

OPTIMIZING PVT PERFORMANCE WITH MORPHING FINS DESIGN: A NUMERICAL ANALYSIS

BY

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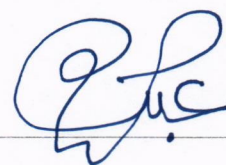
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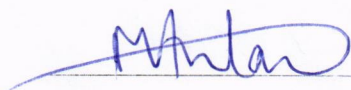
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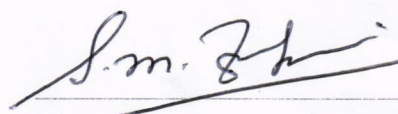
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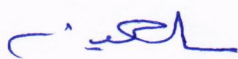
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Dedication

This thesis is dedicated to Almighty Allah (SWT) Who protected and guided me through
my life and also to my parents and family

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Alhamdulillah, praises and thanks to Allah (SWT), the Almighty, for His protection and blessings throughout my thesis period. May He Allah, bless all those that supported me during the period of my master's program.

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LIST OF ABBREVIATIONS

Chapter 1 and 2

PV	Photovoltaic
PVT	Photovoltaic-Thermal
PCM	Phase Change Material
CFD	Computational Fluid Dynamics
VHP	Variable Height Pin
CPV	Concentrated Photovoltaic
DBC	Direct Bonded Copper
IMS	Insulated Material Substrate
MF	Micro-Finned
FSPCM	Form Stable Phase Change Material
WCT	Water Container Thickness
HP	Heat Pipe
PEC	Passive Evaporative Cooling
CWIW	Cotton Wick Immersed in Water
XPS	Polystyrene Foam
TEG	Thermoelectric Generator
STEG	Solar Thermoelectric Generator
EF-HS	Evaporative Finned Heatsink
TPCT	Two-Phase Closed Thermosyphon
CTHP	Conventional Tubular Heat Pipe
LHP	Loop Heat Pipe
MCHP	Microchannel Heat Pipe
PHP	Pulsing Heat Pipe
LCPV	Low-Concentrated PV
FHP	Flat Heat Pipe
RC	Radiative Cooling
CLPHP	Closed-Loop Pulsating Heat Pipe
PRC	Passive Radiative Cooling
CRC	Commercial Radiative Coatings
SBR	Sub-Bandgap Reflection

MIE	Mid-Infrared Emission
RUR	Reflection of Ultraviolet Radiation
TC	Thermochromic Coating
DSO	Dimethyl Silicon Oil
LI	Liquid immersion
IA	Isopropyl Alcohol
DW	Deionized Water
EA	Ethyl Acetate
WF	Water Flow
TEC	Thermoelectric Cooling
SWCNT	Single-Walled Carbon Nanotube
CFM	Copper Foam
CR	Concentration ratio
OM	Organic mixture
RESC	Rapid Evaluation of Solar Panel Cooling
CuO	Copper Oxide
GNP	Graphene
PEG	Polyethylene Glycol
$\text{CaCl}_2 \cdot 6\text{H}_2\text{O}$	Calcium Chloride Hexahydrate
CO_2	Carbon dioxide
Si	Silicon
RT	Rubitherm
TPX	Methyl Pentane Copolymer
PDMS	Polydimethylsiloxane
SiO_2	Silicon Dioxide
TiO_2	Titanium Dioxide
T_{PV}	PV temperature
T_{PV-ref}	Reference PV temperature without cooling
T_{PV-Fin}	PV temperature with fin heatsink
T_{PV-PCM}	PV temperature with PCM cooling
T_{PV-PEC}	PV temperature with PEC
T_{PV-HP}	PV temperature with HP cooling
$T_{PV-PRC \text{ or } LI}$	PV temperature with PRC or LI cooling
$T_{PV-Air \text{ or } Liquid}$	PV temperature with forced air or liquid cooling

Chapter 3

RPF	Rectangular plate fin
BLSF	Bifurcated longitudinal split fin
VHGF	Variable height gradient fin
HPPF	Hybrid plate-pin fin
PPF	Perforated Plate Fin
n	Number of fins
t_f	Thickness of fins
S	Fin spacing
H	Fin height
t_b	Baseplate thickness
B	Heatsink width
L	Heatsink length
L^*	Bifurcation length
H^*	VHGF optimization height
H_p	Pin fin height
d_p	Pin fin diameter
P1	Circular perforation
P2	Square perforation
P3	Vertical rectangular slots
P4	Horizontal rectangular slots
$\mathbf{n}$	Boundary normal vector
$\mathbf{u}$	Velocity vector
q	Heat flux
U_o	Normal inflow velocity
$\mathbf{I}$	Identity matrix
$\mathbf{K}$	Stress tensor
ρ	Density
C_p	Heat capacity at constant pressure
T	Temperature
Q	Heat source
k	Thermal conductivity
K	Turbulent kinetic energy
ε	Turbulent dissipation rate

μ	Viscosity
μ_T	Turbulent viscosity
$C_{\varepsilon 1}$	Turbulent production constant for ε
$C_{\varepsilon 2}$	Turbulent dissipation constant for ε
σ_ε	Turbulent Prandtl number for ε
σ_k	Turbulent Prandtl number for K
C_μ	Turbulent viscosity constant
η	System performance
Nu	Nusselt number
P_d	Pressure drag coefficient

Chapter 4

ζ	Dimensionless temperature non-uniformity
P_R	PV rated power
P_{PV}	PV power output
D_F	PV derating factor
T_{ref}	Reference temperature
G_{ref}	Reference solar irradiance
G_T	Solar radiation incident on the PV
τ_p	PV power temperature coefficient
T_C	Cell temperature

ABSTRACT

Full Name : Ahmed Mohammed Bukar

Thesis Title : Optimizing PVT Performance with Morphing Fins Design: A
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Photovoltaic systems are vital to the global transition toward sustainable energy, but their efficiency is adversely affected by elevated operating temperatures, leading to reduced electrical performance and system reliability. This research investigates advanced thermal management solutions for solar photovoltaic systems, combining innovative finned heatsink designs with integrated air and water cooling strategies to address these challenges. The scope of the project includes a comprehensive literature review, the optimization of air-cooled fin designs, and the integration of hybrid cooling systems. Computational fluid dynamics simulations were employed to evaluate various configurations under concentrated and non-concentrated solar irradiance. Key findings indicate that advanced fin configurations, such as bifurcated longitudinal split fins integrated with water cooling, achieved significant PV temperature reductions of up to 36 K and 91 K under non-concentrated and concentrated conditions, respectively. For the concentrated irradiance, the integrated system improved electrical efficiency by 56%, increased power output by 54%, and achieved temperature uniformity within 1 K across the PV surface. This research addresses critical gaps in PV thermal management and lays a foundation for future research in economic feasibility, material innovations, and adaptive designs for diverse climatic conditions.

ملخص الرسالة

الاسم: الكامل: أحمد محمد بكر

عنوان الرسالة: تحسين أداء الأنظمة الكهروضوئية الحرارية من خلال تصميم الزعانف المتغيرة: تحليل عددي

التخصص: الهندسة الميكانيكية

تاريخ الدرجة العلمية: ديسمبر 2024

تُعد الأنظمة الكهروضوئية أساسية للانتقال العالمي نحو الطاقة المستدامة، إلا أن كفاءتها تتأثر سلبًا بارتفاع درجات حرارة التشغيل، مما يؤدي إلى انخفاض الأداء الكهربائي وموثوقية النظام. يبحث هذا البحث في حلول الإدارة الحرارية المتقدمة للأنظمة الكهروضوئية، من خلال الجمع بين تصميمات مبتكرة للمشتتات الحرارية ذات الزعانف واستراتيجيات تبريد الهواء والماء المتكاملة لمواجهة هذه التحديات. يشمل نطاق المشروع مراجعة شاملة للأدبيات، وتحسين تصميمات الزعانف المبردة بالهواء، وتكامل أنظمة التبريد الهجينة. تم استخدام عمليات محاكاة ديناميكيات الموائع الحسابية لتقييم التكوينات المختلفة تحت الإشعاع BLSF الشمسي المركز وغير المركز. تشير النتائج الرئيسية إلى أن تكوينات الزعانف المتقدمة، مثل الزعانف الطولية المتشعبة المدمجة مع نظام تبريد المياه، حققت تخفيضًا كبيرًا في درجات حرارة الوحدات الكهروضوئية يصل إلى 36 كلفن في الظروف غير المركزة و91 كلفن في الظروف المركزة. تحت الإشعاع المركز، حسن النظام المتكامل الكفاءة الكهربائية بنسبة 56%، وزاد إنتاج الطاقة بنسبة 54%، وحقق توحيدًا في درجات الحرارة ضمن 1 كلفن على سطح الوحدة الكهروضوئية. توضح النتائج إمكانيات أنظمة التبريد الهجينة بين الهواء والماء لتعزيز أداء الأنظمة الكهروضوئية، وتقديم حلول قابلة للتطبيق في البيئات عالية الإشعاع. يعالج هذا البحث الثغرات الحرجة في الإدارة الحرارية للأنظمة الكهروضوئية ويضع أساسًا للبحث المستقبلي في الجدوى الاقتصادية، الابتكارات المادية، والتصاميم التكيفية لظروف المناخ المتنوعة

CHAPTER 1

INTRODUCTION

1.1 Background

World energy demand has been on the rise due to the increasing population and global industrialization. The dominant source of this global energy demand will remain fossil fuels for at least the end of the century [1], [2], [3], [4]. However, due to fossil fuels' massive contribution to global carbon emissions, the world is undergoing a rapid transition to cleaner energy sources [5], [6], [7], [8]. Two-thirds of the world's energy needs can be met by renewable energy sources such as solar, wind, hydro, geothermal, and biomass, and these sources will also accelerate the reduction of greenhouse gas emissions between now and 2050 to keep the average global surface temperature below 2 °C [9]. Ali [10] reported that renewable energy has expanded significantly in the last two decades since it has been recognized as the most effective and efficient alternative with lower greenhouse gas emissions. Among these renewable sources, solar energy has the greatest potential worldwide [11].

The energy from the sun has the capacity to sufficiently meet the world's energy needs if significant advancements in its harvesting technologies are achieved. About 70% of the solar energy that the world receives is thought to be harvestable, while 30% is diffused or reflected [12]. The energy of the sun is predominantly converted into thermal energy using thermal collectors and electrical energy using photovoltaics. In thermal collectors, the irradiation enters the collector in the form of heat, which then transfers to a working fluid

like water or air [13]. However, most of the research interest are currently geared toward producing power using photovoltaic (PV) devices. PV devices employ the photoelectric effect to convert solar radiation into electricity using a solid-state semiconductor cell. To generate appreciable power, a number of these cells are merged to form the PV module [14].

PV systems are pivotal in the global transition toward renewable energy, offering a sustainable and clean energy source. However, their performance and efficiency are significantly affected by operational temperature increases, which reduce power output and system reliability. Effective thermal management is critical for mitigating these effects, ensuring stable and efficient PV operation. Numerous passive and active cooling techniques have been proposed, such as fin heatsinks, water-based cooling systems, and hybrid methods, each addressing different aspects of PV thermal challenges. Despite these advancements, there remains a need for innovative, cost-effective, and scalable solutions that balance thermal efficiency with practical implementation. This thesis explores advanced cooling designs, including fin optimization and integrated air and water cooling systems, to address these challenges comprehensively.

1.2 Problem statement

Despite being the predominant method for harnessing solar energy, PV technology faces numerous challenges that diminish its conversion efficiency. The performance of PV modules is influenced by ambient variables, including temperature, dust, cloud cover, wind speed, and irradiance levels; cell technology; optimal material selection; and the necessity for maximal sun irradiation to achieve high performance [15]. The primary issue is the module temperature, which directly influences the overall system efficiency. A reduction

in module temperature will lead to a contraction of the semiconductor cell band gap, attributable to the temperature dependency of the p-n junction voltage and the diode factor. The efficiency of the PV module is inversely related to its temperature as shown in Figure 1.1 [16]. The temperature of a PV module has a significant impact on its intrinsically low energy conversion efficiency, even under optimal operating conditions. Between 13% and 20% of the solar energy immediately incident on the panel is converted into electricity, while the remainder is dissipated as thermal energy, leading to elevated operating temperatures and thus reduced efficiency [18]. The buildup of elevated temperatures in PV modules can result in reduced lifespans, hotspots, and low efficiency [19]. The adverse impact of elevated temperatures in PV modules can be mitigated by implementing an effective thermal management mechanism that ensures the economic sustainability and reliability of PV systems.

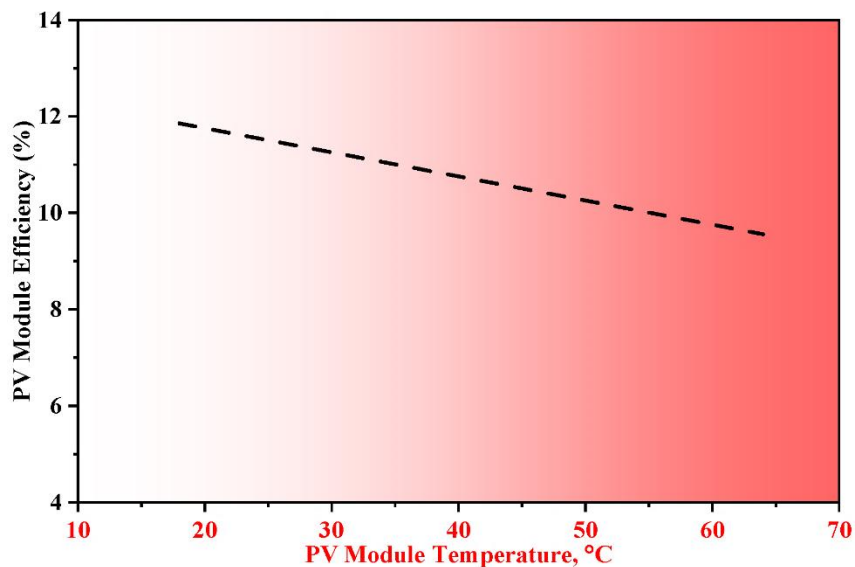


Figure 1.1: Temperature dependence of PV module efficiency [17]

1.3 Aim and objectives

The aim of this study is to use numerical methods to analyze and enhance the thermal efficiency of a photovoltaic module. This will be achieved by integrating a water-cooling channel within a flat plate (PVT) that includes extended surface (fins). The goal is to create a sustainable cooling system. To accomplish this goal, the following primary objectives are outlined:

1. To conduct a comprehensive review of existing literature on PV cooling systems involving both active and passive cooling.
2. To numerically analyze the convective flow across the fins of a heated plate heatsink with focus on the effect of fin type and distribution on the performance of the heatsink.
3. To numerically analyze several PV cooling configurations comprising the PV module, the cooling pipe (PVT) and the heatsink (Fins) achieved in objective two, and then identify the most efficient design.
4. To analytically evaluate the electrical performance of the developed PVT-Fin system.

1.4 Scope of the thesis

The scope of work within this research is limited to the following determinants:

1. **Numerical Simulations:** Using computational fluid dynamics (CFD) models to simulate and analyze cooling configurations under different operational scenarios using the commercial software COMSOL Multiphysics.

2. **Performance Metrics:** Evaluating PV configurations based on key parameters such as operating temperature, temperature non-uniformity, power output, electrical efficiency, and overall performance.
3. **Applications:** Investigating both standalone and PV integrated with cooling systems for their potential deployment in practical PV applications.

1.5 Significance of the study

This study makes a multifaceted contribution to advancing the thermal management of PV systems through a combination of rigorous literature review, innovative fin heatsink optimization, and integrated cooling methodologies. Each part of the thesis provides distinct and valuable insights, collectively addressing critical challenges in PV system efficiency and reliability. The significance of the study is discussed below from the perspectives of the three parts:

1. Literature Review:

The comprehensive review of existing thermal management techniques for PV systems provides a clear understanding of the current state of the field. By critically analysing passive cooling strategies, such as fins and coatings, and active methods, such as water cooling and hybrid approaches, the review identifies key strengths, limitations, and knowledge gaps in existing solutions. This foundational work not only guides the subsequent parts of the thesis but also serves as a valuable resource for researchers aiming to advance PV cooling technologies.

2. Optimization of Fin Heatsinks:

The second part of the thesis develops and evaluates innovative fin configurations, such as bifurcated longitudinal split fins (BLSF), hybrid plate-pin fins (HPPF), and variable height gradient fins (VHGF), among others. These designs push the boundaries of traditional fin-based cooling by introducing advanced geometries that enhance thermal boundary layer disruption and convective heat transfer. The study provides practical engineering insights into the trade-offs between thermal performance and hydraulic resistance, paving the way for more efficient heatsink designs in high-performance PV systems.

3. Integrated Air and Water Cooling:

The third part investigates the synergistic effects of combining air- and water-cooling strategies, addressing limitations inherent in standalone cooling systems. By integrating water cooling channels with advanced fin designs, this study achieves significant improvements in thermal performance while maintaining manageable hydraulic resistance. This work is particularly significant for high-irradiance environments, such as concentrated PV systems, where traditional cooling methods often fall short.

1.6 Thesis organisation

This research thesis is organized into five chapters, each addressing a specific aspect of the study to provide a comprehensive understanding of the work. The following is a brief description of each chapter:

1. Chapter One: Introduction

This chapter introduces the background and motivation for the research, emphasizing the increasing importance of renewable energy and the challenges

associated with photovoltaic (PV) system efficiency due to thermal management issues. It provides an overview of the research problem, objectives, and scope, highlighting the significance of advanced cooling strategies in improving PV performance. The chapter also outlines the contributions of this study to the fields of renewable energy and thermal management.

2. Chapter Two: Literature Review

This chapter presents a comprehensive review of the current state-of-the-art in PV cooling technologies. It begins with an introduction to PV systems, including their types, efficiency challenges, and thermal limitations. The chapter then explores passive cooling methods, such as fins and coatings, and active cooling techniques, including water-based and hybrid systems. It critically analyzes the strengths and limitations of these methods and identifies research gaps that motivate the work presented in this thesis. The chapter establishes the foundational context for the subsequent research and development of advanced cooling configurations.

3. Chapter Three: Optimizing Fin Heatsinks for Photovoltaic Cooling

This chapter details the research and methodology for optimizing innovative fin heatsinks for PV cooling. It describes the design and evaluation of advanced fin configurations, such as bifurcated longitudinal split fins (BLSF), hybrid plate-pin fins (HPPF), and variable height gradient fins (VHGF), using CFD simulations. The chapter also presents results on thermal performance metrics, including baseplate temperature and thermal resistance, as well as the trade-offs between heat dissipation and hydraulic resistance. The findings contribute to improving passive cooling strategies for PV systems.

4. Chapter Four: Integrated Fin Heatsink and Water Cooling Technique for Solar PV

This chapter explores the integration of air- and water-cooling strategies to address the limitations of standalone cooling methods. It describes the numerical modeling of hybrid cooling systems that combine advanced fin configurations with water channels to enhance thermal and hydraulic performance. The chapter presents detailed analyses of temperature reduction, pressure drop, and overall performance under various operating conditions, including concentrated and non-concentrated solar irradiance.

