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CHAPTER 1: INTRODUCTION

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].

1.1 India's Cooling Challenge and Energy Landscape

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 (IEA, 2018; BEE, 2019). This growth could push cooling to account for up to 45% of India's peak electricity demand in major cities during summer [1,2,5].

Moreover, the country remains highly dependent on coal, which supplied approximately 70% of total electricity generation in 2022 [3]. 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.

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

(GWP) and often Ozone Depletion Potential (ODP). Though newer refrigerants (e.g., R-32) aim to reduce GWP, the environmental impact of their entire lifecycle remains significant.

1.2 Problems associated with VCS-based cooling 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.

Key advantages of this alignment include [7]:

- 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
- 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 (130–170°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.

Recognizing the urgency, the Government of India has launched several policy initiatives aimed at transforming the cooling sector:

1.3 India Cooling Action Plan (ICAP): Launched in 2019 [5], ICAP is the world's first national plan to comprehensively address the cooling demand across sectors. 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
- 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.

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
- 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.

India stands at a pivotal juncture where cooling demand is rising sharply, but so is the opportunity to lead the world in deploying climate-resilient, solar-integrated cooling systems.

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. 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

Solution Pump	Pumps the rich solution to generator
Generator	Uses external heat to separate refrigerant from absorbent
Condenser	Condenses refrigerant vapor by rejecting heat to the environment
Expansion Valve	Drops refrigerant pressure before entering evaporator
Heat Exchanger	Improves efficiency by preheating the weak solution

1.7 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., LiBr). 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.

1.8 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)
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
Cost	Lower initial	Lower O&M

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.

Enhancements and Innovations

Recent research and commercial efforts have aimed at overcoming VAS limitations:

- Nanofluid-enhanced absorbents: Improve thermal conductivity and absorption rates.
- Desiccant-assisted absorption: Increases latent load handling in humid regions.
- Hybrid absorption–compression systems: Combine best of both worlds.
- AI-based control systems: Predict and optimize generator temperature, load response, and solar input.

1.9 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)

1.10: 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.

1.11 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)

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.

1.12 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

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.

1.13 Refrigerant–Absorbent Pairs

A. 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.

B. Ammonia–Water (NH₃–H₂O)

- Suitable for low-temperature refrigeration (as low as -20 °C).

- Operates at high pressures (up to 20 bar).
- Toxicity of ammonia requires strict leak management and material selection.
- Requires rectifier or dephlegmator to remove water vapor from ammonia stream.

C. Ammonia–Lithium Nitrate ($\text{NH}_3\text{--LiNO}_3$)

- No water used → no rectifier required.
- Emerging system for sub-zero applications.
- High viscosity of LiNO_3 limits mass transfer; needs compact heat exchangers.
- Lower corrosiveness compared to $\text{NH}_3\text{--H}_2\text{O}$.

D. Water–Lithium Chloride ($\text{H}_2\text{O--LiCl}$)

- 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	$\text{H}_2\text{O--LiBr}$	$\text{NH}_3\text{--H}_2\text{O}$	$\text{NH}_3\text{--LiNO}_3$	$\text{H}_2\text{O--LiCl}$
Cooling Range	$> 5^\circ\text{C}$	-20 to $+10^\circ\text{C}$	-10 to $+5^\circ\text{C}$	$8\text{--}14^\circ\text{C}$
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

Different working pairs and configurations require different generator temperatures, which determine compatibility with solar thermal technologies:

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

Solar Collector	Temperature Range	Suitable Pairs
Flat Plate Collector (FPC)	60–80 °C	H ₂ O–LiBr (single-effect)
Evacuated Tube Collector (ETC)	80–110 °C	H ₂ O–LiBr (single-effect), NH ₃ –LiNO ₃
Compound Parabolic Collector (CPC)	100–140 °C	NH ₃ –H ₂ O (with rectifier)
Parabolic Trough Collector (PTC)	140–180 °C	H ₂ O–LiBr (double-effect), NH ₃ –H ₂ O
Fresnel / Solar Tower	180–250 °C	Triple-effect absorption systems

The collector type significantly affects system sizing, efficiency, and capital cost. Hence, working pair and collector technology must be co-optimized for performance.

1.14 Challenges in Working Pair Selection [20]

- Crystallization: LiBr can precipitate if the 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: LiNO_3 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.

1.15 Emerging Trends and Innovations [18,19]

I. Nanofluid Additives:

- Adding nanoparticles (e.g., Al_2O_3 , 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 $\text{NH}_3\text{-NaSCN}$ 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.

1.16 Detailed Analysis of Working Pairs

The selection of an optimal refrigerant–absorbent working pair is a cornerstone of efficient vapor absorption cooling system design. While Section 3 discussed general selection criteria, this section delves deeper into the mechanisms, operating conditions, performance characteristics, and application scope of the four most significant working pairs:

- Water–Lithium Bromide ($\text{H}_2\text{O-LiBr}$)

- Ammonia–Water ($\text{NH}_3\text{--H}_2\text{O}$)
- Ammonia–Lithium Nitrate ($\text{NH}_3\text{--LiNO}_3$)
- Water–Lithium Chloride ($\text{H}_2\text{O--LiCl}$)
- Each pair is evaluated under the following framework:
- Thermodynamic behavior
- Temperature and pressure range
- COP and system performance
- Crystallization and corrosion risks
- Environmental and safety considerations
- Application domains
- Research advances and future scope

1. Water–Lithium Bromide ($\text{H}_2\text{O--LiBr}$) [8,9,11,12]

➤ Thermodynamic Behavior

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.

➤ Operating Conditions

- I. Evaporator temperature: 5–10 °C
- II. Generator temperature:
- III. 80–100 °C (single-effect)
- IV. 140–170 °C (double-effect)

- COP range: 0.65–0.8 (single), 1.1–1.3 (double)

The LiBr solution becomes highly concentrated in the absorber, where it absorbs water vapor.

Heat applied in the generator drives the desorption process.

➤ Advantages

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

➤ Limitations

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

➤ Applications

- I. Centralized air conditioning
- II. Solar cooling in commercial/institutional buildings
- III. HVAC in hospitals, airports, malls

➤ Research Advancements

- I. Crystallization modeling to develop real-time anti-crystallization controls
- II. Use of magnetite nanoparticles (Fe_3O_4) to improve solution thermal conductivity
- III. Development of low-cost ETC-compatible single-effect chillers for hot and humid climates

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

➤ Thermodynamic Behavior

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 $-20\text{ }^\circ\text{C}$), making it suitable for cold chains and ice-making applications.

➤ Operating Conditions

- I. Evaporator temp: -20 to $+10\text{ }^\circ\text{C}$
- II. Generator temp: $120\text{--}160\text{ }^\circ\text{C}$
- III. Operating pressure: $8\text{--}24\text{ bar}$
- IV. COP: $0.5\text{--}0.75$
- V. High system pressures and the volatility of both substances demand robust and leak-proof construction.

➤ Advantages

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

➤ Limitations

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

➤ Applications

- I. Cold storages, dairy, food processing
- II. Solar-assisted freezing units
- III. Absorption ice machines

➤ Research Advancements

- I. Use of $\text{NH}_3\text{--H}_2\text{O}$ systems with PTC-driven generators
- II. Optimized plate heat exchangers to minimize charge quantity
- III. Dynamic simulation models for rural milk cooling applications

3. Ammonia–Lithium Nitrate ($\text{NH}_3\text{--LiNO}_3$) [11,15]

➤ Thermodynamic Behavior

This pair is non-aqueous, meaning no rectifier is required. $\text{NH}_3\text{--LiNO}_3$ 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.

➤ Operating Conditions

- I. Evaporator temp: -5 to $+5$ °C
- II. Generator temp: $100\text{--}150$ °C
- III. Pressure: $8\text{--}15$ bar
- IV. COP: $0.6\text{--}1.0$

➤ Advantages

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

➤ Limitations

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

➤ Applications

- I. Decentralized rural cooling
- II. Vaccine preservation in off-grid areas
- III. Experimental hybrid chillers

➤ Research Advancements

- I. Integration with low-temperature CPC collectors
- II. Addition of nanofluids (e.g., Al_2O_3 , CuO) to lower viscosity
- III. Control-oriented simulation for portable cooling systems

4. Water–Lithium Chloride (H_2O – LiCl)[16]

➤ Thermodynamic Behavior

H_2O – 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.

➤ Operating Conditions

- I. Evaporator temp: 8–14 °C
- II. Generator temp: 85–105 °C
- III. Vacuum operation
- IV. COP: 0.5–0.7

➤ Advantages

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

➤ Limitations

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

➤ Applications

- I. HVAC systems in coastal zones

- II. Hospitals, archives, museums
- III. Hybrid desiccant-cooling systems

➤ Research Advancements

- I. Modeling of LiCl-based desiccant-integrated VAS
- II. Use in indirect evaporative cooling
- III. Application in net-zero energy buildings

Table 1.6: Comparison Summary of different working pair [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.

I. 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.

II. 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.

III. 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.

IV. 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, Gkountas 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 by X% (not specified) 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 (Year)	System Details (Type, Collector, Chiller, Capacity)	Installed Location / Scenario	Key Findings
1	Nakahara et al. (1977)	H ₂ O-LiBr; flat plate collector (32.2 m ²), solar + gas; single effect (7 kW + 14 kW auxiliary)	Ohbayashi-Gumi, Tokyo; 1 year study	Collected fundamental data on collectors, storage tank, backup, controls; COP 0.4–0.8 at 70–100°C; no economic analysis.
2	Li & Sumathy (2001)	H ₂ O-LiBr; flat plate collector (38 m ²); single effect (4.7 kW)	University of Hong Kong; residential AC; 4 months	Partitioned storage tank had superior COP under variable conditions versus single tank.
3	Rosiek et al. (2009)	H ₂ O-LiBr; flat plate collector (160 m ²); single effect (70 kW)	CIESOL, U. of Almeria; Mon–Fri, 9am–8pm	System COP was 0.6 during routine operation.
4	Bermejo et al. (2010)	H ₂ O-LiBr; linear Fresnel (352 m ²) + gas burner; double effect (174 kW)	Engineering School, Seville, Spain; 1 year, 5 days/week	Focused on collector area, dirt, climate, piping losses, control; COP 1.1–1.25.
5	Darkwa et al. (2012)	H ₂ O-LiBr; evacuated tube collectors (220	CSET, U. of Nottingham, Ningbo, China	COP 0.69 at T _{gen} 90°C; delivered chilled water at 7°C.

		m ²); single effect (55 kW)		
6	Hang et al. (2014)	H ₂ O-LiBr; external compound parabolic (54 m ²); double effect (23 kW)	U. California, Merced; lab facility	Average COP ~1; operated with gas backup or hybrid load; validated dual operation strategy.
7	Z.Y. Xu et al. (2015)	H ₂ O-LiBr; variable effect (50 kW)	Shanghai Jiao Tong University	COP range 0.69–1.08 at 95–120°C; innovative cycle shifting between single/double effect.
8	Olivier Marc et al. (2015)	H ₂ O-LiBr; single effect (30 kW); only solar (no backup); Reunion Island	Input: 30 kW prototype, dynamic data	COP 0.74; no auxiliary backup used; performance for isolated solar operation.
9	Buonomano et al. (2016)	H ₂ O-LiBr; evacuated flat plate (52.5 m ²); double effect (23 kW)	Saudi Arabia; 6 months experimental	EFPC outperformed PTC in cost and technical feasibility; compared both collectors.
10	J.F. Chen et al. (2018)	H ₂ O-LiBr; air-cooled; single effect (5 kW); adiabatic flash + absorber	Residential application	COP 0.72 from 9-hour outdoor test; examined novel air-cooled, adiabatic design.

11	Atmaca & Yigit (2003)	H ₂ O-LiBr; various collectors, single effect (10.5 kW); simulation (EES)	Antalya, Turkey	Analyzed effect of hot water temp, collector type, tank mass; COP 0.7–0.78 at Thw 80–90°C; optimum area = 50 m ² .
12	Bellos et al. (2016)	H ₂ O-LiBr; various collector types; single effect (100 kW); simulation (EES)	Athens, Greece; steady-state model	Compared collectors; max COP 0.8072 at 106°C, 10°C evaporator; ETC best financially.
13	Balghouthi et al. (2008)	H ₂ O-LiBr; flat plate collector, single effect (11 kW) + 18 kW backup; EES & TRNSYS	Tunis, Tunisia	COP 0.72 at 86°C/4°C (gen/evap); proven suitable for regional climate.
14	Gomri et al. (2013)	H ₂ O-LiBr; flat plate collector; single effect (10 kW) + gas backup; Fortran simulation	Constantine, Algeria	Max COP 0.82 at 5°C evaporator; T _{gen} 54–83°C, T _{cond} 28–36°C; auxiliary CO ₂ emissions <3kg/h.
15	Mazloumi et al. (2008)	H ₂ O-LiBr; parabolic trough collector; single effect (5 TR); simulation	Ahwaz, Iran	COP 0.67–0.78; generator 70–95°C, water-cooled absorber/condenser.

