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**NOVEL DESIGN, METHODS AND
DEVELOPMENT OF A NEXT-GENERATION
SOLAR-DRIVEN COGENERATION SYSTEM**

HAFIZ BIN AMAN ZAHARIL

PhD

The Hong Kong Polytechnic University

2026

The Hong Kong Polytechnic University
Department of Building Environment and Energy
Engineering

**Novel Design, Methods and Development of a Next-
Generation Solar-Driven Cogeneration System**

Hafiz Bin Aman Zaharil

**A thesis submitted in partial fulfilment of the requirements for
the degree of Doctor of Philosophy**

Aug 2025

CERTIFICATE OF ORIGINALITY

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_____ (Signed)

Hafiz Bin Aman Zaharil (Name of student)

ABSTRACT

Concentrated solar power (CSP) is currently limited in its thermal efficiency and thus economic feasibility relative to other technologies such as solar photovoltaics (PV) and fossil fuels. The current consensus roadmap for CSP's transition to 3rd generation, includes higher temperature working fluids, efficient thermal storage, advanced power cycles and, efficient waste heat utilisation from the system. Currently, the widespread use of thermal oil in CSP systems is constrained by its thermal stability limit of around 400°C, which restricts the overall efficiency of the power cycle. Molten salts offer a higher operational temperature of up to 565°C and improved thermal storage capability, but their high freezing point (~220°C) introduces significant challenges in system design, especially in preventing solidification during operation and downtime. Alternative fluids such as gas, which overcome some of the disadvantages mentioned above, have been theorised, but their low heat capacity and density, and high viscosity limit amplify the challenges. To overcome these limitations, supercritical fluids have emerged as promising candidates for next-generation CSP systems. Among them, supercritical carbon dioxide (sCO₂) is particularly attractive due to its ability to operate at higher temperatures, thereby enhancing thermodynamic efficiency. Compared to gases, sCO₂ has a much higher density and lower viscosity, which enables more compact turbomachinery, leading to lower land footprint and reduced compression work, thus increasing efficiency.

Currently, research on the supercritical carbon dioxide (sCO₂) Brayton cycle has been largely focused on nuclear applications, with limited exploration in the solar energy sector. In particular, its integration with solar thermal systems has underutilised the substantial waste heat produced, especially near the fluid's critical point, where the cooler rejects heat due to the high specific heat capacity of sCO₂. Moreover, the promising concept of operating in a direct configuration, without intermediate heat exchangers, remains underexplored, despite its potential advantages such as higher operating temperatures and reduced exergy losses,

particularly in its design methodology, where many assumptions were made and detailed steps are absent. To address these gaps, this study proposes an innovative direct integration of a parabolic trough solar collector (PTSC) with an sCO₂ Brayton cycle, an Organic Rankine Cycle (ORC), and a direct contact membrane distillation (DCMD) unit, along with some innovative methodologies, aiming to enhance overall energy utilisation and system efficiency

Firstly, a validated mathematical model from each of the proposed components in the system is developed and validated with experimental data from previous articles in the literature and reports from renowned national laboratories such as Argonne National Laboratory. The mathematical model is based on the foundations of heat transfer and thermodynamics, which lay the foundation for comprehensive energetic and exergetic analyses throughout the thesis. Additionally, typical economic models of levelised cost of electricity (LCOE), levelised cost of water (LCOW) and Net Present value (NPV) are developed for detailed economic analyses.

Secondly, a foundational thermodynamic analysis was conducted on a direct system integrating a PTSC with a recompression sCO₂ Brayton cycle and DCMD units. The study examined the system's energetic and exergetic performance under various pressure ratios, solar irradiance levels, and thermal boundary conditions. Results showed that both thermal and exergetic efficiencies decreased with lower solar input, with the highest exergetic efficiency of 38.3% observed at an optimal pressure ratio of approximately 3.1–3.2. Simultaneously, water production reached up to 96.6 m³/h under high feed temperatures and low permeate temperatures. The analysis also revealed that higher feed temperatures enhanced water output but increased sensitivity to solar fluctuations. These findings contribute to the design of more efficient solar thermal systems capable of simultaneously generating power and producing clean water.

Thirdly, this research presents a comprehensive thermodynamic and techno-economic analysis of a novel third-generation CSP system integrating a direct PTSC, sCO₂ Brayton cycle, DCMD, and a bottoming ORC. The primary goal was to understand the performance of this integrated system across key operational parameters, and to analyse and optimise ORC working fluids used in the bottoming cycle. Under baseline conditions, neopentane and isobutane demonstrated slightly higher net power outputs, whereas toluene provided superior water production due to its lower condenser inlet temperature. At an ambient temperature of 10°C, cyclohexane, n-octane, n-nonane, and n-heptane achieved the highest thermal efficiencies, reaching up to 36.52%. Exergy analysis identified significant losses in the DCMD unit, especially at the membrane, and in the second-stage compression and expansion processes of the sCO₂ cycle at high DNI. Nevertheless, the integrated configuration showed improvements in exergetic efficiency of 0.12–0.35% compared to single-cycle systems. Techno-economic evaluation further highlighted n-octane as having the highest net present value under colder conditions, while cyclohexane consistently performed well across different ambient temperatures. Additionally, this research establishes a discretisation-based modelling foundation enabling detailed design of PTSC and power cycles in direct configuration and the introduction of a ‘duality method’, which comprises a comprehensive methodological steps and utilisation of both pinch point and approach temperature to achieve maximal waste heat utilisation.

Fourthly, this research introduces a discretised 1-D modelling approach for CSP systems, providing a detailed methodology previously lacking in literature, specifically applied to a direct-configuration PTSC integrated with sCO₂ Brayton cycle. By employing this discretisation technique, the system’s performance was explored across essential parameters such as inlet and outlet temperatures, mass flow rates, collector length, and solar irradiance. This method revealed significantly different optimal conditions compared to prior approaches,

offering more precise design and operational insights. The results demonstrated distinct optimal inlet conditions for various targeted outlet temperatures and lengths, highlighting novel results missed by traditional optimisation in the literature. Thus, the discretised method serves as a foundational advancement, enabling detailed system designs and superior optimisation of future configuration CSP systems.

Lastly, this study presents a comprehensive thermo-economic mapping of the integrated PTSC-sCO₂-ORC-DCMD system, incorporating projected future cost reductions of sCO₂ Brayton cycle technology, which is anticipated to become commercially viable by 2030. For the first time, a detailed evaluation was conducted using component-level thermodynamic and economic correlations across varying power outputs, turbine inlet temperatures, and the effectiveness of the high-temperature (HTR) and low-temperature recuperators (LTR), parameters widely recognised as the most influential in power generation systems. Additionally, a duality method that considers both pinch point and approach temperature of a bottoming ORC is utilised, and the results are compared to a typical energy balance method, which is substantial. A 25-year life cycle assessment was performed using levelised cost of electricity (LCOE) and levelised cost of water (LCOW) as evaluation metrics. Results revealed that while larger power outputs significantly reduce LCOE, higher turbine inlet temperatures do not always translate to lower LCOE despite improving thermal performance. The LCOE of the system ranged from approximately 0.115 USD/kWh to 0.025 USD/kWh, and LCOW from 0.70 USD/m³ to 0.40 USD/m³, demonstrating the potential of this system to meet the targeted 0.05 USD/kWh for third-generation CSP and compete with the current cost range of reverse osmosis desalination (0.35–1.00 USD/m³).

This thesis introduces a novel integrated system comprising four distinct components that collectively utilise waste heat from an sCO₂ Brayton cycle with high efficiency. Furthermore, it proposes, for the first time in the literature, a direct integration methodology supported by a

discretised PTSC 1-D model, which leads to significantly novel results compared to previous studies. In addition, a duality approach, combining pinch point and approach temperature analyses, is developed to improve heat recovery between cycles more effectively than current methods reported in the literature. Finally, the thesis presents a comprehensive and detailed thermo-economic analysis and framework, incorporating component-level thermodynamic and economic correlations across varying power outputs, turbine inlet temperatures, and the effectiveness of HTR and LTR, which, to the best of current knowledge, is novel in the literature.

PUBLICATIONS DURING PHD STUDY

Journal papers:

- 1) Zaharil, H. A., & Yang, H. (2023). A fundamental investigation on supercritical carbon dioxide energetic, exergetic and entropy behavior in parabolic trough solar collector. *Journal of Cleaner Production*, 384, 135625. doi:<https://doi.org/10.1016/j.jclepro.2022.135625>
- 2) Zaharil, H. A., & Yang, H. (2024). Novel solar-based cogeneration system: Parabolic trough integrating supercritical Brayton and organic Rankine cycles with membrane distillation. *Applied Energy*, 376, 124210. doi:<https://doi.org/10.1016/j.apenergy.2024.124210>
- 3) Zaharil, H. A., & Yang, H. (2024). Thermodynamic analysis of a parabolic trough power plant integrating supercritical carbon dioxide Brayton cycle and direct contact membrane distillation. *Applied Thermal Engineering*, 252, 123637. doi:<https://doi.org/10.1016/j.applthermaleng.2024.123637>
- 4) Zaharil, H. A., & Yang, H. (2025). Comprehensive thermoparametric analysis of supercritical carbon dioxide in parabolic trough solar collector via an innovative method. *Applied Thermal Engineering*, 125434. doi:<https://doi.org/10.1016/j.applthermaleng.2025.125434>

Conference paper:

- 1) Zaharil, H. A., & Yang, H. (2024). Exploring Direct Parabolic Trough Supercritical Brayton Cycle with Direct Contact Membrane Distillation (DCMD) for co-generation of Power and Water. Paper presented at the *15th International Conference on Applied Energy (ICAE2023)*, Doha, Qatar.

Manuscript under work:

- 1) Comprehensive thermo-economic mapping of sCO₂ Brayton and Organic Rankine cycles via multiple novel methodological approaches. (In collaboration with UCL)

ACKNOWLEDGEMENTS

Three years ago, I was searching for a way forward for my career development, but I was unsure where my ambitions would take me. Everything shifted after I reached out to Prof Yang and he responded. His message didn't just introduce an opportunity to join his doctoral programme, but it opened a door I hadn't imagined and gave me the confidence to walk through it. It was a challenging start, with new surroundings and new expectations, but I was fortunate to have Prof. Yang and the Renewable Energy Research Group by my side. I am also deeply grateful to The Hong Kong Polytechnic University for their generous support throughout my studies, and specifically to the Graduate School for funding my six-month research visit to University College London under the prestigious PolyU PhD Scholars International Collaborative Research Fellowship (ICRF).

First and foremost, I would like to express my heartfelt gratitude to my supervisor, Prof. Yang, for his expertise, encouragement, and unwavering support. His profound knowledge and insightful feedback on my research have helped me avoid the wrong trajectory. I am deeply thankful for his encouragement when I struggled with writing and revising papers, his patience in supporting me to reach my full potential, and most of all, for taking a chance on me when he welcomed me into his research group.

I sincerely thank all my team members for their friendship, encouragement, and kind advice throughout my Ph.D. study, as well as the department's academic and support staff for their guidance and assistance. I am deeply grateful to my beloved family, whose unwavering love, belief, and sacrifices have been my driving force. Their constant support, along with that of my supervisor and colleagues, has made this thesis possible, and I will always cherish their indispensable contributions. I would like to also extend my gratitude to the dedicated staff members at BEEE, Graduate School (GS), Finance office (FO), and FCE for all their help.

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ABBREVIATIONS

AACE - Association for the Advancement of Cost Engineering

AGMD - Air-gap membrane distillation

CAPEX - Capital expenditure

CEPCI - Chemical Engineering Plant Cost Index

CCHP - Combined cooling, heating and power

CF - Capacity factor

CSP - Concentrated solar power

DCMD - Direct contact membrane distillation

DNI - Direct normal irradiance

EE - Evaporation efficiency

EES - Engineering Equation Solver

GWP - Global warming potential

HCE - Heat-collecting element

HTR - High-temperature recuperator

IAEA - International Atomic Energy Agency

ICAE - International Conference on Applied Energy

IPC - Integrated performance coefficient

IPCC - Intergovernmental Panel on Climate Change

IRENA - International Renewable Energy Agency

LCOW - Levelised cost of water

LCOE - Levelised cost of electricity

LTR - Low-temperature recuperator

MD - Membrane distillation

MED - Multi-effect desalination

MSF - Multi-stage flash desalination

MWe - Megawatt electric

MWth - Megawatt thermal

NPV - Net present value

NREL - National Renewable Energy Laboratory

OPEX - Operating expenditure

ORC - Organic Rankine cycle

PCHE - Printed circuit heat exchanger

PR - Pressure ratio

PTSC - Parabolic trough solar collector

PV - Photovoltaics

RO - Reverse osmosis

SAM - System Advisor Model (NREL)

sCO₂ - Supercritical carbon dioxide

SDG - Sustainable Development Goal

SPT - Solar power tower (central receiver)

STEC - Specific thermal energy consumption

TES - Thermal energy storage

TPC - Temperature polarisation coefficient

USDOE - United States Department of Energy

NOMENCLATURES

Roman Symbols

- A - Area (m^2)
A_a - Collector aperture area (m^2)
C_p - Specific heat capacity ($\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$)
D - Diameter (m)
f - Friction factor (-)
h - Specific enthalpy ($\text{J} \cdot \text{kg}^{-1}$)
I - Solar irradiance ($\text{W} \cdot \text{m}^{-2}$)
k - Thermal conductivity ($\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$)
L - Length (m)
 $\dot{m}$ - Mass flow rate ($\text{kg} \cdot \text{s}^{-1}$)
P - Pressure (Pa, kPa, MPa)
PR - Pressure ratio (-)
 $\dot{Q}$ - Heat transfer rate (W)
Re – Reynolds number (-)
s – Specific entropy ($\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$)
T - Temperature (K)
T_{in} - Inlet temperature (K, C)
T_{out} - Outlet temperature (K, C)
U - Overall heat transfer coefficient ($\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$)
W - Power (W)
W_{net} - Net power output (W, kW or MWe)

Greek symbols

- Δ - Change in a quantity (-)
 ΔP - Pressure drop (Pa, kPa, MPa)
 ε - Heat exchanger effectiveness (-)
 η_{th} - Thermal efficiency (-)
 η_{ex} - Exergy-based efficiency (-)

ρ - Density ($\text{kg} \cdot \text{m}^{-3}$)

σ - Stefan-Boltzmann constant ($\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-4}$)

Dimensionless groups

Pr - Prandtl number (-)

Nu - Nusselt number (-)

Re - Reynolds number (-)

Ra - Rayleigh number (-)

Subscripts / Labels

0 - Reference (ambient/dead-state) condition

amb - Ambient condition

in, out - Inlet, outlet

net - Net value

th – Thermal/Energy

ex - Exergy

b_f - Bulk feed

mf - Membrane feed side

mp - Membrane permeate side

bp - Bulk permeate

Economic and techno-economic symbols

CAPEX - Capital expenditure (USD)

OPEX - Operational expenditure (USD)

NPV - Net present value (USD)

LCOE - Levelised cost of electricity ($\text{USD} \cdot \text{kWh}^{-1}$)

LCOW - Levelised cost of water (USD m^{-3})

CRF - Capital recovery factor (-)

i - Discount rate (-)

t - Project lifetime (yr)

CHAPTER 1 INTRODUCTION

1.1 Overview and problem statement

Ensuring a reliable supply of energy while drastically reducing greenhouse-gas emissions is one of the most pressing challenges of this century. Although policies since the 1970s oil crisis have aimed to improve security of supply, fossil fuels still provide roughly four-fifths of the world's energy mix (Earth.org, 2024). In 2023 alone, fossil fuels accounted for 82% of the global energy consumption (Reuters, 2024). This continued reliance on carbon-intensive sources underpins most of the CO₂ emissions driving climate change, with over 36.8 Gt of energy-related emissions recorded in 2022 alone (IEA, 2023).

The urgency of transitioning to low-carbon energy systems has led to rapid investments in renewable energy technologies. However, by 2024, this transition still faces obstacles such as weak political will, insufficient financing, technological barriers, and limited implementation of long-term policy frameworks (Ritchie & Rosado, 2024; Nsabiyeze et al., 2024). As global energy demand continues to rise due to population growth, urbanisation, and economic development, bridging the gap between sustainable supply and increasing demand has become even more critical.

Among various renewable sources, solar energy has emerged as one of the most promising. It is not only abundant and universally available but also among the cleanest energy options. According to IRENA (2023), solar energy made up 31% of electricity generation from renewable sources globally. Solar technologies are broadly divided into two categories: photovoltaic (PV), which directly converts sunlight into electricity, and solar thermal, which captures solar heat for use in heating, cooling, desalination, and power generation. While solar PV has become dominant due to falling costs, solar thermal technology, particularly

Concentrated Solar Power (CSP), offers unique benefits in terms of energy storage and dispatchability.

CSP systems use mirrors or lenses to concentrate direct normal irradiance (DNI) onto a receiver, which absorbs and transfers the heat to a working fluid (Fig. 1.1). There are two main types of CSP, line-focus systems, such as parabolic trough and Fresnel reflectors, and point-focus systems such as solar towers and parabolic dishes. Line-focus systems are the most widely deployed, with parabolic trough systems alone accounting for about 64% of the global CSP capacity as of 2021, representing more than 4,000 MW of installed capacity (NREL, 2021).

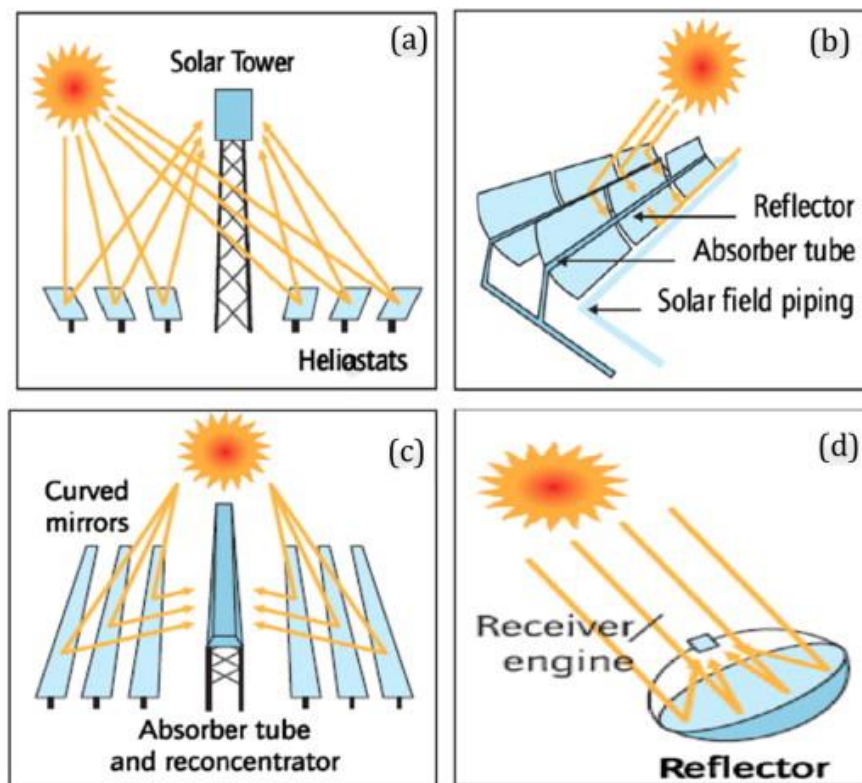


Fig. 1.1 Types of Concentrated Solar Power and its working principles (Ghaffour et al. 2015)

Despite technological advancements, CSP systems still lag behind solar PV in terms of cost competitiveness. The levelised cost of electricity (LCOE) from CSP dropped from around USD 0.35/kWh in 2010 to USD 0.075/kWh in 2021 (IRENA, 2022), a significant achievement.

However, solar PV achieved LCOEs below USD 0.05/kWh in the same year (IRENA, 2022), making it the more attractive option for utility-scale deployments in most countries (refer to Fig. 1.2)

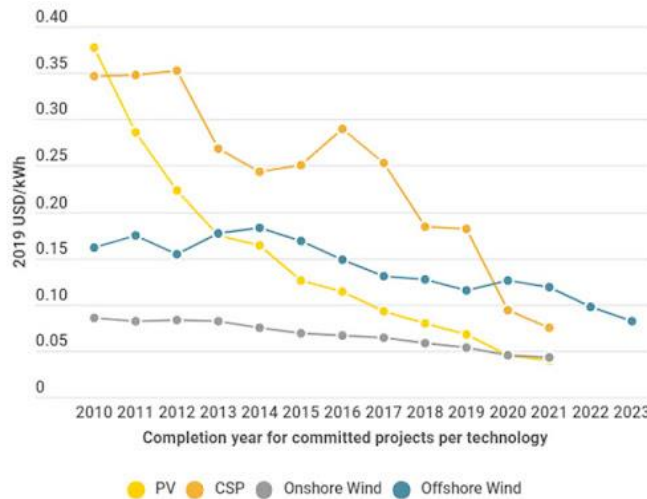


Fig. 1.2 LCOE between selected renewable energy technologies (IRENA, 2022)

One of the fundamental limitations in traditional CSP systems is the choice of working fluids. Thermal oils, commonly used in parabolic trough collectors, have a maximum operating temperature of around 400°C (DLR, 2024). This ceiling constrains the theoretical efficiency of the thermodynamic cycle, as dictated by the Carnot principle. The use of higher temperature fluids is needed to achieve meaningful gains in thermal efficiency.

Molten salts have emerged as a potential alternative due to their higher thermal stability, capable of operating up to 565°C. However, these salts have drawbacks, including high freezing points (typically above 220°C), corrosion risks, complex handling procedures, and agglomeration issues (Zaharil & Yang, 2024; Solarthermalworld, 2023). These challenges increase system complexity and raise the cost of auxiliary equipment, reducing overall system reliability and financial viability.

To address these limitations, the next generation of CSP systems, commonly referred to as Gen3 CSP, is being developed. These systems aim to achieve outlet temperatures of 700°C or

higher, which would enable more efficient thermodynamic cycles such as the Brayton cycle (Refer Fig. 1.3). The goal of Gen3 CSP is to achieve an LCOE of around 0.05\$/kWh. The U.S. Department of Energy and NREL have supported several Gen3 demonstration projects aiming to validate this approach (NREL, 2023). A promising candidate for high-temperature operation is supercritical carbon dioxide (sCO₂), which offers several advantages over traditional fluids.

Generation	1 st gen.	2 nd gen.	3 rd gen.
Receiver outlet temp.	~250 - 450 °C	~500 - 565 °C	~720 °C
Typical plant or technology	PTC, SPT, LFR 	PTC, SPT, LFR ~500 - 565 °C 	PDC 
Heat transfer medium	Oil or steam	Steam or salt	Gas
Thermal energy storage	Early designs: No or small Recent designs: Yes	Early designs: No or small Recent designs: Yes	No
Power cycle	Steam Rankine cycle	Stirling	Brayton cycle
Peak temp. of cycle	~240-440 °C	~480-550 °C	~720 °C
Design cycle eff.	~ 28-38%	~ 38-44%	~38%
Annual solar-electric eff.	~ 9-16%	~ 10-20%	~25%

Fig. 1.3 Illustrations of various generations of Concentrated Solar Power (CSP) technologies and predictions for the future of CSP. (Taken from He et al., 2020)

sCO₂ is a fluid at a state above its critical temperature (31.1°C) and critical pressure (7.38 MPa), where it exhibits unique properties such as high density, low viscosity, and excellent thermal stability. These characteristics allow for more compact and efficient turbomachinery, reduced pumping work, and lower heat transfer losses (Turbomachinerymag, 2024). In contrast to thermal oils or molten salts, sCO₂ does not pose risks of solidification at typical operating temperatures and is non-toxic and non-flammable. Its use enables systems to operate more efficiently at higher temperatures, thereby increasing the overall exergy efficiency of the cycle.

Another critical issue facing power generation systems is energy waste. Globally, thermal power plants typically lose about 60% of energy input as waste heat due to inherent inefficiencies in the Rankine cycle and cooling requirements (Thunder Said Energy, 2023). This wasted thermal energy represents an enormous untapped opportunity for improving overall system performance.

Past research has focused on waste heat recovery through combined cycles, thermoelectric conversion, or by integrating low-temperature cycles like the Organic Rankine Cycle (ORC). The ORC is particularly well-suited for this purpose due to its ability to efficiently convert low-grade thermal energy into electricity using organic fluids with low boiling points.

Building on this understanding, the present study proposes an integrated CSP system using a parabolic trough collector coupled with an sCO₂ Brayton cycle, supported by an ORC bottoming cycle for waste heat recovery. What sets this system apart is the use of sCO₂ as the primary working fluid, which eliminates the heat transfer losses typically observed when using intermediate fluids. Additionally, waste heat from the sCO₂ cooler and the ORC condenser is utilised for clean water production via direct-contact membrane distillation (DCMD) using seawater, contributing to water sustainability in arid regions.

This integrated approach not only improves thermal efficiency and reduces exergy destruction but also supports the cogeneration of electricity and water, a particularly important feature for regions suffering from both energy and water scarcity. It also enables CSP technology to move beyond its conventional role and into a multipurpose, high-efficiency energy platform that can compete with other renewables in terms of cost, reliability, and impact.

In summary, while CSP technology has made significant progress over the past two decades, it must evolve further to remain viable in a rapidly decarbonising world. The use of sCO₂, higher operating temperatures, and integrated cogeneration systems represents a critical pathway toward this evolution. Through innovation, integration, and careful techno-economic

optimisation, the next generation of CSP systems can play a meaningful role in supporting global energy transition goals.

Importantly, such advancements also align with the United Nations Sustainable Development Goals (SDGs). Specifically, this work supports SDG 7 (Affordable and Clean Energy) by promoting accessible, reliable, and modern energy services. It contributes to SDG 9 (Industry, Innovation and Infrastructure) through the development of cutting-edge power systems, and SDG 13 (Climate Action) by directly addressing greenhouse gas emissions. The water production aspect of the system further supports SDG 6 (Clean Water and Sanitation), highlighting the potential of integrated renewable technologies to deliver multifaceted benefits for sustainable development (Fig. 1.4).

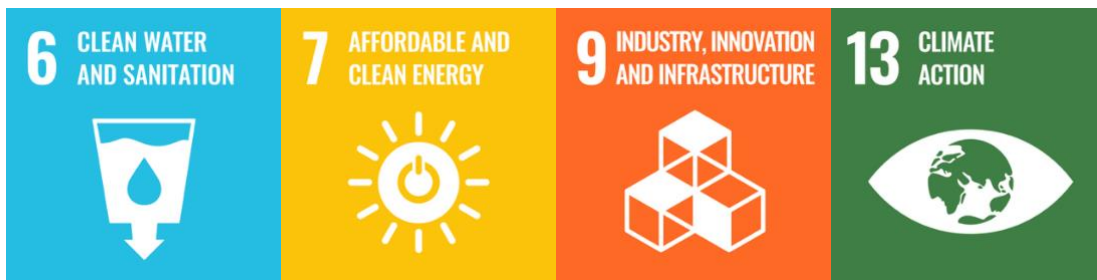


Fig. 1.4 The United Nations sustainable goals relevant to this research

1.2 Motivation

This thesis is motivated by the need to design a next-generation concentrated solar power (CSP) system that achieves high thermal efficiency, low waste heat, and a compact system footprint, which are key goals of third-generation CSP. Existing CSP technologies are often limited by the thermal stability of traditional working fluids and underutilised waste heat, which lowers overall performance and raises operational costs. As a result, CSP struggles to compete with photovoltaics and fossil fuels in terms of efficiency and cost. To remain relevant in the global transition to clean energy, CSP must evolve through innovative system designs

that allow higher operating temperatures, better integration of subsystems, and smarter use of thermal resources.

This research proposes a novel integrated CSP configuration that directly couples a parabolic trough solar collector (PTSC) with a supercritical carbon dioxide (sCO₂) Brayton cycle, a bottoming organic Rankine cycle (ORC), and a direct contact membrane distillation (DCMD) unit. The system is designed to operate at high temperatures, reduce thermal losses, and co-generate electricity and clean water. Its compact design and improved energy recovery make it a promising candidate for future solar-thermal applications aiming for both energy and water sustainability.

Besides the novel design of an integrated system that enhances efficiency, minimises waste heat, and reduces footprint, the main academic contribution lies in the detailed methodological approach developed throughout the thesis. This includes a discretised 1-D modelling method for PTSC design, which allows precise temperature tracking and system optimisation across a wide range of operating conditions, something not previously demonstrated in CSP literature. Additionally, a ‘duality method’ is introduced, which combines pinch point and approach temperature analysis to maximise waste heat recovery between cycles, improving overall thermal integration.

Finally, the thesis presents a comprehensive and detailed thermo-economic framework that incorporates component-level thermodynamic and economic correlations across different power outputs, turbine inlet temperatures, and the effectiveness of both high-temperature (HTR) and low-temperature (LTR) recuperators. This framework enables a deeper understanding of the cost-performance trade-offs and provides a practical foundation for designing efficient, cost-competitive third-generation CSP systems.

1.3 Thesis structure

This thesis presents a step-by-step development from **Chapter 2** to **Chapter 8** (Fig. 1.5). In **Chapter 2**, the literature review identifies the current limitations of CSP systems, the potential of supercritical CO₂, the research gaps in solar integration and outlines the thesis objectives. In **Chapter 3**, the detailed methodology is shown. Validated models for each component, namely PTSC, sCO₂ Brayton cycle, ORC, and DCMD are developed using thermodynamic principles and experimental data, alongside cost models for LCOE, LCOW, and NPV. **Chapter 4** presents a novel application of a 1-D discretised PTSC model to the integrated system, revealing optimal conditions overlooked by traditional approaches. **Chapter 5** conducts a thermodynamic analysis of the direct PTSC- sCO₂- DCMD configuration, exploring the effects of pressure ratio, DNI, and temperature on power and water output. **Chapter 6** expands the system by adding a bottoming ORC and performs a comprehensive thermo-economic and exergy analysis, introducing a duality method for improved heat recovery. **Chapter 7** maps the full system's performance and economics under projected future costs, and compares different turbine inlet temperatures, power outputs, and recuperator effectiveness. **Chapter 8** concludes the thesis and suggests future research directions.

Development and Investigation of an Integrated sCO₂-Based CSP System

Propose a novel integrated system design

- Develop validated component models
- Compile thermodynamic and economic models

Perform foundational thermodynamic analysis

- Analyze direct integration system
- Utilize thermodynamic performance

Carry out detailed thermo-economic analysis

- Conduct deep multi-cycle thermodynamic analysis
- Optimise system parameters including organic fluid selection
- Introduction of duality method
- Incorporation of discretisation method to power generation design.

Undertake comprehensive mapping study

- Establish thermo-economic performance maps of the integrated PTSC–sCO₂–ORC–DCMD system using a novel approach combining component-level thermodynamic and economic correlations across key parameters: power output, turbine inlet temperature, and recuperator effectiveness.
- Assess the overall system thermodynamic and economic performance by evaluating life cycle cost metrics (LCOE and LCOW)

Contributions

- Proposed a novel integrated system utilising waste heat from an sCO₂ Brayton cycle
- Developed a discretised PTSC 1-D model for direct integration
- Utilised a duality approach for enhanced heat recovery between cycles
- Carried out detailed thermo-economic analyses with novel methodological approaches

Fig. 1.5 Thesis Framework

With these investigations, this thesis has delivered four major contributions. First, it introduces a novel integrated system that combines four distinct components to efficiently utilise waste heat from an sCO₂ Brayton cycle and a smaller land footprint requirement. Second,

it proposes, for the first time in the literature, a direct integration strategy based on a discretised 1-D PTSC model, which yields significantly novel results compared to previous studies. Third, it develops a dual-approach method combining pinch point and approach temperature analyses to enhance heat recovery between cycles beyond what existing methods offer. Finally, it presents a novel and detailed thermo-economic framework that integrates component-level thermodynamic and cost correlations across a range of power outputs and operating conditions, marking a new contribution to the field.

CHAPTER 2 LITERATURE REVIEW

In this chapter, we first review the recent progress of concentrated solar power (CSP), focusing on the working fluids, power cycles, and their waste heat utilisation. Then, we provide an overview of the status of Organic Rankine cycle (ORC) and membrane distillation (MD) technologies, covering their principles, advantages, and disadvantages. Specifically, we emphasise direct CSP integration methods and waste heat utilisation from topping cycle methods that are highly relevant to our study. Next, we review the recent progress in CSP research utilising sCO₂ and waste heat and identify the remaining gaps in the literature.

2.1 Current Status of Concentrated Solar Power

In recent years, concentrated solar power (CSP) has progressed significantly as a means of harnessing solar energy for utility-scale electricity generation, with the added benefit of inherent thermal energy storage. Globally, CSP capacity has steadily grown to over 6 GW in early 2023 and 8.1 GW by late 2023 (refer to Fig. 2.1) (Ren21, 2024; SOLARPACES, 2025), with commercial plants operating in countries such as Spain, the United States, Morocco, South Africa, China, and others (SOLARPACES, 2023). Two main CSP technologies have dominated deployments, namely parabolic trough systems and central solar power tower (SPT) plants (IEA, 2023). Parabolic trough plants account for the majority of installed CSP capacity worldwide due to their earlier commercialisation and proven track record (Behar et al. 2013; Arias et al. 2022). However, power tower systems have gained prominence in recent projects, as they offer higher operating temperatures and potentially greater efficiency and storage capability (Hernández et al. 2025). Other CSP configurations, like linear Fresnel reflectors and dish-Stirling engines, have also been developed, but they represent a smaller fraction of current installations (Cellura et al. 2025).

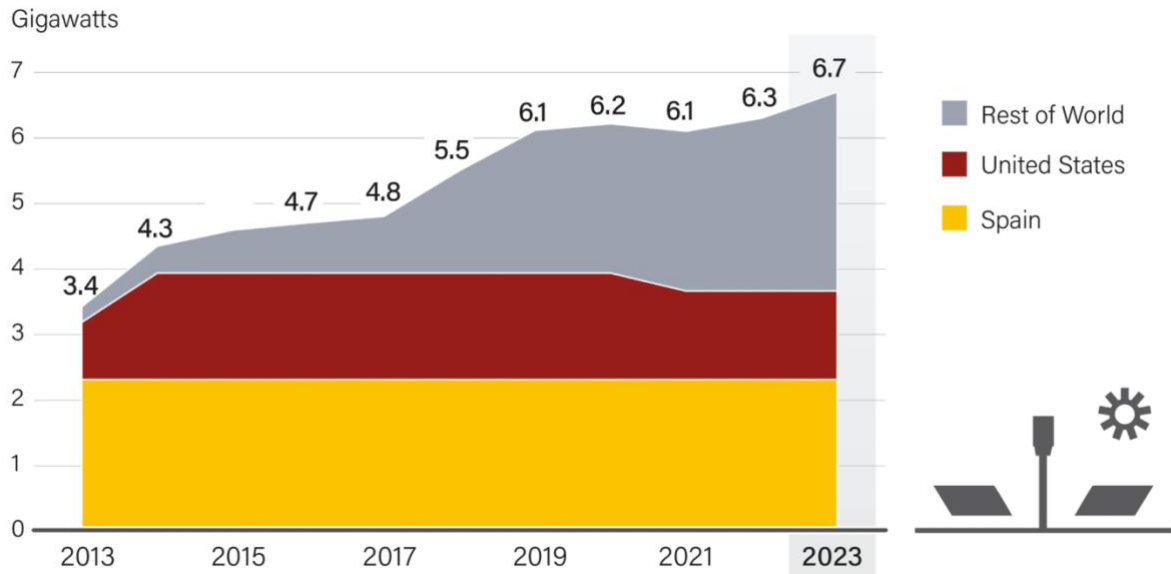


Fig. 2.1 CSP global capacity growth (REN21, 2024)

Parabolic trough CSP plants employ long, curved mirrors to focus sunlight onto a linear receiver tube, heating a working fluid, traditionally synthetic oil, limited to around 400 °C, which constrains steam conditions and thermal-to-electric efficiency. Recent designs increasingly use molten salt, enabling operation at around 500–550 °C, improving Rankine cycle performance and facilitating multi-hour thermal energy storage, as demonstrated by plants such as Spain’s 50 MW Andasol series with around 7 h storage. In contrast, solar power towers utilise heliostats to concentrate sunlight onto a central receiver, achieving higher operating temperatures. While early towers generated steam directly at ~250 to 300 °C, modern systems circulate molten nitrate salt heated to around 550–565 °C, allowing 8 to 15 hours of storage and decoupling solar collection from power generation. Examples include the 110 MW Crescent Dunes (Nevada) with 10 h storage and the 150 MW Noor III (Morocco) with 7 to 8 hours storage. Higher operating temperatures in tower systems enable more efficient power cycles, potentially surpassing trough plants under comparable conditions. Ongoing research and development are targeting even higher-temperature working fluids

(around ≥ 700 °C or above) by exploring advanced heat-transfer media, such as novel molten salt mixtures such as, chloride-based salts, solar salts, lithium -based salts to name a few (Turchi et al., 2017; Starke et al. 2024) or solid particle-based heat transfer and storage systems (Deng et al. 2025; Khan et al, 2022; Ho et al, 2016). Additionally, gaseous and supercritical fluids have attracted considerable interest owing to their favourable thermophysical properties, including high density, low viscosity, and the pronounced increases in density and specific heat capacity near the critical point, which enable more compact turbomachinery (refer to Fig. 2.2), reduced compression power requirements, and enhanced opportunities for waste heat recovery. Reaching these higher temperatures would enable next-generation CSP to integrate with advanced power cycles like the sCO₂ Brayton cycle, which can significantly boost the thermal-to-electric conversion efficiency beyond what is possible with conventional steam-Rankine cycles (Wang et al. 2017; Zaharil & Yang, 2024).

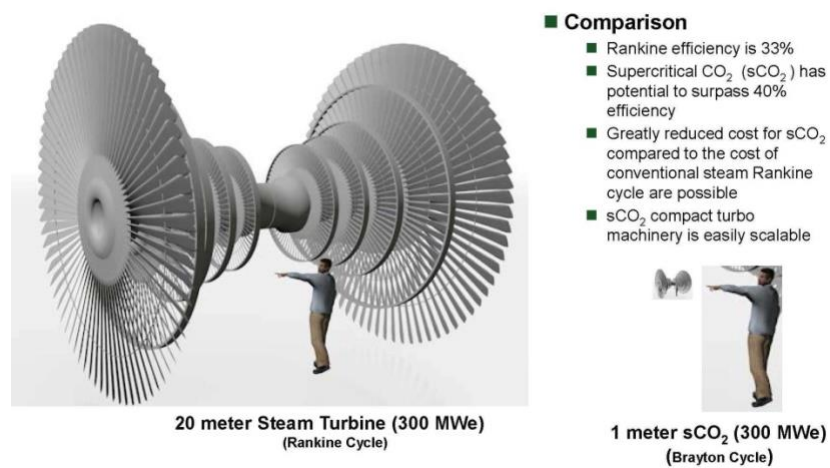


Fig. 2.2 Component size comparison between sCO₂ Brayton Cycle and Steam Rankine cycle (US Department of Energy, 2025)

Apart from improving the working fluids and power cycles, recent CSP research has also emphasised waste heat utilisation to raise overall system efficiency further and provide

additional benefits. In a typical CSP plant, a substantial amount of heat is rejected at the condenser of the power cycle as low-grade waste heat. Instead of dissipating this thermal energy to the environment, it can be harnessed for secondary applications (Ghaibi & Mohan, 2015; Zaharil & Yang, 2025).

Waste heat from CSP plants can be utilised for cogeneration and polygeneration, such as powering thermal desalination processes like multi-effect distillation (MED) or multi-stage flash (MSF) to produce fresh water, particularly valuable in arid, high-solar regions. It can also drive absorption chillers for cooling (trigeneration) or supply industrial process heat and district heating, reducing fossil fuel use. Though not yet widespread in commercial projects, these approaches are being studied and tested to improve the overall efficiency, sustainability, and economic viability of CSP systems (Abdelghafar et al. 2025).

Overall, the current status of CSP is characterised by a growing global footprint of parabolic trough and power tower plants, incremental improvements in thermal efficiency through higher operating temperatures and improved power cycles, and an expanding scope of applications beyond standalone power generation. These developments, from integrating sCO₂ turbines to co-producing water and cooling, aim to leverage the full potential of CSP's thermal output, thereby reducing wasted energy and enhancing the value proposition of concentrated solar power in a low-carbon energy future (Wang et al., 2017; Yousef et al. 2024)

2.2 Current literature status of supercritical carbon dioxide Brayton cycle

The Parabolic Trough Solar Collector (PTSC) represents one of the four available Concentrated Solar Power (CSP) technologies and is currently the most mature and extensively utilised CSP technology in commercial operations. The design of a parabolic trough incorporates a longitudinal receiver or Heat Collection Element (HCE), partially encased by a

curved mirror that focuses the Direct Normal Irradiance (DNI) onto the receiver. Within the HCE, a working fluid serves as the absorption medium for the concentrated DNI, circulating through HCE to either convey heat to a heat exchanger for immediate transfer to the power cycle or to be reserved in thermal storage for subsequent use. Due to the nature of the design, working fluid plays an important role in the system's performance, particularly the temperature of the working fluid, which limits the maximum temperature achievable at turbine inlet and thus the overall efficiency of the system (refer Fig. 2.3). This is due to the fundamental law of Carnot cycle (Equation 2-1), which explicitly states that the maximum reachable efficiency of a thermal engine is based on the hot reservoir, in this case the working fluid. Besides that, higher temperature also leads to higher exergy, or the quality of energy due to its higher difference between the temperature of the working fluid and the dead state (ambient or reference temperature).

$$\text{Carnot cycle energy efficiency: } 1 - T_{\text{cold}}/T_{\text{hot}} \quad (2-1)$$

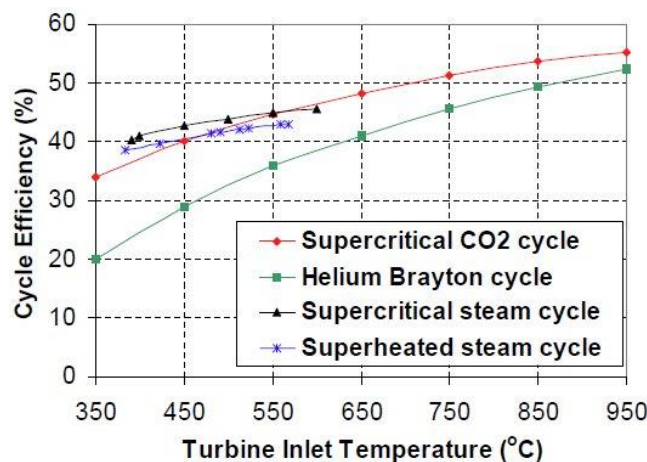


Fig. 2.3 Efficiencies comparisons between power cycles (Moroz et al. 2014)

In contrast, the Supercritical Brayton cycle is recognised as one of the foremost thermodynamic cycles under scholarly examination, and it is notable for its distinctive benefits.

Supercritical fluids offer a theoretically boundless operating temperature range, circumventing the aforementioned constraints imposed by the thermal limits of current working fluids. Moreover, supercritical fluids have additional benefits. Notably, it exhibits hybrid thermodynamic characteristics, embodying a density akin to liquids while maintaining a viscosity comparable to gases. These advantages are important as the former allows for smaller equipment such as turbines and pipes in addition to a smaller land footprint (Fan et al., 2022; White et al, 2022). The latter allows for reduced frictional losses in the system and reduced pumping power all these lead to a lower economic burden. Various supercritical fluids have been investigated in the past such as water, sulphur hexafluoride, and nitrogen by Zaharil (2021), and a comparative investigation of various supercritical fluid Brayton cycles was conducted by Habibi et al. (2020), which studied five alternatives to sCO₂. Coco-Enriquez et al. (2017), also investigated multiple alternatives to sCO₂, namely Ethane, Methane, Xenon and Nitrogen in a supercritical Brayton cycle, with nitrogen performed the best under a wet-cooling system. However, sCO₂ continues to be at the forefront of research due to several important advantages, which are:

- Classified as non-toxic and non-flammable.
- Reduced capital expenditure relative to alternative thermodynamic cycles (Turchi, 2009).
- Achieves high efficiency at elevated temperatures (Turchi, 2009).
- Suitable for use as a direct heat transfer fluid (HTF) in solar receivers (Turchi, 2009).
- Offers a more compact and simplified design compared to Rankine cycles (Turchi, 2009).
- Modular sCO₂ tower receiver/generator benefits from scalability in maintenance and cost (Ma & Turchi, 2011)
- Compatible with thermal energy storage systems (Ma & Turchi, 2011)

From a thermodynamic perspective, sCO_2 possesses a low critical point, facilitating simpler and more cost-effective design and operational practices, alongside less stringent cooling requirements. One of the pioneering suggestions for a supercritical thermodynamic power cycle employing sCO_2 as the working fluid was the Feher cycle, which attained a thermal efficiency of 55% under optimal conditions. Given its thermodynamic efficacy, it was advocated for use as a compact and portable electricity generation system (Feher., 1968). Later on, further studies on the sCO_2 Brayton cycle were scrutinised by a plethora of researchers in conjunction with various heat sources. Most notably, nuclear energy has been used due to its temperature compatibility with sCO_2 critical temperature (Dostal, 2006; Sarkar, 2009).

The integration of sCO_2 Brayton cycle with CSP has been investigated via various researchers in the past. Turchi et al. (2009), conducted an examination of the operational dynamics within a closed-loop sCO_2 recompression Brayton cycle. Their findings indicated that sCO_2 , when used as a working fluid, could potentially surpass traditional systems in thermal efficiency. This superiority was identified in comparison to alternatives such as the helium Brayton cycle and both supercritical and superheated steam cycles. Chacartegui et al. (2011) conducted a comparative analysis of various configurations of supercritical and subcritical solar thermal power plants. They posited that these configurations hold the capacity to rival conventional power generation cycles in efficiency and performance. Garg et al. (2013), examined the trans-critical, sub-critical, and supercritical Brayton cycles within the context of Concentrated Solar Power (CSP). Their research illuminated that the supercritical Brayton cycle demonstrated superior potential relative to its trans-critical and sub-critical counterparts. Singh et al. (2013a) explored a parabolic through direct system employing sCO_2 Brayton technology, employing a dynamic response model underpinned by a mathematical representation of the heat exchanger and turbomachinery. This approach was designed to evaluate the system's sensitivity to fluctuations in air temperatures and solar energy inputs,

aiming to predict its behaviour under various operational scenarios. In their 2013 study, Singh et al. (2013b) conducted an analysis on a parabolic trough direct system utilising sCO₂ Brayton technology. Through the implementation of a dynamic response model, informed by mathematical formulations of the heat exchanger and turbomachinery, the research aimed to discern the system's sensitivity to variations in air temperatures and solar energy inputs, thereby enabling predictions of its performance across diverse conditions. The same author, Singh et al. (2013c) unveiled an innovative control strategy, the extremum-seeking controller, designed to enhance power output from a direct sCO₂ closed-loop Brayton cycle in response to variations in solar heat input and ambient temperatures. This technique primarily adjusted the sCO₂ mass balance within the cycle, demonstrating notable efficacy without the requirement for seasonal retuning, thereby offering a robust solution for year-round operational optimisation. Building up on the promise of higher efficiency of sCO₂ Brayton cycle, Al-Sulaiman & Atif. (2015) conducted a comprehensive examination of a direct solar power tower (SPT) integrated with an sCO₂ Brayton cycle, assessing various configuration efficiencies. Their findings indicated that the recompression configuration emerged as the most robust with relatively consistent efficiency throughout the day. Notably, the regeneration configuration also exhibited significant efficiency, a result that was particularly impressive considering its design simplicity. Padilla et al. (2015) analysed the exergetic performance of four closed-loop sCO₂ Brayton cycles and found that the thermal efficiency of sCO₂ Brayton cycles increases with turbine inlet temperature, with the recompression cycle with main compression intercooling showing the highest efficiency at 850°C. Exergy analysis highlighted the solar receiver as the primary source of exergy loss, while exergy efficiency peaked at temperatures between 700–750°C across different cycle configurations. In 2017, Coco-Enríquez et al. (2017) assessed the efficacy of various supercritical fluids within the Brayton power cycle of a parabolic trough plant. Their research uniquely identified sCO₂ as yielding superior efficiency enhancements

for the plant, particularly at turbine inlet pressures ranging from 250 to 350 bar, thereby surpassing the performance of other supercritical fluids such as nitrogen, ethane, xenon, and methane. A low-level thermodynamic focus study by Wang and He (2017) with an evaluation of an indirect solar power tower system integrated with a reheating sCO₂ Brayton cycle. Their findings suggest that the ideal cycle low pressure, situated between 7.80 and 10.0 MPa, does not necessitate proximity to the critical pressure. Additionally, they noted that while the performance of individual components significantly influences the maximum exergy efficiency, it has a minimal impact on the determination of optimal thermodynamic parameters. Khatoon & Kim (2019) evaluated sCO₂ Brayton cycle layouts for dry cooling in CSP plants, ideal for arid regions, by analysing efficiency, specific work, and solar receiver integration using mathematical models. Enhancing the working fluid's critical temperature with a carbon dioxide and benzene mixture improved efficiency and CSP integration, showing binary mixtures offer significant advantages in drought-prone areas. Huixing et al. (2020) undertook a comparative analysis of CSP systems utilising Rankine, air Brayton, and sCO₂ Brayton cycles. Their results demonstrated that sCO₂ Brayton cycles achieved the highest thermal efficiencies at turbine inlet temperatures above 500°C. Nevertheless, they also highlighted that the energy conversion efficiency of sCO₂ is significantly underestimated when the working pressure is confined to 20 MPa or 30 MPa, illustrating its critical reliance on turbine inlet pressure for optimal performance. An off-design analysis of a direct SPT-sCO₂ Brayton cycle by Yang et al. (2020) and concluded that optimising ambient temperature, energy input to the receiver, and the molten salt's initial temperature enhances the efficiency of the solar receiver, whereas deviations in power demand from the set design parameters lead to an efficiency reduction due to increased molten salt outlet temperatures. Moreover, the system exhibits robust operational capabilities, achieving continuous operation with notable solar-to-electricity conversion efficiencies of 17.8% and 17.1% during the summer and winter solstices, respectively,

underlining its potential for reliable and efficient solar energy utilisation. Khatoon & Kim (2021) proposed a preliminary design and assessment of SPT and compared the regenerative and recompression sCO₂ Brayton cycle. The results indicate an enhanced system efficiency with the implementation of a recompression cycle configuration, exhibiting a variability range of 39% to 45%. The average net power output was calculated to be 37.17 MW for the regenerative cycles and 39.04 MW for the recompression cycles, respectively. Yang et al. (2021) proposed a novel optimisation method of an indirect SPT-sCO₂ Brayton cycle. The novel design method significantly enhances system performance, evidenced by a 6.38% increase in the maximum load cover factor under stable power demand and a 5.62% reduction in the levelised cost of energy. The evident potential of sCO₂ Brayton cycle led to the study of life cycle assessment of the power cycle integrated with parabolic trough by Xiao et al. (2022), incorporating uncertainty analysis. The study shows that auxiliary electricity, thermal energy storage, and solar field manufacture dominate environmental impact and energy use in CSP plants with an sCO₂ cycle, which has a recompression layout superior in reducing greenhouse gas emissions by 21% to 41% compared to the steam Rankine cycle, highlighting its environmental efficiency. Zhang et al. (2022), studied the dynamic characteristics of a CSP plant with sCO₂ to enhance the fast peak shaving capability. The study employs dynamic simulation to explore various control strategies for fast peak shaving in an sCO₂ CSP system, finding that adjusting particle/water mass flow rate offers the quickest response, and optimising turbine rotational speed can enhance net cycle efficiency from 15.30% to 16.12%, with sCO₂ inventory control being key for pressure stability; moreover, synchronously changing mass flow rate and turbine inlet temperature, or maintaining constant temperature while adjusting flow, effectively manage peak demands while ensuring system safety. Zhao et al. (2023) compared three thermal storages configurations and concluded that sCO₂ power generation with compressed CO₂ energy storage outperforms molten salt and combined storage systems

in efficiency (thermal: 6.56%, exergy: 5.76%) and cost-effectiveness, offering a shorter payback period, thus presenting a more viable solution for enhancing solar thermal power's efficiency and economic feasibility. Another off-design conditions work by Lu et al. (2022), which compared the recompression and regeneration sCO₂ Brayton cycle and deduced that the recompression cycle, especially at a specific bypass fraction, outperformed the regeneration cycle under off-design conditions. Another dynamic analysis by Li et al. (2024) investigated the dynamic characteristics of a system with a combined SPT and sCO₂ Brayton cycle with different thermal storage configurations. The study examines two-tank molten salt TES (TT-TES), packed-bed TES with solid fillers (PBS-TES), and packed-bed TES with phase change materials (PBP-TES). It was concluded that during the vernal equinox, the daily average system efficiencies for CSP configurations with TT-TES, PBS-TES, and PBP-TES are 26.0%, 25.5%, and 24.5%, respectively. Additionally, packed-bed TES systems significantly reduce material costs and TES volume. Compared to TT-TES, PBP-TES and PBS-TES reduce costs by 21.2% and 42.3%, and decrease TES volume by 83.0% and 63.8%, respectively.

2.3 Current literature status of Organic Rankine cycle

Globally, the Organic Rankine Cycle (ORC) market is a small but established niche that turns low-temperature heat into electricity in industry and renewable projects (Wieland, et al. 2023). Independent surveys count more than 2,700 ORC plants in operation with a little over 4 gigawatts of combined capacity by 2020; the supplier base includes firms such as Turboden, which reports about 450 plants across more than 50 countries (Mitsubishi Heavy Industries, 2024; Wieland et al., 2022, 2023). Industry estimates place the value of the ORC market at roughly USD 0.8–0.9 billion in 2024, with most forecasts pointing to around USD 1.1–1.25 billion by 2030–2032; Asia–Pacific currently holds the largest regional share (Grand View Research, 2024; Fortune Business Insights, 2024). Activity is strongest in projects that recover

heat from factories, biomass plants, and geothermal fields, although new geothermal power additions slowed in 2023 (REN21, 2024; Wieland et al., 2023). The current state of the global ORC market can be seen in Fig. 2.4.



Fig. 2.4 Global Organic Rankine Cycle market size by application, 2023–2033 (Market.us, 2024).

The Organic Rankine Cycle (ORC) exhibits superior thermal efficiency compared to traditional Rankine cycles, particularly when utilising heat sources with temperatures under 300°C (Zhang et al., 2021; Xu et al., 2019). Moreover, its system design allows for a more compact configuration and reduced installed capacity, enhancing system flexibility. This attribute is advantageous for the development of distributed and miniaturised power generation systems (Refer to Table 2.1) (Usman et al., 2020). A simple Organic Rankine Cycle (ORC) layout comprises an evaporator, condenser, pump, expander, and power generator, as depicted in Fig. 2.5. Within the evaporator, the refrigerant is transformed into a high-pressure superheated vapour. This vapour subsequently undergoes expansion in the expander, a process that converts thermal energy into mechanical power. The gaseous refrigerant, now at low

pressure, condenses into a liquid in the condenser. It is then circulated back to the evaporator by the pump, thus completing the thermodynamic cycle. The generated mechanical power is converted to electrical power via a coaxial generator. The difference between organic Rankine cycle and the traditional steam Rankine cycle is that the former makes use of organic fluids, which have lower boiling points and thus enabling the usage of low-grade heat such as waste heat from thermodynamic cycles (Chowdry & Ehsan, 2023).

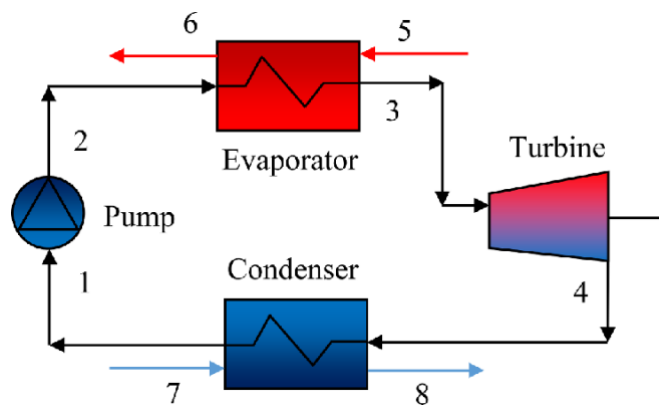


Fig. 2.5 Typical layout of an Organic Rankine Cycle (taken from Elahi et al. 2022)

Table 2.1 Organic Rankine Cycle advantages

Feature	ORC	Steam Rankine
Low temperature of the hot source	ORC can operate efficiently with low-grade heat (e.g., 80 – 300°C).	Steam Rankine needs higher-temperature sources to be efficient.
Low evaporating temperature and pressure	ORC works with much lower pressures.	Steam Rankine typically runs at much higher pressures.
No need for superheating	Often unnecessary for ORC fluids.	Steam Rankine usually needs superheating to prevent turbine blade erosion from wet steam.
Simple evaporator	Yes, because of low temperatures and pressures.	Steam Rankine evaporators (boilers) are more complex.
Low vapour temperature in turbine	Reduces material stress in ORC.	Steam Rankine turbines handle much higher steam temperatures.

High condensing pressure	ORC condenses at higher pressures, so larger vacuum systems are not needed.	Steam Rankine condenses at low pressures, requiring condensers and vacuum systems.
Compact construction	ORC fluids have high densities, allowing for smaller components.	Steam systems are bulkier for the same power output.
No need for water treatment and degassing	ORC uses organic fluids that do not cause scaling/corrosion like water/steam.	Steam Rankine requires careful water chemistry control.
Simpler expanders	Yes, because of lower pressures and temperatures.	Steam turbines are more complex.

Additionally, there are recent attempts at using computer aided research methods to unlock the potential of next-gen ORC by focusing on the development of next-gen working fluids. Computer-aided molecular design (CAMD) can be used to improve Organic Rankine Cycle (ORC) systems by creating tailor-made, environmentally friendly working fluids with optimal thermophysical properties. Instead of choosing from existing fluids, CAMD generates and screens new molecular structures using group-contribution and thermodynamic models (such as SAFT), then evaluates their performance within ORC simulations. This integrated molecular-and-system design approach enables simultaneous optimisation of both the fluid and the cycle, aiming for higher efficiency, lower cost, and reduced environmental impact in applications like waste-heat recovery and renewable heat utilisation (Markides et al. 2025).

2.3.1 ORC utilising low-grade heat sources.

Due to the aforementioned advantages, many attempts were made to use ORC with low-grade heat. Aghahosseini and Dincer (2013) conducted a comparative analysis of pure and zeotropic fluids suitable for low-grade heat sources, focusing on their cycle performance. They evaluated fluids including R245fa, R134a, R123, R404a, R600a, and R407c, categorizing R123, R600a, and R245fa as dry, and R134a, R404a, R143a, and R407c as wet. Given the ORC's

operation at temperatures below 200°C for heat extraction, the study highlighted the limited availability of isentropic pure working fluids within this thermal range. It has been observed that the exergetic efficiency decreases with an increase in the working fluid's boiling point. In parallel to analysing fluid properties, studies such as those by Mago et al. (2007) also noted an increase in irreversibility and a decrease in exergy efficiency when dry operational fluid is superheated in ORC, further identifying that higher turbine pressures can reduce the demand for external heat due to a corresponding decrease in mass flow rate. The same research of Aghahosseini and Dincer (2013) determined that while energy efficiency remains relatively stable with changes in turbine pressure, exergetic efficiency significantly decreases as turbine pressure increases. This finding suggests that, unlike water, organic working fluids in ORC benefit from operation in a saturated state without the necessity for superheating. Additionally, other research such as from Yamamoto et al. (2001), Somayaji et al. (2006), Gurgenci (1986), and Hung (2001) also highlighted in their studies that the entry of the working fluid into the turbine in a saturated state is crucial for enhancing the ORC's performance by reducing the cycle's overall irreversibility. Further exploring the efficiency of ORC systems, Ozdemir (2018) evaluated five organic working fluids, namely R114, R113, R245fa, R600a, and R227ea, within both basic and regenerative ORC, concluding that the regenerative ORC outperformed the basic configuration. Among these, R113 was identified as yielding the highest efficiencies, whereas R227ea was found to be the least efficient. Wang et al. (2011) evaluated the performance of nine pure fluids, which are R123, R113, R141b, R11, R245ca, R245fa, R236ea, and Butane, using two thermodynamic models for reclaiming waste heat from vehicle engines via ORC, featuring both a single screw expander (model A) and an integrated internal heat exchanger (model B). Their findings highlighted superior thermodynamic performance from R123, R113, R141b, and R11, while R245ca and R245fa were noted for their environmental benefits. The analysis emphasised the significance of condensation temperature relative to

evaporation pressure in ORC efficiency. A summary of optimal operational parameters and outcomes for these fluids in harnessing low-quality heat is provided. Liu et al. (2004) analysed the influence of operational fluids on ORC efficiency, focusing on both energetic and exergetic efficiencies. They proposed a methodology for evaluating the slope of the ORC T-s diagram, which serves to categorise working fluids as dry, isentropic, or wet, based on the sign of ε in the equation, where $\varepsilon > 0$ indicates dry, $\varepsilon \approx 0$ signifies isentropic, and $\varepsilon < 0$ denotes wet fluids (Refer Fig 2.4). Efficiency comparisons among various working fluids revealed that those with hydrogen bonds, like ethanol, water, and ammonia, have a higher enthalpy of vaporisation, classifying them as wet and less preferred for Organic Rankine Cycle (ORC) applications. In addition, the authors identified the optimal vaporisation temperature for maximising exergetic efficiency, noting that while toluene exhibits superior thermal efficiency, R123 excels in heat recovery due to its lower critical temperature. Zarrogiannis et al. (2017) found that a 70% Neopentane/2-Fluoromethoxy-2-methylpropane mixture outperformed other fluid mixtures in ORC control potential. Chacartegui et al. (2009) demonstrated that cyclohexane and toluene yield higher efficiencies in combined systems leveraging low-temperature sources. Hung (2001) compared dry working fluids like R113, R123, Benzene, p-Xylene, and Toluene, highlighting p-Xylene's high efficiency for high-temperature waste heat recovery, while R123 and R113 were more effective for low-temperature sources, emphasising the influence of heat source quality on irreversibility. Desai and Bandyopadhyay (2009) proposed a cogeneration approach using ORC, observing a significant efficiency improvement with perfluoro-n-pentane showing the most substantial increase, advocating n-pentane as the optimal fluid for standalone ORC systems. Pedro et al. (2008) found that regenerative ORCs exhibit improved thermal performance when utilising dry fluids like R-123, R-113, isobutane, and R245Ca.

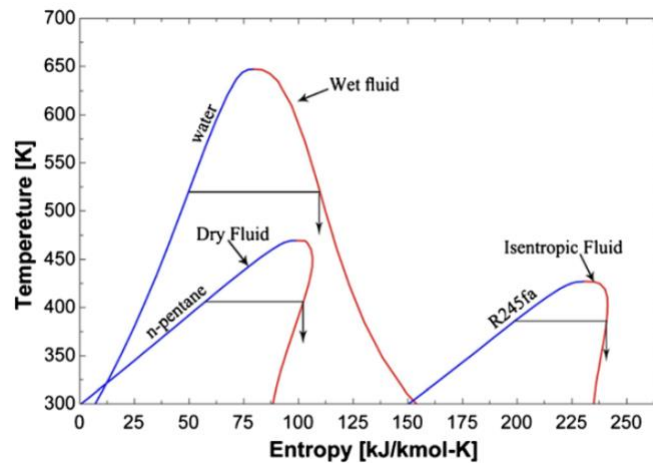


Fig. 2.6 Temperature–entropy diagrams showing the expansion behaviour of wet, dry, and isentropic working fluids in an ORC. (Bahrami et al. 2013)

2.3.2 ORC coupled with concentrated solar power (CSP)

In this sub-section, CSP will focus on parabolic trough solar collector (PTSC) technology due to the scope of the research utilising PTSC and due to an overabundance of research in this area over other CSP technologies and thus, a focused literature review is necessary. PTSCs are distinguished for their capability to operate up to 400°C with thermal oil and 600°C with molten salt, making them a preferred choice for their maturity, cost-effectiveness, and efficiency. Coupling PTSC with Organic Rankine Cycle (ORC) technology, which is compatible with a wide range of organic working fluids, offers versatile configurations for heat transfer and polygeneration systems (Loni et al., 2021). Delgado-Torres and García-Rodríguez (2010) analysed a parabolic trough solar collector (PTSC) system for low-temperature applications coupled with desalination, identifying isopentane as the best fluid following a techno-economic assessment. Quoilin et al. (2011) introduced a steady-state model and design for a compact ORC aimed at rural electrification in Lesotho, offering detailed sizing for the system's various components. Additionally, simulations of both single and double-stage expanders were conducted. The analysis indicated that an expander with an electro-mechanical efficiency below 60% could achieve an overall electrical efficiency of between 7% and 8%.

He et al. (2012) used TRANSYS to simulate a PTSC-ORC system, analysing variables like interlayer pressure, HTF flow rate, solar intensity, and incidence angle. Findings showed that heat loss from the collector increases with pressure up to 10 Pa, then stabilises at higher pressures. Uzgoren and Timur (2015) conducted a comparative study on a PTSC-ORC system featuring both dry-cooling and wet-cooling condensers, utilising Cyprus' climatic data for their analysis. Their findings indicated that the wet-cooling condenser system produced 27% more energy than its dry-cooling counterpart, though the latter conserved 18.7 tons/MWh of water. Furthermore, integrating hybrid systems in solar energy applications can mitigate reliability issues associated with solar's variable nature by combining it with more consistent energy sources, such as biomass.

