h[23])/(m_dot[22]*h[22]-m_dot[20]*h[20])
epsilon_SHE=(T[21]-T[20])/(T[22]-T[20])
T[22]=T[25] "Tgen"
P[22]=P[21]
x[22]=x_librh2o(T[22],P[22]) "Mass frcation of LiBr in Water mixture"
h[22]=h_librh2o(T[22],x[22])
s[22]=s_librh2o(T[22],x[22])
x[23]=x[22]
"mass balance of LiBr"

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```

x[21]*m_dot[21]=x[22]*m_dot[22]
m_dot[23]=m_dot[22]
m_dot[24]=m_dot[22]
m_dot[19]=m_dot[24]+m_dot[28]
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vAFAHHMKJ LEGOAVW PAEI
m_dot[20]=m_dot[19]
m_dot[21]=m_dot[19]
P[23]=P[22]
h[23]=h_librh2o(T[23],x[23])
s[23]=s_librh2o(T[23],x[23])
m_dot[22]*h[22]+m_dot[20]*h[20]=m_dot[23]*h[23]+m_dot[21]*h[21]
{h[21]=h_librh2o(T[21],x[21])}
s[21]=s_librh2o(T[21],x[21])
LMTD_SHE=((T[22]-T[21])-(T[23]-T[20]))/ln((T[22]-T[21])/(T[23]-T[20]))
Q_SHE=(m_dot[21]*h[21]-m_dot[20]*h[20])
UA_SHE=Q_SHE/LMTD_SHE
"PRV"
P[24]=P[19]
h[23]=h[24]
x[24]=x[23]
h[24]=h_librh2o(T[24],x[24])
s[24]=s_librh2o(T[24],x[24])
W_dot_pump2=abs(m_dot[20]*(h[20]-h[19]))
EX_EVAP=abs(Q_EVAP*(1-((T[0]+273.15)/(T_EVAP+273.15))))
COP=Q_EVAP/(Q_GEN-W_dot_pump2)
eta_en_SOFC_GT_VAR=((W_dot_net_SOFC_GT+Q_EVAP)/(m_f*LHV))*100[%]
eta_ex_SOFC_GT_VAR=((W_dot_net_SOFC_GT+EX_EVAP)/(Ex_input))*100[%]
"economic analysis"
"cost of SOFC components"
Z_SOFC_stack=1.1*A_a*N_FC*(2.96*(T_e+273.15)-1907)
Z_ac=91562*((W_dot_ac/1000)/455)^0.67 "air compressor"
Z_fc=91562*((W_dot_fc/1000)/455)^0.67 "fuel compressor"
Z_wp=3450*(W_dot_wp/1000)^0.71 "water pump in SOFC system"
Z_AB=((46.08*m_dot[4])/(0.995-(P[13]/P[4])))*(1+exp(0.018*(T[13]+273.15)-26.4)) "after burner"
Z_AHE=2500*UA_AHE
Z_FHE=2500*UA_FHE
Z_WHE=2500*UA_WHE
Z_inverter=10^5*((W_dot_FC_stack_ac/1000)/500)^0.7
Z_gt=(W_gt/1000)*(1318.5-98.32*ln(W_gt/1000))
"VAR cost"
Z_GEN=5000*UA_GEN
Z_SHE=2500*UA_SHE
Z_pump2=3450*(abs(W_dot_pump2))^0.71
Z_pump1=3450*(abs(W_dot_pump1))^0.71
Z_ABS=5000*UA_ABS
Z_EVAP=5000*UA_EVAP
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vAFAHHMKJ LEGOAVW PAEI
Z_cond=1700*UA_cond "VAR condensor"
Z_exv=114.5*m_dot[19]
Z_PRV=114.5*m_dot[23]
Z_ST=931*(m_w/1000)