16	Qu et al. (2010)	H ₂ O-LiBr; linear parabolic troughs; double effect (16 kW); TRNSYS	CMU, Pittsburgh	Optimized orientation/piping/storage; delivered 39% cooling, 20% heating demand, annual 135 kWh/m ² savings.
17	Li et al. (2014)	H ₂ O-LiBr; compound parabolic (27 m ²); double effect (20 kW), air-cooled	Guangzhou, China	Identified monthly optimum collector temps; storage raises optimum; optimal temp up with T _a , T _c , down with T _e .
18	Devarajan & Kumaraguruparan (2024)	Solar PV + PCM cooling; experiment + simulation	PV system efficiency enhancement	PV efficiency improved by 10–15%; panel temps down by 20°C; cost up 15%, PCM cycle life limited; optimization needed.
19	Khan et al. (2023)	Water spray cooling on PV panels; efficiency trial	PV cooling, performance testing	Panel temp fell 25°C, electric efficiency up 12%, increased water maintenance by 10%; calls for smart integration.

20	Khan & Mishra (2021)	Combined cycle, low-GWP fluids, solar thermal generation, economic/life-cycle focus	Solar thermal power, environmental evaluation	Cycle efficiency 38%, GWP cut 90%, LCOE reduced 8%; higher capital/complexity; recommends fluid/cycle optimization.
21	Geneiting et al. (2023)	Probabilistic solar irradiance forecasting models for grid	Solar grid management, forecasting	Forecast accuracy improved 15%, uncertainty down 20%; computational cost; AI/physics hybrids recommended.
22	Khan, Mishra & Singh (2024)	Solar + ORC waste heat recovery cycles; efficiency and CO ₂ reduction analysis	Solar combined cycle performance	Max cycle efficiency 35%, exergy destruction cut 12%, CO ₂ emission reduction; system is complex, high investment.
23	Al-Sulaiman (2014)	Exergy analysis, solar collector + combined cycle; thermal plant focus	Exergy losses in solar plants	Collector efficiency 75%, exergy losses cut by 10%, total cycle efficiency 40%; scale-up remains a challenge.

24	Zhang et al. (2022)	Deep learning for power system frequency control, grid optimization	Smart grid, power system automation	DL improved frequency accuracy by 18%, faster response; relies on data, interpretability issues for models; IoT needed.
25	Li et al. (2022)	Probabilistic modeling, integrated regional energy flows	Regional energy management, prediction	Prediction accuracy up 20%, can analyze failures; demanding computations and data; application in real-time control.
26	Du, Wang & Zhang (2021)	Impeller back seal on sCO ₂ turbine performance	Power generation, SC-CO ₂ turbines	Turbine efficiency up 5%, leakage loss down 30%; complex manufacturing; optimization for commercial turbines needed.
27	Khan, Caliskan & Hong (2023)	CSP combined cycles with eco-friendly fluids; cycle comparison	Solar thermal power, CSP evaluation	Cycle efficiency up to 42%, GWP reduced 85%, costs saved; higher complexity/maintenance, needs better fluids, exchangers.

28	Gkoutas et al. (2020)	Nanofluids in sCO ₂ Brayton cycle heat exchangers; experimental enhancement	Power cycles, sCO ₂ performance	15% higher heat transfer, less resistance, stability/erosion issues; recommends long-term studies on nanofluids.
29	Chai & Tassou (2020)	Heat exchanger technologies for advanced Brayton cycles	High-efficiency power cycle design	Designs yield >80% heat transfer, low pressure drop; cost/fabrication remain difficult; novel methods needed.
30	Al-Sulaiman & Atif (2015)	Different sCO ₂ Brayton cycles for solar tower analysis	Solar thermal generation, sCO ₂ focus	Cycle efficiency up to 44%, input temps ~600°C; high-T/P material challenges; improved integration needed.
31	Osorio et al. (2016)	Transient analysis of sCO ₂ cycles in CSP towers	CSP transient control, stability	Startup 30% faster, dynamic operating stability improved; complex control is a challenge; advanced controllers advised.

32	Zare & Hasanzadeh (2016)	Energy/exergy study, Brayton-based combined solar cycles	Solar-thermal, thermodynamic optimization	Exergy up to 50%, thermal efficiency 42%; heat/cycle irreversibilities limit gains; receiver/cycle improvements suggested.
33	Miao et al. (2022)	Optimizing supercritical N ₂ O + helium Brayton cycles for space power	Space nuclear power, high-temp Brayton	Efficiency 48% at 700°C; limited experimental data; experimental verification/integration recommended.
34	Khan et al. (2024)	Hybrid solar Brayton-CO ₂ cycles; energy-environmental performance	Renewable energy generation	Efficiency 46%, CO ₂ emissions cut 60%; calls for capital cost reduction, storage integration.
35	Malik, Zheng & Lin (2019)	Compressor design for nuclear Brayton cycles; performance study	Nuclear/BPCS cycles, compressor engineering	88% efficiency, pressure ratio 3.5; complex gas mixing; needs advanced design/optimization.

36	Zhou et al. (2023)	Solar tower plant optimization: Brayton cycle & cooling tech	Solar thermals, LCOE/efficiency optimization	LCOE cut 12%, efficiency up 5%; more water used; dry/hybrid cooling tech needed.
37	Zoghi et al. (2021)	Solar hybrid trigeneration: economic, environmental assessment	Cooling, heating, power generations, hybrid	Exergy efficiency 55%, CO ₂ emissions down 40%, cost reduced 15%; complexity/operation cost issues; smart controls, storage.
38	Mohtaram et al. (2023)	Efficient ejector-expansion, absorption system for refrigeration	Air conditioning, refrigeration, energy recovery	COP raised by 12%, exergy destroyed down 8%; high capital cost; improvement via cost-effective designs/control.
39	Mishra & Singh (2018)	Combined solar power-cooling systems: parameter analysis	Integrated solar-thermal/cooling assessment	1.5 TR capacity; system efficiency 38%; complex integration; better control/storage inclusion advised.
40	Xu et al. (2011)	Energy, exergy analysis of CSP tower plants	Solar towers, CSP plant assessment	Thermal efficiency 40%, exergy 38%; heat loss in receiver/storage;

				points to better receivers/materials.
41	Farges et al. (2018)	Optimizing solar tower design using global algorithms	Solar-thermal plant cost/efficiency assessment	Efficiency gains 4%, cost lowered 7%; high computational cost; concept extendible to hybrid plants.
42	Sun, Wang & Xie (2021)	Cycles for industrial waste heat recovery	Waste heat recovery, power plants, economic analysis	42% cycle efficiency, 10% lower cost; restrictions: capital needed; propose straight industrial integration.
43	Zare, Mahmoudi & Yari (2015)	Economic viability of Kalina cycle for turbine heat recovery	Gas turbines, Kalina cycle economics	Exergy efficiency 55%, cost down 8%; Kalina cycle complexity a challenge; simpler/hybrid aimed for future.
44	Kerme et al. (2017)	LiBr-water absorption chillers, solar-driven; simulation and performance	Solar cooling simulation, building integration	COP 0.7, exergy efficiency 35%; large collector area; recommends HVAC + storage integration.

45	Mishra, Arora & Arora (2024)	Combined cycle power + absorption cooling sustainability analysis	Plant exergy and CO ₂ performance, hybrid	Exergy efficiency 52%, CO ₂ down 35%; complex, costly; recommends hybrid use and optimization.
46	Yadav, Sinha & Kumar (2023)	SOFC-GT hybrid performance & operation benchmarking	SOFC-GT, electrical eff. 50–60%, waste heat +10% overall, 700–900°C	Demonstrates hybrid-generation boost, tracks optimal operation.
47	Siqueira, Oechsler & Catapan (2024)	SOFC & ethanol reformer heat exchanger network design	Hydrogen supply/fuel cell integration	Heat recovery up 15%; exchanger 1–3 bar, SOFC at 800°C; improved hydrogen production noted.
48	Wang et al. (2021)	Optimization: atmospheric/pressurized SOFC trigeneration for CCHP	Pressurized SOFC: elec. eff. 68.1%, energy 68.8%, exergy 65.7%; \$19.84/h, 3 bar	Demonstrates economic and thermodynamic viability of pressurized trigeneration.
49	Chitgar & Emadi (2021)	SOFC-GT multi-gen: electricity, freshwater, hydrogen	Residential, multi-gen demonstration	Energy eff. >55%, exergy 50%, 800–850°C SOFC, GT

				1000°C; ROI ~7 yrs, proven for home use.
50	Roushenas, Zarei & Torabi (2021)	SOFC-GT trigeneration with compressed air and thermal storage	Flexible power/cooling generation, hybrid storage	Energy utilization 70%, storage boosts dispatch, exergy loss down 12% vs. non-storage.
51	Gholamian & Zare (2016)	Kalina vs ORC for SOFC waste heat conversion	Compare SOFC-GT-ORC/Kalina, temp diff. 120–200°C	SOFC-GT-ORC exergy 62.35%, Kalina 59.53%; ORC environmentally better, Kalina more complex.
52	Nanadegani & Sunden (2023)	Thermodynamic review of fuel cells: exergy, energy, key loss points	Fuel cell system analysis & application	Exergy eff. 40–65%, major destruction in fuel processing (25%); guides selection, optimization in hybrid systems.
53	Pirkandi, Penhani & Maroufi (2020)	Hybrid systems: steam & gas turbines + SOFC; system efficiency study	Hybrid power, SOFC-GT-ST integration	Efficiency 65–70%, SOFC at 800–900°C; integration drops fuel use 10%.
54	Stasiak, Ziołkowski & Mikielewicz (2024)	ORC with CCS from sewage sludge bioenergy, hybrid plant	CCS-bioenergy, renewable with	ORC eff. 18–22%; CCS improves sustainability; 3–5 MPa, temp diff

			waste carbon handling	120–160°C; renewables integrated.
55	Zhang et al. (2024)	Trigeneration, residential: system COP (1.2–1.5), heating 85%, electric eff. 40%	Residential supply; optimized for cost/energy	Payback 6–8 yrs, heat exchangers 85–90°C, peak design for home/utility use.
56	Eisavia, Chitsaz, Hosseinpour & Ranjbar (2018)	SOFC-GT system arrangements: efficiency, emission & cost	SOFC-GT hybrids, emission/cost assessment	Efficiency up to 62%, emission reduction 40%, lifecycle cost beats GT only.
57	Khan & Singh (2024)	SOFC hybrid CCHP; system performance (efficiency, exergy, cooling COP)	Power, cooling, heat co-production	Total system efficiency 70%, COP (cooling) 1.2, exergy losses down 15%, pressure 2–3 bar; advanced CCHP configuration.
58	Yadav, Sinha & Kumar (2024)	Hybrid SOFC-GT- ORC with cost/efficiency assessment	Power and heat generation, hybrid evaluation	Electrical eff. 65%, thermal 70%; cost down 8% by waste heat, SOFC 850°C, GT 1100°C.
59	Shirazi, Taylor, Morrison & White (2017)	Optimized solar absorption chillers for air conditioning,	Solar cooling for buildings;	COP 0.7–0.9, collector efficiency up to 55%; optimal designs at 80–

		performance efficiency	air- conditioning	95°C, cost reduction by 10%.
60	Emadi, Chitgar, Oyewunmi & Markides (2020)	Combined ORC- SOFC waste heat recovery, working fluid selection, optimization	SOFC waste heat utilization, system optimization	Optimized fluid lifts eff. by 12%, total exergy eff. 68%, 3–5 MPa, 100–150°C diff; best fluids for hybrid cycles.
61	Chen, Dai & Wang (2017)	Air-cooled LiBr-H ₂ O absorption chiller (3.5 kW) powered by ETC	Solar residential cooling	COP 0.7, collector eff. 60%, operation at 90– 120°C, enables low- power space cooling with high fraction solar input.

CHAPTER 3: PROBLEM FORMULATION

3.1 Research Gap

70 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.

78 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.

Identified Research Gaps:

- 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.
- 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.
- 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.

- 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.

In light of these gaps, the present work not only aims to simulate and analyze an SOFC-GT-VARS system but also seeks to explore hybrid configurations involving solar-powered Brayton cycles with HRSG and VACS subsystems. The ultimate goal is to contribute toward the development of next-generation distributed energy solutions that are efficient, clean, and adaptable to future energy needs.

3.2 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.

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.

16 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 thermal energy storage (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.

3 The HRSG produces medium-temperature steam (~200°C) at high pressure (~2000 kPa), which is suitable for various industrial process heat applications.

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 8 for simulation accuracy.

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:

$$\dot{Q}_{CV} - \dot{W}_{CV} + \sum(\dot{m}_i h_i) - \sum(\dot{m}_e h_e) = 0 \quad (1)$$

$$\dot{E}D = \dot{E}X_{in} - \dot{E}X_{out} \quad (2)$$

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:

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

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.