5. Chapter Five: Summary, Conclusion and Recommendations

This chapter summarizes the key findings of the study, highlighting the contributions of each chapter to the overarching research objectives. It discusses the implications of the results for improving PV efficiency and reliability, offering practical insights into the design of cooling systems for renewable energy applications. The chapter concludes with recommendations for future work, including experimental validation of the proposed designs, exploration of novel materials, and further optimization of integrated cooling systems.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

PV thermal management plays a crucial role in enhancing the performance and lifespan of PV systems by mitigating the negative effects of elevated operating temperatures. Recent advancements in PV cooling have highlighted various methods to reduce maximum temperatures, improve electrical efficiency, and boost power output. Several review articles have examined specific cooling strategies, such as phase change material (PCM)-based cooling by Hafiz Ali [20], heat pipe technologies by Gao et al. [21], and fin-based cooling approaches by Kazem et al [22]. Despite these efforts, a need remains for an integrated review that quantitatively evaluates the performance of different PV cooling techniques. PV cooling methods are broadly categorized into passive and active techniques,

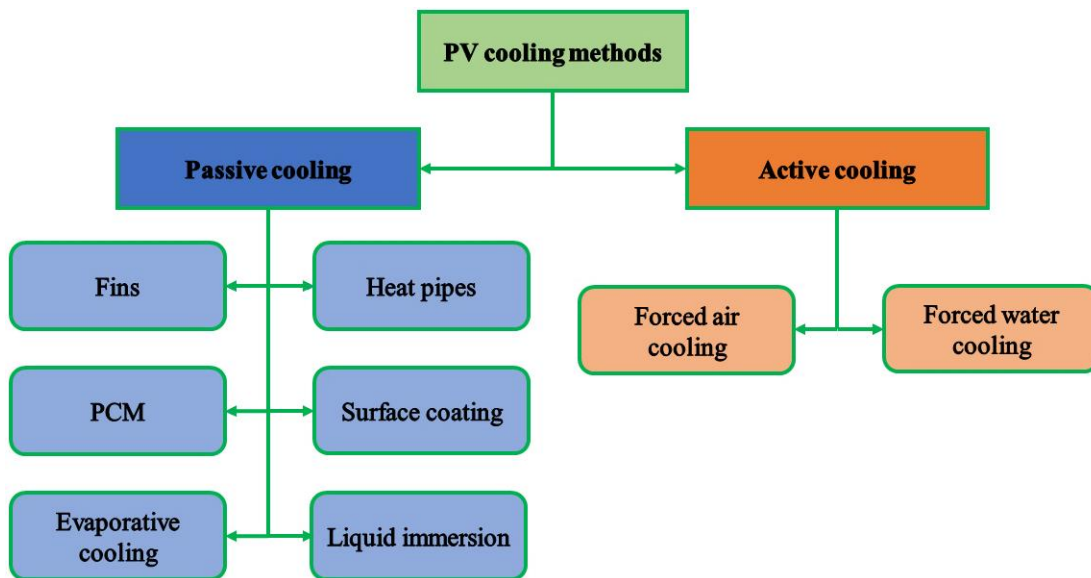


Figure 2.1: Classification of PV cooling techniques

as illustrated in Figure 2.1. Passive cooling methods rely on ambient conditions and require no external energy input while active cooling methods employ mechanical systems to enhance heat dissipation.

2.2 Passive cooling using extended surfaces (fins)

2.2.1 Overview of previous work

The most recent research performed by Selimefendigil et al. [23] comparatively scrutinized three distinct PV cooling techniques, namely hollow fins based on natural convection, porous media, and wavy channels based on forced convection. The elliptical-shaped hollowed fin, as shown in Figure 2.2(b), offers a light weight and relatively low temperature reduction compared to the other methods. In comparison to the reference PV panel, a temperature reduction of 33.6 °C, 37 °C, and 37.6 °C was recorded for fin, wavy channels, and porous media, respectively. Hernandez-Perez et al. [24] investigated the use of passive techniques for cooling PV panels using natural convection with the objective of achieving low operational temperatures and increased efficiency. The proposed heatsink model has fins with a segmented profile every 2 cm, summing up to 16 fins per profile. The authors found that, in their modelling and experiments, the temperature dropped by up to 10 °C maximum during peak solar irradiance leading to a potential 4% boost in the system's electrical efficiency. Elbreki et al. [25] reported on a method to passively regulate the temperature of PV modules using truncated fins. In this numerical study using ANSYS Fluent, either the number, height, or thickness of fins are morphed while keeping the other fin parameters and reflector constant. The optimum height and number of fins were reported to be 150 mm and 12 mm, respectively, while a temperature reduction of about 20 °C was achieved at this configuration.

Raina et al. [26] assessed the effect of using augmented fins on power generation using PV modules in hot and cold climates. The study shows that incorporating a cooling system on a PV panel in a cold climate almost results in no significant enhancement in the electrical output. However, in hot climates, a 5.47% improvement in overall efficiency was recorded. This study was further extended by the same authors [27] to assess the hydraulic effects of incorporating discontinuous fins, where pressure losses were observed to drop by 49% using a segmented fin profile. Furthermore, a study to analytically predict the temperature of PV modules supported by passive cooling, as shown in Figure 2.2(a), was conducted by AlAmri et al. [28]. The authors developed an analytical model with a crosslink between the panel temperature and the influx of solar radiation, yielding a closed-form equation for readily obtaining the efficiency, power, and temperature of a photovoltaic system as a function of the fin characteristics and wind speed. To validate the model, it was tested for the weather in Dammam, Saudi Arabia. The results came as expected, showing improved panel power of 8.7% and 6.5% in the summer and winter seasons, respectively.

The effect of fin parameters, as shown in Figure 2.3 is of great importance when designing conducting plates like PV modules [29]. Several studies have investigated their effect on PV performance and identified the optimum parameters for their system. Nehari et al. [30] numerically investigated the effect of fin length on the performance of PV panels by performing a parametric study. The authors deduced that the fin configuration with lengths $L = 25, 30, \text{ and } 35 \text{ mm}$ offers better heat rejection from the PV panel to the atmospheric air through natural convection. By planting 4, 6, 8, and 10 longitudinal fins at the back of a PV module, Farhan et al. [31] assessed the thermal performance of the module. Ten fins deployed at solar noon resulted in a 2.5 W power improvement.

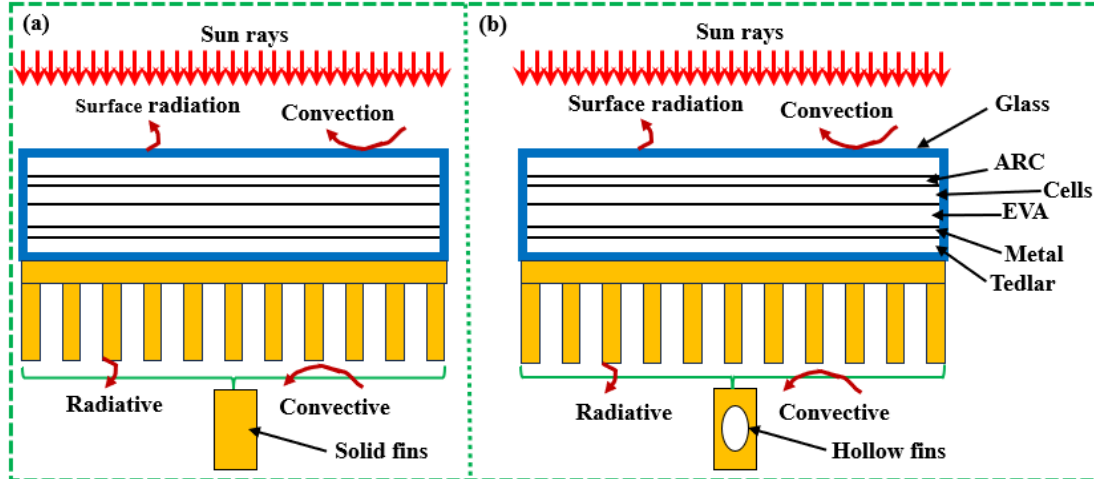


Figure 2.2: (a) Solid fin implemented by [28] , (b) Hollow fin implemented by [23]

In another Turkish experiment, Ekiz et al. [32] examined how fin surface affects photovoltaic module efficiency. A continuous rectangular fin with a flat and threaded surface was compared to an uncooled PV module. For threaded and flat fins, module temperature decreased by 16.5% and 11.78%, respectively, improving power output by 17.2% and 10.84%.

Many experiments have examined the effect of fixed fins on the PV module's temperature. Grubisic-Cabo et al. [33] fixed aluminium fins to the PV module's back and observed its performance in natural conditions. Non-perforated longitudinal and randomly dispersed perforated aluminium fins are used. Perforated fins increase power output by 2%, according to the study. Similar research was done by Ahmad et al. [34] using experimental continuous rectangular fins. The peak module temperature dropped by 5.77 °C and power output increased by 14.2%. Amber et al. [35] examined PV module efficiency with two fin-based cooling methods. The study used rectangular and threaded pin fins on the module back. The authors found that rectangular-finned PV panels dissipated 155% more heat. It increased power by 10.8% over the baseline module and 4% over the pin-finned module.

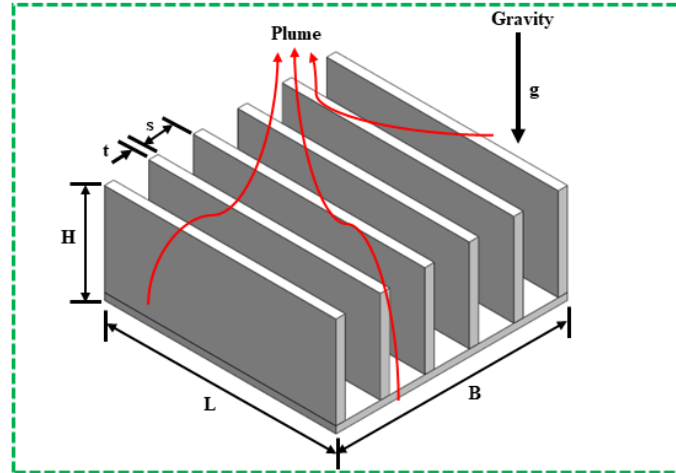


Figure 2.3: Natural convection from rectangular fin [37]

Thus, the module's temperature dropped by 10.6%, increasing electrical efficiency by 14.5%. Another study by Grubisic-Cabo et al. [36] involved experimental and numerical fin heatsink integration on stand-alone PV panels.

Elbreki et al. [38] tested PV module efficiency with lapping/longitudinal fins and reflectors. Lapping fin passive cooling lowers PV module temperature by 24.6 °C compared to the reference module. This produces 10.68% higher electrical efficiency and 37.1 W of power. An economic analysis using life cycle cost analysis showed that PV modules with longitudinal fins had 4 years payback period, lapping fins 5 years, and no fins 8 years. So, passive cooling with lapping fins became the optimal option. In another study [39], metal mesh was considered instead of fins. The authors compared aluminum and iron meshes to a generic rectangular fin heatsink and a base case PV without cooling. Aluminum meshes dissipate more heat than iron meshes but less than fins. Experiments with an aluminum and iron mesh heatsink show a 6.56 °C and 4.35 °C temperature drop. Ahmad et al. [40] experimentally designed a heatsink for enhancing uniformity of a standalone PV module temperature. At outdoor operating condition, the module

temperature reduced by 8.45 °C. Consequently, the output power improved by 9.56%. Krstic et al. [41] experimentally incorporated small heatsinks of different geometry on PV panel. The authors recorded that a reduction in module temperature averaging 7.5 °C is possible. Gomaa et al. [42] experimentally investigated the effect of U-shaped fins on the efficiency of grid-tied PV module under passive cooling. The findings revealed that the U-shaped fins reduced the working temperature slightly from 58 °C to 55 °C. Razali et al. [43] explored the possibility of incorporating multi-directional tapered fin heatsink on PV module using aluminum alloy as fin material. The novelty is to improve the flow of air at the center of the module which in turn improves the convection heat transfer coefficient. It was found that the proposed design can reduce the working temperature of the module by 12 °C, thus, increasing the efficiency by 1.53%. Arifin et al. [44] integrated aluminum fin-based heatsinks numerically and experimentally on a PV module. By employing Computational Fluid Dynamics (CFD) simulation using ANSYS on the heatsink model, operating at an air flow rate of 1.5 m/s and a temperature of 35°C, and exposed to a heat flux of 1000 W/m², a notable reduction in the mean temperature of the PV panel was observed, decreasing from 85.3 °C to 72.8 °C. This decline in temperature, in turn, yielded a substantial 10% increment in the open circuit voltage and an 18.67% enhancement in the maximum power point of the module. Mankani et al. [45] computationally optimized a fin-based heatsink for cooling PV modules. The study analyzed the effect of morphing fin thickness, spacing, and height on the heat transfer rate in comparison to an uncooled base case module. The results revealed that the most optimal fin arrangement recorded a 26.8 °C reduction in operating temperature from 68.8 °C to 42 °C.

To closely examine the effect of passive cooling on PV system, several unconventional Fin integrations are proposed. Some authors are narrowing the research to incorporating fins at the single silicon cell level instead of the compact PV module as demonstrated by Cuce et al. [46] experimentally. The results indicate that the efficiency of the PV cell considerably increase with the proposed cooling technique. An increase of 20 % in power output of the PV cell is achieved at 800 W/m² radiation condition. Maximum level of cooling is observed for the intensity level of 600 W/m². Furthermore, Ansari et al. [47] used variable height pin fin (VHP) heatsinks on densely packed CPV to reduce heat dissipation nonuniformity along the flow direction in water-cooled and conventional fin heatsinks. Numerous instances of the proposed variable height pin fin heatsink were investigated to gradually reduce heat transfer near the inlet region and maintain heat transfer near the outlet zone as close to the uniform height pin fin heatsink as possible. Most notably, the VHP design reduced temperature disparity by 1.56 K compared to the uniform height pin fin heatsink (9.82 K). The optimal VHP heatsink reduced fluid circulation energy by 33.1%. The effect of ambient temperature on the performance of PV modules is not only prevalent in conventional PV modules but also in CPV. In their quest to ascertain the threshold of passive cooling mechanisms and to discern the circumstances necessitating an active cooling system, Alzahrani et al. [48] examined the performance limits of a concentrator solar cell and micro heat dissipation unit at the operational temperature. In the numerical investigation using COMSOL Multiphysics software, silicon wafer (Si wafer), direct-bonded copper (DBC), and insulated material substrate (IMS) were examined with a micro-finned (MF) heat dissipation system as shown in Figure 2.4. Various solar concentrations were considered at the recommended 80 °C maximum operating

temperature. Compared to the flat plate heatsink, the MF heatsink lowered the IMS, Si wafer, and DBC temperatures by 55.43, 56.07, and 57.31 °C under 1000X.

A similar study conducted by Luo et al. [49] designed a CPV cooled with flared-fin heat. Within this investigation, an enhanced heatsink was introduced to fulfil the demanding heat dissipation necessities associated with CPV. Findings indicate that the comprehensive thermal resistance of the heatsink is notably reduced by 10%. Zaghloul et al. [50] studied unconventional PV module thermal management. As shown in Figure 2.5, a heatsink with flat plate, flared, straight, and pin fins passively cools a triple-junction cell under high solar concentration. A heatsink allowed the module to operate at 1775X solar concentration, compared to the maximum of 77X at ideal conditions (uncooled). Flared heatsinks removed more heat than rectangular and pin fins. Li et al. [51] added high-thermal-conductivity fins to a bi-facial PV module in subtropical climates like Dubai, the natural convection-cooled fin can lower the bifacial module's maximum operating temperature by 13.6 °C.

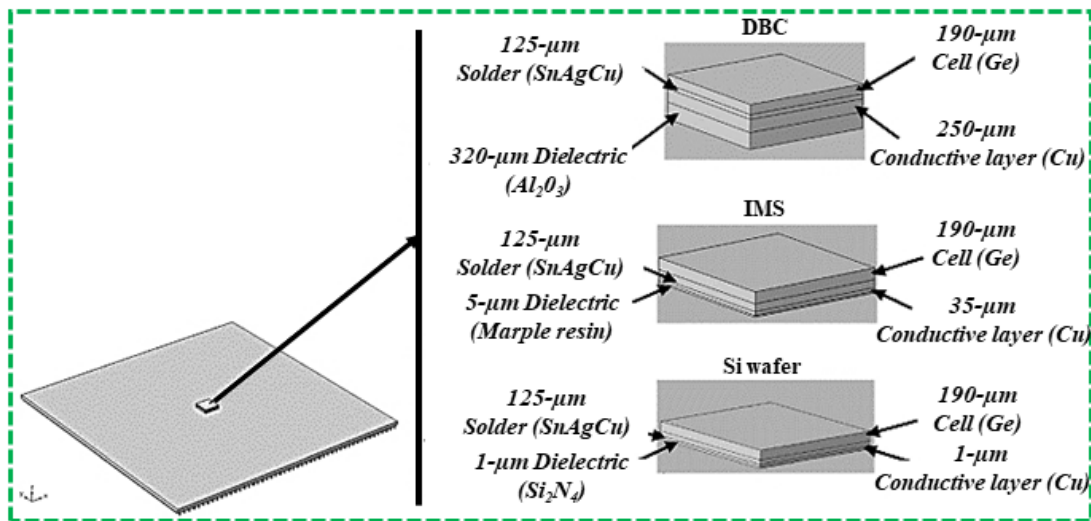


Figure 2.4: CPV-heatsink geometry showing layers of DBC, IMS and Si Wafer substrates [48]

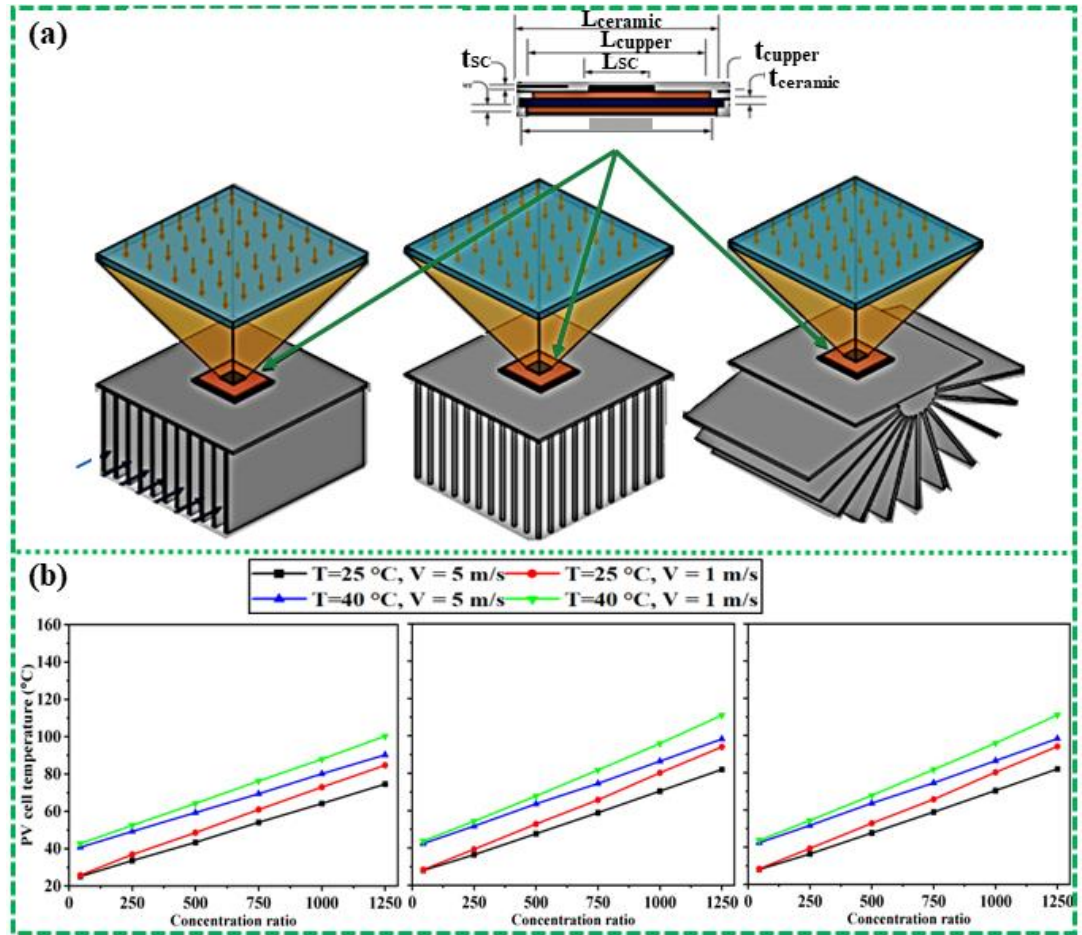
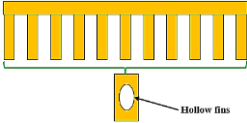
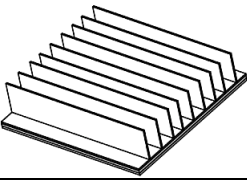
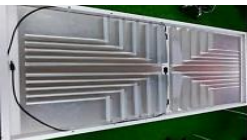
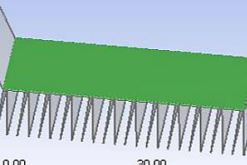
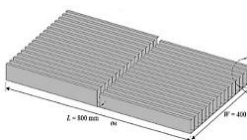
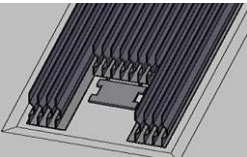