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```

"Cost rate of the plant"
in=0.12
N=20 "years"
CRF=(in*(1+in)^N)/(((1+in)^N)-1)
phi=1.06
N_hours=7200 {hours} {net amount of hours performed in a year, which is set as 7200}
Z_total=Z_SOFC_stack+Z_AB+Z_ac+Z_fc+Z_wp+Z_AHE+Z_FHE+Z_WHE+Z_inverter+Z_gt+Z_
ST+Z_pump1+Z_cond+
Z_GEN+Z_SHE+Z_pump2+Z_ABS+Z_EVAP+Z_exv+Z_PRV
Z_dot_total=(CRF*phi*Z_total)/N_hours {$/h}
Z_SOFC_GT_total=Z_SOFC_stack+Z_AB+Z_ac+Z_fc+Z_wp+Z_AHE+Z_FHE+Z_WHE+Z_invert
er+Z_gt
Z_dot_SOFC_GT_total=(CRF*phi*Z_SOFC_GT_total)/N_hours {$/h}
C_f=0.003[$/MJ]
C_dot_f=C_f*((m_f*LHV)/1000)*3600[$/h]
C_pen=0.024[$/kg] "penalty cost"
C_dot_env=C_pen*m_dot_CO2*3600[$/h]
C_dot_Plant=C_dot_f+C_dot_env+Z_dot_total "Total Plant cost rate $/h"
C_dot_SOFC_GT=C_dot_f+C_dot_env+Z_dot_SOFC_GT_total
EMI_overall=((m_dot_CO2)/(W_dot_net_SOFC_GT+Q_EVAP))*3600*1000 {kg/MWh}
EMI_SOFC_GT=((m_dot_CO2)/(W_dot_net_SOFC_GT))*3600*1000
{kg/MWh} {$PX$96} {$ST$OFF}

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RESEARCH PUBLICATIONS:

Published in SCI Journals:

1. Shobhit Mishra, Raj Kumar Singh. Performance evaluation of absorption cooling system for air conditioning-based novel trigeneration system using solar energy, Volume 46, Article 354 (pg. 1-13); Journal of the Brazilian Society of Mechanical Sciences and Engineering (2024);. IF -1.8
2. Shobhit Mishra, Raj Kumar Singh. Exergy, energy, economic and environmental evaluations of a SOFC based cooling system, International Journal of Exergy. IF – 1.1

Conference papers

1. Shobhit Mishra, Raj Kumar Singh. Performance Assessment of a Novel Absorption Cooled Trigeneration System Powered by Solar Energy for Air Conditioning, IEEE (UP CONCLAVE); Amity University Uttar Pradesh, 1st to 3rd December 2023.
2. Shobhit Mishra, Raj Kumar Singh. Thermodynamic Assessment of Solar-Based Combined Brayton Cycle with Inlet Cooling Through Vapor Absorption Cooling System.; NIT Warangal, ICMECH-REC, 29-31 May 2024.



Performance evaluation of absorption cooling system for air conditioning-based novel trigeneration system using solar energy

Shobhit Mishra¹ · Raj Kumar Singh¹

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Abstract

This work aims to exploit the waste heat from a basic helium Brayton cycle (HBC) driven by a solar power tower (SPT). The two subsystem heat recovery steam generator (HRSG) and vapour absorption cooling system (VACS) have been implemented to recover the waste heat, respectively, for heating to generate steam and cooling for air conditioning. A comprehensive exergy and energy analysis of this proposed trigeneration system was carried out with parametric analysis. For analysis purpose Engineering Equation Solver software has been used. It was concluded that energy and exergy efficiency and net work output of the proposed trigeneration system were observed as 44.96%, 34.15%, 14,562 kW, respectively. The heating production through the HRSG was obtained 8510 kW while the cooling production by VACS as 115.10 kW. Moreover, coefficient of performance (COP) of the VACS subsystem was observed as 0.8015. Energy and exergy efficiency of the SPT operated basic HBC using the subsystems was improved by 58.98% and 12.92%, respectively. Parametric analysis revealed that the helium turbine inlet temperature and solar heliostat field efficiency much affected the trigeneration system performance.

Keywords Performance assessment · Solar power tower · Trigeneration system · Vapour absorption cooling system · HRSG · Air conditioning

Abbreviations

CR	Central receiver
HBC	Helium Brayton cycle
SPT	Solar power tower
SHE	Solution heat exchanger
SCO ₂	Supercritical CO ₂
VACS	Vapour absorption cooling system
ORC	Organic Rankine cycle
IHE	Intermediate heat exchanger
HC	Helium compressor
HT	Helium turbine
COP	Coefficient of performance
CPR	Compressor pressure ratio
CIT	Compressor inlet temperature
HTIT	Helium turbine inlet temperature
DNI	Direct normal irradiation (W/m ²)
HRSG	Heat recovery steam generator