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.

$$\dot{E}X_j = \dot{m}[(h_j - h_0) - T_0(s_j - s_0)] \quad (3)$$

The subscript 'j' denotes a specific thermodynamic state, while refers to $\dot{E}X_j$ 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].

$$\dot{Q}_{\text{rec,in}} = \eta_{\text{field}} \cdot \dot{Q}_{\text{Sun}} = \eta_{\text{field}} \cdot \text{DNI} \cdot A_{\text{hel}} \cdot N_{\text{hel}} \quad (4)$$

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

$$\eta_{\text{field}} = \eta_{\text{cos}} \cdot \eta_{\text{s\&b}} \cdot \eta_{\text{int}} \cdot \eta_{\text{att}} \cdot \eta_{\text{ref}} \quad (5)$$

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

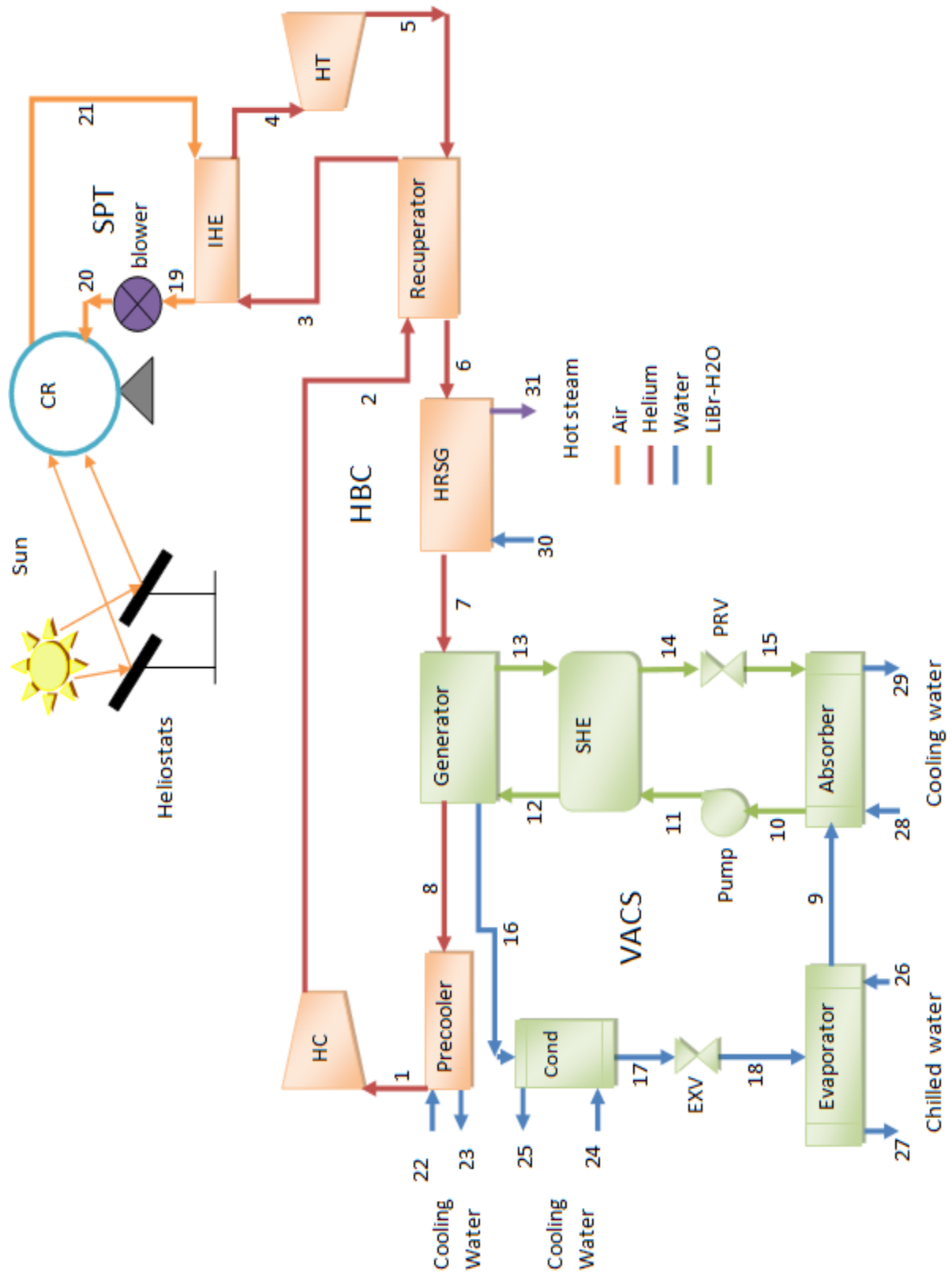


Figure 4.1: Proposed SPT based trigeneration system

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];

$$\eta_{rec} = \frac{\dot{Q}_{rec,net}}{\dot{Q}_{rec,in}} \quad (6)$$

$$\dot{Q}_{rec,in} = \dot{Q}_{rec,net} + \dot{Q}_{rec,loss} = \dot{m}_{air}(h_{30} - h_{31}) + \dot{Q}_{rec,loss} \quad (7)$$

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.

4.1.3 System parameters

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

$$\dot{W}_{net} = \dot{W}_{HT} - \dot{W}_{HC} \quad (8)$$

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

$$COP_{VACS} = \frac{\dot{Q}_e}{\dot{Q}_g} \quad (9)$$

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

$$\eta_{en,HBC} = \frac{\dot{W}_{net}}{\dot{Q}_{Sun}} \quad (10)$$

$$\eta_{en,co,h} = \frac{\dot{W}_{net} + \dot{Q}_{HRSG}}{\dot{Q}_{Sun}} \quad (11)$$

$$\eta_{en,co,c} = \frac{\dot{W}_{net} + \dot{Q}_e}{\dot{Q}_{Sun}} \quad (12)$$

$$\eta_{en,tri} = \frac{\dot{W}_{net} + \dot{Q}_{HRSG} + \dot{Q}_e}{\dot{Q}_{Sun}} \quad (13)$$

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

$$\eta_{ex,HBC} = \frac{\dot{W}_{net}}{\dot{Q}_{Sun} \cdot \left(1 - \frac{T_0}{T_{ref,Sun}}\right)} \quad (14)$$

$$\eta_{ex,co,h} = \frac{\dot{W}_{net} + \dot{E}X_{HRSG}}{\dot{Q}_{Sun} \cdot \left(1 - \frac{T_0}{T_{ref,Sun}}\right)} \quad (15)$$

$$\eta_{ex,co,c} = \frac{\dot{W}_{net} + \dot{E}X_e}{\dot{Q}_{Sun} \cdot \left(1 - \frac{T_0}{T_{ref,Sun}}\right)} \quad (16)$$

$$\eta_{ex,tri} = \frac{\dot{W}_{net} + \dot{E}X_{HRSG} + \dot{E}X_e}{\dot{Q}_{Sun} \cdot \left(1 - \frac{T_0}{T_{ref,Sun}}\right)} \quad (17)$$

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: Equations for Each Component of the Proposed Solar Integrated Power and Cooling System

Component	Energy balance equation	Exergy balance equation
Heliostat field	$\dot{Q}_{\text{rec,in}} = \eta_{\text{field}} \cdot \text{DNI} \cdot A_{\text{hel}} \cdot N_{\text{hel}}$	$\dot{Q}_{\text{Sun}} \cdot \left(1 - \frac{T_0}{T_{\text{ref,Sun}}} \right)$ $= \dot{Q}_{\text{rec,in}} \cdot \left(1 - \frac{T_0}{T_{\text{ref,hel}}} \right)$ $+ \dot{E}D_{\text{hel}}$
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	$\dot{Q}_{\text{HRSG}} = \dot{m}_6 (h_6 - h_7)$ $= \dot{m}_{30} (h_{31} - h_{30})$ $\Delta T_{\min} = T_6 - T_{31}$	$\dot{E}X_6 - \dot{E}X_7 = \dot{E}X_{31} - \dot{E}X_{30}$ $+ \dot{E}D_{\text{HRSG}}$
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	$(h_3 - h_2) = (h_5 - h_6)$ $\epsilon_{\text{Recuperator}} = \frac{(T_3 - T_2)}{(T_5 - T_2)}$	$\dot{E}X_5 - \dot{E}X_6 = \dot{E}X_3 - \dot{E}X_2$ $+ \dot{E}D_{\text{Recuperator}}$
Generator	$\dot{Q}_g = \dot{m}_{16} h_{16} + \dot{m}_{13} h_{13} - \dot{m}_{12} h_{12}$ $\dot{m}_{12} = \dot{m}_{13} + \dot{m}_{16}$	$\dot{E}X_{13} + \dot{Q}_g \left(1 - \frac{T_0}{T_g}\right)$ $= \dot{E}X_{16} + \dot{E}X_{13}$ $+ \dot{E}D_g$
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	$\dot{Q}_a = \dot{m}_9 h_9 + \dot{m}_{15} h_{15} - \dot{m}_{10} h_{10}$ $\dot{m}_{10} = \dot{m}_9 + \dot{m}_{15}$ $x_{13} \dot{m}_{15} = x_{10} \dot{m}_{10}$	$\dot{E}X_{10} + \dot{Q}_a \left(1 - \frac{T_0}{T_a}\right) + \dot{E}D_g$ $= \dot{E}X_9 + \dot{E}X_{15}$
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	$\dot{m}_{13} \cdot h_{13} - \dot{m}_{14} \cdot h_{14}$ $= \dot{m}_{12} \cdot h_{12}$ $- \dot{m}_{11} \cdot h_{11}$ $\epsilon_{SHE} = \frac{(T_{13} - T_{14})}{(T_{13} - T_{11})}$	$\dot{E}X_{13} + \dot{E}X_{11} = \dot{E}X_{12} + \dot{E}X_{14}$ $+ \dot{E}D_{SHE}$
Pressure reducing valve	$h_{14} = h_{15}$	$\dot{E}X_{14} - \dot{E}X_{15} = \dot{E}D_{PRV}$
Evaporator	$\dot{Q}_e = \dot{m}_9 (h_9 - h_{18}) = \dot{m}_{\text{chilled water}} (h_{26} - h_{27})$	$\dot{E}X_9 - \dot{E}X_{18} = \dot{Q}_e \cdot \left(1 - \frac{T_0}{T_e}\right)$ $+ \dot{E}D_e$

4.2 SOFC -GT-VACS

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.

18 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]:

- 65
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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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]:

$$\sum(\dot{n}_i) - \sum(\dot{n}_e) = 0 \quad (1)$$

$$\dot{Q}_{CV} - \dot{W}_{CV} + \sum(\dot{n}_i \bar{h}_i) - \sum(\dot{n}_e \bar{h}_e) = 0 \quad (2)$$

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]:

$$\dot{X}_j = \dot{X}_{ph} + \dot{X}_{ch} + \dot{X}_{ke} + \dot{X}_{pe} \quad (3)$$

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

$$\dot{X}_{ph} = \dot{m} \cdot [(h_j - h_0) - T_0 \cdot (s_j - s_0)] \quad (4)$$

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

$$\dot{X}_{ch} = \dot{n}_t [\sum_i y_i e_{ch,i}^0 + \bar{R} T_0 \sum_i y_i \ln(y_i)] \quad (5)$$

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;

$$\dot{X}_j = \dot{X}_{ph} + \dot{X}_{ch} \quad (6)$$

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

$$\dot{E}D_k = \dot{X}_{in} - \dot{X}_{out} \quad (7)$$

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

$$\dot{X}_{input} = \dot{n}_{CH4,11} \cdot \bar{e}_{CH4}^{ch,0} \quad (8)$$

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	Energy balance equations	Exergy balance equations
SOFC STACK		
Fuel cell stack	$\dot{n}_{11} \cdot \bar{h}_{11} + \dot{n}_3 \cdot \bar{h}_3 = \dot{n}_{12} \cdot \bar{h}_{12} + \dot{n}_4 \cdot \bar{h}_4$	$\dot{X}_3 + \dot{X}_{11} = \dot{X}_4 + \dot{X}_3 + \dot{ED}_{SOFC}$
AB	$\dot{Q}_{loss,AB} = \dot{n}_{13} \cdot \bar{h}_{13} - (\dot{n}_{12} \cdot \bar{h}_{12} + \dot{n}_4 \cdot \bar{h}_4)$ $\eta_{AB} = \frac{\dot{n}_{12} \cdot \bar{h}_{12} + \dot{n}_4 \cdot \bar{h}_4}{\dot{n}_{13} \cdot \bar{h}_{13}}$	$\dot{X}_{12} + \dot{X}_4 = \dot{X}_{13} + \dot{ED}_{AB}$
AC	$\dot{W}_{ac} = \dot{n}_1 \cdot (\bar{h}_2 - \bar{h}_1)$ $\eta_{ac} = \frac{(\bar{h}_{2s} - \bar{h}_1)}{(\bar{h}_2 - \bar{h}_1)}$	$\dot{X}_1 = \dot{X}_2 - \dot{W}_{ac} + \dot{ED}_{ac}$
FC	$\dot{W}_{fc} = \dot{n}_5 \cdot (\bar{h}_6 - \bar{h}_5)$ $\eta_{ac} = \frac{(\bar{h}_{6s} - \bar{h}_5)}{(\bar{h}_6 - \bar{h}_5)}$	$\dot{X}_5 = \dot{X}_6 - \dot{W}_{fc} + \dot{ED}_{fc}$
WP	$\dot{W}_{wp} = \dot{n}_8 \cdot (\bar{h}_9 - \bar{h}_8)$ $\eta_{wp} = \frac{(\bar{h}_{9s} - \bar{h}_8)}{(\bar{h}_9 - \bar{h}_8)}$	$\dot{X}_8 = \dot{X}_9 - \dot{W}_{wp} + \dot{ED}_{wp}$
WHE	$\dot{n}_{13} \cdot (\bar{h}_{13} - \bar{h}_{14}) = \dot{n}_9 \cdot (\bar{h}_{10} - \bar{h}_9)$	$\dot{X}_{13} + \dot{X}_9 = \dot{X}_{14} + \dot{X}_{10} + \dot{ED}_{WHE}$