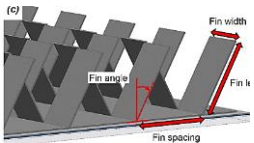
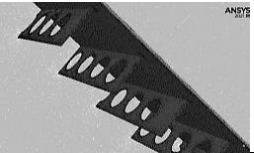
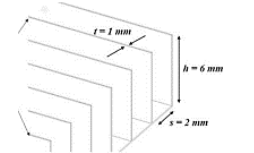
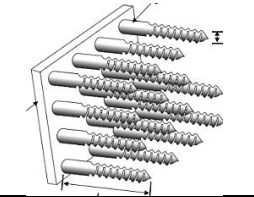
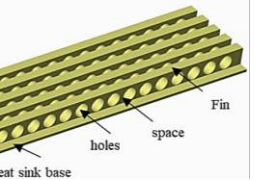
Figure 2.5: CPV incorporated with (a) Rectangular, pin, and flared fin, (b) Maximum cell temperature [50]

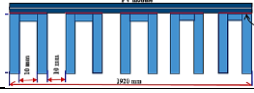
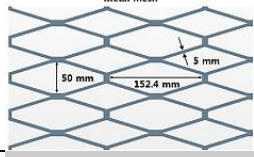
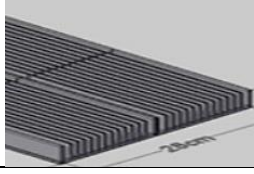
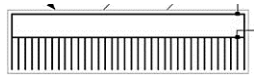
2.2.2 Discussion on passive cooling using extended surfaces (fins)

The fin heatsink efficiently disperses excess solar heat that the PV module cannot convert into electricity. Avoiding forced convection usually leads to a more economical and smaller system. Rectangular fins are the predominant fin geometry for passive cooling in the literature, as shown in Table 2-1. Cuce et al. [46] were selected to represent rectangular fins in Figure 2.6 due to reporting the highest temperature drop for that geometry. The lapping fins [38] achieved the highest temperature decrease of approximately 22 °C, while the truncated fins followed closely with a reduction of about 20 °C.

Table 2-1: Summary of work proposed work by different researchers for cooling PV module using fin

Reference (Year)	Types of fins	Fin sample	Morphing parameters	Nature of work	Maximum $T_{PV-ref} (^{\circ}C)/$ $T_{PV-Fin} (^{\circ}C)$	Key findings
Selimefendigil et al. [23] (2023)	1. Flat fin 2. Hollow fin		- Number of fins - Radius of hollow	Numerical	1. 67/38 2. 67/33.4	Hollow fins outperform flat fins, achieving an 8.5 °C temperature reduction.
Ekiz et al. [32] (2023)	Rectangular plate		- Fin surface 1. Threaded 2. Flat	Experimental	1. 44.2/37.44 2. 44.2/39.75	- Threaded surfaces dissipate more heat than flat surface. - PV temperature reduced by 16.52% and 11.78% respectively. - Efficiency improved by 1.95% and 1.21 % respectively.
Razali et al. [43] (2023)	Multi-directional tapered		Fin height	Experimental	69/57	- The temperature is reduced by 12 °C. - Electrical efficiency improved by 1.53 %.
Elbreki et al. [25] (2022)	Truncated fin		1. Number of fins 2. Fin thickness 3. Fin height	Numerical	64.3 /44.14	- PV electrical efficiency improved by 9.2 %. - PV temperature reduced by 20.16 °C. - Fin thickness above 2 mm has negligible effect on PV performance.
Ahmad et al. [40] (2022)	L-shaped multi-level		--	Experimental	64.2/56.1	- Maximum temperature reduction of about 8.45 °C. - Heatsinks are more efficient at irradiation > 650 W/m²
Elbreki et al. [38] (2021)	1. Lapping 2. Rectangular		--	Experimental	64.3/41.6	- PV module temperature reduced by 24.6 °C. - The optimum number of fins (and payback period) for longitudinal and lapping is 15 (4.2 years) and 18 (5 years) respectively.

Reference (Year)	Types of fins	Fin sample	Morphing parameters	Nature of work	Maximum $T_{PV-ref} (^{\circ}C) /$ $T_{PV-Fin} (^{\circ}C)$	Key findings
Hernandez- Perez et al [27] (2021)	Straight interleaved		Fin width	Numerical Experimental	45.02/37.74	- Temperature reduction up to 7 °C - Hydraulic performance is improved by reducing 49 % of the pressure losses
Hudisteanu et al. [52] (2021)	Rectangular perforated		- Fin arrangement - Fin perforation	Numerical	62.9/51.27	- The power production of the PV panel rises by 6.49% by incorporating perforated fin. - Efficiency dropped linearly with rise in temperature at about 0.37%/°C
Ahmad et al. [34] (2021)	Rectangular plate		--	Experimental	56/50.23	- The maximum temperature reduction recorded was 5.77 °C. 14.2 % overall improvement in power output
Amber et al. [35] (2021)	- Rectangular plate - Threaded pin		Type of fin	Experimental	64/58 64/62	- Rectangular fins are more efficient than threaded pin fin. - Rectangular fin produced 10.8 % and 4 % more power than unfinned and pin fin PV module.
Hernandez- Perez et al [24] (2020)	Multi-angular interleaved		- Fin length - Wind direction	Experimental Numerical	47.92/38.49	- Mean temperature reduction of 9.4 °C (19.67 %) - Electrical efficiency improved by 4%
Egab et al. [53] (2020)	Rectangular fins with holes		- Number of fins - Number of holes - Hole distance	Numerical	55.56/46.1	- The air heatsinks significantly reduced the panel temperature. - The use of fins reduced the PV temperature by 12 %

Reference (Year)	Types of fins	Fin sample	Morphing parameters	Nature of work	Maximum T_{PV-ref} (°C)/ T_{PV-Fin} (°C)	Key findings
Farhan et al. [31] (2020)	Rectangular plate		Number of fins	Experimental		- Power enhanced by 2.5 W maximum. - Significant reduction in temperature was recorded before solar noon with passive cooling.
Gomaa et al. [42] (2020)	U-shaped fins		--	Experimental	58/55	- Maximum temperature reduction is 3 °C. - Power output enhanced by 7 %.
Bayrak et al [54] (2019)	Staggered array		Fin arrangement	Experimental	61.02/51.29	The heat transfer was improved by applying thermal paste between the fin and panel.
Kim et al.[39] (2019)	Metal mesh		- Material 1. Aluminium 2. Iron	Experimental Numerical	1. 52.72/46.16 2. 52.72/48.37	- Maximum temperature drop recorded is 6.56 °C and 1.44% increase in efficiency. - Fins demonstrates higher heat dissipation than meshes.
Mays et al. [55] (2017)	Rectangular fins		--	Experimental	56/49.9	- The integrated system enhanced the PV electrical efficiency by 1.77%. - The system output power is also improved by 1.86 W. - Maximum temperature drop recorded was 6.1 °C.
Cuce et al. [46] (2011)	Rectangular fins		--	Experimental	67.9/50.4	

Elbreki et al. [25] achieved the highest increase in efficiency by adding truncated fins to the PV module, as depicted in Figure 2.7. The U-shaped, aluminum mesh, and staggered arrays showed the lowest improvement in efficiency. This could be owing to the limited convection area provided by the narrow fins used by the authors, as seen in Table 2-1.

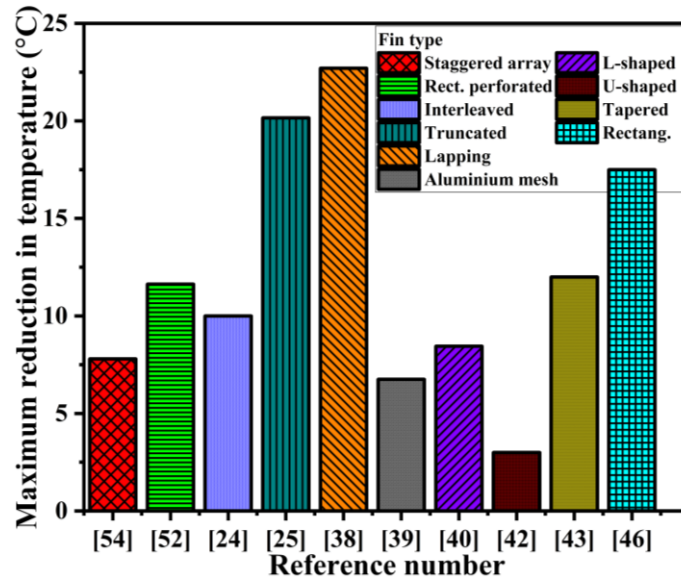


Figure 2.6: Maximum reduction in temperature using fins compared to non-cooled PV

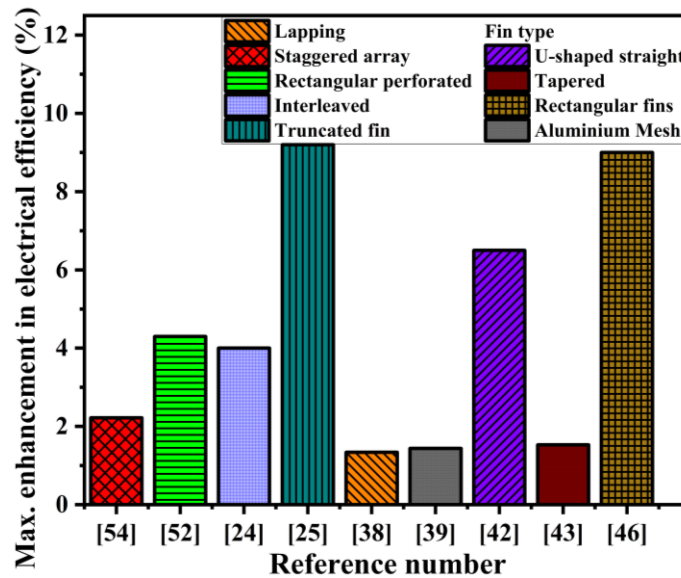


Figure 2.7: Maximum enhancement in electrical efficiency using fins compared to non-cooled PV module

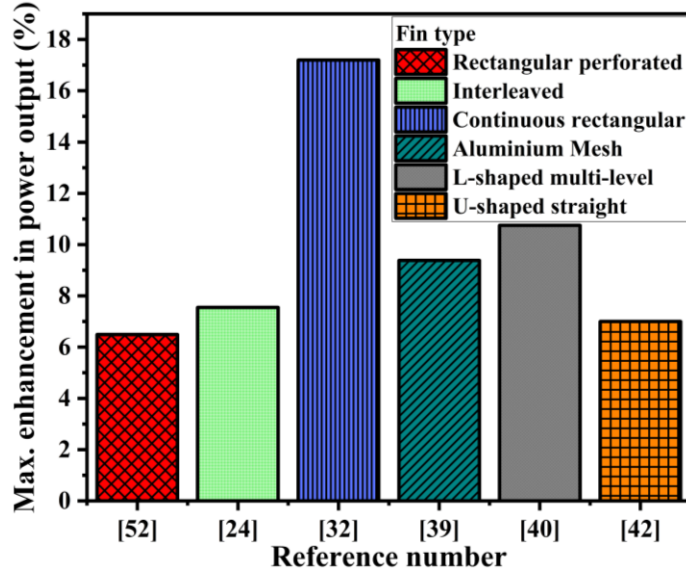


Figure 2.8: Maximum enhancement in power output using fin compared to non-cooled PV module

Ekiz et al. [32] achieved a 17% increase in power production by adding a continuous rectangular fin to the PV module. Rectangular perforated and U-shaped fins in Figure 2.8 resulted in minimal power increase, whereas aluminium mesh and L-shaped multi-level fins showed moderate power output improvement.

2.3 Passive cooling using phase change material (PCM)

2.3.1 Overview of previous work

Hassabou and Isaifan [56] developed a thermal management system using a layer of PCM attached at the back of a PV module serving as heat absorber. The numerical study performed using COMSOL Multiphysics successfully illustrated a simplified method of simulating melting and time record of PCM temperature using averaging volume homogenization and thermal capacity formulation methods and the numerical results have been validated against analytical results. Because of the changing PV orientation in sun-tracking PV system, Adibpour et al. [57] observed that the PCM melting kinetics during

tracking are significantly different from those of fixed nonrotating PV module. In the work, calcium chloride hexahydrate was used as the PCM attached to the PV module using aluminium container. The authors observed that, at solid phase, there is rapid change in the PCM temperature while at PCM melting point, the temperature is constant. Also, on average, about 7 °C reduction in the module maximum temperature was recorded. Furthermore, similar study without sun tracking using the same PCM was earlier reported by Rezvenpour et al. [58], the authors recorded 16.1 °C drop in peak temperature of the PV. Singh et al. [59] investigated how container material affected heat transfer from the module to the PCM in Chennai, India, during different seasons. As container height decreases, PV operating temperature decreases. Another experiment by Velmurugan et al. [60] made a 75:25 eutectic mixture of lauryl and ethyl alcohol. The eutectic combination melts at 21.75 °C and has 199 J/g latent heat of fusion. 3 and 5 cm C-PCM containers are on the PV back. Maximum PV temperature (T_{PV}) dropped by 10.3 °C due to higher heat transmission in the 5 cm container. This T_{PV} reduction increased average voltage by 2.85% and output power by 2.8%. Stropnik et al. [61] used Rubitherm-28 (RT-28) PCM in a container-filled PCM study. Figure 2.9 shows Ahmad et al. [62]'s study on the impact of PCM container shape and structure. The trapezoid PCM container cooled better than the rectangular one.

Cooling increases PV voltage and power production [63]. In typical applications, cooling solutions only help if the profit outweighs the costs. Japs et al. [64] examined PCM passive cooling's economics and technology. Mainly negative energy and economic results were observed daily. The study found that higher morning power costs and energy yields do not guarantee profitability.

A PCM with better thermal conductivity is better for PV modules. Thus, recent studies emphasize on PCM properties. Marudaipillai et al. [65] used form stable-phase change material for PV thermal management due to its high latent heat of fusion and form stability. The finned composite PCM-based PV panel lowered module temperature by 11.5 °C. In the conventional single container filled with PCM configuration, as demonstrated by Adibpour et al. [57], there is a significant challenge of containment when the PV is tilted since the melting is not homogeneous from the top to the bottom of the container. By dividing the container into compartments, Nizetic et al. [66] solved this problem and improved PV heat dissipation. A new trend is to use multi-layered PCM, like the one shown in Figure 2.10, instead of single compartment. This is explained by Ranawade et al. [67]. The multilayered PCM enables effective thermal control across all seasons.

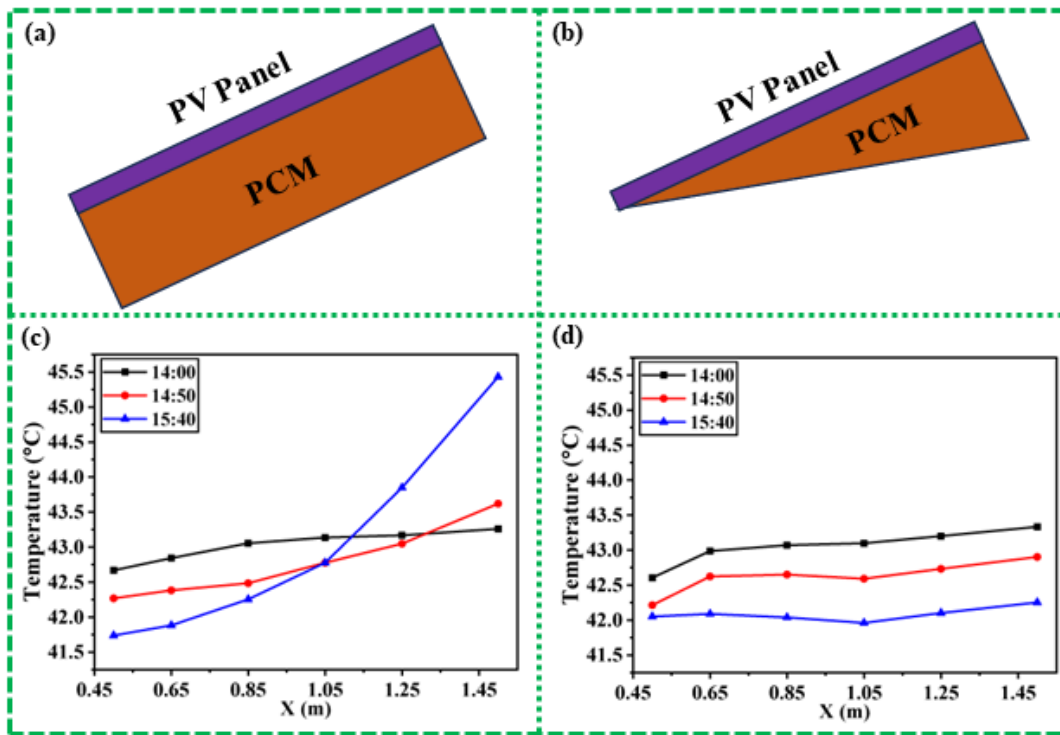


Figure 2.9: PCM container geometry (a) rectangular, (b) trapezoidal, and temperature (c) rectangular, (d) trapezoidal [62]

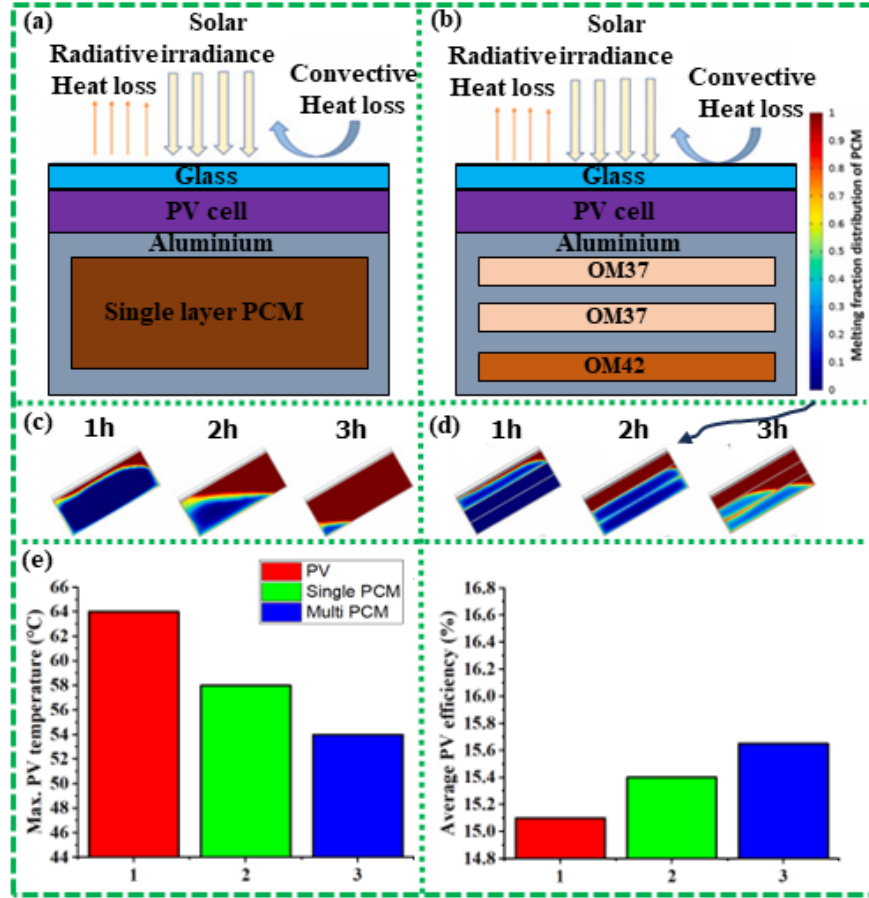


Figure 2.10: Configuration (a-b), and melting fraction (c-d) for (a) Single layer (b) multi-layer, and (e) PV temperature and efficiency [67]

The clever combination of PCMs with both low and high melting points is what makes this possible. Experimental results showed a 4 °C and 7.2 °C drop in maximum temperature for the multi-layered PCM module compared to the single-layer PCM and reference PV. Elsabahy et al. [68] demonstrated a technique for confining PCMs by encapsulating them within various enclosures as shown in Figure 2.11. The findings suggest that the implementation of triple encapsulation leads to a reduction in the mean cell temperature by 16 °C and a reduction in the non-uniformity index by 24 °C.