Mohtaram et al. (2002) explored a hybrid system harnessing solar and wind energies for heating, cooling, power, and desalination, finding increased freshwater output with solar radiation, exergy efficiency up to 77.27% with higher turbine pressure, and reduced costs with elevated ambient temperatures. Taccani et al. (2016) used a low-temperature parabolic trough collector and identified R245fa as the ideal fluid for its system, achieving an overall efficiency of 8%. Muñoz et al. (2017) proposed an innovative PTSC solar-based Rankine-Brayton system for electricity generation, pinpointing propane and R125 as the optimal fluids. Bellos and Tzivanidis (2017) enhanced the ORC's power output by experimenting with various organic and nanofluids. Caldino-Herrera et al. (2017) investigated a PTSC-ORC system with dual storage (hot and cold) and an additional heater, finding that with R245fa as the working fluid, the system achieved an efficiency of about 8%. Erdogan et al. (2017) investigated the integration of CSP systems with geothermal-based ORC using a shell-and-tube heat exchanger, identifying baffle spacing as a critical factor in reducing heat transfer area and pumping power. Their comparison of 10 PTSC fluids and 21 ORC fluids found that Dowtherm A was optimal for minimising both parameters. Bassetti et al. (2018) developed a hybrid geothermal-CSP

facility incorporating a PTSC, ORC, and TES unit. This design pre-heats geothermal fluid via the CSP system before it enters the ORC evaporator, with TES storing surplus CSP energy for night-time use. The addition of PTSC resulted in a 2.8% increase in electrical output compared to standalone geothermal-ORC setups. Heng et al. (2019) evaluated a small-scale cogeneration system in Malaysia's equatorial region, analysing its performance across May, June, and July 2018. Solar energy availability was recorded at 82,809 kWh, 77,592 kWh, and 64,239 kWh for these months, respectively, resulting in net electricity generation of 6373 kWh, 6201 kWh, and 5155 kWh. The system achieved a solar-to-electricity efficiency of approximately 8%, aligning with findings from prior research. Al-Sulaiman et al. (2011) evaluated the exergy efficiency of a parabolic trough-ORC CCHP system against a solely electrical power-producing system across three operational modes: solar, combined solar and storage, and storage only. Findings indicated that the CCHP system's exergy efficiency outperformed the single-output system by margins of 13%, 4.5%, and 4% for each mode, respectively. Al-Sulaiman et al. (2012) investigated a CCHP system integrated with a PTSC and an ORC, comparing its energy efficiency against three configurations named after their outputs, which are electrical power, cooling-cogeneration, and heating-cogeneration, across three operational modes, namely solar, solar with storage, and storage. The CCHP system demonstrated superior energy efficiency in all modes, with efficiencies in the solar mode reaching 13%, 16%, 88%, and 91% for the electrical, cooling-cogeneration, heating-cogeneration, and CCHP configurations, respectively, highlighting the CCHP's dominance. Tzivanidis et al. (2016) used PTSC and ORC power cycle and explored the optimisation of an ORC integrated with PTSC for sustainable electricity generation, focusing on both financial and energetic efficiency. Through a thermodynamic analysis of various working fluids and a comprehensive assessment of system configurations, including solar field size and storage capacity, cyclohexane emerged as the optimal working fluid for achieving a 1 MW_e output. The ideal system configuration consists of a 25,000 m^2

solar field and a 300 m³ storage tank, yielding a 9-year payback period and an internal rate of return (IRR) of 13.46%. Chacartegui et al. (2016) evaluated toluene, cyclohexane, and siloxane D4 as working fluids in a parabolic solar power plant with direct and indirect thermal storage, determining toluene as the optimal choice both energetically and financially for systems utilising Therminol VP-1 at temperatures up to 400°C. Cyclohexane was ranked second, with siloxane D4 deemed the least favourable option. Ashouri et al. (2018) conducted an exergo-economic optimisation of a double-pressure PTSC-ORC system incorporating a storage tank, achieving a peak exergy efficiency of 22.7%. Majumdar et al. (2018) explored a new solar-ORC system employing a dynamic moving boundary model tailored for medium-temperature collectors, resulting in a time-dependent model for heat transfer fluid temperature variation. The system's overall efficiency was determined to be 21%. Ni et al. (2018) examined a solar-ORC system's performance under cloudy conditions with a dynamic model, proposing a PID control strategy that enhanced system efficiency by 24%.

Some comparative studies were also conducted, such as from Desai and Bandyopadhyay (2016), which conducted a financial and energetic comparison between water/steam Rankine Cycles (RC) and ORC utilising PTSC and Linear Fresnel Reflectors (LFR). They determined that the Levelized Cost of Electricity (LCOE) for water/steam RC was \$0.353/kWh with PTSC and \$0.422/kWh with LFR, compared to \$0.344/kWh for the PTSC-ORC system. Rodríguez et al. (2016) compared thermocline thermal storage and two tanks in a PTS-ORC system. They found that the thermocline storage systems are more favourable over two-tank systems for CSP-ORC plants, owing to their comparable thermal efficiency but at approximately 30% lower costs, thereby enhancing the economic viability and efficiency of CSP-ORC facilities, as exemplified by the 1 MWe CSP-ORC project in Benguerir, Morocco. Recently, Boukelia et al. (2023) detailed a dual heat source double cycle system with a topping Rankine cycle and a bottoming ORC coupled with PTSC with various wet ORC fluids, namely ammonia, R32,

R290, and R143a. The configuration enhanced the annual net output to 22.20, 20.02, 19.25, and 18.67 GWhe, respectively, achieving a power generation increase of over 19.36% compared to a reference stand-alone solar plant.

2.4 Direct contact membrane distillation (DCMD)

Seawater desalination has emerged as a crucial strategy to address water scarcity in areas with scarce natural water resources. However, desalination operations are energy-intensive and traditionally rely on fossil fuels, contributing to high costs and environmental concerns related to climate change (Sharon & Reddy, 2015). Desalination technologies are broadly classified into membrane-based and thermal-based methods. Reverse osmosis (RO) is the prevalent technology within the membrane-based category, while multi-stage flash (MSF) and multi-effect desalination (MED) are the leading applications in the thermal-based category. Each of these technologies, however, is associated with significant limitations, which are detailed in Table 2.2.

Table 2.2 Advantages and disadvantages of various state-of-the-art desalination technologies (Misdan et al. (2013) as cited in Al-Obaidi et al. (2022) & Ghazi et al. (2022))

Technologies	Advantages	Disadvantages
MSF (Multi-Stage Flash) Desalination	• Features systems for both heat recovery and rejection. • Minimal risk of plugging and scaling. • Performs well with water of high salinity. • Delivers water of superior quality compared to RO. • Generates a greater volume of freshwater than RO.	• Necessitates substantial heat energy. • Increased risk of scaling and corrosion at elevated temperatures. • Elevated costs of water production. • Demands intensive energy consumption. • Involves costlier equipment compared to RO.

		• Occupies more space than RO systems.
MED (Multi-Effect Distillation) Desalination	• Exhibits reduced scaling and corrosion risk compared to MSF. • Capable of utilising hot water in addition to steam. • Requires lower steam temperatures compared to MSF. • Efficient in processing water with high salinity. • Yields water of superior quality and greater freshwater output than RO. • Consumes less energy than MSF. • Offers greater heat transfer efficiency than MSF.	• Requires significant heat energy. • Incurs high costs for water production. • Consumes more energy per unit than RO. • Utilises costlier equipment than RO. • Demands more space than RO systems.
RO (Reverse Osmosis) Desalination	• Functions at ambient temperatures without the need for steam or heat recovery. • Presents a very low risk of scaling and corrosion. • Eliminates the requirement for heat in the desalination process. • Offers lower water production costs than thermal methods. • Less energy-demanding than thermal approaches. • Features more compact equipment. • Involves a lower initial investment than thermal methods.	• Less effective with extremely high salinity due to osmotic pressure challenges. • Susceptible to plugging and fouling. • Generates less freshwater compared to thermal methods. • Necessitates high-pressure pumps and systems. • Requires thorough pre-treatment to eliminate colloidal substances.

However, currently there is an emerging technology that merges the two aforementioned categories of desalination techniques. The technology is termed membrane distillation technology (MD). The fundamental science behind the technology is explained as follows.

Membrane Distillation (MD) is a thermal separation technique that utilises a microporous hydrophobic membrane to distil pure water from a high-salinity solution by exploiting the vapour pressure difference across the membrane. The process heats the saline feedwater, causing water vapour to pass through the hydrophobic membrane to a colder side, where it condenses into pure water, leaving salts and impurities behind. This separation is driven by heat and mass transfer mechanisms, including conduction, convection, latent heat transfer, and vapour diffusion, influenced by the membrane's thermal conductivity, flow rates, and the vapour pressure gradient. The efficiency of MD is significantly determined by the membrane's properties and the temperature gradient across it, highlighting the process's reliance on both the physical and thermal characteristics of the system for effective desalination (refer Fig. 2.5)

MD has significant advantages over MSF and MED, such as Membrane distillation (MD) offers several advantages over conventional thermal desalination processes such as multi-stage flash (MSF) and multi-effect distillation (MED), including lower operating temperatures, reduced sensitivity to feedwater salinity, and the ability to utilise low-grade or waste heat sources. Furthermore, MD systems are generally more compact, modular, and capable of achieving high salt rejection without requiring high operating pressures, making them particularly suitable for integration with renewable energy systems.

When integrated with an sCO₂ Brayton cycle, membrane distillation (MD) can exploit the low-grade waste heat rejected near the fluid's critical point, enabling simultaneous electricity generation and freshwater production without substantially affecting the cycle's efficiency. In addition, MD's smaller footprint compared to MED and MSF complements the inherently compact design of sCO₂ power blocks, while its ability to operate at much lower temperatures than MED and MSF aligns well with the moderate thermal conditions naturally available in sCO₂ cooling stages. Based on the IAEA's guideline that thermal desalination plants such as MSF and MED require approximately 1–2 m² of plot area per m³/day of capacity, a 1,000

m^3/day unit typically occupies about 1,000–2,000 m^2 (International Atomic Energy Agency, 2000). In contrast, modular AGMD/VMEMD systems with typical fluxes of $3\text{--}10 \text{ L}\cdot\text{m}^{-2}\cdot\text{h}^{-1}$ can achieve the same capacity within only a few hundred square metres (Han et al. 2021), representing an estimated footprint reduction of roughly 50–85 % ($\approx 2\text{--}8\times$ smaller). MD also operates at much lower temperatures and pressures, aligning naturally with the moderate-grade waste-heat streams available in sCO_2 cycles (Abid 2023). Moreover, emerging configurations such as PGMD enhance internal heat recovery and energy efficiency (Rupakheti et al. 2025), while innovations in ceramic membrane materials promise added robustness for high-temperature waste-heat utilisation (Alessandro & Macedonio 2025). These factors collectively support MD as a highly compatible desalination partner for sCO_2 -driven CSP systems.

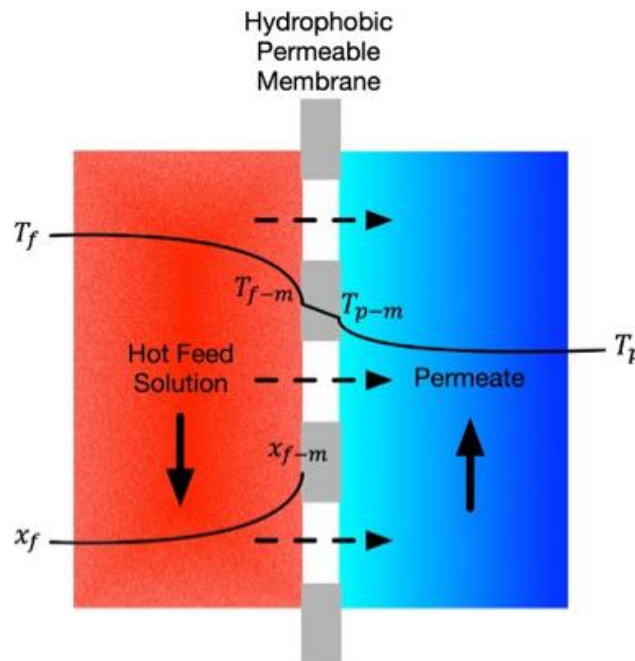


Fig. 2.7 Membrane distillation (MD) fundamental science schematic diagram (Omar, 2022)

2.4.1 Desalination integration with concentrated solar power (CSP)

The scholarly literature features numerous studies exploring the integration of desalination technologies with CSP across diverse operating and climatic conditions and global regions. Among the early works along this line is by Trieb et al. (2003), who conducted a study on the

integration of parabolic trough and Fresnel plants with thermal desalination systems in the arid environment of Dubai. The research indicated that the CSP plant could yield an annual electricity output of 1.5 billion kWh and produce 60 million m³ of freshwater annually. This capacity is sufficient to meet the electricity requirements of 250,000 individuals and supply clean water to 50,000 people. Moser et al. (2010) explored the coupling of a Multi-Effect Distillation (MED) desalination system with two CSP technologies, namely, parabolic trough collectors and linear Fresnel reflectors in Mediterranean conditions. Through simulation of technical performance on an hourly basis over a year and financial modelling to assess net present value, the study confirmed the economic viability of CSP-desalination systems. These systems offer stable energy and water prices, providing a secure alternative to the volatile costs and availability associated with fossil fuels. Olwig et al. (2011) examined the integration of a low-temperature multi-effect distillation (LT-MED) condenser and a reverse osmosis (RO) plant with parabolic trough solar plants using wet cooling in the MENA region's climatic conditions. Their analysis revealed a direct correlation between water pricing and electricity costs, demonstrating that with a renewable energy feed-in tariff of approximately \$0.24/kWh, freshwater production can be economically achieved at \$1/m³ at the Aqaba site. Delgado-Torres and García-Rodríguez (2012) explored and advocated for various design configurations combining a CSP plant, a PTSC solar field, an ORC, and an RO system. Their findings highlight the necessity for precise RO unit integration to minimise the solar field's footprint, achieving optimal specific energy consumption rates of approximately 2 kWh/m³. Hassabou et al. (2013) analysed desalination plants (5000 m³/day) using MSF and RO technologies alongside a PTSC plant in Marsa-Matruh, Egypt. Their comparison showed that water costs from a PTSC-RO setup were competitive with conventional fossil fuel systems, whereas PTSC-MSF costs were significantly higher, nearly tripling those of fossil fuel desalination systems. Fylaktos et al. (2014) conducted an economic analysis of a CSP plant coupled with

desalination, exploring configurations for electricity generation alone and combined with RO or LT-MED desalination. Utilising the Levelized Cost of Electricity (LCOE) and accounting for revenues from electricity, freshwater sales, and carbon emission credits, the study found CSP integrated with desalination to be economically viable in all setups, with electricity-only configurations yielding the best financial results. Casimiro et al. (2014) developed new simulation techniques for freshwater and electricity production using PTSC and forward feed-MED systems, employing NREL's SAM software. Tested in San Diego with various cooling methods, the research indicated the cost-effectiveness of integrating PTSC with MED/seawater cooling circuit. Weiner et al. (2015) analysed a hybrid RO-MED system for producing 7600 m³/day of fresh water in California, utilising a parabolic trough. The system, which integrated ORC for electricity to drive the RO unit and used ORC's exhaust steam for the MED process, achieved a 41% reduction in the LCOE compared to a conventional MED system with efficient RO utilisation. Mohan et al. (2016) created and assessed a novel solar thermal poly-generation system featuring membrane distillation, using flat plate collectors, Evacuated Tube Collectors (ETC), and compound parabolic collectors for evaluation. Economic assessments revealed that the ETC configuration offers the shortest payback period of 9 years compared to the alternatives. Talebbeydokhti et al. (2017) designed a small-scale Ericsson cycle with a PTSC and MED system, achieving a 260 kW output using air as the heat transfer fluid. They examined different stages (7, 9, and 11 effects) and inlet steam temperatures (70°C - 85°C), including seawater preheating scenarios. The study identified the 11-effect configuration at 70°C as optimal, offering the highest freshwater production and minimising scaling and corrosion. Freshwater production increased from 1.15, 1.44, and 1.73 kg/s to 1.21, 1.53, and 1.84 kg/s with preheating for each respective stage setup. Hoffmann and Dall (2018) explored a 100 MWe SPT integrated with a MED system in Namibia, demonstrating its capability to fulfil local freshwater demands and economic feasibility under the region's renewable energy

and water pricing framework. Lai et al. (2019) developed a hybrid energy system combining a Solar Stirling Dish engine with a salinity gradient energy storage system, achieving optimal and real efficiencies of 9.23% and 5.07%, respectively. Ghorbani et al. (2020) investigated a PTSC solar field-driven hybrid system of MED and ORC for cogeneration, producing 459.9 MW of electricity and 3628 kg/h of water, with efficiencies noted at 12.47% for the ORC and an exergy efficiency of 87.11% for the system. Wang et al. (2021) conducted a study on a novel solar power and water-making system that employs the super-critical carbon dioxide (S-CO₂) Brayton cycle for power generation and multi-stage flash (MSF) desalination for water purification. By designing and simulating this system, they found it could produce electricity with an efficiency of 36.6% and generate 50.1 megawatts of power and 4050.8 tons of fresh water daily. Their research highlights this system's potential as a cost-effective and efficient solution for simultaneous power generation and desalination, with an electricity cost of \$0.059 per kWh and water cost of \$1.15 per ton. Lastly, Jamshidian et al. (2022) conducted a techno-economic study on a PTSC-RO-MED unit powered by solar energy in Iran's Bushehr province, finding a 15% total recovery increase in hybrid mode compared to a standalone MED unit and reducing freshwater production costs from \$2.08/m³ to \$1.918/m³ for 6700 people by integrating RO with MED.

2.4.2 Membrane distillation (MD) integration with concentrated solar power (CSP)

As discussed in section 3.1, various examinations into the combination of CSP and desalination systems were investigated, and promising results were obtained. However, very few studies have shown the integration of CSP with membrane distillation (MD) desalination technology. Guillén-Burrieza (2011), under the MEDESOL project, evaluated an Air Gap Membrane Distillation (AGMD) system powered by solar energy, demonstrating its capacity to efficiently produce up to 7 litres per hour per square meter of distilled water. Employing an experimental methodology that involved testing under varying solar conditions without heat

storage, the project explored the system's performance across different operational parameters, including salt concentrations and feed temperatures. Results highlighted not only the system's promising water production capabilities but also identified scalability and energy consumption as critical areas requiring optimisation to enhance the AGMD system's efficiency and broader applicability for sustainable water solutions in small communities. Andrés-Mañas et al. (2020a) conducted a pioneering experimental assessment at the pilot scale of vacuum-enhanced air-gap membrane distillation (V-AGMD) at the Plataforma Solar de Almería, employing two commercial spiral-wound modules with distinct channel lengths (1.5 and 2.7 meters), correlating to different membrane surface areas (7.2 and 25.9 square meters) and envelope counts. Implementing a vacuum to extract air from the gap markedly enhanced vapour transfer and module efficiency, notably for high-salinity feeds, with permeate flux increasing by up to 234% and specific thermal energy consumption reducing by 68%. The vacuum application yielded unprecedented results, optimal permeate productivity ($8.7 \text{ l h}^{-1} \text{ m}^{-2}$) with the shorter channel and superior energy efficiency (49 kWh m^{-3} , equating to a GOR of 13.5) with the longer. These outcomes represent the most effective performances recorded at the pilot scale in membrane distillation to date. The same author proposed an innovative solar-powered seawater desalination system utilising vacuum multi-effect membrane distillation (V-MEMD) and solar thermal collectors, which was evaluated. This system efficiently condenses steam and preheats feed with seawater, demonstrating stable operation under variable solar conditions. Simulation predicts annual distilled water production between 41.7 and 70.5 m^3 , with consistent output across temperatures, peaking at an average flux of $(5.5 \pm 1) \text{ l h}^{-1} \text{ m}^{-2}$ at 80°C (Andrés-Mañas et al, 2020b). Later, the authors investigated a solar-powered desalination system using vacuum multi-effect membrane distillation (V-MEMD), which was evaluated for its efficiency in steam condensation and feed preheating. Simulations of this system, under varying solar conditions, forecast an annual distilled water output of 41.7 to 70.5 m^3 , achieving

optimal production rates of $(5.5 \pm 1) \text{ l h}^{-1} \text{ m}^{-2}$ at 80°C (Andrés-Mañas & Zagaroga, 2023). One study by Inkawhich et al. (2023) introduced an integrated Air-Gap membrane distillation (AGMD) and CSP/PV desalination system designed for zero-waste operation in isolated areas and experimentally evaluated under real-world conditions for 9 months. Capable of producing up to 288 L of water daily, the system's efficiency varies with operational adjustments and environmental conditions. Results underline its potential as an independent solution for sustainable water production.

A few studies that focus on the large-scale application of CSP-MD combinations that make use of thermodynamic cycle and waste heat. For example, Soomro & Kim (2018a) explored the integration of a 50 MWe PTSC plant with a direct-contact membrane distillation (DCMD) system in Abu Dhabi for simultaneous electricity and freshwater production, employing SAM and MATLAB® for performance and economic assessments. Findings indicated electrical output ranged from 7.71 GWh in December to 13.5 GWh in May, with cooling water requirements varying seasonally, and demonstrated the DCMD's enhanced efficiency and flux with increased feed temperatures, achieving freshwater production up to $14.33 \text{ m}^3/\text{day}$ at $\$0.64/\text{m}^3$. Integration offers a sustainable, cost-effective solution to meet the growing demands for freshwater and electricity. The same authors did similar research using similar methodology for a solar power tower (SPT) of 111 MWe and found that the energy cost was 13.03 cents per kWh. Additionally, the DCMD system, validated against experimental data, showed improved permeate flux and evaporation efficiency with higher feed water temperatures, leading to an average freshwater production of $40,759 \text{ L/day}$ at a cost of $\$0.392/\text{m}^3$ (Soomro & Kim, 2018b). In the same year, the same research as mentioned, but this time employing a 111MWe Fresnel lens solar collector, was investigated and concluded that the system managed to achieved energy outputs ranging from 14.08 GWh in December to 38.33 GWh in June, at a real levelised cost of 0.34 ¢/kWh . Additionally, the DCMD unit showed increased evaporation efficiency

from 39.13% to 50.01% with feed temperature elevation from 30°C to 45°C, resulting in an average freshwater production of 31,844.6 L/day at a cost of \$0.425/m³ (Soomro & Kim, 2018c). Building on the previous work, Soomro et al. (2020) compared the three aforementioned CSP technologies and concluded that solar power tower (SPT) configuration achieved the highest electricity output (353.87 GWh) and capacity factor (56.1%), while the parabolic-trough (PT) plant reported the lowest LCOE (4.51 cents/kWh). In terms of desalination, the PTSC plant integrated with DCMD maximised freshwater production (38.9 m³/day), with the SPT configuration offering the lowest water production cost (0.314 USD/m³). A research by Ouali et al. (2023) evaluated the performance and economic viability of integrating SPT plants with DCMD units across six coastal MENA locations, notably achieving high energy output (586.28 GWh) and capacity factor (67%) in Sharm el-Sheikh. Cost-effective water production (<0.5 \$/m³) was identified in Aden, Dammam, Sharm el-Sheikh, and Salalah. Using a 100 MWe SPT with direct thermal energy storage and a DCMD system, validated against Spain's Gemasolar facility, the study highlights SPT-DCMD's potential for providing fresh water and electricity in arid regions, with a nominal LCOE of 11.2 ¢/kWh. The same researchers also conducted similar research in the Moroccan climate, and this time by integrating PTSC with DCMD. This integrated system, particularly effective in Dakhla, southern Morocco, achieved an annual energy generation of 493.8 GWh and a capacity factor of 47.6%, with a Levelized Cost of Electricity (LCOE) at 10.63 cents/kWh. The study found that increasing feed water temperature in the DCMD system enhances permeate flux, with freshwater production costs varying across regions, being lowest in Dakhla at \$0.84/m³ and highest in Agadir at \$1.20/m³, indicating seasonal and regional variations in freshwater production efficiency (Ouali et al. 2024)

2.5 Literature review conclusion, research gap identification, and objectives.

It is evident from the current body of literature that several issues have hampered the wider commercial adoption of concentrated solar power (CSP). Some of the most important issues that need to be addressed towards higher technical and economic efficiency of CSP systems, as outlined in the literature section, are:

- 1) Elevating the operational temperature of heat transfer fluids to surpass the limitations of current working fluids.
- 2) Adoption of advanced thermodynamic cycles to surpass the limitations of the current steam Rankine cycle.
- 3) Enhancing the efficiency of heat-collecting elements or the receiver tube in CSP.
- 4) Utilising waste heat.
- 5) Minimising exergy losses within the system.

The research gaps in the academic literature that are being addressed by this thesis are summarised below:

Research Gap 1: Unexplored CSP with advanced power cycles with Membrane distillation (MD) utilisation as the means of waste heat utilisation despite its obvious advantages as outlined in section 2.4.1.

Research Gap 2: While direct integration of CSP to power cycle is not a new idea, the detailed methodology on how to design it is underexplored. Most of them rely on assumptions. The integrations in a direct manner from the solar field, particularly PTSC, into a power cycle are detailed for the maiden time in the literature.

Research Gap 3: While waste heat utilisation from power cycles has been extensively investigated, a detailed methodology for maximising heat recovery from the topping cycle to the bottoming cycle remains lacking. In particular, there are limited methods and a focus on

achieving this without causing either excessive superheating of the ORC working fluid or excessive wetness after turbine expansion, as neither condition offers a clear operational advantage, as explained in section 2.3.

To fill in the three major gaps in the literature as outlined above, the objectives of this research are as follows:

Research objective 1. Develop and validate an integrated direct CSP system model that links a parabolic trough solar collector (PTSC) with a recompression sCO₂ Brayton cycle, a bottoming Organic Rankine Cycle (ORC), and a direct-contact membrane distillation (DCMD) unit. The model will (i) use a discretised 1-D PTSC formulation, (ii) set out step-by-step design rules for direct coupling from the solar field to the power cycle to reduce guesswork and hidden assumptions, and (iii) track energy and mass flows across all subsystems. Standard economic models (LCOE, LCOW, NPV) are developed to support later assessment.

Research objective 2. Quantify how the integrated system performs across key operating conditions and design choices. This includes pressure ratio, turbine inlet temperature, mass flow rate, collector length, solar irradiance, ambient temperature, and recuperator effectiveness. The analysis will report (a) power output, (b) thermal efficiency, (c) where the main losses happen, and (d) water production from DCMD. The study will also examine how changes in solar input affect stability and output, and identify operating ranges that keep the system both efficient and robust.

Research objective 3. Maximise useful heat recovery from the sCO₂ topping cycle to the ORC without causing either excessive superheating before the ORC turbine or excessive wetness after expansion. To do this, develop and apply a “duality method” that uses both pinch point and approach-temperature checks as joint design rules, and compare the results against a typical single-balance method. The goal is a clear, repeatable method that increases waste-heat use and improves overall system output (power and water).

Research objective 4. Test technical and economic feasibility across different climatic and operating conditions, and plant sizes, and run multi-objective optimisation to guide design. This will cover a wide range of the most important operating conditions of sCO₂, and map 25-year LCOE and LCOW against the most important variables in sCO₂ power cycle, namely net power rating, turbine inlet temperature, and recuperator effectiveness, while accounting for expected cost drops in sCO₂ technology. The optimisation will choose decision variables such as pressure ratio, mass flow rate, key temperatures, collector length, and ORC working fluid, to (i) maximise net power, (ii) maximise water production relative to power output, (iii) raise overall energy use, and (iv) lower LCOE and LCOW, subject to practical limits (material temperature limits, minimum approach temperatures, and Organic fluid's dryness limits). The outputs will be clear design rules for different climates and scales.

Overall, this thesis presents a comprehensive foundational framework for designing and thermodynamic and economic analysis for a 3rd-generation Concentrated Solar Power (CSP) system, addressing key gaps in the literature, including advanced working fluids, next-generation power cycles, and waste heat utilisation for maximising total energy capture and output.

The importance of this thesis lies in its ability to bridge critical gaps in current CSP development by introducing a fully integrated, direct-configuration system that couples a PTSC with an sCO₂ Brayton cycle, ORC, and DCMD for maximised energy and water output, a novelty in the literature. Additionally, it delivers, for the first time, a detailed design methodology supported by a discretised 1-D PTSC model and a duality heat-recovery approach, enabling higher efficiency, improved waste-heat utilisation, and robust techno-economic performance across diverse operating conditions, advancing CSP towards competitive third-generation targets.

CHAPTER 3 METHODOLOGICAL DEVELOPMENT AND INNOVATIONS OF THE PROPOSED SYSTEM

In this chapter, the methods to design and evaluate the objectives of the thesis are outlined. The novelty of certain methodologies employed in this study will be highlighted and discussed in detail at the relevant sections. The research methodology is carried out in the following stages:

- The proposed system description in a detailed manner, including its overall configuration, main components, and operating principles. This section will present schematics, boundary conditions, and key design assumptions to clearly define the scope and physical basis of the study.
- Development of the parabolic trough solar collector (PTSC) and, using a discretised one-dimensional modelling approach with its integration into a power generation system.
- Development of the supercritical carbon dioxide (sCO₂) Brayton cycle, including thermodynamic modelling and its associated component and system equations for economic analysis
- Development of the organic Rankine cycle (ORC) as a bottoming cycle, with both thermodynamic and economic assessments to maximise the utilisation of the heat source.
- Development of the direct contact membrane distillation (DCMD) process, incorporating thermodynamic and economic modelling to evaluate its potential for freshwater production from waste heat.
- Development of the economic models used in this thesis.

3.1 System descriptions

This study investigates a system integrating a direct parabolic trough-driven recompression Brayton supercritical carbon dioxide (sCO₂) cycle with a Direct Contact Membrane Distillation (DCMD) desalination unit. Starting at point 5 (see Fig. 3.2A) Single recompression sCO₂ Brayton cycle and B) Double cycle sCO₂ recompression Brayton cycle and a simple bottoming ORC, heat is supplied from the Parabolic Trough Solar Collector (PTSC) power plant. The working fluid then expands through the turbine and transfers energy sequentially via the High-Temperature Recuperator (HTR) and Low-Temperature Recuperator (LTR), producing two separate flow streams at point 6 in the thermodynamic cycle. One stream undergoes direct compression, while the other is first cooled before being compressed. The heat rejected in this process is recovered to drive the DCMD system, where seawater functions as the cooling medium. All analyses were performed under steady-state conditions, with fluid properties obtained from the EES property library. The residual thermal energy is utilised in a simple Organic Rankine Cycle (ORC) for additional power generation. Heat rejected from the ORC condenser and the cooler is subsequently directed to the DCMD unit to produce fresh water. Further details of the sCO₂ recompression Brayton cycle are provided in Zaharil & Yang (2024), while the ORC follows a conventional simple configuration. Fig. 3.1 Fig. 3.1 The methodological flow of the integrated system is the high-level overview and steps of the complete integrated system. Please note that the variables shown in the figure may vary depending on the specific objectives of the thesis study.

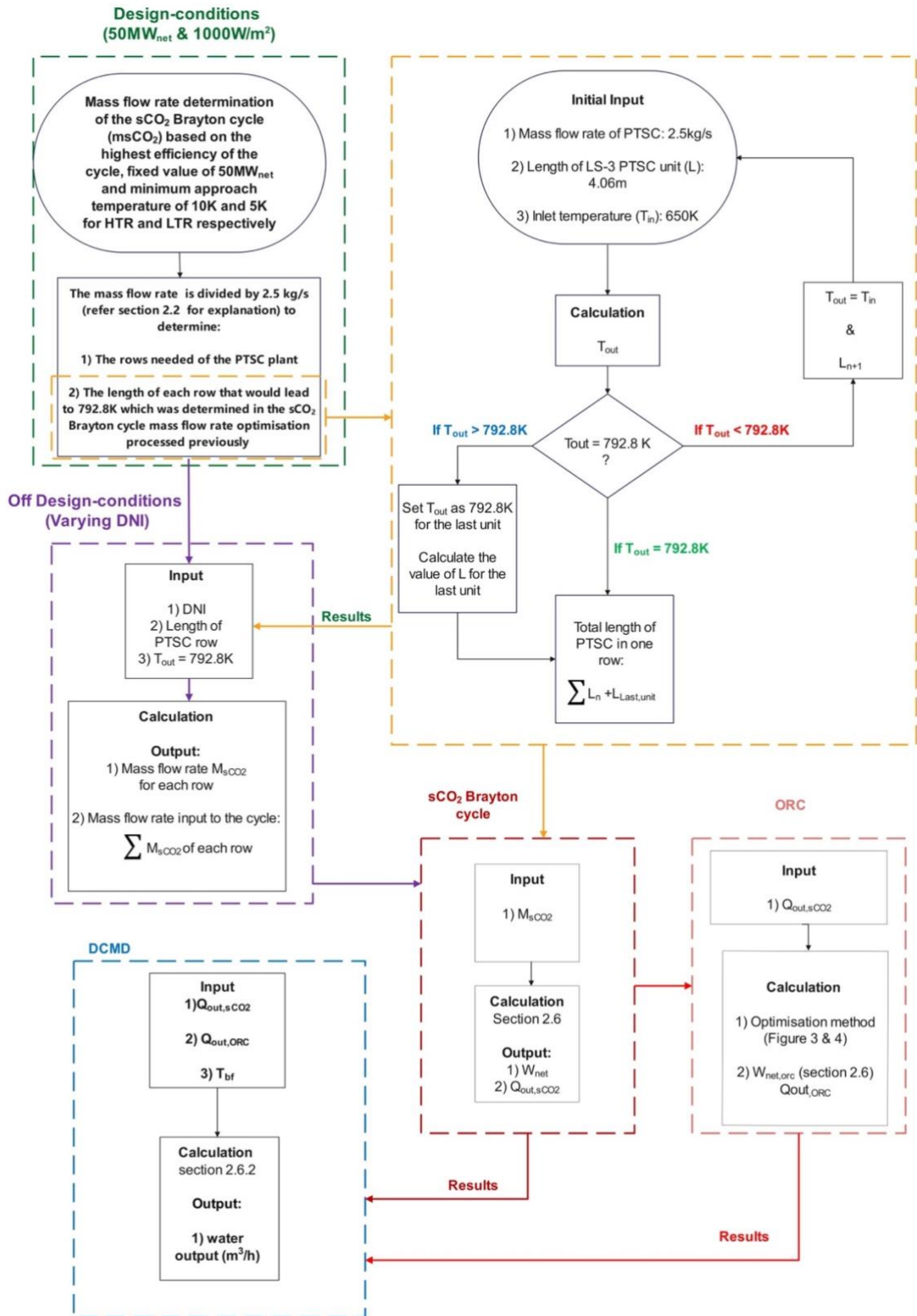


Fig. 3.1 The methodological flow of the integrated system

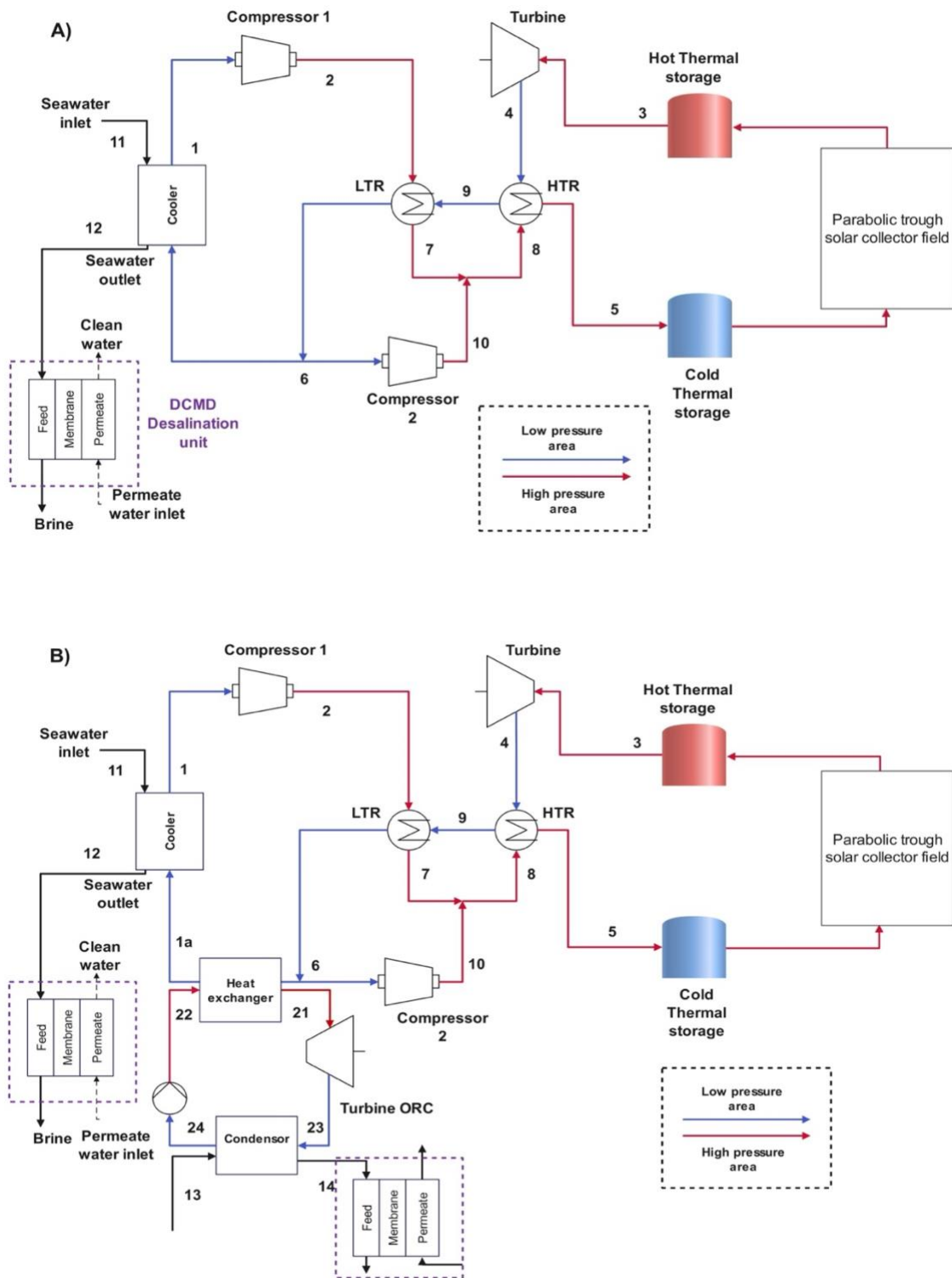


Fig. 3.2A) Single recompression $s\text{CO}_2$ Brayton cycle and B) Double cycle $s\text{CO}_2$ recompression Brayton cycle and a simple bottoming ORC

3.1.1 Assumptions

The system is assumed to operate under steady state conditions, with no consideration of startup, shut-down, or transient effects. Heat is supplied directly from the parabolic-trough solar field without thermal storage or back-up fuel. Only direct normal irradiance (DNI) is used in the analysis, as concentrated solar power systems cannot utilise diffuse solar radiation. The working fluids are treated as pure substances with fixed composition, and their properties are obtained from EES's library. All equipment operates within typical safe pressure and temperature limits, with material ageing and long-term degradation neglected. Kinetic and potential energy effects are ignored, assuming the plant remains stationary.

3.2 Parabolic trough solar collector (PTSC)

This section presents only the core elements of the PTSC mathematical model that captures the essentials of the methodology. The full set of equations, detailed derivations, and supplementary formulations are provided in APPENDIX 1.

The physical model is based on a modified LS-3 parabolic trough unit, incorporating an increased wall thickness in the heat collecting element (HCE) receiver tube from the standard 0.01 m to 0.02 m to withstand the high-pressure requirements of sCO₂ (refer Fig. 3.3). Similar design adjustments have been reported in prior sCO₂-PTSC studies (Goyal & Reddy, 2022; Zaharil & Yang, 2023; Zaharil, 2021; Bellos & Tzivanidis, 2017b; Qiu et al., 2017). The full PTSC specifications are given in Table 3.1. While different PTSC configurations may yield varied optimal values, the modified LS-3 remains a widely adopted academic reference, as evidenced by recent studies, and continues to provide valuable insights into thermodynamic performance under diverse operating conditions.

Table 3.1 Full specifications of examined LS-3 PTSC model. (Qiu et al., 2017; Bellos & Tzivanidis, 2017b)

Dimensions	Values per unit PTSC
Width (W)	5.6m
Length (L)	4.06m
Focal length (f)	1.71m
Aperture area (A_a)	22.74m ²
Concentration ratio (C_r)	25.46
Inner/Outer receiver diameter (D_{ri}/D_{ro})	0.05m / 0.07m
Inner/Outer glass cover (D_{ci}/D_{co})	0.108m / 0.115m
Emissivity of glass cover (ε_c)	0.86
Optical efficiency (η_0)	0.8419 when $K(\theta) = 1$
Thermal conductivity of the glass cover (K_c)	0.78 W/m·K (estimated) (Omni calculator, 2024)
Emissivity of receiver (ε_r)	0.000327 T_r - 0.065971

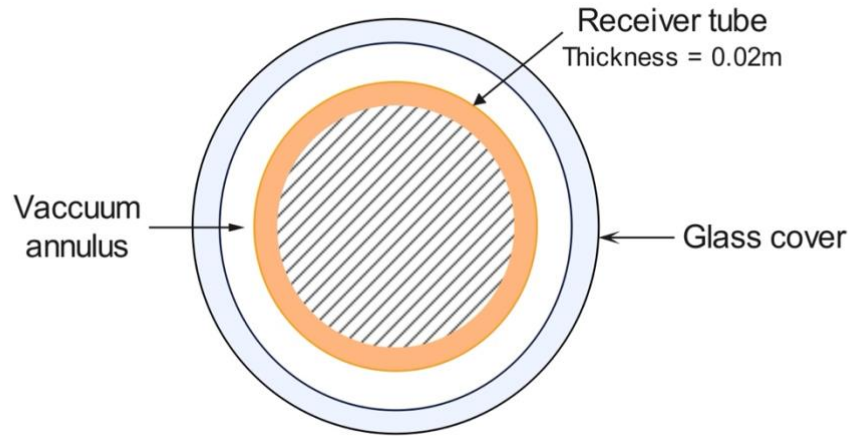


Fig. 3.3 The cross-section view of the LS-3 heat collecting element (HCE)(Zaharil & Yang, 2025)

3.2.1 Mathematical Model of PTSC

The fundamental concepts of energy balance and heat transfer are applied to each PTSC unit. The thermal efficiency of each unit is determined based on the energy absorbed by the working fluid, which is evaluated using Newton's law of cooling in conjunction with the steady-state energy balance.

$$\dot{Q}_u = h_f \pi D_{ri} L (T_r - T_{fm}) \quad (3-1)$$

$$\dot{Q}_u = \dot{m} C_p (T_{out} - T_{in}) \quad (3-2)$$

Heat loss from the absorber tube arises from the temperature gradient between the tube and the ambient air and occurs through three primary mechanisms, namely radiation and convection to the inner surface of the glass cover, conduction through the glass material, and subsequent radiation and convection to the sky and surrounding environment. Convection within the vacuum annulus is considered negligible.

$$\dot{Q}_{loss,rad} = \frac{\pi \sigma D_{co} L (T_r^4 - T_{ci})}{\frac{1}{\varepsilon_r} + \frac{1 - \varepsilon_c}{\varepsilon_c} \left(\frac{D_r}{D_{ci}} \right)} \quad (3-3)$$

$$\dot{Q}_{loss,cond,gi-go} = \frac{2\pi K_c L (T_{cc} - T_{co})}{\ln \left(\frac{D_{co}}{D_{ci}} \right)} \quad (3-4)$$

$$\dot{Q}_{loss,conv+rad} = \pi h_w D_{co} L (T_{co} - T_{amb}) + \varepsilon_c D_{co} L \pi \sigma (T_{co} - T_{sky}) \quad (3-5)$$

Thermal efficiency, as used here, is the ratio of heat absorbed by the working fluid to the total DNI on the collector, with pumping power accounted for. This definition, following Mwesigye et al. (2018), is particularly relevant for long PTSCs, where pressure drops can notably reduce performance.

$$\eta_{th} = \frac{\dot{Q}_u - W_p / \eta_{el}}{I_b A_a} \quad (3-6)$$

The Petela model is applicable for calculating the exergy flow from undiluted incoming solar irradiation (Petela, 2003). The total solar exergy input can be expressed as follows:

$$\dot{E}_s = I_b A_a * \left[1 - \left(\frac{4}{3} * \frac{T_{amb}}{T_{sun}} \right) + \left(\frac{1}{3} * \left(\frac{T_{amb}}{T_{sun}} \right)^4 \right) \right] \quad (3-7)$$

The useful exergetic content carried by the working fluid can be calculated using the following equation (Loni et al., 2016):

$$\dot{E}_u = \dot{Q}_u - \dot{m} C_p T_{amb} \ln \left[\frac{T_{out}}{T_{in}} \right] - \left(\frac{\dot{m} T_{amb} \Delta P}{\rho_f T_{fm}} \right) \quad (3-8)$$

The exergetic efficiency is the ratio of the fluid's exergy gain over the total undiluted exergy flow:

$$\eta_{ex} = \frac{\dot{E}_u}{\dot{E}_s} * 100\% \quad (3-9)$$

3.2.2 Discretisation method

In this study, a one-dimensional (1-D) discretised mathematical model is employed to analyse the thermal performance of parabolic trough solar collectors (PTSC) using supercritical carbon dioxide (sCO₂) as the working fluid. While 1-D models have been reported in the literature, their application in a fully discretised form for PTSC analysis remains rare. In the present work, the collector is segmented into a series of small control volumes, with the energy balance and heat transfer calculations performed sequentially along its length. This method captures the progressive changes in fluid temperature, heat gain, and efficiency with much greater accuracy than conventional single-unit evaluations, which often assume negligible temperature change between the inlet and outlet. By resolving temperature and performance variations along the collector, the model provides a more realistic representation of longer collector configurations and reveals trends that diverge notably from those reported in earlier sCO₂ studies. The use of this discretised approach enables the first comprehensive parametric and sensitivity analyses of sCO₂ in PTSCs, allowing the identification of new operational insights and the exploration of direct-system designs without intermediate heat exchangers. This modelling strategy forms the core novelty of the present work, and the results obtained demonstrate its potential to influence future PTSC optimisation and high-temperature cycle integration strategies.

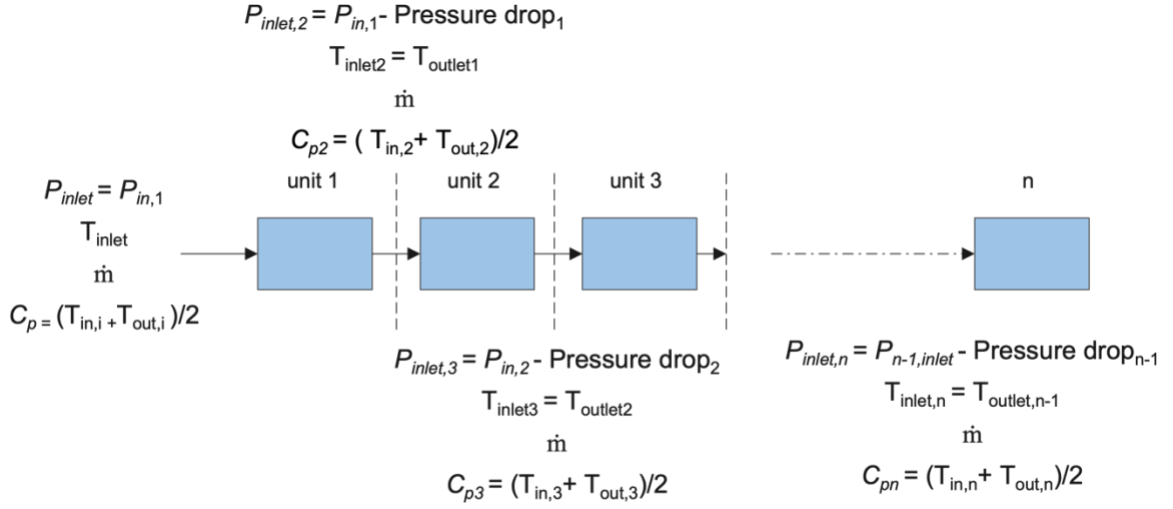


Fig. 3.4 Discretised PTSC system using a 1-D model, illustrating the variation of thermophysical property inputs across consecutive PTSC units

Fig. 3.4 illustrates the one-dimensional discretised modelling approach used in this study divides the parabolic trough solar collector into a series of small segments or control volumes arranged in sequence. For each segment, the inlet pressure, temperature, mass flow rate, and average specific heat are defined, with the inlet temperature of a segment set equal to the outlet temperature of the preceding one. The pressure drop across each segment is calculated and subtracted from the previous inlet pressure to determine the new inlet pressure. Using these updated inlet conditions, an energy balance is applied to compute the segment's outlet temperature, heat gain, and other thermal parameters. This process is repeated segment by segment along the full collector length, progressively capturing the variation in fluid properties, thermal performance, and pressure losses. By resolving these changes step by step, the method accurately reflects the non-uniform temperature rise and pressure drop along the collector, enabling more realistic performance predictions, especially for long collector configurations, than conventional single-unit analyses. Below are the equations

$$\eta_{ex} = \frac{\int_{i=1}^n E_u(i) di}{\int_{i=1}^{n-1} E_s(i) di + E_s(n)} \quad (3-10)$$

$$\eta_{ex} = \frac{\sum_{i=1}^n \dot{E}_{u,i}}{\sum_{i=1}^{n-1} \dot{E}_{s,i} + \dot{E}_{s,n}} \quad (3-11)$$

with n being the last unit. The reason E_{s_n} is included because the length of the final unit of the PTSC varies according to the examined length (refer to Fig. 3.5). Therefore, inclusion is necessary to accurately calculate the exergy of undiluted thermal radiation (E_s) reflected towards the heat-collecting element. Notably, the same calculation method in was applied for other desired outputs, such as thermal efficiency, pressure drop, and heat loss, among others.

3.2.3 Model assumptions and justifications

1-D modelling is associated with a few well-known assumptions. Some of the most prominent ones are the assumption of constant flux across the receiver, negligible axial heat loss, and steady-state conditions, as referenced in the works of Yilmaz & Söylemez (2014) and Behar et al. (2020) and Chaanaoui et al. (2024). However, these assumptions are generally negligible over short distances and have proven highly accurate. Additionally, the bracket conduction loss is considered negligible in this research, as other thermal losses significantly outweigh it, as mentioned in the NREL's System Advisor Model (SAM) technical manual by Wagner & Gilman (2011). This assumption was kept constant across all conditions to enable proper comparative research between different inlet temperatures and mass flow rates. The mass flow rate was kept constant across all units, which is acceptable as the aforementioned reference also utilised this approach. Additionally, a detailed optical model is not employed as this research focuses on thermal performance.

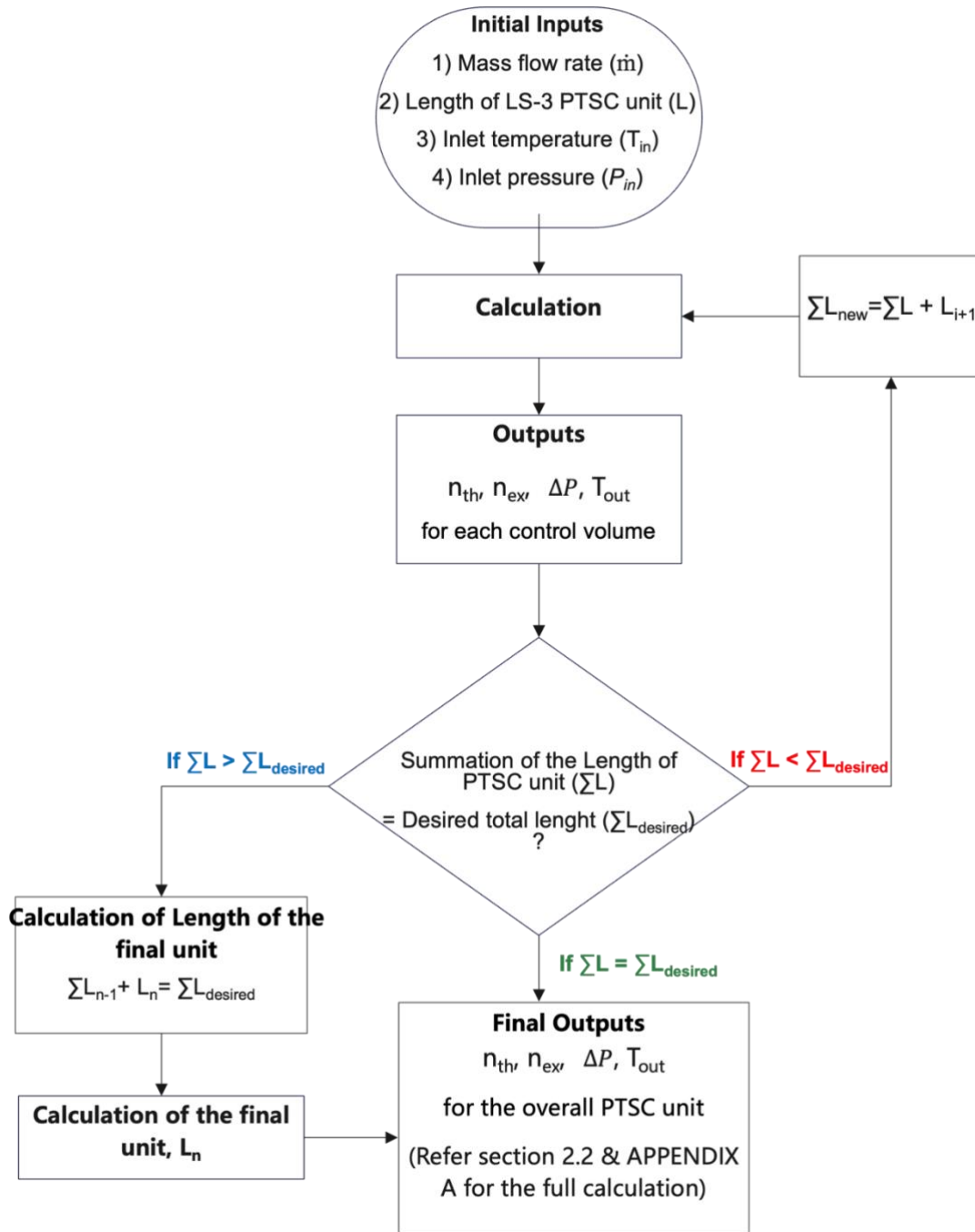


Fig. 3.5 Methodology flowchart for the design and calculation of the discretised PTSC system

3.2.4 Model validation

The model validation in Table 3.2 evaluates thermal efficiency does not account for pumping power. This allows a direct and accurate comparison with the work of Bellos et al. (2017b), who used sCO₂ in the same PTSC configuration, a method widely adopted in previous

studies with various fluids and shown to be highly accurate. Validation was performed on a single unit; however, as shown by Chaanoui et al. (2024), the discretisation method maintains high accuracy when based on precise single-unit assessments. While conventional thermal efficiency was used here for consistency, incorporating pumping or compressing power is more representative for long-distance PTSC applications, improving sCO₂ performance estimates over extended lengths. Additional validation against the CFD results of Qiu et al. (2017) showed an outlet temperature deviation of only 0.038% (481.86 K reference vs. 481.68 K in this study), confirming the model's reliability.

Table 3.2 Model validation against the results of Bellos et al., (2017b).

Inlet temperature (K)	Inlet pressure (kPa)	Mass flow rate (kg/s)	Thermal efficiency (Bellos & Tzivanidis, 2017)	Thermal efficiency	Percentage error (%)
350	8000	0.5	0.839	0.839	0
450	8000	0.5	0.827	0.827	0
550	8000	0.5	0.802	0.801	≈0
650	18000	2.0	0.775	0.775	0
750	20000	2.5	0.704	0.704	0
850	20000	2.5	0.583	0.583	0
Inlet temperature (K)	Inlet pressure (kPa)	Mass flow rate (kg/s)	Exergetic efficiency (Bellos & Tzivanidis, 2017)	Exergetic efficiency	Percentage error (%)
350	8000	0.5	0.155	0.1543	0.4516
450	8000	0.5	0.315	0.3148	0.0635
550	8000	0.5	0.403	0.4031	0.0248
650	18000	2.0	0.449	0.4493	0.0668
750	20000	2.5	0.453	0.4533	0.0662
850	20000	2.5	0.404	0.4036	0.099
Inlet temperature (K)	Inlet pressure (kPa)	Mass flow rate (kg/s)	Outlet temperature (K) (Bellos &	Outlet temperature (K)	Percentage error (%)

			Tzivanidis, 2017)		
350	8000	0.5	374.9	374.8	0.16
450	8000	0.5	479.8	479.7	0.021
550	8000	0.5	578.9	578.9	0
650	18000	2.0	656.5	656.5	0
750	20000	2.5	754.7	754.7	0
850	20000	2.5	853.8	853.8	0.012

To further strengthen the validation, the same mathematical model and discretised method used in this study were applied to the PTSC specifications reported by Ma et al. (2018). Their work examined a EuroTrough-150 PTSC unit, 150 m in length, operating with Therminol VP-1 as the working fluid, based on the real Badaling 1 MW PTSC power plant in Beijing. The specifications of the PTSC model from Ma et al. (2018). By discretising the entire collector length into smaller control volumes, the outlet temperature deviations were only 2.73 °C and −2.10 °C, corresponding to percentage errors of 1.18% and −0.73%, respectively (see Table 3.3). These results confirm the validity of the proposed method.

Table 3.3 Model validation taken from Ma et al., 2018

DNI (W/m ²)	$\dot{V}$ (m ³ /h) (Ma et al., 2018)	T _{in} (°C) (Ma et al., 2018)	T _{out} (°C) (Ma et al., 2018)	T _{out} (°C) – Present study	Temperature difference (Absolute)	Temperature difference (%)
926	75	170	231.26	233.99	2.73	1.18%
928	60	190.6	288	285.9	-2.10	0.73%

3.3 Supercritical Brayton Cycle

In this study, a partially discretised steady-state thermodynamic model is developed to evaluate the performance of a recompression supercritical carbon dioxide (sCO₂) Brayton cycle. The turbomachinery components, the main and recompression compressors, and the turbine,

are treated as single-step elements using specified isentropic efficiencies, consistent with a zeroth-order approach. The heat exchangers, namely the high-temperature recuperator, low-temperature recuperator, and cooler, are discretised into finite segments to capture progressive temperature variations along the flow paths, enabling accurate determination of heat duties, surface areas, and associated costs. Given the high density of sCO₂, pressure losses in these units are minimal, previous studies report drops on the order of around 0.01%–0.3% (Khatoon & Kim 2022; Moisseytsev & Sienicki, 2007) and are therefore neglected by assuming constant pressure across each heat exchanger. Enthalpy values are used directly in the calculations, as enthalpy inherently accounts for variations in specific heat capacity, density, and other thermophysical properties, thus ensuring accurate representation of large variance of thermophysical properties of CO₂ near the critical point. This modelling approach combines the computational efficiency of a zeroth-order framework for turbomachinery with the accuracy of segment-by-segment heat exchanger analysis, striking a balance between rapid cycle evaluation and the level of detail required for reliable sizing and costing in preliminary design stages. The model outputs include cycle efficiency, net power output, heat input, and detailed heat exchanger sizing, offering a robust platform for a thermodynamic performance and techno-economic assessments. The T-s diagram of sCO₂ recompression cycle is shown in Fig. 3.6.

The specifications of the sCO₂ Brayton cycle at the reference state are typically presented in Table 3.4. These may vary during parametric investigations conducted in the thesis. In all cases, any modifications will be clearly stated or evident in the course of the investigation.

Table 3.4 The reference state condition for sCO₂ Brayton cycle

<i>sCO₂ Brayton cycle (Al-Sulaiman & Atiff., 2015; Chacartegui., 2011)</i>	
Temperature at the inlet of compressor (C1)	304.25K
Pressure at the inlet of the compressor (C1)	7.4MPa
Pressure ratio	2.7

High temperature recuperator effectiveness, HTR	0.85
Low temperature recuperator effectiveness, LTR	0.7
Isentropic efficiencies of compressors	0.8
Isentropic efficiencies of turbine	0.9
Split Ratio (SR)	Variable
Temperature at state 5 (T5)	650K
Mass flow rate (kg/s)	627.7 kg/s (reference state)

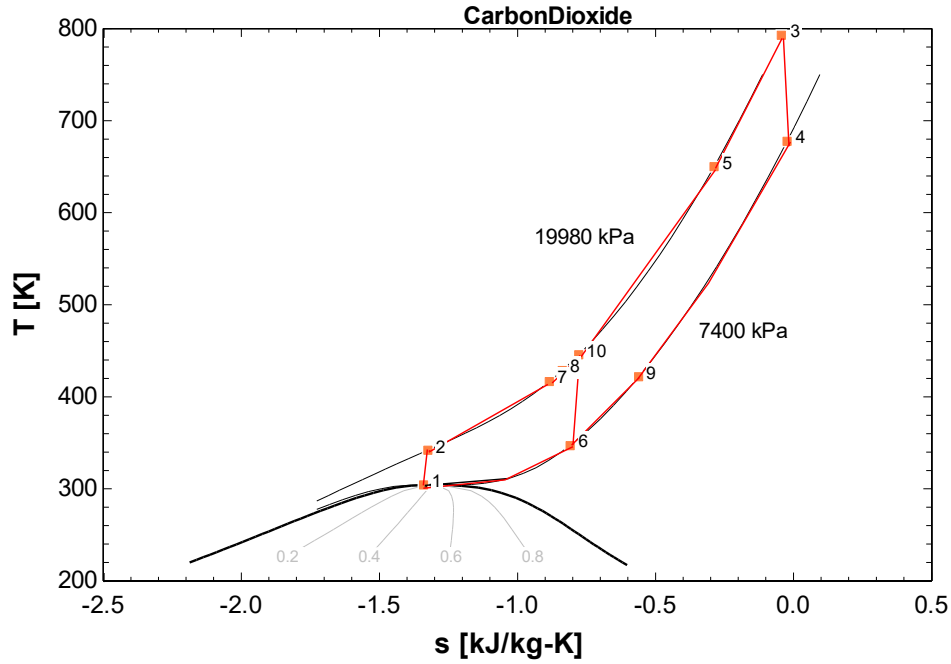


Fig. 3.6 sCO₂ recompression cycle Brayton cycle T-s diagram

The Brayton cycle mathematical model utilizes the isentropic efficiencies of compressors in its calculations, as detailed below.

$$\eta_{c1} = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (3-12)$$

$$\eta_{c2} = \frac{h_{10s} - h_6}{h_{10} - h_6} \quad (3-13)$$

The energy used by the two compressors can be described as:

$$\dot{W}_{c1} = SR * \dot{m}_{sCO_2} (h_2 - h_1) \quad (3-14)$$

$$\dot{W}_{c2} = (1 - SR) * \dot{m}_{sCO_2} (h_{10} - h_6) \quad (3-15)$$

The equations of the compressor's effectiveness were calculated as below:

$$\varepsilon_{HTR} = \frac{h_5 - h_8}{h_4 - h_8} \quad (3-16)$$

$$\varepsilon_{LTR} = \frac{h_7 - h_2}{h_9 - h_2} \quad (3-17)$$

The energy balance equations for HTR and LTR are respectively:

$$(h_4 - h_9) = (h_5 - h_8) \quad (3-18)$$

$$SR(h_7 - h_2) = (h_9 - h_6) \quad (3-19)$$

The amount of heat the system needs to release to reach state 1, which is the same as the heat the condenser takes in, is:

$$\dot{Q}_{out,CO_2} = SR * \dot{m}_{CO_2}(h_6 - h_1) \quad (3-20)$$

At point 8, where the fluid that was divided in point 6 converges, the governing equation is:

$$h_8 = [(1 - SR) * h_{10}] + (SR * h_7) \quad (3-21)$$

The isentropic efficiency and output power of the turbine were calculated as follows:

$$\eta_{turbine,CO_2} = \frac{h_3 - h_4}{h_3 - h_{4s}} \quad (3-22)$$

$$\dot{W}_{CO_2,turbine} = \dot{m}_{CO_2}(h_3 - h_4) \quad (3-23)$$

The Network done of the topping cycle is:

$$\dot{W}_{net,CO_2} = \dot{W}_{CO_2,turbine} - \dot{W}_{c1} - \dot{W}_{c2} \quad (3-24)$$

Ultimately, the energetic and exergetic efficiencies of the topping cycle integrated with PTSC are presented as:

$$\eta_{th,CO_2} = \frac{\dot{W}_{net,CO_2}}{\dot{Q}_{total}} * 100\% \quad (3-25)$$

$$\eta_{ex,CO_2} = \frac{\dot{W}_{net,CO_2}}{\dot{E}_s} * 100\% \quad (3-26)$$

where Q_{total} represents the total solar energy gathered by the PTSC collector, and E_s signifies the maximum energy that can be transformed into useful work (Exergy) by using the Petela (2003) model. The exergy destruction rate of the power cycles was calculated using the well-known exergy balance method, with the full equations available in (Singh & Mishra., 2018; Habibi et al., 2020)

For bottoming ORC, a simple cycle configuration was employed in the research and the mathematical model is as follows. For the turbine expansion process, the equations are:

$$\eta_{turbine,ORC} = \frac{h_{21}-h_{23}}{h_{21}-h_{23s}} \quad (3-27)$$

$$\dot{W}_{ORC,turbine} = \dot{m}_{orc} * (h_{21} - h_{23}) \quad (3-28)$$

With the $\dot{m}_{orc}$ denoting the optimized mass flow rate of the bottoming ORC as determined by a process shown in Fig. 3.11. The pump isentropic efficiency and power requirement are shown respectively in the equation below:

$$\eta_{pump} = \frac{h_{22s}-h_{24}}{h_{22}-h_{24}} \quad (3-29)$$

$$\dot{W}_{pump} = \dot{m}_{orc}(h_{22} - h_{24}) \quad (3-30)$$

The condenser of the ORC was calculated as below:

$$\dot{Q}_{ORC,out} = \dot{m}_{orc}(h_{23} - h_{24}) \quad (3-31)$$

The net-work of the ORC is:

$$\dot{W}_{net,ORC} = \dot{W}_{ORC,turbine} - \dot{W}_{pump} \quad (3-32)$$

The cycle efficiency of the ORC is the ratio of $\dot{W}_{net,ORC}$ over the heat input rejected from the topping cycle which is illustrated in Equation $\dot{Q}_{ORC,out} = \dot{m}_{orc}(h_{23} - h_{24})$ (3-31). This equation is used to determine the highest efficiency of the ORC in the optimisation method shown in Fig. 3.11.

$$\eta_{th,ORC} = \frac{\dot{W}_{net,ORC}}{\dot{Q}_{out,sCO_2}} * 100\% \quad (3-33)$$

The energetic and exergetic efficiencies of the combined system with ORC added at the bottom are:

$$\eta_{th} = \frac{\dot{W}_{net,sco2} + \dot{W}_{net,ORC}}{\dot{Q}_{total}} * 100\% \quad (3-34)$$

$$\eta_{ex} = \frac{\dot{W}_{net,sco2} + \dot{W}_{net,ORC}}{\dot{E}_s} * 100\% \quad (3-35)$$

3.3.1 Model validation

The mathematical model employed in this research has been rigorously validated to ensure accuracy. Validation of the Parabolic Trough Solar Collector (PTSC) model was previously established by Zaharil (2021) and Zaharil & Yang (2023). For the sCO₂ Brayton cycle, the model was benchmarked against the work of Al-Sulaiman & Atif (2015), which utilised identical specifications and the same recompression cycle configuration as in this study. The comparison revealed a maximum deviation of 4.9% from the results reported in March and June, indicating a high degree of reliability. Furthermore, the model was validated against the laboratory report of Moisseytsev & Sienicki (2007) from the Argonne National Laboratory, which examined a recompression sCO₂ Brayton cycle with a net power output of 200 MW. In this case, the efficiency deviation was 4.45% (see Table 3.5).