	$\epsilon_{WHE} = \frac{(T_{10} - T_9)}{(T_{13} - T_9)}$	
FHE	$\dot{n}_{13} \cdot (\bar{h}_{14} - \bar{h}_{15}) = \dot{n}_6 \cdot (\bar{h}_7 - \bar{h}_6)$ $\epsilon_{FHE} = \frac{(T_7 - T_6)}{(T_{14} - T_6)}$	$\dot{X}_{14} + \dot{X}_6 = \dot{X}_{15} + \dot{X}_7$ $+ \dot{ED}_{FHE}$
AHE	$\dot{n}_{13} \cdot (\bar{h}_{15} - \bar{h}_{16}) = \dot{n}_2 \cdot (\bar{h}_3 - \bar{h}_2)$ $\epsilon_{FHE} = \frac{(T_3 - T_2)}{(T_{15} - T_2)}$	$\dot{X}_{15} + \dot{X}_2 = \dot{X}_{16} + \dot{X}_3$ $+ \dot{ED}_{AHE}$
Mixer	$\dot{Q}_{loss} = \dot{n}_{11} \cdot \bar{h}_{11} - (\dot{n}_{10} \cdot \bar{h}_{10}$ $+ \dot{n}_7 \cdot \bar{h}_7)$	$\dot{X}_7 + \dot{X}_{10} = \dot{X}_{11} + \dot{ED}_{mixer}$
GT	$\dot{W}_{GT} = \dot{n}_{16} \cdot (\bar{h}_{16} - \bar{h}_{17})$ $\eta_{GT} = \frac{(\bar{h}_{16} - \bar{h}_{17})}{(\bar{h}_{16} - \bar{h}_{17s})}$	$\dot{X}_{16} = \dot{X}_{17} + \dot{W}_{GT} + \dot{ED}_{GT}$
WST	$\dot{n}_{17} \cdot (\bar{h}_{17} - \bar{h}_{18}) = \dot{n}_{29} \cdot (\bar{h}_{29} - \bar{h}_{31})$	$\dot{X}_{17} - \dot{X}_{18} = \dot{X}_{29} - \dot{X}_{31}$ $+ \dot{ED}_{WST}$
Pump-1	$\dot{W}_{pump1} = \dot{n}_{29} \cdot (\bar{h}_{31} - \bar{h}_{30})$ $\eta_{pump1} = \frac{(\bar{h}_{31s} - \bar{h}_{30})}{(\bar{h}_{31} - \bar{h}_{30})}$	$\dot{X}_{30} = \dot{X}_{31} - \dot{W}_{pump1}$ $+ \dot{ED}_{pump1}$
VARs		
Generator	$\dot{Q}_{GEN} = \dot{n}_{25} \cdot \bar{h}_{25} + \dot{n}_{22} \cdot \bar{h}_{22}$ $- \dot{n}_{21} \cdot \bar{h}_{21}$ $\dot{n}_{21} = \dot{n}_{22} + \dot{n}_{25}$	$\dot{X}_{21} + \dot{Q}_{GEN} \cdot \left(1 - \frac{T_0}{T_{GEN}}\right)$ $= \dot{X}_{25} + \dot{X}_{22}$ $+ \dot{ED}_{GEN}$

Condenser	$\dot{Q}_{COND} = \dot{n}_{25} \cdot (\bar{h}_{25} - \bar{h}_{26})$ $= \dot{n}_{36} \cdot (\bar{h}_{37} - \bar{h}_{36})$	$\dot{X}_{25} - \dot{X}_{26} = \dot{Q}_{COND} \cdot \left(1 - \frac{T_0}{T_{COND}} \right) + \dot{ED}_{COND}$
Expansion valve	$\bar{h}_{26} = \bar{h}_{27}$	$\dot{X}_{26} - \dot{X}_{27} = \dot{ED}_{EV}$
Absorber	$\dot{Q}_{ABS} = \dot{n}_{19} \cdot \bar{h}_{19} - \dot{n}_{24} \cdot \bar{h}_{24} - \dot{n}_{28} \cdot \bar{h}_{28}$ $\dot{n}_{19} = \dot{n}_{28} + \dot{n}_{24}$	$\dot{X}_{19} + \dot{Q}_{ABS} \cdot \left(1 - \frac{T_0}{T_{ABS}} \right) + \dot{ED}_{ABS} = \dot{X}_{24} + \dot{X}_{28}$
Pump-2	$\dot{W}_{pump2} = \dot{n}_{20} \cdot (\bar{h}_{20} - \bar{h}_{19})$ $\eta_{pump2} = \frac{(\bar{h}_{20s} - \bar{h}_{19})}{(\bar{h}_{20} - \bar{h}_{19})}$	$\dot{X}_{19} = \dot{X}_{20} - \dot{W}_{pump2} + \dot{ED}_{pump2}$
SHE	$\dot{n}_{22} \cdot \bar{h}_{22} - \dot{n}_{23} \cdot \bar{h}_{23}$ $= \dot{n}_{21} \cdot \bar{h}_{21} - \dot{n}_{20} \cdot \bar{h}_{20}$ $\varepsilon_{SHE} = \frac{(T_{21} - T_{20})}{(T_{22} - T_{20})}$	$\dot{X}_{22} + \dot{X}_{20} = \dot{X}_{21} + \dot{X}_{23} + \dot{ED}_{SHE}$
PRV	$\bar{h}_{23} = \bar{h}_{24}$	$\dot{X}_{24} - \dot{X}_{23} = \dot{ED}_{PRV}$
Evaporator	$\dot{Q}_{EVAP} = \dot{n}_{28} \cdot (\bar{h}_{28} - \bar{h}_{27})$ $= \dot{n}_{32} \cdot (\bar{h}_{32} - \bar{h}_{33})$	$\dot{X}_{33} - \dot{X}_{32} = \dot{Q}_{EVAP} \cdot \left(1 - \frac{T_0}{T_{EVAP}} \right) + \dot{ED}_{EVAP}$

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 (9) represents the steam reforming of methane to produce hydrogen, while Equation (10) 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 (11).

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

$$\ln K_s = \frac{\Delta \bar{g}_s^0}{\bar{R}T_{\text{FC,e}}} = \ln \left[\frac{y_r \cdot (3x_r + y_r - z_r)}{(x_r - y_r) \cdot (1.5x_r - y_r + z_r)} \right] \quad (12)$$

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];

$$U_f = \frac{\text{H}_2 \text{ consumed}}{\text{H}_2 \text{ supplied}} = \frac{z_r}{3x_r + y_r} \quad (13)$$

$$U_{\text{air}} = \frac{\text{O}_2 \text{ consumed}}{\text{O}_2 \text{ supplied}} = \frac{\frac{z_r}{2}}{\dot{n}_{\text{O}_2, \text{inlet}}} \quad (14)$$

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

$$\Delta \bar{g}_s^0 = \bar{g}_{\text{s,CO}_2}^0 + \bar{g}_{\text{s,H}_2}^0 - \bar{g}_{\text{s,H}_2\text{O}}^0 - \bar{g}_{\text{s,CO}}^0 \quad (15)$$

Where,

$$\bar{g}_s^0 = \bar{h} - T_{\text{FC,e}} \cdot \bar{s}^0 \quad (16)$$

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;

$$I = j \cdot A_a \quad (17)$$

$$j = \frac{2 \cdot F \cdot z_r}{N_{FC} \cdot A_a} \quad (18)$$

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

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

$$\dot{W}_{SOFC} = N_{FC} \cdot V_{FC} \cdot I \quad (19)$$

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

$$V_{FC} = V_N - V_{loss} \quad (20)$$

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>	$V_N = -\frac{\Delta \bar{g}^0}{nF} + \frac{\bar{R}T_{FC,12}}{nF} \cdot \ln \left(\frac{a_{H2,12} \sqrt{a_{O2,4}}}{a_{H2O,12}} \right); \quad a_{H2O,12} = \frac{P_{H2O,12}}{P_{ref}} =$ $\frac{y_{H2O,12} \cdot P_{12}}{P_{ref}}; \quad a_{H2,12} = \frac{P_{H2,12}}{P_{ref}} = \frac{y_{H2O,12} \cdot P_{12}}{P_{ref}}; \quad a_{O2,4} = \frac{P_{O2,4}}{P_{ref}} = \frac{y_{O2,4} \cdot P_4}{P_{ref}};$ $\Delta \bar{g}^0 = \bar{g}_{H2O}^0 - \bar{g}_{H2}^0 - \frac{1}{2} \bar{g}_{O2}^0;$ $\bar{g}^0 = \bar{h} - T_{FC,12} \cdot \bar{s}^0$
<i>Ohmic loss</i>	$V_{Ohm} = [R_c + \sum_i (\rho_i \cdot L_i)] \cdot j;$ $\rho_e = \left[3.34 \times 10^4 \cdot \exp \left(-\frac{10300}{T_{FC,e}} \right) \right]^{-1};$

	$\rho_a = \left[\frac{95 \times 10^6}{T_{FC,e}} \cdot \exp \left(-\frac{1150}{T_{FC,e}} \right) \right]^{-1};$ $\rho_c = \left[\frac{42 \times 10^6}{T_{FC,e}} \cdot \exp \left(-\frac{1200}{T_{FC,e}} \right) \right]^{-1};$ $\rho_{int} = \left[\frac{9.3 \times 10^6}{T_{FC,e}} \cdot \exp \left(-\frac{1100}{T_{FC,e}} \right) \right]^{-1};$
Activation polarization	$V_{act} = \frac{\bar{R}T_{FC}}{F} \cdot \sinh^{-1} \left(\frac{j}{2j_{oa}} \right) + \frac{\bar{R}T_{FC}}{F} \cdot \sinh^{-1} \left(\frac{j}{2j_{oc}} \right)$
Concentration loss	$V_{conc} = \frac{\bar{R}T_{FC}}{2F} \cdot \left[\ln \left(1 + \frac{P_{H_2} \cdot j}{P_{H_2O} \cdot j_{as}} \right) - \ln \left(1 - \frac{j}{j_{as}} \right) \right]$ $- \frac{\bar{R}T_{FC}}{4F} \ln \left(\frac{j}{j_{cs}} \right)$

4.2.5 Performance parameters

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

$$\dot{W}_{net,overall} = \dot{W}_{net,SOFC-GT} - \dot{W}_{pump1} - \dot{W}_{pump2} \quad (21)$$

Where,

$$\dot{W}_{net,SOFC-GT} = \dot{W}_{SOFC} + \dot{W}_{GT} - \dot{W}_{ac} - \dot{W}_{fc} - \dot{W}_{wp} \quad (22)$$

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

$$\eta_{en,SOFC-GT} = \frac{\dot{W}_{net,SOFC-GT}}{\dot{m}_{f,11} \cdot (LHV_{CH_4})} \quad (23)$$

$$\eta_{ex,SOFC-GT} = \frac{\dot{W}_{net,SOFC-GT}}{\dot{X}_{inlet}} \quad (24)$$

Where, LHV_{CH_4} 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];

$$\eta_{en,overall} = \frac{\dot{W}_{net,Plant} + \dot{Q}_{EVAP}}{\dot{m}_{f,11}(LHV_{CH_4})} \quad (25)$$

$$\eta_{ex,overall} = \frac{\dot{W}_{net,Plant} + \dot{Q}_{EVAP} \left(1 - \frac{T_0}{T_{EVAP}}\right)}{\dot{x}_{inlet}} \quad (26)$$

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

$$COP = \frac{\dot{Q}_{EVAP}}{(\dot{Q}_{GEN} + \dot{W}_{pump2})} \quad (27)$$

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]:

$$\dot{C}_{env} = 3600 \cdot \dot{m}_{CO_2} \cdot C_{pen} \quad (28)$$

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]:

$$EMI_{SOFC-GT} = 3600 \cdot \frac{\dot{m}_{CO_2}}{(\dot{W}_{net,SOFC-GT})} \quad (29)$$

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];

$$EMI_{Plant} = 3600 \cdot \frac{\dot{m}_{CO_2}}{(\dot{W}_{net,Plant} + \dot{Q}_{EVAP})} \quad (30)$$

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]:

$$\dot{C}_{total} = \dot{C}_f + \dot{C}_{env} + \dot{Z}_{total} \quad (31)$$

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

$$\dot{C}_f = (C_f \cdot \dot{m}_f \cdot LHV) \cdot 3600 \quad (32)$$

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

$$\dot{Z}_{total} = \sum_j \frac{Z_j \cdot CRF \cdot \varphi}{3600 \cdot N} \quad (33)$$

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];

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \quad (34)$$

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]:

$$UA = \frac{\dot{Q}}{LMTD} \quad (35)$$

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.