List of symbols

EX	Rate of exergy (kW)
h	Specific enthalpy (kJ/kg)
$\dot{W}$	Power (kW)
N_{hel}	Number of heliostats
$\dot{Q}$	Rate of heat interaction (kW)
T	Temperature (K)
ED	Exergy destruction rate (kW)
A	Area (m ²)
s	Specific entropy (kJ/kg K)
$\dot{m}$	Mass flow rate (kg/s)

Subscripts

c	Condenser
e	Exit/evaporator
g	Generator
j	Particular state
rec	Receiver
ref	Reference/reflectivity
hel	Heliostat
i	Inlet
0	Dead condition
tri	Trigeneration
co	Cogeneration

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✉ Shobhit Mishra
shobhit721mishra@gmail.com

¹ Department of Mechanical Engineering, Delhi Technological University, Bawana Road, Delhi 110042, India

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Exergy, energy, economic and environmental evaluations of a SOFC based cooling system

Shobhit Mishra*, Raj Kumar Singh

Department of Mechanical Engineering, Delhi Technological University, Delhi, India 110042

*Corresponding author email address: shobhit721mishra@gmail.com

Raj Kumar Singh: rajkumarsingh@dtu.ac.in

Biographical statements:

Shobhit Mishra: Mr. Mishra is a doctoral researcher in the department of Mechanical Engineering, Delhi Technological University, India. He is a researcher with expertise in the area of sustainable energy systems. His research is focused on pioneering solutions in renewable energy technologies.

Raj Kumar Singh: Dr. Singh is professor in the department of Mechanical Engineering, Delhi Technological University, India. He has contributed significantly to scientific discourse through 50 SCI journal papers, specifically addressing key challenges in clean energy transitions and fluid mechanics.

Abstract

This study developed a novel solid oxide fuel cell-based vapour absorption refrigeration system (VARS) for combined cooling and power generation. Thermodynamic, economic, and environmental analyses were performed on the proposed system using computational techniques. The proposed plant obtained power output, exergy, and energy efficiency of 551.29 kW, 48.03%, and 50.18%, respectively, at given operating conditions. The cooling effects of 5.275 kW were obtained from the VARS with a coefficient of performance of 0.752. The total cost rate of the proposed plant was observed as 33.57\$/h, respectively. However, the CO₂ emission per MWh was obtained as 394.70 kg/MWh.

Keywords: Gas turbine, solid oxide fuel cell, vapor absorption refrigeration system, combined cooling and power generation, hybrid system, room air conditioning

1. Introduction

It is known that traditional energy sources are responsible for the natural disasters that underscore the detrimental impacts of global warming on human society. Given the current circumstances, there is an urgent need for renewable technologies that are both sustainable and profitable. One of the most popular renewable technologies is the solid oxide fuel cell (SOFC). SOFCs use electrochemical processes to transform chemical energy of fuel into electrical power and heat. Unlike other fuel cells, SOFCs experience extremely high temperatures during the course of their electrochemical processes. Fuel flexibility rises with an increased operating temperature. SOFCs are beneficial to produce the heat and power due to their high electrical efficiency (around 60%) and zero negative effects on the environment. The performance of SOFC-based thermal systems can be enhanced by combining SOFCs with other cycles because of their high-temperature applicability [1].

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THERMODYNAMIC ASSESSMENT OF SOLAR BASED COMBINED BRAYTON CYCLE WITH INLET COOLING THROUGH VAPOR ABSORPTION COOLING SYSTEM

Shobhit Mishra^{1*}, Raj Kumar Singh¹

¹Department of Mechanical Engineering, Delhi Technological University, Delhi, India,
110042

E-mail: shobhit721mishra@gmail.com

As was previously noted, there are several irreversibilities in the solar portion of solar power tower (SPT) plants that cannot be avoided. To improve the operation of the SPT plant, a new, efficient power producing unit must be built. The current study created the unique integrated cycle for SPT plant. The vapor absorption cooling system (VACS) was used for waste heat recovery that cools the inlet of the compressor of the helium Bryton cycle (HBC). Also a heat recovery steam generator (HRSG) was used to recover heat for steam production. The energy and energy analysis of the proposed system was done using engineering equation solver software. Furthermore, a parametric analysis was conducted to investigate the effects of significant factors on plant performance. The plant yielded network production, energy, and energy efficiency of 19918 kW, 41.36%, and 38.62%, respectively. It was also concluded that the presented intercooled system produced 19.23 % more efficiency than the basic HBC solar based system. Also, from the HRHG The steam 56.67kg/s. The comparison with prior studies shows that the new system performed thermodynamically better than Rankine and supercritical CO₂ based power generating systems. Additionally, the current system is more effective than the prior one, which was a simple ORC-based system.