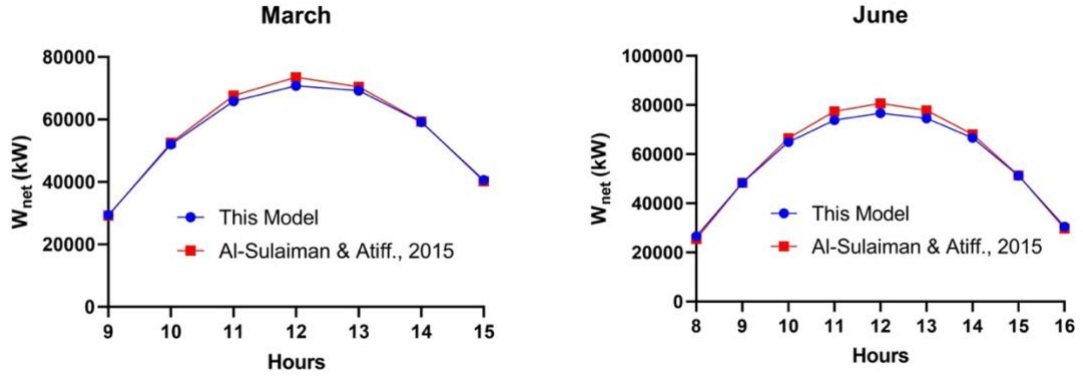


Fig. 3.7 Validation of recompression sCO₂ Brayton cycle against data from Al-Sulaiman & Atiff (2015)

Table 3.5 sCO₂ recompression Brayton cycle validation data with Moisseytsev & Sienicki (2007)

State	Moisseytsev & Sienicki (2007) (°C)	This research (°C)	Error (%)
1	31.25	31.25	0
2	84.4	85.02	0.73
3	471.87	471.7	0.036
4	362.3	362.3	0
5	323.3	323.3	0
6	90.2	88.21	2.26
7	171.8	172.3	0.29
8	175.2	175.2	0
9	190.7	190.8	0.052
10	183.8	184	0.11
η_{thcycle}	39.1 %	40.9%	4.45
SR	0.71	0.736	3.5
Q_{out}	145.5 MW	147.7 MW	1.49
Q_{HTR}	266.3 MW	266.3 MW	0
Q_{LTR}	165.5 MW	170.5 MW	3.02

3.3.2 Discretisation of heat exchanger

This section details the discretisation of the heat exchangers within the cycles to accurately capture the thermophysical variations of sCO₂, particularly in the vicinity of its critical point (see Fig. 3.8), where properties such as density and specific heat capacity exhibit pronounced changes in response to small temperature variations.

In this study, the counter-flow heat exchanger is modelled using a discretisation approach in which the exchanger length is divided into N finite segments. Each segment is treated as an independent control volume with distinct inlet and outlet states for the hot and cold streams, enabling the progressive resolution of temperature variations along the flow path. Unlike the conventional method based on $\dot{Q} = \dot{m}C_p \Delta T$, which assumes a constant or average specific heat capacity, the present work employs an enthalpy-based formulation. For each segment i , the energy balance is expressed as:

$$\dot{Q}_i = \dot{m}(h_{in,i} - h_{out,i}) = \dot{m}C_p(T_{in,i} - T_{out,i}) \quad (3-36)$$

where h denotes the specific enthalpy at the respective boundary, obtained directly from accurate thermophysical property databases. This approach inherently accounts for the variation of C_p with temperature, ensuring accurate representation of real-fluid behaviour, particularly for working fluids exhibiting significant property changes (refer Fig. 3.8). The analysis assumes steady-state operation, constant mass flow rates, and negligible heat losses to the surroundings. By marching through all N segments, the temperature profiles of both streams are determined across the full length of the exchanger, providing a detailed and thermodynamically consistent characterisation of the heat transfer process and a more accurate representation of its LMTD and area calculation for economic analysis. The summation energy balance of each of the control volume is equals to the global energy balance of the heat exchanger, proving the calculation of the method to be accurate (refer Fig. 3.9).

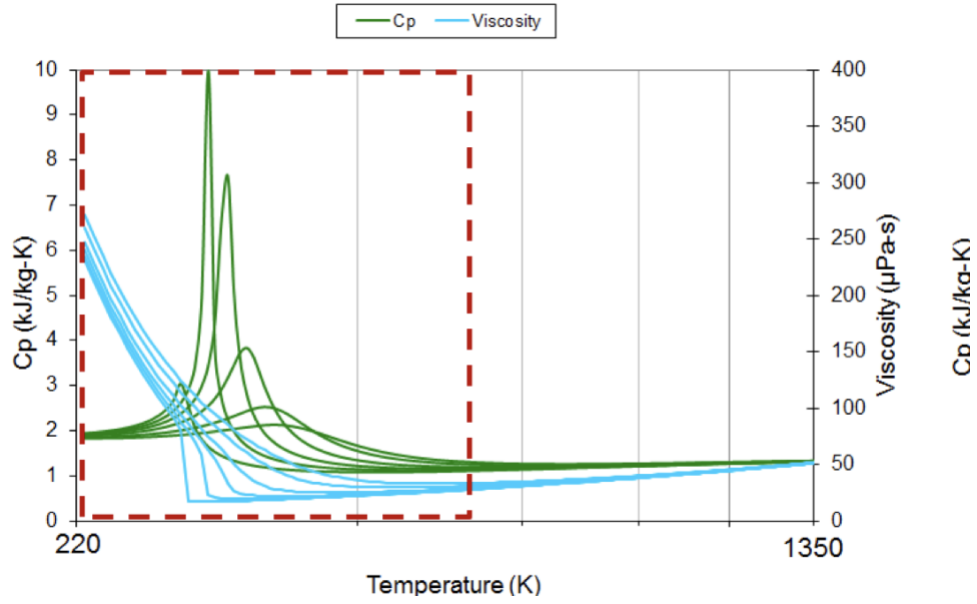


Fig. 3.8 Thermophysical properties of $s\text{CO}_2$ near its critical point (Schmitt et al. 2014)

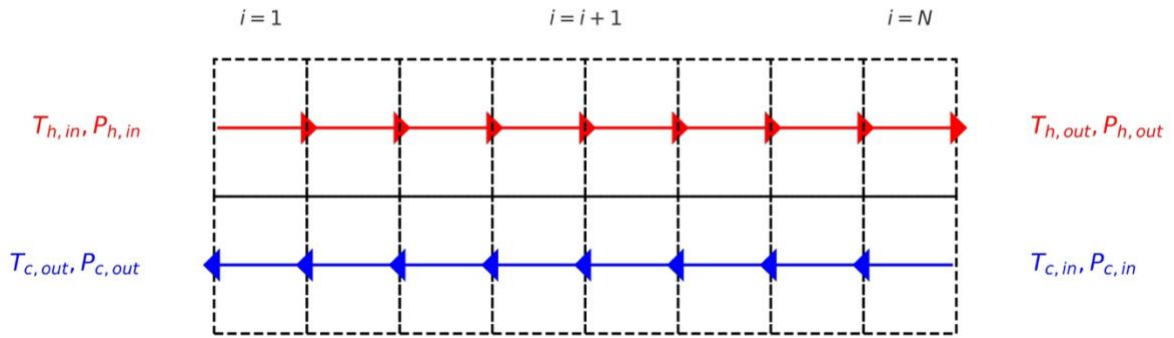


Fig. 3.9 Counter-flow heat exchanger discretisation into N segments for energy balance analysis.

3.3.3 Methodology for Maximising Heat Transfer from Topping to Bottoming Cycle

This work develops a systematic method for maximising heat transfer from the topping cycle to the bottoming cycle while ensuring that the working fluid remains dry after turbine expansion. While waste-heat utilisation from power cycles is widely studied, methods that determine the optimal extraction conditions for maximum thermal efficiency, W_{net} , and LCOE are relatively scarce.

1) Gap in Existing Approaches

In most literature, the waste heat transfer from the topping to the bottoming cycle is handled using a fixed temperature difference assumption, with 15 °C being the most common. Although pinch point and approach temperature are the two key temperature indices in heat exchanger design (refer to Fig. 3.10), studies rarely explore them in combination for power cycles design and optimisation.

2) Key Parameters in This Study

Pinch point: Fixed at 5 °C (a typical design value).

Approach temperature: Varied for each fluid according to its thermophysical profile and the ambient temperature.

Simultaneous variation: Pinch point and approach temperature are considered together to identify the combination that maximises cycle thermal efficiency (η_{th_ORC}), W_{net} and LCOE while preventing wetness at the turbine outlet.

3) Dryness Constraint in ORC

For Organic Rankine Cycles (ORC), lowering the ambient temperature can lead to a two-phase condition at the turbine outlet for certain fluids. This occurs because even isentropic and “dry” fluids have saturated vapour curves with non-vertical slopes. Combined with pressure ratio limits set by topping-cycle heat supply and condensation temperature, this can produce wet expansion.

During optimisation, if the calculated quality (x) at turbine exits T_{23} falls below 1 in EES, the corresponding approach temperature is discarded. Only cases with $x \geq 1$ are retained to ensure dry or isentropic expansion.

4) Energy Balance and Mass Flow Optimisation

The procedure in Fig. 3.11 ensures that the energy balance between each point of the hot and cold sections of the heat exchanger is maintained. The mass flow rate is optimised using:

$$\dot{m}_{SCO_2}(h_6 - h_{pp}) = \dot{m}_{ORC}(h_{21} - h_{pp2}) \quad (3-37)$$

where the pinch point is fixed and the approach temperature ($T_6 - T_{21}$) is varied from 6 K and above.

5) Final Temperature Calculation

Once the mass flow rate is optimised, the inlet temperature T_{1a} is calculated using the overall heat exchanger energy balance:

$$\dot{m}_{SCO_2}(h_6 - h_{1a}) = \dot{m}_{ORC}(h_{21} - h_{22}) \quad (3-38)$$

Novelty Statement

The proposed method (Fig. 3.10) simultaneously considers both pinch point and approach temperature, systematically varying the latter while maintaining energy balance and ensuring dryness at the turbine outlet. This integrated approach reveals non-obvious combinations of thermal efficiency, W_{net} and LCOE, (shown in section 6.1.1) representing a novel contribution to waste-heat recovery optimisation between topping and bottoming cycles in the literature.

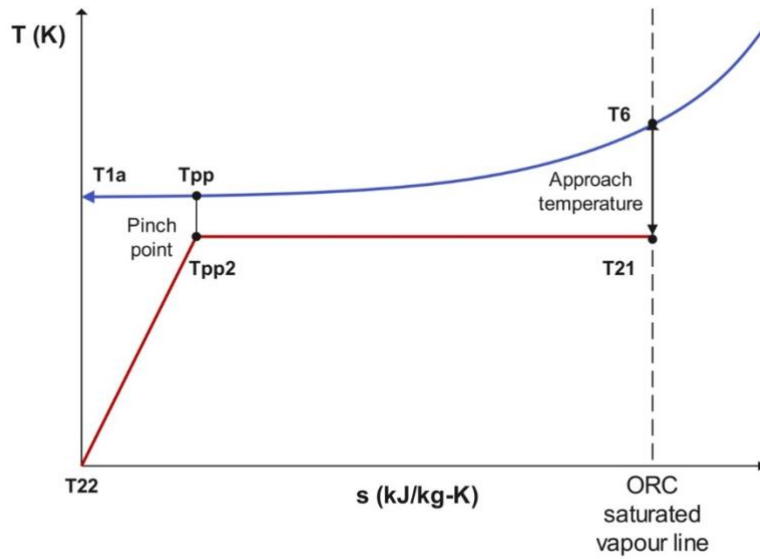


Fig. 3.10 An illustration of pinch point and approach temperature between topping and bottoming cycle.

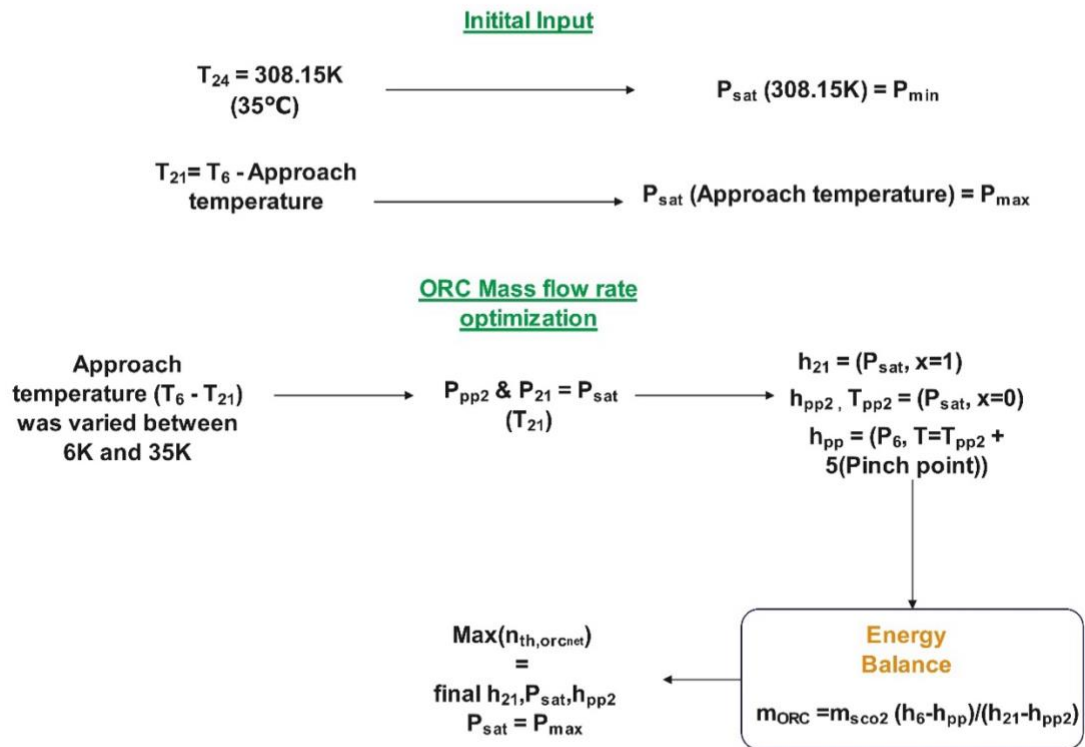


Fig. 3.11 Optimisation Flow and methodological steps for the Bottoming Organic Rankine Cycle Calculation.

3.4 Organic Rankine cycle (ORC)

A zeroth-dimension Organic Rankine Cycle (ORC) model were used, and it considers each component of the cycle, namely the pump, turbine, heater, and condenser, using simplified zeroth-order mathematical formulations. In this framework, overall energy balances were applied with prescribed isentropic efficiencies for the pump and turbine, together with either effectiveness values or pinch-point temperature differences for the heat exchangers. Each unit is treated as operating under constant pressure, and the heat exchangers were not discretised into segments. This modelling approach ensures computational efficiency and simplicity, while still capturing the key performance trends in comparison with experimental results, as summarised in Table 3.6. The thermodynamic modelling methodology adopted for the ORC is consistent with that employed for the sCO₂ Brayton cycle, but here it was applied to a simplified cycle configuration. The complete dataset is reported in Table 3.7.

Table 3.6 ORC validation against Galloni et al. (2015)

			Galloni et al. (2015)	This research	Error (%)
R245fa pressure at scroll inlet	P_{21}	[bar]	9.95	9.95	0.000
R245fa temperature at scroll inlet	T_{21}	[°C]	89.7	89.7	0.000
R245fa pressure at scroll outlet	P_{23}	[Bar]	2.17	2.17	0.000
R245fa temperature at scroll outlet	T_{23}	[°C]	38.4	38.31	0.234
R245fa pressure at pump outlet	P_{22}	[bar]	9.99	9.95	0.400
R245fa temperature at pump outlet	T_{22}	[°C]	35.8	35.32	1.341
R245fa pressure at pump inlet	P_{24}	[Bar]	2.02	2.17	7.426
Condensation heat rejected	Q_{out}	kW	9.87	9.87	0.000

R245fa temperature at pump inlet	T_{24}	[°C]	33.8	33.18	1.834
R245fa vapor quality at scroll inlet	x_3	[—]	0.915	0.915	0.000
Expander work rate	$\dot{W}_{expander}$	[kW]	1.17	1.16	0.855
Pump work rate	$\dot{W}_p$	[kW]	0.16	0.16	0.000
Thermal efficiency	η_{th}	[%]	9.28	9.19	0.980
Expander efficiency	$\eta_{h,exp}$	[%]	84.9	84.9	0.000
Pump efficiency	$\eta_{h,pum}$	[%]	79.7	79.7	0.000

Table 3.7 Bottoming cycle ORC specifications (Singh et al. 2018)

Isentropic efficiency of turbine	0.9 (Besarati & Goswami., 2013)
Isentropic efficiency of pump	0.8 (Besarati & Goswami., 2013)
Pinch point at the heat exchanger (K)	5
Condensation temperature (K)	308.15 (35 °C) – reference state
Ambient temperature (K)	303.15 (30 °C) - reference state
Topping and bottoming cycle pressures (kPa)	Varies for each fluid

3.5 Direct contact membrane distillation

Direct Contact Membrane Distillation (DCMD), recognised as the most straightforward and extensively studied configuration in membrane distillation, was selected for the present research. The DCMD performance was simulated using a numerical iteration model to estimate the membrane feed and permeate temperatures (T_{mf} and T_{mp}), initially assumed to be equal to the respective bulk temperatures (T_{bf} and T_{bp}). The detailed calculation procedure is illustrated in Fig. 3.12, and upon determining T_{bf} and T_{bp} , additional performance indicators, including total heat transfer (Q_t), overall heat transfer coefficient (U), temperature polarisation coefficient (TPC), evaporating efficiency (EE), and specific thermal energy consumption (STEC), were

subsequently computed (refer to APPENDIX 2 for the full detailed mathematical model). The DCMD module featured an operative membrane area of 0.56 m², achieved by stacking four membrane sheets of dimensions 0.2 m × 0.7 m (Soomro & Kim, 2018). A counter-current flow arrangement between the feed and permeate streams was adopted to enhance the permeate flux.

The flow chart of the iteration process to calculate T_{mf} and T_{mp} which requires J_m is shown in Fig. 3.12. For the full modelling please refer to the supplementary content. The calculation of J_m is given as:

$$J_{mh} = D_m * (P_a^v - P_b^v) * 3600 \quad (3-39) \quad (Khayet., 2011; Essalhi & Khayet., 2013)$$

where P_a^v and P_b^v are the vapour pressure of the seawater at feed and permeate water and D_m is the diffusion coefficient. The temperature of membrane at feed and permeate side can't be determined experimentally and thus it is calculated as (Essalhi & Khayet., 2013) :

$$T_{mf} = \frac{k_m * (T_{bp} + (h_f/h_p) * T_{bf}) + ((\delta_m * (h_f * T_{bf} - J_m * \Delta H_v)))}{(k_m) + (h_f * (\delta_m + (k_m/h_p)))} \quad (3-40)$$

$$T_{mp} = \frac{k_m * (T_{bf} + (h_p/h_f) * T_{bp}) + ((\delta_m * (h_p * T_{bp} + J_m * \Delta H_v)))}{(k_m) + (h_p * (\delta_m + (k_m/h_f)))} \quad (3-41)$$

For DCMD math model, the full mathematical model can be found in APPENDIX 2. The bulk feed temperature (T_{bf}) into the DCMD is the cooler or condenser outlet temperature and it varies with ambient temperature and organic fluids used with the permeate flux changing accordingly. The equations presented in this section were added for this research. Equation 3-42 specifically addresses the required mass flow rate for cooling the thermodynamic cycle, which consequently determines the heat introduction into both the condenser and DCMD units.

$$\dot{Q}_{out,SCO_2} = \dot{m}_{sea} * (h_{seawater,outlet} - h_{seawater,inlet}) \quad (3-42)$$

The required number of DCMD units is calculated based on the needed seawater mass flow rate:

$$N_{DCMD} = \dot{m}_{sea} / \dot{m}_{DCMD} \quad (3-43)$$

with $\dot{m}_{DCMD}$ representing the mass flow rate of each DCMD unit. The value of $\dot{m}_{DCMD}$ is 0.333 kg/s, calculated by converting the inlet volumetric flow rate of 20 L/min (Table 3.8). The amount of water produced per unit of DCMD in m³/hour is:

$$Q_{water} = A_{DCMD} * J_{mh} / \rho_{water} \quad (3-44)$$

The system's total water production is:

$$Q_{water,total} = Q_{water} * N_{DCMD} \quad (3-45)$$

The exergy destruction rate in DCMD modules was divided in three sections and the exergetic destruction rate on the feed, membrane, permeate side, and the total exergetic destruction rate were calculated respectively as follows (Choi et al., 2021; Mehdizadeh-Fard et al., 2021):

$$\dot{E}_{xd_feed} = \left(1 - \frac{T_0}{T_{bf}}\right) \dot{Q}_T - \left(1 - \frac{T_0}{T_{mf}}\right) \dot{Q}_T \quad (3-46)$$

$$\dot{E}_{xd_membrane} = \left(1 - \frac{T_0}{T_{mf}}\right) \dot{Q}_T - \left(1 - \frac{T_0}{T_{mp}}\right) \dot{Q}_T \quad (3-47)$$

$$\dot{E}_{xd_permeate} = \left(1 - \frac{T_0}{T_{mp}}\right) \dot{Q}_T - \left(1 - \frac{T_0}{T_{bp}}\right) \dot{Q}_T \quad (3-48)$$

$$\dot{E}_{xd_DCMD} = \dot{E}_{xd_feed} + \dot{E}_{xd_membrane} + \dot{E}_{xd_permeate} \quad (3-49)$$

where T_0 is the ambient temperature, Q_T is the total heat transfer across the DCMD module. The specifications of each unit of membrane distillation is detailed in Table 3.8.

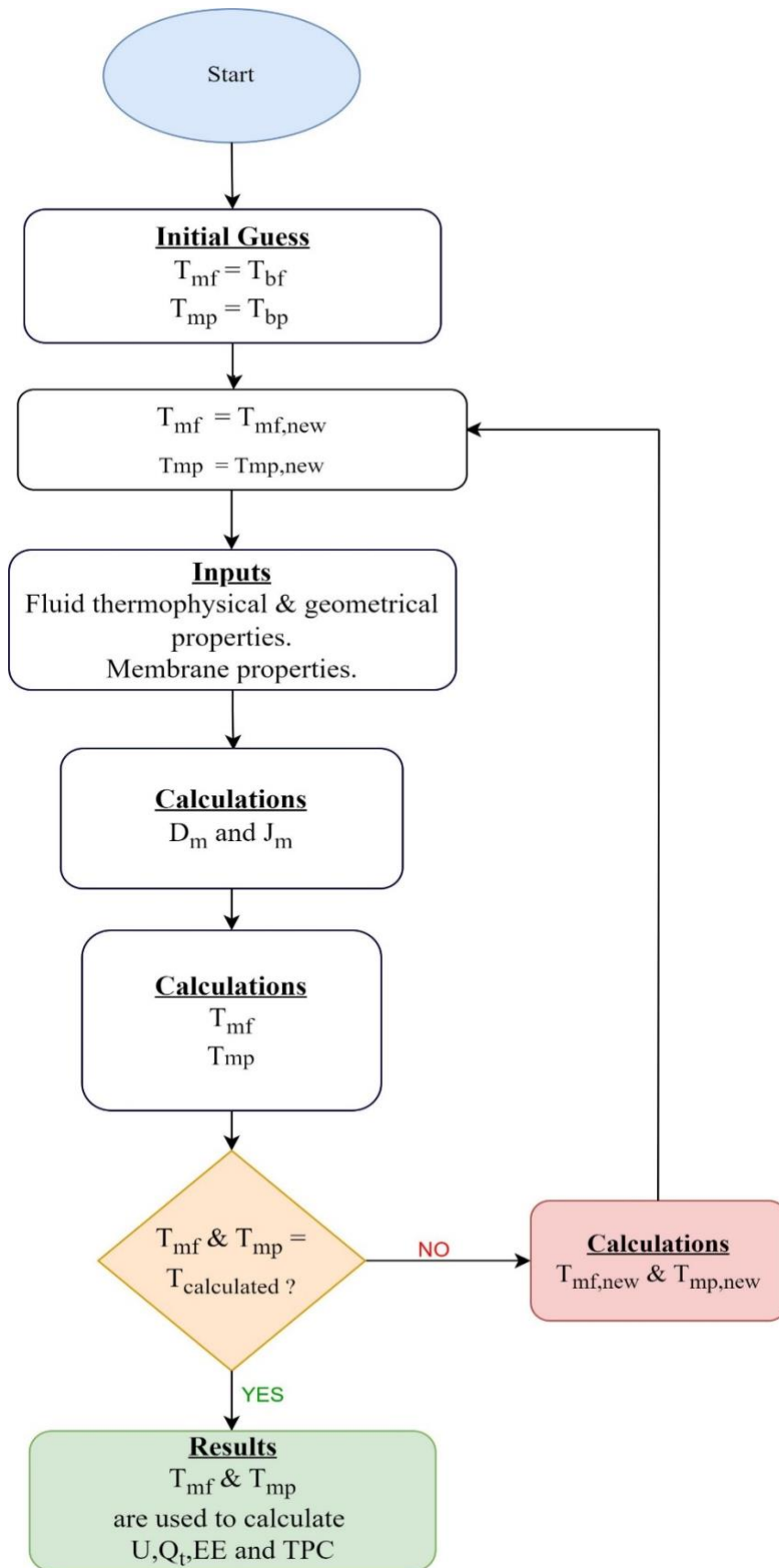


Fig. 3.12 The Iteration calculation process of DCMD

3.5.1 Model validation

The Direct Contact Membrane Distillation (DCMD) model was validated against the experimental results reported by Cath et al. (2004) and the numerical simulation conducted by Soomro & Kim (2018), yielding a maximum deviation of 4.84% (Refer Fig. 3.13), thereby further confirming the model's accuracy. The experimental study of Cath et al. (2004) utilised a DCMD unit with a membrane thickness of 175 μm , a porosity of 0.7, and a pore size of 0.45 μm , whereas the work of Soomro & Kim (2018) employed DCMD membrane properties and geometric parameters similar to those adopted in the present research.

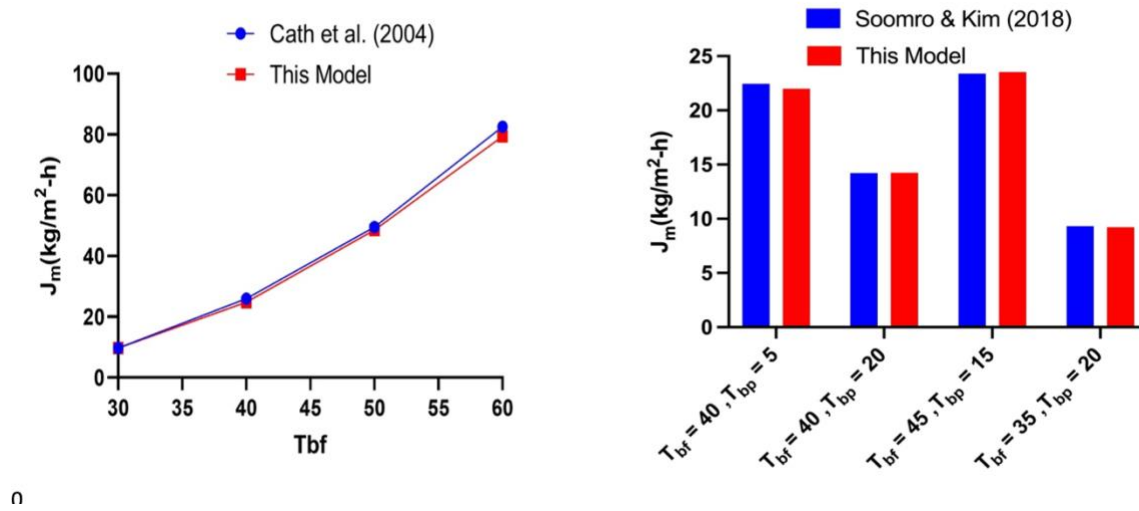


Fig. 3.13 DCMD validation against Cath et al., (2004) and Soomro & Kim (2018)

Table 3.8 Operating conditions and specifications of the DCMD module.

Description	Parameter	Value / Type
Membrane properties	Sheet type	Flat Sheet
	Material	PTFE
	Length	0.7 m
	Width	0.2 m
	Thickness (height)	0.001 m
	Effective area of membrane	0.56 m ² (0.14m ² each layer)
	Membrane Pore δ_m	120 μm
	Porosity ε	0.91 (91 %)
	Tortuosity	1.098
	Thermal conductivity K_m	0.2 W/m*K
Operating conditions	Concentration of feed water	0.35 g/kg

	Concentration of permeate water	0
	Feed flow rate	20 L/min
	Permeate flow rate	20 L/min
	Seawater inlet cooler temperature	293.15 K (20°C)

3.5.2 Model assumptions

The DCMD system assumed a uniform membrane pore size, negated surface diffusion due to hydrophobicity, and stationary air within pores, while the PTSC receiver was considered capable of handling the full pressure ratio increment for academic analysis.

3.6 Economic mathematical modelling

The Levelised Cost of Electricity (LCOE), Net Present Value (NPV), and Levelised Cost of Water (LCOW) were calculated using standard and widely accepted equations in techno-economic assessment. These metrics provide a consistent basis for comparing the cost-effectiveness and financial performance of the proposed system. Full methodological details for these calculations follow the standard practices outlined in the relevant literature.

3.6.1 Future projection model

This chapter, however, focuses specifically on the future cost projection of the proposed hybrid sCO₂-ORC-DCMD system. The aim is to estimate how the capital costs of each subsystem may evolve by 2030, based on inflation-adjusted baseline costs and technology-specific learning rates. The results are intended to provide insight into the system's long-term economic competitiveness as deployment scales and technology matures.

To estimate future capital costs of the hybrid sCO₂-ORC system, both general inflation and technology-driven learning were considered. The base cost values were first updated to 2023 price levels using the Chemical Engineering Plant Cost Index (CEPCI) taken from The University of Manchester (2024). This index reflects inflation in equipment, materials, and

labour across the process industries. Since CEPCI data is only available up to 2023, no further inflation was applied beyond this year. Instead, expected cost reductions due to learning-by-doing were applied from 2023 to 2030 using the learning curve method, which accounts for reductions in cost as deployment increases and production processes become more efficient. All cost projections are presented in constant 2023 dollars, consistent with standard techno-economic analysis practice.

The learning adjustment was made using the following equation:

$$Future_{CAPEX} = Current_{CAPEX} \times (1 - x)^n \quad (3-50)$$

where $Current_{CAPEX}$ is the cost in 2023 after inflation adjustment, x is the learning rate, expressed as a decimal and n is the number of doublings in cumulative deployment between 2023 and 2030. For the supercritical CO₂ (sCO₂) components (e.g. turbine, recuperator, high-temperature heat exchangers), a 5% learning rate was used ($x_{sCO} = 0.05$). This is based on the 6% learning rate recommended in the Sandia sBET model (Drennen & Lance, 2019), which applies to novel sCO₂ components in indirect cycles for CSP and other applications. However, a slightly more conservative rate of 5% was used in this study to account for uncertainties in the actual pace of sCO₂ deployment and technology maturity by 2030.

For the Organic Rankine Cycle (ORC), a learning rate of 1% was applied ($x_{ORC} = 0.01$), consistent with the NETL Cost Estimation Guidelines (2013), which recommend this value for mature technologies where most cost improvements have already been realised through previous commercial deployment. This value also reflects the limited potential for further cost reductions in ORC systems, which have been commercially available for decades. By applying learning separately to each system component, and using constant 2023 dollars, this method

ensures that projected cost reductions reflect technological learning alone, without conflating them with broader economic inflation, for which data beyond 2023 are not available.

3.6.2 Parabolic trough solar collector economic model

The sizing and cost estimation of the parabolic trough solar collector (PTSC) field are based on the thermal energy requirement of the sCO₂ Brayton cycle and the peak direct normal irradiance (DNI) at the site. The thermal power required from the PTSC field is calculated by

$$Q_{PTSC} = \frac{Q_{in}}{\eta_{optical}} \quad (3-51)$$

where Q_{in} is the thermal input to the sCO₂ cycle and $\eta_{optical}$ is the optical efficiency of the PTSC. The collector area is then determined from

$$A_{PTSC} = \frac{Q_{PTSC}}{DNI_{peak}} \quad (3-52)$$

using $DNI_{peak} = 1.0 \text{ kW/m}^2$. The capital cost of the PTSC field is obtained by

$$C_{PTSC} = A_{PTSC} \times y \quad (3-53)$$

where $y = 150 \text{ USD/m}^2$, representing the mid-range of projected 2030 values (USD 120–170/m²) from NREL (Kurup & Turchi, 2021) and DLR (Neumann & Feldhoff, 2018). The cost of the thermal energy storage (TES) system is estimated as:

$$C_{TES} = 0.13 \times C_{CAPEX, plant} \quad (3-54)$$

This 13 % figure is consistent with empirical data from large-scale parabolic trough projects. For example, the Andasol CSP plant's molten salt storage system has been reported at approximately USD 50/kWh_{th} equivalent to around 13 % of total plant CAPEX. This value lies within the 8–15 % range reported in IRENA's CSP cost literature (IRENA, 2023). Using

these assumptions provides a balanced and realistic basis for techno-economic modelling, grounded in both projected future costs and operational plant experience.

3.6.3 sCO₂ Brayton cycle economic model

This section is dedicated to calculating the component costs of the sCO₂ Brayton recompression cycle. Vendor-based cost correlations developed by Weiland et al. (2019) are used to determine the bare component cost for each major item, based on a power-law relationship between cost and a selected scaling parameter, with optional temperature correction factors for high-temperature operation. Each correlation applies to a specific component type and is valid within defined operating and size ranges (refer Table 3.9). The total component cost is then obtained by adding the installation costs, expressed as separate percentages for materials and labour, to the bare component cost, ensuring that both the purchase price and installation expenses are accounted for in the analysis (refer Table 3.10). The general form for the component calculations are is:

$$C = a \cdot (SP)^b \cdot f_T \quad (3-55)$$

where:

- C = equipment cost (2017 US\$)
- SP = scaling parameter (e.g. duty Q , conductance UA , shaft power $\dot{W}_{sh}$, volume flow $\dot{V}_{in}$, etc.)
- a, b = component-specific constants
- f_T = temperature correction factor

with the temperature correction factor (f_T):

$$f_T = \begin{cases} 1, & T_{metal} < 550^\circ C \\ 1 + c(T_{metal} - 550) + d(T_{metal} - 550)^2, & T_{metal} \geq 550^\circ C \end{cases} \quad (3-56)$$

- c, d = component-specific constants (if not listed, assume zero).

Installation cost:

$$C_{installed} = C \cdot (1 + f_{mat}) \cdot (1 + f_{lab}) \quad (3-57)$$

- f_{material} = materials factor (fraction)
- f_{labour} = labour factor (fraction)

A site development cost equal to 10% of the combined sCO₂ Brayton cycle components and PTSC field cost was included to account for shared civil works, infrastructure, and utility connections. This percentage is consistent with cost breakdowns for CSP plants, where site development and infrastructure typically account for around 9% of total plant CAPEX (Zhang, 2009). In addition, a contingency of 15% was applied component cost to cover design uncertainties, price fluctuations, and potential scope changes. This value falls within the 15–30% contingency range recommended by the Association for the Advancement of Cost Engineering for early-stage budget estimates (AACE International, 2013). Both percentages are therefore within typical ranges for early-stage techno-economic assessments and are added to CAPEX calculation.

Table 3.9 Cost correlation parameters for selected sCO₂ Brayton cycle components (Wieland et al, 2019).

Component	Scaling parameter (units)	a	b	c	d	Valid range	Uncertainty
Recuperators	UA (W/K)	49.45	0.7544	0.02141	0	1.6×10^5 – 2.15×10^8 W/K	-31% to +38%
Coolers	UA (W/K)	32.88	0.75	0	0	8.6×10^5 – 7.5×10^7 W/K	-25% to +28%
Axial turbines	$\dot{W}_{sh}$ (MW_{sh})	182,600	0.5561	0	1.106×10^{-4}	10-750 MW_{sh}	-25% to +30%

IG centrifugal compressors	$\dot{W}_{sh}$ (MW_{sh})	1,230,000	0.3992	0	0	1.5-200 MW_{sh}	-40% to +48%
Gearboxes	$\dot{W}_{sh}$ (MW_{sh})	177,200	0.2434	0	0	4-10 MW_{sh}	-15% to +20%
Generators	$\dot{W}_{sh}$ (MW_e)	108,900	0.5463	0	0	4-750 MW_e	-19% to +23%

Table 3.10 Installation cost percentages relative to each component's bare cost for sCO_2 Brayton cycle components (Weiland et al., 2019).

Component	Materials (%)	Labour (%)
Recuperators	2	3
Direct air coolers	8	12
Axial turbines	8	12
IG centrifugal compressors	8	12
Gearboxes	8	12
Generators	8	12

3.6.4 ORC economic model

This section is dedicated to calculating the component costs of the organic Rankine cycle (ORC) using the module costing method (refer APPENDIX 3 for detailed steps). This is a common method in economic analysis and has proven to be accurate in previous studies published in the literature. In this approach, the bare component cost is first determined from standard cost correlations based on key design parameters such as heat exchanger area or turbine power output. The bare module costing method then applies correction factors to account for material type, operating pressure, and other design specifications, resulting in the bare module cost. Installation and indirect costs are incorporated within this methodology,

providing a consistent and reliable framework for estimating the total component cost of the ORC system.

The bare module costing method was used to estimate installed equipment costs. The purchased cost of equipment (C_p) was calculated from standard cost correlations:

$$C_p = K_1 + K_2 X + K_3 X^2 \quad (3-58)$$

where X is either the power (for turbines/pumps) or heat exchanger area, and K_1 , K_2 , K_3 are equipment-specific constants (refer to Table 3.11). The actual purchase cost was obtained by adjusting for material and pressure:

$$C_{purchase} = C_p \times F_M \times F_P \quad (3-59)$$

where F_M is the material factor and F_P is the pressure factor, given by:

$$F_P = a + bP + cP^2 \quad (3-60)$$

with P in bar. The bare module cost was then:

$$C_{BM} = C_{purchase} \times F_{BM} \quad (3-61)$$

where F_{BM} is the bare module cost factor including all direct and indirect installation expenses.

Table 3.11 Coefficients used in the cost estimation models for each component of the Organic Rankine Cycle (ORC), based on Turton et al. (2008) as cited in Song et al. (2017).

Component	K_1, K_2, K_3	C_1, C_2, C_3	B_1, B_2	F_M	F_{BM}
Pump	$K_1 = 3.3892$	$C_1 = -0.3935$	$B_1 = 1.89$	1.0	/
	$K_2 = 0.0536$	$C_2 = 0.3957$	$B_2 = 1.35$		
	$K_3 = 0.1538$	$C_3 = -0.0023$			
Turbine	$K_1 = 2.2476$	/	/	/	3.5

$$K_2 = 1.4965$$

$$K_3 = -0.1618$$

Heat	$K_1 = 4.3247$	$C_1 = -0.0016$	$B_1 = 1.63$	1.35	/
exchanger	$K_2 = -0.3030$	$C_2 = -0.0063$	$B_2 = 1.66$		
	$K_3 = 0.1634$	$C_3 = 0.0123$			

3.6.5 DCMD economic model

This chapter is dedicated to outlining the methodology for calculating the levelised cost of water (LCOW) for the DCMD system, serving as a precursor to the subsequent performance and techno-economic analysis.

The total capital expenditure (CAPEX) includes three components, the membrane cost, balance of plant (BoP), and indirect costs. The membrane cost is estimated at \$50 per square metre of membrane area. The membrane module cost was set to \$50 per m² as a forward-looking mid-case for the 2030s. This value lies within the range reported by Zagaroga et al. (2018). The BoP, which includes auxiliary components such as pumps, heat exchangers, tanks, and piping, is taken as 40% of the membrane cost. An additional 15% of the combined membrane and BoP cost is added as indirect costs (e.g., installation, commissioning). Therefore, the total CAPEX is calculated as:

$$CAPEX = C_{membrane} + 0.4 \times C_{membrane} + 0.15 \times (C_{membrane} + BoP) \quad (3-62)$$

The operating expenditure (OPEX) consists of several yearly cost components. Membrane replacement is assumed at 20% of the original membrane cost per year, reflecting an average membrane lifespan of 5 years. Maintenance is taken as 2% of the total CAPEX annually.

Additionally, labour, chemicals, and brine disposal costs are estimated at \$0.03 per square metre of membrane area each year. This gives the total OPEX as:

$$OPEX = C_{replace} + C_{maint} + C_{labour} + C_{chemical} + C_{brine} \quad (3-63)$$

The annual freshwater production is determined based on the specific membrane flux J_{mh} in $kg/m^2 \cdot h$, the total membrane area, and the system capacity factor. Water production is converted from mass to volume using the density of water ($998 kg/m^3$) and the number of operating hours per year (8760 hours/year):

$$Q_{annual} = 0.402 \times 8760 \times \frac{J_{mh} \times A_{membrane}}{998} \quad (3-64)$$

To account for the time value of money over the system's 25-year lifespan, a discount rate of 5% is applied. The total cost and total water production are converted into net present values (NPV) as follows:

$$NPV_{cost} = CAPEX + OPEX \times \left(\frac{1-(1+r)^{-n}}{r} \right) \quad (3-65)$$

$$NPV_{water} = Q_{annual} \times \left(\frac{1-(1+r)^{-n}}{r} \right) \quad (3-66)$$

Where $r = 0.05$ and $n = 25$. Finally, the LCOW is given by the ratio of these two values:

$$LCOW = \frac{NPV_{cost}}{NPV_{water}} \quad (3-67)$$

This value represents the true average cost of producing freshwater over the plant's lifetime, including all capital and operational costs, and is suitable for comparing DCMD with other desalination technologies.

3.6.6 Conclusion

This chapter defines the proposed integrated system and sets out the modelling approach used throughout the thesis: a parabolic trough solar collector (PTSC) directly coupled to a supercritical CO₂ (sCO₂) Brayton cycle, with a bottoming organic Rankine cycle (ORC) and a direct-contact membrane distillation (DCMD) unit. It presents the core calculation methods (including discretised modelling where needed), and shows how energy and mass flows are tracked consistently across all parts of the system so later comparisons are fair. It also introduces the heat-recovery design logic (the “duality method” idea) as a repeatable way to set bottom-cycle conditions using two checks rather than a single rough balance. Finally, the chapter establishes credibility by validating key sub-models against prior studies and a national laboratory report, giving a sound base for the later results chapters.

CHAPTER 4 DETAILED ANALYSIS OF PTSC USING DISCRETISED METHOD

This chapter presents a comprehensive thermo-parametric analysis of parabolic trough solar collectors employing supercritical carbon dioxide (sCO₂) as the heat transfer fluid. Using a discretised one-dimensional modelling approach detailed in section 3.2.2 that incorporates pressure losses, the work systematically explores the influence of key operating and design parameters, including inlet and outlet temperatures, mass flow rate, collector length, operating pressure, and solar irradiance, on both thermal and exergetic performance. The findings not only identify optimal configurations for maximising efficiency that is different to the literature but also provide design guidelines for the direct integration of parabolic trough systems with sCO₂ Brayton cycles, offering valuable insights for advancing solar thermal power technologies.

The methodology employs a discretised one-dimensional model that divides the absorber tube into segments, accounting for heat transfer, energy balance, and pressure losses step by step. Thermophysical properties are evaluated at mean segment temperatures, and the model is validated against existing literature before being applied to a broad parametric analysis of key operating and design variables.

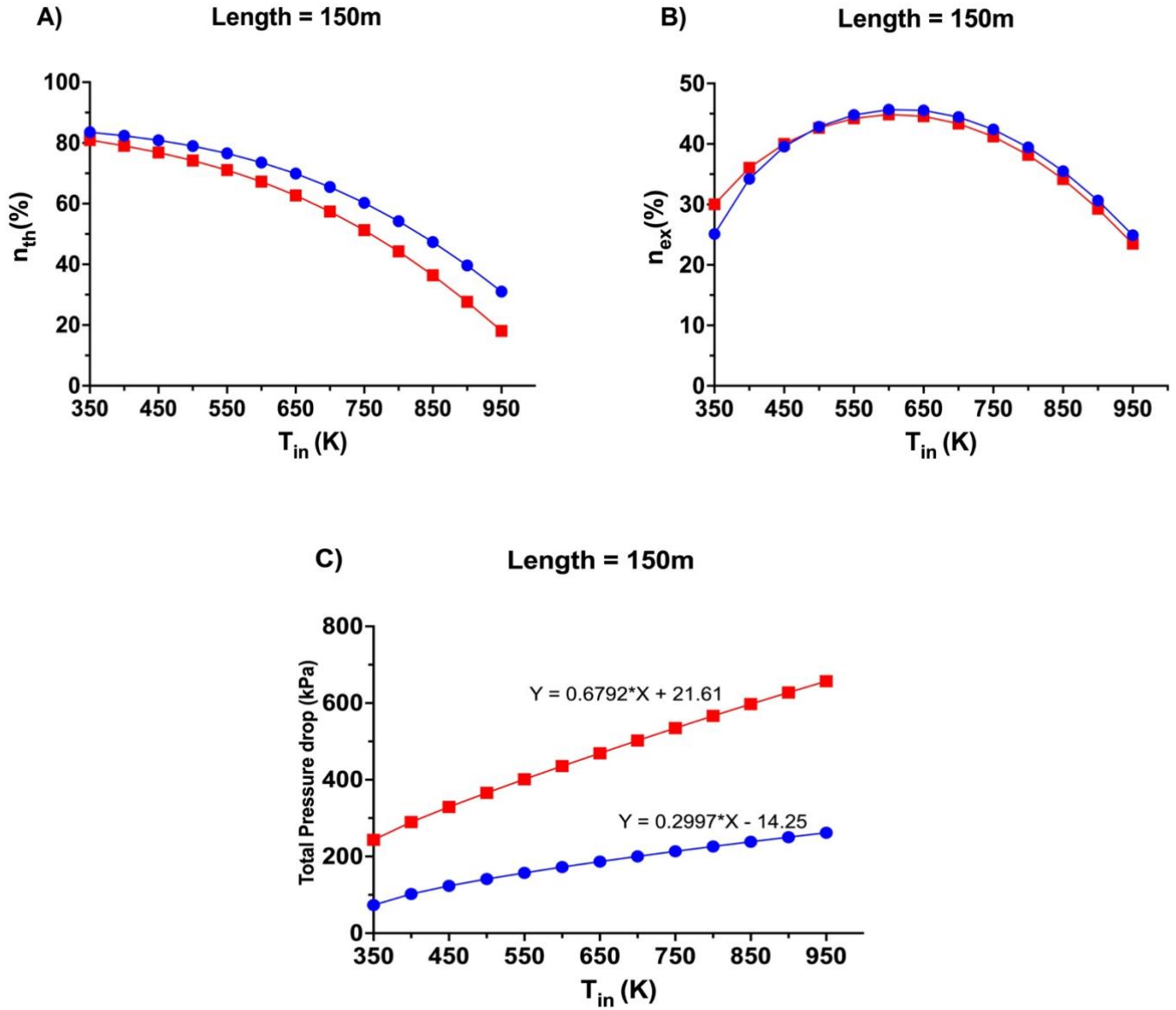


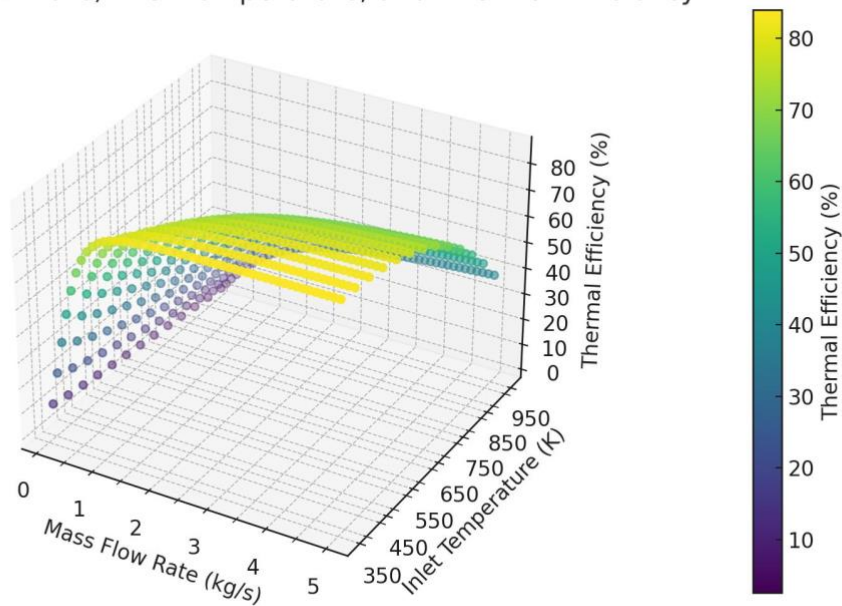
Fig. 4.1 A) Thermal efficiency, B) exergetic efficiency, and C) pressure drop comparison between 20 MPa and 9 MPa at a length of 150 m at a representative flow rate of 2.5 kg/s

Fig. 4.1 presents the relationship between efficiency and pressure drop across different inlet temperatures and pressures at a mass flow rate of 2.5 kg/s, a representative condition identified in previous studies (Zaharil, 2021; Bellos et al., 2017a). In Fig. 4.1A, thermal efficiency is observed to decline with increasing inlet temperature, while the 20 MPa case consistently outperforms the 9 MPa case, with the gap widening progressively. This trend is associated with the substantially higher pressure drop at 9 MPa compared to 20 MPa, as reflected in the respective best-fit slopes of 0.6792 and 0.2997. The superior performance at 20

MPa arises from the higher fluid density, which reduces pressure drop and consequently lowers pumping power requirements along the PTSC, as shown in Equation A1-16. Notably, exergetic efficiency exhibits a reversal, which was observed at inlet temperatures between 350 K and 450 K, the 9 MPa case demonstrates superior behaviour. This effect is attributed to the exceptionally high specific heat capacity (C_p) of CO₂ in this temperature range, due to its proximity to the critical point. Furthermore, the occurrence of a maximum point in exergetic efficiency, also identified in this study, was previously reported by Bellos and Tzivanidis (2017c), thereby corroborating their findings even for PTSCs of extended length.

4.1 Detailed parametric analysis between different combinations of mass flow rates and inlet temperatures

A) Mass Flow Rate, Inlet Temperature, and Thermal Efficiency



B) Mass Flow Rate, Inlet Temperature, and Outlet Temperature

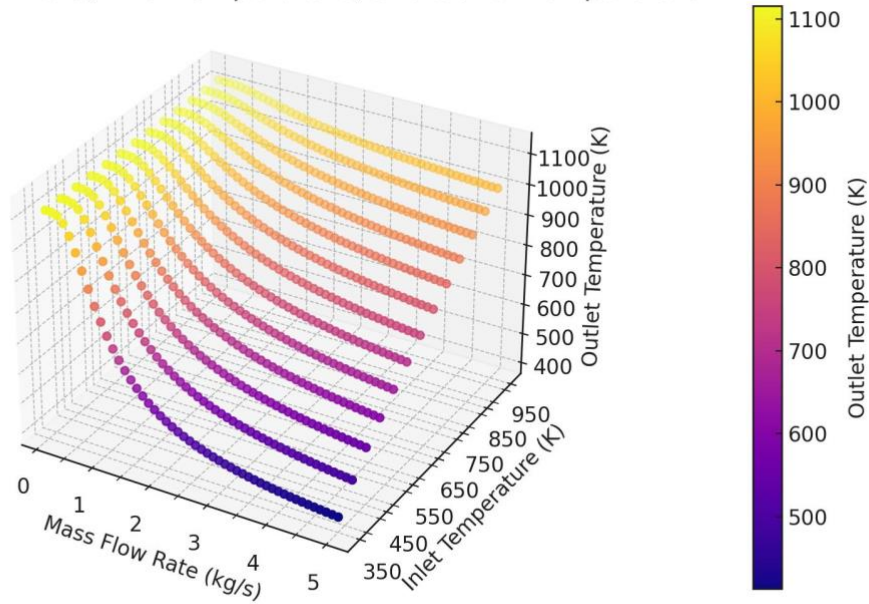


Fig. 4.2 Combinations of examined indexes and their impact on energetic efficiency and outlet temperature at a length of 150m and Pin of 20MPa.

Fig. 4.2 illustrates the combined impact of the three examined indexes on thermal efficiency and outlet temperature. It shows that inlet temperatures over prolonged distances result in varying outlet temperatures, limiting the comprehensiveness of any single-unit analysis of previous studies. This variation is more pronounced at lower inlet temperatures and reduced mass flow rates, where the outlet temperature is more susceptible to change. For example, at a T_{in} of 350 K, the outlet temperature significantly drops between mass flow rates of approximately 0.1 kg/s to 2 kg/s, indicating the need for careful longer distance analysis and optimisation. Fig. 4.2A reveals that lower inlet temperatures achieve optimal thermal efficiency at lower mass flow rates due to the lower pumping power required at these temperatures. This is because higher density and larger C_p , due to its proximity to the critical point, enhances the ability of lower mass flow rates to capture heat effectively. Additionally, thermal efficiency gains plateau or decline at mass flow rates below 5 kg/s for all inlet temperatures, proving

further analysis is not necessary. At higher inlet temperatures, thermal efficiency declines beyond a peak due to the impact of high pump power requirements.

Fig. 4.3 shows the exergetic efficiency under the same conditions as Fig. 4.2. It confirms the observations from Fig. 4.3, where inlet temperatures between 650 K and 850 K achieve maximum exergetic efficiency of around 44–46%. Moreover, at these temperatures, increased flow rates result in a smaller reduction in exergy, making them advantageous for applications that need flow rate flexibility. Despite that, this figure only exhibits the efficiency at unfixed outlet temperatures. A comprehensive study on the combinations of inlet temperature, mass flow rates and length of the PTSC is needed to determine the best combinations that lead to the highest efficiency for each targeted outlet temperature, which will be discussed in the next sections. Additionally, when analysed with fixed outlet temperatures, as shown in Table 4.2, the results reveal optimised values that differ for each target outlet temperature, indicating a significant departure from previous research findings.

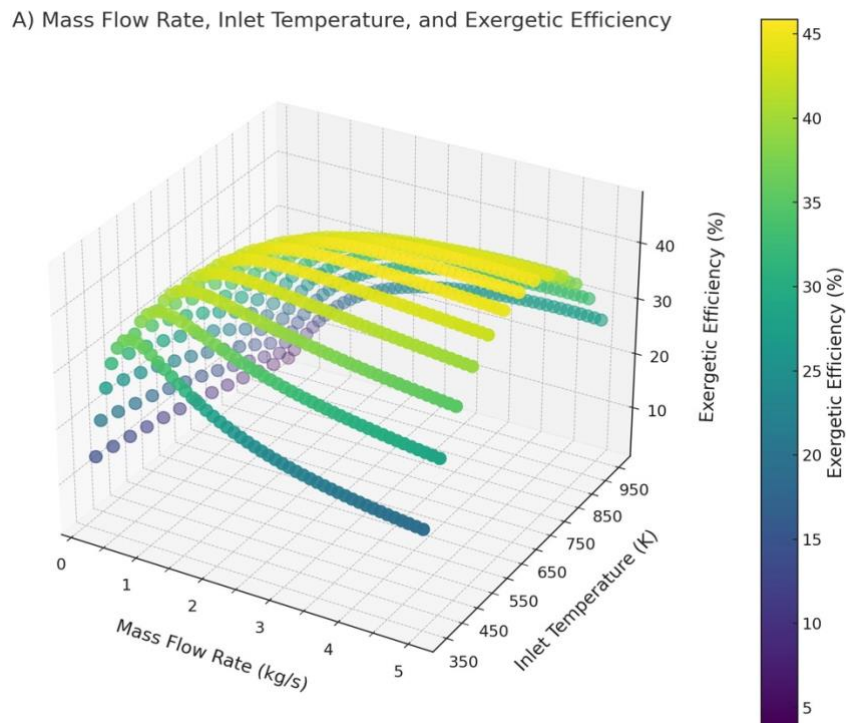
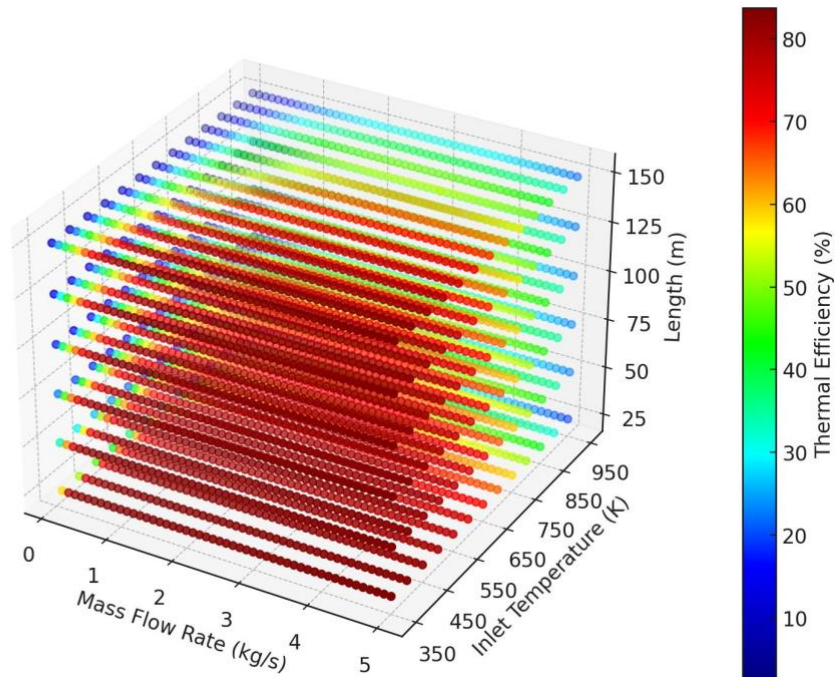


Fig. 4.3 Combinations of examined indexes and their impact on energetic efficiency at a length of 150m and P_{in} of 20MPa.

4.2 Parametric analysis across different PTSC lengths.

A) 3D Scatter Plot: Z Axis = Length (m), Colour = Thermal Efficiency (%)



B) 3D Scatter Plot: Z Axis = Length (m), Colour = Thermal Efficiency (%)

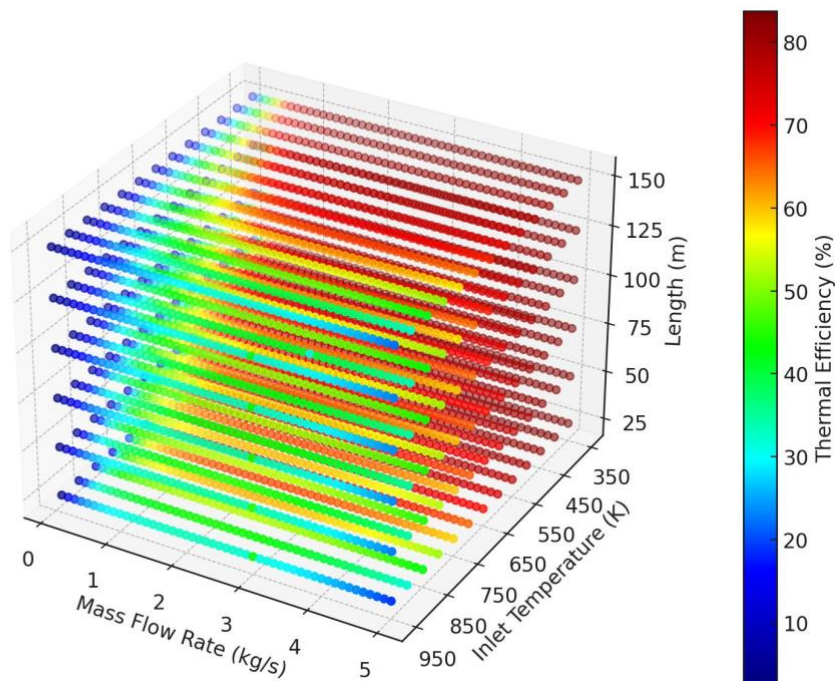
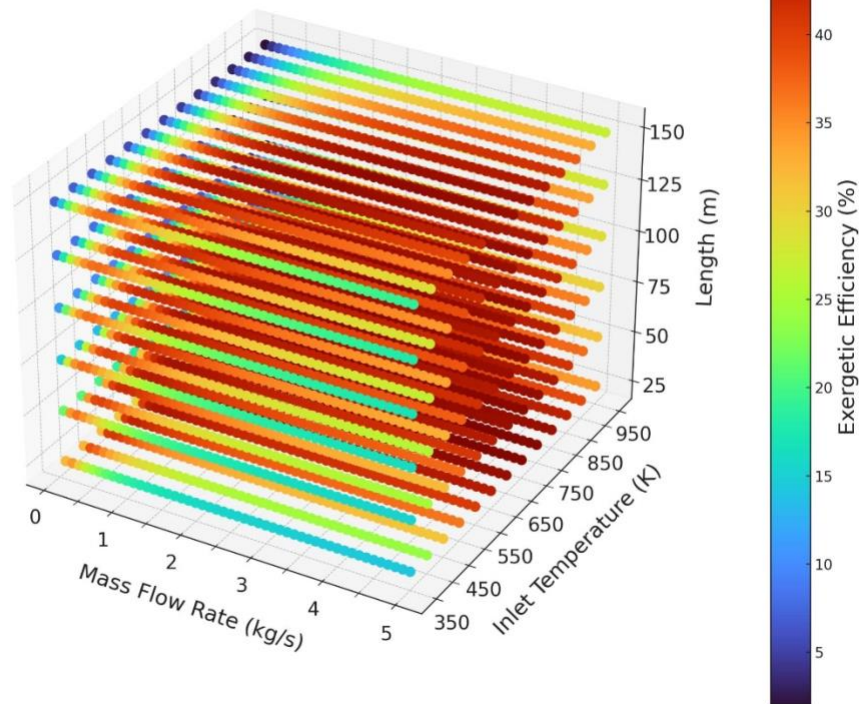


Fig. 4.4 Thermal efficiency at various lengths with multiple permutations, displayed from different angles: (A) and (B).

Fig. 4.4 shows the thermal efficiency across the examined permutations and its relationship with the length at 20MPa. It is evident from the graph that the highest efficiency tends to be at lower inlet temperatures. Additionally, as length increases, the peak efficiency tends to be at higher flow rates. At lower flow rates, the fluid's capacity to absorb heat is restricted relative to the amount of heat available over an extended length, resulting in overcapacity. However, mass flow rates that are too high lead to high pumping power requirements, which explains the presence of a maximum point. When comparing at the same flow rate and inlet temperature, an increase in tube length generally decreases efficiency. This is due to the increased heat loss potential and the reduced temperature differential between the fluid and the receiver tube as the fluid progresses axially, leading to lower heat transfer efficiency. Additionally, longer tubes contribute to a higher cumulative pressure drop. Notably, the optimal efficiency window narrows with increasing inlet temperatures across all tube lengths. This is due to the interplay of thermal overcapacity and the impact of pressure drop, as higher inlet temperatures reduce fluid density. This reduction in density, caused by the fluid's increasing deviation from the critical point, results in its behaviour more akin to a gas, thereby increasing the required pumping power.

A) 3D Scatter Plot: Z Axis = Length (m), Color = Exergetic Efficiency (%)



B) 3D Scatter Plot: Z Axis = Length (m), Color = Exergetic Efficiency (%)

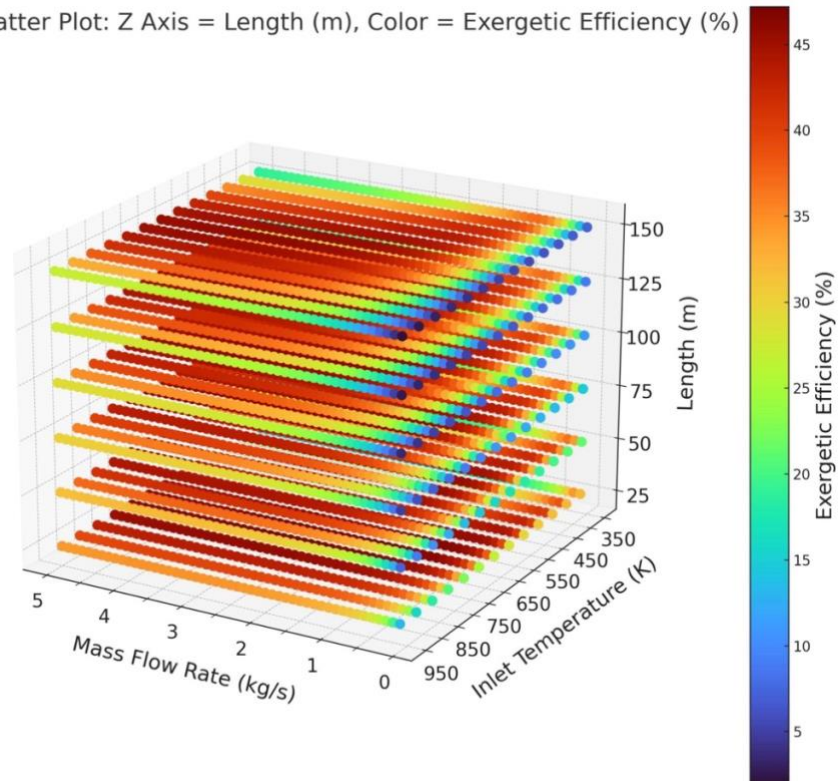


Fig. 4.5 Exergetic efficiency at various lengths with multiple permutations, displayed from different angles: (A) and (B).

Fig. 4.5 illustrates the exergetic efficiency with the same conditions as Fig. 4.4. The notable takeaway is that the inlet temperatures of 550 K–800 K show reasonably high performance across all lengths, and each length has a distinct optimal point. For example, for 150 m, the optimal efficiency is around 2.7–4.5 kg/s at inlet temperatures of 600 K–700 K, but for 50 m, it is 1.5–3.3 kg/s at 650–700 K inlet temperatures. However, the most interesting observation is that, when compared at the same mass flow rate and inlet temperature, shorter lengths (25 m and 50 m) consistently show higher efficiencies at low mass flow rates (0.1 to 1.0 kg/s) across all temperatures. At medium mass flow rates (1.0 to 3.0 kg/s), medium lengths (75 m and 100 m) tend to perform optimally, particularly at moderate to high temperatures. For high mass flow rates (3.0 to 5.0 kg/s), longer lengths (100 m to 150 m) exhibit higher efficiencies, maintaining relatively high performance even at the highest inlet temperatures. This trend highlights the importance of selecting the appropriate parabolic trough length based on specific operational conditions to maximise exergetic efficiency. The observed trend can be attributed to the balance between heat transfer and fluid dynamics. In shorter troughs, lower mass flow rates allow for longer residence time, enhancing heat absorption and efficiency. In contrast, longer troughs benefit from larger total heat exchange areas (larger aperture and HCE area), which require higher mass flow rates and higher total heat capacity of the fluid to optimise heat transfer and minimise thermal losses. Consequently, the optimal length varies with the flow rate to ensure effective thermal performance across different operational conditions.