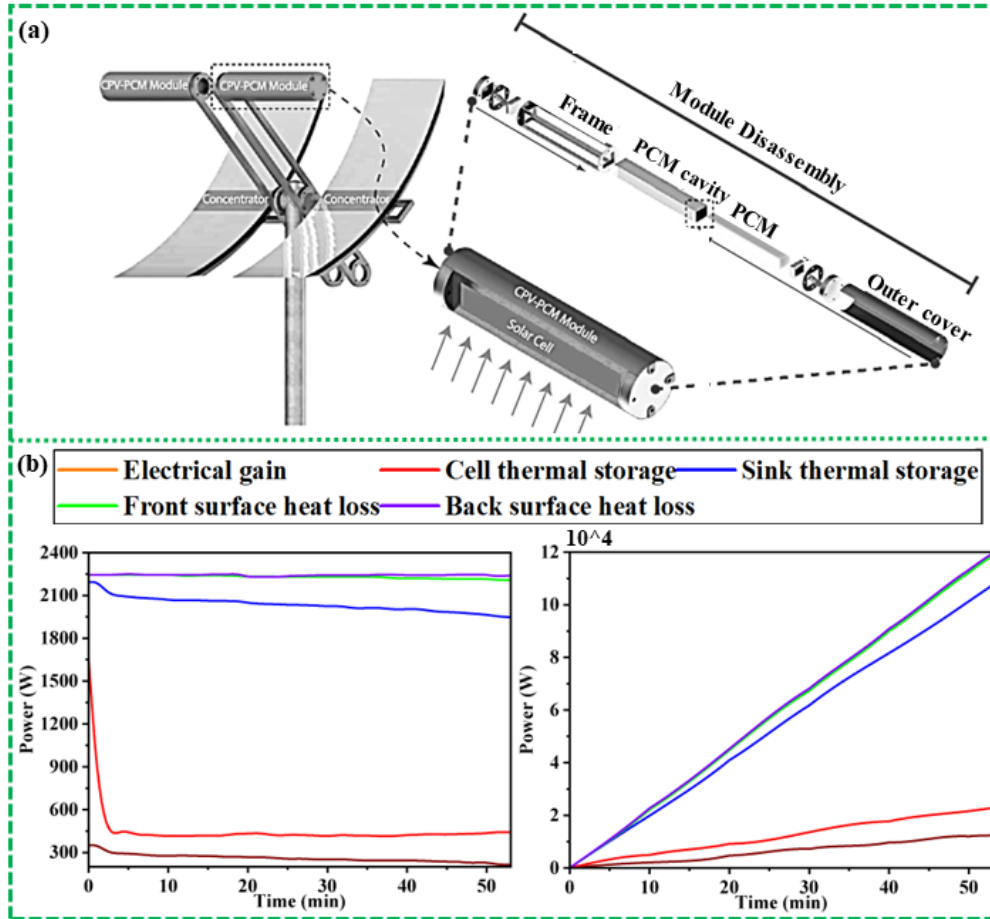


Figure 2.11: (a) Configuration for the LCPV-PCM system, (b) Shares of energy [68]

The effect of tilt angle on the melting behaviour of PCM and its influence on the overall performance of PV modules is investigated by Abdulmunem et al. [69]. The study examined tilt angles of 0, 30, 60, and 90 degrees. When the PV-PCM system tilt angle increases from 0° to 90°, PCM melting time decreases. Thus, this decrease in time lowers PV cell temperature by 0.4% to 12%. Lower tilt angles restrict PCM cooling performance. This is due to passive cooling via natural convection in the PCM container. Bhakre et al. [70] numerically examined how tilt angle and water container thickness (WCT) simultaneously affect PVT-PCM system performance. According to the preliminary study, the PVT-PCM system's best configuration is 30 mm WCT. WCT above 30 mm did not

affect PCM melt fraction or PV panel temperature. Additional research indicates that a system orientation of 30° to 90° reduces PV average temperature. PCM melting lowers PV panel temperatures by 23 °C and 5 °C at 30° and 90° orientations. Naseer et al. [71] measured PV performance in Taxila, Pakistan, by manipulating hybrid PCM thickness. Compared to the base case (A) with natural cooling, three PV configurations (B, C, and D) with different hybrid PCM thicknesses were tested. The "D" configuration with the thickest low melting PCM had superior electrical power and efficiency.

In recent studies, two or more passive PV cooling systems have been integrated together; one common approach is the PCM-Fin heatsink. As shown in Figure 2.12, Rabie et al. [72] integrated concentrator photovoltaic systems with fins and PCM for passive cooling. PV layer temperature and conversion efficiency with integrated PCM and extended fins are predicted by a two-dimensional model. Individual design concentration ratios are evaluated by the model. A concentration ratio of 5 gives solar cells with an aluminium heatsink and four parallel PCM cavities a steady state of 120-125 °C. The heatsink's expanded aluminium fins outside the PCM container slow melting and prevent temperature rise. The temperature stays around 82 °C with this setup. Local eutectic PCM (myristic-stearic acid) in an aluminium-finned container was used by Homlakom et al. [73]. PCM proportions were 60:40, 70:30, and 80:20. The best PCM ratio was 60:40, which increased efficiency by 4.2% and lowered temperature by 7.06 °C. Chibani et al. [74] optimized a CPV system with RT-35HC PCMs and metal fins on the solar module back. This study examined fin tilt angles and materials. Solar panels with fins have higher energy efficiency regardless of material or inclination angle. Graphite and copper reduced solar panel temperatures by 3 °C compared to steel and titanium. Wongwuttanasatian et al. [75] compared PCM in

grooved, tubed, and finned containers and found that passive cooling of PV modules with PCM depended on irradiation. Al-Lami et al. [76] added wavy fins to paraffin wax, one of the most abundant PCMs, to reduce thermal conductivity and speed melting and solidification. The modified PV module cools by 16 °C and increases maximum output power by 38.4% compared to the reference PV. Singh et al. [77] found that PCM-Fin heat lowers PV temperature by 19 °C.

Several reports have confirmed that the type of PCM used also significantly influences the performance enhancement of PV modules [22], [78], [79]. Diaz et al. [80] computationally evaluated the performance of seven (RT-21, RT-25, RT-35, n-Octadecane, calcium chloride hexahydrate, capri-cepalmitic acid, and capric-lauric acid) PCMs in a PV-PCM system for the cities of Vicuna and Calama, Chile. The authors deduced that the type of

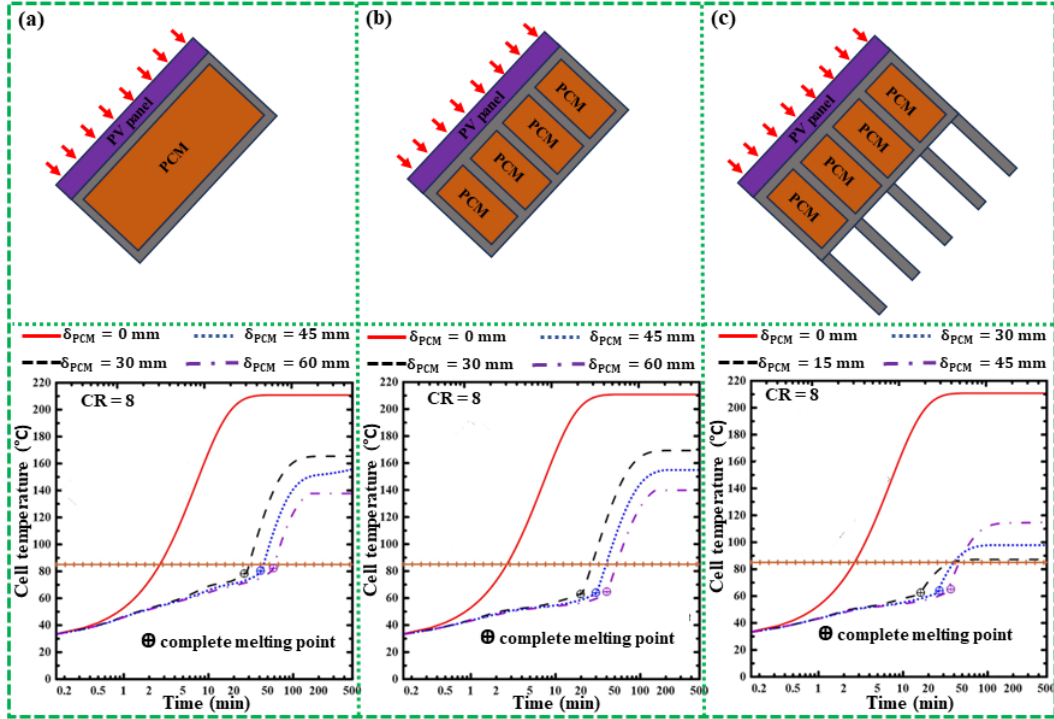


Figure 2.12: Schematic view and transient variation of solar cell temperature for PCM in (a) Single cavity, (b) Four cavity, (c) Four cavity with extended fins [72]

PCM selection depends on the climatic conditions and the time variation of the study. Ahmad et al. [62] investigated the RT-25, RT-31, and RT-42 for the thermal performance of PV modules. The results showed that RT-25 stood out in terms of electrical efficiency enhancement. A similar study was conducted by Durez et al. [81] using RT-21, RT-35, and RT-44. The results showed that a PCM with a melting point of 21°C can reduce panel temperature by 39%.

To further improve the performance of PV modules, a hybrid nano-PCM heatsink was reported [20]. In a recent study, nano-enhanced PCM heatsinks using graphene (GNP) and copper oxide (CuO) nanoparticles and polyethylene glycol (PEG) PCM were demonstrated by Moein-Jahromi et al. [82]. Dev et al. [83] designed a hybrid active nanofluid and passive PCM cooling system for solar PV. The experimental study confirmed that nanoparticles improve the thermal conductivity of the working fluid and thus enhance the electrical efficiency of the module. To enhance the heat transfer efficiency between the PCM (coconut-sunflower oil) and the cooling water, Abdollah et al. [84] included boehmite as a nanoparticle into the PCM. The findings indicated that the highest increase in power output, compared to the reference scenario, was observed while using nano-composed oil. This increase amounted to 48.23% and occurred at a radiation intensity of 690 W/m².

PCM-based passive cooling is not limited only to conventional PV systems but also to more advanced configurations. The integration of a PVT system with a PCM is now being investigated as a trend in the field of PV thermal management. In their study, Fayaz et al. [85] conducted a comprehensive investigation into the impact of various operating circumstances on the performance of PVT-PCM, including both numerical simulations and experimental methods. The PVT-PCM system achieved an electrical efficiency of 12.75%

in the experimental scenario, whereas the numerical study yielded an electrical efficiency of 12.59%. A similar study was carried out by the same authors by varying the radiation level while keeping the mass flow rate of the working fluid in the PVT constant [86].

Ozbas et al. [87] showed advanced PCM cooling of solar PV. Methanol and water are used to integrate passive PCM and heat pipe (HP) cooling in this study. An indoor solar simulator showed that thermosyphon heat pipes increased PV efficiency and lowered PCM-only, PCM-methanol, and PCM-water temperatures by 1.3, 4.4, and 5.4 °C, respectively. In another advanced study, Gad et al. [88] combine PCM with HP and a compound parabolic collector. The CPV-PCM-HP cooling system outperformed the CPV-PCM system in cell temperature, conversion efficiency, and overall performance. A CPV system integrated with a PV panel raises the cell's operating temperature due to increased solar energy absorption by the PV surface.

2.3.2 Discussion on passive cooling using PCM

PCM is highly efficient at absorbing excess heat from PV panels that is not transformed into electrical energy. The latent heat of the phase change material assists in reducing the temperature of the panel, bringing it within the range of 25 °C, and keeping it relatively stable during periods of high solar irradiance. Singh et al. [59] demonstrated the most significant decrease in PV temperature, with a reduction of 23.3 °C, by using calcium chloride hexahydrate ($\text{CaCl}_2 \cdot 6\text{H}_2\text{O}$) as PCM, as indicated in Figure 2.12. Incorporating PCM leads to an average temperature reduction of around 10.1 °C. Ozbas et al. [87] found that using paraffin wax in conjunction with a heat pipe resulted in a temperature decrease of 1.3 °C. Figure 2.14 demonstrates that the PV electrical efficiency is improved by integrating integrating a PCM passive cooling system.

Table 2-2: Summary of proposed work by different researchers for cooling PV modules using PCM

Reference (year)	PCM tested, melting point (°C)/Latent Heat (kJ/kg)	Type PV/PCM integration	Nature of work	Maximum T_{PV-ref} (°C)/ T_{PV-PCM} (°C)	Key findings
Adibpour et al. [57] (2023)	CaCl ₂ ·6H ₂ O, 29/188	PV integrated with PCM inside aluminum container	Numerical (ANSYS)	42/35	- The panel temperature dropped by 7 °C. - PCM close to the PV begins to melt first.
Ranawade et al. [67] (2023)	OM37-OM42 38-44/186-190	PV integrated with multi-layered PCM	1.Experimental 2.Numerical	64/57	The multilayer arrangement is more efficient than single layer.
Bhakre et al. [70] (2023)	RT-42, 42/165	PVT/PCM	Numerical	-	PCM melts faster at higher tilt angles and takes longer to fully melt at lower tilt angles.
Ozbas et al. [87] (2022)	Paraffin wax, 47/266.	PV integrated with PCM combined heat pipe passive cooling.	Experimental	54.9/53 (PCM). 54.9/50.5 (PCM + methanol). 54.9/49.5 (PCM + water).	- PCM and thermosyphon integration is very effective. - Temperature reduced by 10%, while efficiency increased by 6%.
Homlakorn et al. [73] (2022)	Myristic acid + stearic acid, 36.1, 168.3	PV + PCM +Fin	Experimental	60/52.94	- Organic PCMs reduce PV temperature. - Max reduction of 7.06 °C was recorded.
Moein-Jahromi et al. [82] (2022)	Polyethylene glycol (PEG), 38.4/150	PV integrated with PCM and nanoparticles	Experimental	50.8/44.2 (PCM + GNP/CuO nanoparticles)	- The thermal conductivity of the PCM was improved by nanoparticles. - Electrical efficiency was enhanced by 3 %
Al-lami et al. [76] (2022)	Paraffin wax, 40-44/190	PV combined with PCM and wavy fins	Experimental	67/50	- Max. PV temperature reduced by 16 °C. - Max. power output increased by 38.4%
Naseer et al. [71] (2022)	1. PT48, 48/230 2. PT58, 58/225	PV integrated with hybrid PCM	Experimental	67/59	- Max. PV temperature reduced by 8.1 °C. - Power output increased by 1.9 W. - PCM configuration with large thickness and low melting point is best.
Nizetic et al. [66] (2021)	Rubitherm, 25-29/250	PV integrated with 8 PCM containers	Experimental	52/50	The peak and average clear day (July) temperature of the PV reduced by 2 °C and 0.14 °C.

Reference (year)	PCM tested, melting point (°C)/Latent Heat (kJ/kg)	Type PV/PCM integration	Nature of work	Maximum T_{PV-ref} (°C)/ T_{PV-PCM} (°C)	Key findings
					The several compartment arrangements reduced the PCM and container material by 47% and 36%.
Singh et al. [59] (2020)	CaCl ₂ ·6H ₂ O, 30/191	PV integrated with PCM inside aluminum container	1.Experimental 2.Numerical	77.1 /53.8	- The peak temperature (June) of the PV reduced by 23.3 °C. - The efficiency and power output of the module improved by 8.3 % and 15.9 W.
Velmurugan et al. [60] (2020)	Laurel and Cetyl alcohol, 21.75, 199	PV integrated with PCM inside aluminum container	Experimental	68.1 /57.8	- The PCM reduced the PV temperature by 10.3 °C. - The reduction in TPV resulted in an increase in the average PR from 70.65% to 72.63%.
Marudaipillai et al. [65] (2020)	Glycol-graphite, 35-40, -.	PV combined with finned PCM	Experimental	47.41 /35.91	The peak temperature dropped by 11.5 °C. consequently, the efficiency was improved by 3.7 %. Finned PCM composite is suitable for hot climate.
Singh et al. [77] (2020)	Paraffin wax, 20/190	PV combined with finned PCM	Numerical	45 /26	- PV temperature reduced by 19. - Peak power improved by 10.2 %. - Electrical efficiency improved by 1.22 %.
Fayaz et al. [86] (2019)	Paraffin, 44/242	PVT combined with PCM	1.Experimental 2.Numerical	75.1/64.8	- Maximum cell temperature reduction of 12 °C. - PVT-PCM system is more efficient than PVT only.