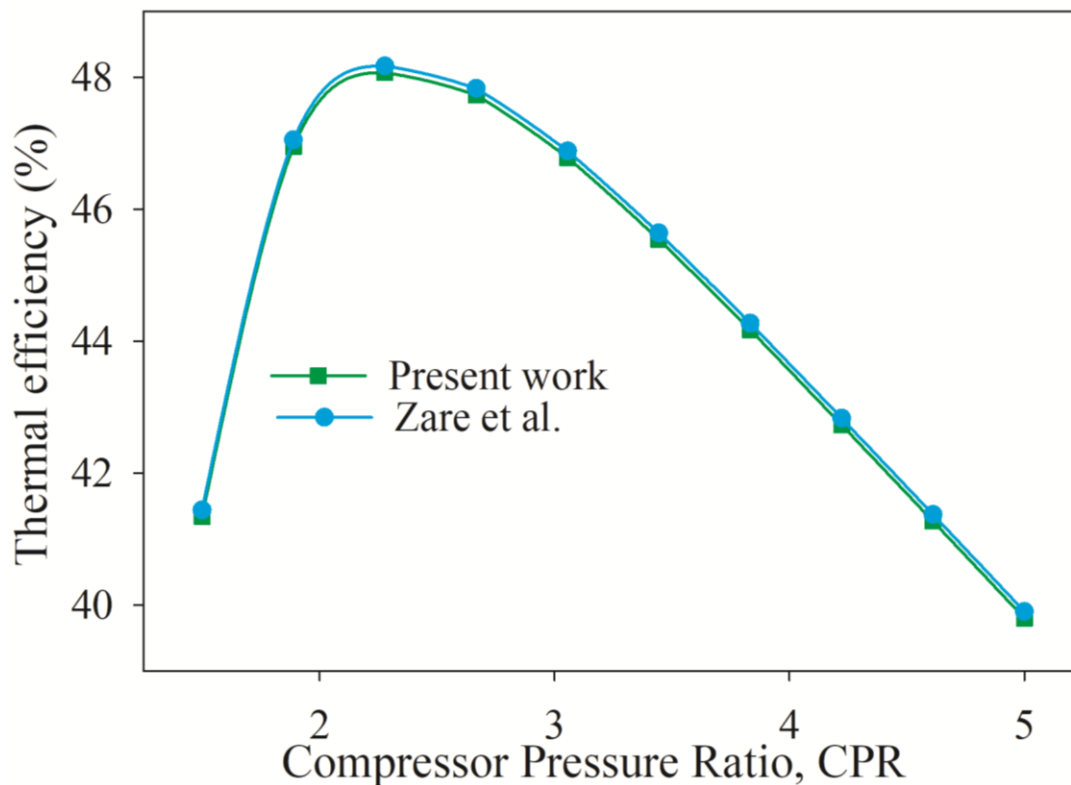


Figure 5.1: Comparison & Validation of HBC

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. 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. Excluding the SPT, the system's energy efficiency reaches a high value of 66.31 %. 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: Final result of the complete 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	

Table 5.3: 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.

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.

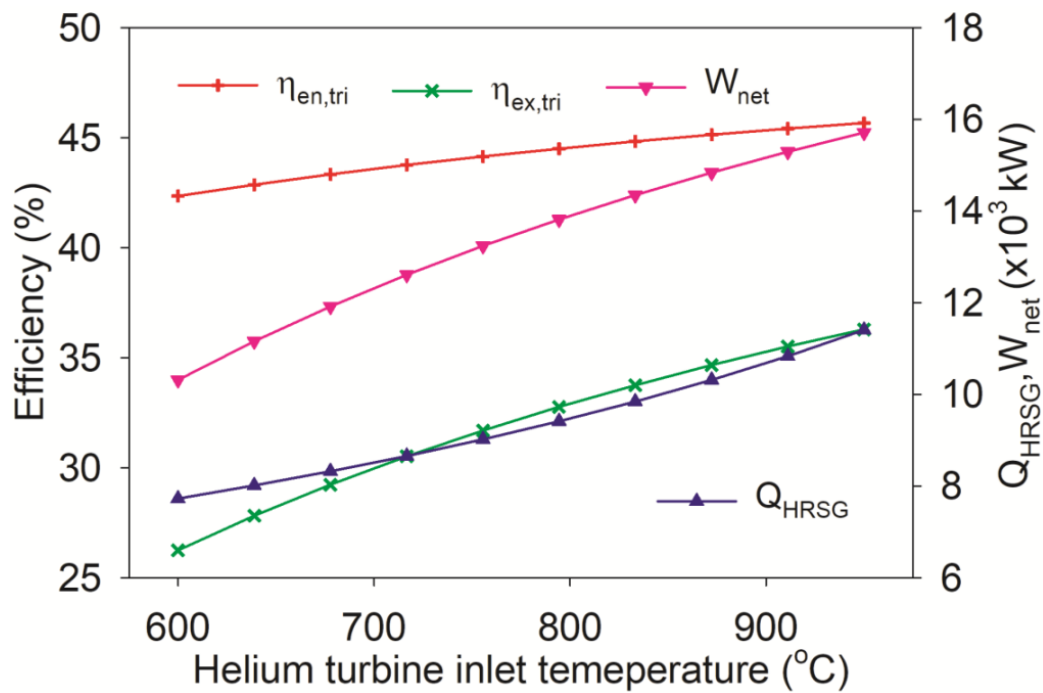


Figure 5.2: Trigeneration system performance variations with HTIT

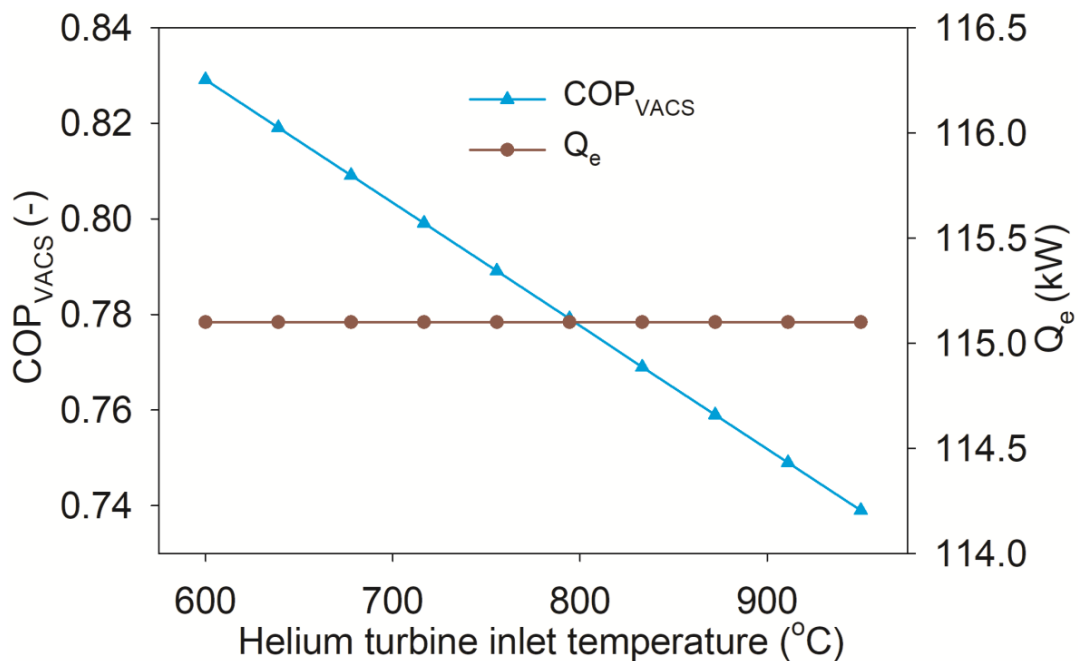


Figure 5.3: COP and cooling production variation with HTIT

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.

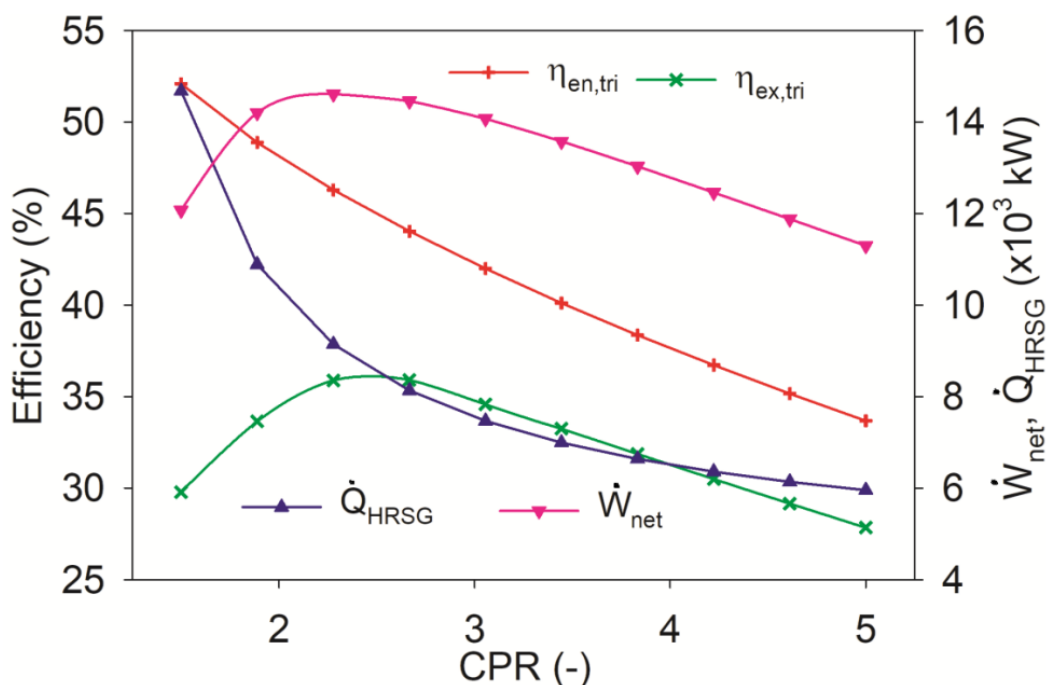


Figure 5.4: Trigeneration performances variation with CPR

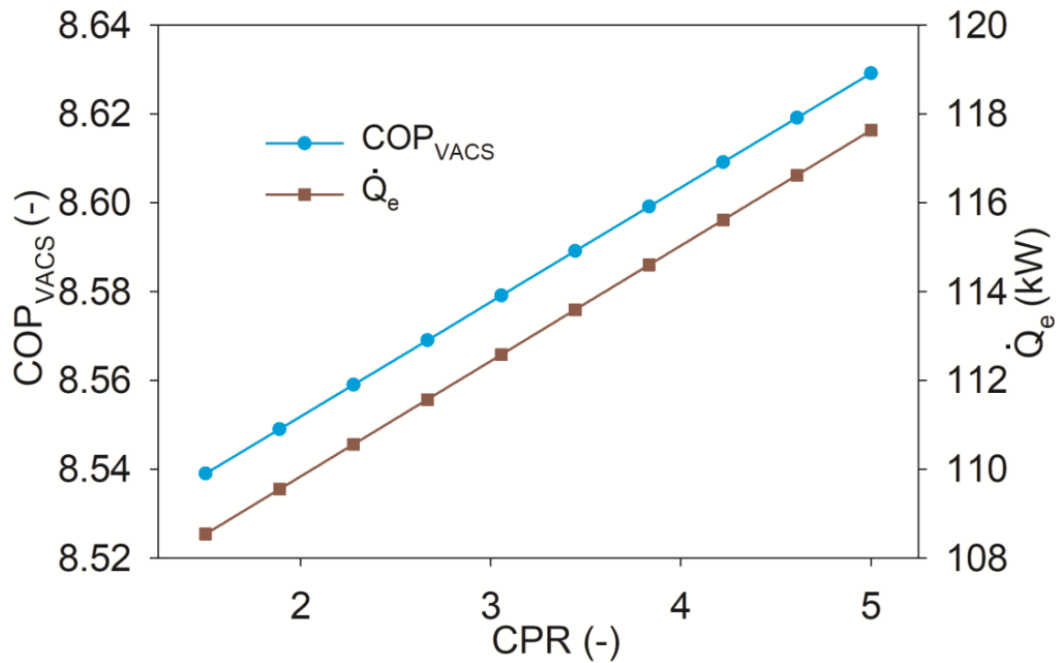


Figure 5.5: COP and cooling production variation with CPR

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.