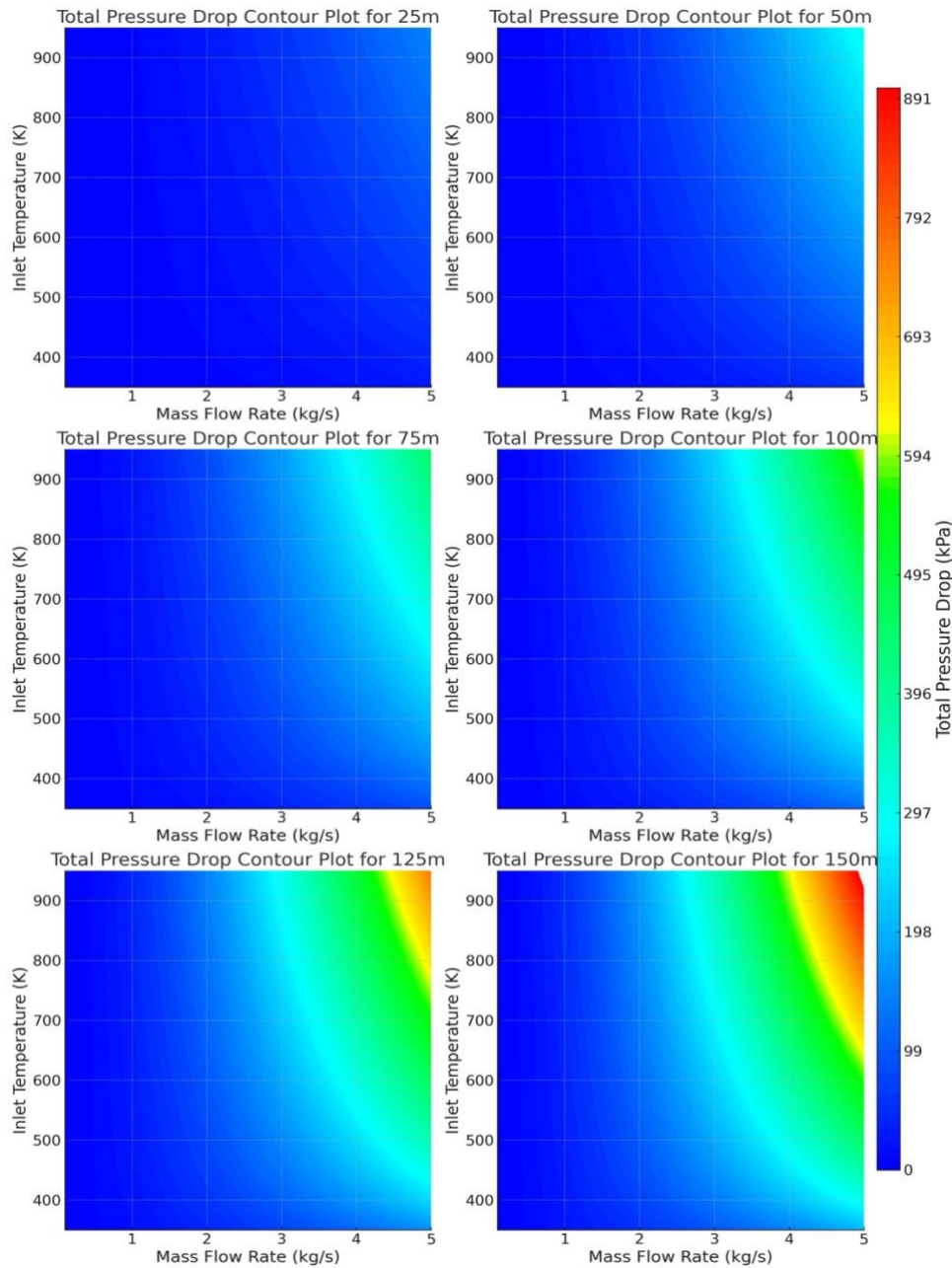
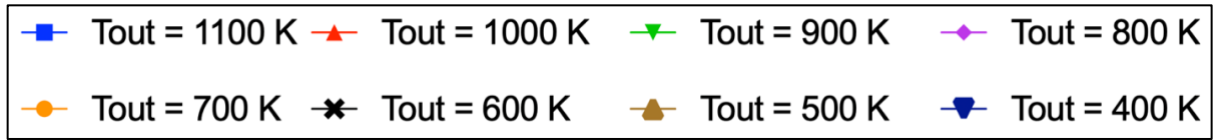


Fig. 4.6. Pressure drop (ΔP) across various inlet temperatures, flow rates and length

Fig. 4.6 depicts the total pressure drop across the examined inlet temperatures and length. This is an important metric as it impacts both thermal and exergetic efficiency. Evidently, the

changes in pressure drop do not vary significantly at mass flow rates of around ≤ 3 kg/s. However, at higher mass flow rates (>3 kg/s), the increase in pressure drop becomes much more pronounced, particularly in the longer troughs. This behaviour is mainly due to the fluid entering a more turbulent regime, where frictional forces within the trough increase significantly. Additionally, the longer lengths exacerbate these losses due to the extended distance the fluid must travel, allowing more interactions with the tube's surface, thus increasing drag. At higher inlet temperatures, changes in the fluid's properties, such as density and viscosity, further intensify these effects, leading to the observed non-linear rise in pressure drop.

Complementing Fig. 4.4 and Fig. 4.5, Table 4.1 presents the optimal exergetic points for various lengths at different output temperature levels for thermal and exergetic efficiency, respectively (refer to the data availability section for thermal efficiency data). This table differs from Fig. 4.5 as it categorises the data by outlet temperature, providing a more comprehensive performance overview across various combinations. These temperature outputs reflect the diverse applications of parabolic trough systems, including domestic water heating, desalination, electricity generation, and high-temperature industrial heating. The table serves as a practical guide for identifying the ideal length, mass flow rate, and temperature outputs to achieve the best thermal and exergetic efficiency for specific applications. The table shows that the optimal points for both inlet pressures are around T_{in} of 450 K to 750 K. Additionally, 20MPa showed a marginally better performance for each outlet temperature level. While these exact values are not definitive, this guide and the comprehensive data provided in APPENDIX 4 lead to a more informed decisions in PTSC design considerations. In addition to the efficiency, the mass flow rate should also be considered since the total heat content is another crucial factor for real-world applications.

Table 4.1 The optimal exergetic values for each length across the examined parametric indexes.

Length (m)	P_{in}	Low (350 K – 500 K)	Medium (500 K – 700 K)	High (≥ 700 K – 900 K)	Ultra-high ≥ 900 K
25	20 MPa	$T_{in} = 450$ K $\dot{m} = 1.7$ kg/s $\eta_{ex} = 33.06\%$	$T_{in} = 650$ K $\dot{m} = 1.8$ kg/s $\eta_{ex} = 45.92\%$	$T_{in} = 700$ K $\dot{m} = 2.6$ kg/s $\eta_{ex} = 46.67\%$	$T_{in} = 750$ K $\dot{m} = 0.4$ kg/s $\eta_{ex} = 40.13\%$
	9 MPa	$T_{in} = 450$ K $\dot{m} = 2$ kg/s $\eta_{ex} = 33.04\%$	$T_{in} = 650$ K $\dot{m} = 1.8$ kg/s $\eta_{ex} = 45.73\%$	$T_{in} = 700$ K $\dot{m} = 1.8$ kg/s $\eta_{ex} = 46.12\%$	$T_{in} = 750$ K $\dot{m} = 0.4$ kg/s $\eta_{ex} = 39.9\%$
50	20 MPa	$T_{in} = 450$ K $\dot{m} = 3.4$ kg/s $\eta_{ex} = 33.14\%$	$T_{in} = 650$ K $\dot{m} = 3.5$ kg/s $\eta_{ex} = 45.96\%$	$T_{in} = 700$ K $\dot{m} = 3$ kg/s $\eta_{ex} = 46.41\%$	$T_{in} = 700$ K $\dot{m} = 0.7$ kg/s $\eta_{ex} = 42.59\%$
	9 MPa	$T_{in} = 450$ K $\dot{m} = 3.9$ kg/s $\eta_{ex} = 32.64\%$	$T_{in} = 600$ K $\dot{m} = 1.9$ kg/s $\eta_{ex} = 44.69\%$	$T_{in} = 650$ K $\dot{m} = 1.7$ kg/s $\eta_{ex} = 45.93\%$	$T_{in} = 600$ K $\dot{m} = 0.5$ kg/s $\eta_{ex} = 42.46\%$
75	20 MPa	$T_{in} = 400$ K $\dot{m} = 5$ kg/s $\eta_{ex} = 33.31\%$	$T_{in} = 600$ K $\dot{m} = 2.7$ kg/s $\eta_{ex} = 45.22\%$	$T_{in} = 700$ K $\dot{m} = 3.1$ kg/s $\eta_{ex} = 46.3\%$	$T_{in} = 700$ K $\dot{m} = 1.0$ kg/s $\eta_{ex} = 42.7\%$
	9 MPa	$T_{in} = 400$ K $\dot{m} = 2.9$ kg/s $\eta_{ex} = 29.71\%$	$T_{in} = 600$ K $\dot{m} = 2.8$ kg/s $\eta_{ex} = 44.62\%$	$T_{in} = 600$ K $\dot{m} = 2.3$ kg/s $\eta_{ex} = 45.5\%$	$T_{in} = 650$ K $\dot{m} = 2.3$ kg/s $\eta_{ex} = 43.27\%$
100	20 MPa	$T_{in} = 400$ K $\dot{m} = 3.1$ kg/s $\eta_{ex} = 29.54\%$	$T_{in} = 400$ K $\dot{m} = 3.6$ kg/s $\eta_{ex} = 45.17\%$	$T_{in} = 700$ K $\dot{m} = 3.6$ kg/s $\eta_{ex} = 46.09\%$	$T_{in} = 600$ K $\dot{m} = 1.0$ kg/s $\eta_{ex} = 43.59\%$
	9 MPa	$T_{in} = 400$ K $\dot{m} = 3.9$ kg/s $\eta_{ex} = 29.46\%$	$T_{in} = 550$ K $\dot{m} = 2.6$ kg/s $\eta_{ex} = 43.61\%$	$T_{in} = 650$ K $\dot{m} = 2.3$ kg/s $\eta_{ex} = 45.58\%$	$T_{in} = 650$ K $\dot{m} = 1.2$ kg/s $\eta_{ex} = 43.49\%$
125	20 MPa	$T_{in} = 400$ K $\dot{m} = 3.9$ kg/s $\eta_{ex} = 29.73\%$	$T_{in} = 600$ K $\dot{m} = 4.4$ kg/s $\eta_{ex} = 45.07\%$	$T_{in} = 650$ K $\dot{m} = 3.3$ kg/s $\eta_{ex} = 46.15\%$	$T_{in} = 700$ K $\dot{m} = 1.8$ kg/s $\eta_{ex} = 43.61\%$
	9 MPa	$T_{in} = 400$ K $\dot{m} = 4.9$ kg/s $\eta_{ex} = 29.04\%$	$T_{in} = 550$ K $\dot{m} = 3.2$ kg/s $\eta_{ex} = 43.24\%$	$T_{in} = 600$ K $\dot{m} = 2.2$ kg/s $\eta_{ex} = 45.28\%$	$T_{in} = 600$ K $\dot{m} = 1.3$ kg/s $\eta_{ex} = 43.66\%$
150	20 MPa	$T_{in} = 400$ K $\dot{m} = 4.6$ kg/s $\eta_{ex} = 29.77\%$	$T_{in} = 550$ K $\dot{m} = 3.6$ kg/s $\eta_{ex} = 44.06\%$	$T_{in} = 650$ K $\dot{m} = 3.6$ kg/s $\eta_{ex} = 46.05\%$	$T_{in} = 600$ K $\dot{m} = 1.5$ kg/s $\eta_{ex} = 43.87\%$
	9 MPa	$T_{in} = 350$ K $\dot{m} = 3.7$ kg/s $\eta_{ex} = 25.22\%$	$T_{in} = 550$ K $\dot{m} = 3.8$ kg/s $\eta_{ex} = 42.61\%$	$T_{in} = 600$ K $\dot{m} = 2.4$ kg/s $\eta_{ex} = 45.08\%$	$T_{in} = 650$ K $\dot{m} = 1.8$ kg/s $\eta_{ex} = 43.57\%$

Table 4.2 and Table 4.3 highlight optimal thermal and exergetic efficiency combinations for fixed outlet temperatures, revealing that a 50m length is optimal for both energetic and exergetic efficiency. This shows that at 50m, at least in terms of efficiency, strikes the perfect balance between the thermal and exergetic gains, thermal losses, and exergetic destruction through pressure drop that is associated with varied PTSC length and should be considered for design optimisation.

Table 4.2 The optimal energetic values for each length across at fixed outlet temperature (T_{out})

T_{out} (K)	25m	50m	75m	100m	125m	150m
1100	$T_{in} = 350$ K $\eta_{th} = 30.16$ $\dot{m} = 0.0406$	$T_{in} = 350$ K $\eta_{th} = 49.09$ $\dot{m} = 0.132$	$T_{in} = 350$ K $\eta_{th} = 40.78$ $\dot{m} = 0.165$	$T_{in} = 350$ K $\eta_{th} = 42.49$ $\dot{m} = 0.229$	$T_{in} = 350$ K $\eta_{th} = 43.57$ $\dot{m} = 0.294$	$T_{in} = 350$ K $\eta_{th} = 44.32$ $\dot{m} = 0.359$
1000	$\eta_{th} = 30.16$ $T_{in} = 350$ K $\dot{m} = 0.0089$	$\eta_{th} = 69.54$ $T_{in} = 350$ K $\dot{m} = 0.214$	$\eta_{th} = 64.41$ $T_{in} = 350$ K $\dot{m} = 0.298$	$\eta_{th} = 65.41$ $T_{in} = 350$ K $\dot{m} = 0.404$	$\eta_{th} = 66.03$ $T_{in} = 350$ K $\dot{m} = 0.51$	$\eta_{th} = 66.45$ $T_{in} = 350$ K $\dot{m} = 0.615$
900	$T_{in} = 350$ K $\eta_{th} = 69.01$ $\dot{m} = 0.124$	$T_{in} = 350$ K $\eta_{th} = 78.03$ $\dot{m} = 0.281$	$T_{in} = 350$ K $\eta_{th} = 73.45$ $\dot{m} = 0.396$	$T_{in} = 350$ K $\eta_{th} = 74.05$ $\dot{m} = 0.532$	$T_{in} = 350$ K $\eta_{th} = 74.42$ $\dot{m} = 0.669$	$T_{in} = 350$ K $\eta_{th} = 74.66$ $\dot{m} = 0.805$
800	$T_{in} = 350$ K $\eta_{th} = 75.8$ $\dot{m} = 0.163$	$T_{in} = 350$ K $\eta_{th} = 82.49$ $\dot{m} = 0.343$	$T_{in} = 350$ K $\eta_{th} = 78.42$ $\dot{m} = 0.505$	$T_{in} = 350$ K $\eta_{th} = 78.75$ $\dot{m} = 0.676$	$T_{in} = 350$ K $\eta_{th} = 78.95$ $\dot{m} = 0.848$	$T_{in} = 350$ K $\eta_{th} = 79.08$ $\dot{m} = 1.02$
700	$T_{in} = 350$ K $\eta_{th} = 79.9$ $\dot{m} = 0.213$	$T_{in} = 350$ K $\eta_{th} = 84.81$ $\dot{m} = 0.451$	$T_{in} = 350$ K $\eta_{th} = 81.25$ $\dot{m} = 0.659$	$T_{in} = 350$ K $\eta_{th} = 81.42$ $\dot{m} = 0.867$	$T_{in} = 350$ K $\eta_{th} = 81.51$ $\dot{m} = 1.09$	$T_{in} = 350$ K $\eta_{th} = 81.57$ $\dot{m} = 1.30$
600	$T_{in} = 350$ K $\eta_{th} = 82.24$ $\dot{m} = 0.287$	$T_{in} = 350$ K $\eta_{th} = 85.84$ $\dot{m} = 0.6$	$T_{in} = 350$ K $\eta_{th} = 82.82$ $\dot{m} = 0.868$	$T_{in} = 350$ K $\eta_{th} = 82.88$ $\dot{m} = 1.16$	$T_{in} = 350$ K $\eta_{th} = 82.91$ $\dot{m} = 1.45$	$T_{in} = 350$ K $\eta_{th} = 82.91$ $\dot{m} = 1.74$
500	$T_{in} = 350$ K $\eta_{th} = 83.44$ $\dot{m} = 0.43$	$T_{in} = 350$ K $\eta_{th} = 86.07$ $\dot{m} = 0.89$	$T_{in} = 350$ K $\eta_{th} = 83.61$ $\dot{m} = 1.29$	$T_{in} = 350$ K $\eta_{th} = 83.62$ $\dot{m} = 1.73$	$T_{in} = 350$ K $\eta_{th} = 83.61$ $\dot{m} = 2.16$	$T_{in} = 350$ K $\eta_{th} = 83.57$ $\dot{m} = 2.59$
400	$T_{in} = 350$ K $\eta_{th} = 83.94$ $\dot{m} = 1.0$	$T_{in} = 350$ K $\eta_{th} = 85.8$ $\dot{m} = 2.05$	$T_{in} = 350$ K $\eta_{th} = 83.97$ $\dot{m} = 3.01$	$T_{in} = 350$ K $\eta_{th} = 83.94$ $\dot{m} = 4.02$	$T_{in} = 350$ K $\eta_{th} = 83.88$ $\dot{m} = 5.03$	$T_{in} = 350$ K $\eta_{th} = 83.79$ $\dot{m} = 6.03$

Table 4.3 The highest exergetic values for each length across at fixed outlet temperature (T_{out})

T_{out} (K)	25m	50m	75m	100m	125m	150m
1100	$T_{in} = 350$ K $\eta_{ex} = 18.32$ $\dot{m} = 0.0406$	$T_{in} = 450$ K $\eta_{ex} = 28.66$ $\dot{m} = 0.151$	$T_{in} = 400$ K $\eta_{ex} = 23.43$ $\dot{m} = 0.176$	$T_{in} = 400$ K $\eta_{ex} = 24.38$ $\dot{m} = 0.245$	$T_{in} = 450$ K $\eta_{ex} = 25.01$ $\dot{m} = 0.330$	$T_{in} = 450$ K $\eta_{ex} = 25.44$ $\dot{m} = 0.403$
1000	$T_{in} = 550$ K $\eta_{ex} = 33.59$ $\dot{m} = 0.127$	$T_{in} = 600$ K $\eta_{ex} = 41.61$ $\dot{m} = 0.345$	$T_{in} = 550$ K $\eta_{ex} = 37.88$ $\dot{m} = 0.43$	$T_{in} = 550$ K $\eta_{ex} = 38.48$ $\dot{m} = 0.582$	$T_{in} = 550$ K $\eta_{ex} = 38.85$ $\dot{m} = 0.734$	$T_{in} = 550$ K $\eta_{ex} = 39.1$ $\dot{m} = 0.887$
900	$T_{in} = 700$ K $\eta_{ex} = 41.07$ $\dot{m} = 0.339$	$T_{in} = 650$ K $\eta_{ex} = 47.0$ $\dot{m} = 0.635$	$T_{in} = 650$ K $\eta_{ex} = 43.39$ $\dot{m} = 0.879$	$T_{in} = 650$ K $\eta_{ex} = 43.71$ $\dot{m} = 1.18$	$T_{in} = 650$ K $\eta_{ex} = 43.89$ $\dot{m} = 1.48$	$T_{in} = 650$ K $\eta_{ex} = 44.01$ $\dot{m} = 1.79$
800	$T_{in} = 750$ K $\eta_{ex} = 45.46$ $\dot{m} = 1.53$	$T_{in} = 700$ K $\eta_{ex} = 49.45$ $\dot{m} = 1.71$	$T_{in} = 700$ K $\eta_{ex} = 46.03$ $\dot{m} = 2.39$	$T_{in} = 700$ K $\eta_{ex} = 46.02$ $\dot{m} = 3.19$	$T_{in} = 650$ K $\eta_{ex} = 46.03$ $\dot{m} = 2.72$	$T_{in} = 650$ K $\eta_{ex} = 46.00$ $\dot{m} = 3.27$
700	$T_{in} = 650$ K $\eta_{ex} = 45.92$ $\dot{m} = 1.717$	$T_{in} = 650$ K $\eta_{ex} = 48.69$ $\dot{m} = 3.658$	$T_{in} = 650$ K $\eta_{ex} = 45.58$ $\dot{m} = 5.188$	$T_{in} = 600$ K $\eta_{ex} = 45.21$ $\dot{m} = 3.499$	$T_{in} = 600$ K $\eta_{ex} = 45.08$ $\dot{m} = 4.379$	$T_{in} = 600$ K $\eta_{ex} = 44.85$ $\dot{m} = 5.261$
600	$T_{in} = 500$ K $\eta_{ex} = 41.9$ $\dot{m} = 0.8804$	$T_{in} = 550$ K $\eta_{ex} = 43.73$ $\dot{m} = 3.738$	$T_{in} = 550$ K $\eta_{ex} = 41.68$ $\dot{m} = 5.373$	$T_{in} = 550$ K $\eta_{ex} = 41.27$ $\dot{m} = 7.176$	$T_{in} = 550$ K $\eta_{ex} = 40.55$ $\dot{m} = 8.988$	$T_{in} = 500$ K $\eta_{ex} = 39.89$ $\dot{m} = 5.328$
500	$T_{in} = 450$ K $\eta_{ex} = 33.14$ $\dot{m} = 1.644$	$T_{in} = 450$ K $\eta_{ex} = 34.31$ $\dot{m} = 3.401$	$T_{in} = 450$ K $\eta_{ex} = 33.33$ $\dot{m} = 4.945$	$T_{in} = 450$ K $\eta_{ex} = 33.54$ $\dot{m} = 6.603$	$T_{in} = 450$ K $\eta_{ex} = 33.85$ $\dot{m} = 8.272$	$T_{in} = 450$ K $\eta_{ex} = 34.3$ $\dot{m} = 9.96$
400	$T_{in} = 350$ K $\eta_{ex} = 18.09$ $\dot{m} = 1.004$	$T_{in} = 350$ K $\eta_{ex} = 18.53$ $\dot{m} = 2.053$	$T_{in} = 350$ K $\eta_{ex} = 18.21$ $\dot{m} = 3.015$	$T_{in} = 350$ K $\eta_{ex} = 18.35$ $\dot{m} = 4.021$	$T_{in} = 350$ K $\eta_{ex} = 18.58$ $\dot{m} = 5.027$	$T_{in} = 350$ K $\eta_{ex} = 18.92$ $\dot{m} = 6.034$

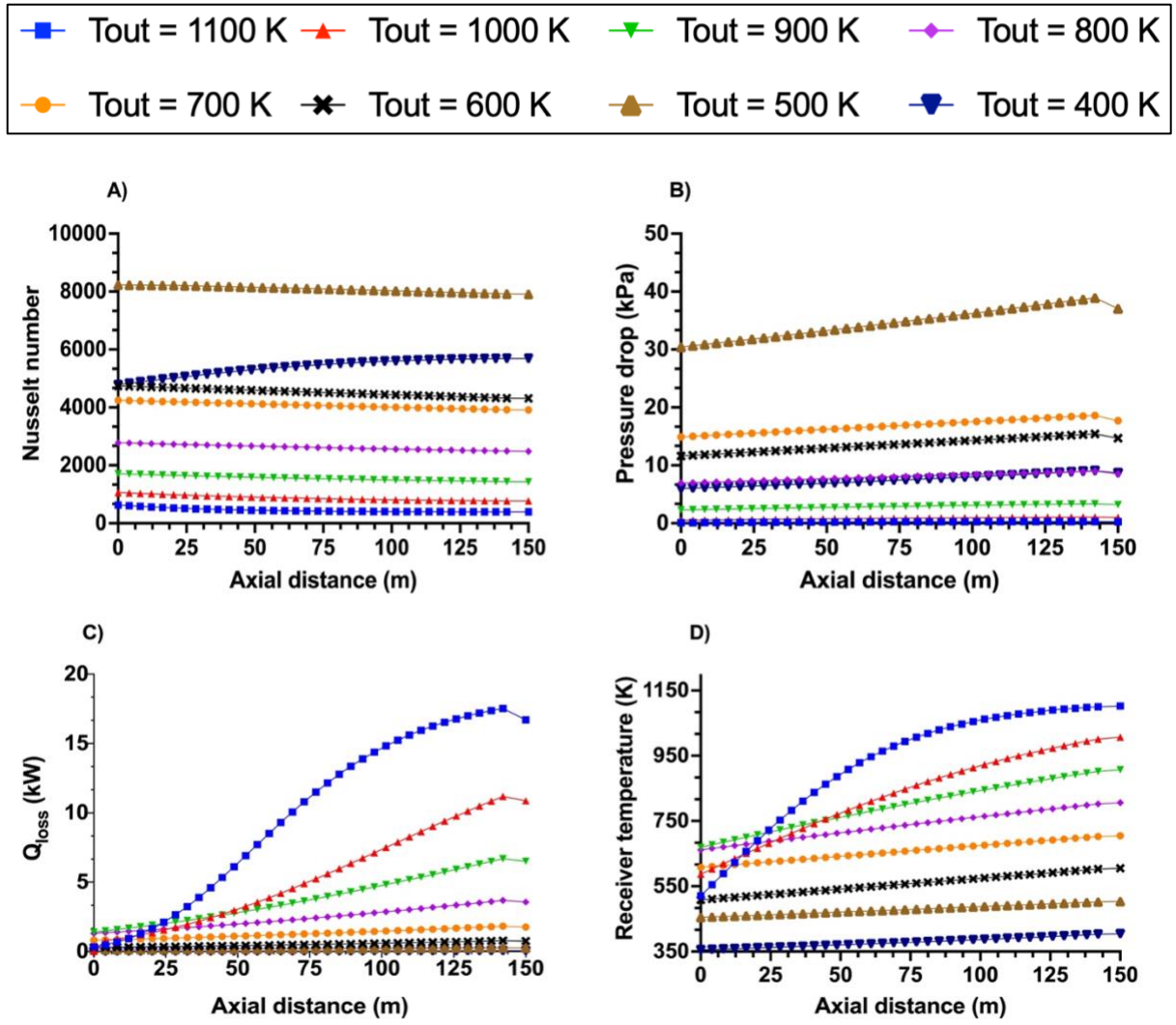


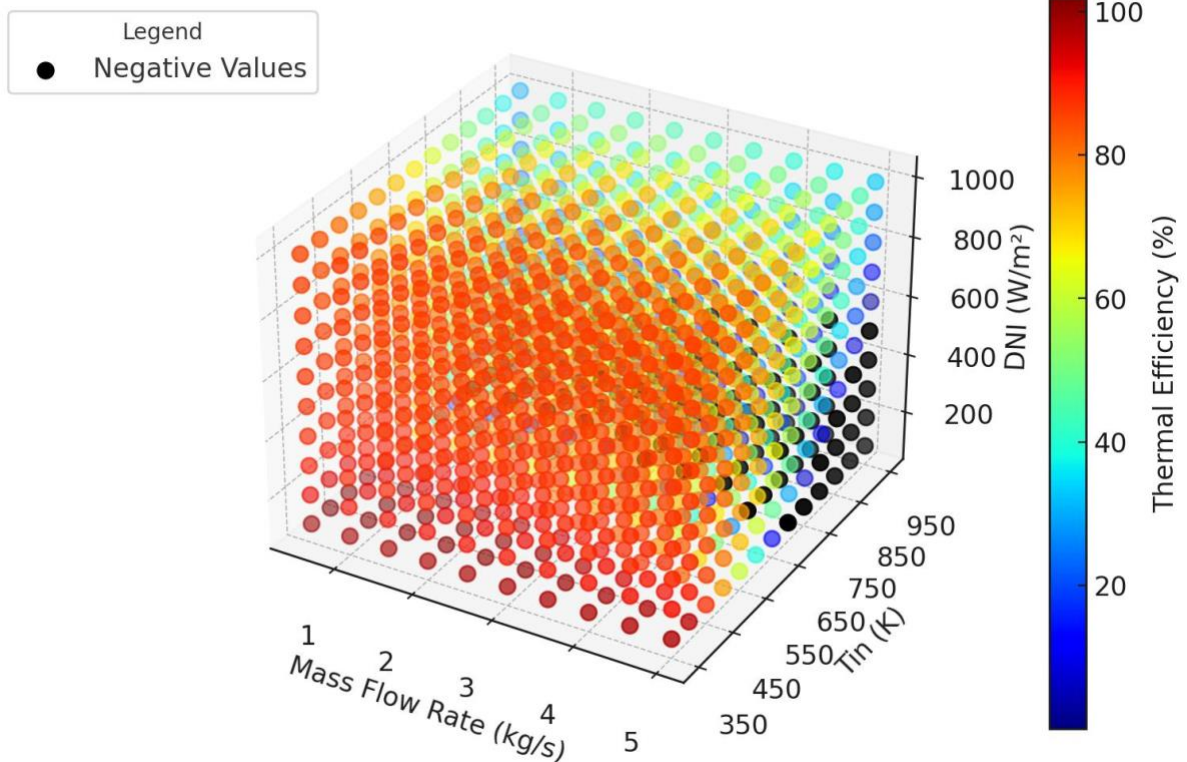
Fig. 4.7 The changes for each unit of the selected indexes across the length of the tube (axial distance)

Fig. 4.7 illustrates four key indices for each unit of the PTSC along its 150m length, using the optimised exergetic inputs in Table 4.3. These values represent individual unit measurements, not cumulative totals. As depicted in Fig. 4.7A, the Nusselt number is highest at an outlet temperature of 500 K and lowest at 1100 K, which can be attributed to the corresponding highest and lowest mass flow rates. In addition, the trends in heat loss and receiver temperature are similar, as they are interrelated through radiative loss. The small drop at the end is due to the final segment being shorter than the rest to achieve the total length of

150m. The discretised model used in this study can inform future designs by deciphering the axial variations in Nusselt number, pressure drop, and heat loss. For instance, understanding these parameters can facilitate the strategic integration of heat transfer enhancements at specific axial lengths, such as inserts (Pandey et al. 2021), radiative shields (Wang et al. 2018) and innovative coatings (Hu et al. 2024). This targeted approach can enhance performance in regions with high heat loss while optimising efficiency by omitting enhancements in areas with lower heat loss. Similarly, techniques to improve the Nusselt number are likely to be more beneficial when applied at outlet temperatures between 800 K and 1000 K.

4.3 Sensitivity Analysis of Direct Normal Irradiance (DNI) on Parabolic Trough Solar Collectors

A) 3D Scatter Plot: Z Axis = DNI, Color = Thermal Efficiency (%)



B) 3D Scatter Plot: Z Axis = DNI, Color = Exergetic Efficiency (%)

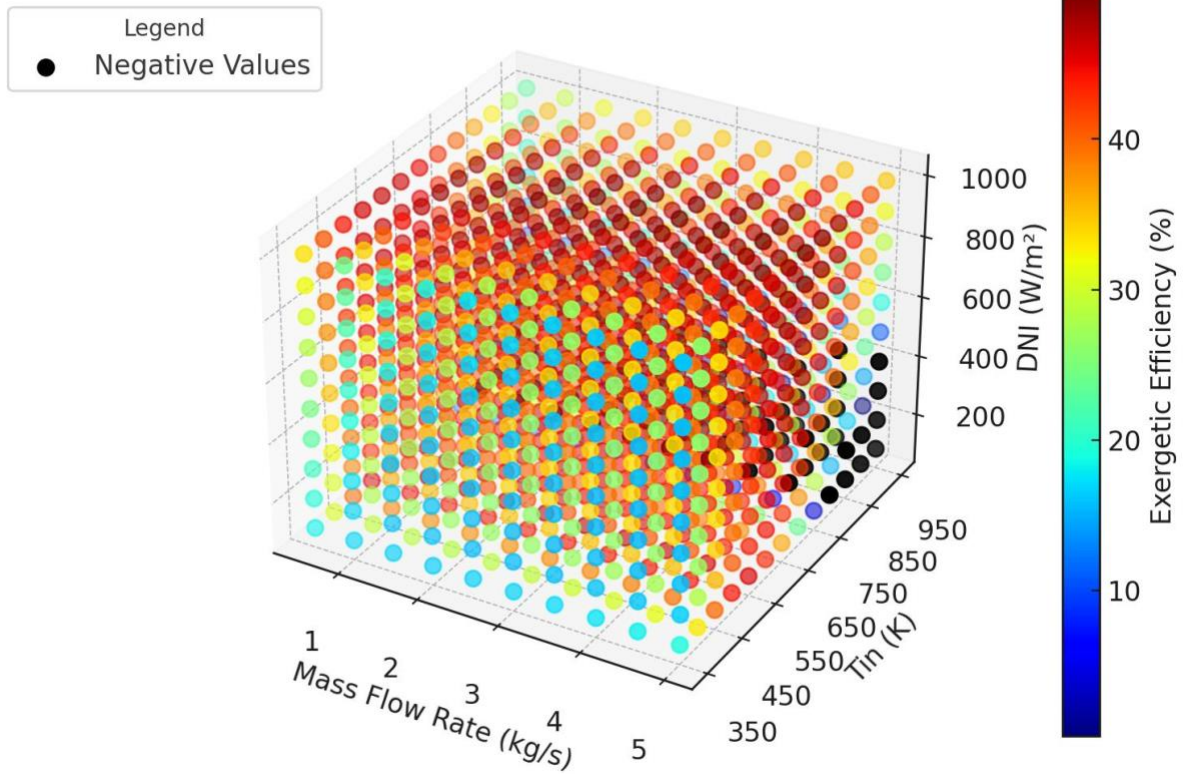


Fig. 4.8 Sensitivity analysis of varying DNI for different combinations of mass flow rate and Inlet temperature at the length of 50m.

Fig. 4.8 A and B illustrate the impact of varying DNI on performance across different combinations of mass flow rates and inlet temperatures, with respect to thermal and exergetic efficiency, respectively. The analysis was conducted at a length of 50m, selected for its previously demonstrated optimal efficiency. The results show that DNI does not change the optimised values when analysed at different DNIs, whereby lower inlet temperatures lead to higher thermal efficiency, while middling inlet temperatures lead to the highest exergetic efficiency. However, DNI reduction leads to different output temperatures, which leads to different optimised values, which is an important distinction in this research. Please refer to Appendix 4 for the full data of the optimised values at various combinations of different DNIs,

targeted outlet temperatures and lengths of PTSC. Another notable observation here is that lower DNI leads to higher thermal efficiency at lower inlet temperatures and mass flow rates. However, the values also reveal a turning point (around $T_{in} = 500K$ @ $4.5kg/s$) to the previously mentioned observation, as inlet temperatures and mass flow rates increase simultaneously, efficiency starts to drop rather than rise. This finding is significant because previous studies in the literature have only demonstrated that decreasing DNI consistently leads to lower efficiency due to limited parametric boundary conditions and single-unit analyses. The results stem from the fact that at lower inlet temperatures, reducing DNI lowers the outlet temperature, which decreases the tube's average temperature and reduces radiative losses, as shown in Equation A-13 in Appendix 1. However, this effect is less significant or reversed at higher inlet temperatures and flow rates due to the already substantial radiative losses, and the lower DNI values lead to lower heat absorption in relation to heat loss, unlike at lower inlet temperatures.

However, the opposite trend can be seen with exergetic efficiency as it constantly reduces with the reduction of DNI. The exergetic efficiency generally decreases with lower DNI, except for certain high DNI values and at a low mass flow rate of 0.5 kg/s and inlet temperatures between $550K$ and $700K$, where a slight improvement was observed. This exception occurs because, at low mass flow rates and moderate inlet temperatures, the reduced DNI leads to lower outlet temperatures, which helps minimise exergy destruction by reducing radiative losses and maintaining a better temperature gradient. However, outside this specific range, the lower DNI results in diminished energy quality and increased irreversibilities, leading to an overall reduction in exergetic efficiency. At low DNIs of around $200\text{-}450\text{ W/m}^2$ and high inlet temperatures, negative values for both thermal and exergetic efficiencies can be observed. At high inlet temperatures, the insufficient DNI results in low absorber tube temperatures, resulting in the fluid being hotter than the receiver tube and creating a reverse temperature gradient that causes heat transfer to primarily occur in the opposite direction.

4.4 Comparison with existing research and discussions on the novel contributions of this research

This study advances PTSC optimisation by employing a rarely used approach, discretising long-length collectors. In contrast to much of the existing literature, which has mainly analysed single-unit PTSCs with inlet temperature as the primary variable, this work demonstrates that such simplification overlooks the complexity of system behaviour. As shown in Fig. 4.2, Fig. 4.3, and Table 4.1, different collector lengths demand specific inlet temperatures and mass flow rates to achieve their target outlet temperatures. Because the outlet temperature directly determines the usefulness of the heat for downstream applications, it must be optimised for each PTSC length rather than assumed from single-unit analysis. For instance, Table 4.3 shows that outlet temperatures up to 1100 K yield maximum efficiency at relatively low inlet temperatures (350–450 K), contrary to earlier assumptions that such values apply only to low-temperature uses.

This study also provides a systematic comparison across lengths, revealing that a 50 m PTSC achieves the best efficiency, a finding not reported before. The discretisation method therefore offers a more robust framework for design, enabling tailored optimisation for different outlet temperatures, collector lengths, mass flow rates, and DNI levels. As summarised in Table 4.4, whereas earlier studies tended to focus narrowly on inlet conditions, this approach highlights the outlet temperature as the critical parameter over extended lengths. Full optimised thermal and exergetic efficiency results are provided in APPENDIX 4 and the link that is present in the section, offering a comprehensive dataset for future PTSC design and optimisation.

Table 4.4 Comparison of results obtained in this study with those reported in previous works.

T_{in} (K) Bellos & Tzivanidis (2017b)	$\dot{m}$ (kg/s) Bellos & Tzivanidis (2017b)	n_{ex} (%) Bellos & Tzivanidis (2017b)	T_{out} (K) Bellos & Tzivanidis (2017b)	T_{in} (K) Present study (L=150m)	$\dot{m}$ (kg/s) Present study (L=150m)	n_{ex} (%) Present study (L=150m)	T_{out} (K) Present study (L=150m)
350	0.5	15.5	374.9	350	0.8	37.01	902.8
450	0.5	31.5	479.8	450	1.3	42.29	832.3
550	0.5	40.3	578.9	550	2.1	44.92	791.9
650	2.0	44.9	656.5	650	3.6	46.05	787.6
750	2.5	45.3	754.7	750	4.6	44.29	847.0
850	2.5	40.4	853.8	850	4.9	38.28	924.4

4.5 Conclusion

This chapter used the discretised PTSC model to examine how inlet temperature, mass flow rate, collector length, and solar input level shape outlet temperature, efficiency, and pressure drop. The results showed that realistic design requires balancing temperature gains against increasing losses, with mid-range operating temperatures offering the strongest compromise between performance and pumping penalties. These trends provide clear guidance on choosing combinations that reach a desired outlet temperature without excessive losses, and comparison with prior work highlighted where the discretised method gives more defensible operating points than simpler approaches. Taken together, these findings define the outlet-temperature and operating-range boundaries that the next chapter uses as the thermal input conditions for analysing the integrated PTSC–sCO₂ Brayton cycle with desalination.

CHAPTER 5 THERMODYNAMIC ANALYSIS OF sCO₂

BRAYTON CYCLE WITH DIRECT CONTACT MEMBRANE

DISTILLATION

This chapter presents an integrated system model that combines a direct parabolic trough solar collector, a supercritical carbon dioxide (sCO₂) Brayton cycle, and a Direct Contact Membrane Distillation (DCMD) desalination unit. A mathematical framework is developed to represent the thermodynamic behaviour of the coupled power and desalination processes, allowing analysis of how operating conditions influence both energy conversion and water production within the same system. The research provides a fundamental understanding on how the system work.

5.1 Methods and boundary conditions for sCO₂ Brayton cycle

The recompression Brayton cycle layout was selected for this study due to its robust and superior performance (Al-Sulaiman & Atiff, 2015) and have been considered as the most promising by USDOE. In considering a wide range of Direct Normal Irradiance (DNI) values, the research necessitated an innovative approach for analysing and comparing the cycle. This innovation was driven by the complexity of the re-compression cycle, which does not readily accommodate alterations in heat input (Q_u) resulting from changes in DNI values. For example, a decrease in Q_u without corresponding adjustment in the mass flow rate ($\dot{m}_{sCO_2}$), could lead to turbine's outlet temperature T4 to fall below the fixed value of T5 = 650K (see Fig. 3.2) which is thermodynamically impossible (refer Fig. 3.6 for T-s diagram). To overcome this challenge, we introduced an optimisation method for mass flow rate that accounts for DNI changes and their impact on Q_u . This method, closely aligned with the principles described in Section 3.1, involves setting approach temperatures for HTR and LTR to at least 10K and 5K respectively, to withstand large heat loads as mentioned in (Linares et al., 2020). Utilising the

EES's parametric table function, the mass flow rate was determined in increments of 0.1 kg/s, selecting the rate that yielded the highest W_{net} while satisfying the set approach temperatures. Significantly, our findings consistently link the maximum W_{net} with mass flow rates that align with the 5K threshold of the LTR, under reference conditions ($50 MW_{net} @ 1000 W/m^2$). Moreover, it is worth noting that the optimisation of mass flow rates leads to nearly uniform temperatures across the cycle, despite varying DNI inputs, as illustrated. This computation was achieved through well-established thermodynamic techniques, including isentropic efficiency in turbines and compressors, as well as principles of effectiveness and energy balance for HTR and LTR. Comprehensive boundary conditions and other inputs for the reference state sCO_2 Brayton cycle are detailed in Table 3.4. The boundary and operating conditions for DCMD can be referred to Table 3.8.

5.2 Research conditions

The parametric conditions used in results and discussion section were carefully chosen to mirror actual operational conditions, considering the assumptions detailed earlier in Section 2.1. Our analysis of pressure ratios ranging from 2.7 to 3.6 aligns with standard practices in power cycle efficiency assessments. Given that advanced power cycles in commercial use today function at pressures up to 30,000 kPa, our chosen pressure ratios are realistic and attainable. This affirms that materials capable of withstanding these pressures are practical for real-world use, albeit with some modifications and advanced materials required for the PTSC's receiver tubes. Moreover, higher ratios lead to more expensive materials, making it essential to understand the trade-offs. These insights help designers to understand the thermodynamic impact of increasing the pressure ratio.

For the combinations of bulk feed and permeate temperatures of DCMD, the bulk feed temperature was evaluated across a range from 30 to 90°C, representing the typical minimum and maximum feed temperatures according to Soomro & Kim (2017) and Qtaishat and Banat (2013) with minimum of 10°C of bulk temperature difference across the membrane needed to satisfactorily produce water vapor across the membrane (Khayet, 2011). A maximum of 90°C was chosen as it was the maximum temperature that allows for usage in DCMD without significant negative impact on the membrane of DCMD, and it is within the range of the waste heat temperature input of a typical sCO₂ Brayton cycle. Conversely, the bulk permeate temperature was assessed from 20°C to 50°C. This range corresponds to the typical ambient temperature spectrum across many locations, indicating that maintaining operations within these temperatures negates the necessity for auxiliary heating or cooling of the permeate fluid.

5.3 Impact of high pressure on thermal and exergetic performance.

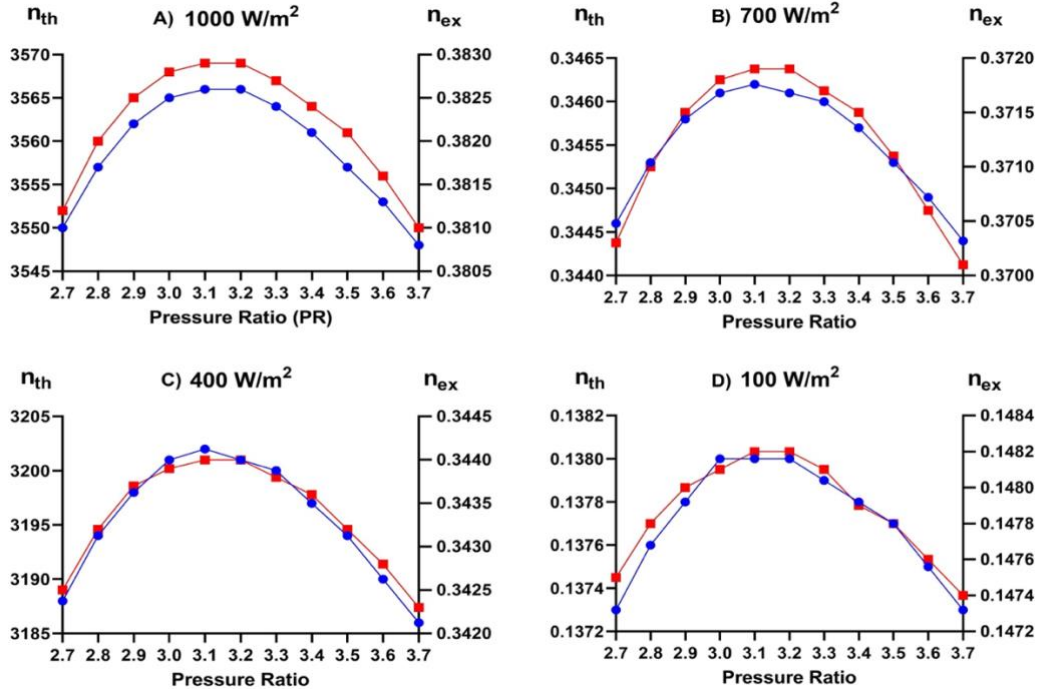


Fig. 5.1 Thermal and exergetic efficiency across various pressure ratio (PR) and DNI.

Fig. 5.1 presents the relationship between the system's thermal and exergetic efficiency with varying Pressure Ratio (PR) and Direct Normal Irradiance (DNI). A concurring trend emerges, with both metrics showing a decline as DNI decreases across all PR values. Importantly, as DNI values declined, the decrease in both exergetic and energetic efficiency progressively intensified. For instance, the exergetic efficiency slipped by 0.011% from 1000 to 700 W/m^2 , 0.0278% from 700 to 400 W/m^2 , and 0.1950% from 400 to 100 W/m^2 . Besides that, it's noticeable that even though PR initially rises, a drop after it reaches about 3.1 to 3.2 which peaked at around 0.383, showing the existence of an optimal point. This happens because as PR increases, $W_{turbine}$ simultaneously increases, but after 3.2, the compressor's work outpaced the increment of $W_{turbine}$, lowering its efficiency. Moreover, the improvement in exergetic and thermal percentage were stronger in higher DNI values than lower DNI values. For example, the increment in exergetic performance increased by 0.0017% from PR of 2.7 until 3.2 at 1000 W/m^2 but only by 0.0007% at 100 W/m^2 . This is due to diminished solar energy input resulting in a lower $W_{turbine}$ increment, relative to the increasing power demand of the compressor.

5.4 Impact of low pressure on thermal and exergetic performance.

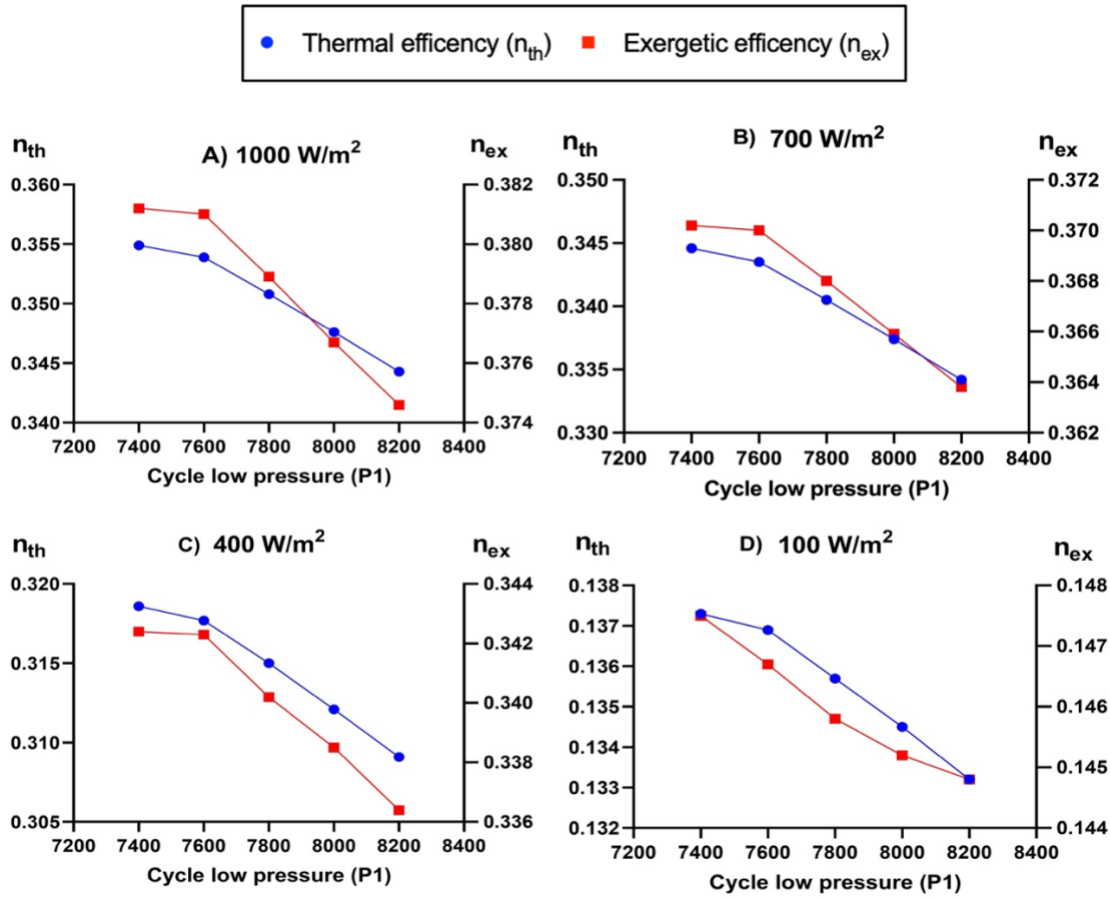


Fig. 5.2 Thermal and exergetic efficiency across various low cycle pressure (P_1) and DNI.

Fig. 5.2 illustrates the influence of adjusting the low pressure (P_1) in a cycle on both thermal and exergetic efficiency, maintaining the high-pressure constant at 19980kPa. Evidently, elevating P_1 consistently degrades thermal and exergy performance across all DNIs, suggesting the futility of exploring greater P_1 values. This performance deterioration transpires as the pressure drop disparity between the high and low cycle pressure diminishes, leading to a decline in turbine power output. Interestingly, a relatively small performance drop was observed from 7400 to 7600kPa. For example, the thermal efficiency drop from 7400kPa to 7600kPa is 0.001% whereas the drop is around 0.0031% to 0.0033% thereafter. This can be attributed to the fact that

at lower P1 values, which approach the critical point, there is a higher density of supercritical carbon dioxide (sCO₂). This, in turn, reduces the compressor's workload.

Performance evaluation of DCMD

In this section, the comparisons of important metrics on the performance of DCMD shown in section 3.5 and APPENDIX 2, were evaluated. Moreover, the water production capacity of DCMD under the system's different operating conditions and DNIs was investigated. Please note that in the sections below, the unit Celsius was used for reader's convenience despite the research was conducted using Kelvin.

5.4.1 Detailed analysis of Specific thermal energy consumption (STEC) and Evaporation efficiency (EE).

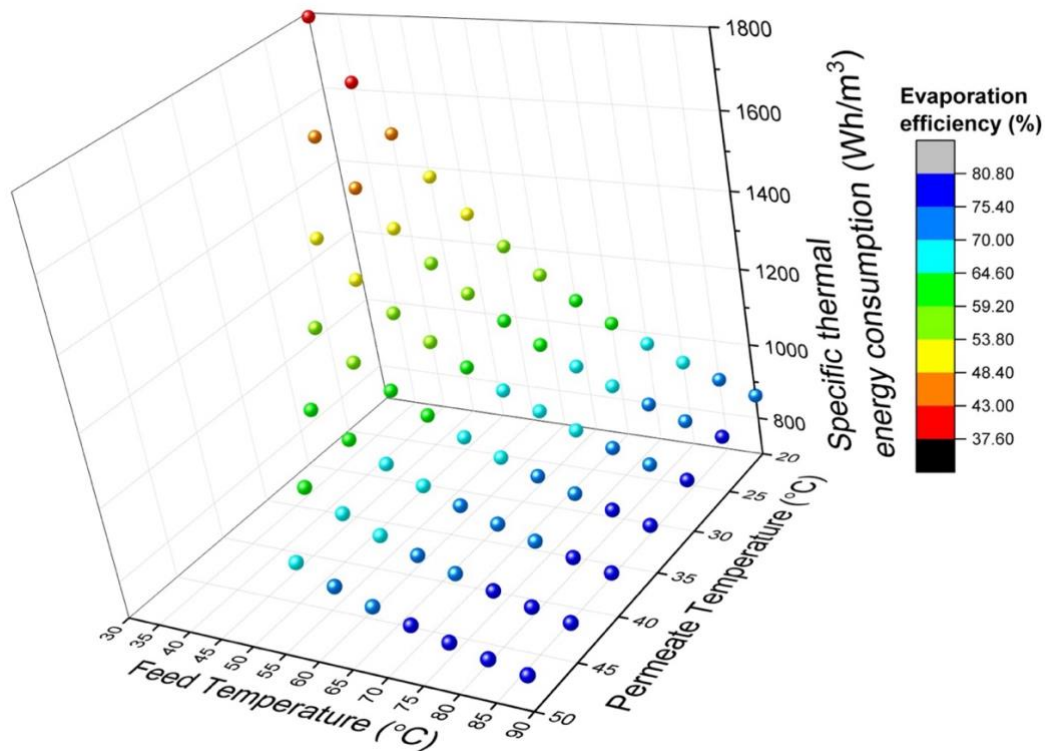
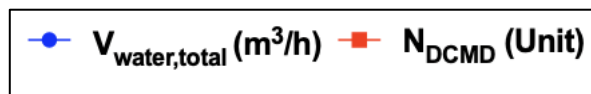


Fig. 5.3 The relationship between combinations of bulk Feed Temperature, T_{bf} (y-axis) and bulk Permeate temperature, T_{bp} (x-axis) with Specific thermal energy consumption, STEC (Z-axis) and Evaporation efficiency, EE (colour contour)

Fig. 5.3 exhibits the relationship between various combinations of bulk feed (T_{bf}), and bulk permeate (T_{bp}) temperatures and their impact on STEC and EE. Although the temperature at state 6 did not reach 90°C for the system used in this research, making it thermodynamically impossible for the cooler outlet to achieve this value, it is theoretically extended to 90°C in this study to facilitate deeper understanding. The clearly seen pattern in the figure is that STEC increases as feed temperatures diminish, and it descends with an increase in feed temperatures and bulk temperature difference (ΔT_b). Conversely, EE exhibits the inverse behaviour, which is consistent with the inherent properties of these two indices. A lower STEC results in lower thermal energy needed for evaporation which increases the EE. Another noteworthy point is that a rise in feed temperature yields superior EE values and inferior STEC figures in comparison to a reduction in permeate temperature, when assessed at an identical bulk temperature difference. For instance, the EE value proves to be greater for T_{bf} and T_{bp} at 60°C & 50°C (68.07%), compared to 40°C & 30°C (48.94%) and 30°C & 20°C (37.69%). This pattern is consistent throughout all similar comparisons. This evidence suggests that T_{bf} has a more pronounced effect than T_{bp} and even exceeds the influence of the overall temperature disparity between the channels. Such behavior is attributed to the fact that a rise in T_{bf} results in a disproportionate increase in the vapour pressure of the feed, thereby generating a higher permeate flux (J_{mh}), despite a constant bulk temperature difference (refer equations in APPENDIX 2). This result proves to be pivotal in the next sections.

5.4.2 Impact of topping cycle pressure on water production capacity.



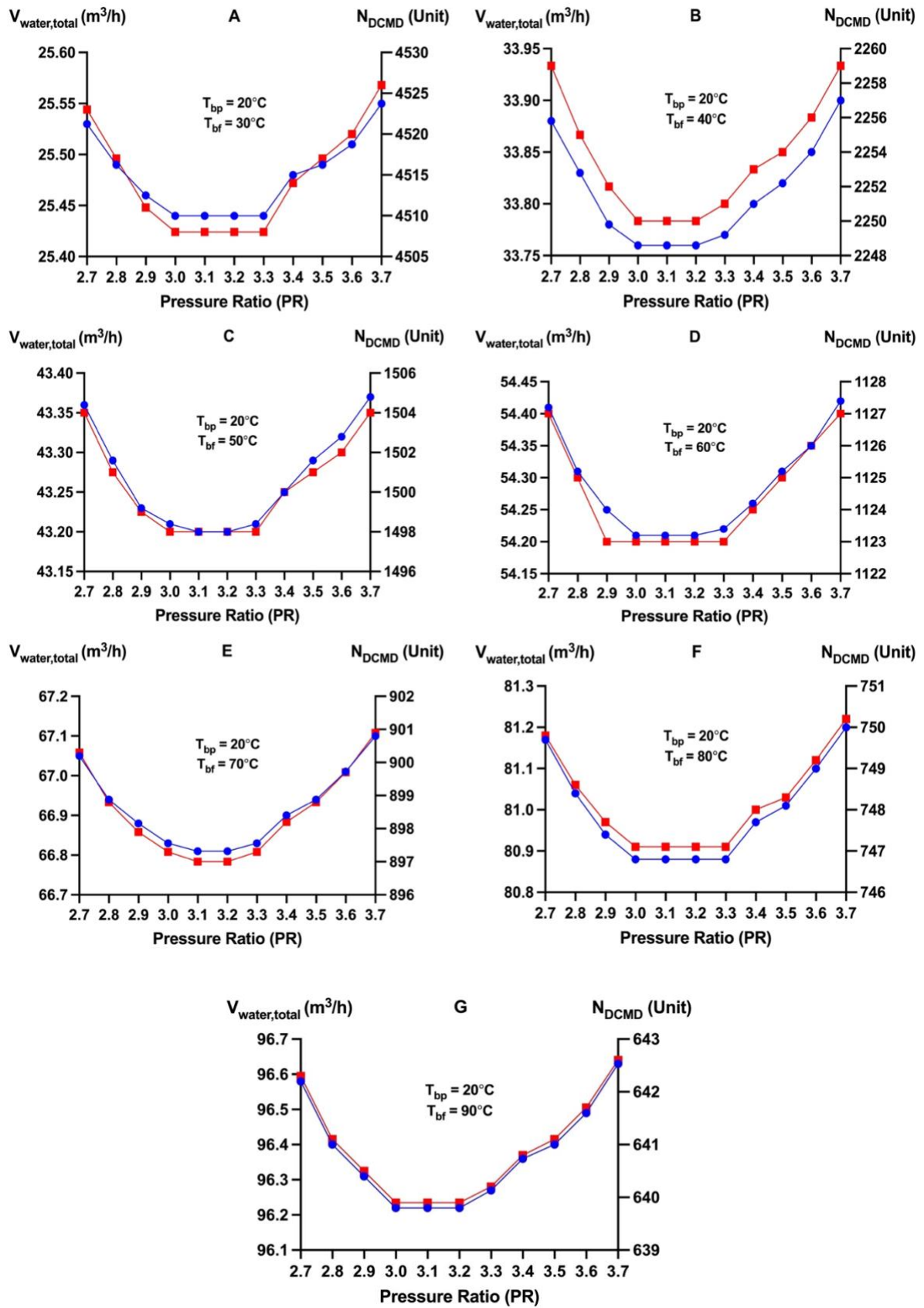


Fig. 5.4 The relationship between Pressure ratio (PR), total water production and Number of DCMD units needed at different $\Delta T_b = (T_{bf} - T_{bp})$ at reference cycle conditions.

Fig. 5.4 illustrates the correlation between various PR values and ΔT_b , and their subsequent impact on water production ($V_{water,total}$) and desalination units needed (N_{DCMD}). It is unequivocal that for all ranges of ΔT_b , a minimum point emerges as PR increases, specifically situated between the PR values of 3.0 and 3.3. This observation is explicable through the principles shown in section 3.1, where thermodynamic performance escalated with PR until reaching an apex, before subsequently declining. This rise in performance signifies a more efficient heat utilisation in the sCO_2 cycle, an effect mirrored inversely in Fig. 5.1. As the cycle consumes more heat, less becomes available to DCMD, curtailing its maximum water production capacity, and vice versa which is quantified by the minimum water production of 25.44 m³/h at pressure ratio of 3.0-3.3 at ΔT_b of 10°C compared to 25.53 m³/h at pressure ratio of 2.7. Additionally, it was discovered that an increase in total water production corresponds with higher ΔT_b , while the opposite result is displayed by units needed, with the maximum and minimum water produced and units needed at bulk feed temperature of 90°C of 96.63 m³/h and 640 respectively. The former can be attributed to the greater permeate flux (refer equation A2-1) values accompanying an increase in ΔT_b , thereby augmenting the system's evaporation efficiency. Conversely, the latter phenomenon occurs as ΔT_b increases; the cooler absorbs an identical amount of heat ejected by the cycle but at a lower mass flow rate. This, along with the reduction of STEC (as mentioned in section 5.4.1, Fig. 5.3), reduces the number of DCMD units required.

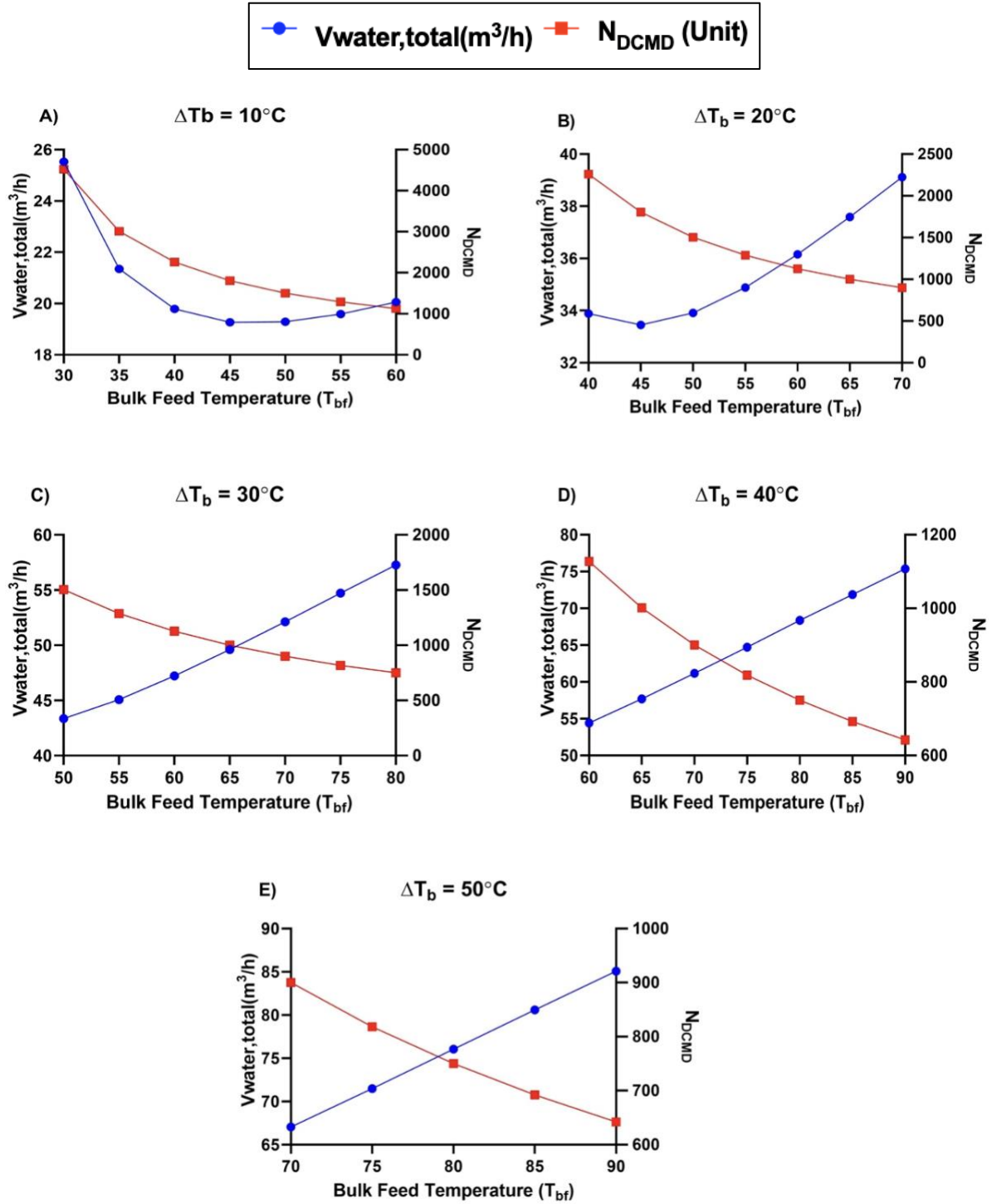


Fig. 5.5 The relationship between $\Delta T_b = T_{bf} - T_{bp}$ and the total water production ($V_{water,total}$) and total of DCMD units needed (N_{DCMD}) at reference cycle condition.

Fig. 5.5 details the relationship of ΔT_b at various bulk feed temperature (T_{bf}) with total water production ($V_{water,total}$) and units needed (N_{DCMD}). The range of ΔT_b combinations

was confined to the minimum and maximum of temperatures for T_{bf} and T_{bp} of 30°C and 50°C respectively, for the reasons mentioned in section 5.2. The maximum ΔT_b is 50°C with the rest are outside the scope of this section. Firstly, the findings in Fig. 5.5 affirm those of Fig. 5.3 showing that a higher temperature difference (ΔT_b) results in increased water output and fewer units required. Besides that, At a ΔT_b of 10°C and 20°C, water production first declines then increases, which differs at larger ΔT_b values. At lower bulk temperature differences (smaller ΔT_b), the permeate flux is low, as indicated by Equation A2-4. Additionally, the reduced water flow rate in the cooler, due to a larger temperature difference between the entering and exiting water, necessitates fewer units. The combination results in decreased water production. However, when the bulk temperature difference is at higher levels ($\geq 30^\circ\text{C}$), the permeate flux increases disproportionately, leading to an increase in water production with increasing feed temperature despite lower mass flow rate requirement. Furthermore, at higher ΔT_b , total water production consistently grows despite a decrease in DCMD units needed, proving there is dual advantages with higher ΔT_b usage. This underscores the fact that at low ΔT_b , increasing the feed temperature is not as beneficial as higher ΔT_b values.