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Performance Assessment of a Novel Absorption Cooled Trigeneration System Powered by Solar Energy for Air Conditioning

Shobhit Mishra
Department of Mechanical Engineering
Delhi Technological Engineering, Delhi
India, 110042
Email: samns899@gmail.com

Raj Kumar Singh
Department of Mechanical Engineering
Delhi Technological Engineering, Delhi
India, 110042
Email: rajkumarsingh@dce.ac.in

Abstract—utilizing the waste heat from the solar power tower (SPT) to run a basic helium Brayton cycle (HBC) is the aim of this study. The two subsystems that have been put in place to recover waste heat are the vapour absorption cooling system (VACS) and the heat recovery steam generator (HRSG), which is used for cooling air conditioning systems and heating to produce steam, respectively. A comprehensive exergy and energy analysis of this proposed trigeneration system were carried out with parametric analysis. For analysis purpose Engineering Equation Solver software has been used. It was concluded that energy and exergy efficiency and net work output of the proposed trigeneration system were observed as 44.96%, 34.15%, 14562 kW respectively. The heating production through the HRSG was obtained 8510 kW while the cooling production by VACS as 115.10 kW. Furthermore, the VACS subsystem's coefficient of performance (COP) was discovered to be 0.8015. Energy and exergy efficiency of the SPT operated basic HBC using the subsystems were improved by 58.98% and 12.92% respectively. Parametric analysis revealed that the helium turbine inlet temperature and solar heliostat field efficiency much affected the trigeneration system performance.

Keywords—Performance assessment, solar power tower, trigeneration system, vapor absorption cooling system, HRSG, air conditioning

I. INTRODUCTION

This By 2030, the global energy demand will have increased by at least 50%. Future security and sustainability depend on the production of carbon-free energy [1]. The remarkable concentration solar power (CSP) system can provide the requirement for safe and reliable electricity. The power production industry is currently showing a great deal of interest in the CSP system because of its high-temperature

appealing for applications that recover waste heat inside of automobiles. Transcritical and supercritical CO₂ cycles have drawn more attention lately because of their enhanced performance in elevated temperatures waste heat recovery. CO₂ power cycles have received a lot of interest since CO₂ is the working fluid in the Brayton cycles, which also increase the efficiency of turbines and heat exchangers [3].

In order to convert heat into electricity, SPT may use closed Brayton cycles or traditional Rankine cycles, depending on the temperature of the heat source. The compact size and effective performance of the sCO₂ Brayton cycle have made research on it more popular recently [4]. Li et al.'s comprehensive classification of sCO₂ systems and their advantages for use in solar and nuclear power plants was published in [5]. Al2O₃ nanofluid was employed by Gkoutas et al. [6] to enhance heat transfer; they discovered that this resulted in a 0.9% decrease in the heat exchanger's length and a 14% decrease in a pressure loss in the sCO₂ system. A detailed analysis of heat exchanger features used in helium and sCO₂ Brayton cycles by Chai and Tassuo [7] suggests that nickel-based alloys should be used at higher temperatures, while stainless steel is the best material for heat exchangers out to an operational temperature of 650 °C.

In addition, Zare and Hasanizadeh [8] presented a combined HBC and two ORC that were both run on an SPT system in the direction of helium fluid. They were able to increase the power plant's energy efficiency to almost 30%. Also recently, Khan et al. [9] proposed as novel combined cycle using HBC as topping cycle and tCO₂ cycle has been taken for extra power generation for using the SPT heat. They performed a thermodynamic and exergoeconomic analysis of the presented system. They observed proposed combined power plant improved by energy and exergy



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I, **Shobhit Mishra**, completed **B. Tech** in Mechanical Engineering from **Uttar Pradesh Technical University, Lucknow, India**, and **M.Tech** in Thermal Engineering from **JAMIA MILLIA ISLAMIA (Delhi, India)**. I have enrolled **Ph.D.** in Department of Mechanical Engineering at **Delhi Technological University, Delhi-42, (India)** under the supervision of **Prof. Raj Kumar Singh**. I have published **2 SCI/SCIE** and **2 Scopus** international conference papers in reputed journals.