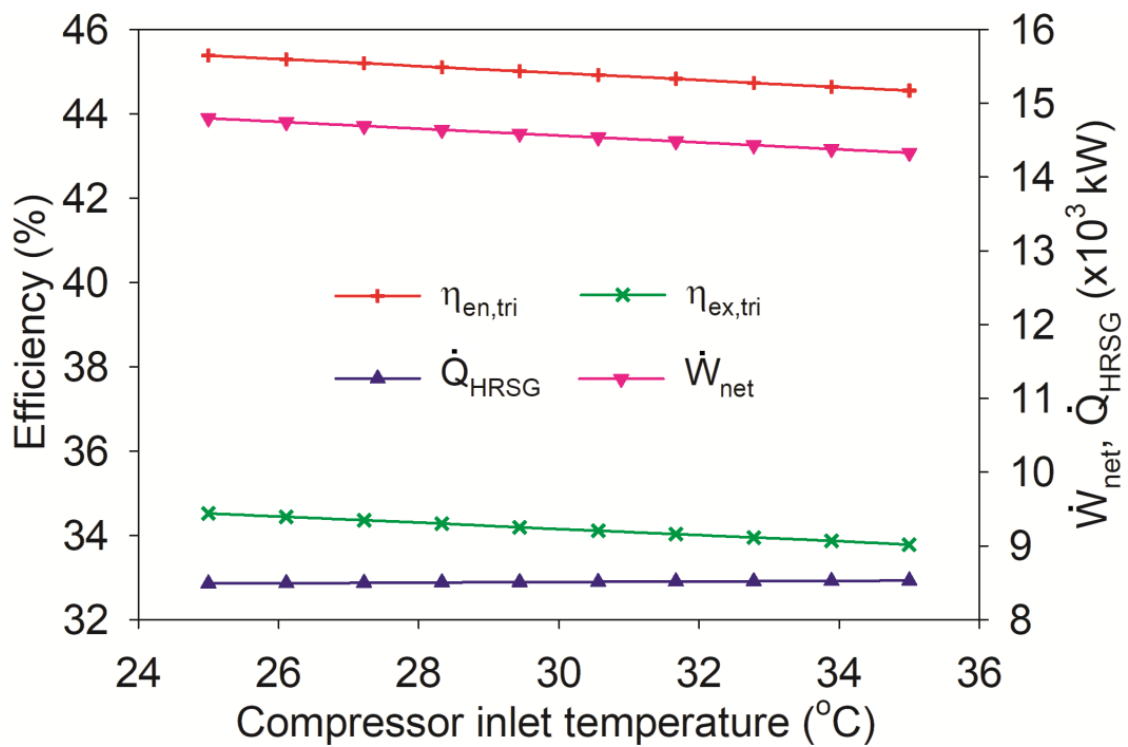


Figure 5.6: Trigeneration performance variations with the CIT

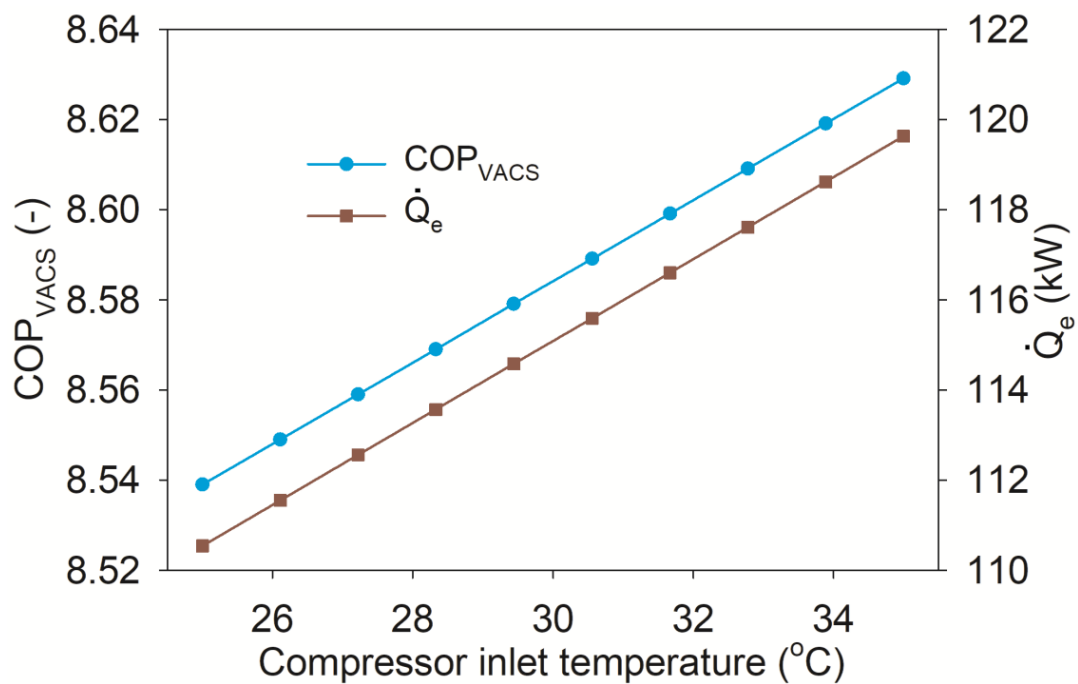


Figure 5.7: COP and cooling production variation with CIT

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.

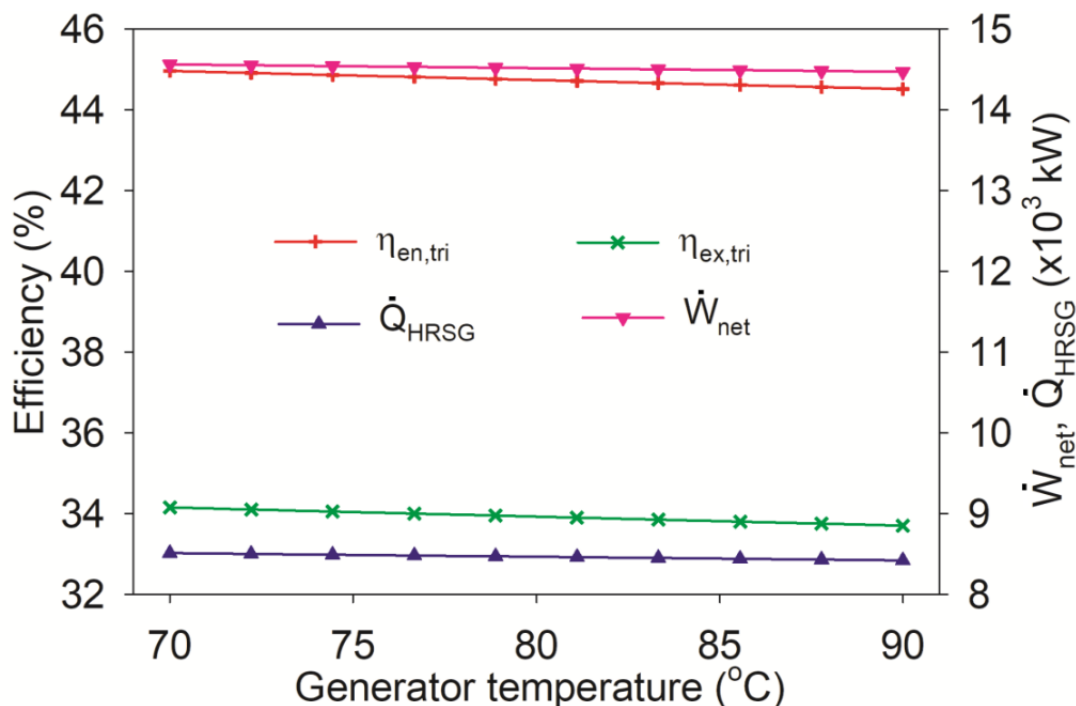


Figure 5.8: Trigeneration system performance variations with generator temperature

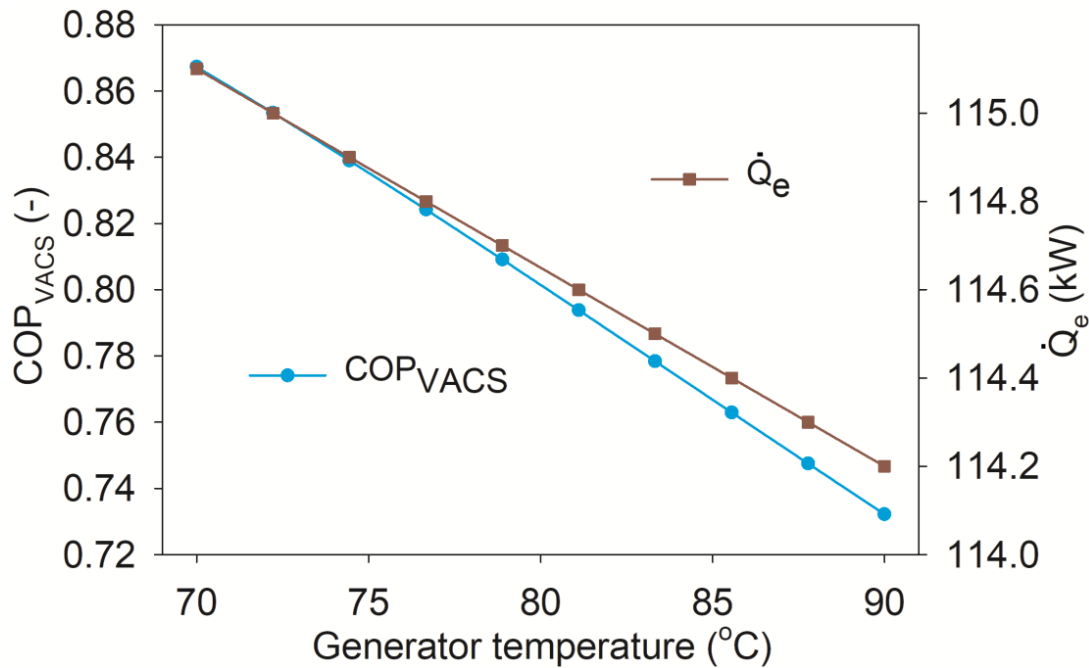


Figure 5.9: COP and cooling production variation with generator temperature

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/m² to 950 W/m² 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/m² to 950 W/m². Hence, while higher DNI improves power and heating outputs, it compromises the cooling performance and efficiency of the absorption cycle.