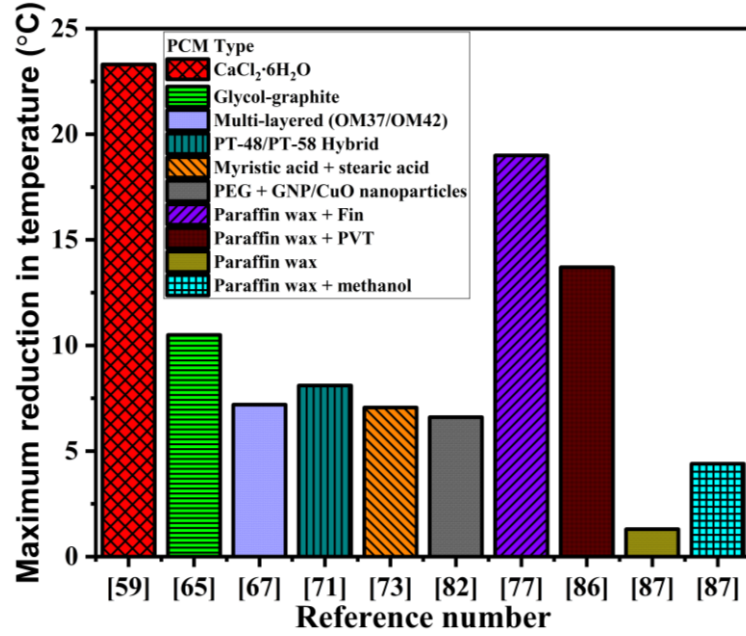


Figure 2.13: Maximum reduction in temperature using PCM compared to non-cooled PV module

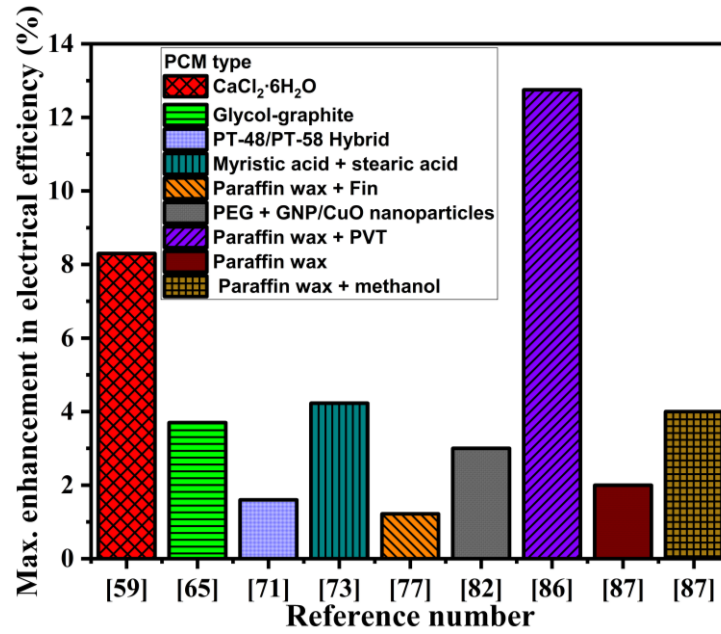


Figure 2.14: Maximum enhancement in electrical efficiency using PCM compared to non-cooled PV module

Fayaz et al. [85] achieved a maximum improvement of approximately 13% by utilizing paraffin wax combined with PVT. Subsequently, Singh et al. [59] demonstrate a notable

improvement of approximately 9%. According to Singh et al. [77] and Moein-Jahromi et al. [82], the combination of PCM with fins and nanoparticles led to the least significant increase in efficiency. Nevertheless, the little increase in efficiency can be attributed to the characteristics of PCM. More precisely, the paraffin wax material in this case had a lower melting point and latent heat compared to the paraffin wax utilized by Fayaz et al. [85], which showed the greatest improvement in efficiency.

2.4 Passive evaporative cooling

2.4.1 Overview of previous work

Passive evaporative cooling (PEC) is a phenomenon that takes place when a fraction of a fluid undergoes a phase change, resulting in the removal of its latent heat of vaporization from the surrounding environment. To optimize the overall efficiency of PV modules the implementation of evaporative passive cooling is a common approach. This approach is beneficial as it does not require any additional energy input, thereby preventing the on-site consumption of electricity generated by PV systems for cooling purposes [89]. The most widely adopted method is the direct integration of the cooling media with the PV panel. The most used cooling media is cotton wick [90], [91]. Alktranee et al. [90] mounted Cotton Wick Immersed in Water (CWIW) to the solar panel's back. Passive cooling was used to absorb photovoltaic panel heat and lower its temperature. Water-soaked cotton bristles exposed to the wind lowered PV module temperature by 22%. Power generation and efficiency rose by 16.3 W and 53%. Figure 2.15 shows Chandrasekar et al. [91]'s passive cooling system for a flat photovoltaic module using cotton wick structures, water, Al_2O_3 , and CuO nanofluids. Adding a cooling system lowers module temperature by 20 °C.

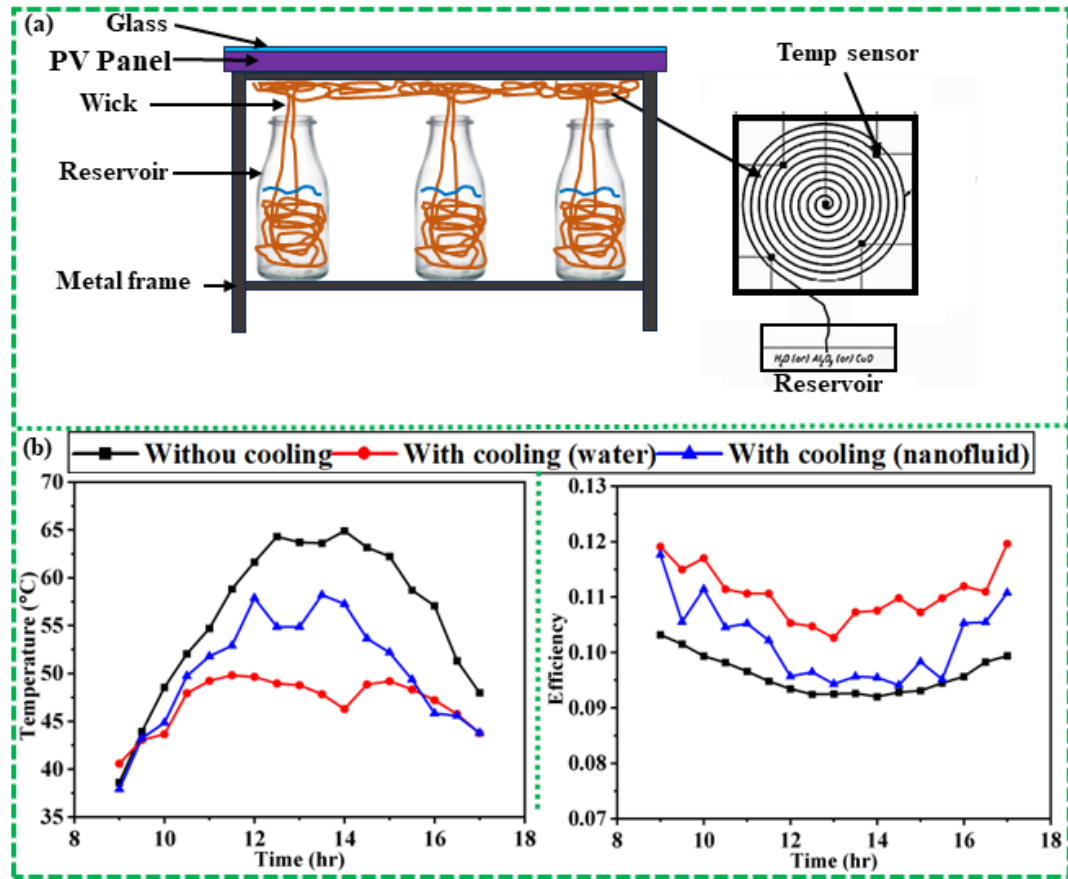


Figure 2.15: (a) Configuration for wick evaporative cooling, (b) Transient temperature and (c) Transient efficiency [91]

PEC using local materials such as clay, jute, and wet fabrics is also reported. Alami A. [89] applied a synthetic clay layer to the module's back and evaporated a thin water film. The method's 19.4% and 19.1% voltage and power improvements demonstrate its technical viability. Bhakre et al. [92] used jute directly attached to a PV panel's backside for evaporative cooling (Figure 2.16(a)). The results show that jute cooling reduces average temperature by 18 °C and increases electrical efficiency and maximum output power by 10.3% and 7.10 W compared to the reference panel. Another widely adopted evaporative PV cooling technique is integration with fins. Alktranee et al. [93] used rectangular fins with cotton wicks submerged in water for PV cooling. The system reduced temperature by

31.4%, boosted power by 66%, and enhanced electrical efficiency from 3.12 to 8.6%. Almuwailhi et al. [94] used wetted fabric in extruded polystyrene foam (XPS) for passive evaporative cooling. This system increased daily efficiency by 2.7%. Chandrasekar et al. [95] conducted an experimental investigation on a cooling system comprising fabricated aluminum fins, cotton wicks, and headers, as shown in Figure 2.16(b). The devised cooling system resulted in a 12% reduction in the temperature of the PV module while simultaneously increasing electrical production by 14%.

The application of PEC extends beyond the scope of basic standalone PV integration. The current research trend is investigating the potential integration of evaporative cooling with auxiliary systems, such as thermoelectric generators (TEG) and solar stills. Figure 2.17 shows Gao et al.'s [96] PV-thermoelectric generator (PV-STEG) thermal regulation technology using evaporative cooling. This study found that PEC technology can significantly lower PV-STEG system operating temperature by 92 K at a solar concentration ratio of 5. This method outperforms passive fin cooling.

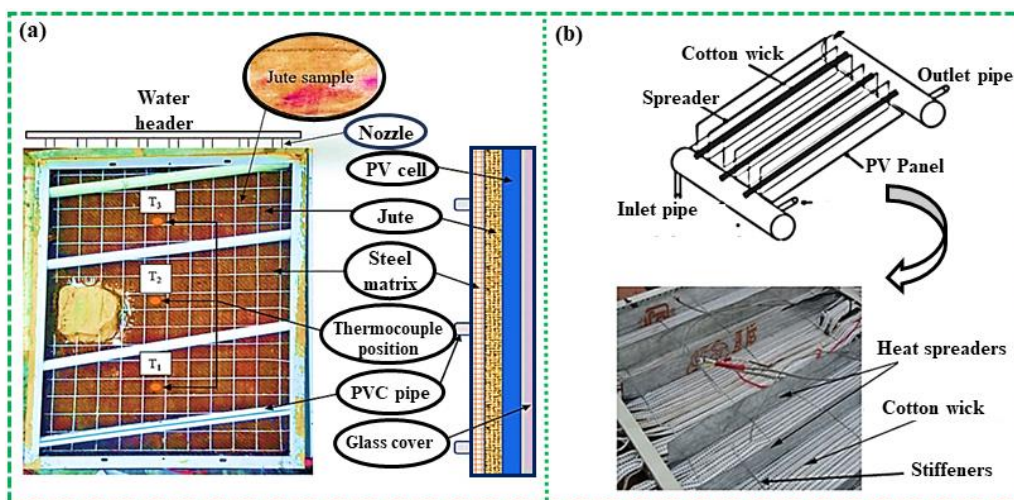


Figure 2.16: Configuration for (a) PEC using jute [92], and (b) PEC using fins and cotton wick [95]

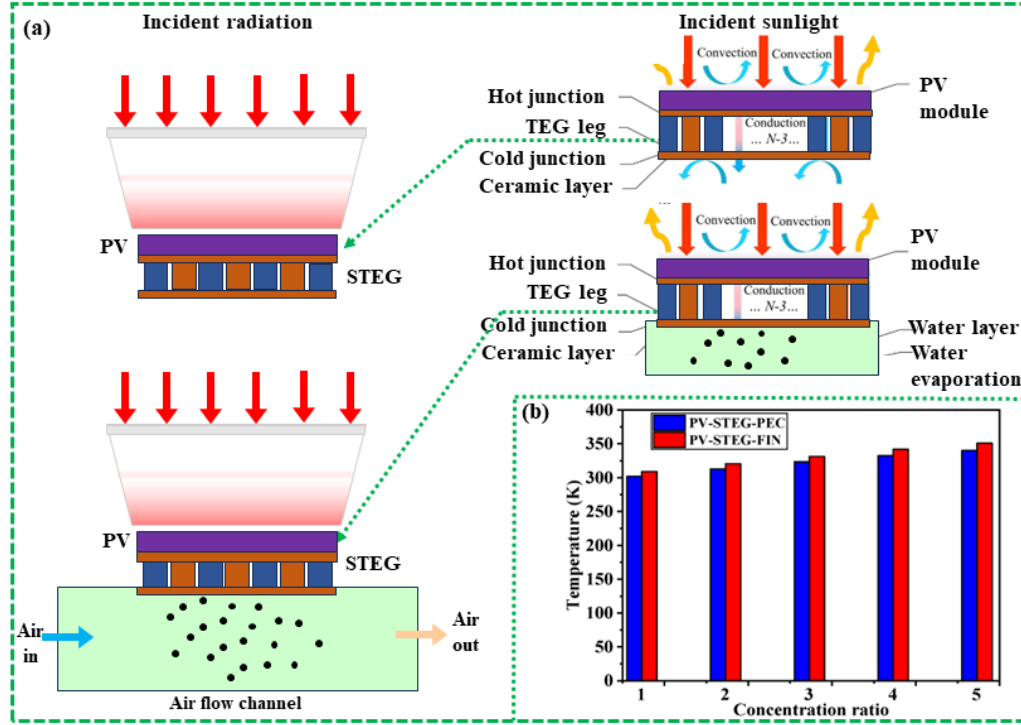


Figure 2.17: (a) Configuration for PV-STEG and PV-STEG-PEC, (b) Average PV temperature [96]

One of the primary obstacles encountered in traditional evaporative cooling systems is the issue of water loss, which consequently leads to power loss because of the need to circulate or compensate for the depleted water. In response to this issue, Srithar et al. [97] devised a sophisticated evaporative cooling system by integrating it with a solar still, as shown in Figure 2.18. Jute sacks were soaked in water at both ends for the experiment. In the solar still, capillary action causes water to rise through the sack and chill the back panel. Consequently, the output power experienced a 5.6% rise, the electrical efficiency improved by 14.51%, and the surface temperature was reduced by 8 °C. Nevertheless, this study could be enhanced by employing many varieties of woven material derived from naturally occurring fibers to enhance the cooling of PV systems.

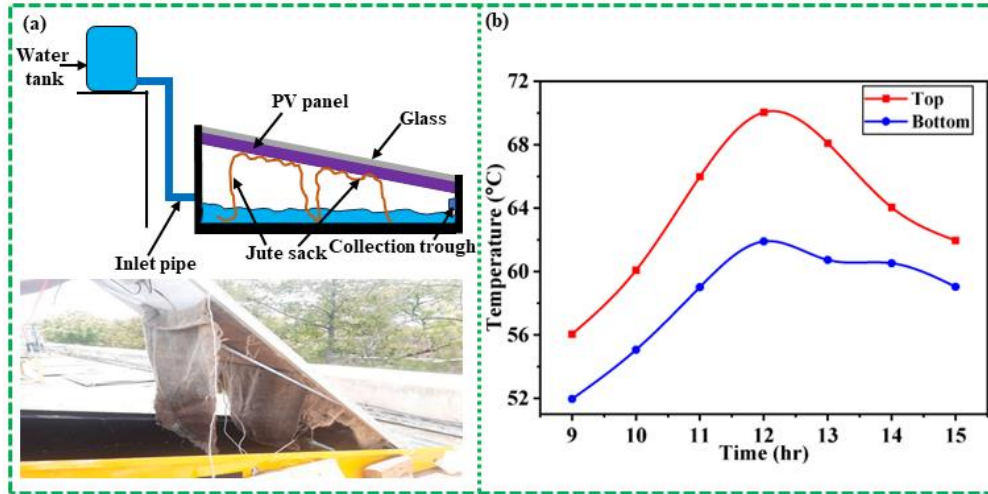


Figure 2.18: (a) Schematic and experimental configuration of the PV-Solar still-Jute PEC, (b) Daily PV temperature variation [97]

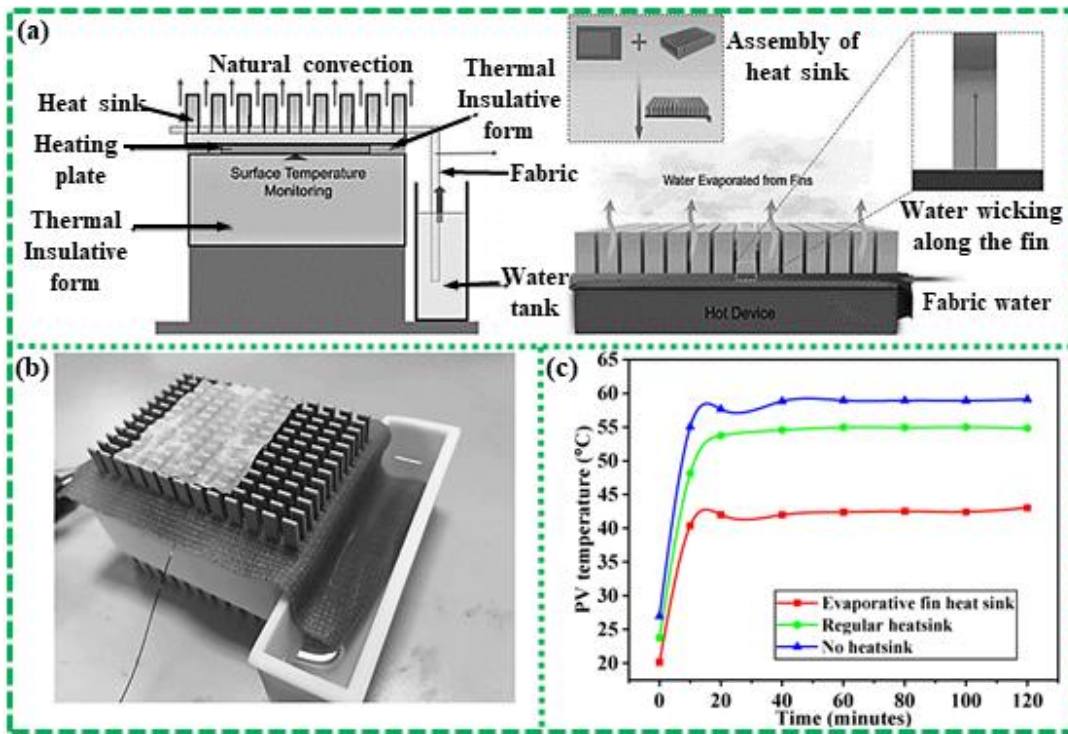


Figure 2.19: PEC using finned heatsink (a) Schematic, (b) Prototype, (c) Time dependent T_{pv} [98]

Finned heatsinks are frequently employed for the purpose of cooling photovoltaic systems, as discussed extensively in Section 2.1. Nevertheless, the system's limited cooling effectiveness per unit volume poses a significant obstacle to enhancing its performance or compactness. To address this obstacle, Lou et al. [98] conducted a study that utilized a novel approach involving surface and system design as depicted in Figure 2.19(b). This allowed capillary-driven water evaporation across the fins of the evaporative finned heatsink (EF-HS), improving cooling efficiency. Researchers found that the nano-capillary EF-HS had eightfold more passive cooling capacity than a conventional heatsink operating under natural convection, could reduce fin height, and reduced material costs while maintaining efficiency. Indoor PV panel temperatures rise 15.5 °C, increasing efficiency by 12.7%.