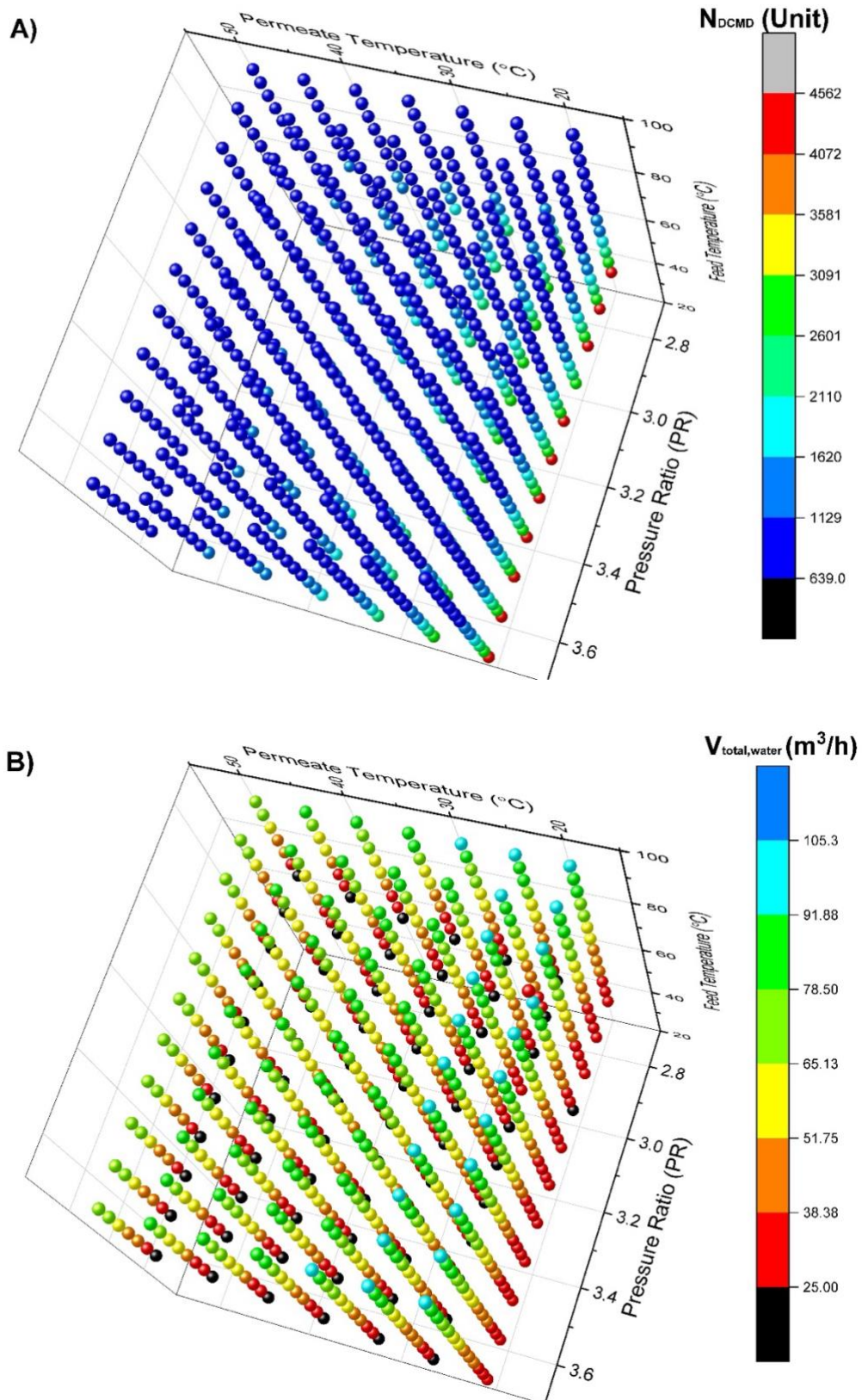


Fig. 5.6 A) The Number of units needed for DCMD (N_{DCMD}) and B) the total water produced by the system ($V_{total,water}$) at various permutations of Bulk feed temperature (T_{bf}) and Bulk Permeate temperature (T_{bp}) and pressure ratio (PR) at reference condition.

Fig. 5.6 exhibits various permutations of T_{bf} , T_{bp} and PR on N_{DCMD} and $V_{total,water}$. Both Fig. 5.6 A and B illustrate the full extent of the examined research conditions of what was discussed in section 5.2. It is worth mentioning that the prior results are consistent across all condition combinations. In addition, Fig. 5.6 A showed that the combination of ΔT_b with the lowest N_{DCMD} value is the highest T_{bf} and lowest value of T_{bp} with 90°C and 20°C respectively and pressure ratio of 3.1-3.3 with 640 units. Conversely, the highest N_{DCMD} required is the lowest value of both T_{bf} and T_{bp} with 30°C and 20°C respectively and PR=2.7 with 4523 units. Meanwhile, for total water production, the maximum of 96.58 m³/h occurs at T_{bf} and T_{bp} of 90°C and 20°C respectively, at a PR of 2.7. The minimum of 19.2 m³/h was observed at T_{bf} of 45°C and T_{bp} of 35°C, at PRs of 3, 3.1, and 3.2. This reiterates the results found in Fig. 5.3, whereby the bulk feed temperature is significantly influential for performance across all combinations of conditions. Besides that, there is a minimum point for total water production with increasing PR across all combinations for the reason that was detailed in section 5.5.1 and Fig. 5.4.

5.5 Analysis on the power and water combined performance

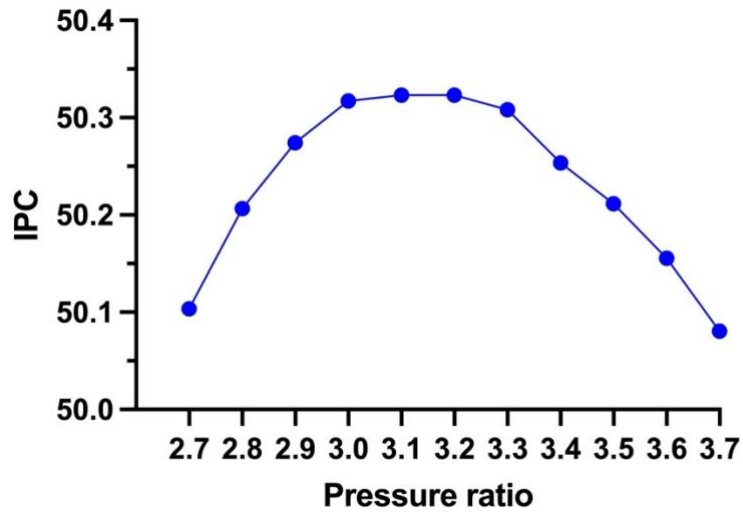


Fig. 5.7 Integrated performance coefficient (IPC) across different pressure ratios (PR).

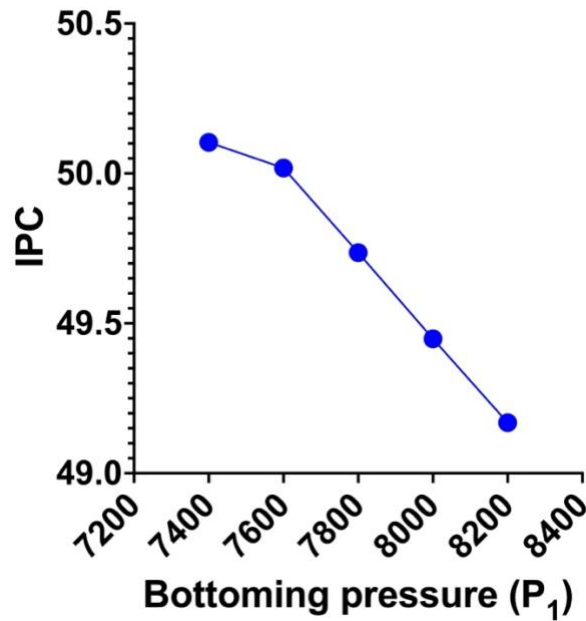


Fig. 5.8 Integrated performance coefficient (IPC) across different bottoming pressures (P_1).

In this section, we combined two outputs, power (kW) and water (m^3/h), each normalized by W/m^2 to create what we term the Integrated Performance Coefficient (IPC). We then summed these normalized values using a fixed DNI of $1000 \text{ W}/\text{m}^2$ as a comparative metric. Although not

mathematically unitless, this normalization process effectively renders these metrics dimensionless for analytical purposes. By scaling both power (kW) and water (m³/h) outputs by the same measure of solar energy density (W/m²), each output is transformed into a relative measure that directly compares how much of each output is produced per unit of solar input. This transformation allows us to abstract away from the specific units of power and water, focusing instead on the proportionality and comparative productivity of each within the same integrated system relative to the energy consumed. This method provides a coherent framework for evaluating the performance of different systems under varied conditions, focusing on the relative conversion of solar energy input into useful outputs. The bulk feed and permeate temperatures difference is fixed to 90°C and 20°C because, as it has been proven earlier, this produces the most water. This coefficient enables the examination of peak and minimum performance across various pressure ratios, given the inverse relationship between power generation and water production as outlined in earlier sections. Fig. 5.7 and Fig. 5.8 show the combined performances resemble Fig. 5.1 and Fig. 5.2. This is logical since the system prioritizes power output, utilizing waste heat to produce water. It demonstrates that power generation is a more influential output of the system, with the highest and lowest values being 50.32 at pressure ratios of 3.1 and 3.2, respectively, and 50.08 at a pressure ratio of 3.7. However, it is worth noting that the choice must be based on current advancement and cost of material in PTSC's receiver tube and power cycle, nonetheless performances between different pressure ratios are not significantly large and lower pressure ratio of 2.7 may lead to better choice for current design of PTSC tube. Fig. 5.8 confirms the above observation, showing that power is the preeminent index. There is a significant reduction of IPC when bottoming pressure exceeds 7600 kPa, further indicating that a reduction in bottoming pressure is unnecessary due to poor performance.

5.6 Impact of DNI on water production

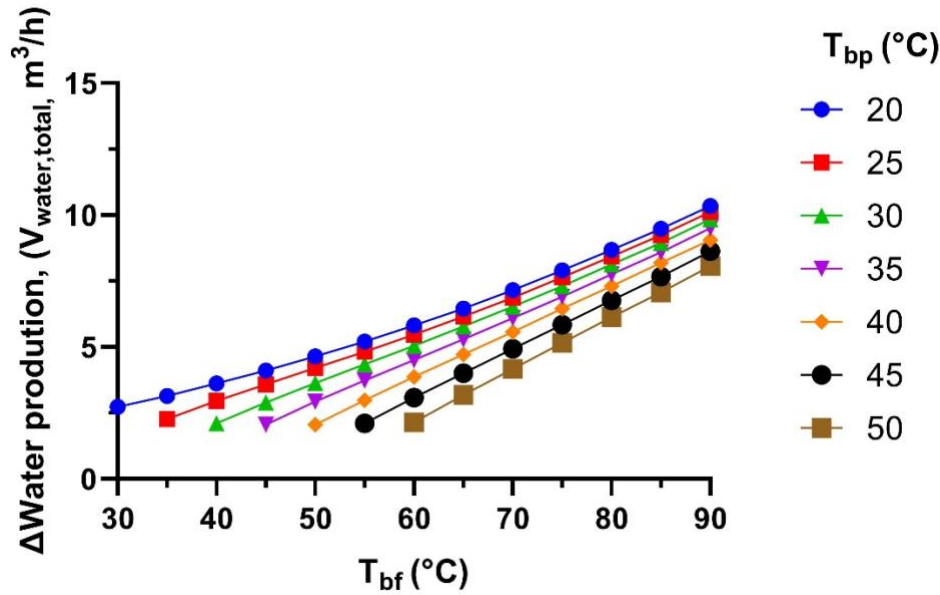


Fig. 5.9 Water production reduction in response to every $100\text{W}/\text{m}^2$ decrease in DNI for various combinations of feed (T_{bf}) and permeate (T_{bp}) temperatures.

Fig. 5.9 illustrates the water reduction decline for each $100\text{ W}/\text{m}^2$ drop across different combinations of feed and permeate bulk temperatures. For example, at a feed temperature of 30°C and a permeate temperature of 20°C , water production fell by about $2.73\text{ m}^3/\text{h}$ for every $100\text{ W}/\text{m}^2$ reduction in DNI. This trend persists across all DNI levels, making it crucial to examine the effects of varying feed and permeate temperature combinations. Notably, a combination of higher feed temperatures and lower permeate temperatures, which typically yields higher water output, is more sensitive to DNI changes. For example, the feed and bulk temperature of 90°C and 20°C leads to a reduction of around $10\text{m}^3/\text{h}$ for each $100\text{ W}/\text{m}^2$ drop. Conversely, lower feed temperatures with the same permeate temperature are less affected by DNI fluctuations with feed temperature of 30°C only reduced by around $2.55\text{m}^3/\text{h}$. Moreover, the impact of DNI is less pronounced at higher permeate temperature as the water production reduction decreases from around $10\text{m}^3/\text{h}$ to around $7.5\text{m}^3/\text{h}$ when permeate temperature

increases from 20°C to 50°C at feed temperature of 90°C , and the same observation can be seen throughout all combinations.

5.7 Temperature polarisation analysis

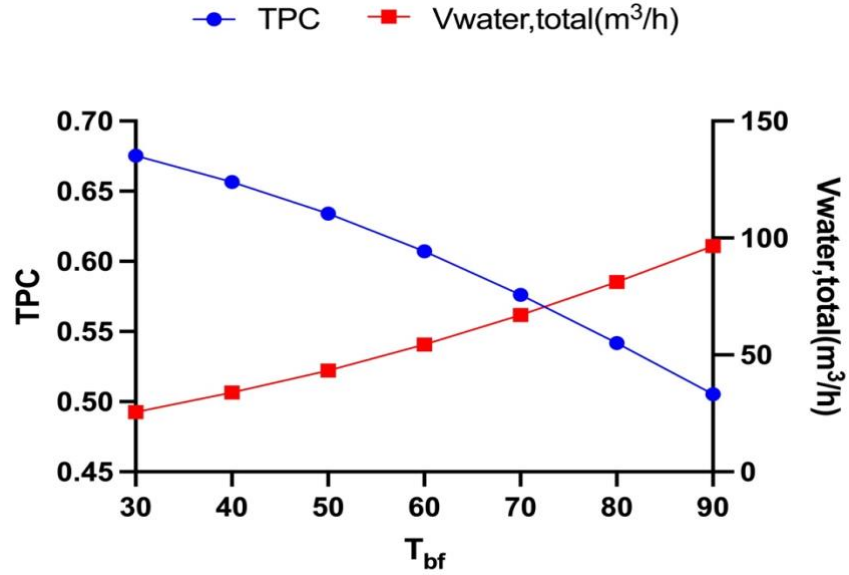


Fig. 5.10 Relationship between bulk feed temperature (T_{bf}) and Temperature polarisation coefficient and total water production ($V_{water,total}$) with $T_{bp} = 20^\circ\text{C}$

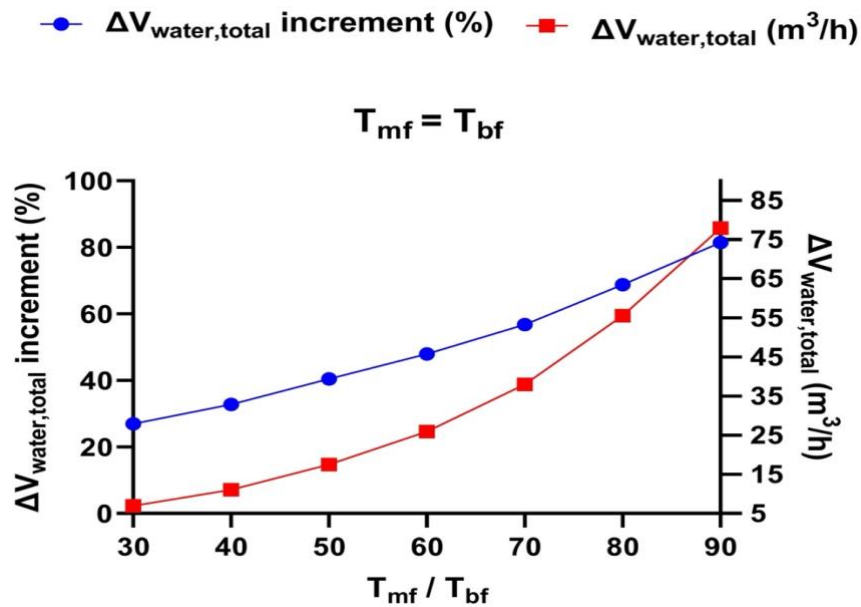


Fig. 5.11 The percentage improvement of total water production, V_{water} (right axis) and the absolute value improvement difference, $\Delta V_{water,total}$ when membrane feed temperature is improved to unity ($T_{mf} = T_{bf}$).

Fig. 5.10 illustrates the impact of bulk feed temperature increment leads to higher acceleration of water production but lower temperature polarisation coefficient. It is clear that higher feed temperature leads to not only increase performance but higher performance potential. Fig. 5.11 examines the change of water production ($\Delta V_{water,total}$) improvement in percentage and absolute terms when feed membrane temperature (T_{mf}) aligns with bulk feed temperature. Evidently, a higher bulk feed temperature enhances the potential for both indices' increment. An increment in total water production of 6.92 and 81.41 m^3/h for the lowest and highest respective feed temperature was observed. Given the understanding that higher feed temperature results in lower DCMD units needed and increased water production. Significant potential exists for simultaneously boosting water production through temperature polarisation improvement and reduction of cost trough less unit needed for this system.

5.8 Conclusion

This chapter evaluated a direct PTSC–sCO₂ Brayton cycle with DCMD to understand how solar input and cycle settings affect combined power performance and water output. The findings showed that power-related efficiency drops steeply with declining solar intensity, while a pressure-ratio region around 3.1–3.2 delivered the best usable-energy efficiency of 38.3%. On the desalination side, feed and permeate temperatures controlled water production strongly, reaching up to 96.6 m^3/h under favourable thermal conditions, but with increasing sensitivity at higher feed temperatures. Overall, the chapter translated these patterns into operating guidance by identifying which conditions maintain strong combined performance and where the system becomes most vulnerable to solar variability. Because these results show that additional heat use and fluid handling could improve both output stability and water performance, the next chapter incorporates a bottoming ORC to investigate how working-fluid choice and configuration can shift performance and loss distribution.

CHAPTER 6 DETAILED THERMODYNAMIC AND ECONOMIC ANALYSIS AND ORGANIC FLUID SELECTION.

This chapter presents the development and analysis of a novel solar-driven cogeneration system that integrates a parabolic trough solar collector (PTSC) with a recompression supercritical carbon dioxide (sCO₂) Brayton cycle, a bottoming Organic Rankine Cycle (ORC), and a Direct Contact Membrane Distillation (DCMD) desalination process. The system is designed to simultaneously generate electricity and produce freshwater by efficiently coupling advanced power cycles with innovative waste heat recovery.

A comprehensive mathematical framework is developed to evaluate the system's thermodynamic and techno-economic performance under varying conditions. This includes a duality method, introduced here for the first time, that considers both the pinch point and approach temperature in the optimisation process without wetting the working fluid. By applying this approach across a range of organic fluids in the ORC, the study provides new insights into how fluid selection influences cycle efficiency, water yield, and economic viability.

The research advances the realisation of third-generation concentrated solar power (CSP) systems by combining high-efficiency thermodynamic cycles with integrated desalination. It also identifies key sources of exergy destruction, quantifies system-level improvements, and introduces comparative cost analysis, thereby bridging the gap between conceptual designs and practical implementation

6.1 Research conditions

The parabolic trough plant was designed for a rated capacity of $50 MW_{net}$ at a DNI of $1000 W/m^2$, featuring PTSCs with an inlet temperature of 650K (T_5), and a mass flow rate of 2.5kg/s per row, which was proven to be the optimal combinations under various conditions as per CHAPTER 4. A slightly modified LS-3, whose tube is 10mm thicker than the original value was used as the unit in order to withstand the high pressure of the supercritical fluid. (refer Zaharil & Yang., 2023 for full specifications of the unit). The optimization process began by determining the cycle's mass flow rate with the best cycle efficiency, adhering to approach temperatures of at least 10K for HTR and 5K for LTR for sufficient management of heat load (Linares et al., 2020), and $50 MW_{net}$. The mass flow rate optimization is not only important for performance, but due to the fixed value of T_5 , mass flow rate adjustment is necessary in order to avoid the temperature of T_4 to below T_5 , which is thermodynamically impossible. The collector rows were calculated by dividing the optimized mass flow rate by 2.5kg/s. To be exact, the flow rate of 2.50008kg/s per row was used to ensure proper mass balance. The length of the collector units was established post-optimization based on the T_3 determined during cycle's mass flow rate optimization step. However, the inaccuracy of the 1D model for long-length parabolic troughs, as evidenced by its inability to account for axial effects and nonlinear temperature gradients within the Heat Transfer Fluid beyond an HCE length of approximately 80 meters must be highlighted (Forristall, 2003 as taken from Yilmaz & Mwesigye. 2018). To overcome this issue, an innovative solution is proposed by using iterative calculations, performed for each unit to achieve the desired temperature, with each unit length of 4.06m the length of the final unit set at 3.15m to reach the intended output temperature (refer Fig. 3.1 for the full flowchart). The iterative method involves using the outlet values of a PTSC unit as the inlet values of the inlet of the next unit.

For the bottoming cycle, ORC cycle conditions were assumed to condense at 35°C under reference condition and the turbine inlet is at saturated vapour conditions (Singh et al., 2018). The mass flow rate was optimised by using a combination of pinch point (Fig. 3.5), approach temperature (duality method) and energy balance method. The pinch point allows for an energy balance equation of $\dot{m}_{scO2}(h_6 - h_{pp}) = \dot{m}_{ORC}(h_{21} - h_{pp2})$ and the approach temperature ($T_6 - T_{21}$) was varied from 6K and above for each fluid and the approach temperature with the highest cycle efficiency was chosen. After the optimisation of the flow rate, the temperature of T_{1a} can easily be calculated by using the energy balance of the whole heat exchange which is $\dot{m}_{scO2}(h_6 - h_{1a}) = \dot{m}_{ORC}(h_{21} - h_{22})$. Please refer to Fig. 3.10 for full calculation of the optimisation flow of bottoming cycle and Fig. 3.11 for heat exchanger illustration.

6.1.1 Duality method

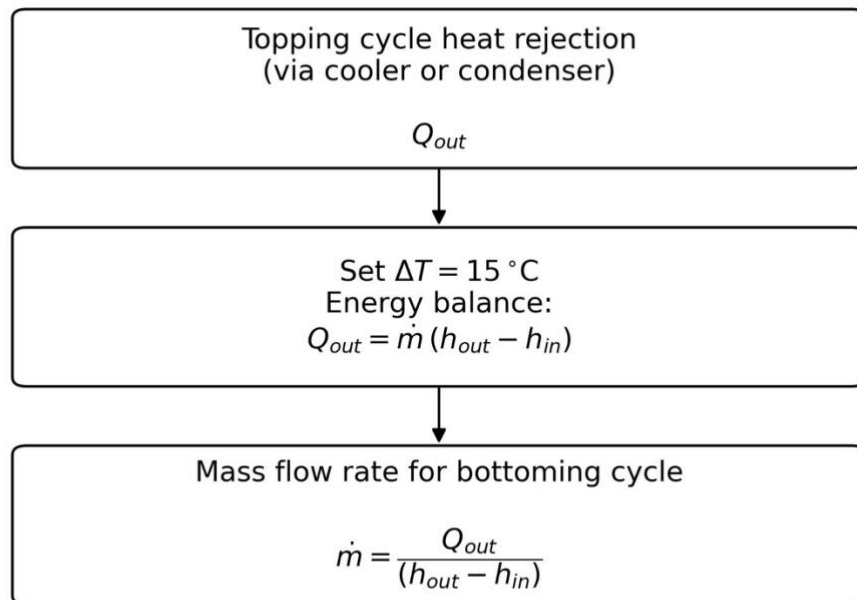


Fig. 6.1 Typical energy balance method between topping to bottoming cycle

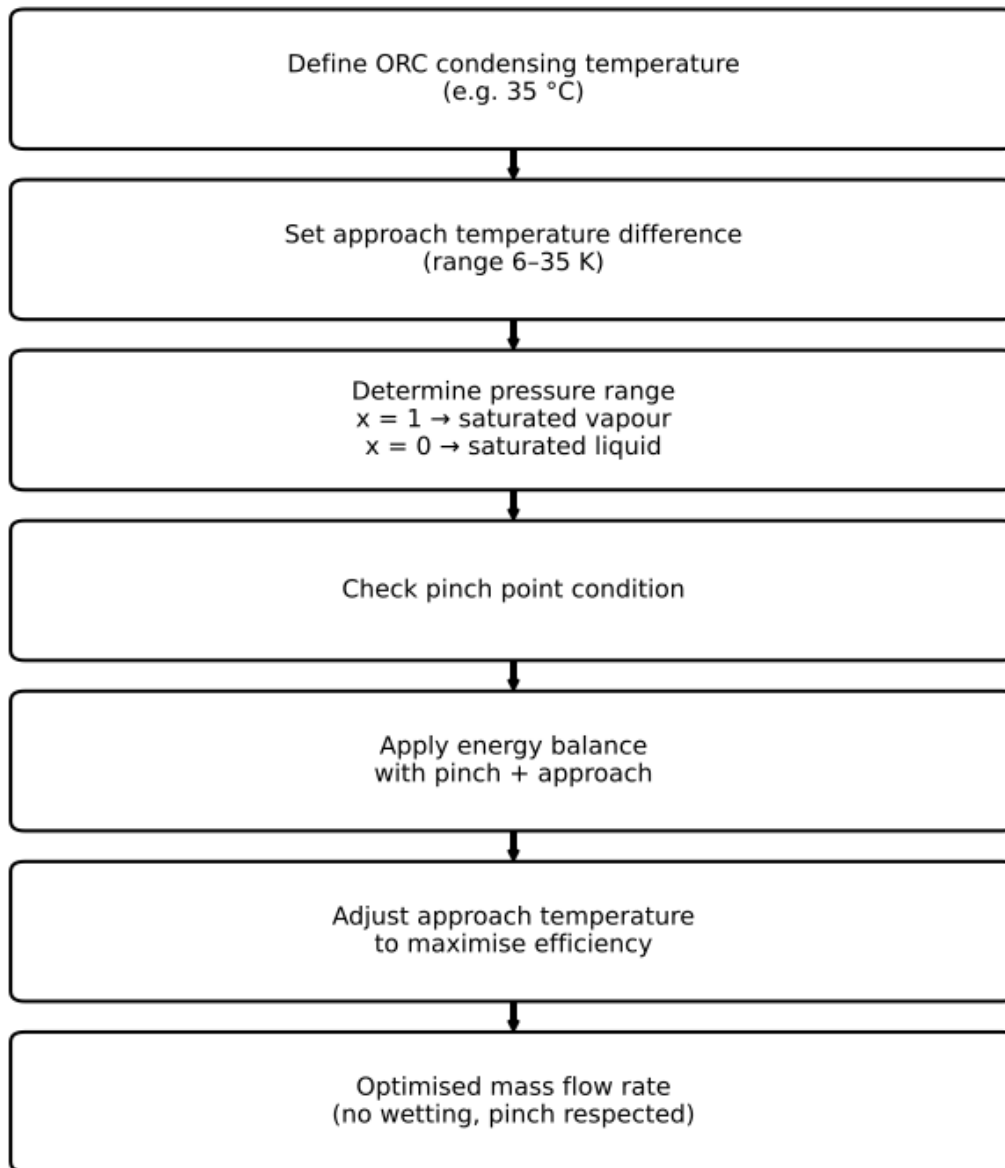


Fig. 6.2 The flow-chart of the proposed duality method in simplified form, considering pinch point and approach temperature precisely without wetting the working fluid after expansion

Fig. 6.1 and Fig. 6.2 show the simplified flow-chart of the typical energy balance method and the proposed ‘duality’ method. The conventional energy balance method provides a quick estimation of the bottoming cycle mass flow rate by equating the heat rejected from the topping

cycle to the heat absorbed by the working fluid. However, this approach does not account for the wetting that may occur after expansion, as it does not explicitly reference the saturation conditions of the fluid. Instead, it often results in an indiscriminate degree of superheating at the turbine outlet. Previous studies such as (Mago et al. 2007; Aghahosseini & Dincer. 2013; Yamamoto et al. 2001) have shown that such excess superheating does not yield additional exergetic benefits and thus represents an inefficient design choice. Consequently, while the energy balance method is useful for initial sizing, it lacks the thermodynamic rigour and optimisation capability offered by more advanced approaches such as the proposed duality method.

The duality method (Fig. 6.2) offers a systematic improvement over the conventional energy balance approach for determining the mass flow rate in combined cycles. By incorporating both the approach temperature and the pinch point constraint, referenced explicitly to the saturation states ($x=1$ for saturated vapour and $x=0$ for saturated liquid), this method ensures that the thermodynamic boundary conditions are respected throughout the cycle. Since the working fluid is selected to be isentropic or dry, the expansion process remains on the vapour side of the saturation dome, thereby avoiding wetting. Furthermore, by systematically varying the approach temperature, the method allows for the identification of the best performance metric, whether thermal efficiency, net power output, or levelised cost of energy (LCOE), thus providing a more realistic and reliable optimisation framework than a simple energy balance. The step-by-step detailed methodology is shown in Fig. 3.11.

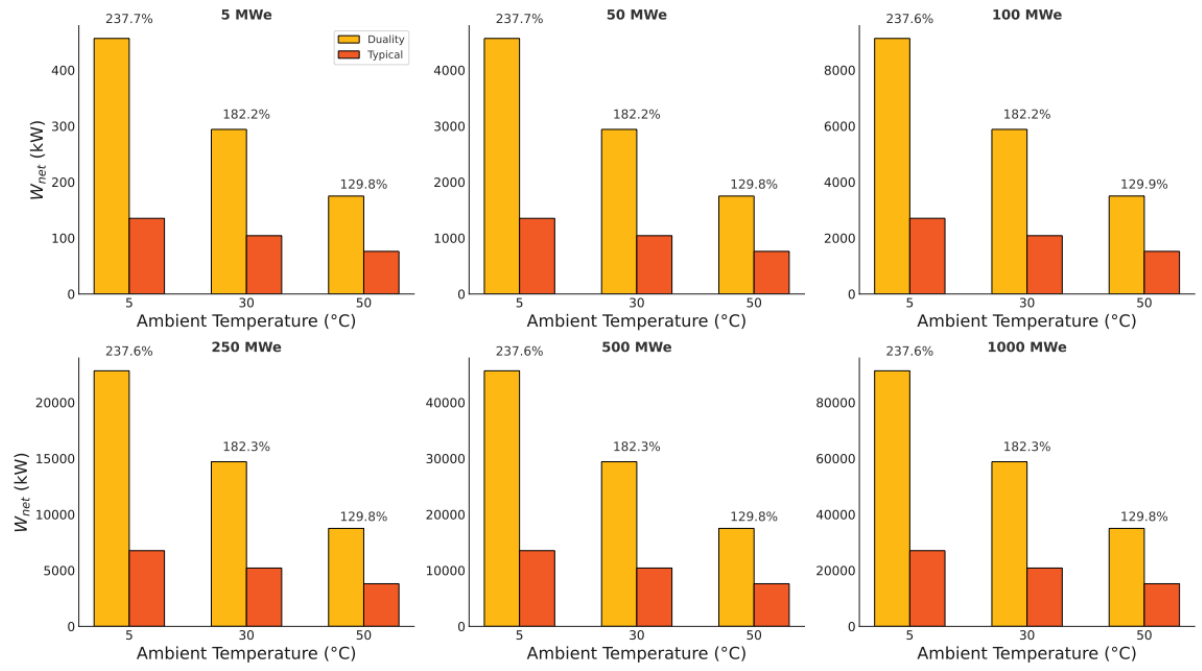


Fig. 6.3 W_{net} difference between the typical and proposed methods

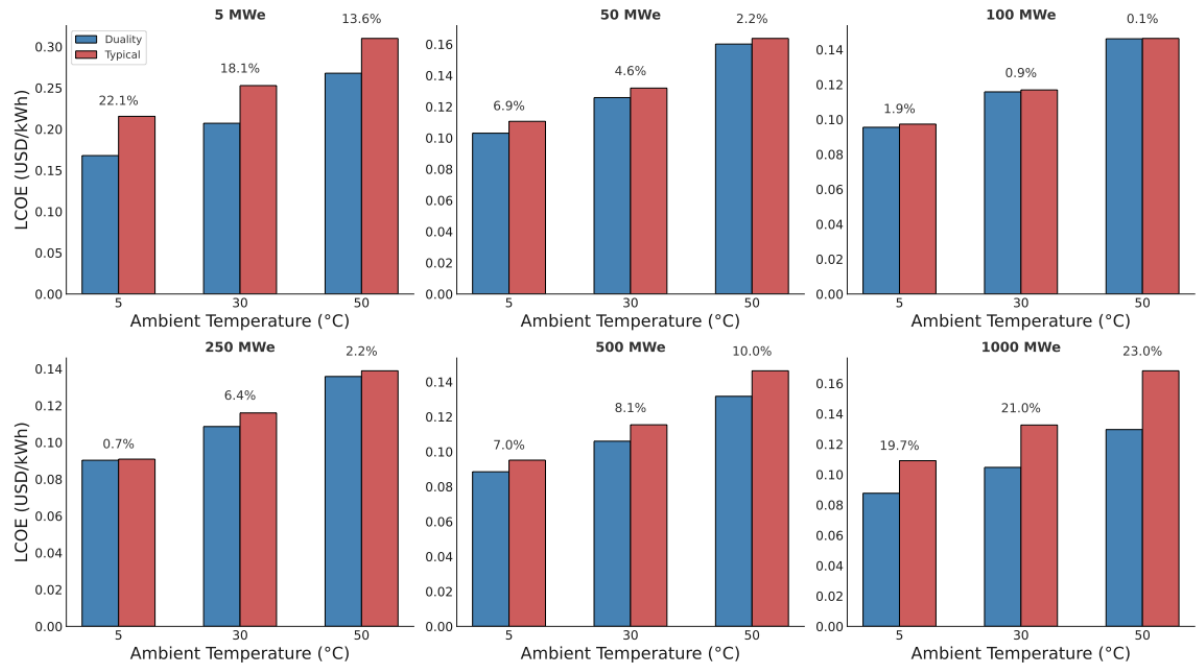


Fig. 6.4 LCOE difference between the typical and proposed methods

Fig. 6.3 and Fig. 6.4 illustrate the performance differences between the typical energy balance method and the proposed duality method in terms of net power output (W_{net}) and

levelised cost of energy (LCOE) at various ambient temperature. For W_{net} , the duality method consistently outperforms the typical approach, with improvements ranging from about 130% to 238% across scales (5–1000 MWe) and ambient temperatures. In contrast, the LCOE results show reductions of approximately 2% to 23%, depending on the system size, with the greatest benefits observed at the lower and higher ends of the power outputs. These results highlight that the duality method not only enhances net power recovery but also offers significant thermodynamic and economic advantages compared to the traditional energy balance method.

6.1.2 Organic fluids selection in ORC

The fluids selected were based on the previous studies that concluded that for low-temperature source, pure fluids performed better than zeotropic/mixture fluids (Chowdhury & Ehsan., 2023). Additionally, environmental considerations, such as Global Warming Potential (GWP), Ozone Depletion Potential (ODP), and biodegradability, are increasingly pivotal. It has also been established that organic fluids exhibiting isentropic behaviour and dryness are preferable over those undergoing two-phase expansion. Furthermore, it has been demonstrated that superheating can diminish exergetic efficiency. Consequently, the organic fluids employed in this research are characterized by their biodegradability, dry/isentropic nature, and low GWP and ODP (IPCC., 2018). Please refer Table 6.2 for the full quantitative values. Importantly, the fluids selected for this study are those with a critical temperature exceeding 55°C, ensuring that condensation at 55°C is possible, due to the parametric analysis of up to 50°C. Table 6.1 depicts the fluids chosen as some of its important thermophysical properties. Furthermore, the fluids chosen for this research have been utilised in practical applications, demonstrating their feasibility and practical relevance, and ensuring the study is not purely theoretical.

Table 6.1 Chosen organic fluids and some of its thermophysical properties and temperatures at selected points of the ORC at reference condition.

Fluids	T_{crit} (K)	P_{crit} (kPa)	P_{max} (kPa)	P_{min} (kPa)	T_{21} (K)	T_{23} (K)	h_{21} (kJ/kg)	h_{23} (kJ/kg)	s_{21} (kJ/kg-K)	s_{23} (kJ/kg-K)
Toluene	591.8	4126	12.68	6.249	323.8	308.6	284.504	266.500	0.908	0.915
n-octane	569.3	2497	6.893	3.186	323.8	314.4	407.626	391.615	1.267	1.272
n-nonane	594.5	2281	2.527	1.073	323.8	314.5	96.229	80.350	0.400	0.405
n-heptane	540.1	2727	19.39	9.834	323.8	314.2	407.726	391.789	1.267	1.273
Neopentane	433.7	3196	362	233.3	323.8	313.9	377.279	364.084	1.184	1.189
n-decane	617.7	2103	0.9167	0.3551	323.8	314.6	32.345	16.471	0.253	0.259
Isopentane	460.4	3370	209.2	128.8	323.8	313.2	33.780	18.793	-0.498	-0.492
Isobutene	418.1	4010	620.1	408.2	323.8	314.2	307.082	291.942	0.950	0.955
R600a	407.8	3640	730.1	464.5	325.8	310.6	467.074	449.731	1.475	1.481
n-hexane	507.9	3058	54.84	30.59	323.8	311.4	622.293	608.082	2.334	2.339
Cyclopentane	511.7	4571	106	61.84	323.8	313.9	403.238	387.508	1.253	1.259
Cyclohexane	553.6	4081	38.54	20.08	324.8	308.9	390.937	373.330	1.207	1.214
n-Butane	425.1	3796	505.1	329	323.8	310.7	314.441	297.386	0.980	0.986
o-xylene	630.3	3738	3.524	1.579	323.8	311.6	656.438	640.809	2.440	2.445
p-xylene	616.2	3532	4.484	2.047	323.8	309.9	207.137	189.316	0.713	0.719
m-xylene	616.9	3535	4.16	1.933	323.8	309.9	214.192	196.789	0.724	0.730

Table 6.2 Environmental value of the examined organic fluids. (IPCC, 2007; IPCC, 2014; Baha & Dunpont, 2023; Abdelaziz, 2023; 2024; German Environmental Agency; 2017)

Fluid	ODP	GWP
Toluene	0	Low - Approximately 3
Neopentane	0	Low - Approximately 3
Isobutene	0	Low - Approximately 3
Cyclohexane	0	Low - Approximately 3
n-Octane	0	Negligible
n-Decane	0	Negligible
R600a (Isobutane)	0	Low - Approximately 3 (infrserv, 2024)
R600 (Butane)	0	Low - Approximately 4 (infrserv, 2024)
n-Nonane	0	Low - Approximately ≤ 4 (Chem Service, 2021)
Isopentane	0	Low - Approximately 5
Isohexane	0	Low - Approximately 3
n-Hexane	0	Low - Approximately 3 (The University of British Columbia, 2017)
Cyclopentane	0	Low - Approximately 5 (Sinteco, 2024)
o-Xylene	0	Negligible
m-Xylene	0	Negligible
p-Xylene	0	Negligible
n-Heptane	0	Low (Haltermann Carless, 2021)

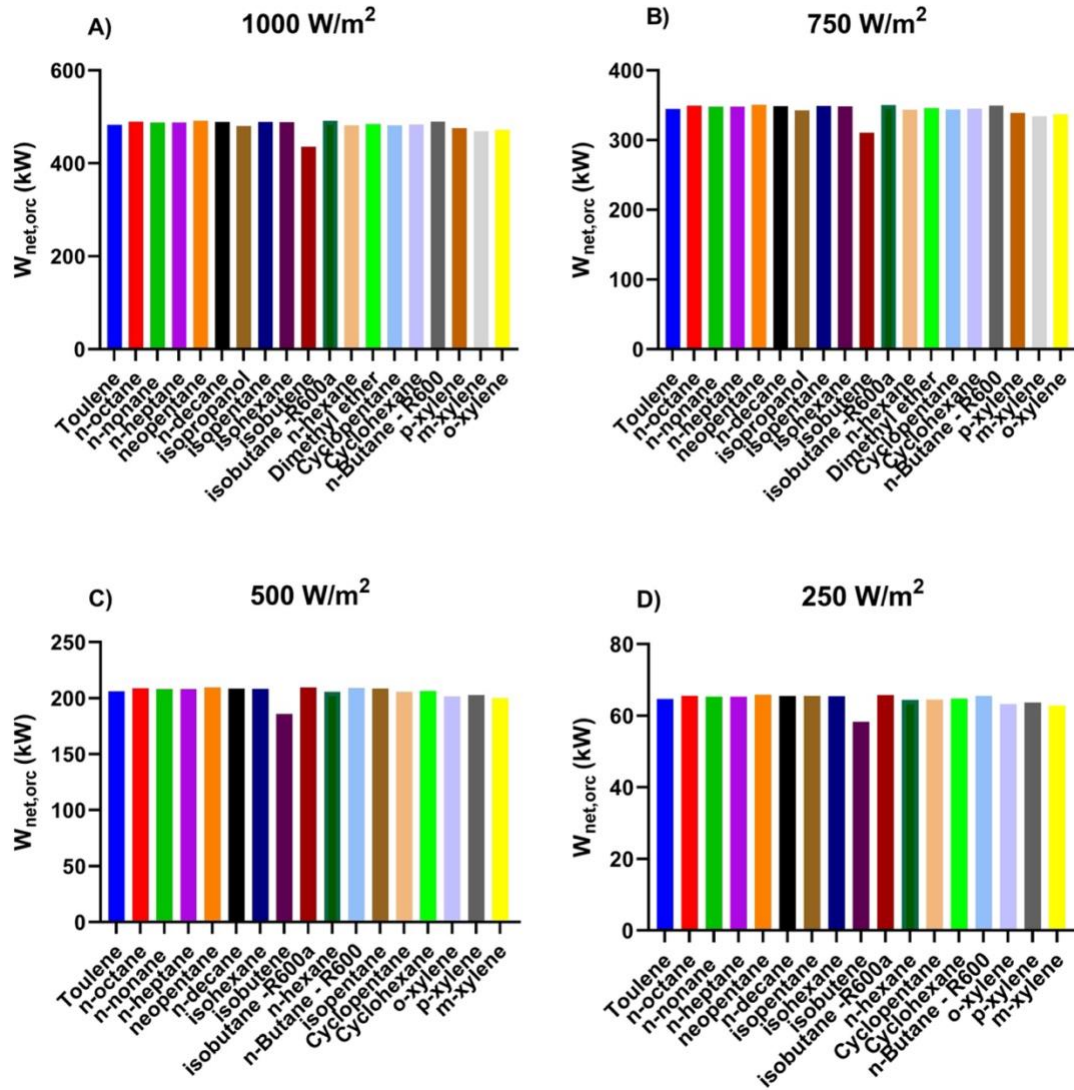


Fig. 6.5 Comparison of W_{net} for Different Fluids in Bottoming ORC

Fig. 6.5 shows the net bottom work $w_{net,ORC}$ produced by a range of organic fluids. While the disparities between these fluids are relatively marginal, it is evident that isobutene exhibited the least favourable performance. At a DNI of 1000 W/m^2 , neopentane and isobutane (R600a) demonstrated the most superior performance, generating respective 491.67 kW and 491.2 kW , closely followed by n-butane (R600) at 489.77 kW . This trend was consistent across other levels of DNI. As anticipated, a reduction in DNI corresponded with a decrease in power output. However, it is noteworthy that the combined cycle efficiency experienced an accelerated decline as DNI diminished. For instance, in the

case of Toluene, the exergetic efficiency of the combined cycle decreased by 1.75%, 3.51%, and 11.42% as DNI reduced from 1000 to 750 W/m^2 , 750 to 500 W/m^2 , and 500 to 250 W/m^2 , respectively (refer Fig. 6.13).

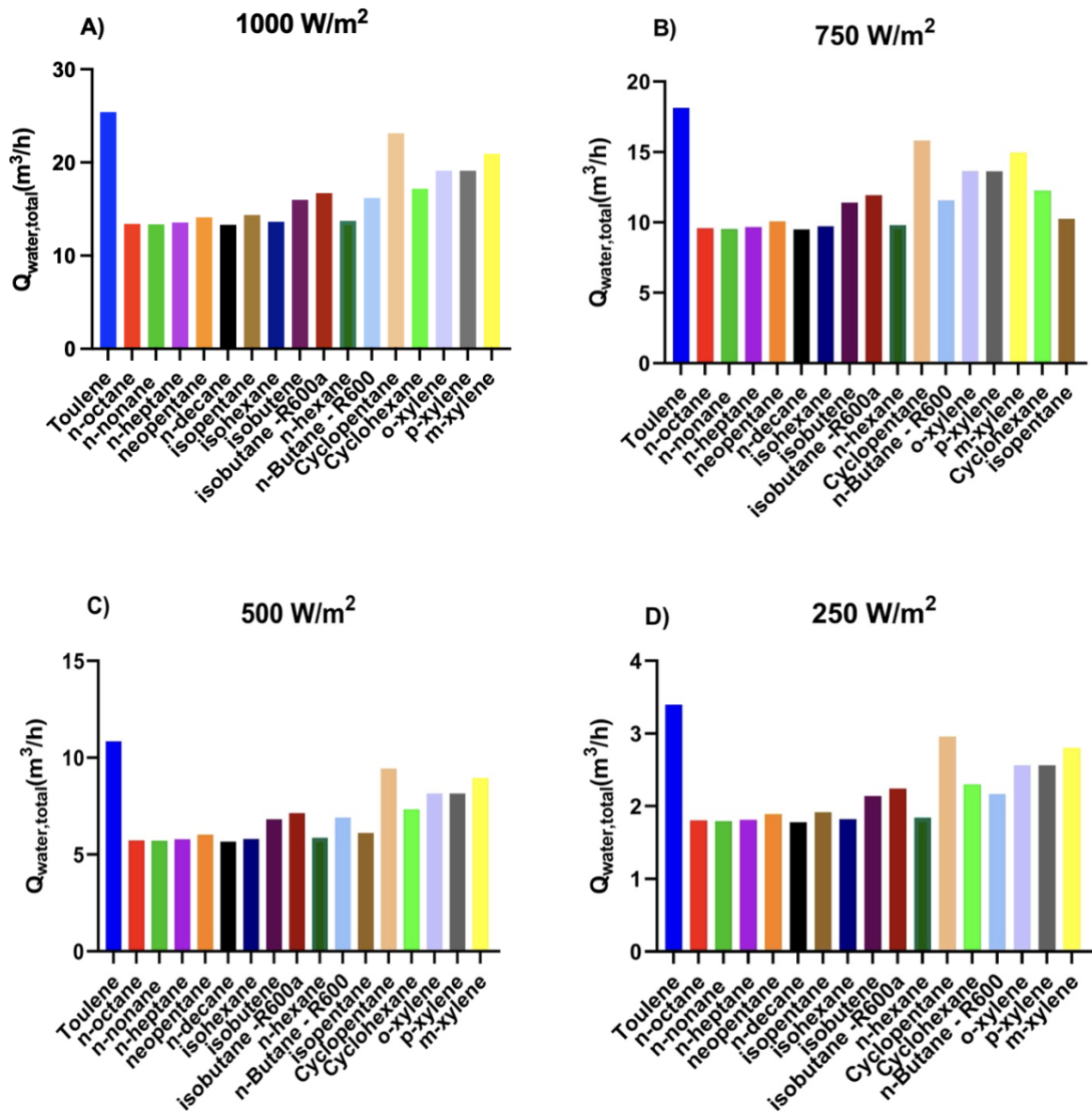
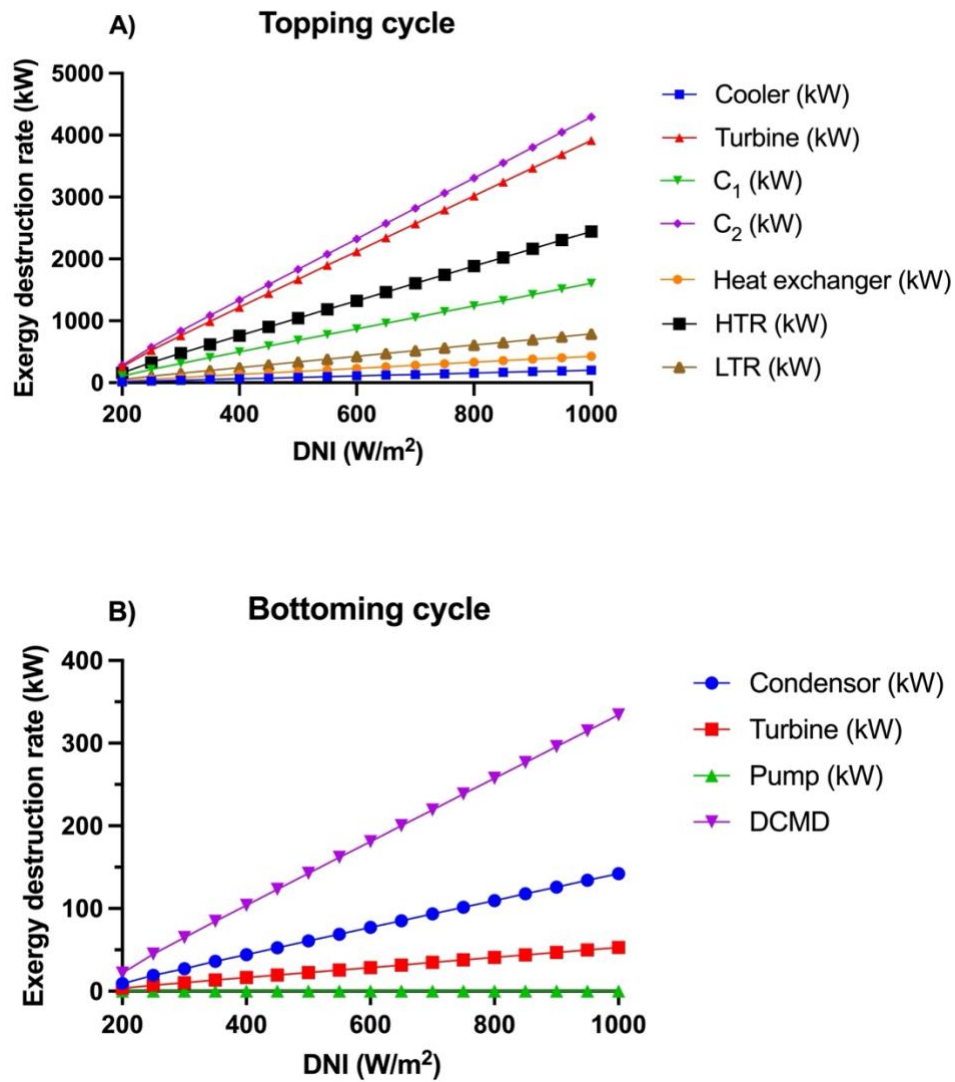


Fig. 6.6 Comparison of $Q_{water,total}$ for Different Fluids in Bottoming ORC

Fig. 6.6 illustrates the volumetric yield of water produced by the system employing various organic fluids in the Organic Rankine Cycle (ORC) under differing Direct Normal Irradiance

(DNI) levels. Notably, toluene manifests the highest water production relative to other fluids, with n-decane registering the minimum. Moreover, m-xylene and cyclopentane demonstrate commendable performance. The superior efficacy of toluene is attributed to its minimal T_{23} value, necessitating a substantially augmented mass flow rate by the condenser to cool the ORC, compared to alternative fluids. This diminished T_{23} , and consequently T_{14} (bulk feed temperature), results in a lower permeate flux, which, as established by previous studies (Soomro & Kim., 2016), is a critical determinant for Direct Contact Membrane Distillation (DCMD) efficiency. Nonetheless, the derived findings indicate that notwithstanding the reduced bulk feed temperature, the seawater's mass flow rate at the condenser emerges as a significant factor in water desalination via a CSP-DCMD system. Observations on the influence of DNI reduction on water production consistently demonstrated a decrease. Taking toluene as a reference, the research discerned a steady absolute decline in water production of approximately 7.3 m³/h in a solar-coupled DCMD desalination system. Intriguingly, while this absolute decrement remained constant, the proportional decrease escalated markedly with each measurement, 28.68%, 40.21%, and 68.63%. This divergence between absolute and relative decline rates underscores a non-linear impact on the system's efficacy concurrent with decreasing DNI.

6.2 Exergy destruction rate analysis



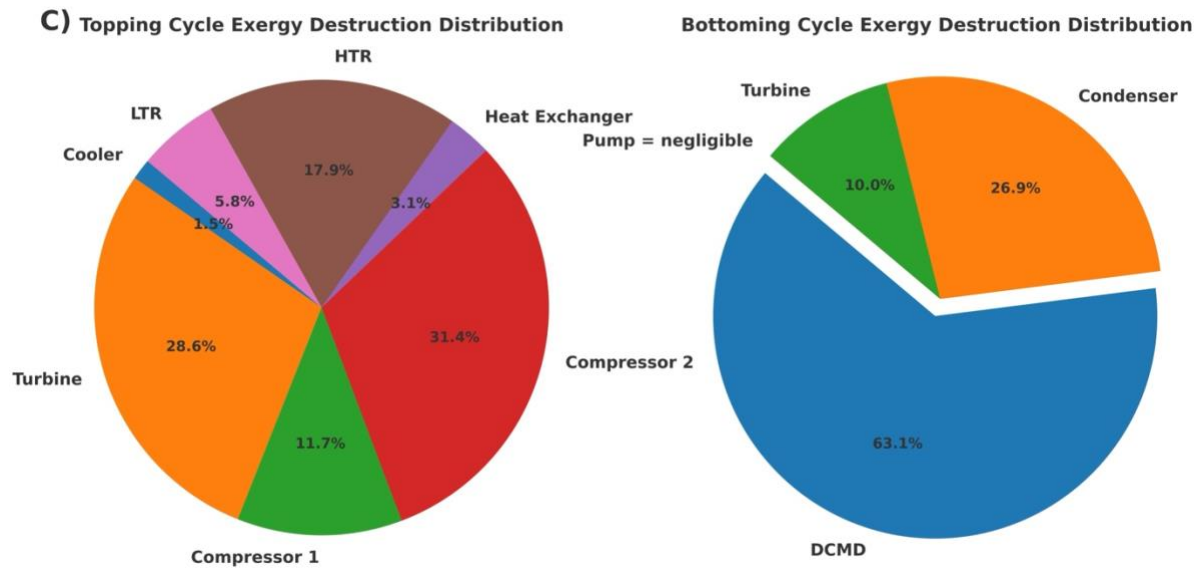


Fig. 6.7 Exergetic destruction rate at each component for A) topping cycle, B) bottoming cycle and C) the percentage of exergetic destruction of each component.

Fig. 6.7 shows the exergetic destruction rate within each component of the system's topping and bottoming cycles under varying DNIs. As established in the preceding section, Toluene exhibits optimal CPC performance and is consequently selected for further analysis in subsequent sections of this paper. Notably, the most substantial exergy losses occur in Compressor 2 and the turbine of the topping sCO_2 Brayton cycle. This phenomenon is attributable to Compressor 2's role in compressing supercritical carbon dioxide at elevated temperatures, positioning it further from the critical point and resulting in a lower density compared to Compressor 1. Consequently, this necessitates a significant input of power. Conversely, the cooler manifests the minimal exergy destruction, likely due to the mass flow rate division enacted by the recompression Brayton cycle, thereby diminishing the specific heat (C_p) mismatch between the cycle's working fluid and the seawater within the cooler.

In the bottoming ORC and DCMD segments, it's apparent that DCMD registers the highest exergy destruction. However, the exergy losses in the bottoming cycle are substantially lower compared to the topping cycle. Notably, as DNI escalates, the gradient of exergy destruction

for components with higher losses increases. For instance, Compressor 2 exhibits an average slope of 4.966, Turbine 4.53, and Cooler 0.235, indicating a more pronounced increase in exergy destruction with rising DNI, suggesting enhanced energy conservation potential in Compressor 2 compared to the Cooler. Nevertheless, the proportion of exergy destruction among components remains consistent across varying DNI levels, with values of 1.48%, 28.62%, 11.74%, 31.38%, 3.13%, 17.87%, and 5.77% for Cooler, Turbine, Compressors 1 and 2, Heat Exchanger, HTR, and LTR, respectively. This trend persists in the bottoming cycle, with exergetic percentages for DCMD, Condenser, Turbine, and Pump being 63.12%, 26.88%, 9.99%, and a negligible amount for the Pump (refer Fig. 6.7C). It is critical to contextualize these findings against other studies indicating a decrease in exergy destruction with increasing DNI (Al-Sulaiman & Atiff, 2015; Singh et al., 2018). Such studies employed an indirect system with a fixed mass flow rate for their thermodynamic cycle, unlike this research, which utilizes adaptable mass flow rates corresponding to varied DNI levels, typically utilizing lower mass flow rates at reduced DNI, leading to decreased absolute exergy destruction.

6.3 Exergy destruction of direct contact membrane distillation (DCMD)

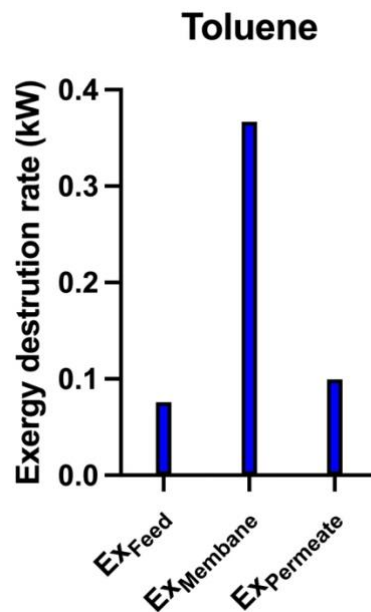


Fig. 6.8 Exergy destruction rate for a single unit of direct contact membrane distillation (DCMD) with toluene as the ORC fluid

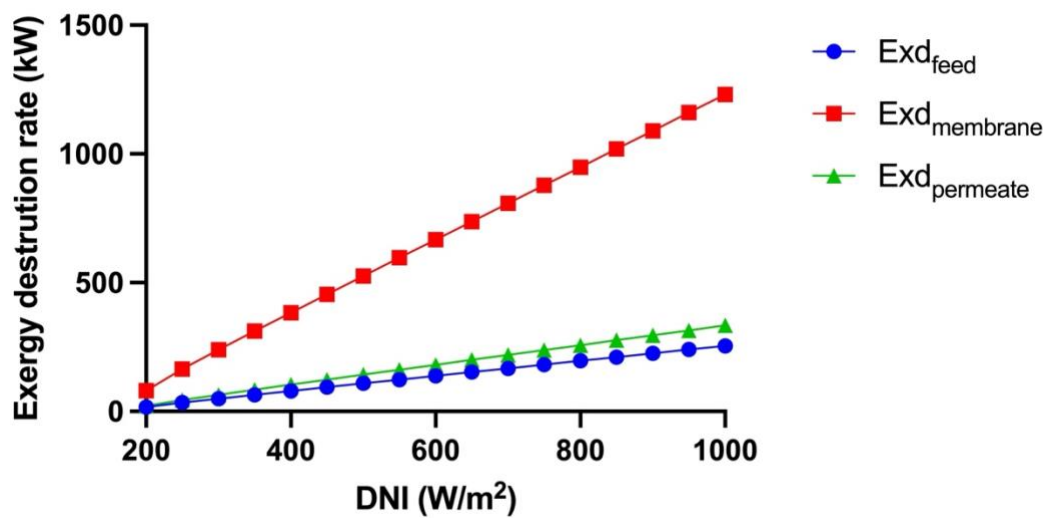


Fig. 6.9 Exergy Destruction rate across all cumulative units in Direct Contact Membrane Distillation (DCMD) with Toluene as the ORC Fluid.

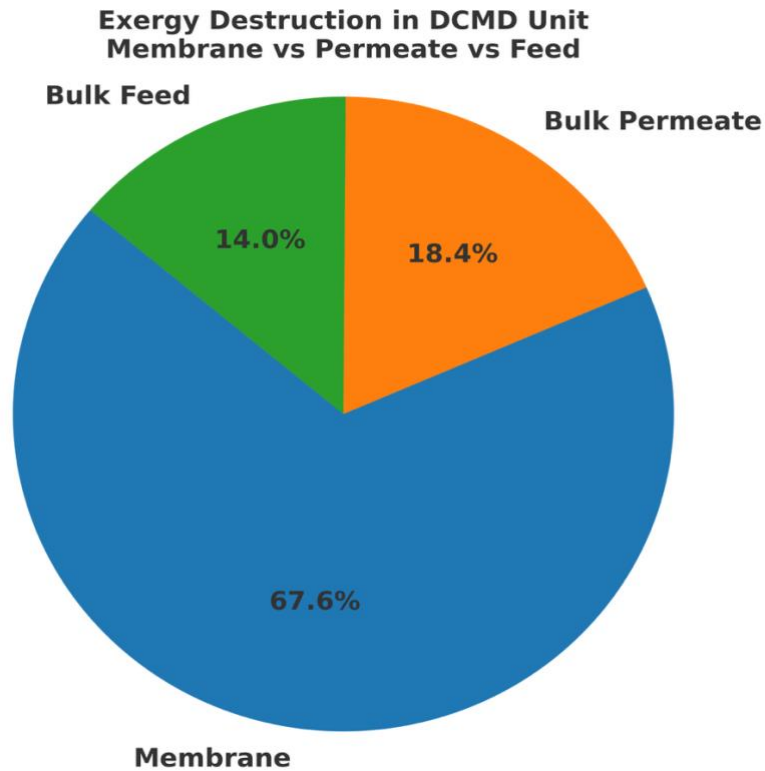


Fig. 6.10 Exergy destruction distribution of each component for DCMD

This section focuses on the exergy destruction rate within DCMD units when employing Toluene. Fig. 6.8 and Fig. 6.10 illustrates the exergy destruction in a single DCMD unit utilizing Toluene as the ORC working fluid and the percentage distribution, while Fig. 6.9 presents the cumulative exergy destruction across all required DCMD units (multiple units). In the case of a single unit, DNI's impact is considered negligible. Notably, the predominant exergy loss occurs during vapour transfer across the membrane, accounting for 67.64% of total exergetic destruction, succeeded by 18.35% across the bulk permeate and 14% across the bulk feed (see Fig. 6.10). Fig. 6.9, depicting the correlation between all DCMD units and DNI, mirrors the trends observed in Fig. 6.5 and Fig. 6.6. While the relative percentages of exergetic destruction remain consistent, the absolute exergetic destruction at the membrane escalates more rapidly with increasing DNI, followed by the permeate and feed sides. Significantly, the exergetic destruction gradient at the membrane is markedly steeper at 1.438 compared to 0.39

and 0.298 for the permeate and feed sides, respectively. This underscores the paramount importance of membrane enhancements, particularly when operating under elevated DNI conditions.

6.4 Comparison between different bottoming cycle's organic fluids impact on exergy destruction of DCMD

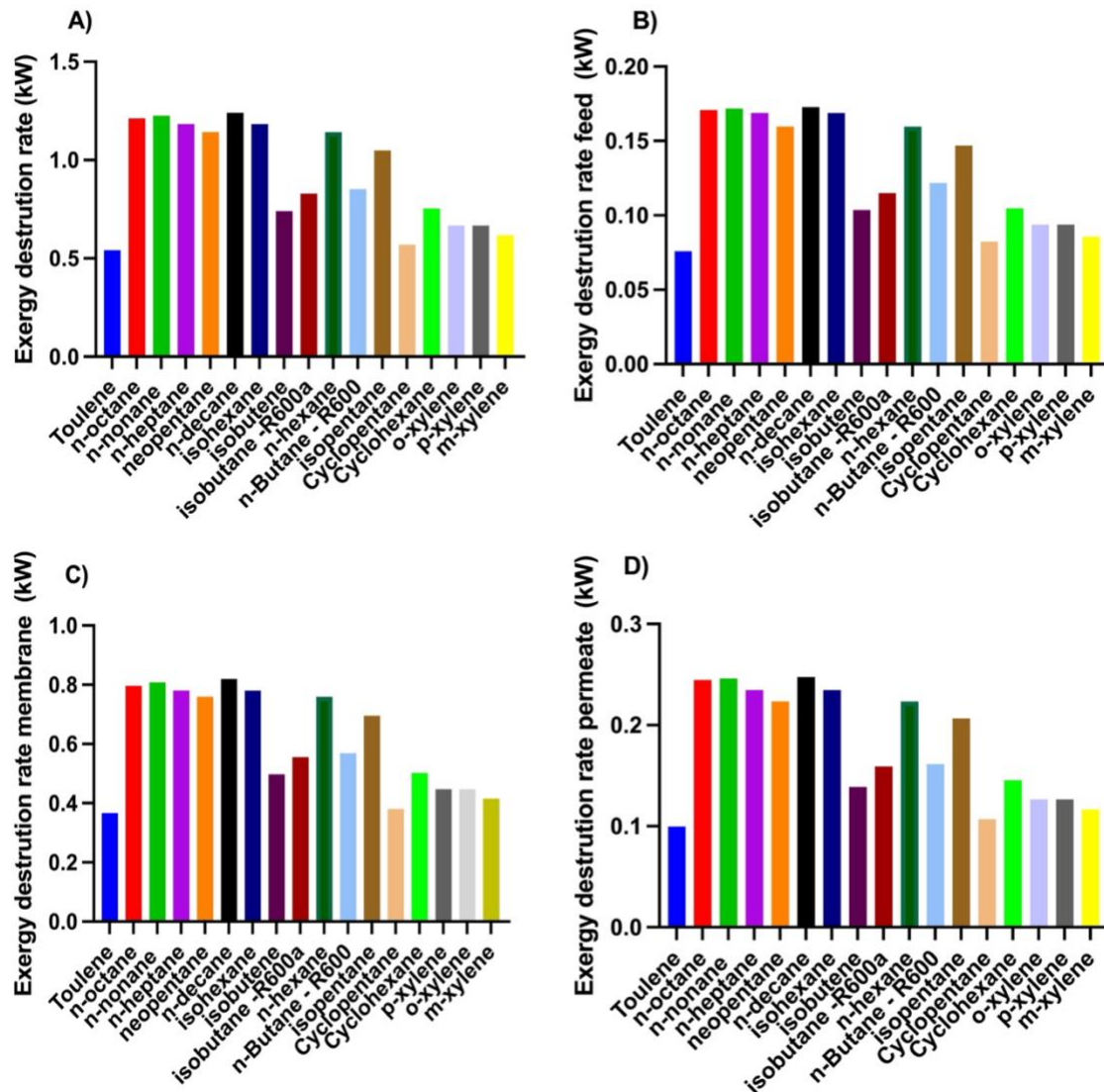


Fig. 6.11 Exergy destruction rate of single unit of DCMD.