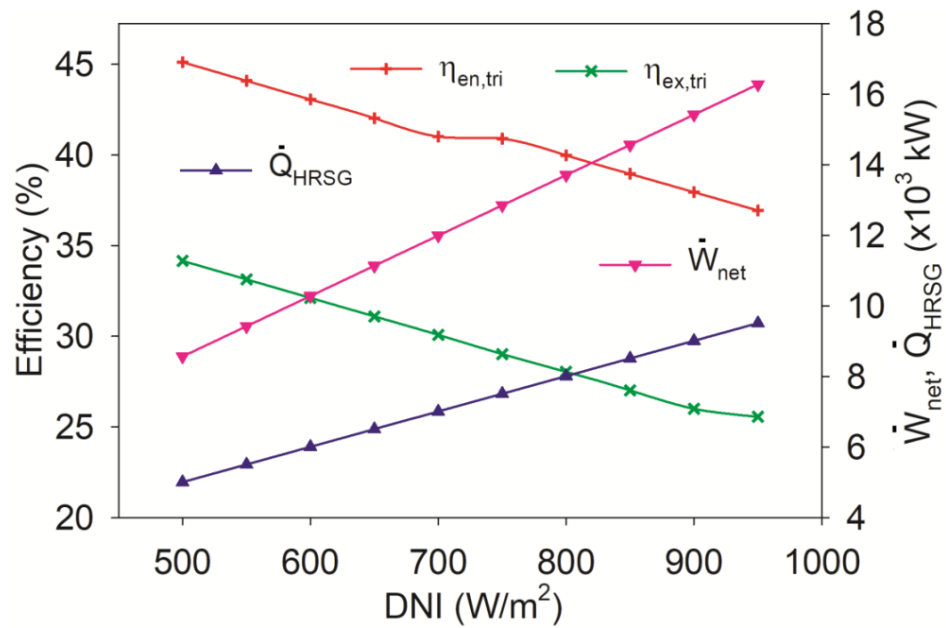


Figure 5.10: Trigeneration system performance variations with the DNI

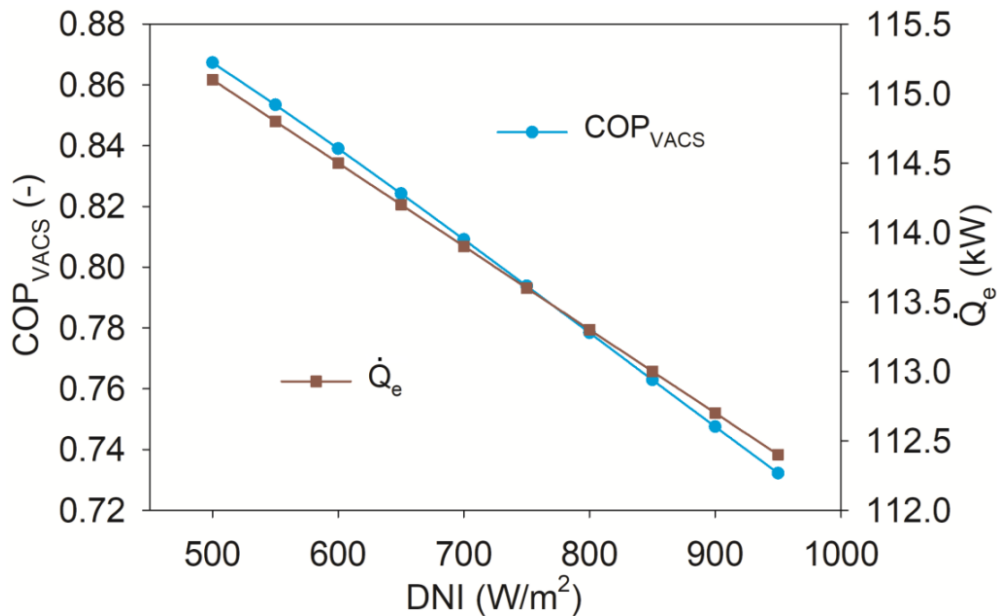


Figure 5.11: COP and cooling production variation with DNI

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

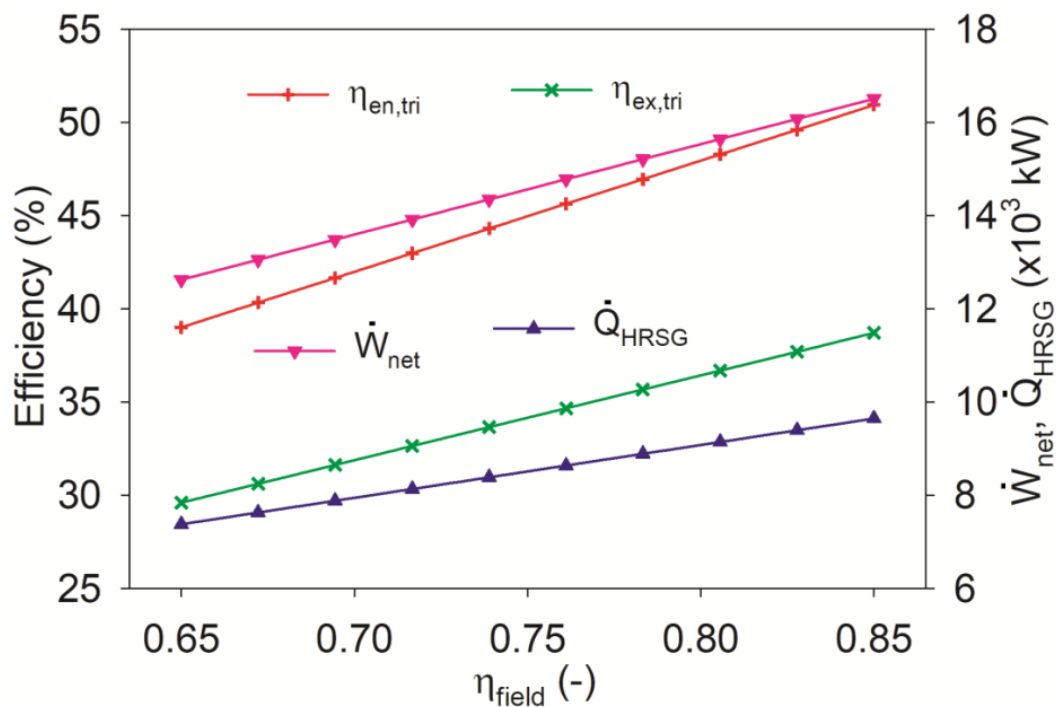


Figure 5.12: Trigeneration system performance variations with heliostat field efficiency

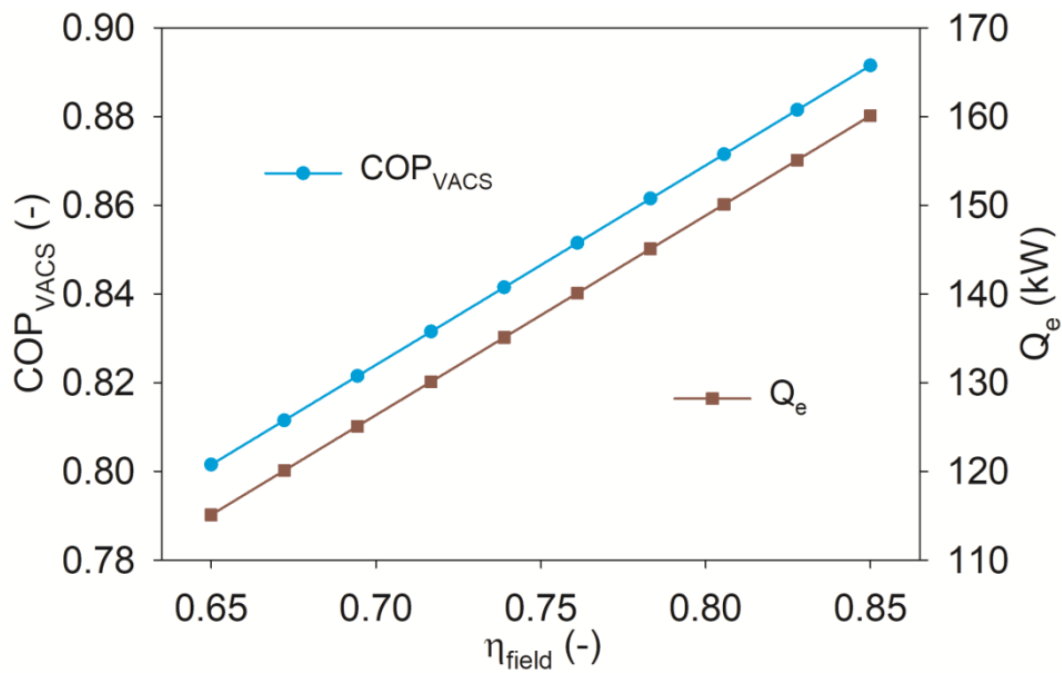


Figure 5.13: COP and cooling production variation with heliostat field efficiency

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

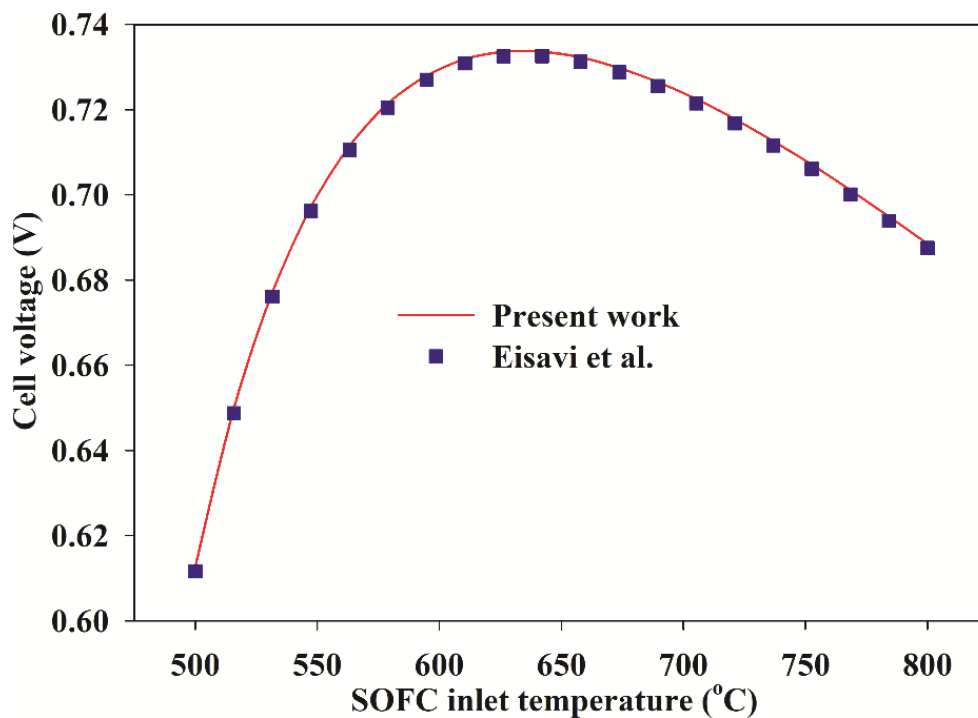


Figure 5.14: Validation of SOFC electrochemical modelling

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

Table 5.4: Validation of LiBr-H₂O VARS with the experimental work of Chen et al. [116]*

*Air cooled system, $T_{GEN}=90$ °C, $T_0=28$ °C; $T_{41}=15$ °C

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

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

7 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.

14 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.

71 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.

60 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

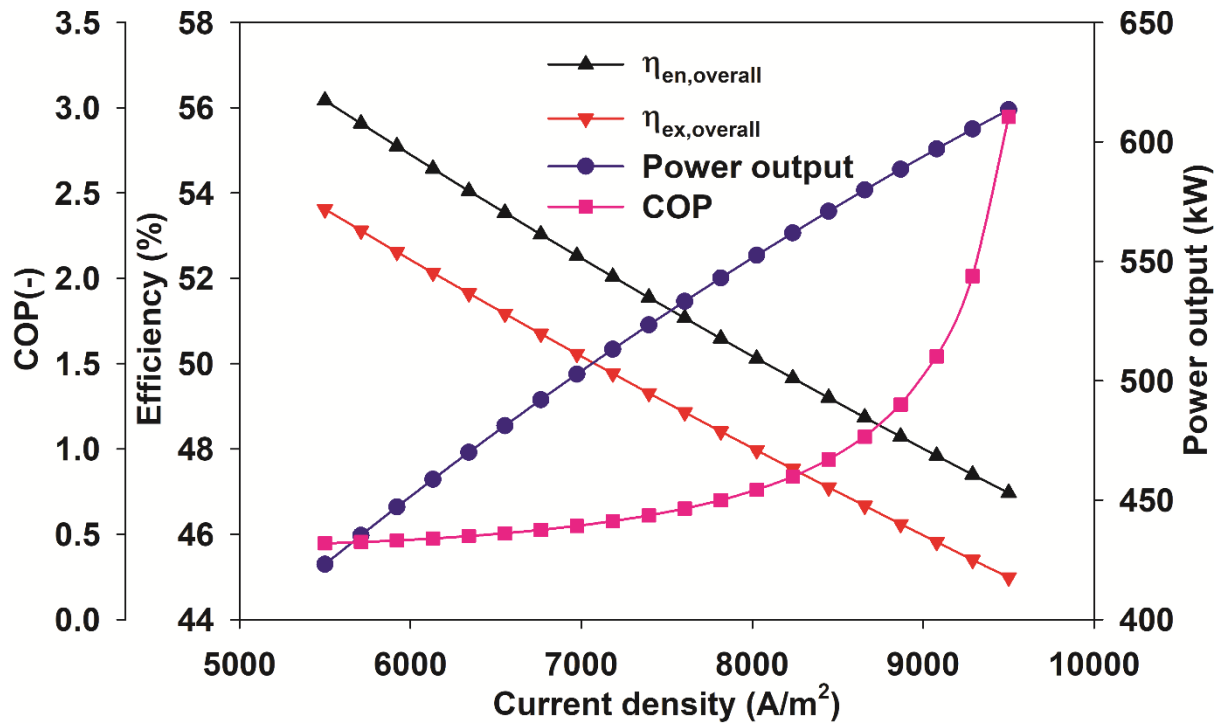
59 In this analysis, the proposed hybrid system utilizes a SOFC as the primary heat and power source. Therefore, it is critical to examine the influence of key SOFC operating parameters—specifically the U_f , j , and SOFC inlet temperature on the overall system performance. To isolate

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.

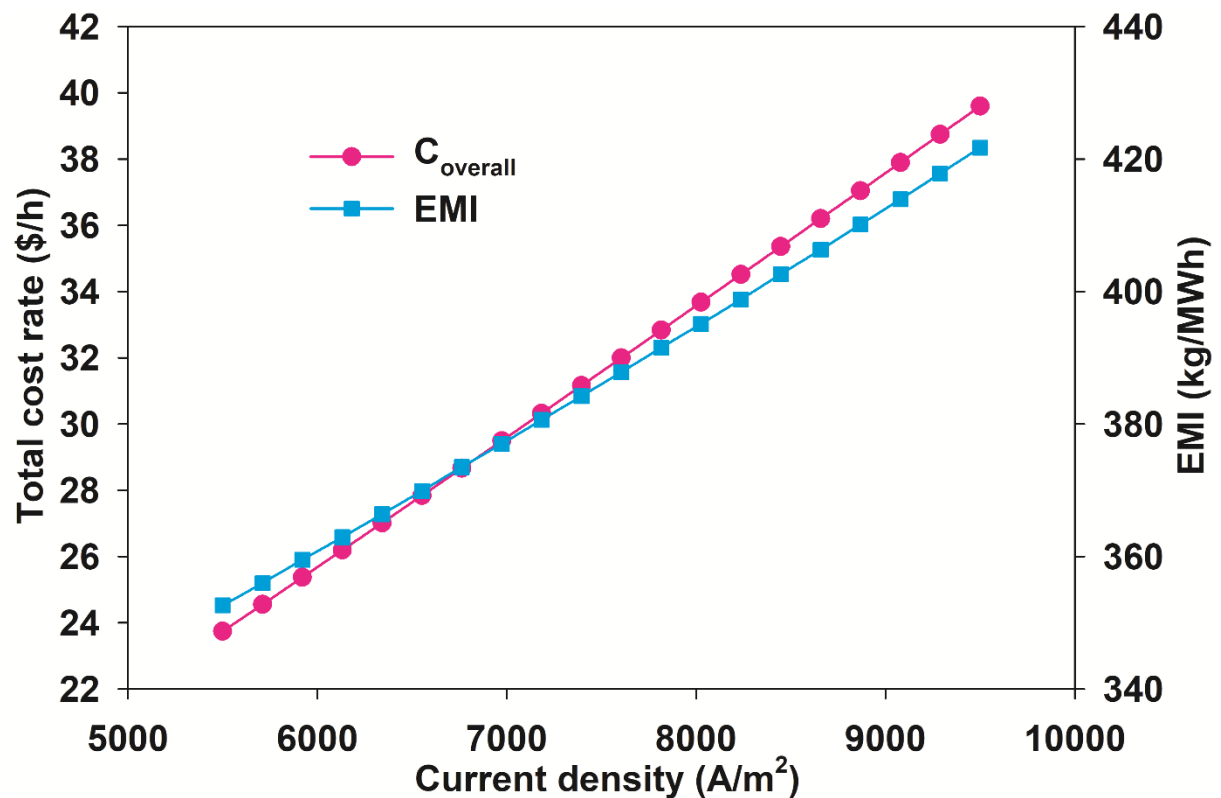
5.2.3.1 Current density effect on the performance