2.4.2 Discussion on passive evaporative cooling

Figure 2.20 illustrates that the typical decrease in PV temperature while using a PEC system ranges from 10 to 20 °C. Alktranee et al. [93] achieved the greatest temperature reduction of approximately 26 °C by using cotton wicks submerged in water on a finned surface attached to the PV. Fins enhance the conduction heat transfer rate between the PV and the evaporative wick. This also enhanced the flow of surplus heat from the photovoltaic surface to the atmosphere via convection. Srithar et al. [97] achieved the smallest decrease in temperature by combining PEC with a jute sack and a solar still. The low performance is due to the solar water still retaining heat throughout the phase change from liquid to steam, which hinders the PV temperature from reaching its minimum.

Table 2-3: Summary of proposed work by different researchers for cooling PV module using evaporative cooling

Reference (year)	Evaporative carrier	Type of PV integration	Nature of work	Maximum T_{PV-ref} ($^{\circ}C$)/ T_{PV-PEC} ($^{\circ}C$)	Key findings
Bhakre et al. [92] (2023)	Jute	PV + Jute with water	Experimental	72/51	- Maximum temperature reduction of about 21 $^{\circ}C$. - Efficiency and power improved by 10.31% and 7.1W.
Alktranee et al. [93] (2023)	Cotton wick	PV + Cotton wick + Fin submerged in water	Experimental	80.2/54.2	Attaching fins increased the heat transfer area, thus increasing the efficiency.
Almuwailhi et al. [94] (2023)	Wet fabric	PV + Wet fabric in XPS channel with water spray	Experimental	62/56	Increased energy output by 3.6%, panel efficiency by 2.7%, and a maximum temperature reduction of 5.3 $^{\circ}C$.
Gao et al. [96] (2023)	Direct water contact	PV-STEG submerged in water	Numerical	--	T_{pv} reduced by 92 K. CR and air velocity increased electrical efficiency, but relative humidity decreased it.
Srithar et al. [97] (2023)	Jute sack	PV + Jute sack + Solar still	Experimental	56 /48	T_{pv} reduced by 8 $^{\circ}C$ maximum. Temperature drops less during high sun intensity. - PV with still boost current by 7% and reduce voltage by 1.8%.
Lou et al. [98] (2023)	Fabric	PV + EF-HS + Fabric immersed in water	Experimental Numerical	58/42	T_{pv} reduced by 16 $^{\circ}C$ maximum. - Electrical efficiency and power output improved by 1.55 % and 12.7 %, respectively.
Alktranee et al. [90] (2022)	Cotton wick	PV + Cotton wick submerged in water	Experimental	56/45.8	PV temperature reduced by 22 % and efficiency increase by 7.25 %.
Chandrasekar et al. [95] (2015)	Cotton wick	PV + Cotton wick + Fin	Experimental	49.2/43.3	PV module temperature decreased by 12% and electrical power increased by 14%.
Alami A. [89] (2014)	Synthetic clay	PV + clay with water spray	Experimental	85/45	- Synthetic clay is an effective and efficient PEC technology. - Less thick clay layer is more effective.
Chandrasekar et al.[95] (2013)	Cotton wick	PV + Cotton wick submerged in: 1.Nanofluids, or 2.Water	Experimental	65/54 65/45	- The sticking of the nanoparticle on the wick reduced the efficiency in nano + wick configuration. - Increase in efficiency of 1.4 % and 0.9 % using water and nanofluid were attained, respectively.

Figure 2.21 demonstrates how PEC can enhance electrical efficiency relative to uncooled PV modules. Researchers in this field have used a variety of methods, resulting in significant differences in results. Alktranee et al. [90] showed a significant improvement, attaining an impressive 8% increase in electrical efficiency by using cotton wick and water.

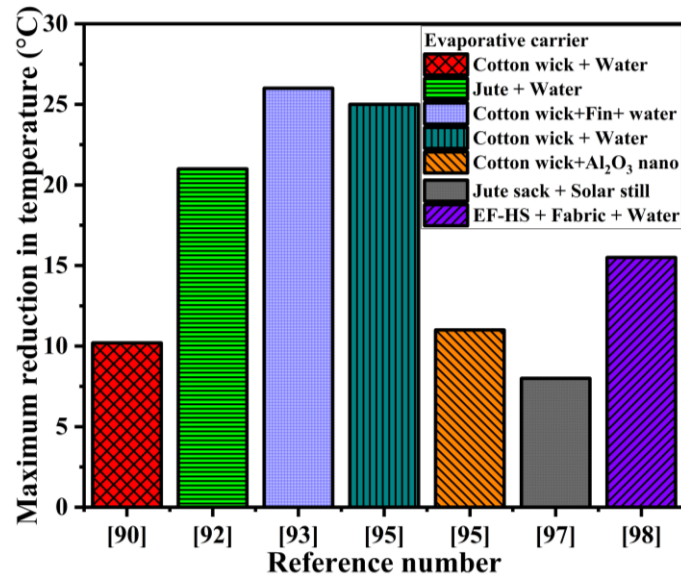


Figure 2.20: Maximum reduction in temperature using PEC compared to non-cooled PV module

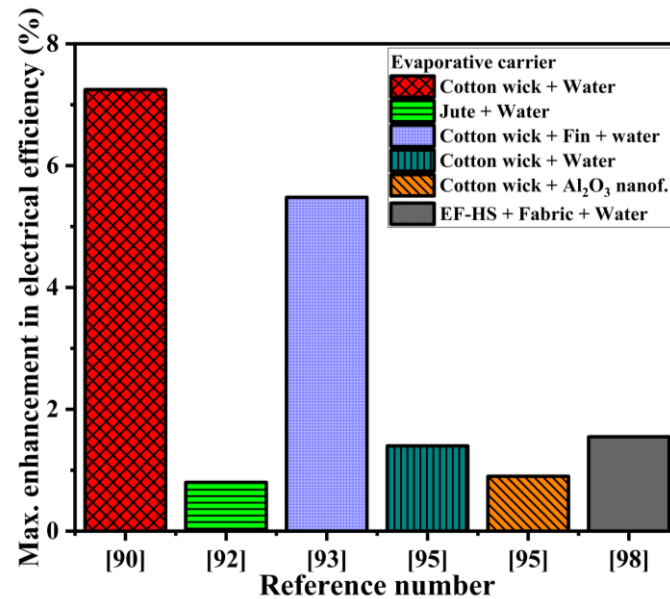


Figure 2.21: Maximum enhancement in electrical efficiency using PEC compared to non-cooled

PV

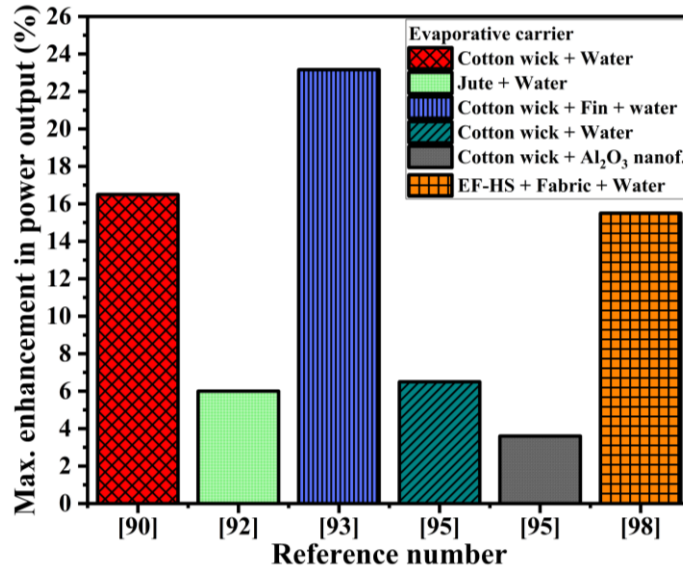


Figure 2.22: Maximum enhancement in power output using PEC compared to non-cooled PV module

In contrast, Bhakre et al. [92] reported a more modest 1% increase, employing Jute fiber and water. Four out of six PEC techniques demonstrated efficiency improvements below 2%, showcasing a range of outcomes that underscore the subtle effectiveness of evaporative cooling. This observation indicates that the effectiveness of evaporative cooling depends on several elements and is not consistently the same in all situations.

2.5 Passive cooling using heat pipe

2.5.1 Overview of previous work

The word "heat pipe" encompasses a wide concept that can be used to describe various heat transfer devices that utilize a gas-liquid two-phase flow mechanism for heat transmission. The sole differentiation lies in the propulsive force that governs the way the condensate liquid is directed back to the evaporator section, such as by the influence of gravity, capillary action, centrifugal force, surface tension, or pulsation. The classification of heat

pipes used in solar energy systems is based on their structure and includes five types: two-phase closed thermosyphon (TPCT), conventional tubular heat pipe (CTHP), loop heat pipe (LHP), microchannel heat pipe (MCHP), and pulsing heat pipe (PHP) [21]. For passive cooling of PV, some of these technologies are being adopted by researchers.

Ozbas et al. [87] created passive heat pipe PV cooling integrated with PCM. A paraffin PCM-submerged closed-loop thermosyphon system. Methanol and water are used to analyse the working fluid's effect on heat pipe cooling. The Reference PV had an average front glass surface temperature of 54.9 °C. PCM alone, PCM + methanol, and PCM + water reached 53.6, 50.5, and 49.5 °C. The highest efficiency increase was 6% with PCM and water heat pipe. Gad and his colleagues [88] conducted a study on the effectiveness of direct and indirect passive cooling systems using PCM/flat heat pipes (FHP) for low-concentrated PV panels (LCPV) with compound parabolic collectors, as shown in Figure 2.23. This study examines two panels that are analytically investigated using MATLAB. One panel incorporates PCM (direct), while the other panel incorporates PCM-FHP (indirect). The outcome showed that the CPV-PCM-FHP system is more effective.

PV-TE systems have been extensively investigated to increase solar energy efficiency. However, the greater power demand needed for active cooling and the low cooling rates of traditional passive cooling technologies have slowed the development of these systems. To address these constraints, Yao et al. [99] developed a CPV-TE integrated with heat pipe thermosyphon cooling, as shown in Figure 2.24. This efficiently reduces the speed of water circulation for cooling to 0.011 m/s without using more power. Shittu et al. [100] incorporated a flat-plate heat pipe into the CPV-TE system. A thorough examination of PV-TE-heat pipe electrical performance has been done. When the concentration ratio is 6,

the PV-TE heat pipe system is 1.47% and 61.01% more efficient than the PV-TE and PV-only systems. Finally, the PV-TE-heat pipe is recommended for high-concentration systems due to its efficiency. In a subsequent study conducted by the same authors [101], PV-TE was combined with a MCHP and investigated experimentally.

The possibility of combining radiative cooling (RC) and heat pipes as passive cooling was examined by Kumar et al. [102]. Solar radiation, ambient temperature, and wind speed are examined in Atacama Desert and Las Vegas for a PV-TE-HP integrated with RC system. The hybrid system lowered average operating temperature by 2 °C compared to the reference PV. This decline occurred in the Atacama Desert and a 13 °C drop in Les Vegas. The proposed system lowers maximum PV temperature by 4 °C in June and 9 °C in

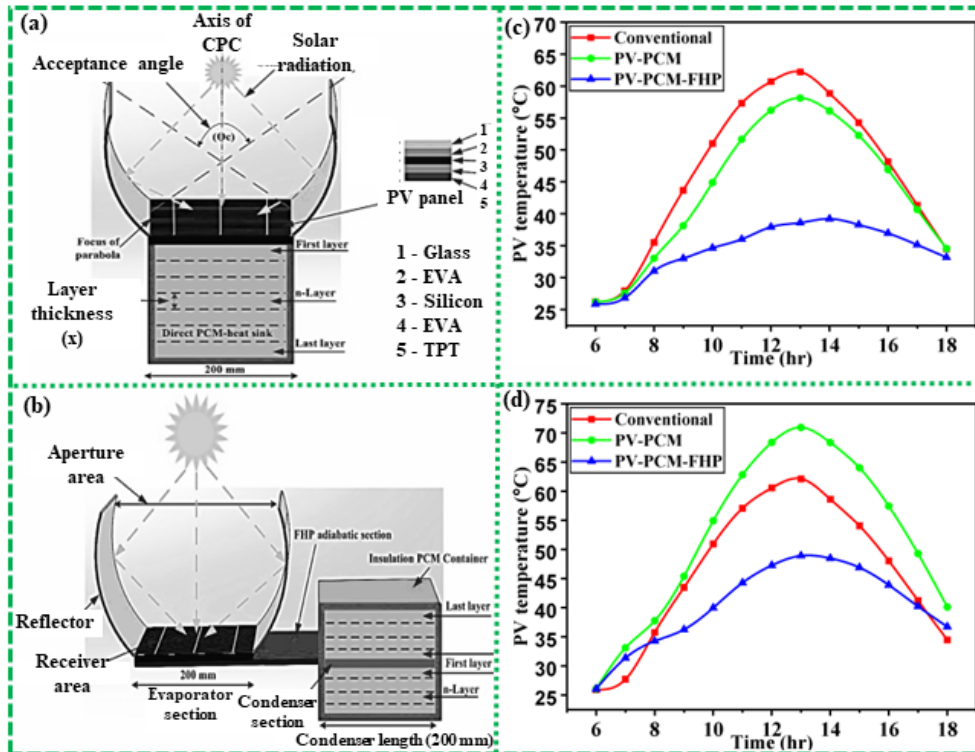


Figure 2.23: (a) Direct, and (b) Indirect PCM-heatsink, (c) Temperature variation without CPC, (d) With CPC [88]

January. Ahmed et al. [103] combined PV, hybrid heat pipes, and radiative cooling in another study. The suggested system reduces maximum T_{PV} by 12.86 °C, increasing electrical efficiency by 7.25% over the traditional PV module. LHP are frequently used in solar PV systems as heat pipe heat exchangers. It is a distinct variation of a heat pipe, where the liquid as well as vapor lines are individually linked to the evaporator and condenser. Like CTHP, LHP is a highly efficient dual-phase heat transfer system capable of lowering the solar panel's surface temperature. Alizadeh et al. [104] numerically assessed closed-loop pulsating heat pipe (CLPHP) performance. The study examined module temperature and electrical efficiency under four conditions. Cases include passive and active CLPHP cooling, natural cooling, and flat plate cooling. At 1235 W/m², passive and active cooling reduced PV temperature by 15% and 35%, respectively. Efficiency increased 23% and 35% under similar conditions. Due to a direct correlation between solar radiation and efficiency, CLPHP performs best under high radiation conditions. In a previous study by Roslan et al. [105], they recorded a reduction of 10 °C in T_{PV} which is 4 °C lower than that of Alizadeh et al. [104].

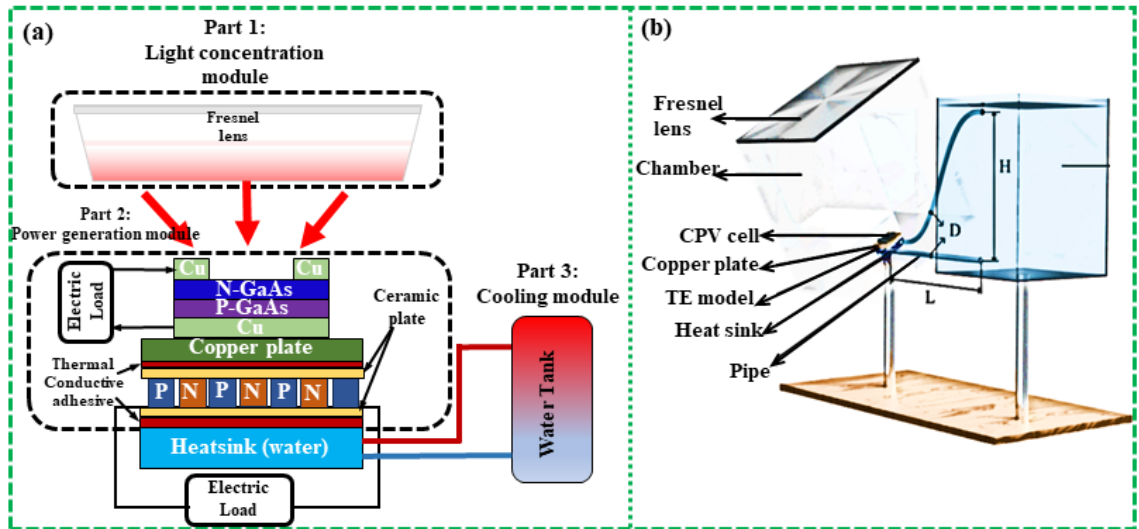


Figure 2.24: (a) Schematic, and (b) 3D diagram of thermosyphon heat pipe cooling [99]

Al-Mabsali et al. [106] conducted a numerical study to examine the influence of the distance between heat pipes in PV cooling setups. This study indicated that an optimal spanwise separation of 2.5 times the heat pipe diameter is recommended. Anand et al. [107] comparatively studied the effect of LHP and heatsink on a CPV. The LHP and heatsink have reduced the T_{PV} increase above the surrounding temperature by 48% and 25% compared to uncooled CPV systems, respectively. Similarly, the LHP demonstrates superior thermal conductivity in dissipating heat compared to the aluminum heatsink. Several writers have showcased the feasibility of a hybrid passive cooling system that utilizes one or more systems. Basri et al. [108] conducted a study to explore the potential of integrating fin heatsink and heat pipe as a passive cooling mechanism for PV thermal management. The experiment carried out in Malaysia yielded a decrease of 6 °C in T_{PV} and an increase of 0.29 V.