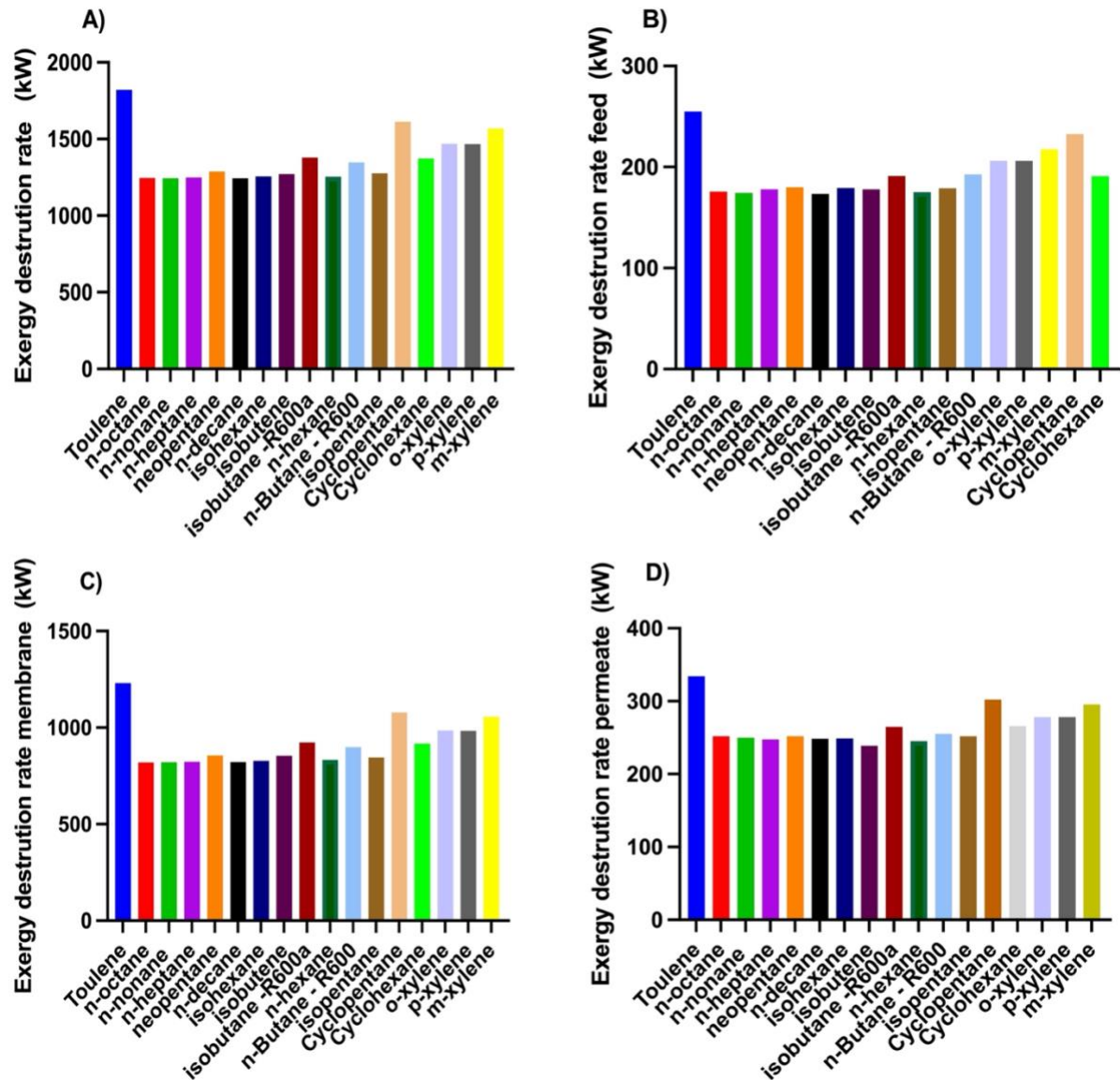


Fig. 6.12 Exergy destruction rate for cumulative DCMD units.

Fig. 6.11 and Fig. 6.12 detail the exergetic destruction rate of DCMD when different organic fluids are used in ORC with the former figure focusing on single unit and the latter on multiple units. Fig. 6.11 clearly demonstrates that each fluid showed different values of exergy destruction for each part of the DCMD unit. The exergetic destruction pattern relative to each other is the same in all three sections of the DCMD unit. What can be seen here is that the feed temperature or condenser outlet temperature influence (T_{14}) the exergetic destruction of all sections of DCMD. Higher T_{14} leads to higher exergetic destruction (refer Table 6.1).

Fig. 6.12 paints a different picture compared to Fig. 6.11. Despite having very low per unit exergy destruction, toluene shows significantly higher exergetic destruction when accounting for all units. This is because the low T_{23} leads to a large amount of mass flow rate needed and thus, more units of DCMD needed as shown in Equation A2-10.

6.5 Single vs double cycle performance

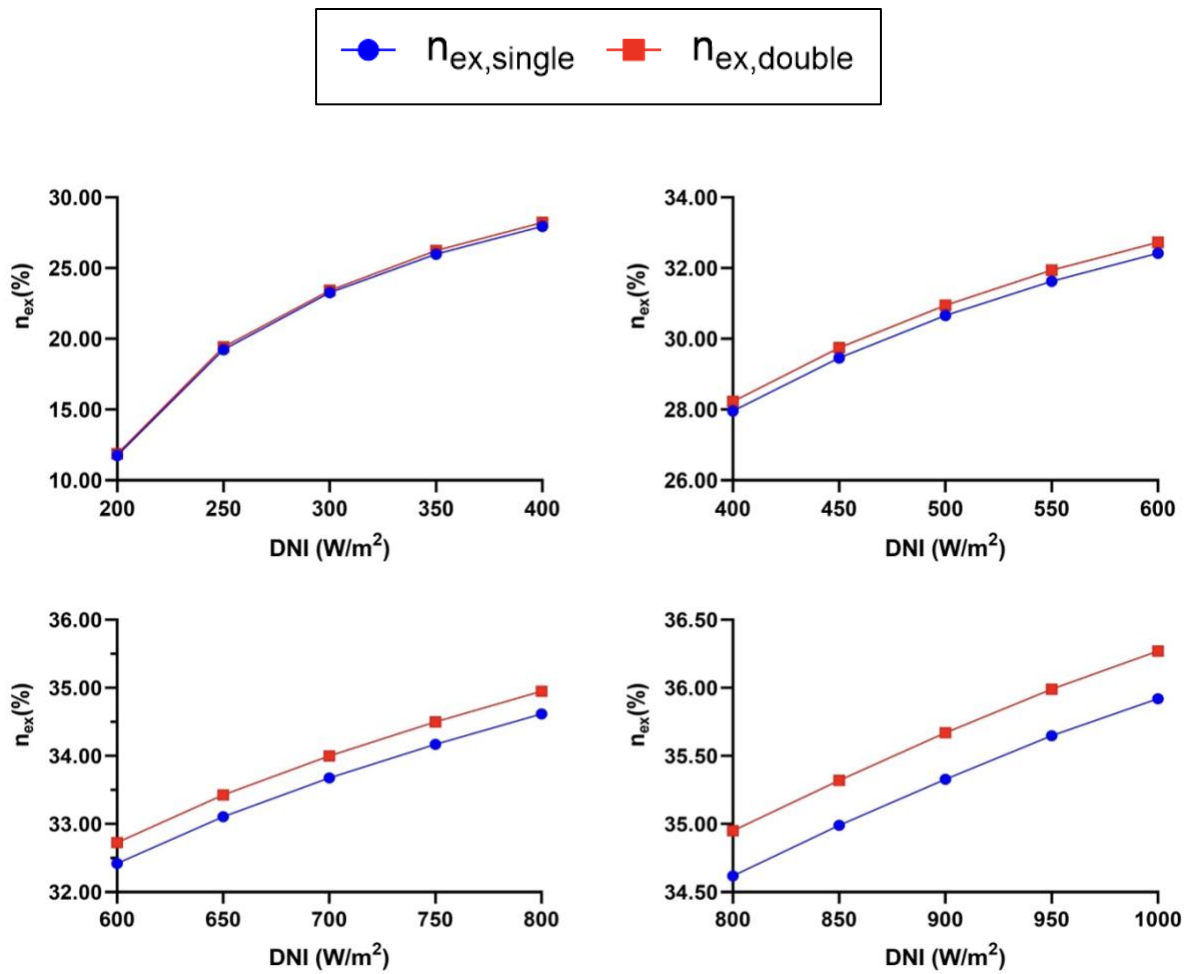


Fig. 6.13 Exergetic efficiency between single and double cycle across different DNIs

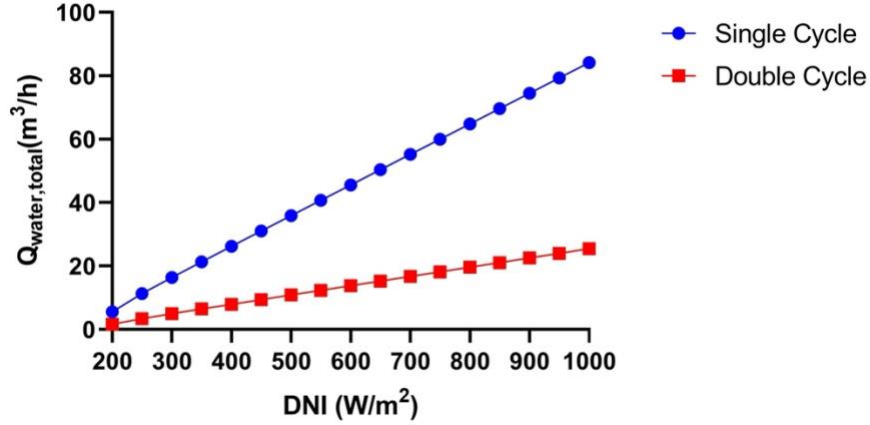


Fig. 6.14 Amount of water produced by single and double cycle.

Fig. 6.13 shows the exergetic performance of a single sCO_2 Brayton cycle against dual sCO_2 Brayton – ORC cycle across various DNI levels. As expected, the double cycle showed higher efficiency, but the efficiency tends to increase as DNI increase making the use of double cycle extremely advantageous at higher DNI levels. For example, the difference in exergetic efficiency between double and single cycle is 0.12% at 200 W/m^2 which rises to 0.31% at 600 W/m^2 and 0.35% at 1000 W/m^2 . It is also obvious from the figure that the rate of exergetic efficiency increment for both cycles reduce with increasing DNI with the most obvious increment impact between 200 to around 400 W/m^2 .

Fig. 6.14 compares the total water production of the system ($Q_{water,total}$) between the two cycles. Obviously single cycle produced more water, this is due to the amount of heat rejected by the single cycle at the cooler is higher than at the condenser of the ORC. It is also obvious that single cycle increasingly produces more amount of water as DNI increases. At 200 W/m^2 , the difference is $3.85 \text{ m}^3/\text{h}$ and it increases up to $58.67 \text{ m}^3/\text{h}$ at 1000 W/m^2 . In terms of slope, the average slope of a single cycle compared to double cycle are 0.0297 and 0.0982 respectively. The reason for this is the combination of increase energy/exergy loss in double cycle as

opposed to single cycle due to additional components, and the heat rejected by the cooler of the single cycle is higher than ORC's condenser for double cycles.

6.6 Additional DCMD units at cooler of topping cycle.

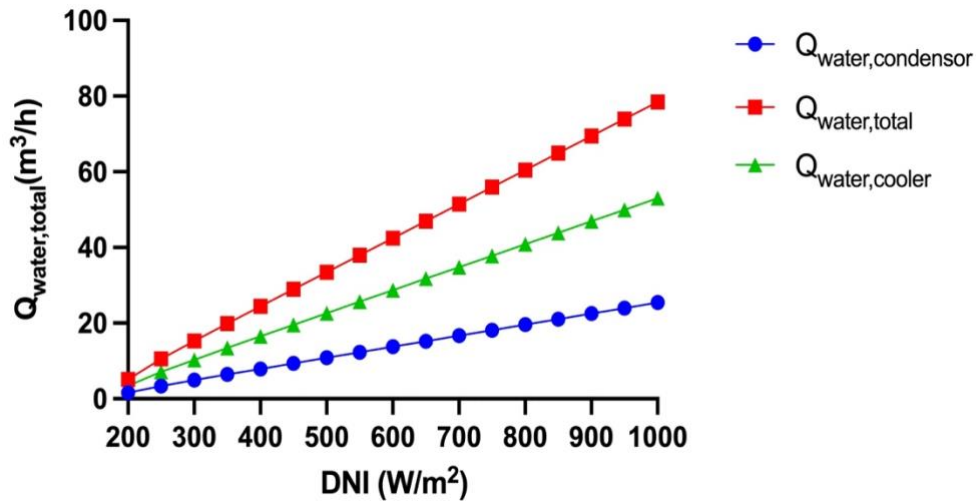
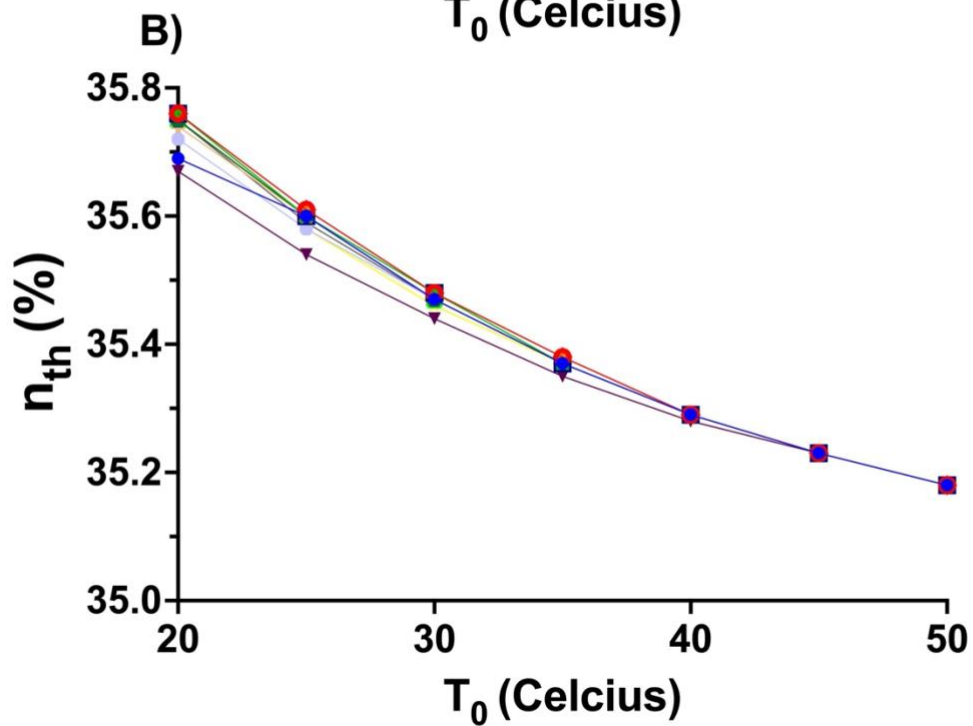
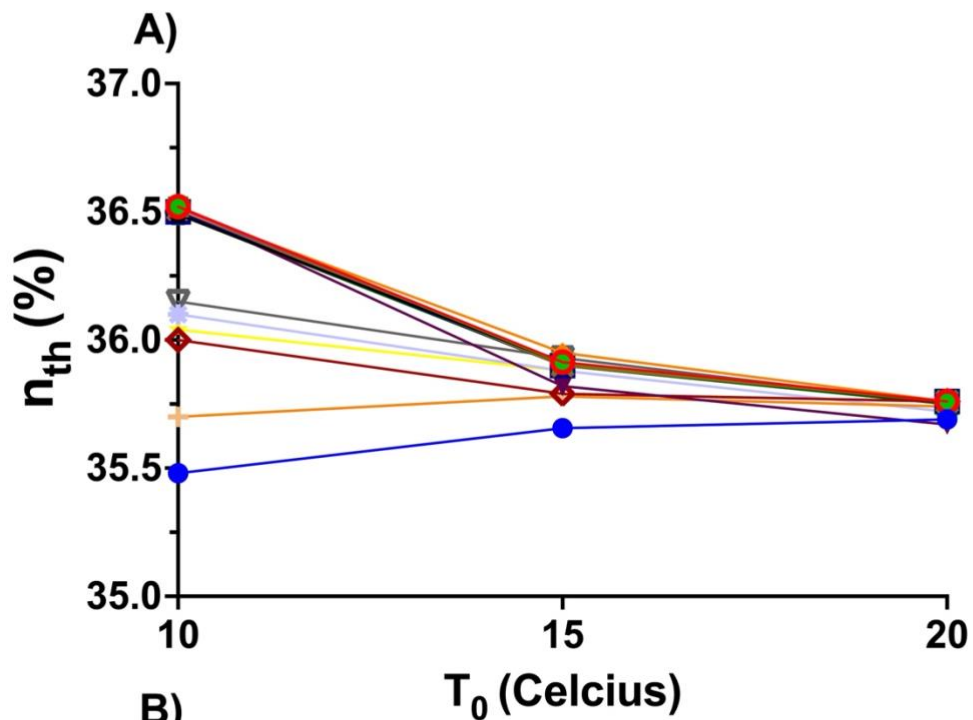
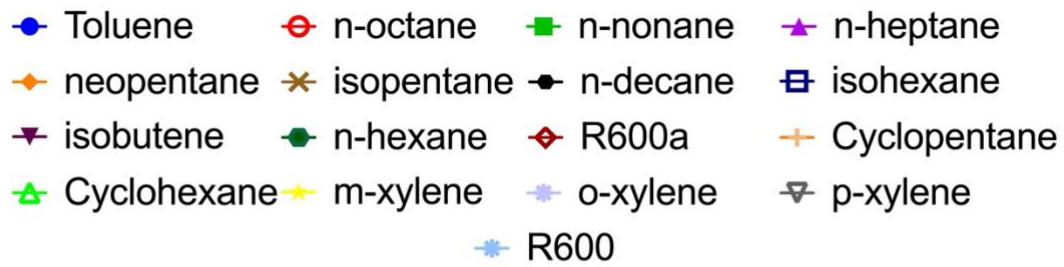


Fig. 6.15 Comparative Analysis of Total Water Production ($Q_{\text{water,total}}$) in Double Cycle with and without Utilizing a Topping Cycle Cooler.

The previous section established that water production of the double cycle outperformed the single cycle. However, additional improvements can be made to the double cycle by adding DCMDs on the cooler on the topping cycle (Fig. 3.2). Obviously, the amount of water produced is much higher than the one in Fig. 6.14 as shown in Fig. 6.15. The total water produced is maximum when DNI is at 1000 W/m^2 with $78.45 \text{ m}^3/\text{h}$ and the minimum of $5.14 \text{ m}^3/\text{h}$. This is about $53 \text{ m}^3/\text{h}$ and $3.47 \text{ m}^3/\text{h}$ respectively higher than the one in Fig. 6.14. It can also be seen that the water produced is increasingly higher at the topping cycle's cooler compared to the bottoming cycle's condenser due to the larger heat load that was rejected by the cooler compared to the condenser.

6.7 Parametric analysis of the impact of varying ambient temperatures.



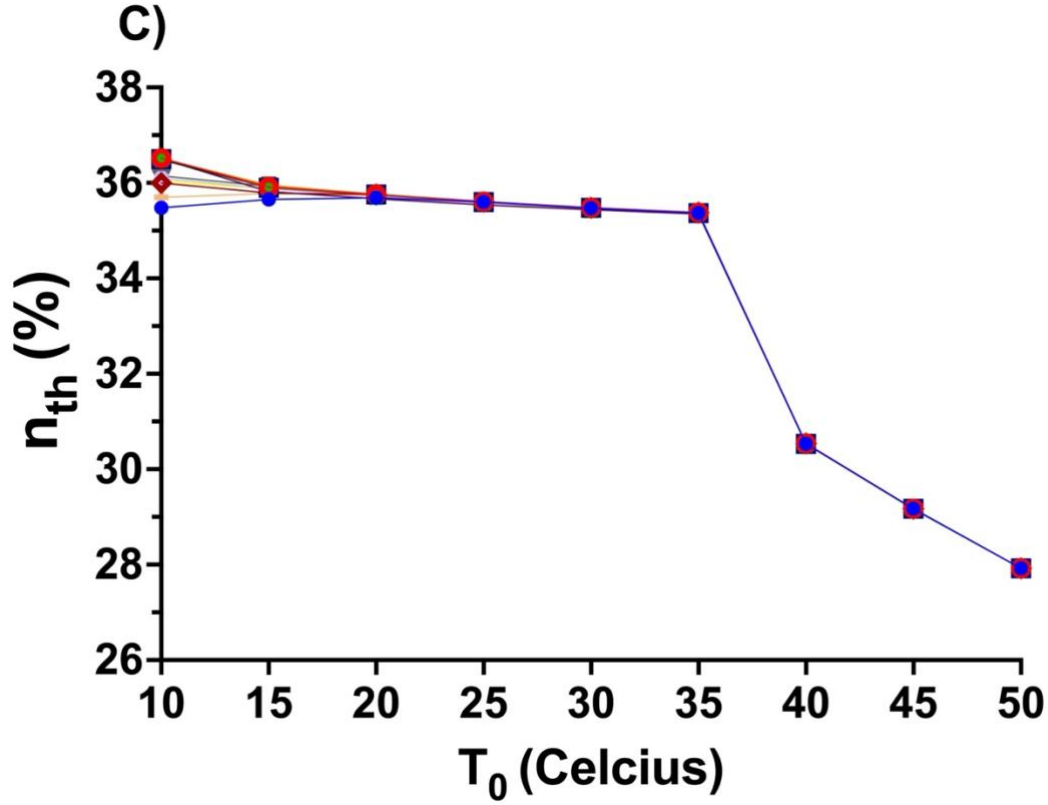


Fig. 6.16 Thermal efficiency of the combined system across different ambient temperatures using selected fluids. A) 10°C - 20°C B) 20°C - 50°C C) 10°C - 50°C

Fig. 6.16 offers a comparative evaluation of the combined system's efficiency with different ORC fluids, noting ambient temperatures in Celsius for reader convenience. Fig. 6.16A and 20B show the system's performance without auxiliary heating, while Fig. 6.16C includes it. This separation provides better insight into the system. Given the incorporation of auxiliary heating, we propose that thermal efficiency is a more appropriate metric for this segment. In Fig. 6.16A, fluids fall into two groups, Toluene and Cyclopentane, which show increased efficiency from 10°C to 15°C, and others, which demonstrate a decrease over the same range. This divergence stems from the lower optimized approach temperatures of Toluene and Cyclopentane, at 11.3K and 15.9K respectively, compared to 36.7K, 35K, and 29.3K for o-xylene, p-xylene, m-xylene, and 47.5K for remaining fluids. Contrary to expectations, lower approach temperatures typically yield higher efficiencies but, in this case,

Toluene & Cyclopentane efficiencies are constrained by fluid wetting beyond these temperatures, reducing necessary mass flow rate in the ORC and thus, diminishing power output. In contrast, fluids with higher approach temperatures achieve better efficiencies through increased mass flow rates and consequent power generation. Additionally, fluids with an approach temperature of 47.5K bypass the need for a cooler in the topping cycle, as they can expel all heat to the bottoming cycle. The elimination of the cooler in the topping cycle is thermodynamically feasible due to the influence of lower ambient temperatures on the condensation temperature T_{24} , and consequently T_{22} , which is lower than T_1 (refer to Fig. 3.2 for the numbering of the system).

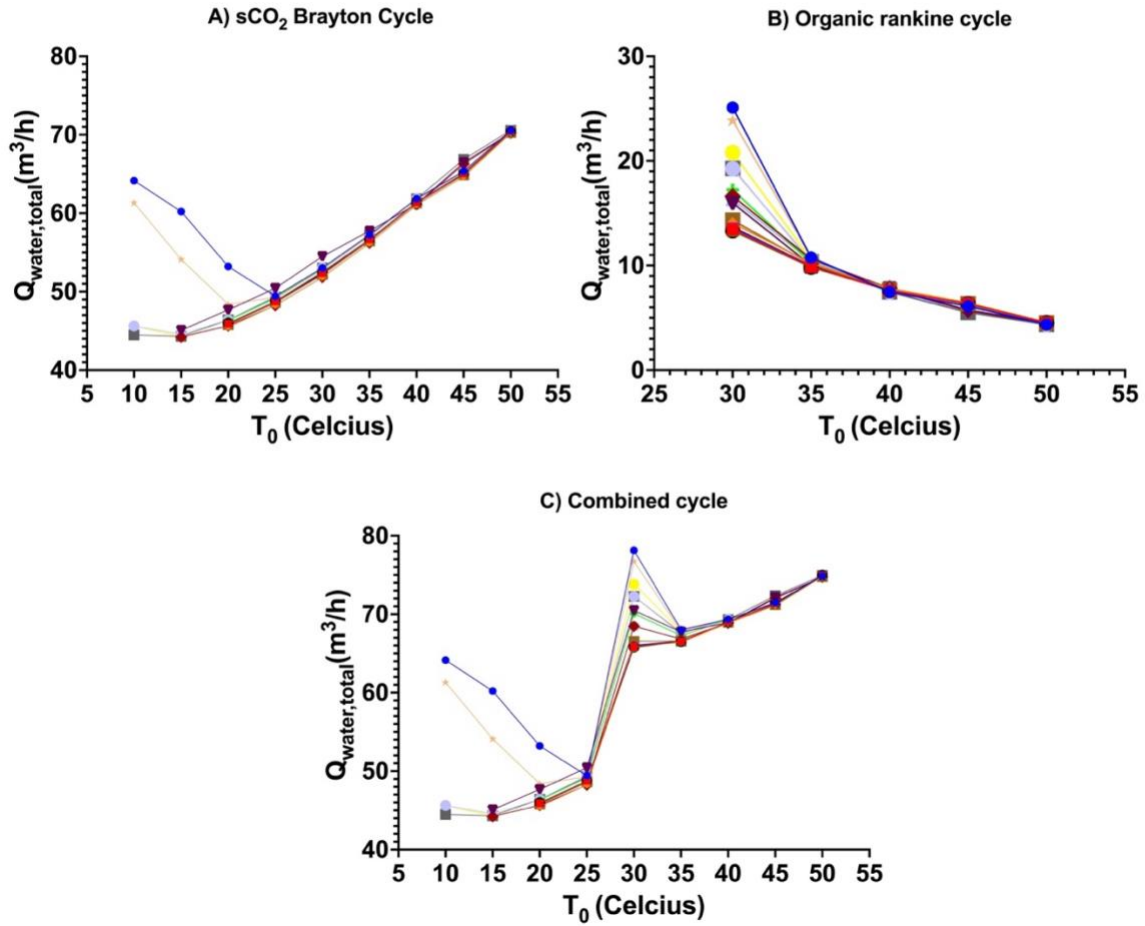
Toluene & Cyclopentane showed increase in performance from 10°C to 15°C due to the larger approach temperature and thus increase flow rate but are still relatively weak and the opposite is true for other fluids. This may seem contradictory from the previous explanation but as ambient temperature increases, so does T_{24} and T_{22} and this leads to a smaller temperature range between inlet and outlet of turbine which constraints the efficiency of a thermal engine as defined by Carnot cycle. For Toluene and Cyclopentane, the increases in approach temperature have an opposite effect in which greater mass flow rate needed by the ORC and thus, increasing the efficiency. The distinct improvement for Toluene and Cyclopentane is due to their constrained approach temperature to prevent wetting at T_0 of 10°C thus, the elevation of mass flow rate at 15°C enhances their power output, despite previously explained limitations, setting them apart from other fluids. (Refer Data Availability section for details on approach temperature optimization and mass flow adjustments). For example, from 10°C to 15°C the mass flow rate increments for Toluene and Cyclopentane are respective 93% and 50% whereas it is only 2.88% for n-Octane. This emphasises the complex relationship between mass flow rate and power generation in relation to the constraints that are associated with different ambient temperatures. The moderation in increment of Toluene and Cyclopentane seen from

ambient temperature of 15°C to 20°C is due to the lower mass flow rate increment with only 48% and 38% increment due to the diminishing advantage of approach temperature increment as it is less constrained by fluid wetting at this temperature.

Fig. 6.16B shows a straightforward relationship between increasing ambient temperatures and falling efficiency for all fluids. This trend stems from a rise in T_{22} and a drop in the optimized approach temperature at higher ambient temperatures, leading to a reduction in mass flow intake, and/or diminished pressure and/or temperature differences across the turbine. Notably, Toluene and Cyclopentane reach their efficiency peaks at ambient temperatures of 20°C and 15°C, respectively. After these peaks, their efficiency declines due to the dominant influence of lower approach temperature which leads to lower flow rates, and/or temperature and/or pressure difference at the turbine inlet and outlet, mirroring other fluid's behaviour. Fig. 6.16 underscores the comparative advantage of adding an ORC as a supplementary cycle in cooler settings, such as the Andasol power plant in Fig. 6.16C is when the impact of auxiliary heating ($Q_{Auxilliary}$) is added to the system. It can be seen from ambient temperature of 35°C onwards, there is a sharp decline in efficiency. The reason for this is that heating is required when ambient and thus seawater temperature past 30°C to bring down the temperature at state 1 to 304.25K (31.1°C).

Málaga, Spain, particularly during the winter, as opposed to warmer areas like Abu Dhabi, where the efficiency improvement is more muted. The highest and lowest recorded efficiencies are for n-nonane at 36.52% at T_0 of 10°C, and m-xylene at 35.18% at 50°C. At this stage, the merit of the ORC optimization method shown in Fig. 3.11 warrants discussion. By adjusting the approach temperature, it identifies an optimal setting that maximizes cycle efficiency and prevents fluid wetting post-turbine expansion, notably at low ambient temperatures. This contrasts with previous studies that assumed a fixed approach temperature and determined mass flow rate through energy balance.

- Toluene
 —■— n-octane
 —▲— n-nonane
 —▼— n-heptane
- ◆— neopentane
 —●— n-decane
 —■— isopentane
 —▲— isohexane
- ▼— isobutene
 —◆— R600a
 —■— n-hexane
 —◆— Cyclopentane
- ▲— Cyclohexane
 —■— R600
 —●— o-xylene
 —■— p-xylene
- m-xylene



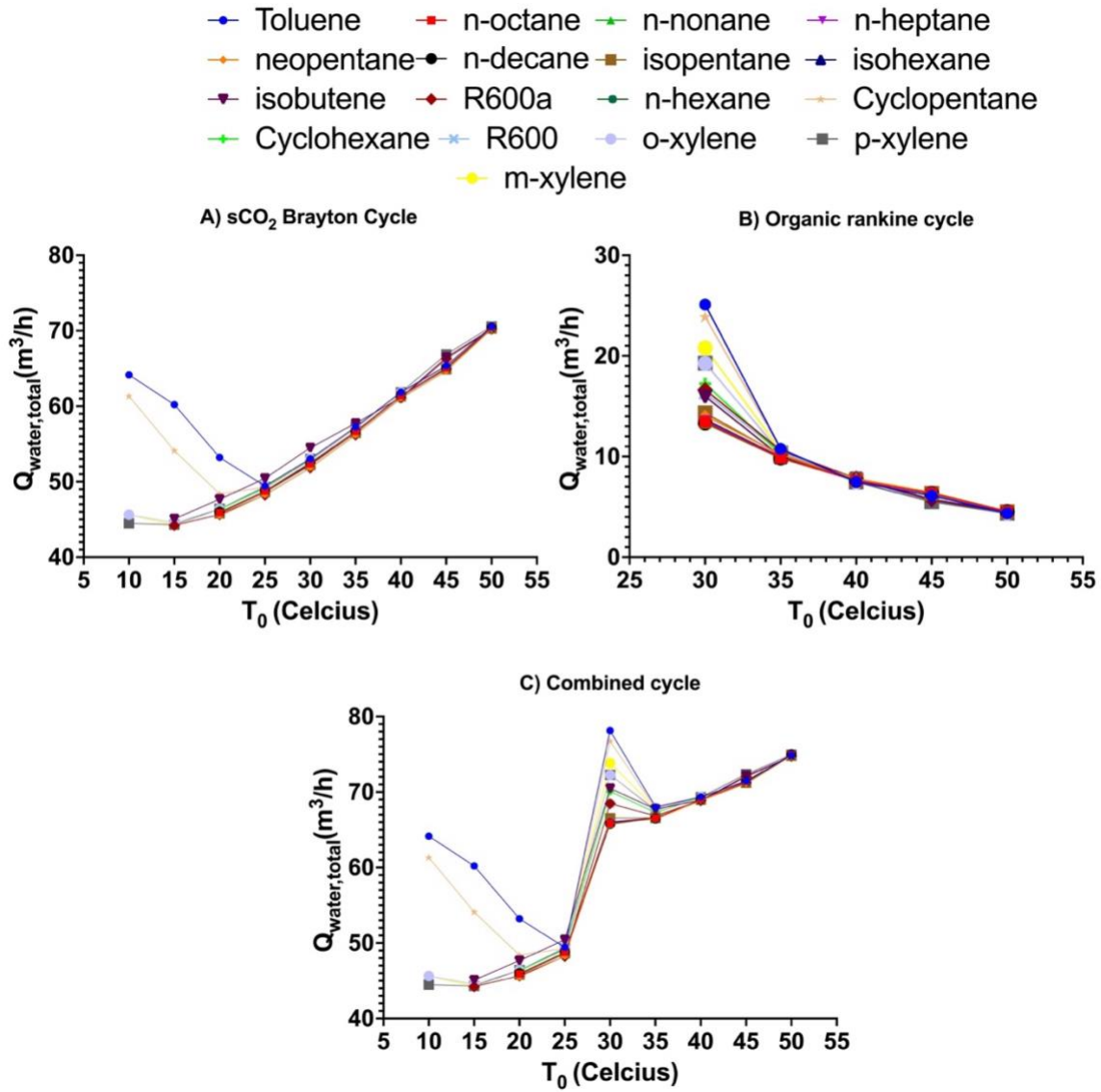


Fig. 6.17 The total water produced by A) sCO₂ Brayton cycle B) ORC and C) Combined cycle across various ambient temperature (T_0)

Fig. 6.17 illustrates the total water production $Q_{water,total}$ at varying ambient temperature (T_0) of the examined ORC fluids. As can be seen in Fig. 6.17A, only five fluids produced water at the cooler of sCO₂ Brayton cycle. This is because other fluids make full use of the topping cycle's heat and thus, cooler is not in use. The data also reveal a distinct pattern for Toluene and Cyclopentane, where water production initially decreases before increasing, reaching a minimum point, a trend not clearly observed with other fluids. As mentioned in Fig. 6.16, at lower ambient temperatures, Toluene and Cyclopentane absorb minimal heat from the topping

cycle, in contrast to other fluids which absorb substantially more, leaving limited residual heat for water production in the cooler. However, as ambient temperatures rise, these fluids absorb less heat, thus enhancing the cooler's capacity for water production. For Fig. 6.17B a clear reduction in water production can be seen. This is because from ambient temperature of 30°C onwards, as mentioned earlier, the heat uptake by ORC reduces which leads to less water production via ORC's condenser. It should be noted that the ORC commences water production only at ambient temperatures of $\geq 30^\circ\text{C}$, as temperatures below this threshold resulted in T_{23} too low for generating any meaningful temperature and vapour pressure difference across the feed and permeate sides of the DCMD. Fig. 6.17C illustrates a notable increase in water production at the 30°C mark, attributed to the start of water production by the ORC. Beyond this temperature, water production initially declines before experiencing a modest rise. This pattern is due to a marked decrease in ORC water production between 30°C and 35°C, which surpasses the increase from the topping cycle's cooler. Subsequently, as the rate of decrease in ORC water production moderates, the contribution from the topping cycle's cooler becomes predominant, leading to an overall increase in water production for the combined cycle. The overall water production suggests that systems designed at lower ambient temperature benefit from the usage of Toluene and Cyclopentane if water is of great concern.

Further discussion is warranted in this section, as indicated in section 6.1.2 and Fig. 6.11, where the significance of T_{23} on mass flow rate and consequently on water production ($Q_{water,total}$) is highlighted. This section underscores the critical relationship between the seawater mass flow rate at the ORC's condenser and T_{23} and thus DCMD's feed temperature, T_{14} . As depicted in Fig. 6.17B, Toluene exhibited the highest water production at an ambient temperature of 30°C, confirming the findings of Fig. 6.11. Yet, as the ambient temperature increases, a shift occurs in the fluid achieving maximum water production. Beyond 40°C,

several other fluids in water production outperform Toluene. This nuanced interaction is further elaborated in Fig. 6.18.

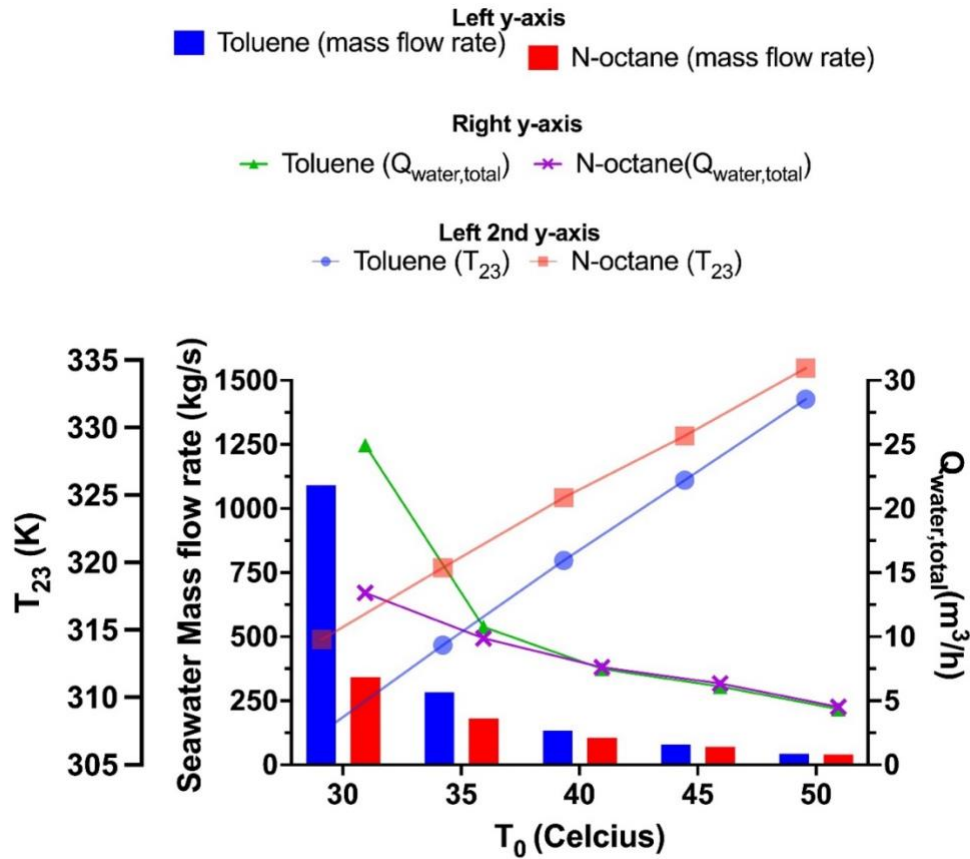
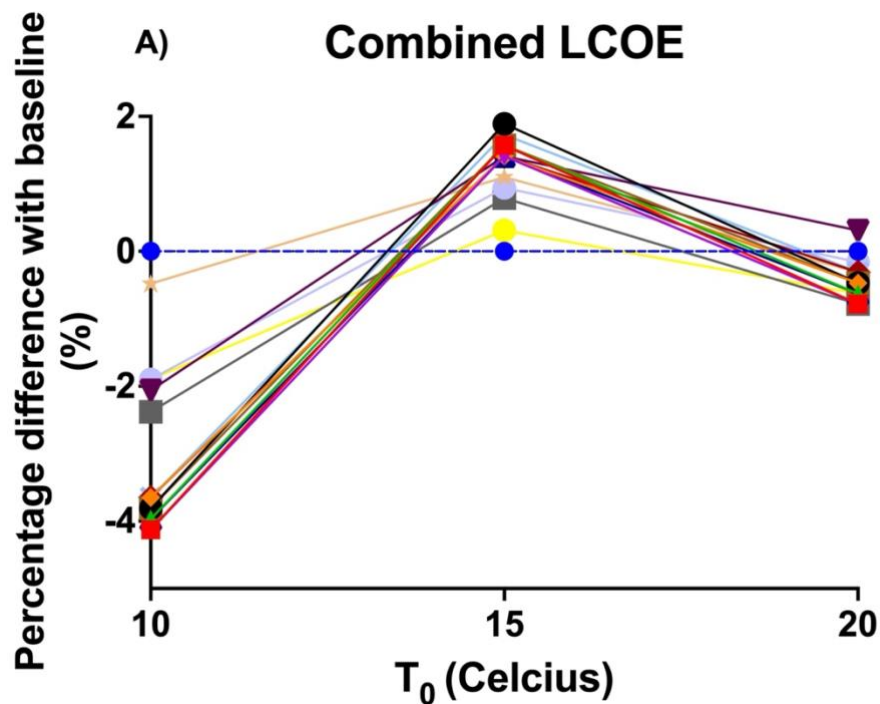
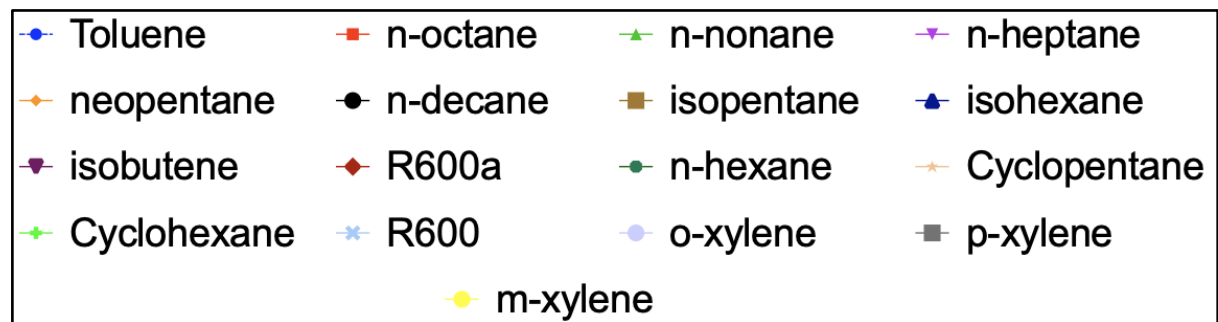


Fig. 6.18 The illustration on how T_{23} , the mass flow rate of seawater at the condenser, affects the total water production ($Q_{water,total}$) in the bottoming ORC.

Fig. 6.18 shows the relationship between T_{23} , mass flow rate of seawater at the condenser and the water production $Q_{water,total}$ at the ORC. At ambient temperature of 30°C, the mass flow rate of Toluene is 1091.52 kg/s compared to N-Octane 283.13kg/s due to low T_{23} of 308.65K and 314.4K respectively. This significant disparity results in increased water production with Toluene due to the necessity for more DCMD units to accommodate higher flow rates. Nevertheless, as the ambient temperature increases, the gap in T_{23} narrows, leading to a decreased mass flow rate. Upon reaching an ambient temperature of 40°C, the higher T_{23} value for N-Octane, which contributes to a higher feed temperature (T_{bf}) in the DCMD and thus

permeate flux of DCMD, surpasses the benefits of Toluene's higher mass flow rate, thereby enhancing water production. This result represents a notable shift from earlier PTSC-DCMD studies, like Soomro & Kim (2018a) and Ouali et al. (2023), which primarily identified the DCMD's feed temperature as the key determinant of water production efficacy. This scenario underscores the intricate relationship between T_{23} and thus T_{14} and mass flow rate within a waste heat powered, power cycle-DCMD system, meriting further in-depth exploration in future works.

6.8 Economic comparative analysis



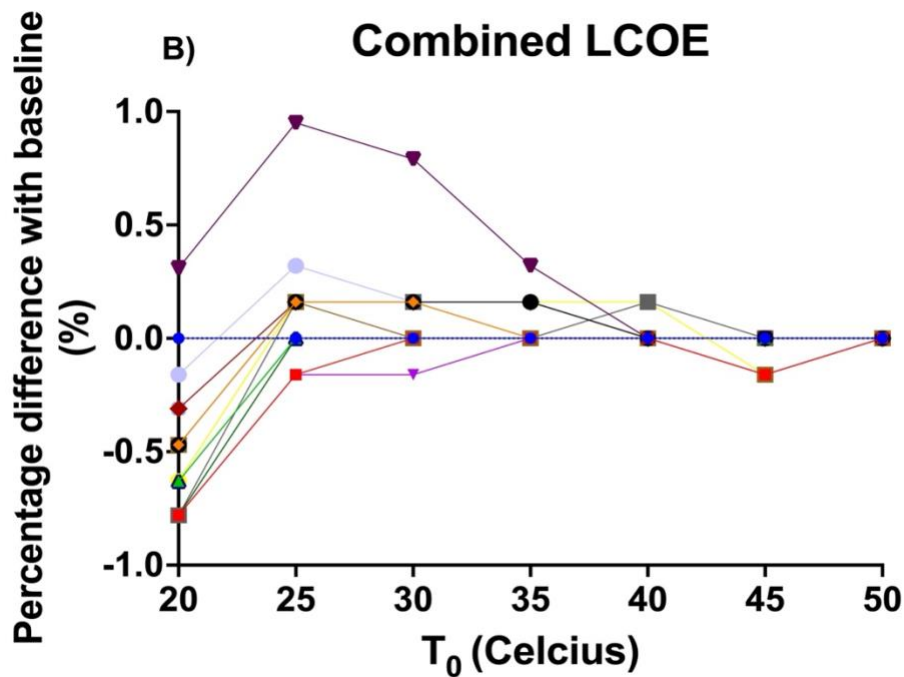


Fig. 6.19 LCOE of the combined system with ambient temperature.

Fig. 6.19 visualises the LCOE of the system with respect to ambient temperature. In this section, only the double cycle configuration (Fig. 3.2B) is considered due to its higher efficiency and water production. Since the system prioritizes power production, using the double cycle results in better overall performance. The obvious observations here are that the fluids with higher efficiency in Fig. 6.16 shows lower LCOE relative to baseline by up to of around 2% to 4% for well understood reasons. Additionally, at 10°C, the use of high-efficiency fluids, as shown in Fig. 6.16, facilitates a higher rate of utilization of waste heat from the topping cycle. This either minimizes the size of the topping cycle's cooler or eliminates it entirely, thereby reducing bare module costs (see APPENDIX 3 MODULE COSTING TECHNIQUE and section 3.6.4) and capital expenditures, resulting in noticeably lower LCOEs. However, there is an anomaly at 15°C, where the less efficient fluids shown in Fig. 6.16 such as toluene and o-xylene, show lower LCOE. The exception can be attributed to the fact that the efficiency difference between fluids at 15°C is narrower and these fluids lead to higher equipment costs (due to higher mass flow rate needed to absorb all the heat rejected by the

topping cycle) leads to more significantly more expensive ORC's CAPEX and thus, leading to higher LCOE. Moreover, it is also apparent that isobutene generally incurs higher costs than other fluids across most ambient temperatures due to its lower efficiency, which is consistent with Fig. 6.16. However, at higher ambient temperatures, the LCOE differential is minimal, with n-octane demonstrating a modestly lower LCOE. The slight variations between fluids at higher temperatures are likely due to differences in power production and bare module equipment cost associated with each fluid's thermophysical properties. In addition, at all temperatures (with the exception of 15°C), the graph nearly mirrors the thermal efficiency performance of Fig. 6.16, indicating that equipment cost variations of ORC likely played a minimal role in LCOE performance. This is due to the significantly dominant effect of the topping cycle which significantly outweighs the bottoming cycle's ORC in terms of power output and thus, cost.

● Toluene	■ n-octane	▲ n-nonane	▼ n-heptane
◆ neopentane	● n-decane	■ isopentane	▲ isohexane
▼ isobutene	◆ R600a	● n-hexane	★ Cyclopentane
▲ Cyclohexane	★ R600	● o-xylene	■ p-xylene
	● m-xylene		

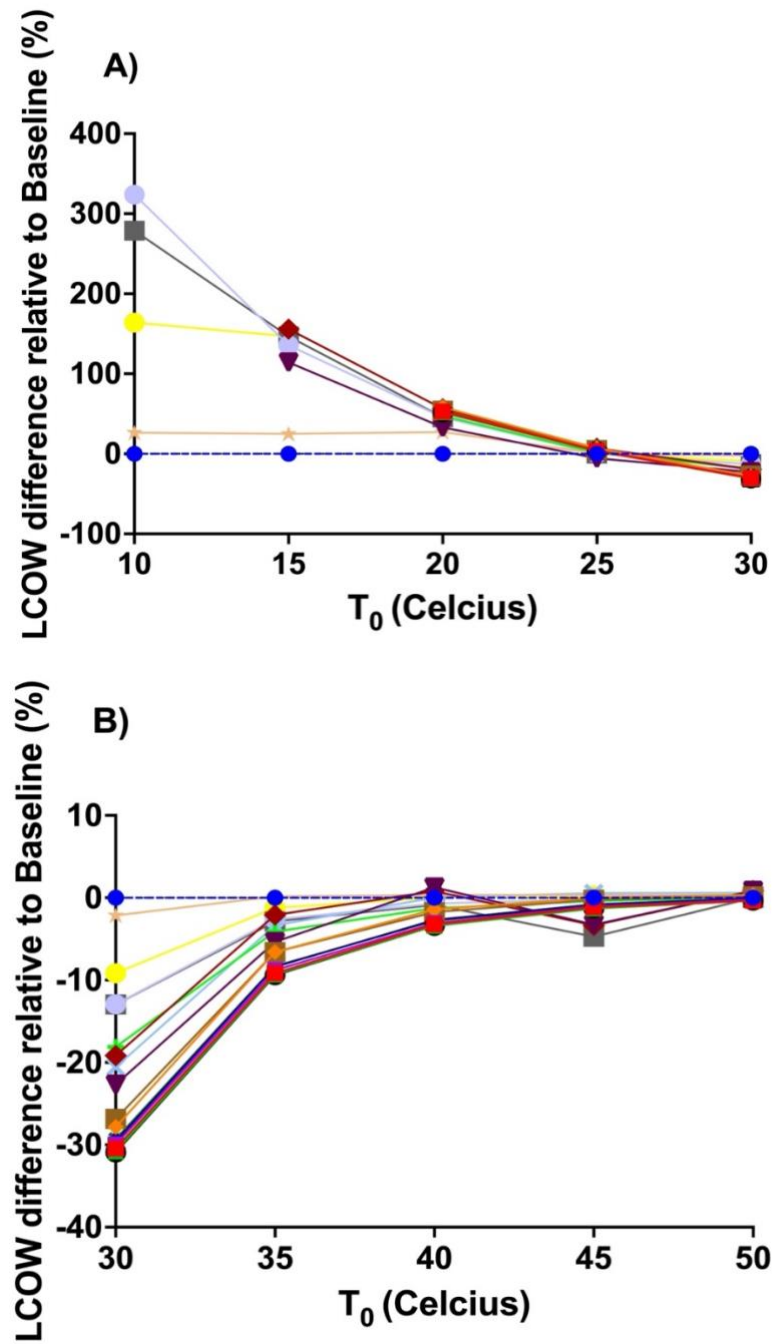


Fig. 6.20 LCOW of the system across various ambient temperatures

Fig. 6.20 illustrates the Levelized cost of water (LCOW) of the system across different ambient temperatures and organic fluids. At ambient temperatures ranging from 10°C to 20°C, all fluids exhibited performance inferior to the baseline, evidenced by a significantly higher LCOW. This makes toluene the preferable choice for achieving lower water costs at these

temperatures. The rationale behind this observation is that toluene absorbs less heat from the topping cycle within the 10°C to 20°C range, thus allowing greater heat rejection at the topping cycle's cooler, which in turn enhances the potential for water desalination and significantly reduces cost due to economies of scale, albeit with lower system's efficiency and higher LCOE. This is due to the lack of water production via ORC's condenser, because temperature at T_{23} for the reason explained in Fig. 6.17. Notably, at 30°C, toluene shows the highest LCOW values despite the highest water production as shown in Fig. 6.17. The higher mass flow rate of toluene leads to a disproportionate increase in both CAPEX and OPEX due to the need for additional units of DCMD, as indicated in Fig. 6.17, overcoming the influence of price reduction due to economies of scale. The results highlight how the interplay of mass flow rate, condensation inlet temperature, and DCMD feed temperature impact not just technical performance but also economic performance. However, at higher temperatures, most other fluids are preferred over toluene, but the difference diminishes with increasing ambient temperature. The minimal difference at higher temperatures is due to the increased condensation temperature (T_{24}) in the Organic Rankine Cycle (ORC) at elevated ambient temperatures. This increase reduces the temperature and pressure differential between the high and low-pressure sides of the ORC. Consequently, the narrower heat absorption capacity of the ORC decreases the temperature difference between the working fluid's temperature at point 21 and 23, leading to reduced variations in heat rejection by the ORC and the cooler.

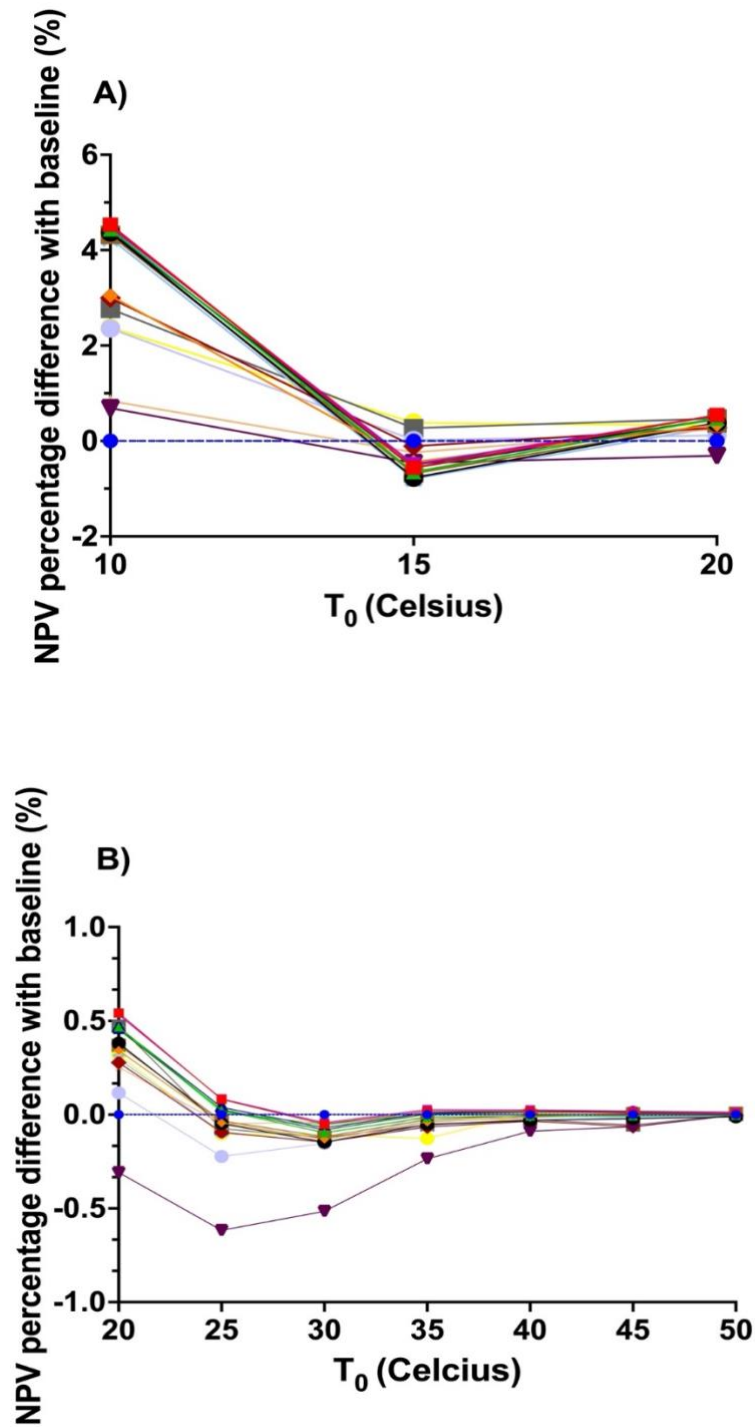


Fig. 6.21 Net present value (NPV) of the combined system between the examined organic fluids across various ambient temperatures

Fig. 6.21 illustrates the net present value (NPV) of the combined system using various organic fluids and ambient temperatures. NPV is utilised to evaluate the economic viability of the system by considering both outputs, therefore NPV acts as the unifying index of the two outputs.

Previous sections focused only on the production cost per unit, omitting consideration of the system's total output. The most profitable fluid at an ambient temperature of 10°C to 20°C is n-octane, with around 4.5% higher NPV. The least profitable fluid across most temperatures is isobutene. At 30 °C, however, toluene achieves the highest NPV, demonstrating that despite its higher LCOW in Fig. 6.20, the economic feasibility is strongly driven by power generation. This is further evidenced by the close alignment between NPV and LCOE trends shown in Fig. 6.19. With a few exceptions, variations in NPV for the fluids evaluated between temperatures of 30°C and 50°C are merely between -0.15% and 0.15%, indicating that the choice of working fluids has a small impact on the system's profitability at these temperatures and alternative factors might take precedence. Table 6.3 presents the recommended working fluids for each ambient temperature, based on the relative values of LCOE, LCOW, and NPV compared to the baseline and to each other.

Table 6.3 Organic fluid recommendation for each ambient temperature based on economic evaluation.

Ambient temperature, T_0 (°C)	LCOE (USD/kWh) (% difference with baseline) – Lower is better	LCOW (USD/m ³) (% difference with baseline) – Lower is better	NPV (USD) (% difference with baseline) – Higher is better
10	n-octane (-4.13%) n-nonane (-3.97%) n-heptane (-4.13%) neopentane (-3.65%) n-decane (-3.81%) isopentane (-3.81%) isohexane (-3.97%) R600a (-3.65%) n-hexane (-3.97%) Cyclohexane (-4.13%) R600 (-3.65%)	Toluene (0%)	n-octane (4.534%) n-nonane (4.438%) n-heptane (4.484%) n-decane (4.371%) isopentane (4.318%) isohexane (4.398%) n-hexane (4.379%) Cyclohexane (4.507%) R600 (4.247%)
15	Toluene (0%) m-xylene (0.31%) p-xylene (0.79%)	Toluene (0%)	Toluene (0%) p-xylene (0.270%) m-xylene (0.380%) o-xylene (0.02%)
20	n-octane (-0.78%) n-nonane (-0.63%) n-heptane (-0.78%) isohexane (-0.63%) n-hexane (-0.78%) Cyclohexane (-0.78%) p-xylene (-0.78%) m-xylene (-0.63%)	Toluene (0%)	n-octane (0.542%) n-nonane (0.468%) n-heptane (0.532%) isohexane (0.465%) n-hexane (0.452%) Cyclohexane (0.467%) p-xylene (0.472%)
25	n-octane (-0.16%) n-heptane (-0.16%)	Isobutene (-5.59%)	All fluids except: m-xylene (-0.1029%) o-xylene (-0.223%) Isobutene (-0.617%)
30	All fluids except: Isobutene (0.79%)	n-octane (-30.25%) n-nonane (-30.80%) n-heptane (-29.97%) neopentane (-27.82%) n-decane (-30.93%) isopentane (-26.86%) isohexane (-29.59%) n-hexane (-29.33%)	All fluids except: n-octane (-0.118%) n-nonane (-0.145%) n-heptane (-0.153%) p-xylene (-0.122%) m-xylene (-0.107%) R600a (-0.146%) R600 (-0.146%)

			o-xylene (-0.153%) p-xylene (-0.122%) m-xylene (-0.107%) Isobutene (-0.515%)
35	All fluids except: Isobutene (0.32%)	n-octane (-9.169%) n-nonane (-9.324) n-heptane (-8.780%) neopentane (-6.605%) n-decane (-9.401%) isopentane (- 6.6054%) isohexane (-8.392%) n-hexane (-8.314%)	All fluids except: Isobutene (-0.235%) m-xylene (-0.127%)
40	All fluids (negligible differences)	n-octane (-3.249%) n-nonane (-3.408%) n-heptane (-3.091 %) n-decane (3.408%)	All fluids (negligible differences)
45	All fluids (negligible differences)	P-xylene (-4.735%) Isobutene (-3.265%) R600a (-3.347%)	All fluids (negligible differences)
50	All fluids (negligible differences)	n-octane (-0.342%) n-nonane (-0.342%) n-heptane (-0.257%) n-decane (-0.343%)	All fluids (negligible differences)

At this stage, it is worth discussing the practical implications of these fluids. Across the ORC candidates evaluated in this thesis (Table 6.3), real-world deployment depends on how working-fluid properties interact with climate, operating safety, and sourcing practicality. Light hydrocarbons with low flash points and high vapour pressure, such as neopentane, isobutene, R600a/isobutane, R600/butane, isopentane, cyclopentane, n-hexane and isohexane, carry the highest fire risk and require strict containment and ventilation measures, especially in hot climates where vapour build-up is more likely. Moderate volatility fluids, including cyclohexane and n-octane, are comparatively easier to manage and therefore more viable across a wider temperature range, which supports deployment in temperate and warmer CSP regions, while still requiring leak detection and controlled handling. Heavier fluids, such as n-

nonane, n-decane, toluene, and xylenes, remain flammable but show lower vapour release, though their toxicity and odour thresholds mean that maintenance activities demand careful exposure control regardless of location. Under closed-loop operation, these risks are managed using sealed circuits, fire-safe zoning, forced ventilation, and appropriate protective measures, allowing fluid choice to prioritise performance without compromising safety. Consequently, climate influences deployment feasibility not only through thermodynamic behaviour but also by affecting how easily each fluid can be handled on-site, making fluids such as cyclohexane and n-octane practical options where higher ambient temperatures and strong solar resource are expected, while lighter fluids may require tighter operational controls in similar environments. As a result, working-fluid selection in practice balances efficiency with climate suitability, handling demands, and sourcing stability, rather than thermodynamic performance alone.

6.9 Conclusion

This chapter extended the system by adding an ORC bottoming cycle and compared working fluids, operating conditions, and ambient temperatures to assess how these factors influence net power, water production, and useful-energy losses. The analysis showed that different fluids favour different priorities: some improved net power slightly, while others increased water production by enabling cooler conditions on the condenser side. Loss breakdowns identified DCMD-related losses as especially important, while the integrated double-cycle layout delivered a small but measurable gain in usable-energy efficiency of 0.12–0.35% compared with single-cycle cases, with the advantage growing at higher DNI. Finally, the chapter clarified which fluids are better suited to colder climates and which remain more practical across broader temperature ranges, linking selection to both performance and handling considerations. These performance and loss patterns identify where design choices

have the greatest influence, which motivates translating them into practical design and cost trade-offs, the focus of the next chapter's thermo-economic analysis.

CHAPTER 7 THERMO-ECONOMIC MAPPING

This section presents a comprehensive thermo-economic analysis of the system over a 25-year operational lifespan, which is representative of conventional power generation systems in the industry and the academic literature. This section also introduces multiple methodological approaches that are currently not present in the literature. Those approaches are:

- 1) A detailed thermodynamic investigation will be conducted to examine a wide range of combinations involving the high-temperature recuperator (HTR), low-temperature recuperator (LTR), turbine inlet temperatures, and power outputs. These parameters are critical in the design and operation of sCO₂ cycles, with recuperators recognised as the most economically influential components (Xu et al., 2021). The turbine inlet temperature directly governs both the cycle's efficiency potential, as described by the Carnot principle, and the associated heat input requirements, while the power output determines the scale of application, spanning from small-scale generation of around 5 MWe to large-scale plants up to 1000 MWe, all of which are compatible with the considered systems. This analysis will provide a comprehensive understanding of how these parameters interact, thereby identifying opportunities for optimisation, innovation, and future performance improvements.
- 2) Detailed economic analysis that considers each of the component of the cycle using a tailored economic correlation for each of the component in sCO₂ Brayton cycle and ORC, under diverse range of operating and ambient temperature.
- 3) The utilisation of the duality method as shown in Section 6.1.1, which has proven to be advantageous relative to the typical method.
- 4) Future cost projection of sCO₂ Brayton cycle and the rest of the system that best compared the likely cost of the system in comparison to the targeted CSP LCOE and Membrane distillation (MD) LCOW.

Thermodynamic analysis of sCO₂ Brayton cycle across various HTR, LTR, power outputs and turbine inlet temperature.