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)

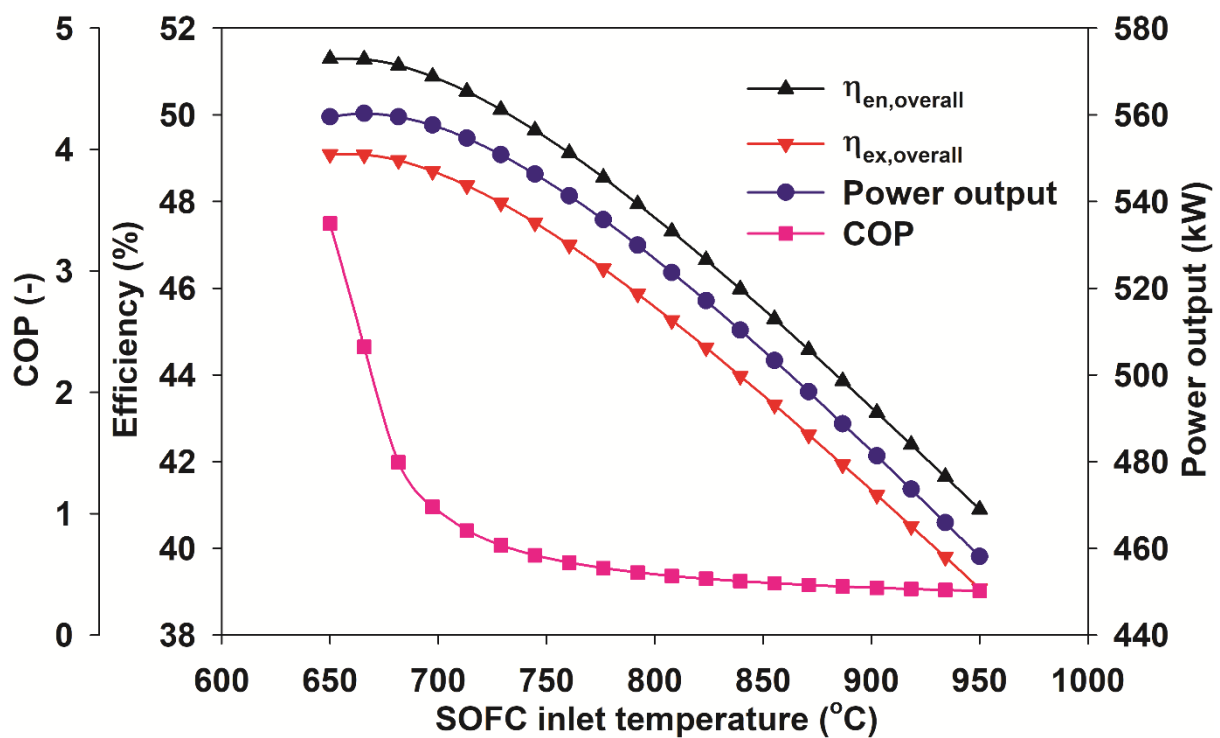


(b)

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

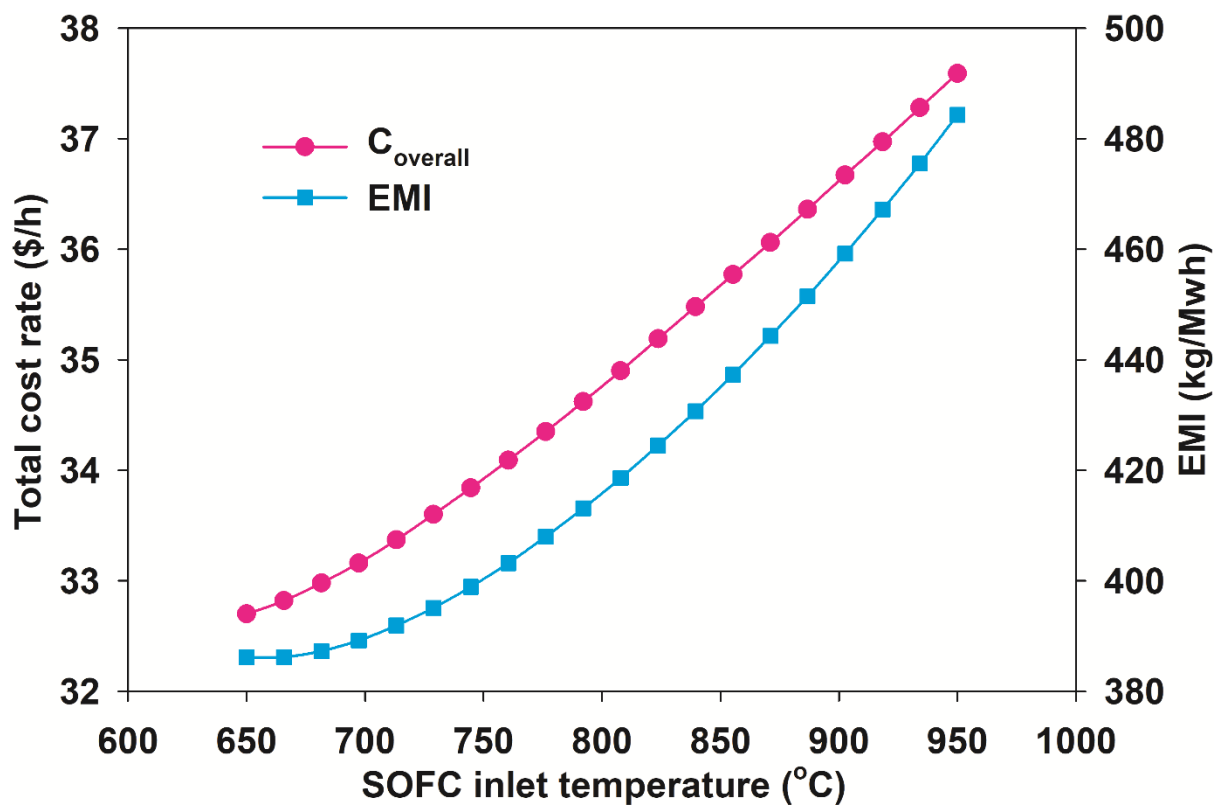
5.2.3.2 Inlet temperature of SOFC effect on performance

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.



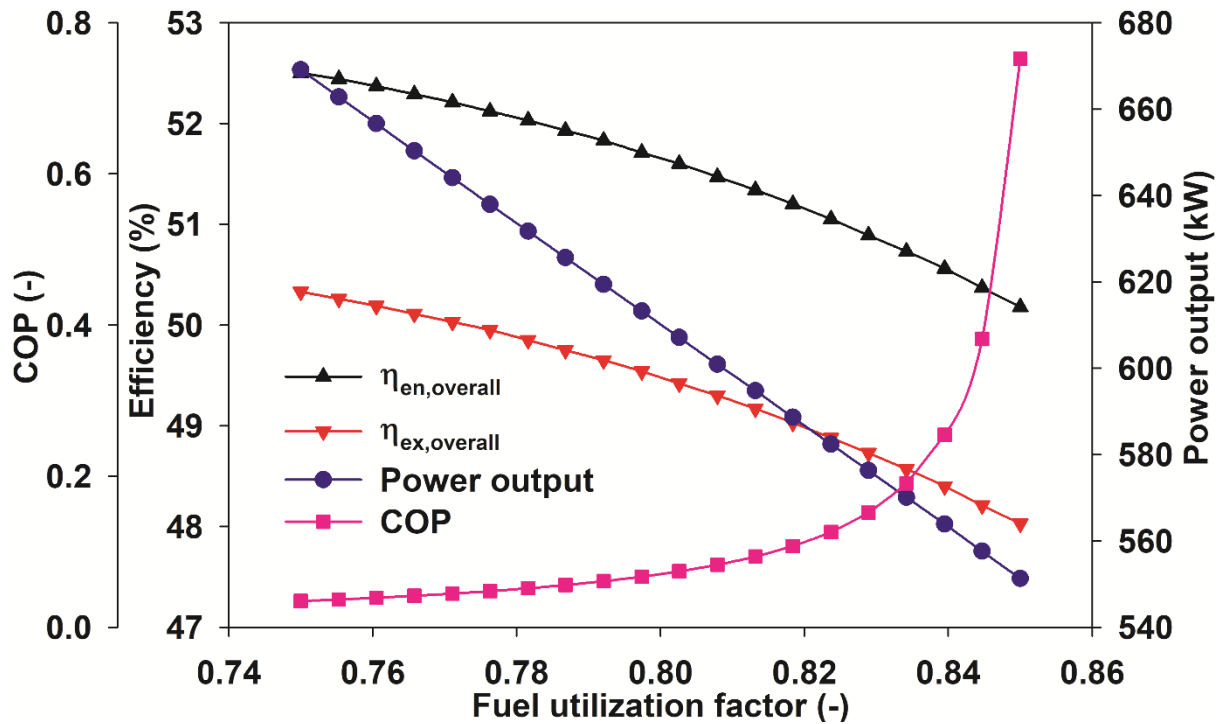
(b)

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

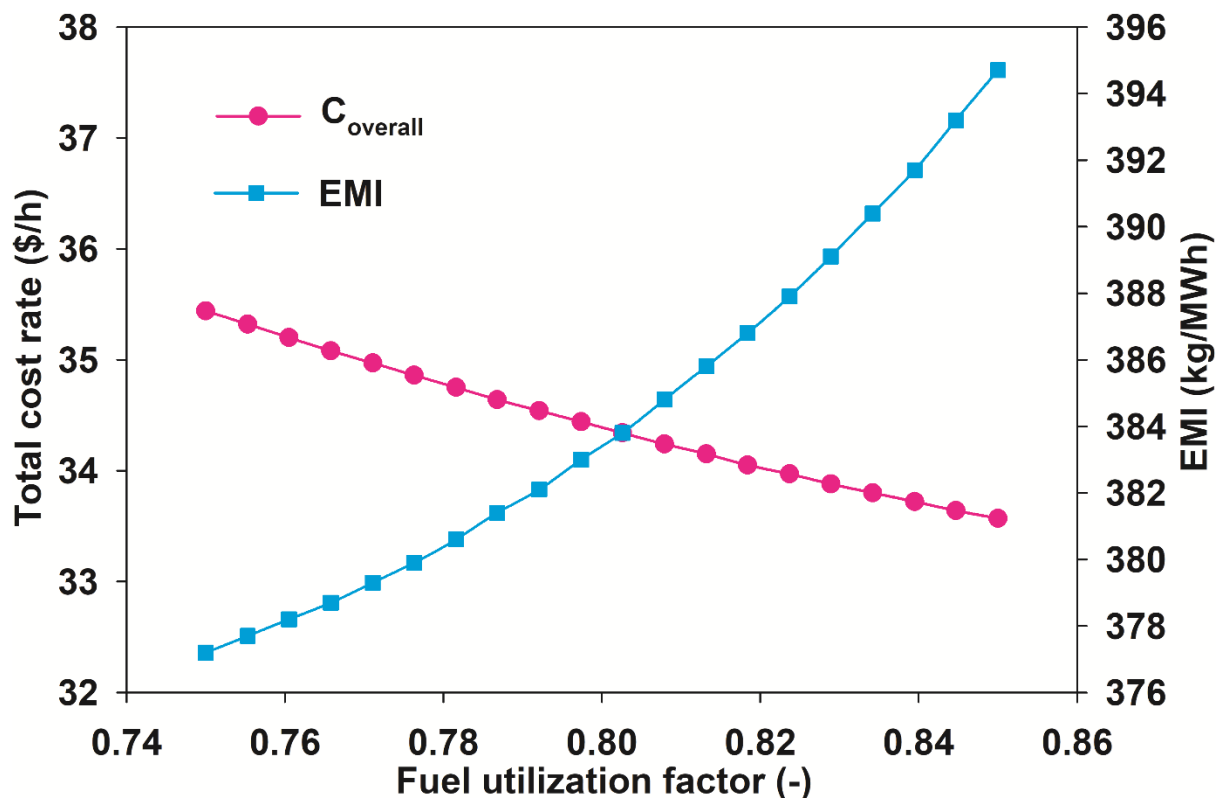
5.2.3.3 Fuel utilization factor impact on performance

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 5: CONCLUSIONS & FUTURE WORK

6.1 Conclusion

A. 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 and exergy efficiencies measured at 34.15% and 44.96%, respectively. 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.

B. 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

- System performance could be further enhanced by exploring alternative low-temperature cycles integrated into the existing SOFC-based hybrid setup.
- Future investigations may include detailed exergo economic evaluation and life cycle assessment of the system.
- A focused cost-effectiveness analysis of the proposed trigeneration configuration is another potential direction for future research.

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