2.5.2 Discussion on passive cooling using heat pipe

Figure 2.25 demonstrates the diverse range of temperature reduction achieved using heat pipe. Anand et al. [107] have achieved a significant reduction of around 35 °C in module temperature, which is noteworthy. By integrating loop heat pipes into concentrated PV systems, they demonstrate the significant capabilities of this technology in handling high thermal conditions. Roslan et al. [105] found a relatively modest decrease of 10.5 °C using closed-loop pulsing heat pipes. The diversity in results highlights how heat pipe cooling is influenced by pipe layouts. The combined data shows that the average temperature decrease is between 10 and 20 °C, demonstrating the effectiveness of heat pipe technology. The success of the Anand et al. [107] model, particularly in concentrated PV, highlights

Table 2-4: Summary of proposed work by different researchers for cooling PV module using heat pipe

Reference (year)	Type of heat pipe	Type of PV integration	Nature of work	Maximum T_{PV-ref} (°C) / T_{PV-HP} (°C)	Key Findings
Yao et al. [99] (2024)	LHP	CPV + TE + heat pipe	Experimental	--	The onset time of the thermosyphon effect increases steadily as the slope of the heatsink decreases. The CPV cell demonstrates an impressive heat dissipation power density of 15.26 W/cm².
Gad et al. [88] (2023)	CTHP	PV + PCM + heat pipe	Analytical	61/40	- PV + PCM + heat pipe, is more effective than PV + PCM only. - At a condenser length of 30 cm, the PV temperature dropped by 20 °C maximum.
Kumar et al. [102] (2023)	CTHP	PV + TE + RC + HP	Numerical	1.70/53 (Las Vegas) 2.51/42 (Atacama desert)	Peak PV temperature reduced by 17 °C and 9 °C in Las Vegas and Atacama desert.
Ozbas et al. [87] (2022)	TPCT	PV + PCM + heat pipe	Experimental	1. 55/53 (Front side) 2. 45/33 (Back side)	- Mean PV temperature (back) dropped by 4.5 and 5.5 °C when methanol and water are used as the working fluid. - The efficiency increased by 4 and 6 %, respectively.
Alizadeh et al. [104] (2022)	LHP	PV + CLPHP in 1.Free convection 2.Forced convection	Numerical	1. 73/59 2. 73/51	- CLPHP performs better at high irradiation. - Efficiency was improved by 23% and 35% for free and forced convection cooling. - CLPHP cooling is cost effective
Ahmed et al.[103] (2022)	CTHP	PV + RC +HP	Numerical	60.5/47.7	- TPV reduced by 12.8 °C while efficiency increased by 7.25%. - PV + RC + HP is suitable for high irradiance climate; however, it has inverse relation with wind speed.
Shittu et al. [100] (2019)	CTHP	PV + TE + heat pipe	Numerical	N/A	PV-TE-Heat pipe systems work well in high-concentration systems. Increased ambient temperature have adverse effect on the performance of PV-TE-Heat pipe.
Anand et al.[107] (2019)	LHP	CPV + LHP	Experimental	102.6/67.8	LHP offers low thermal resistance and consistent output voltage.
Roslan et al.[105] (2019)	LHP	PV + CLPHP	Experimental	51/41	With CLPHP both electrical efficiency and output power increased with decreasing Temperature.

the need for further research on the specific aspects influencing the efficiency of heat pipe cooling techniques in different photovoltaic applications. Figure 2.26 illustrates the significant enhancement in electrical efficiency compared to non-cooled PV modules by incorporating heat pipe cooling. Only four heat pipe technologies were found in the

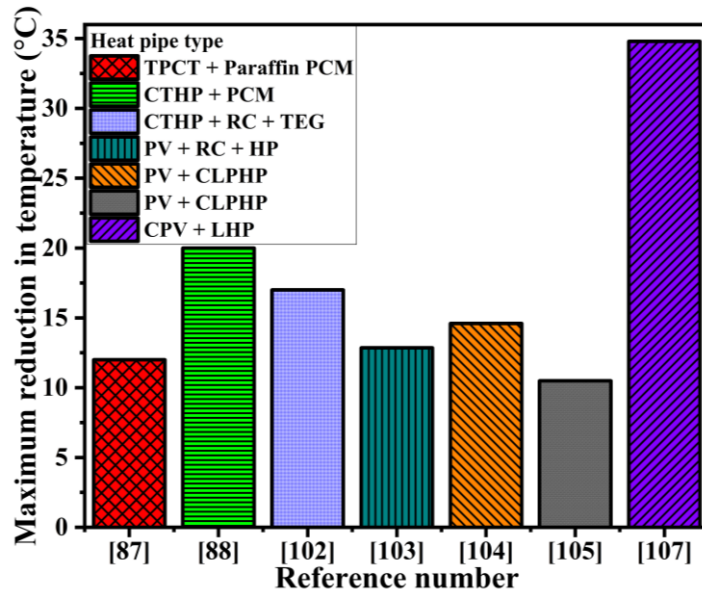


Figure 2.25: Maximum reduction in temperature using heat pipe cooling compared to non-cooled PV module

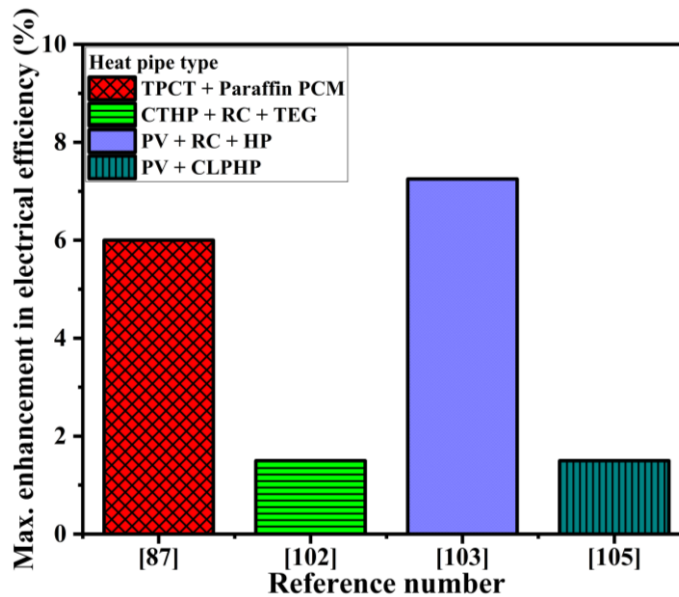


Figure 2.26: Maximum enhancement in electrical efficiency using heat pipe cooling compared to non-cooled PV module

literature reviewed, and their efficiency enhancements were documented. Ahmed et al. [103] demonstrated the most significant improvement, attaining a remarkable 7.25% increase in electrical efficiency by integrating CTHP and radiative cooling. Both Kumar et al. [102] and Roslan et al. [105] observed a slight increase of 1.5%, showing similar results although employing distinct methodologies: CTHP with TEG and CLPHP, respectively. The narrow scope of stated efficiencies necessitates a detailed analysis of the elements that impact the effectiveness of heat pipe cooling in improving electrical efficiency.

2.6 Passive cooling using surface coating and liquid immersion

2.6.1 Overview of previous work on surface coating

PV thermal management using surface coating is most referred to as passive radiative cooling (PRC). PRC is a method that does not require electricity to cool objects by exchanging heat with the cold outer universe by thermal radiation in the wavelength range of 8 μm to 13 μm . This technique is effective in reducing the temperature of PV modules [109]. It is shown that the power generation capacity will be enhanced using PRC coating because of the efficient dissipation of radiative heat. Various coating materials have been documented in the literature, such as nanocomposite coating [109], [110], methyl pentane copolymer (TPX) [111], thermochromic coating [112], polydimethylsiloxane (PDMS) as a photonic composite [113], and nanophotonic coatings [114]. In addition, silicon dioxide (SiO_2) is primarily doped into these materials to enhance their hydrophobic characteristics. Ensuring that PRC coatings applied to the front surface of a PV tile do not hinder the passage of sunlight into the solar cell is crucial. Hence, the PRC coatings must have absolute transparency. However, this criterion does not apply when the coatings are applied to the rear surface of the PV [115]. Cui et al. [115] tested PRC using commercial radiative

coatings (CRC) on PV modules. The researchers tested two CRC on PV module front and rear surfaces for thermal conductivity and electrical efficiency. Different coatings had optimal infrared spectra. The findings show a 3.5 °C decrease in TPV and 0.33% increase in electrical efficiency. In another study, Cui et al. [111] examined the effects of applying a sub-bandgap reflection (SBR) and mid-infrared emission (MIE) coating to the rear surface of a PV module. The study found that mid-infrared emission coating works better. TPX/SiO₂ coating was applied to the PV. The coating reduced T_{PV} by 1 °C and increased efficiency by 0.21%. Perrakis et al. [114] analyzed photonic strategies to regulate photovoltaic temperature. These methods include RUR, SBR, MIE, and their combinations. Due to PV parasitic absorption, SBR is the most effective method.

Li et al. [113] coated the PV module's front with PDMS. The numerical result showed that PV glass with PDMS coating can reach 2 °C. Harbinson et al. [116] utilized PDMS doped with SiO₂ nanoparticles on the frontal surface of a PV, as shown in Figure 2.27. Inside a Styrofoam cooler, the solar panels are positioned in a compartment for the purpose of insulation. Furthermore, a polyethylene film is placed over the panels to prevent convection from occurring. The coating is transparent to visible light, which enables it to penetrate the

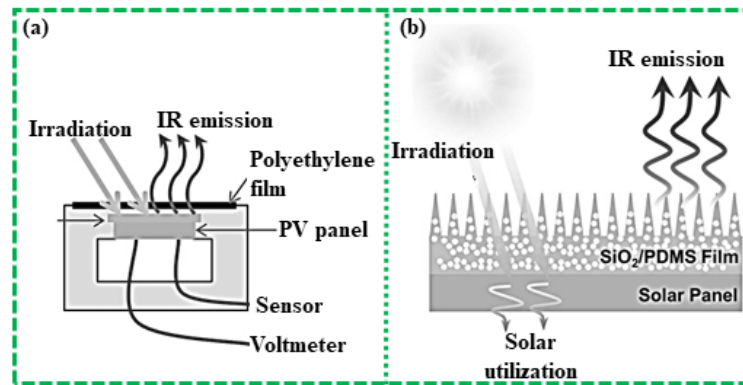


Figure 2.27: (a) schematic view of the SiO₂ PV coating set-up, (b) transparent SiO₂ PV coating[116]

solar cell below. The surface temperature of the PV dropped by 3.5 °C. Hachim et al. [110] improved the electrical efficiency of PV cells using titanium dioxide (TiO₂). The growing interest in PV system PRC has led to the use of optical research methods to study PVT system performance. Zhang et al. [112] coated the PVT system's front with thermochromic coating (TC). Module surface temperature dropped by 3.67 °C.

2.6.2 Overview of previous work on liquid immersion

The immersion technique to cool PV modules is an affordable approach that is gaining attention for PV thermal management in moderate temperature ranges, particularly in hot geographical regions worldwide. Al-Amri et al. [117] submerged PV panels in three distinct liquids: water, ethylene glycol, and engine oil solution, to examine the impact of liquid immersion on the thermal performance of the panels. Further immersion into these liquids occurred when a heat pipe actively cooled the panels to a surface temperature of 50 °C. The working temperature was reported to drop to 30, 34, and 47 °C, respectively. Additionally, the immersion approach improves heat uniformity and shields the PV panel from hot spots. Sivakumar et al. [118] investigated how submerging PV panels in water from 10 to 40 mm improved performance. Thermal and electrical efficiency are empirically measured across solar radiation levels. As immersion depth increases, PV system temperature decreases, reaching 37.5 °C at 40 mm. Electrical efficiency decreased after 20 mm of immersion. Therefore, the researchers concluded that 20 mm immersion depth is optimal. They attributed the decrease in PV temperature and efficiency beyond this depth to limited solar energy reaching the PV surface.

Zhu et al. [119] explored the thermal management of a CPV using immersion in deionized water. The findings indicate that the CPV system may be effectively cooled to a

temperature of 45 °C while operating at an irradiance level of 940 W/m². In addition, the module exhibits a highly homogeneous temperature distribution, with a maximum deviation of 3.4 °C. However, the photovoltaic efficiency was observed to decrease during prolonged exposure to deionized water. Han et al. [120] conducted a comparison study to examine the impact of four different immersion solutions on the temperature of CPV. The liquids consist of dimethyl silicon oil (DSO), isopropyl alcohol (IA), deionized water (DW), and ethyl acetate (EA). The thermal effectiveness ranking, based on the measured module temperatures, is as follows: DW, EA, IPA, and DSO, with temperatures of 31, 42, 51, and 69 °C, respectively.

2.6.3 Discussion on passive cooling using surface coating and liquid immersion

Figure 2.28 presents data comparing the effectiveness of two different passive PV cooling methods: surface coating and liquid immersion, against non-cooled PV modules. Cui et al. [115] achieved a significant drop in temperature of 7.8 °C by using commercial radiative coating in Part A (surface coating). Cui et al. [111] and Li et al. [113] both observed a minimum temperature drop of 1 °C. The detailed results show how certain coating materials and application techniques affect the effectiveness of surface coating for cooling purposes. In the larger context, surface coating cooling achieves a relatively smaller temperature reduction compared to other passive cooling technologies such as fin heatsink cooling, PCM-based cooling, evaporative cooling, and heat pipe cooling. This observation highlights the need to have a detailed understanding of how different cooling methods interact and their appropriateness for different photovoltaic applications.

Table 2-5: Summary of work proposed work by different researchers for cooling PV module using surface coating and liquid immersion

Reference (year)	Type of cooling		Type of coating/immersion material	Type of PV integration	Nature of work	Maximum $T_{PV-ref} (^{\circ}C)/$ $T_{PV-PRC \text{ or } LI} (^{\circ}C)$	Key Findings
	Coating	Immersion					
Cui et al.[111] (2023)	✓		TPX/SiO ₂	PV cell + Rear coating	Experimental	68.3/67.2	• MIE coating is better than SBR coating. • About 1 °C reduction in T_{PV} and 0.21% increase in electrical efficiency achieved.
Harbinson et al.[116](2023)	✓		PDMS/SiO ₂	PV + Front coating	Experimental	79/74.5	• PDMS/SiO₂ improves hydrophobicity. • Temperature of the PV reduced by 3.5 °C.
Zhang et al.[112] (2023)	✓		Thermochromic coating (TC)	PVT + Front coating	Experimental	20.84/17.17	• TC reduced the working T_{PV} by 4.49 °C and 3.67 °C compared to uncoated PV and PVT, respectively.
Cui et al.[115] (2022)	✓		Two CRC with wavelengths $4 \mu m < w < 25 \mu m$ and $8 \mu m < w < 13 \mu m$.	PV cell + front/rear coating	Experimental Numerical	54.7/46.87	• The application of a coating on the rear surface of PV panel is more efficient. with about 7.8 °C reduction in T_{PV} and 0.6% increase in electrical efficiency was achieved.
Li et al.[113] (2021)	✓		PDMS	PV + Front coating	Experimental Numerical	44.6/44.1	• RC is more efficient if applied directly to the cell surface without protective glass. • Less than 1 °C reduction in T_{PV} with glass.
Al-Amri et al.[117] (2022)		✓	1.Water (W) 2.Ethylene glycol (EG) 3.Engine oil (EO)	PV + HP + Liquid immersion	Experimental	1. 50/30 2. 50/34 3. 50/47	• The maximum T_{PV} reduction was 35 C. • W is more efficient than EG and EO because of its low viscosity. • Immersion reduces hot spots.
Sivakumar et al.[118] (2021)		✓	Water	PV + Water immersion	Experimental	50.4/37.5	• The PV temperature reduced by 13 C maximum. • The PV temperature is directly proportional to immersion depth. • 20 mm is the optimal immersion depth.

The work by Al-Amri et al. [117] stands out in Part B (liquid immersion) of Figure 2.28 because it reports both the highest and minimum temperature reduction, giving a thorough understanding of the effectiveness of liquid immersion. The study found that using water as the cooling liquid resulted in a temperature decrease of 20 °C compared to 3 °C achieved by engine oil, which was minimal. The significant difference in results highlights the necessity of choosing a suitable cooling liquid to enhance the efficiency of liquid immersion cooling techniques. The significant decrease in temperature is due to several favorable properties of water in comparison to engine oil. Water has greater thermal conductivity and heat capacity, enabling it to absorb and release heat more efficiently. Water has a larger specific heat capacity, allowing it to absorb more heat per unit mass without a substantial increase in temperature. These characteristics make water a better and more effective cooling fluid. This emphasizes the need to choose the appropriate cooling medium to improve the overall efficiency of liquid immersion methods.

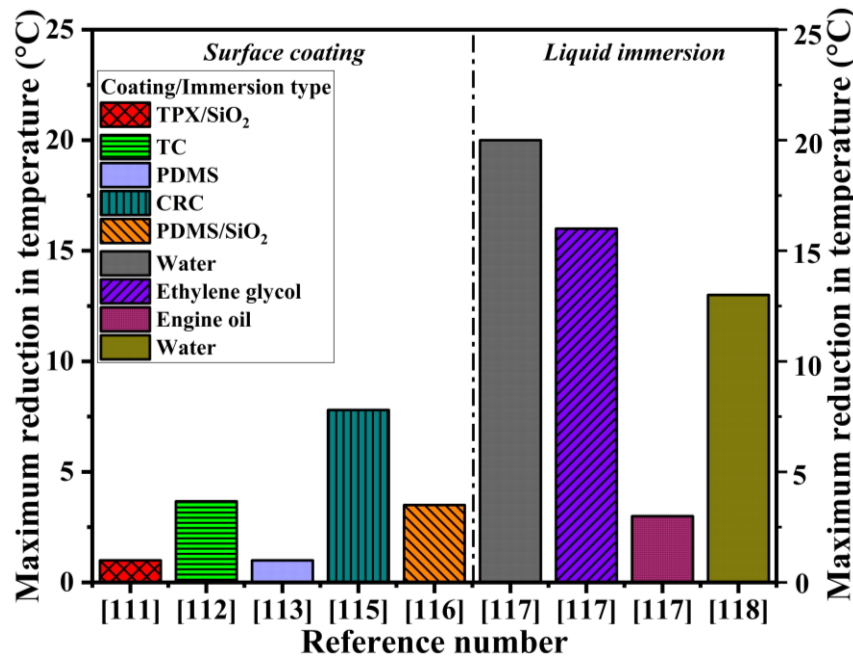


Figure 2.28: Maximum reduction in temperature using surface coating and liquid immersion cooling compared to non-cooled PV module

2.7 Active cooling using air

2.7.1 Overview of previous work

A prevalent method for cooling solar panels involves utilizing a blower and an air duct positioned beneath the panel to facilitate forced air cooling. Singh et al. [121] computationally proposed an internal duct forced air PV thermal management. The results indicated that the implementation of internal cooling can lower the temperature of the PV system by approximately 12.6 °C. In addition, the authors stated that internal cooling is more efficient than external cooling by around 5 °C. Almuwailhi and Zeitoun [122] conducted a comparative study on the performance of natural and forced convection cooling for PV systems. The utilization of fans to provide forced air circulation at a speed of 1-3 m/s through the air duct (1968 mm × 988 mm × 120 mm) beneath the photovoltaic panel resulted in enhanced efficiency. Daily energy output and efficiency increased by 4.4% and 4%, respectively, at 3 m/s. Choubineh et al. [123] added PCM to the air duct to improve PV temperature regulation. The study examines three forced air-cooling cases. The experimental results show that 6 mm thick PCM sheets reduce panel temperature by 3.4, 3.6, and 3.7 °C in driven high-, medium-, and low-velocity flows. Lebbi et al. [124] proposed hybrid air-cooled and self-cleaning PV thermal management in a more advanced study. This project addresses both PV system inefficiency due to high temperatures and dust accumulation reducing sunlight intensity. In Figure 2.29(a), forced cooling cooled the rear and running water cooled and cleaned the front. The device self-cleans by creating water droplets from a perforated pipe on the front. The authors experimentally evaluated power output, current, and voltage in relation to module operating temperature. At 650 W/m², the maximum temperature drop was 15 °C, resulting in a 6% efficiency increase.

To reduce the effect of the running water on the intensity of the solar radiation reaching the solar cell, Kabeel et al. [125] used reflectors to concentrate the incoming radiation, as shown in Figure 2.29(b). This further enhanced the PV's power output.