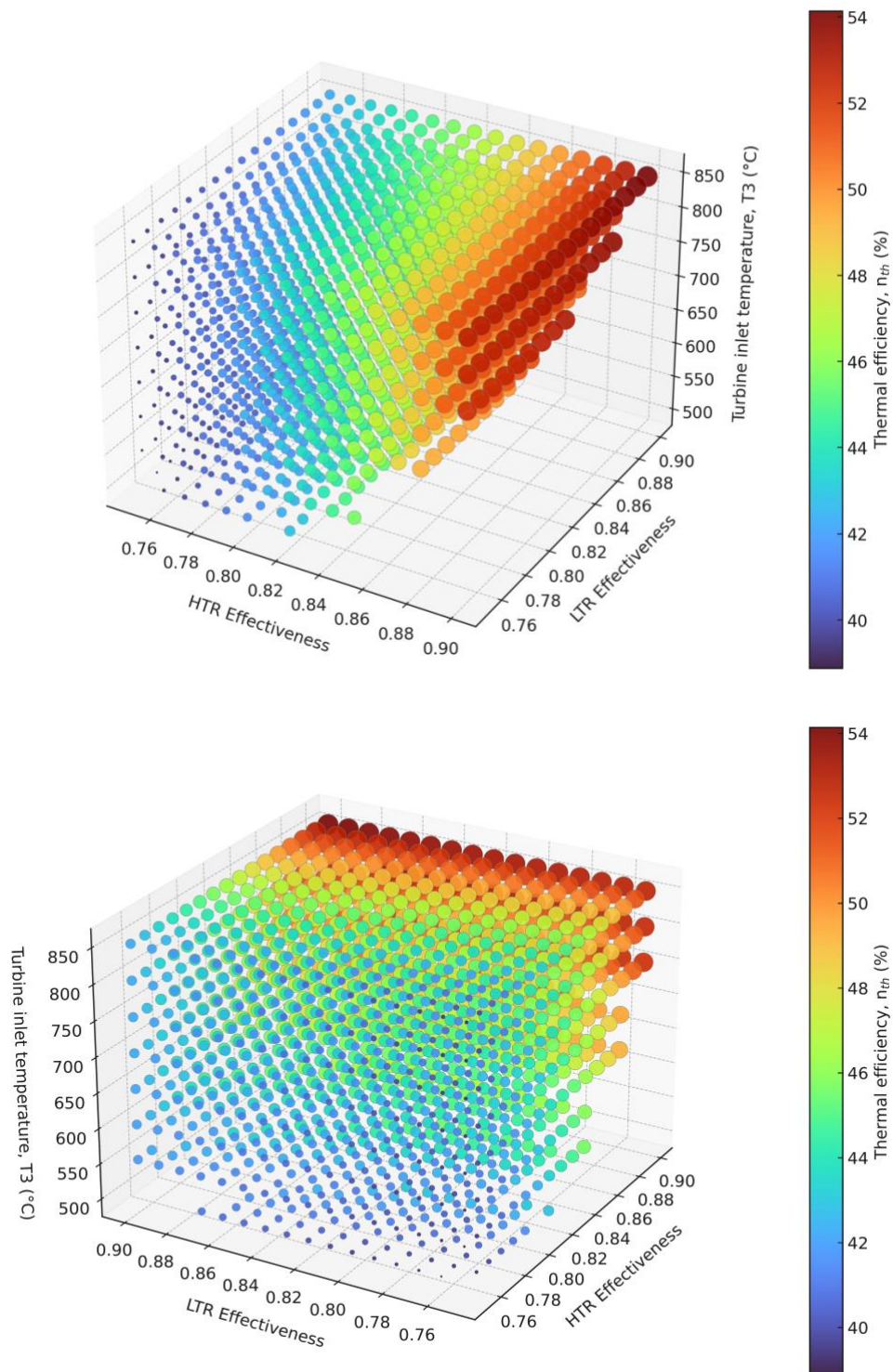


Fig. 7.1 Thermal efficiency relative to the combinations of HTR and LTR effectiveness and turbine temperature, T_3 at different angles at $50MW_{e,net}$

Fig. 7.1 exhibits the variation of thermal efficiency as a function of the high-temperature recuperator (HTR) effectiveness, low-temperature recuperator (LTR) effectiveness, and turbine inlet temperature (T_3) at reference power output of $50\text{MW}_{\text{e,net}}$. The efficiency remains the same across different power outputs because the mass flow rate scales proportionally, keeping the thermal inputs and outputs unchanged. Therefore, only the 50 MWe case is considered in this section as the reference. As expected, higher turbine inlet temperatures result in higher thermal efficiencies, in line with the Carnot principle since the temperature difference between the heat source and the sink is increased. Similarly, increasing the effectiveness of both the HTR and LTR enhances the cycle efficiency by allowing greater heat recovery and reducing exergy destruction, which is a normal and anticipated outcome. However, at lower turbine inlet temperatures, feasible efficiency values are only observed for relatively modest combinations of HTR and LTR effectiveness. This constraint arises because the overall temperature levels of the cycle are lower, leaving only a small temperature difference to drive heat transfer within the recuperators. When the effectiveness of the recuperators is increased under these conditions, the cold streams are pushed too close to the hot-stream temperatures, causing the hot-side outlet to fall below the cold-side inlet (i.e., $T_2 > T_6$ in the LTR or $T_7 > T_9$ in the HTR). Such a condition would require heat to flow from a colder stream to a hotter stream, which contradicts the second law of thermodynamics. As a result, only lower recuperator effectiveness values remain feasible at low turbine inlet temperatures, whereas higher turbine inlet temperatures provide sufficient thermal margin to sustain higher HTR and LTR effectiveness without violating the natural direction of heat transfer.

7.1 CAPEX analysis of sCO₂ Brayton cycle across various HTR, LTR, power outputs and turbine inlet temperature.

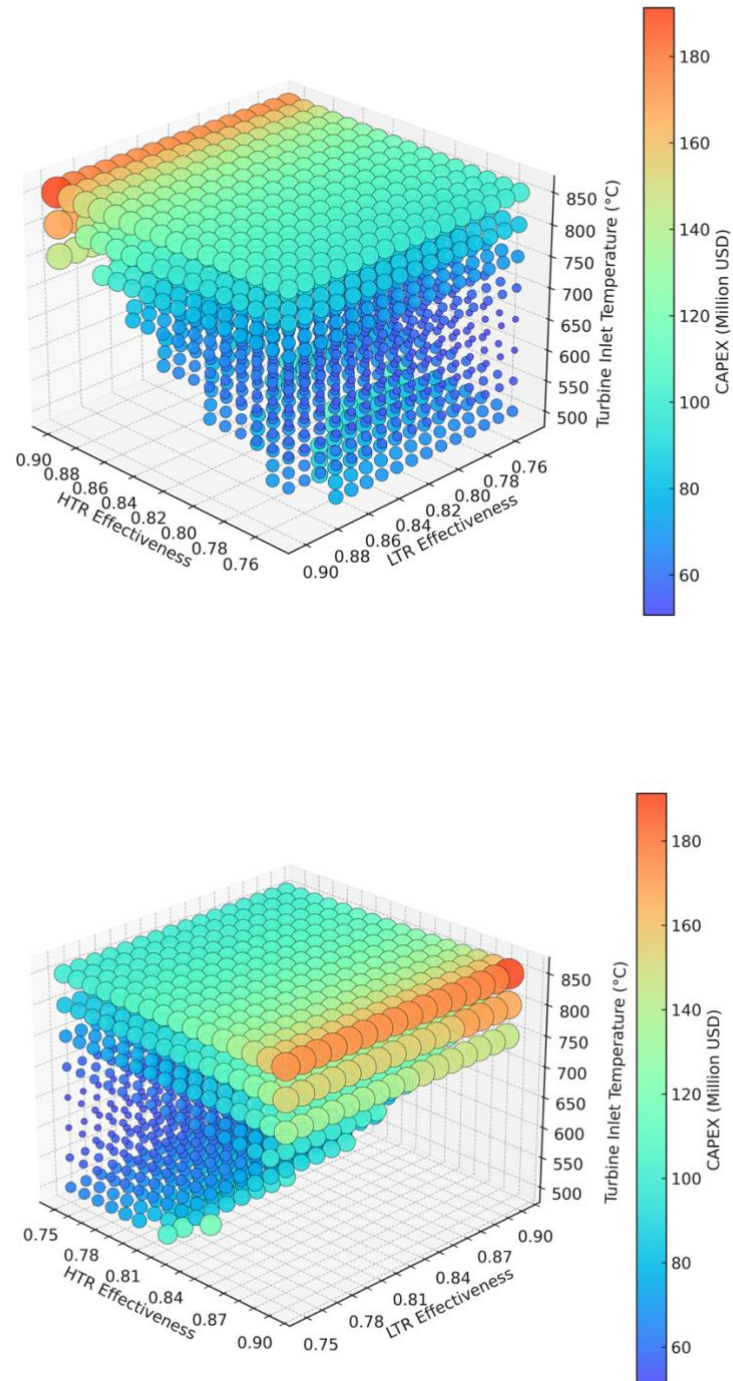


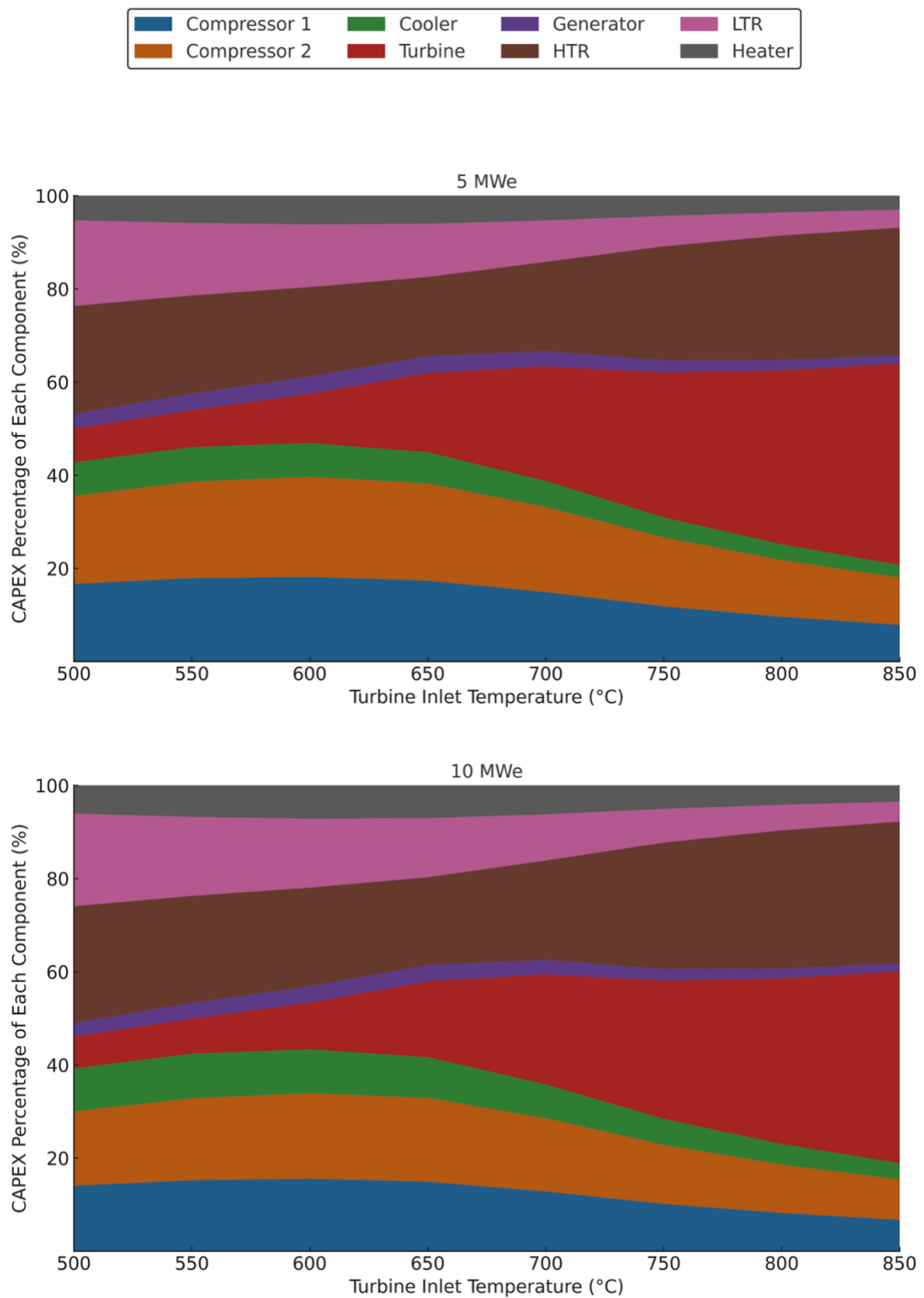
Fig. 7.2 CAPEX relative to the combinations of HTR and LTR effectiveness and turbine temperature, T_3 at different angles at $50MW_{e,net}$

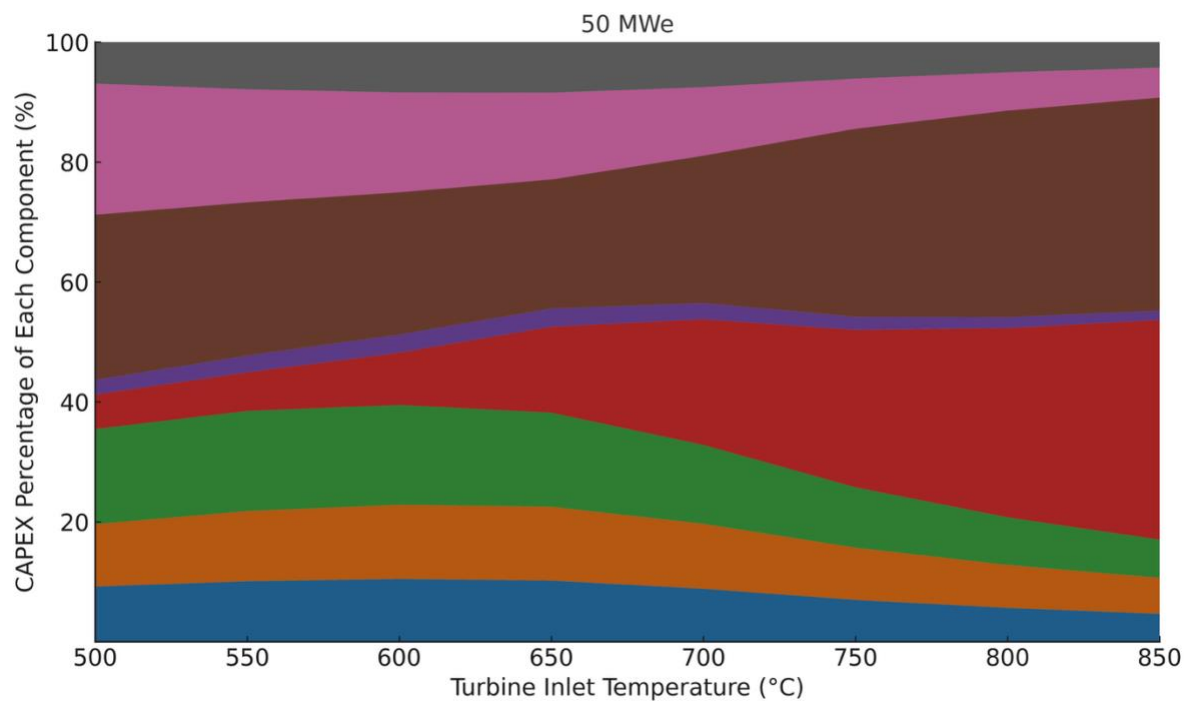
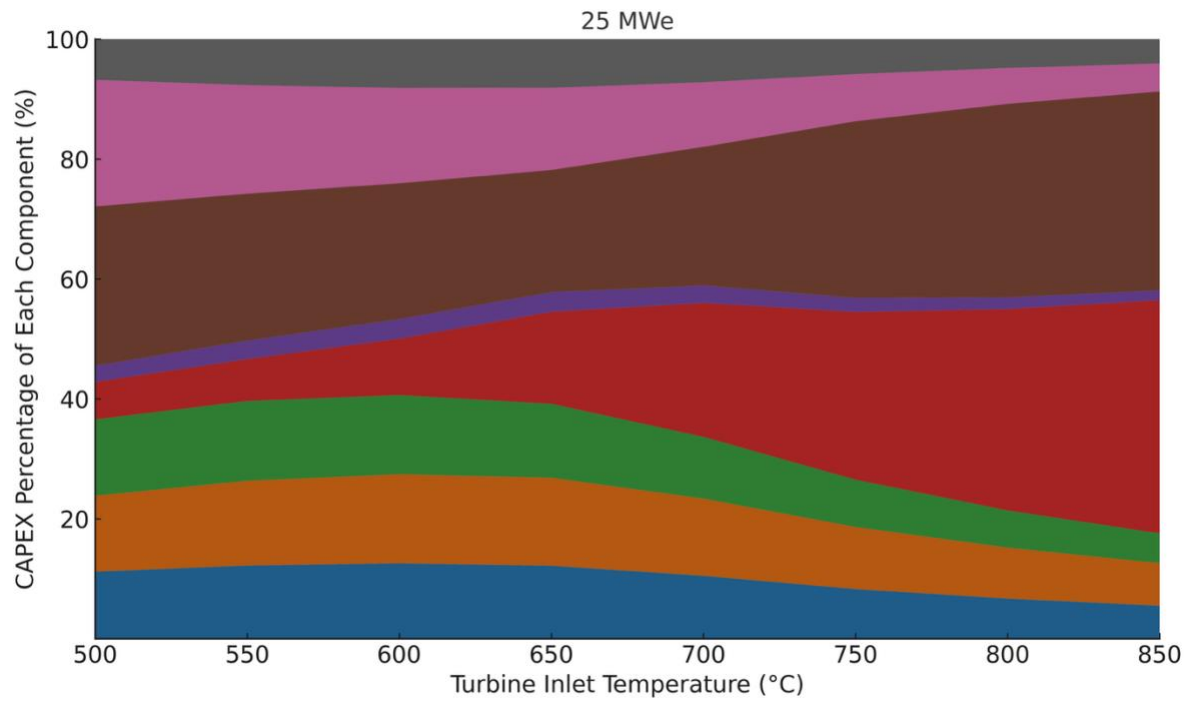
Fig. 7.2 shows the relationship between CAPEX and the combinations of HTR and LTR effectiveness with turbine inlet temperature (T_3), complementing Fig. 7.1 and highlighting the deviation from the previous efficiency-focused section. In this and subsequent few sections, CAPEX is employed as the primary economic indicator rather than LCOE, since OPEX and other cost factors are typically assumed constant across all cases, making CAPEX sufficient for meaningful comparison.

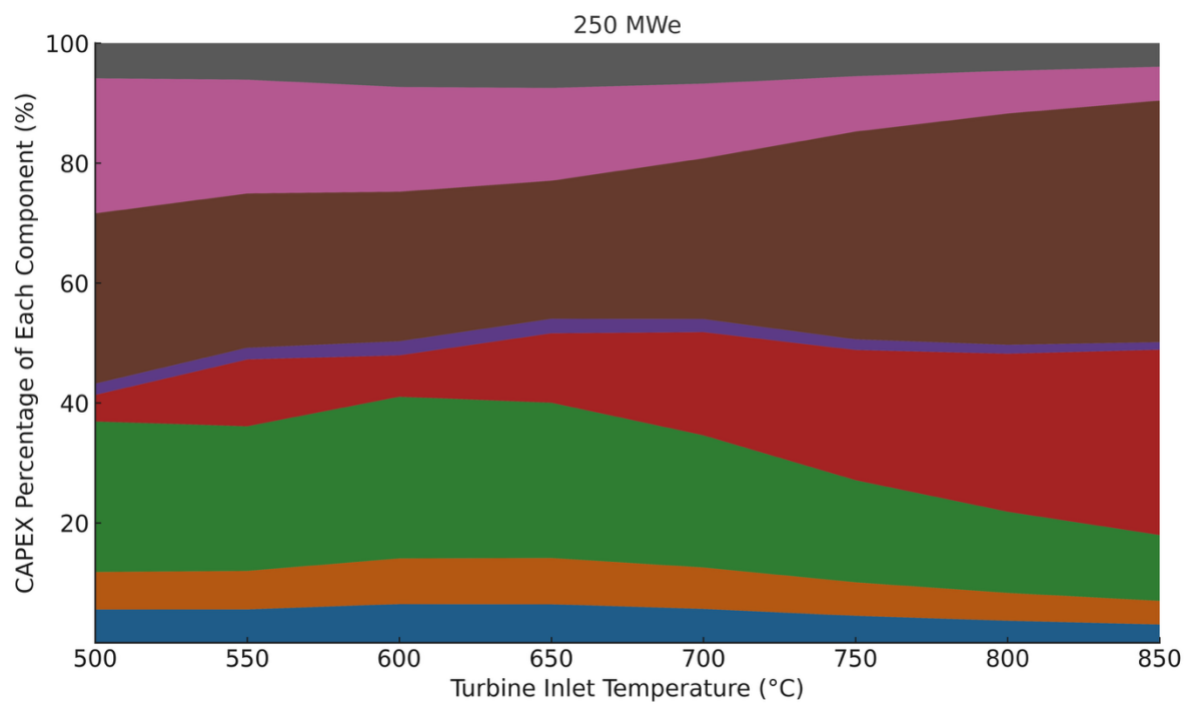
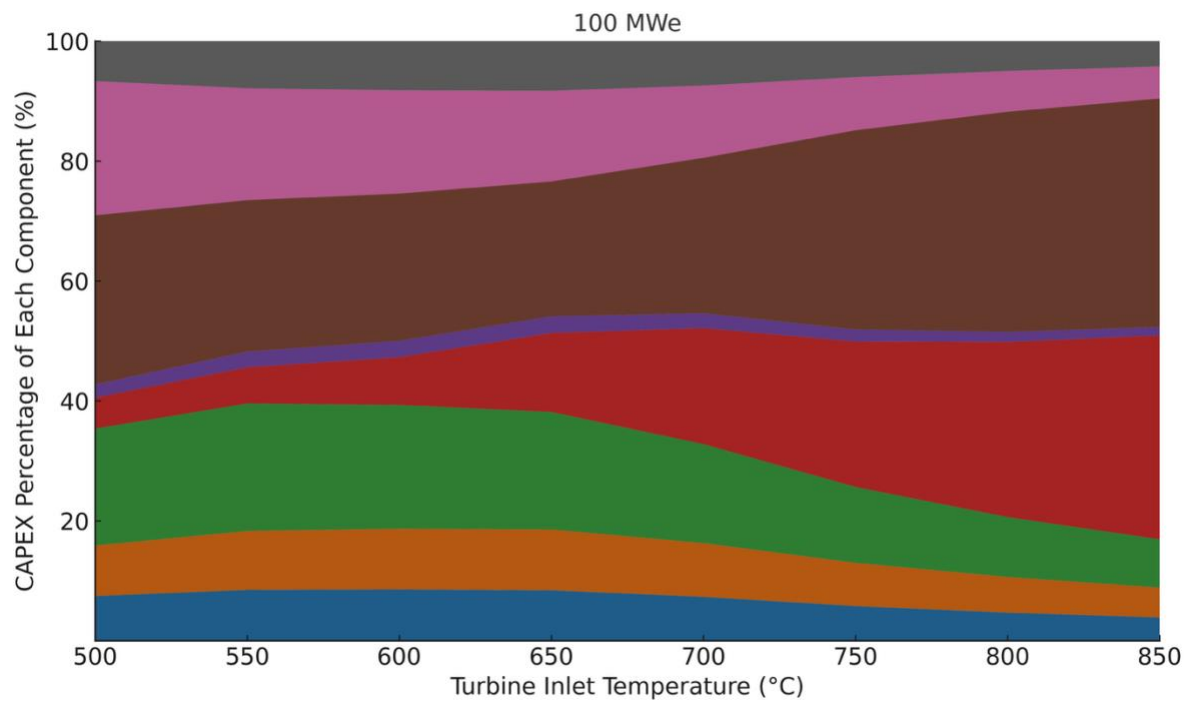
While increasing the effectiveness of the HTR and LTR together with higher T_3 generally leads to improved thermal efficiency, the cost does not follow the same trend. Instead, higher HTR and LTR effectiveness results in higher CAPEX, largely due to the significant cost associated with both recuperators (see section 7.2 and 7.3). Moreover, increasing T_3 beyond a certain point does not translate into better economic efficiency. An optimal window is observed between 600–650 °C, with 600 °C being the most favourable for lower power outputs (5–25 MWe), while 650 °C is optimal for medium to large-scale outputs (≥ 50 MWe). Beyond 650 °C, the economic efficiency begins to deteriorate, indicating diminishing returns at higher turbine inlet temperature, despite the positive thermal efficiency increment. For the next chapters, the combination of 0.75 of HTR and LTR were considered due to its economic efficiency as discussed above.

The economic impact of each component on system viability varies with T_3 . For the recuperators, increased efficiency raises their heat load and cost more than it reduces costs elsewhere. While higher effectiveness lowers the system's mass flow rate and reduces other component costs, the additional expense of the HTR and LTR dominates, making them the most economically influential elements in the sCO₂ Brayton cycle. The main reason for the presence of optimal T_3 is largely due to the disproportionate increase in turbine and HTR costs as turbine inlet temperature rises. This will be detailed in the next section.

7.2 Cost Breakdown by component across various power outputs







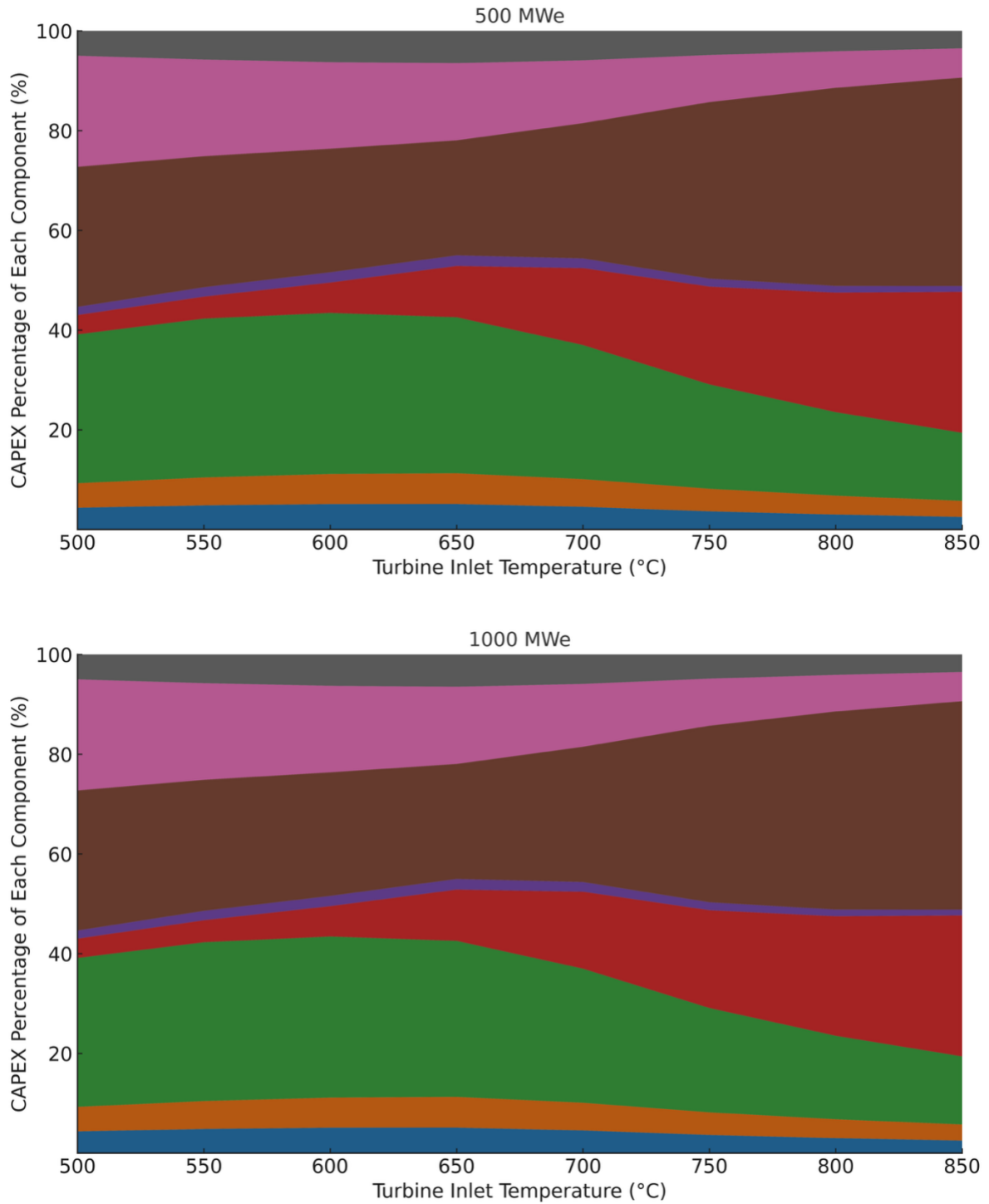


Fig. 7.3 Component cost distribution of the sCO₂ recompression Brayton cycle (1000 MWe) across different turbine inlet temperatures and power outputs.

Fig. 7.3 shows the percentage cost contribution of each component in the sCO₂ recompression Brayton cycle across different turbine inlet temperatures and power outputs. The heater is included in the cost mapping for completeness, due to the typical Brayton cycle

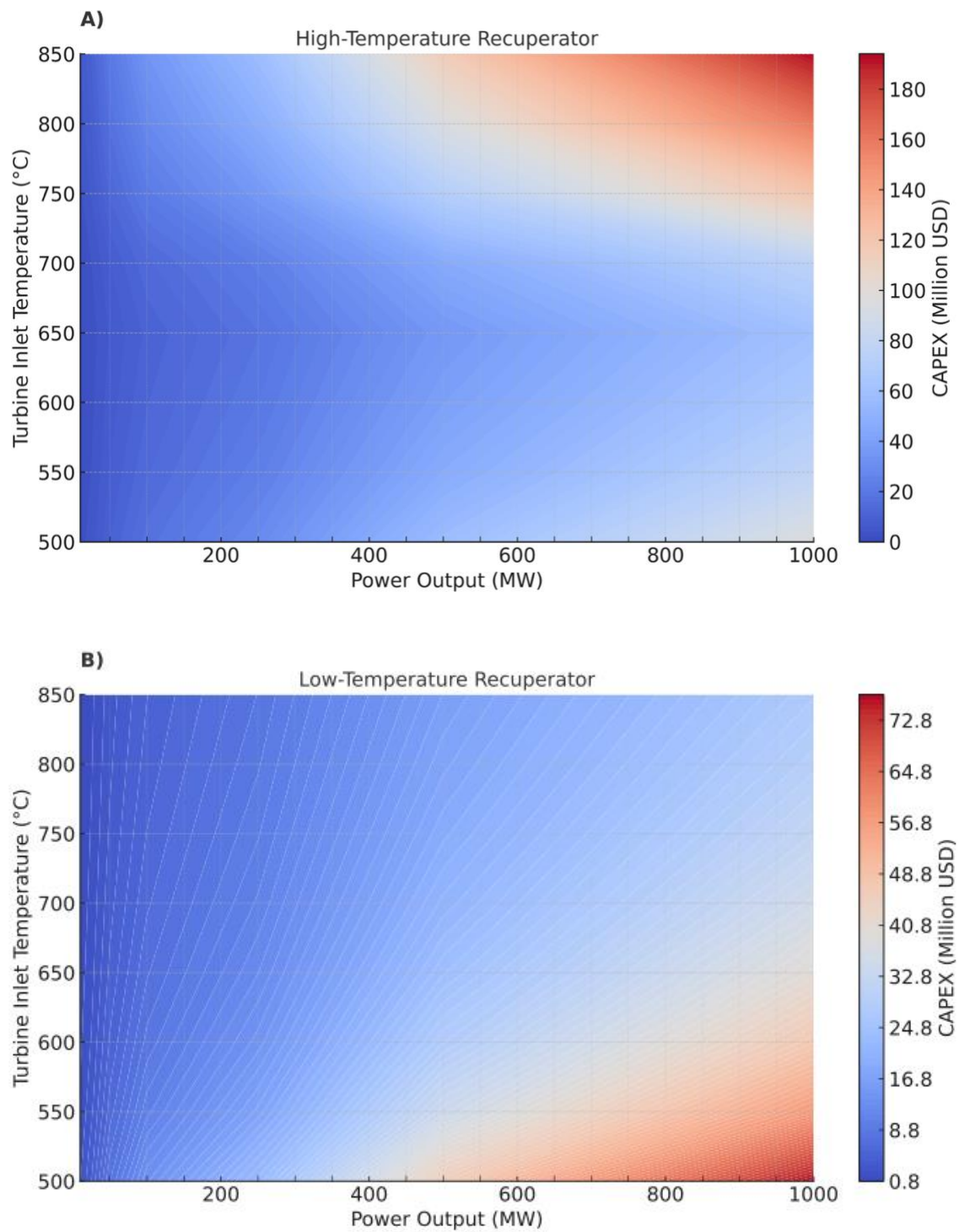
configuration of having heater. However, since the present work considers a direct configuration, its cost is excluded from the final LCOE calculation later on. The results reveal a consistent redistribution of capital costs within the sCO₂ recompression Brayton cycle as the turbine inlet temperature (T_3) increases. At lower T_3 values, a significant portion of expenditure is concentrated on the compressors and the low-temperature recuperator (LTR), reflecting the higher proportion of compression work and cold-end heat recovery in determining overall cycle performance, due to lower thermal efficiency and thus more waste heat recovery. As T_3 rises, however, the relative cost share of these cold-end components diminishes. This is driven by two main effects. First, the increase in turbine work at higher temperatures reduces the relative importance of compression; and second, the narrowing temperature differences at the cold end lower the effectiveness and contribution of the LTR. In contrast, both the turbine and the high-temperature recuperator (HTR) grow steadily in significance, with increment of T_3 leads to the turbine requires advanced materials (as detailed in section 3.6.3) to withstand elevated operating conditions, while the HTR demands larger surface areas to accommodate greater hot-side heat transfer. The cooler also decreases modestly in share with increasing T_3 , consistent with reduced heat rejection requirements. The generator remains a consistently minor contributor across all cases, as its cost scales primarily with electrical capacity rather than thermal conditions.

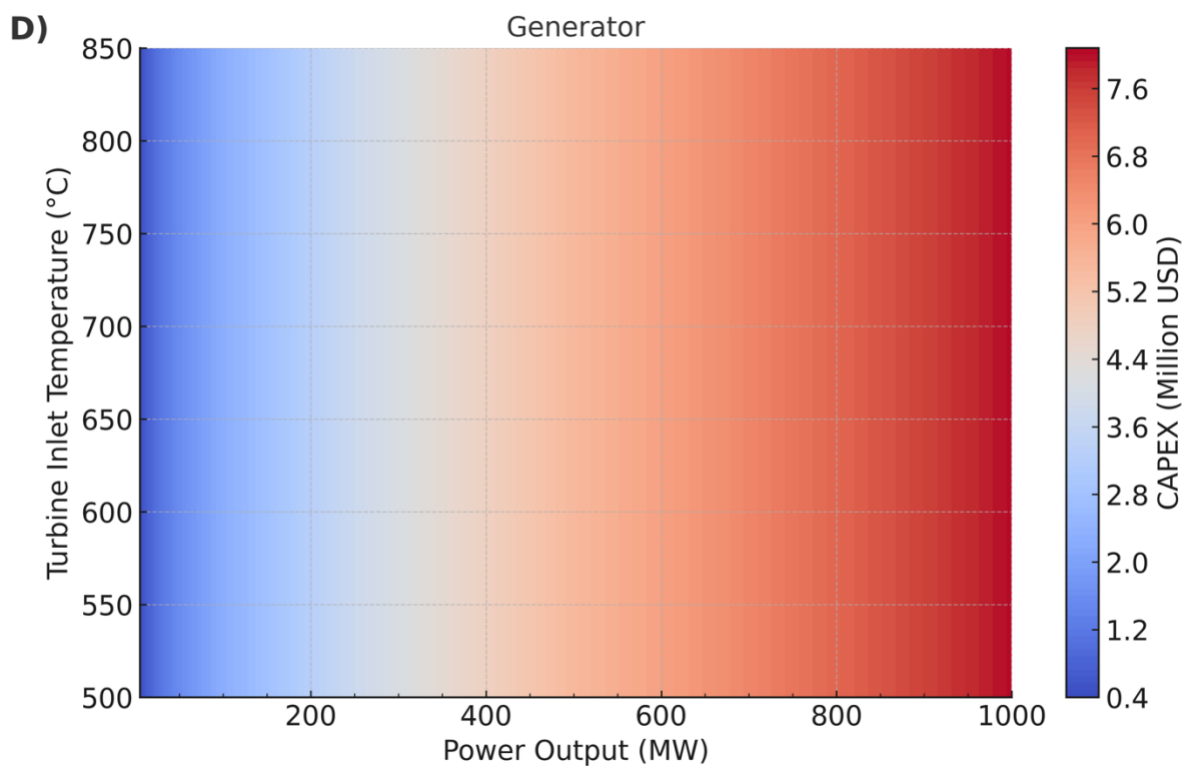
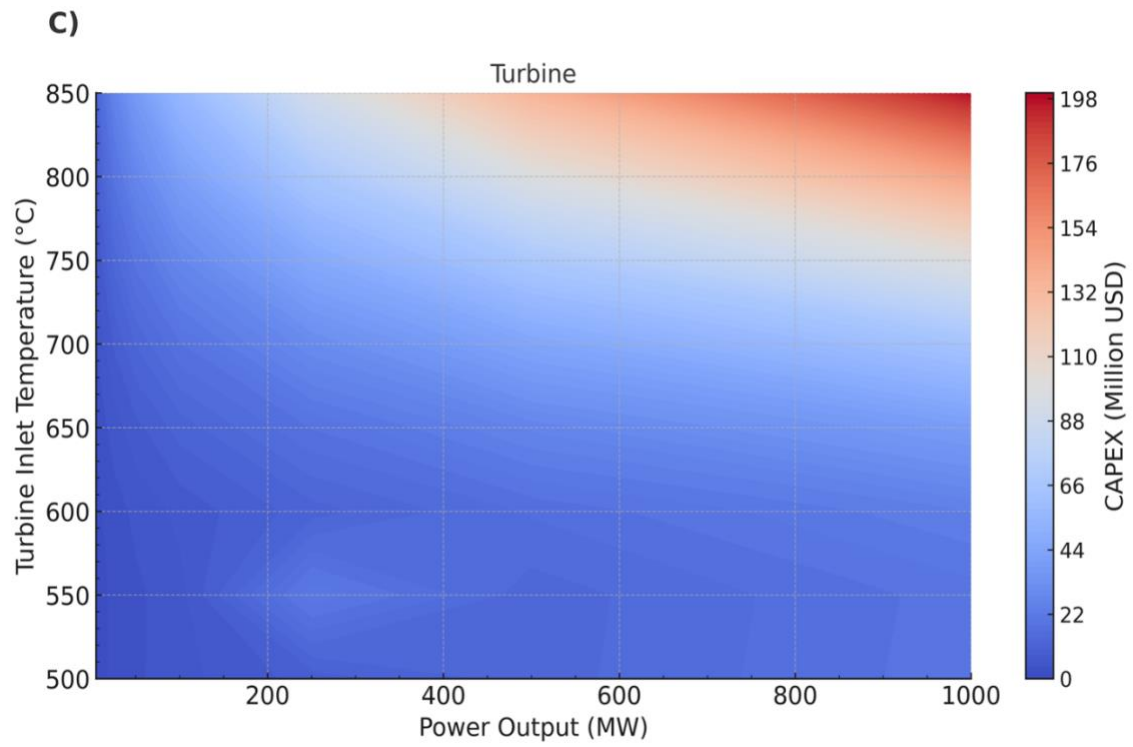
The influence of plant size further reshapes these trends by amplifying or diminishing the relative cost shares of individual components. In smaller plants (5–25 MWe), the compressors dominate the cost structure, owing to scaling penalties and the relatively high share of compression work in the cycle. As plant capacity increases, the proportional impact of the compressors reduces, while the turbine and HTR increasingly govern the cost structure. For medium-scale plants (50–250 MWe), cost distribution becomes more balanced, with the cooler

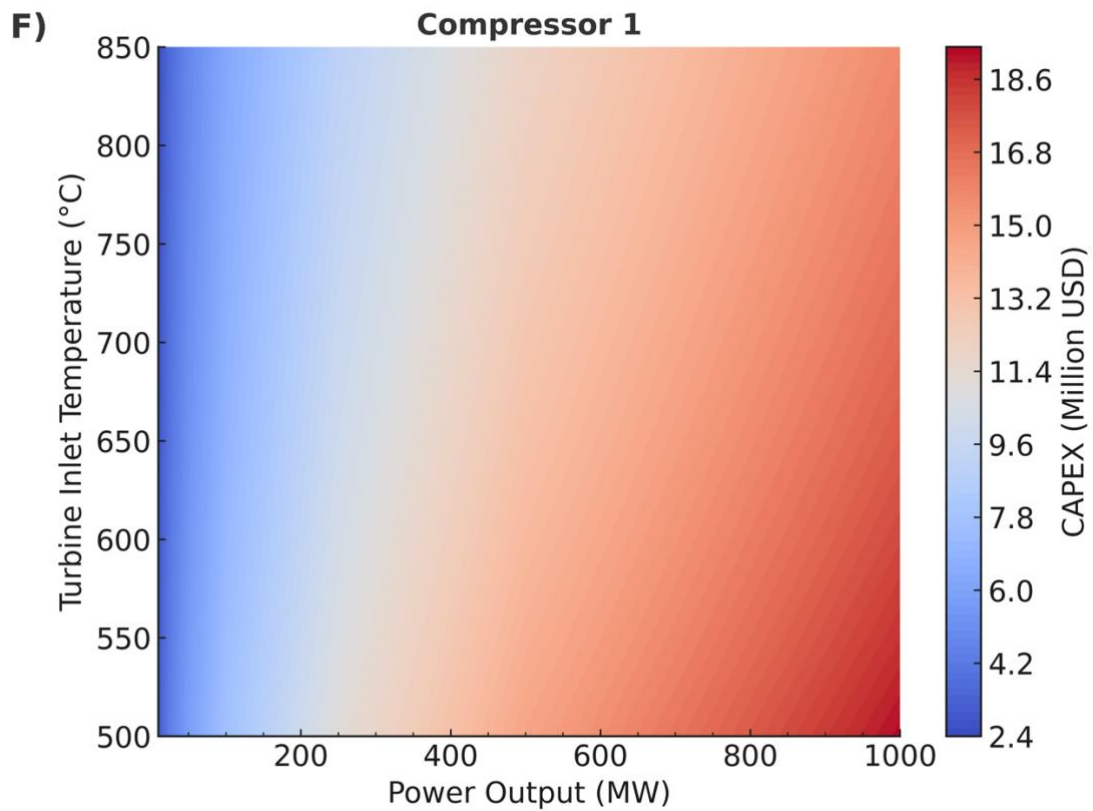
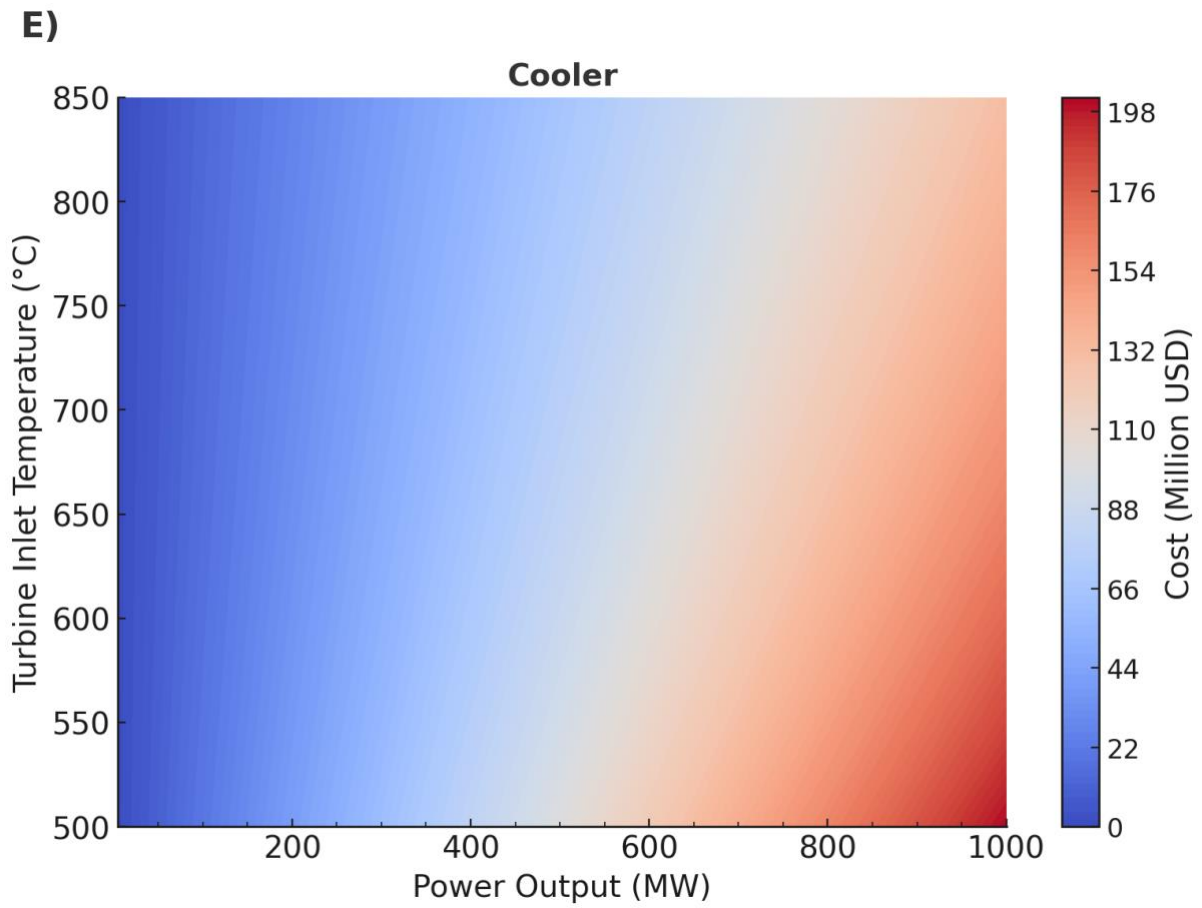
making a modest contribution at intermediate TITs. At very large scales (≥ 500 MWe), the HTR and turbine together dominate the capital cost, while compressors, cooler, and generator become secondary. The HTR, in particular, grows steadily with both T_3 and power output, reflecting its central role in enabling high efficiency at large scale.

Taken together, these results demonstrate that the cost drivers of the sCO₂ recompression Brayton cycle depend strongly on both T_3 and plant size. At low T_3 and small scales, the cycle is constrained by compression and cooling hardware, whereas at high T_3 and large scales, the technical and economic challenges concentrate in the turbine and HTR. This highlights the importance of tailoring cost-optimisation strategies, for small systems, improvements in compressor efficiency and compact cooling are most impactful, while for utility-scale high-temperature systems, advances in turbine materials and large-surface recuperator designs are the key to reducing overall costs.

7.3 Cost of each component in sCO₂ Brayton cycle







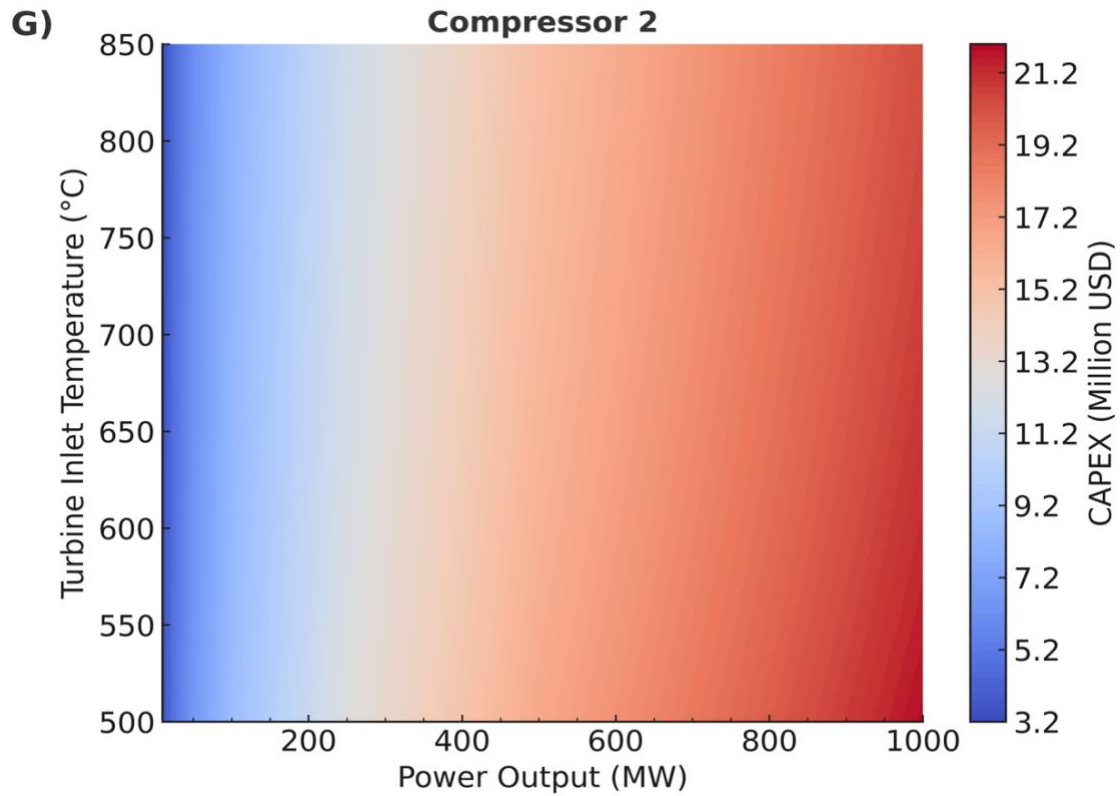


Fig. 7.4 CAPEX for each component for $s\text{CO}_2$ Brayton cycle relative to turbine inlet temperature (T_3) and power outputs

Fig. 7.4 shows the CAPEX values for each of the component in a recompression $s\text{CO}_2$ Brayton cycle. Firstly, it can be seen that HTR and LTR are the two most expensive components. However, the relationship between the two is the opposite at larger power outputs. For HTR, capex is higher at higher turbine inlet temperatures (T_3) whereas LTR is highest at lower T_3 . This complements the observations seen in section 7.2. The reason behind this is that the efficiency is lower at lower T_3 as shown in section 0, which leads to more heat recovery by the LTR at low T_3 whereas at higher T_3 , two phenomena happen. Firstly, higher cycle efficiency lead to the opposite effect for LTR and, the increment in T_3 leads to higher temperature operating in HTR and thus, increasing its cost due to higher material needed to withstand it, as shown by the equations, correlations and its values in section 3.6.3. It is also noteworthy that HTR and LTR cost can reached as high as around 180 and 73 million USD, which cumulatively surpassed other components by a wide margin. Turbine and generator

show predictable behaviour as the cost increases with higher power output and T_3 . At both the lowest and highest power outputs, the cooler represents the most expensive component, as these operating conditions correspond to reduced cycle efficiency and consequently greater quantities of heat that must be rejected. A similar trend is observed for the compressors. However, the second compressor is notably more costly than the first. This is attributable to the higher fluid density at its inlet due to the fluid's relatively longer distance to its critical point relative to the first compressor, which increases the energy required for compression, despite the reduced mass flow rate of the working fluid at this stage. It is worth mentioning that while recompression cycle leads to higher heat utilisation, it cost significantly more than simple cycle based on current cost correlation. Turbine, cooler and HTR's costs significantly influence the system at higher T_3 and power output with up to 180-190 million USD.

7.4 CAPEX relationship between turbine inlet temperature and power output

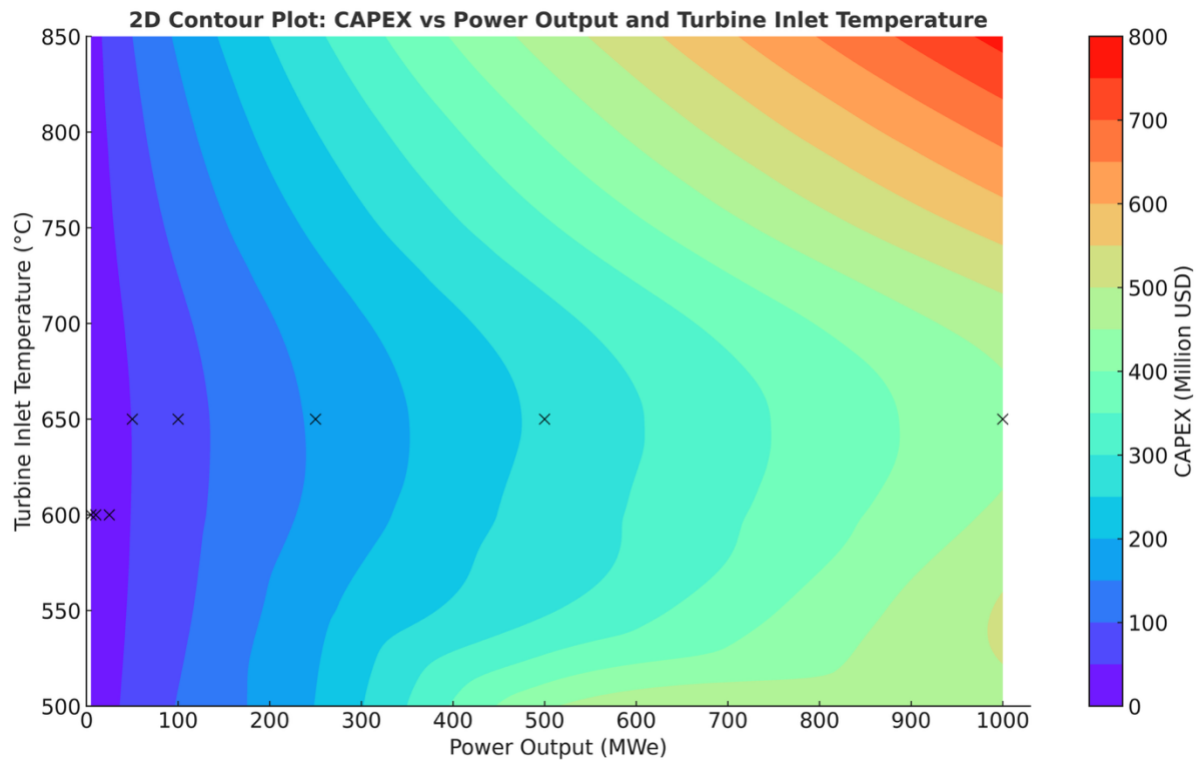


Fig. 7.5 2D colour contour plot showing the variation of CAPEX with power output and turbine inlet temperature for the sCO₂ Brayton cycle, where 'x' is denoted for the most optimal point for each combination.

Fig. 7.5 illustrates the variation of CAPEX with turbine inlet temperature (T_3) and power output for the sCO₂ Brayton cycle, with the 'x' markers denoting the optimum cost points for each combination. A clear trend can be observed, CAPEX generally increases with both power output and turbine inlet temperature. However, the minimum CAPEX does not occur at the lowest T_3 values. This behaviour arises because, although raising T_3 improves thermal efficiency and reduces the size of some components, the costs of the high-temperature recuperator (HTR) and turbine rise sharply at elevated T_3 , outweighing the savings from other equipment. As a result, the overall system cost follows a U-shaped profile with respect to T_3 , with the optimal points clustering in the 600–650 °C range.

At low power outputs, the optimum CAPEX is observed around 600 °C for plants up to around 25 MWe, before shifting to around 650 °C from 50 MWe onwards. This reflects the balance between equipment sizing and high-temperature cost penalties. For medium- and high-power plants, the optimal turbine inlet temperature consistently remains near 650 °C, with no further cost benefit from pushing towards ultra-high temperatures. At larger capacities, CAPEX rises more steeply with T_3 due to the compounded cost of the turbine, HTR, and cooler, whereby the turbine and HTR must be designed for higher thermal stresses, while the cooler must handle greater heat rejection loads. Consequently, even at 1000 MWe, the lowest CAPEX point remains at around 650 °C, indicating that the widely proposed 700 °C target for third-generation CSP systems has not yet translated into a cost-optimal solution under the present cost correlations.

The contour distribution further emphasises this trend. The lower-left region of the plot shows smooth gradients, indicating relatively stable CAPEX behaviour at low-to-moderate scales. In contrast, the top-right region exhibits steep gradients, reflecting the rapid CAPEX escalation as both power output and turbine inlet temperature rise. These results highlight that, within the current techno-economic framework, the most cost-effective operating window for the sCO₂ Brayton cycle lies between 600–650 °C across a wide range of power capacities.

7.5 Parametric analysis of ORC utilising duality method.

This section presents the parametric analysis of the duality method, in which the approach temperatures are varied while maintaining a pinch point of 5 °C, a value commonly adopted in the literature. The purpose of this analysis is to demonstrate that the duality method provides deeper insights into system behaviour, reinforcing the evidence presented in Section 6.1.1 that the conventional energy balance method is inadequate for meaningful analysis and design. For

consistency with the preceding results, the turbine inlet temperatures (T_3) used here are set to 600 °C for sCO₂ net output of up to 25 MW_{e,net} and 650 °C for systems above this capacity.

7.5.1 Levelised cost of electricity (LCOE) of ORC.

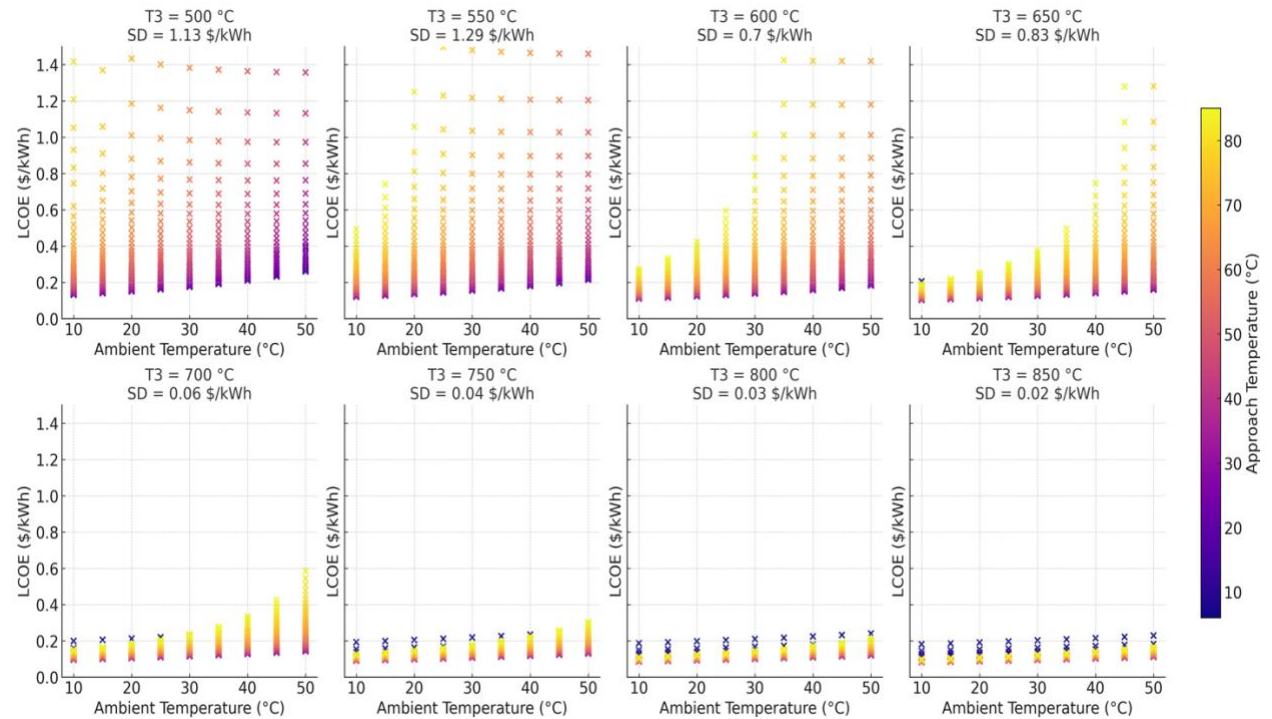


Fig. 7.6 The LCOE of ORC as the bottoming cycle across various turbine inlet temperature and ambient temperature with topping cycle as 50MWe as the reference case.

Fig. 7.6 exhibits the variation in LCOE of the ORC bottoming cycle across different turbine inlet temperatures (T_3), approach temperatures, and ambient conditions, with the topping cycle fixed at 50 MWe as the reference case. Three dominant trends are evident from the results. First, the effect of approach temperature follows a non-linear pattern. At very small approaches (e.g. 6 °C), the LCOE is relatively high, but increasing the approach to a moderate value reduces LCOE substantially. For example, at 600 °C T_3 and 30 °C ambient, LCOE decreases from 0.25 \$/kWh at 6 °C approach to 0.14 \$/kWh at 20 °C. Beyond this optimum,

further increases in approach temperature drive costs upward, with LCOE reaching over 1.0 \$/kWh at 85 °C. Notably, the approach that minimises LCOE shifts upwards as T_3 increases, from around 19 °C at 500 °C to 35 °C at 850 °C under 10 °C ambient conditions, reflecting the capacity of higher turbine inlet temperatures to sustain efficiency even with larger approach values.

Second, increasing the turbine inlet temperature of the topping cycle produces a steady and substantial reduction in LCOE. At 30 °C ambient, the minimum achievable LCOE decreases from around 0.18 \$/kWh at 500 °C to below 0.10 \$/kWh at 850 °C. This improvement is not only in the average value but also in stability, the standard deviation of LCOE across all conditions drops from over 1.1 \$/kWh at 500–550 °C to less than 0.03 \$/kWh once T_3 exceeds 750 °C. This indicates that high turbine inlet temperatures not only reduce costs but also make the cycle far less sensitive to operating variations.

Finally, higher ambient temperatures consistently increase LCOE, but the magnitude of this penalty diminishes at higher T_3 . With a 6 °C approach, raising ambient from 10 °C to 50 °C increases LCOE by about 0.16 \$/kWh at 500 °C, but by less than 0.05 \$/kWh at 850 °C. When the approach is re-optimised, the increase is even more modest, with the 10–50 °C rise in ambient adding only around 0.03 \$/kWh at 850 °C compared to ~0.13 \$/kWh at 500 °C. This demonstrates that high T_3 operation not only improves the mean performance but also mitigates the adverse effect of warmer climates. This is due to the fact that higher T_3 lead to higher temperature input into the bottoming cycle at T_6 (refer Fig. 3.2 for the whole system's figure) and thus the temperature difference between the hot reservoir and cold sink increases, leading to increase efficiency.

Taken together, these results emphasise that the techno-economic performance of the ORC bottoming cycle is governed by a balance between approach selection, turbine inlet temperature, and ambient conditions. Moderate approaches combined with high turbine inlet temperatures yield the lowest and most stable LCOE, while also reducing the vulnerability of the system to external temperature fluctuations.

7.5.2 Thermal efficiency of ORC relative to Carnot cycle efficiency.

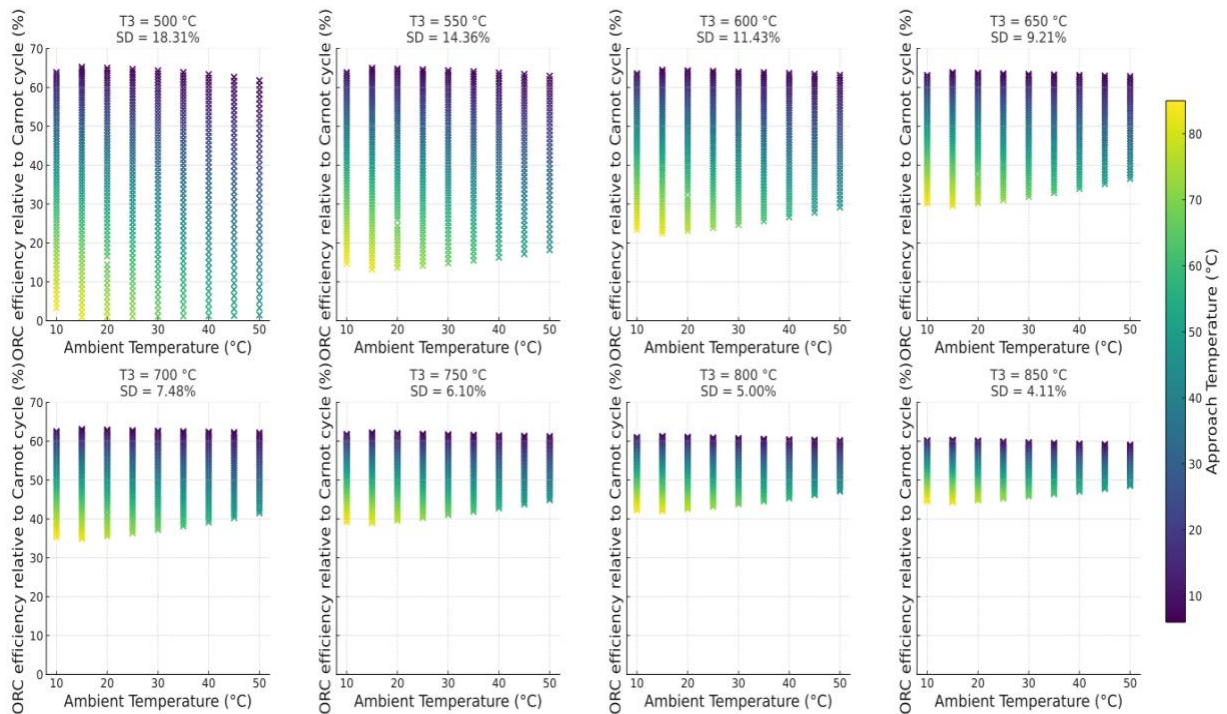


Fig. 7.7 The efficiency of ORC relative to Carnot cycle as the bottoming cycle across various turbine inlet temperature and ambient temperature with topping cycle as 50MWe as the reference case.

Fig. 7.7 shows the efficiency of ORC relative to Carnot cycle at a given ambient temperature across the examined turbine inlet temperature. The relationship is straightforward, as the turbine inlet temperature (T₃) increases, the efficiency of the cycle also increases. This trend is consistent across the data, showing that higher T₃ allows for better utilisation of the

thermal energy input. At the same time, the standard deviation decreases with increasing T_3 values, meaning that the efficiency results become more stable and less sensitive to variations at higher operating temperatures. For the relationships between approach temperature and efficiency, the results show that efficiency decreases as the approach temperature increases. This decline is more pronounced at higher ambient temperatures, while at higher turbine inlet temperatures the efficiency becomes less sensitive to changes in approach temperature. Nonetheless, the overall trend remains consistent, larger approach temperatures always lead to lower cycle efficiency. This shows that efficiency is not correlated to the optimal values of LCOE and W_{net} , something that the typical method does not show. The relationship is shown in Table 7.1 which will be discussed later.

Additionally, the results indicate that increasing ambient temperature leads to a clear reduction in cycle efficiency. Across all turbine inlet temperatures (T_3), the efficiency relative to the Carnot cycle decreases as the ambient temperature rises from 10 °C to 50 °C. This trend is expected, since a higher ambient temperature reduces the effective temperature difference between the heat source and sink, thereby lowering the maximum achievable efficiency. The consistent downward slope across all cases highlights the sensitivity of the cycle performance to changes in ambient conditions.

7.5.3 The net-work output (W_{net}) of ORC

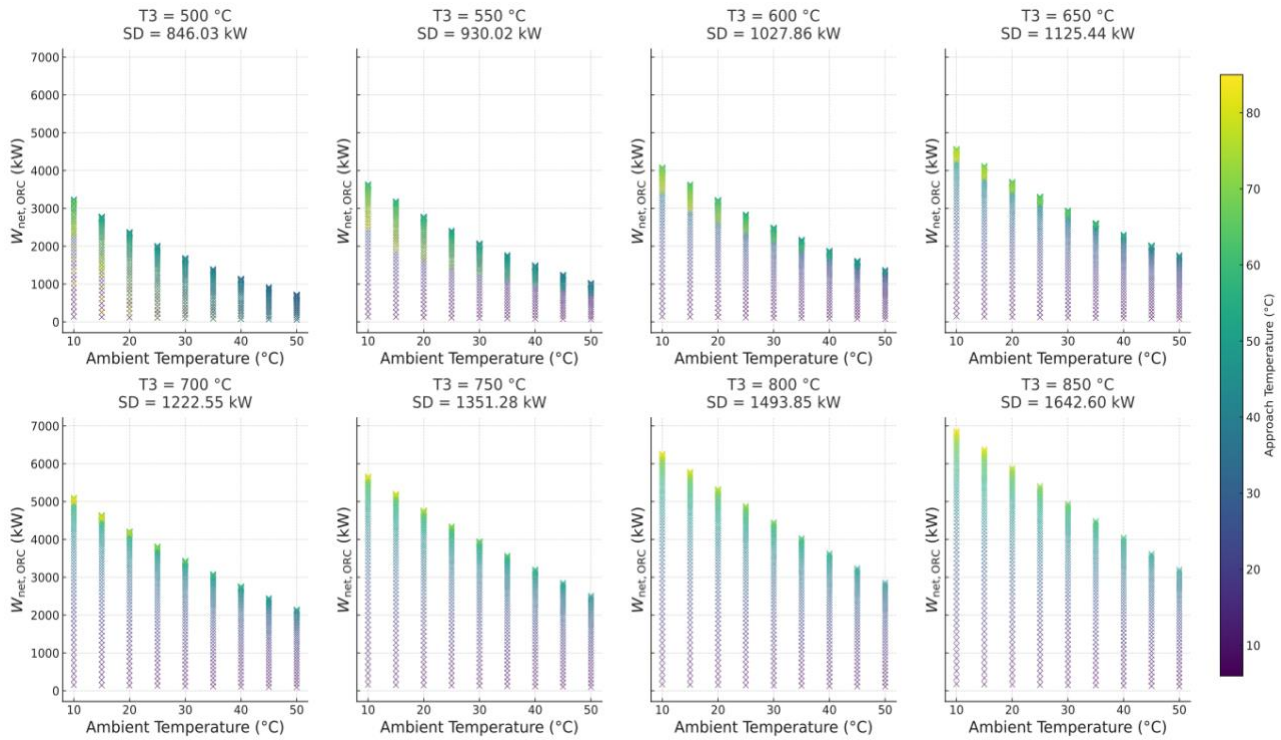


Fig. 7.8 The W_{net} of ORC relative to Carnot cycle as the bottoming cycle across various turbine inlet temperature and ambient temperature with topping cycle as 50MW_e as the reference case.

Fig. 7.8 illustrates the effect of ambient temperature, approach temperature, and turbine inlet temperature (T_3) on the net power output, W_{net} . Three main observations can be made from the figure. Firstly, W_{net} consistently decreases as ambient temperature increases. This occurs because a warmer environment reduces the temperature difference available for heat rejection, which lowers the overall cycle efficiency. The decline is more severe at lower T_3 , with the slope gradually flattening at higher T_3 , showing that elevated turbine inlet conditions help to lessen, but not eliminate, the negative impact of high ambient temperature.

Secondly, an increase in approach temperature leads to higher W_{net} . This effect is modest at low T_3 , but becomes progressively stronger as T_3 increases. For instance, while the rise in

W_{net} is limited at $T_3 = 500\text{ }^{\circ}\text{C}$, it becomes much more pronounced at $T_3 = 850\text{ }^{\circ}\text{C}$, indicating that approach temperature and T_3 act together to improve cycle performance.

Finally, T_3 has the most dominant influence on W_{net} . At very low approach temperatures, raising T_3 produces only a small improvement. However, at moderate to high approach temperatures, the gain is substantial, with net output rising steeply as T_3 increases. This highlights the strong positive interaction between T_3 and approach temperature, which together drive the most significant improvements in cycle performance.

Additionally, Table 7.1 shows that the optimal efficiency, W_{net} and LCOE does not belong to the same sets of combinations, proving the duality method is better for optimal design solutions than the energy balance method, based on the results discussed on sections above. The key takeaway from the table is that each category achieves its best performance at a different approach temperature.

Table 7.1 The comparison between each the proposed duality method and typical energy balance method at 50Mwe at selected ambient temperature levels.

Tamb (°C)		% of Carnot	W_{net} (kWe)	LCOE (USD/kWh)
10	Approach Temperature (°C)	6	69	26
	Duality method	65.43	4566	0.103
	Energy balance	60.59	1352	0.1106
30	Approach Temperature (°C)	6	57	36
	Duality method	64.3	2941	0.1258
	Energy balance	57.34	1042	0.1319
50	Approach Temperature (°C)	6	45	20
	Duality method	61.76	1750	0.1601
	Energy balance	44.17	761.6	0.1637

7.6 System's levelised cost of electricity (LCOE) and levelised cost of water (LCOW) analysis.

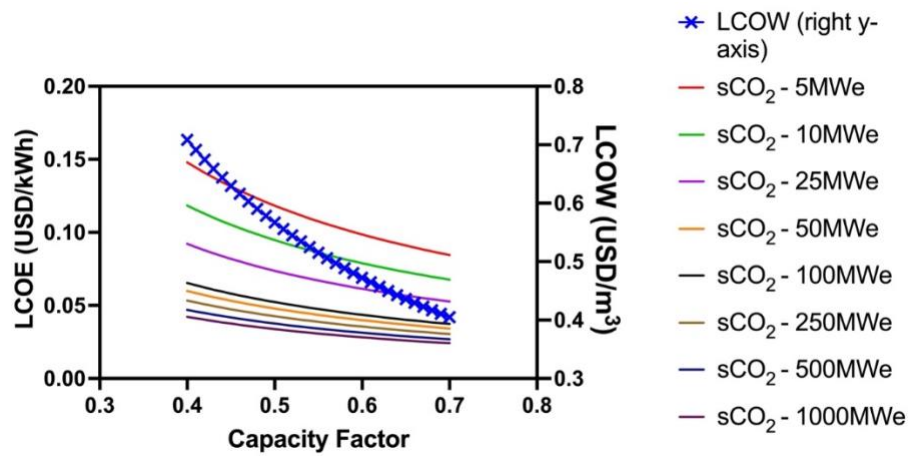


Fig. 7.9 Variation of levelised cost of electricity (LCOE) and levelised cost of water (LCOW) with capacity factor for different sCO₂ Brayton cycle power outputs, highlighting the influence of thermal storage cost on system performance at ambient temperature of 25°C

Fig. 7.9 illustrates the variation of levelised cost of electricity (LCOE) and levelised cost of water (LCOW) with capacity factor across different sCO₂ Brayton cycle power outputs. 25°C was chosen here because it is in line with ISO 9806:2017, an international standard for solar thermal test methods. Additionally, the difference between ambient temperature has been discussed in previous sections, and the trends are similar for all ambient temperatures for this section, thus 25°C is taken as the representative. The capacity factor of a solar thermal power plant represents the ratio of the actual electricity generated over a given period to the maximum possible electricity it could have produced if it ran at full rated capacity all the time. It accounts for varying DNI because the sun's intensity changes with weather, time of day, and season, throughout the year, meaning the plant cannot always operate at its peak output. The range of capacity factor here varies between 0.4 to 0.7 as both of them represent the realistic minimum and maximum of a 3rd generation CSP system with thermal storage.

The first observation is that both indicators decline as the capacity factor increases, reflecting enhanced cost-effectiveness at higher utilisation. The reduction in LCOE is most pronounced between 0.4 and 0.5 capacity factor, after which a plateauing behaviour emerges, indicating diminishing marginal gains at higher utilisation, particularly for larger system sizes. In contrast, LCOW decreases more steadily across the range, falling from 0.7084 USD/m³ at 0.4 capacity factor to 0.4048 USD/m³ at 0.7, with no clear plateau but a gradual slowing in the rate of decline.

At smaller scales, such as the 5 MWe unit, electricity generation remains relatively expensive due to limited economies of scale, with costs well above the DOE third-generation CSP benchmark. Nonetheless, the results show that the 5 MWe system only begins to fall below the 0.1 USD/kWh threshold at a capacity factor of 0.6, highlighting the challenges of operating such small systems competitively. By contrast, the 50 MWe scale meets the 0.05 USD/kWh target as early as a 0.48 capacity factor, and larger systems consistently achieve or surpass this benchmark across most of the studied range. The maximum LCOE occurs for the smallest system (5 MWe) at 0.4 capacity factor with 0.1479 USD/kWh, while the minimum value is achieved by the 1000 MWe system at 0.7 capacity factor with 0.02417 USD/kWh.

Taken together, these findings demonstrate that while larger plants operating at high-capacity factors can reliably meet the 0.05 USD/kWh target and achieve water costs within or below the typical seawater reverse osmosis range (0.5–1.0 USD/m³), smaller systems require very high-capacity factor and thus thermal storage size to cross even the 0.1 USD/kWh threshold. However, LCOW values show competitiveness with reverse osmosis even at modest scales, suggesting that the cogeneration pathway may provide a stronger case for smaller plants than electricity production alone. The reason for this is because unlike reverse osmosis (RO), this system utilises purely on waste heat, with no energy input cost except the pump, which are relatively negligible. Table shows the capacity factor cut off point needed in order to achieve

0.1 USD/kWh which is roughly in line the current CSP, and 0.05USD/kWh, which is the target of 3rd generation CSP.

Table 7.2 Capacity factor cut-off points for achieving CSP cost targets (0.10 USD/kWh and 0.05 USD/kWh).

	5MWe	10MWe	25MWe	50MWe	100MWe	250MWe	500MWe	1000MWe
0.1 USD/kWh	0.6	0.48	0.4	0.4	0.4	0.4	0.4	0.4
0.05 USD/kWh	-	-	-	0.53	0.48	0.43	0.4	0.4

7.7 Total water production capacity

This section explains the total water production of the system over the course of the year with relation to its capacity factor and power outputs at various ambient temperatures.

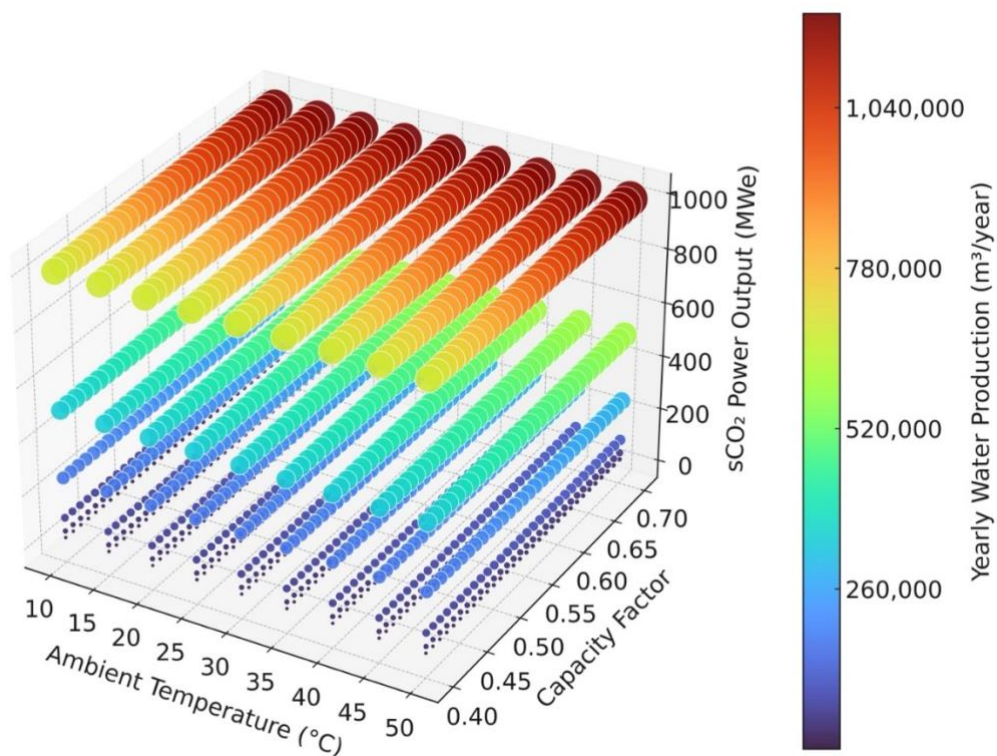


Fig. 7.10 The annual water production capacity of the system under various capacity factor, power outputs and ambient temperature

Fig. 7.10 illustrates the annual water production of the DCMD system as a function of plant output, capacity factor, and ambient temperature. The results demonstrate near-linear scaling with both plant capacity and capacity factor, while ambient temperature has only a secondary effect, shifting cycle efficiency and available waste heat slightly. At 1000 MWe and a capacity factor of 0.7, the system yields approximately 1.04 million m³/year ($\approx 2,850$ m³/day), decreasing to about 0.59 million m³/year ($\approx 1,600$ m³/day) at a capacity factor of 0.4. Scaling further, a 100 MWe plant would deliver around 285 m³/day at CF 0.7 and approximately 163 m³/day at CF 0.4. While these outputs are modest compared to steam-driven cogeneration systems, they are directly comparable with plants that also rely solely on condenser waste heat. For example, the Hongyanhe nuclear power station in China, a roughly 6 GWe facility, produces about 10,080 m³/day of freshwater using MED powered by warmed cooling water (IAEA, 2011). That equates to about 1.7 m³/day per MWe, whereas the present DCMD setup achieves around 2.9 m³/day per MWe at CF 0.7 and around 1.6 m³/day per MWe at CF 0.4. By contrast, plants that draw on high-grade steam achieve far higher values, such as the BN-350 in Kazakhstan (around 135 MWe, up to 80,000 m³/day) (World Nuclear Association, 2025) and the Korean SMART design (around 90 MWe, around 40,000 m³/day) (IAEA, 2002; Kang, 2014). This demonstrates that the DCMD system performs effectively given the constraints of low-grade heat. Unlike steam-extraction systems that sacrifice turbine output, condenser-heat-based configurations preserve full electricity generation. The DCMD approach further benefits from modularity, a compact footprint, and operational flexibility, making it well-suited to applications needing moderate water volumes without compromising power production.

A clear difference emerges between the cost behaviour of electricity and water production when operating conditions deviate from the design point. As the annual capacity factor decreases due to lower or more variable DNI, the levelised cost of electricity rises more sharply

because the reduction in net power output limits the amount of generated electricity over which fixed costs can be distributed. Higher ambient temperatures further intensify this effect by lowering cycle efficiency and reducing turbine output, increasing the cost penalty for electricity under off-design operation. In contrast, the levelised cost of water falls more steadily with increasing utilisation and remains comparatively less sensitive to ambient and DNI variations. This is because desalination continues to make use of available waste heat even when electrical output declines, and its pumping requirements represent a smaller energy share. Consequently, off-design conditions penalise electricity production more strongly than water production, leading to a steeper rise in LCOE relative to LCOW as real operating conditions move away from the optimal design point.

7.8 Conclusion

This chapter translated the system's technical behaviour into cost and performance maps across scale, design settings, and operating conditions. The results showed that larger plant sizes lower the levelised cost of electricity, while turbine inlet temperature does not always reduce cost even when it improves thermodynamic performance. Reported ranges of LCOE of around 0.115–0.025 USD/kWh and LCOW of roughly 0.70–0.40 USD/m³ demonstrated when the system can meet Gen3 cost expectations and when it can compete with typical desalination costs. The overall conclusion is that the best long-term cost outcome arises from balanced design choices that match cycle configuration, heat recovery strategy, and plant scale, rather than maximising a single temperature target. These maps complete the results trajectory by converting the technical insights from earlier chapters into decision-ready guidance, forming the basis for the final thesis-wide recommendations.

7.9 Scalability and Practical Constraints

The techno-economic results presented in this thesis demonstrate that the proposed system is scalable up to utility-scale capacities of 1000 MWe, with lower levelised costs at larger power outputs due to compact turbomachinery and improved cost allocation across major components. This scalability is driven by the high density of supercritical carbon dioxide near and above the critical point, which reduces volumetric flow rates and enables turbomachinery to remain compact even as power output increases. However, practical deployment at such scales is constrained not by thermodynamic limitations but by engineering challenges related to high-temperature materials performance. Operation near or above 700 °C accelerates material degradation mechanisms such as creep, oxidation, and general corrosion, particularly in environments involving long-term exposure to high-pressure supercritical CO₂. These issues affect the durability of turbine components, solar receiver tubing, and heat exchangers, while sealing and bearing reliability also becomes more demanding due to high rotational speeds and pressures. Current industrial efforts are addressing these constraints through the development of advanced nickel-based superalloys, protective corrosion-resistant surface treatments, diffusion-bonded and printed circuit heat exchangers capable of sustaining high pressures, and non-contact solutions such as dry gas seals and magnetic bearings to reduce leakage and wear. As demonstrated by recent sCO₂ Brayton cycle pilot plants operating at elevated pressures and temperatures, these challenges represent engineering maturity and lifetime considerations rather than fundamental barriers to scale, indicating that long-term materials development and component durability remain key requirements for reliable utility-scale implementation.

CHAPTER 8 CONCLUSIONS

The transition to third-generation concentrated solar power (CSP) systems presents a promising yet complex pathway to overcoming the limitations in thermal efficiency and economic competitiveness currently faced by existing CSP technologies, especially when compared with photovoltaics and fossil fuels. A key challenge lies in identifying working fluids and system configurations that can operate at higher temperatures, provide efficient heat recovery, and enable multi-functional outputs such as clean water and power. This thesis proposed and analysed a novel direct integration of a parabolic trough solar collector (PTSC) with a supercritical carbon dioxide (sCO₂) Brayton cycle, an Organic Rankine Cycle (ORC), and a direct contact membrane distillation (DCMD) unit, aiming to create a highly efficient, multi-output solar thermal system. The main findings of the research are summarised as follows:

(1) A mathematical model was developed for each key component in the proposed system and validated against experimental data from published literature and national laboratory benchmarks. These component-level models, rooted in heat transfer and thermodynamic principles, served as a robust foundation for energetic, exergetic, and economic analyses throughout the thesis. The study also established economic models based on levelised cost of electricity (LCOE), levelised cost of water (LCOW), and net present value (NPV), enabling comprehensive techno-economic evaluations.

(2) An initial thermodynamic analysis was conducted on the integration of a direct PTSC with a recompression sCO₂ Brayton cycle and DCMD unit. Simulation results showed that lower solar input negatively impacted both thermal and exergetic efficiency, while optimal pressure ratios (3.1–3.2) yielded an exergetic efficiency peak of 38.3%. Water production reached as high as 96.6 m³/h under ideal thermal boundary conditions. High feed temperatures improved water yield but introduced operational sensitivity to solar variability. These results

provide crucial insights into multi-functional CSP design aimed at co-producing electricity and freshwater.

(3) A comprehensive thermodynamic and economic investigation was carried out on the fully integrated PTSC-sCO₂-ORC-DCMD system. Different ORC working fluids were compared under varied ambient conditions. Neopentane and isobutane delivered higher net power output, while toluene offered improved desalination potential. At an ambient temperature of 10°C, cyclohexane and linear hydrocarbons like n-octane and n-nonane reached thermal efficiencies up to 36.52%. Exergy analysis revealed that the DCMD membrane and the second-stage compression/expansion of sCO₂ incurred significant losses at high solar irradiance. Nevertheless, the integrated system achieved exergetic efficiency improvements of 0.12–0.35% over single-cycle systems. Techno-economic analysis identified n-octane as the most profitable fluid under cold conditions, with cyclohexane performing well across various climates.

(4) A novel discretisation-based modelling method was introduced and applied to a direct-configuration PTSC system integrated with sCO₂ Brayton cycle. This method allowed for precise evaluation of design variables such as collector length, inlet/outlet temperatures, flow rates, and solar irradiance. The approach yielded distinctly different optimal conditions than current methods in the literature, particularly when targeting various outlet temperatures. The findings fill a critical methodological gap in CSP design by offering a step-by-step framework that supports future high-fidelity modelling and performance optimisation.

(5) Finally, a detailed thermo-economic mapping was conducted for the proposed system, incorporating expected future cost reductions of sCO₂ Brayton technology, projected to be viable by 2030. A duality method, combining pinch point and approach temperature analysis, was employed for the first time in such systems and compared against traditional energy

balance methods. The system achieved LCOE values between 0.115–0.025 USD/kWh and LCOW values between 0.70–0.40 USD/m³, suggesting strong potential to meet future CSP targets and to rival the cost performance of conventional desalination technologies like reverse osmosis.

A natural continuation of this work is the experimental validation of the modelling framework, particularly the discretised PTSC model and the duality heat-recovery method. While the present study has developed detailed models and benchmarked them against published datasets, prototype-level testing would allow the performance of key components to be confirmed under real operating conditions. For the discretised PTSC model, a small or modular collector loop with distributed temperature and pressure measurement points would enable verification of predicted outlet temperatures, axial temperature gradients, and thermal loss behaviour across a range of irradiance and mass-flow conditions. A natural next step is the experimental validation of the discretised PTSC model and the duality heat-recovery method. The PTSC model may be verified through a small-scale collector test in which measured temperature distributions and outlet conditions are compared against model predictions. Likewise, the duality approach can be validated by testing heat recovery between two controlled thermal streams to confirm that predicted temperature differences and heat-transfer behaviour are reproduced in practice. These experimental efforts would provide direct confirmation of the modelling framework and support the transition from theoretical analysis to practical system development.

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APPENDIX 1 PTSC MATHEMATICAL MODELLING

1. Optical modelling

The optical efficiency of the system is defined as the ratio of direct normal irradiance (DNI) received by the receiver to that captured by the collector. When the incidence angle modifier (θ) is zero, $K(\theta)$ equals one. According to Bellos et al. (2017), the optical efficiency of the LS-3 collector is 0.841, accounting for imperfections such as shading and reflector surface flaws. In this study, $K(\theta)$ is set to 1, focusing on the thermodynamic performance of the fluid.

$$\eta_0 = 0.841K(\theta) \quad (\text{A1 - 1})$$

The mathematical model developed in this research is founded on energy balance and heat transfer principles. The total available solar energy (Q_s) is calculated as the product of direct normal irradiance (DNI), denoted as I_b , and the parabolic trough's aperture area (A_a). The solar energy reflected onto the absorber (Q_s) is determined by the optical efficiency. These relationships are expressed in Equations A1-2 and A1-3, respectively.

$$\dot{Q}_s = A_a I_b \quad (\text{A1 - 2})$$

$$\dot{Q}_{total} = \dot{Q}_s \eta_0 \quad (\text{A1 - 3})$$

2. Thermal mathematical modelling

The following calculation applies to each unit of the PTSC unless otherwise indicated. The mathematical model provided below complements the flowchart illustrated in Fig. 3.4. An energy model is essential to calculate the exergetic component of the mathematical model. Thermal energy from the concentrated solar radiation is transferred from the hotter absorber tube to the cooler fluid through Newton's law of cooling and the steady-state energy balance equation:

$$\dot{Q}_u = h_f \pi D_{ri} L (T_r - T_{fm}) \quad (\text{A1 - 4})$$

$$\dot{Q}_u = \dot{m}c_p(T_{out} - T_{in}) \quad (A1 - 5)$$

The 1-D model's accuracy over short distances facilitates precise outlet temperature measurements, allowing for a more accurate heat gain ($\dot{Q}_u$) calculation using Equation A1-6. This is because enthalpy better represents the fluid's specific heat capacity with pressure drop, especially near the critical point, which varies significantly, as it inherently accounts for these changes. While the difference between A1-6 and A1-5 is generally minimal, the former should more accurately capture the heat gained, especially when evaluated near critical temperature and pressure.

$$\dot{Q}_u = \dot{m}(h_{out} - h_{in}) \quad (A1 - 6)$$

The T_{fm} represents the average temperature of the inlet and outlet working fluid temperatures and h_f is the convective heat transfer coefficient between the fluid and the absorber wall:

$$h_f = \frac{Nu_f K_f}{D_{ri}} \quad (A1 - 7)$$

with Nu_f and K_f representing the Nusselt number and thermal conductivity of the heat transfer fluid. The Nusselt number for turbulent flow was calculated by using the Dittus-Beolter correlation:

$$Nu_f = 0.023 Re_f^{0.8} Pr_f^{0.4} \quad (A1 - 8)$$

where Re_f and Pr_f are the Reynolds and Prandtl numbers of the working fluid, respectively.

These dimensionless numbers can be calculated by:

$$Re_f = \frac{4\dot{m}}{\pi D_{ri} u_f} \quad (A1 - 9)$$

$$Pr_f = \frac{c_{p_f} u_f}{K_f} \quad (A1 - 10)$$

Heat loss from the absorber tube occurs due to the temperature difference between the high-temperature absorber tube and the ambient air. In accordance with the second law of thermodynamics, unabsorbed heat is transferred from the hot absorber tube (hot reservoir) to the cooler ambient air (cold sink). In PTSC, this heat transfer occurs in three stages: initially, through radiation and convection from the absorber tube to the glass cover, subsequently, via conduction from the inner to the outer glass cover, and finally, through radiation and convection from the outer glass cover to the sky and ambient air. Notably, convection loss between the absorber tube and the inner glass cover is negligible due to the vacuum in the annular space. These heat transfer processes are represented by equations detailed in Duffie & Beckman (2020):

$$\dot{Q}_{loss,rad} = \frac{\pi \sigma D_{co} L (T_{r^4} - T_{ci})}{\frac{1}{\varepsilon_r} + \frac{1 - \varepsilon_c}{\varepsilon_c} \left(\frac{D_r}{D_{ci}} \right)} \quad (\text{A1 - 11})$$

$$\dot{Q}_{loss,cond,gi-go} = \frac{2\pi K_c L (T_{cc} - T_{co})}{\ln \left(\frac{D_{co}}{D_{ci}} \right)} \quad (\text{A1 - 12})$$

$$\dot{Q}_{loss,conv+rad} = \pi h_w D_{co} L (T_{co} - T_{amb}) + \varepsilon_c D_{co} L \pi \sigma (T_{co} - T_{sky}) \quad (\text{A1 - 13})$$

With the clear sky temperature calculated as (Ferraro et al., 2013):

$$T_{sky} = 0.0553 T_{amb}^{1.5} \quad (\text{A1 - 14})$$

The convective heat transfer of the wind (h_w) was assumed to be 10W/m²-K. Additionally, according to our experience (Zaharil & Hasanuzaman, 2020), this coefficient is extremely inconsequential to the performance of a vacuumed HCE. The energy balance equation for the processes described above is the sum of the heat gained by the HTF and the heat lost, which equals the total solar radiation concentrated by the collector on the receiver (Q_{total}). It is shown by the equation below:

$$\dot{Q}_{total} = \dot{Q}_{loss} + \dot{Q}_u \quad (A1 - 15)$$

Energetic efficiency is defined as the ratio of the heat absorbed by the working fluid to the total DNI incident on the collector. This study incorporates pumping power into the energetic efficiency, occasionally referred to as overall efficiency. To enhance simplicity and readability, we term this metric thermal efficiency, following the terminology used by Mwesigye et al. (2018). This measure is particularly relevant for longer parabolic trough solar collectors (PTSCs), where significant pressure drops can substantially affect thermal performance.

$$\eta_{th} = \frac{\dot{Q}_u - W_p / \eta_{el}}{I_b A_a} \quad (A1 - 16)$$

where the compressing/pumping power is $W_p = \dot{m} \Delta P / \rho_f$ (Wirz et al., 2014) and η_{el} is the electrical efficiency of a power block.

APPENDIX 2 DCMD MATHEMATICAL MODELLING

The flow chart of the iteration process to calculate T_{mf} and T_{mp} which requires J_m is shown in Fig. 3.12. For the full modelling please refer to the supplementary content. The calculation of J_m is given as:

$$J_{mh} = D_m * (P_a^v - P_b^v) * 3600 \quad (\text{A2 - 1}) \quad (\text{Khayet., 2011; Essalhi \& Khayet.,2013})$$

Where P_a^v and P_b^v are the vapour pressure of the seawater at feed and permeate water and D_m is the diffusion coefficient.

The temperature of membrane at feed and permeate side can't be determined experimentally and thus it is calculated as (Essalhi & Khayet.,2013) :

$$T_{mf} = \frac{k_m * (T_{bp} + (h_f/h_p) * T_{bf}) + ((\delta_m * (h_f * T_{bf} - J_m * \Delta H_v)))}{(k_m) + (h_f * (\delta_m + (k_m/h_p)))} \quad (\text{A2 - 2})$$

$$T_{mp} = \frac{k_m * (T_{bf} + (h_p/h_f) * T_{bp}) + ((\delta_m * (h_p * T_{bp} + J_m * \Delta H_v)))}{(k_m) + (h_p * (\delta_m + (k_m/h_f)))} \quad (\text{A2 - 3})$$

After iteration procedure, with the calculated values of J_m , T_{mf} and T_{mp} are used to calculate the overall heat transfer coefficient (U) total heat transfer in the DCMD membrane module(Q_t), evaporation efficiency (EE) and temperature polarisation coefficient (TPC) and specific thermal energy consumption (STEC). Q_t was calculated by:

$$Q_t = U * (T_{bf} - T_{bp}) \quad (\text{A2 - 4})$$

Whereas U was calculated by (Manawai et al., 2014):

$$U = \left[\frac{1}{h_f} + \frac{1}{\left(\frac{k_m}{\delta_m} \right) + \frac{J_m * \Delta H_v}{T_{mf} - T_{mp}}} + \frac{1}{h_p} \right]^{-1} \quad (\text{A2 - 5})$$

EE and STEC can be respectively evaluated by:

$$EE(\%) = \left[\frac{J_{mh} * \Delta H_v}{U * (T_{bf} - T_{bp})} \right] * 100 \quad (\text{A2 - 6}) \quad (\text{Khalifa et al., 2015; Zhang et al., 2013})$$

$$STEC = [(Q_t / J_{mh}) * \rho] / 1000 \quad (\text{A2 - 7}) \quad (\text{Soomro \& Kim., 2018a})$$

A key aspect of the DCMD process is temperature polarisation, wherein the T_{mf} and T_{mp} differ from their respective bulk temperatures (T_{bf} and T_{bp}). The variance in temperature on either side of the membrane surfaces establishes a gradient in vapor pressure, which serves as the primary impetus for the Direct Contact Membrane Distillation (DCMD) process. The Temperature Polarization Coefficient (TPC), which is the quotient of the actual driving force to the theoretical one, is ascertainable through the application of Equation A2 - 8.

$$TPC = \frac{T_{mf} - T_{mp}}{T_{bf} - T_{bp}} = \frac{\Delta T_{bm}}{\Delta T_b} \quad (\text{A2 - 8})$$

The mass flow rate needed to cool down the thermodynamic cycle and thus the heat input into the condenser and DCMD units is shown in equation A2 - 9. The values of enthalpies of inlet and outlet of seawater (h_{12} and h_{11}) were fixed.

$$\dot{Q}_{out} = \dot{m}_{sea} * (h_{12} - h_{11}) \quad (\text{A2 - 9})$$

The total amount of DCMD units needed was calculated based on the $\dot{m}_{sea}$ needed and can be calculated as:

$$N_{DCMD} = \dot{m}_{sea} / \dot{m}_{DCMD} \quad (\text{A2 - 10})$$

With $\dot{m}_{DCMD}$ is the mass flow rate of each unit of DCMD. The value of $\dot{m}_{DCMD}$ is 0.333kg/s and it was calculated by converting from the inlet volumetric flow rate of 20L/min (Table 3.8). The amount of water in m^3 /hour produced for each unit of DCMD is:

$$V_{water} = A_{DCMD} * J_{mh} / \rho_{water} \quad (\text{A2 - 11})$$

The total amount of water produced by the system is:

$$V_{water,total} = V_{water} * N_{DCMD} \quad (A2 - 12)$$

APPENDIX 3 MODULE COSTING TECHNIQUE

This appendix explains how the cost of each main component of an Organic Rankine Cycle (ORC) plant can be estimated using the Module Costing Technique (MCT). The process moves step by step from the base equipment cost up to the total CAPEX cost of the plant.

Step 1 — Base Equipment Cost

For each component, calculate the base purchased cost at reference (base) conditions:

$$\log_{10} C_p^0 = K_1 + K_2 \log_{10}(A) + K_3 [\log_{10}(A)]^2 \quad (\text{A3 - 1})$$

where:

- C_p^0 = base purchased cost
- A = size parameter (e.g., area, power, or flow, depending on the component)
- K_1, K_2, K_3 = coefficients specific to the equipment type

Step 2 — Adjustment for Pressure and Material

Adjust the base cost for design pressure and selected material.

Pressure factor:

$$\log_{10} F_p = C_1 + C_2 \log_{10}(p) + C_3 [\log_{10}(p)]^2. \quad (\text{A3 - 2})$$

Material factor: F_m (taken from standard tables).

Purchased cost at operating conditions:

$$C_p = C_p^0 \times F_p \times F_m \quad (\text{A3 - 3})$$

Step 3 — Bare-Module Cost

Convert the purchased cost into the bare-module cost, which includes installation and indirect costs.

For pumps and heat exchangers:

$$F_{BM} = B_1 + B_2 F_p F_m \quad (A3 - 3)$$

$$C_{BM} = C_p^0 \times F_{BM} \quad (A3 - 4)$$

For other components:

$$C_{BM} = C_p^0 \times F_{BM} \quad (A3 - 5)$$

Step 4 — Plant CAPEX Cost

Once all bare-module costs have been estimated, sum them to obtain the total plant cost.

$$CAPEX = (1 + \text{contingency} + \text{fees}) \sum C_{BM} + (\text{site development factor}) \sum C_{BM} \quad (A3 - 6)$$

Here:

- The first term accounts for bare-module costs plus contingency and engineering fees
- The second term accounts for site development and auxiliary facilities

Step 5 — Operating Cost and LCOE

Operating and maintenance costs are typically expressed as a fraction of the CAPEX cost.

The levelised cost of electricity (LCOE) is then:

$$LCOE = C_{\text{annual}} / E_{\text{annual}} \quad (A3 - 7)$$

where C_{annual} is the annualised plant cost (including major repair every 5 years and O&M) and

E_{annual} is the annual electricity generation.

Summary of Workflow

1. Determine the size parameter for each component.
2. Calculate the base purchased cost.
3. Apply pressure and material corrections.
4. Convert to bare-module cost.
5. Sum all component costs to obtain the CAPEXs cost.

6. Calculate LCOE

APPENDIX 4. OPTIMISED THERMAL AND EXERGETIC EFFICIENCY DATA

DNI (W/m ²)	Length (m)	T _{out} = Low (350 K – 500 K)	T _{out} = Medium (500 K – 700 K)	T _{out} = High (700 K – 900 K)	T _{out} = Ultra-high (≥900 K)
1000	25	T _{in} = 350 K n _{th} = 86.05% ṁ = 4.5 kg/s	T _{in} = 450 K n _{th} = 83% ṁ = 1.5 kg/s	T _{in} = 650 K n _{th} = 76.12% ṁ = 1.5 kg/s	T _{in} = 850 K n _{th} = 54.77% ṁ = 1 kg/s
	50	T _{in} = 350 K n _{th} = 86.04% ṁ = 1.0 kg/s	T _{in} = 400 K n _{th} = 85.99% ṁ = 1.5 kg/s	T _{in} = 400 K n _{th} = 83.48% ṁ = 0.5 kg/s	T _{in} = 600 K n _{th} = 71.44% ṁ = 0.5 kg/s
	75	T _{in} = 350 K n _{th} = 84.0% ṁ = 4.0 kg/s	T _{in} = 400 K n _{th} = 83.24% ṁ = 2.0 kg/s	T _{in} = 450 K n _{th} = 79.92% ṁ = 1.0 kg/s	T _{in} = 450 K n _{th} = 70.36% ṁ = 0.5 kg/s
	100	T _{in} = 350 K n _{th} = 83.95% ṁ = 4.5 kg/s	T _{in} = 350 K n _{th} = 83.45% ṁ = 1.5 kg/s	T _{in} = 400 K n _{th} = 80.09% ṁ = 1.0 kg/s	T _{in} = 350 K n _{th} = 72.39% ṁ = 0.5 kg/s
	125	T _{in} = 350 K n _{th} = 83.88% ṁ = 5.0 kg/s	T _{in} = 350 K n _{th} = 83.53% ṁ = 2.0 kg/s	T _{in} = 350 K n _{th} = 80.85% ṁ = 1.0 kg/s	T _{in} = 550 K n _{th} = 67.65% ṁ = 1.0 kg/s
	150	T _{in} = 350 K n _{th} = 83.79% ṁ = 5.0 kg/s	T _{in} = 350 K n _{th} = 83.54% ṁ = 2.5 kg/s	T _{in} = 400 K n _{th} = 80.21% ṁ = 1.5 kg/s	T _{in} = 450 K n _{th} = 71.16% ṁ = 1.0 kg/s
900	25	T _{in} = 350 K n _{th} = 84.05% ṁ = 3.5 kg/s	T _{in} = 400 K n _{th} = 82.73% ṁ = 0.5 kg/s	T _{in} = 550 K n _{th} = 77.01% ṁ = 0.5 kg/s	T _{in} = 850 K n _{th} = 54.77% ṁ = 1 kg/s
	50	T _{in} = 350 K n _{th} = 86.22% ṁ = 1.0 kg/s	T _{in} = 400 K n _{th} = 86.14% ṁ = 1 kg/s	T _{in} = 400 K n _{th} = 84.26% ṁ = 0.5 kg/s	T _{in} = 650 K n _{th} = 68.04% ṁ = 0.5 kg/s
	75	T _{in} = 350 K n _{th} = 84.0% ṁ = 4.5 kg/s	T _{in} = 350 K n _{th} = 83.34% ṁ = 1.0 kg/s	T _{in} = 350 K n _{th} = 79.31% ṁ = 0.5 kg/s	T _{in} = 500 K n _{th} = 67.92% ṁ = 0.5 kg/s
	100	T _{in} = 350 K n _{th} = 83.95% ṁ = 4.5 kg/s	T _{in} = 350 K n _{th} = 83.55% ṁ = 1.5 kg/s	T _{in} = 500 K n _{th} = 78.83% ṁ = 1.5 kg/s	T _{in} = 400 K n _{th} = 69.16% ṁ = 0.5 kg/s
	125	T _{in} = 350 K n _{th} = 83.89% ṁ = 5.0 kg/s	T _{in} = 350 K. n _{th} = 83.17%. ṁ = 1.5 kg/s	T _{in} = 400 K n _{th} = 79.60% ṁ = 1.0 kg/s	T _{in} = 350 K n _{th} = 68.03% ṁ = 0.5 kg/s
	150	T _{in} = 350 K n _{th} = 83.81% ṁ = 5.0 kg/s	T _{in} = 350 K n _{th} = 83.35%. ṁ = 2.0 kg/s	T _{in} = 350 K. n _{th} = 79.77% ṁ = 1.0 kg/s	T _{in} = 500 K n _{th} = 68.72% ṁ = 1.0 kg/s

800	25	$T_{in} = 350 \text{ K}$ $n_{th} = 84.03\%$ $\dot{m} = 4.0 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 82.80\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{th} = 73.84\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 850 \text{ K}$ $n_{th} = 44.82\%$ $\dot{m} = 0.5 \text{ kg/s}$
	50	$T_{in} = 400 \text{ K}$ $n_{th} = 86.45\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 86.44\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 83.02\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 700 \text{ K}$ $n_{th} = 63.58\%$ $\dot{m} = 0.5 \text{ kg/s}$
	75	$T_{in} = 350 \text{ K}$ $n_{th} = 83.99\%$ $\dot{m} = 4.5 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 83.44\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 80.26\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{th} = 60.53\%$ $\dot{m} = 0.5 \text{ kg/s}$
	100	$T_{in} = 350 \text{ K}$ $n_{th} = 83.95\%$ $\dot{m} = 4.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 82.96\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 78.64\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 66.73\%$ $\dot{m} = 0.5 \text{ kg/s}$
	125	$T_{in} = 350 \text{ K}$ $n_{th} = 83.90\%$ $\dot{m} = 5.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 83.31\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 79.34\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 70.73\%$ $\dot{m} = 0.5 \text{ kg/s}$
	150	$T_{in} = 350 \text{ K}$ $n_{th} = 83.83\%$ $\dot{m} = 5.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 83.45\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 80.64\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{th} = 65.86\%$ $\dot{m} = 1.0 \text{ kg/s}$
700	25	$T_{in} = 350 \text{ K}$ $n_{th} = 84.02\%$ $\dot{m} = 3.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 82.85\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{th} = 73.54\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 850 \text{ K}$ $n_{th} = 41.88\%$ $\dot{m} = 0.5 \text{ kg/s}$
	50	$T_{in} = 400 \text{ K}$ $n_{th} = 86.76\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 86.81\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 84.0\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 750 \text{ K}$ $n_{th} = 57.59\%$ $\dot{m} = 0.5 \text{ kg/s}$
	75	$T_{in} = 350 \text{ K}$ $n_{th} = 83.97\%$ $\dot{m} = 3.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 83.02\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 77.81\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{th} = 56.17\%$ $\dot{m} = 0.5 \text{ kg/s}$
	100	$T_{in} = 350 \text{ K}$ $n_{th} = 83.94\%$ $\dot{m} = 4.5 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 83.08\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 77.79\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{th} = 56.7\%$ $\dot{m} = 0.5 \text{ kg/s}$
	125	$T_{in} = 350 \text{ K}$ $n_{th} = 83.90\%$ $\dot{m} = 4.5 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 83.43\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 500 \text{ K}$ $n_{th} = 77.78\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 67.62\%$ $\dot{m} = 0.5 \text{ kg/s}$
	150	$T_{in} = 350 \text{ K}$ $n_{th} = 83.84\%$ $\dot{m} = 4.5 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 83.11\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 78.29\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 68.04\%$ $\dot{m} = 0.5 \text{ kg/s}$
600	25	$T_{in} = 350 \text{ K}$ $n_{th} = 84.0\%$ $\dot{m} = 3 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 81.41\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{th} = 68.44\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 900 \text{ K}$ $n_{th} = 30.82\%$ $\dot{m} = 0.5 \text{ kg/s}$
	50	$T_{in} = 400 \text{ K}$ $n_{th} = 87.25\%$ $\dot{m} = 1 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 87.32\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 500 \text{ K}$ $n_{th} = 82.66\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 750 \text{ K}$ $n_{th} = 57.32\%$ $\dot{m} = 0.5 \text{ kg/s}$
	75	$T_{in} = 350 \text{ K}$ $n_{th} = 83.96\%$ $\dot{m} = 4.0 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 82.45\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 75.42\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 700 \text{ K}$ $n_{th} = 50.22\%$ $\dot{m} = 0.5 \text{ kg/s}$

	100	$T_{in} = 350 \text{ K}$ $n_{th} = 83.93\%$ $\dot{m} = 4.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 83.26\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 79.42\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{th} = 50.45\%$ $\dot{m} = 0.5 \text{ kg/s}$
	125	$T_{in} = 350 \text{ K}$ $n_{th} = 83.89\%$ $\dot{m} = 4.5 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 82.75\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 77.85\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 500 \text{ K}$ $n_{th} = 60.49\%$ $\dot{m} = 0.5 \text{ kg/s}$
	150	$T_{in} = 350 \text{ K}$ $n_{th} = 83.85\%$ $\dot{m} = 5.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 83.28\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 500 \text{ K}$ $n_{th} = 76.69\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 59.8\%$ $\dot{m} = 0.5 \text{ kg/s}$
500	25	$T_{in} = 350 \text{ K}$ $n_{th} = 83.97\%$ $\dot{m} = 3.0 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 81.35\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{th} = 67.07\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 900 \text{ K}$ $n_{th} = 21.67\%$ $\dot{m} = 1.5 \text{ kg/s}$
	50	$T_{in} = 350 \text{ K}$ $n_{th} = 87.89\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 87.58\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{th} = 80.66\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 800 \text{ K}$ $n_{th} = 47.48\%$ $\dot{m} = 0.5 \text{ kg/s}$
	75	$T_{in} = 350 \text{ K}$ $n_{th} = 83.93\%$ $\dot{m} = 3.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 82.52\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 500 \text{ K}$ $n_{th} = 72.55\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 800 \text{ K}$ $n_{th} = 33.12\%$ $\dot{m} = 0.5 \text{ kg/s}$
	100	$T_{in} = 350 \text{ K}$ $n_{th} = 83.9\%$ $\dot{m} = 3.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 81.89\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 76.81\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 750 \text{ K}$ $n_{th} = 36.85\%$ $\dot{m} = 0.5 \text{ kg/s}$
	125	$T_{in} = 350 \text{ K}$ $n_{th} = 83.87\%$ $\dot{m} = 4.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 83.03\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 78.19\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{th} = 45.88\%$ $\dot{m} = 0.5 \text{ kg/s}$
	150	$T_{in} = 350 \text{ K}$ $n_{th} = 83.84\%$ $\dot{m} = 4.5 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 82.52\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 74.84\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{th} = 52.55\%$ $\dot{m} = 0.5 \text{ kg/s}$
400	25	$T_{in} = 350 \text{ K}$ $n_{th} = 83.92\%$ $\dot{m} = 3 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 81.19\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{th} = 64.86\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 900 \text{ K}$ $n_{th} = 8.32\%$ $\dot{m} = 1 \text{ kg/s}$
	50	$T_{in} = 400 \text{ K}$ $n_{th} = 88.69\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 88.72\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{th} = 77.39\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 850 \text{ K}$ $n_{th} = 32.18\%$ $\dot{m} = 0.5 \text{ kg/s}$
	75	$T_{in} = 350 \text{ K}$ $n_{th} = 83.88\%$ $\dot{m} = 3.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 82.78\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{th} = 68.40\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 900 \text{ K}$ $n_{th} = 6.99\%$ $\dot{m} = 1.0 \text{ kg/s}$
	100	$T_{in} = 350 \text{ K}$ $n_{th} = 83.86\%$ $\dot{m} = 3.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 82.04\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 74.22\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 850 \text{ K}$ $n_{th} = 15.39\%$ $\dot{m} = 1.0 \text{ kg/s}$
	125	$T_{in} = 350 \text{ K}$ $n_{th} = 83.84\%$ $\dot{m} = 4.0 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 81.42\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 75.52\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 850 \text{ K}$ $n_{th} = 13.57\%$ $\dot{m} = 0.5 \text{ kg/s}$
	150	$T_{in} = 350 \text{ K}$ $n_{th} = 83.81\%$ $\dot{m} = 4.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 82.89\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 72.15\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 800 \text{ K}$ $n_{th} = 20.05\%$ $\dot{m} = 0.5 \text{ kg/s}$

300	25	$T_{in} = 350 \text{ K}$ $n_{th} = 83.85\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 500 \text{ K}$ $n_{th} = 79.26\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 700 \text{ K}$ $n_{th} = 54.42\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{th} = -34.47\%$ $\dot{m} = 0.5 \text{ kg/s}$
	50	$T_{in} = 400 \text{ K}$ $n_{th} = 90.33\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 89.96\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{th} = 78.46\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 900 \text{ K}$ $n_{th} = 5.95\%$ $\dot{m} = 0.5 \text{ kg/s}$
	75	$T_{in} = 350 \text{ K}$ $n_{th} = 83.81\%$ $\dot{m} = 3 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 80.9\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{th} = 61.39\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{th} = -25.12\%$ $\dot{m} = 0.5 \text{ kg/s}$
	100	$T_{in} = 350 \text{ K}$ $n_{th} = 83.79\%$ $\dot{m} = 3.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 82.42\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{th} = 64.66\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{th} = -30.35\%$ $\dot{m} = 1.0 \text{ kg/s}$
	125	$T_{in} = 350 \text{ K}$ $n_{th} = 83.77\%$ $\dot{m} = 3.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 81.61\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 500 \text{ K}$ $n_{th} = 67.34\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{th} = -28.06\%$ $\dot{m} = 1.0 \text{ kg/s}$
	150	$T_{in} = 350 \text{ K}$ $n_{th} = 83.74\%$ $\dot{m} = 3.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 81.06\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 69.79\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{th} = -31.71\%$ $\dot{m} = 1.5 \text{ kg/s}$
200	25	$T_{in} = 350 \text{ K}$ $n_{th} = 83.71\%$ $\dot{m} = 2 \text{ kg/s}$	$T_{in} = 500 \text{ K}$ $n_{th} = 77.21\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 700 \text{ K}$ $n_{th} = 41.23\%$ $\dot{m} = 1 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{th} = -84.06\%$ $\dot{m} = 0.5 \text{ kg/s}$
	50	$T_{in} = 400 \text{ K}$ $n_{th} = 93.26\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 92.78\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{th} = 70.64\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{th} = -47.63\%$ $\dot{m} = 0.5 \text{ kg/s}$
	75	$T_{in} = 350 \text{ K}$ $n_{th} = 83.66\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 500 \text{ K}$ $n_{th} = 76.25\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 700 \text{ K}$ $n_{th} = 38.47\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{th} = -89.83\%$ $\dot{m} = 1.5 \text{ kg/s}$
	100	$T_{in} = 350 \text{ K}$ $n_{th} = 83.66\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 400 \text{ K}$ $n_{th} = 80.13\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{th} = 44.11\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{th} = -84.89\%$ $\dot{m} = 1.5 \text{ kg/s}$
	125	$T_{in} = 350 \text{ K}$ $n_{th} = 83.62\%$ $\dot{m} = 3.0 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{th} = 79.27\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{th} = 51.81\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{th} = -89.53\%$ $\dot{m} = 2.0 \text{ kg/s}$
	150	$T_{in} = 350 \text{ K}$ $n_{th} = 83.60\%$ $\dot{m} = 3.0 \text{ kg/s}$	$T_{in} = 350 \text{ K}$ $n_{th} = 81.73\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{th} = 48.94\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{th} = -95.34\%$ $\dot{m} = 2.5 \text{ kg/s}$

DNI (W/m ²)	Length (m)	T _{out} = Low (350 K – 500 K)	T _{out} = Medium (500 K – 700 K)	T _{out} = High (≥700 K – 900 K)	T _{out} = Ultra-high (≥900 K)
1000	25	T _{in} = 450 K n _{ex} = 32.67% ṁ = 2 kg/s	T _{in} = 650 K n _{ex} = 45.91% ṁ = 2.0 kg/s	T _{in} = 700 K n _{ex} = 46.46% ṁ = 2.5 kg/s	T _{in} = 850 K n _{ex} = 38.97% ṁ = 1 kg/s
	50	T _{in} = 450 K n _{ex} = 34.23% ṁ = 3.5 kg/s	T _{in} = 650 K n _{ex} = 48.58% ṁ = 4 kg/s	T _{in} = 700 K n _{ex} = 49.67% ṁ = 2.5 kg/s	T _{in} = 750 K n _{ex} = 46.61% ṁ = 1 kg/s
	75	T _{in} = 450 K n _{ex} = 33.31% ṁ = 5 kg/s	T _{in} = 600 K n _{ex} = 45.09% ṁ = 3 kg/s	T _{in} = 700 K n _{ex} = 46.27% ṁ = 3.5 kg/s	T _{in} = 700 K n _{ex} = 42.71% ṁ = 1 kg/s
	100	T _{in} = 400 K n _{ex} = 28.87% ṁ = 3.5 kg/s	T _{in} = 600 K n _{ex} = 45% ṁ = 4.0 kg/s	T _{in} = 650 K n _{ex} = 46.21% ṁ = 3.0 kg/s	T _{in} = 600 K n _{ex} = 43.59% ṁ = 1.0 kg/s
	125	T _{in} = 400 K n _{ex} = 29.44% ṁ = 4.0 kg/s	T _{in} = 600 K n _{ex} = 45.03% ṁ = 4.5 kg/s	T _{in} = 650 K n _{ex} = 46.15% ṁ = 3.5 kg/s	T _{in} = 550 K n _{ex} = 42.7% ṁ = 1.0 kg/s
	150	T _{in} = 400 K n _{ex} = 29.35% ṁ = 5.0 kg/s	T _{in} = 550 K n _{ex} = 43.77% ṁ = 4.0 kg/s	T _{in} = 650 K n _{ex} = 46.04% ṁ = 4.0 kg/s	T _{in} = 600 K n _{ex} = 43.87% ṁ = 1.5 kg/s
900	25	T _{in} = 450 K n _{ex} = 33.06% ṁ = 1.5 kg/s	T _{in} = 650 K n _{ex} = 45.45% ṁ = 2.0 kg/s	T _{in} = 700 K n _{ex} = 45.83% ṁ = 2.5 kg/s	T _{in} = 800 K n _{ex} = 37.85% ṁ = 0.5 kg/s
	50	T _{in} = 450 K n _{ex} = 34.03% ṁ = 3.5 kg/s	T _{in} = 650 K n _{ex} = 48.51% ṁ = 3.5 kg/s	T _{in} = 700 K n _{ex} = 49.23% ṁ = 3.5 kg/s	T _{in} = 650 K n _{ex} = 45.1% ṁ = 0.5 kg/s
	75	T _{in} = 450 K n _{ex} = 33.24% ṁ = 4.5 kg/s	T _{in} = 650 K n _{ex} = 45.11% ṁ = 5 kg/s	T _{in} = 650 K n _{ex} = 45.66% ṁ = 2.5 kg/s	T _{in} = 500 K n _{ex} = 41.05% ṁ = 0.5 kg/s
	100	T _{in} = 400 K n _{ex} = 28.31% ṁ = 3.5 kg/s	T _{in} = 600 K n _{ex} = 44.73% ṁ = 3.5 kg/s	T _{in} = 650 K n _{ex} = 45.64% ṁ = 3.0 kg/s	T _{in} = 650 K n _{ex} = 42.12% ṁ = 1.0 kg/s
	125	T _{in} = 400 K n _{ex} = 29.54% ṁ = 3.5 kg/s	T _{in} = 600 K n _{ex} = 44.74% ṁ = 4 kg/s	T _{in} = 650 K n _{ex} = 45.56% ṁ = 3.5 kg/s	T _{in} = 700 K n _{ex} = 41.83% ṁ = 1.5 kg/s
	150	T _{in} = 400 K n _{ex} = 29.27% ṁ = 4.5 kg/s	T _{in} = 600 K n _{ex} = 44.46% ṁ = 5 kg/s	T _{in} = 650 K n _{ex} = 45.45% ṁ = 3.5 kg/s	T _{in} = 650 K n _{ex} = 42.4% ṁ = 1.5 kg/s
800	25	T _{in} = 450 K n _{ex} = 32.71% ṁ = 1.5 kg/s	T _{in} = 650 K n _{ex} = 44.90% ṁ = 2.0 kg/s	T _{in} = 700 K n _{ex} = 45.07% ṁ = 2.5 kg/s	T _{in} = 850 K n _{ex} = 32.01% ṁ = 0.5 kg/s

	50	$T_{in} = 450\text{ K}$ $n_{ex} = 34.22\%$ $\dot{m} = 3\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 48.4\%$ $\dot{m} = 3\text{ kg/s}$	$T_{in} = 700\text{ K}$ $n_{ex} = 48.87\%$ $\dot{m} = 2.5\text{ kg/s}$	$T_{in} = 700\text{ K}$ $n_{ex} = 43.1\%$ $\dot{m} = 0.5\text{ kg/s}$
	75	$T_{in} = 450\text{ K}$ $n_{ex} = 33.16\%$ $\dot{m} = 4\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 44.66\%$ $\dot{m} = 4.5\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 45.01\%$ $\dot{m} = 2.5\text{ kg/s}$	$T_{in} = 600\text{ K}$ $n_{ex} = 39.19\%$ $\dot{m} = 0.5\text{ kg/s}$
	100	$T_{in} = 400\text{ K}$ $n_{ex} = 29.4\%$ $\dot{m} = 2.5\text{ kg/s}$	$T_{in} = 600\text{ K}$ $n_{ex} = 44.35\%$ $\dot{m} = 3\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 44.97\%$ $\dot{m} = 3\text{ kg/s}$	$T_{in} = 700\text{ K}$ $n_{ex} = 39.59\%$ $\dot{m} = 1\text{ kg/s}$
	125	$T_{in} = 400\text{ K}$ $n_{ex} = 28.84\%$ $\dot{m} = 3.5\text{ kg/s}$	$T_{in} = 550\text{ K}$ $n_{ex} = 43.35\%$ $\dot{m} = 2.5\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 44.87\%$ $\dot{m} = 3.5\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 39.89\%$ $\dot{m} = 1\text{ kg/s}$
	150	$T_{in} = 400\text{ K}$ $n_{ex} = 29.18\%$ $\dot{m} = 4\text{ kg/s}$	$T_{in} = 600\text{ K}$ $n_{ex} = 44.1\%$ $\dot{m} = 4.5\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 44.75\%$ $\dot{m} = 3.5\text{ kg/s}$	$T_{in} = 550\text{ K}$ $n_{ex} = 41.15\%$ $\dot{m} = 1\text{ kg/s}$
700	25	$T_{in} = 450\text{ K}$ $n_{ex} = 32.35\%$ $\dot{m} = 1.5\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 44.23\%$ $\dot{m} = 2.0\text{ kg/s}$	$T_{in} = 700\text{ K}$ $n_{ex} = 44.1\%$ $\dot{m} = 2.5\text{ kg/s}$	$T_{in} = 850\text{ K}$ $n_{ex} = 29.79\%$ $\dot{m} = 0.5\text{ kg/s}$
	50	$T_{in} = 450\text{ K}$ $n_{ex} = 34.47\%$ $\dot{m} = 2.5\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 48.1\%$ $\dot{m} = 3\text{ kg/s}$	$T_{in} = 700\text{ K}$ $n_{ex} = 48.36\%$ $\dot{m} = 2.5\text{ kg/s}$	$T_{in} = 750\text{ K}$ $n_{ex} = 39.81\%$ $\dot{m} = 0.5\text{ kg/s}$
	75	$T_{in} = 450\text{ K}$ $n_{ex} = 33.06\%$ $\dot{m} = 3.5\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 44.17\%$ $\dot{m} = 3.5\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 44.23\%$ $\dot{m} = 2.5\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 37.1\%$ $\dot{m} = 0.5\text{ kg/s}$
	100	$T_{in} = 450\text{ K}$ $n_{ex} = 33.06\%$ $\dot{m} = 5.0\text{ kg/s}$	$T_{in} = 600\text{ K}$ $n_{ex} = 43.83\%$ $\dot{m} = 2.5\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 44.16\%$ $\dot{m} = 3\text{ kg/s}$	$T_{in} = 550\text{ K}$ $n_{ex} = 37.57\%$ $\dot{m} = 0.5\text{ kg/s}$
	125	$T_{in} = 400\text{ K}$ $n_{ex} = 28.87\%$ $\dot{m} = 3\text{ kg/s}$	$T_{in} = 600\text{ K}$ $n_{ex} = 43.82\%$ $\dot{m} = 3\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 44.05\%$ $\dot{m} = 3.5\text{ kg/s}$	$T_{in} = 700\text{ K}$ $n_{ex} = 36.93\%$ $\dot{m} = 1.0\text{ kg/s}$
	150	$T_{in} = 400\text{ K}$ $n_{ex} = 29.09\%$ $\dot{m} = 3.5\text{ kg/s}$	$T_{in} = 600\text{ K}$ $n_{ex} = 43.6\%$ $\dot{m} = 4\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 43.92\%$ $\dot{m} = 3.5\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 37.72\%$ $\dot{m} = 1.0\text{ kg/s}$
600	25	$T_{in} = 450\text{ K}$ $n_{ex} = 32.81\%$ $\dot{m} = 1.0\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 43.37\%$ $\dot{m} = 2.0\text{ kg/s}$	$T_{in} = 700\text{ K}$ $n_{ex} = 42.84\%$ $\dot{m} = 2.5\text{ kg/s}$	$T_{in} = 900\text{ K}$ $n_{ex} = 23.22\%$ $\dot{m} = 2.5\text{ kg/s}$
	50	$T_{in} = 450\text{ K}$ $n_{ex} = 34.24\%$ $\dot{m} = 2.5\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 47.85\%$ $\dot{m} = 2.5\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 47.9\%$ $\dot{m} = 2.0\text{ kg/s}$	$T_{in} = 750\text{ K}$ $n_{ex} = 39.3\%$ $\dot{m} = 0.5\text{ kg/s}$
	75	$T_{in} = 450\text{ K}$ $n_{ex} = 32.95\%$ $\dot{m} = 3.0\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 43.27\%$ $\dot{m} = 3.0\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 43.26\%$ $\dot{m} = 2.5\text{ kg/s}$	$T_{in} = 700\text{ K}$ $n_{ex} = 33.76\%$ $\dot{m} = 0.5\text{ kg/s}$
	100	$T_{in} = 450\text{ K}$ $n_{ex} = 33.04\%$ $\dot{m} = 4.0\text{ kg/s}$	$T_{in} = 600\text{ K}$ $n_{ex} = 43.07\%$ $\dot{m} = 2.0\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 43.17\%$ $\dot{m} = 3.0\text{ kg/s}$	$T_{in} = 650\text{ K}$ $n_{ex} = 33.41\%$ $\dot{m} = 0.5\text{ kg/s}$

	125	$T_{in} = 450 \text{ K}$ $n_{ex} = 33.17\%$ $\dot{m} = 5.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 43.07\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{ex} = 43.02\%$ $\dot{m} = 3.5 \text{ kg/s}$	$T_{in} = 500 \text{ K}$ $n_{ex} = 36.49\%$ $\dot{m} = 0.5 \text{ kg/s}$
	150	$T_{in} = 400 \text{ K}$ $n_{ex} = 28.99\%$ $\dot{m} = 3.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 42.93\%$ $\dot{m} = 3.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 43.03\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 450 \text{ K}$ $n_{ex} = 34.92\%$ $\dot{m} = 0.5 \text{ kg/s}$
500	25	$T_{in} = 450 \text{ K}$ $n_{ex} = 32.27\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{ex} = 42.22\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 700 \text{ K}$ $n_{ex} = 41.11\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 900 \text{ K}$ $n_{ex} = 16.43\%$ $\dot{m} = 2.5 \text{ kg/s}$
	50	$T_{in} = 450 \text{ K}$ $n_{ex} = 34.58\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{ex} = 47.46\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{ex} = 47.41\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 800 \text{ K}$ $n_{ex} = 33.16\%$ $\dot{m} = 0.5 \text{ kg/s}$
	75	$T_{in} = 450 \text{ K}$ $n_{ex} = 32.8\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 42.04\%$ $\dot{m} = 2 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{ex} = 41.86\%$ $\dot{m} = 2 \text{ kg/s}$	$T_{in} = 800 \text{ K}$ $n_{ex} = 23.17\%$ $\dot{m} = 0.5 \text{ kg/s}$
	100	$T_{in} = 450 \text{ K}$ $n_{ex} = 32.77\%$ $\dot{m} = 3.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 42.05\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 41.88\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 750 \text{ K}$ $n_{ex} = 25.37\%$ $\dot{m} = 0.5 \text{ kg/s}$
	125	$T_{in} = 450 \text{ K}$ $n_{ex} = 32.84\%$ $\dot{m} = 4.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 42.04\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 41.97\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{ex} = 30.24\%$ $\dot{m} = 0.5 \text{ kg/s}$
	150	$T_{in} = 450 \text{ K}$ $n_{ex} = 33.11\%$ $\dot{m} = 5.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 41.99\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 41.81\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{ex} = 32.79\%$ $\dot{m} = 0.5 \text{ kg/s}$
400	25	$T_{in} = 450 \text{ K}$ $n_{ex} = 31.68\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 40.67\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 700 \text{ K}$ $n_{ex} = 38.56\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 900 \text{ K}$ $n_{ex} = 6.38\%$ $\dot{m} = 2.0 \text{ kg/s}$
	50	$T_{in} = 450 \text{ K}$ $n_{ex} = 35.12\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{ex} = 46.89\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{ex} = 46.47\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 850 \text{ K}$ $n_{ex} = 22.86\%$ $\dot{m} = 0.5 \text{ kg/s}$
	75	$T_{in} = 450 \text{ K}$ $n_{ex} = 32.59\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 40.76\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{ex} = 39.71\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 900 \text{ K}$ $n_{ex} = 5.835\%$ $\dot{m} = 2.0 \text{ kg/s}$
	100	$T_{in} = 450 \text{ K}$ $n_{ex} = 32.41\%$ $\dot{m} = 3.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 40.73\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{ex} = 39.21\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 850 \text{ K}$ $n_{ex} = 10.94\%$ $\dot{m} = 0.5 \text{ kg/s}$
	125	$T_{in} = 450 \text{ K}$ $n_{ex} = 32.62\%$ $\dot{m} = 3.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 40.69\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 40.26\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 850 \text{ K}$ $n_{ex} = 9.667\%$ $\dot{m} = 0.5 \text{ kg/s}$
	150	$T_{in} = 450 \text{ K}$ $n_{ex} = 32.81\%$ $\dot{m} = 4.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 40.61\%$ $\dot{m} = 3.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 39.9\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 800 \text{ K}$ $n_{ex} = 14.03\%$ $\dot{m} = 0.5 \text{ kg/s}$
300	25	$T_{in} = 450 \text{ K}$ $n_{ex} = 32.11\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 38.97\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 700 \text{ K}$ $n_{ex} = 34.35\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{ex} = -25.22\%$ $\dot{m} = 0.5 \text{ kg/s}$

	50	$T_{in} = 450 \text{ K}$ $n_{ex} = 35.01\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 46.27\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 45.7\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 900 \text{ K}$ $n_{ex} = 4.517\%$ $\dot{m} = 1.0 \text{ kg/s}$
	75	$T_{in} = 450 \text{ K}$ $n_{ex} = 32.26\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 38.86\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 36.15\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{ex} = -18.24\%$ $\dot{m} = 0.5 \text{ kg/s}$
	100	$T_{in} = 450 \text{ K}$ $n_{ex} = 32.3\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 38.79\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{ex} = 36.7\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{ex} = -21.84\%$ $\dot{m} = 1.0 \text{ kg/s}$
	125	$T_{in} = 450 \text{ K}$ $n_{ex} = 32.35\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 38.69\%$ $\dot{m} = 2.5 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 37.22\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{ex} = -20.13\%$ $\dot{m} = 1.0 \text{ kg/s}$
	150	$T_{in} = 450 \text{ K}$ $n_{ex} = 32.41\%$ $\dot{m} = 3.0 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{ex} = 38.67\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 36.52\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{ex} = -22.14\%$ $\dot{m} = 1.5 \text{ kg/s}$
200	25	$T_{in} = 450 \text{ K}$ $n_{ex} = 30.91\%$ $\dot{m} = 5 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{ex} = 36.15\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 700 \text{ K}$ $n_{ex} = 26.13\%$ $\dot{m} = 2 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{ex} = -61.32\%$ $\dot{m} = 0.5 \text{ kg/s}$
	50	$T_{in} = 400 \text{ K}$ $n_{ex} = 36.22\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 4600 \text{ K}$ $n_{ex} = 46.1\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{ex} = 42.68\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{ex} = -34.68\%$ $\dot{m} = 0.5 \text{ kg/s}$
	75	$T_{in} = 450 \text{ K}$ $n_{ex} = 31.61\%$ $\dot{m} = 1.0 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{ex} = 36.23\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{ex} = 28.4\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{ex} = -64.07\%$ $\dot{m} = 1.5 \text{ kg/s}$
	100	$T_{in} = 450 \text{ K}$ $n_{ex} = 31.44\%$ $\dot{m} = 1.5 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{ex} = 36.23\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 650 \text{ K}$ $n_{ex} = 26.9\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{ex} = -60.34\%$ $\dot{m} = 1.5 \text{ kg/s}$
	125	$T_{in} = 450 \text{ K}$ $n_{ex} = 31.42\%$ $\dot{m} = 5.0 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{ex} = 36.22\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 600 \text{ K}$ $n_{ex} = 30.38\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{ex} = -61.87\%$ $\dot{m} = 2.0 \text{ kg/s}$
	150	$T_{in} = 450 \text{ K}$ $n_{ex} = 31.72\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{ex} = 36.19\%$ $\dot{m} = 2.0 \text{ kg/s}$	$T_{in} = 550 \text{ K}$ $n_{ex} = 32.51\%$ $\dot{m} = 0.5 \text{ kg/s}$	$T_{in} = 950 \text{ K}$ $n_{ex} = -63.12\%$ $\dot{m} = 2.5 \text{ kg/s}$