The incorporation of external fins in forced air cooling is also common in the literature. Abdrahman et al. [126] cooled the PV module with a heatsink and forced air. A cooling circuit under the solar cell improves cold air circulation to the PV panels. A 6-inch pipe plenum is connected to five T-shaped pipes. The Malaysian on-site experiment showed that forced air circulation lowers PV temperatures better than natural air flow, which is heavily influenced by wind speed. Fin height and thickness were examined by Mojumder et al. [127]. Aluminum fins increased PVT heat gain by 18.4%. Murtadha et al. [128] compared longitudinal and S-shape PV panel fin configurations with forced air cooling. Both longitudinal and S-shaped fins reduced panel temperature by 11.9% and 9.6%, respectively. S-shaped fins perform poorly because air recirculates beneath the panel before exiting the duct. Abushgair [129] suggests fins to separate the air duct and air-to-air

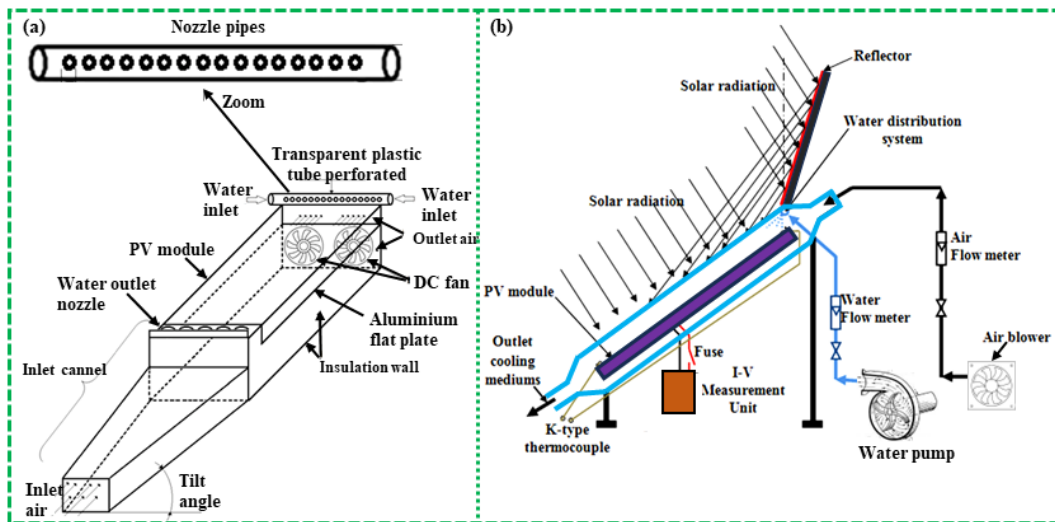


Figure 2.29: (a) Hybrid forced air and water spray cooling [124], (b) Hybrid cooling with reflector [125]

heat exchanger. A blower circulates air over the back panel and a chimney releases it outdoors. The unique cooling setup lowered PV temperature by 6 °C. PV module efficiency and output power increased by 2% and 12.8%, respectively. Lenin et al. [130] conducted a similar study using wooden ducts and vanes to direct airflow within the duct. In an extended study, duct-only and duct-fin forced air cooling were comparatively studied by Ahmed and Nabil [131]. Furthermore, Nabil and Mansour [132] conducted research on the use of forced air cooling on the front surface of the PV module using a DC fan.

Some writers have experimented with PV system design without blowers or ducting for cooling. They are investigating forced air cooling with DC fans perpendicular to PV modules. Hussien et al. [133] studied two forced-air PV cooling configurations as shown in Figure 2.30(a-b). In the first scenario, the PV had a blower and rectangular duct. Small fans were symmetrically placed on the PV's back in the second scenario. The results suggested that small fans at the back of PV systems can improve performance. This improvement maximizes efficiency at 2.1% and reduces energy consumption by 7.9%. The blower cooling method achieved 1.34% total efficiency improvement and 4.2% energy savings. The concept of fans blowing air normal to the rear side of PV was also demonstrated by Syafiqah et al. [134], Amelia et al. [135], and Craciunescu et al. [136]. These studies recorded a decrease in the maximum operating temperature of the PV cell, as documented in Table 2-6. Almeshaie et al. [137] devised a novel PV design methodology called "Rapid Evaluation of Solar Panel Cooling (RESC)," which uses both laboratory and on-site experimental data to assess the effectiveness of cooling systems integrated into solar panels. The authors devised an active fan cooling system for solar PV applications employing this approach, as depicted in Figure 2.30(c-d). Laboratory and field experiments

were conducted on Kuwait's climate, after which a computational model was developed using PVGIS software. The data presented in Figure b demonstrates that higher chamber temperatures lead to improved thermal management of the PV system when utilizing the fan cooling system.

Exhaust air from forced air cooling is not usually released into the surrounding environment. Some authors have investigated utilizing hot exhaust air for purposes like air drying [138] or space heating [139]. Directing forced air-cooling heat to a drying device worked for Deokar et al. [138]. At 5.4 m/s and 923 W/m², PV operating temperature dropped 16 °C, increasing average voltage by 4%.

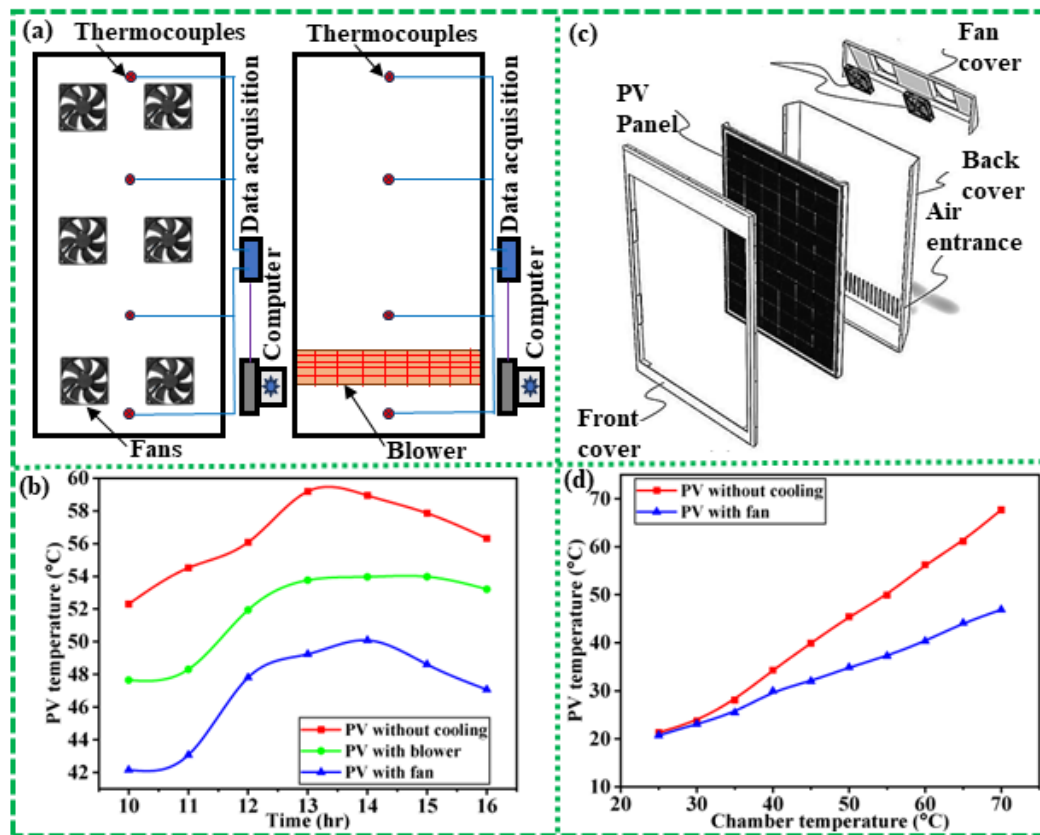


Figure 2.30: (a) Forced air cooling using fan and blower [133], (b) Forced air cooling using fan

[137]

An air-recirculation cooling system by Haddad et al. [139] uses preheated air for secondary heating. The air circulates twice beneath the PV before leaving the collector for heating. Mekonen et al. [140] integrated forced air and evaporation cooling. Figure 2.32 shows a two-stage evaporative cooling experiment with an activated carbon adsorbent between the cooling channels. A fan exposed the watered fiber strand to outside air. PV powered the fan and pump. It was found that cooling pad quantity directly affects evaporative cooler capacity. Additionally, PV temperature dropped 9 °C.

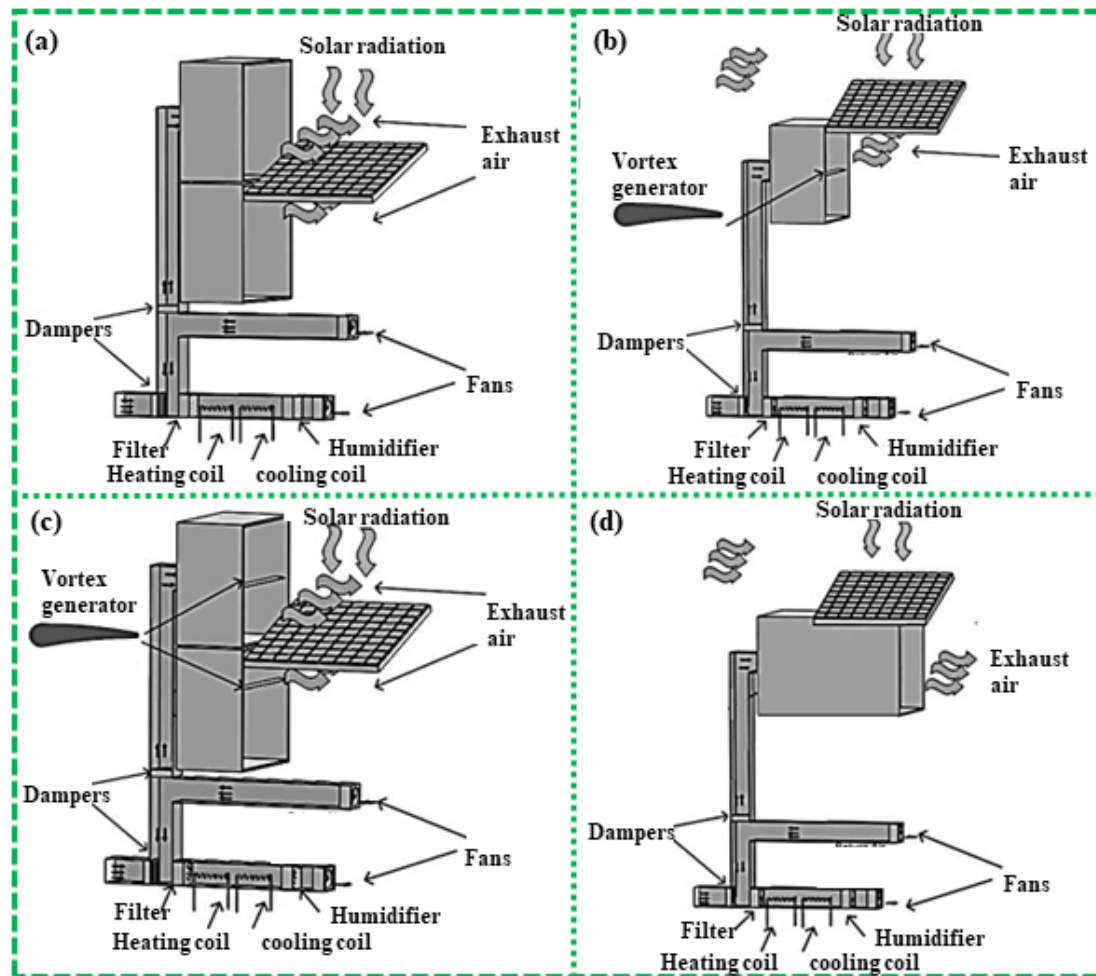


Figure 2.31: PV cooling using industrial exhaust (a) Top surface cooling, (b) Both sides cooling, (c) Rear surface cooling, and (d) Internal cooling [141]

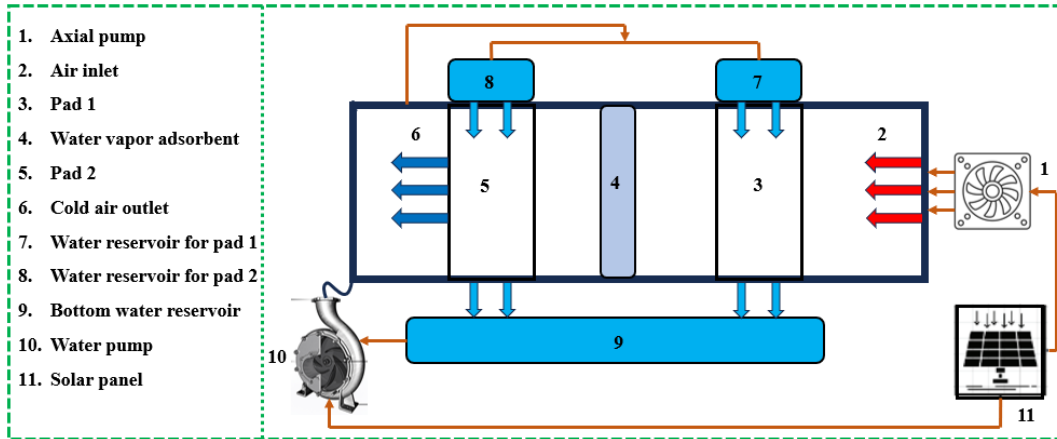


Figure 2.32: Hybrid forced air and evaporative PV cooling [140]

In an advanced research project, Salameh et al. [141] used industrial exhaust to cool PV modules. Air conditioning return air was used to create creative cooling strategies. Figure 2.31 shows three exterior and internal flow setups. The types of exhaust air exposure differentiated the cases. Exhaust air cools the PV system on both sides without a vortex generator, on the bottom side with one, and on both sides with one in configurations (a), (b), and (c). Finally, configuration (d) addresses internal flow. Studies showed a 7% increase in electrical conversion efficiency. Additionally, the PV surface temperature cannot be lowered below exhaust air temperature.

2.7.2 Discussion on active cooling using air

Figure 2.33 illustrates the various active air-cooling methods used to achieve a substantial temperature reduction in solar modules compared to those without cooling, as reported by different researchers. Kabeel et al. [125] demonstrated a significant accomplishment by achieving a maximum temperature reduction of 38 °C using an air blower combined with a water self-cleaning system. Choubineh et al. [123] reported a minimum temperature drop of 6.5 °C by using a blower with PCM. The variability in results indicates that PCM is not

compatible with active air cooling when compared to alternative air-cooling technologies. The average temperature reduction achieved by the ten active air-cooling technologies illustrated in the figure ranges from 10 to 20 °C. This consistent range shows active air-cooling technologies' reliability and efficiency, allowing comparisons with other cooling methods. Kabeel et al. [125] found that using concentrated PV and a water cleaning system simultaneously lowered temperature. This broad approach suggests combining active air cooling, concentrated PV, and extra cleaning to reduce temperature and increase solar radiation. Lebbi et al. [124] found a 15 °C temperature drop with the same technology without a solar concentrator. This comparison shows that CPV systems benefit from active air cooling. In terms of electrical efficiency, active-air cooling on a non-concentrated PV is more efficient due to the concentrator's large operating temperature rise (Figure 2.34).

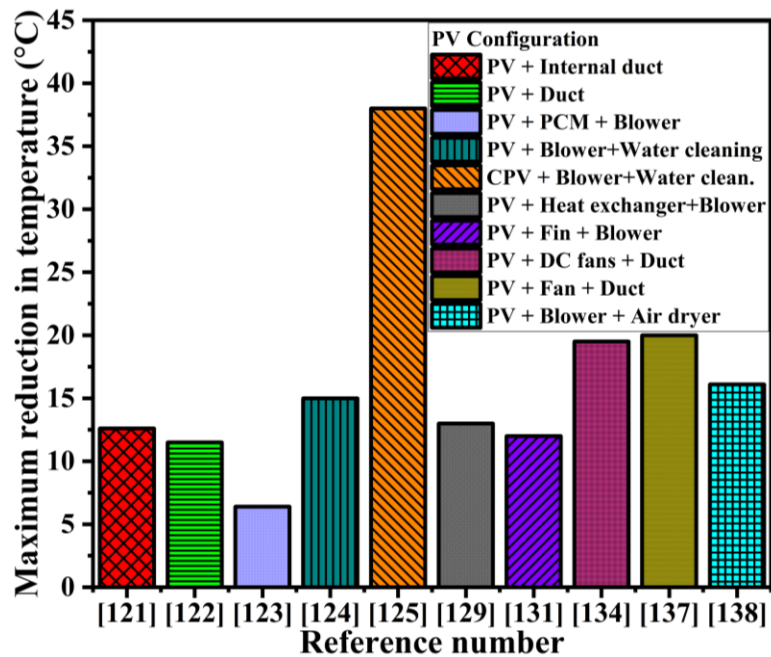


Figure 2.33: Maximum reduction in temperature using active air cooling compared to non-cooled

PV module

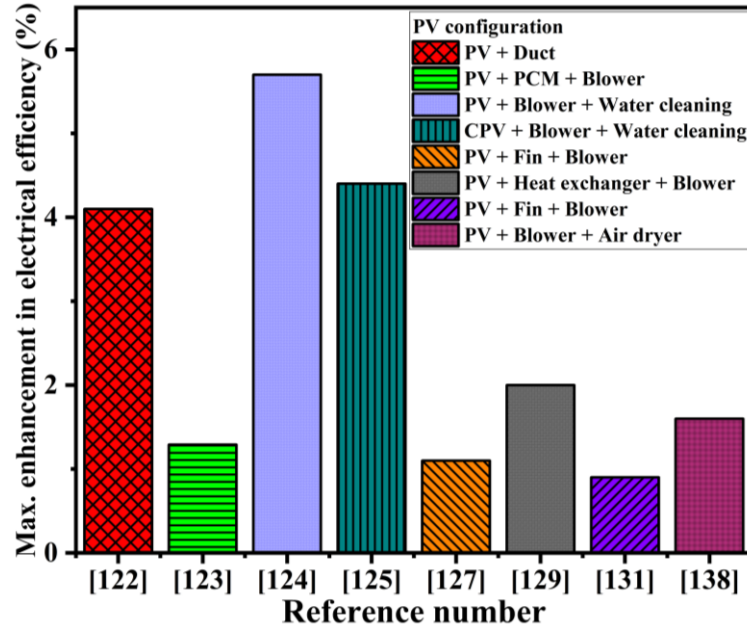


Figure 2.34: Maximum enhancement in electrical efficiency using active air cooling compared to non-cooled PV module

Figure 2.34 shows how many studies have shown that active air cooling maximizes electricity efficiency in PV modules. Lebbi et al. [124] increased electrical efficiency by 5.7%. A blower and water cleaning system were used. This suggests that active air cooling and water purification may boost electrical efficiency. Ahmed and Nabil [131] reported a 0.9% electrical efficiency increase. They used blowers and extended fins. The variation in results emphasizes the impact of setups and technologies in the field of active air cooling.

2.8 Active cooling using liquid

2.8.1 Overview of previous work

Active cooling with liquid is a popular PV thermal management method due to its capacity to provide energy in the form of electricity and heat, known as a PVT system. Various liquid-based methods for cooling PV systems are discussed in the literature, including fluid

flow in pipes [142], [143], [144], fluid flow in ducts [145], [146], [147], [148], [149], jet impingement [150], [151], [152], [153], nanofluids [154], [155], [156], [157], water spray [158], [159], [160], [161], and geothermal cooling [162]. Hattam and Aboaltabooq [143] conducted an experimental study where they designed and tested a PVT system utilizing distilled water as its operational fluid. The PVT's performance was observed by varying the flow rate from 1 to 5 liters per minute and the immersion depth. It was found that the system's efficiency rises with an increase in flow rate. Dwivedi et al. [163] compared passively cooled PV and PVT thermoelectric efficiency. Analysis showed that the PVT system lowered temperature by 10 °C. Passive cooling improved electrical energy efficiency by almost 11% in the PV module. The PVT system increased electricity efficiency by 9%. The useful thermal energy extracted from the PVT boosts its thermoelectric efficiency to 67%. In a similar study conducted by Samad et al. [164], a circulation pump helped water flow in the pipe attached to the PV's rear side while studying the system's thermoelectric performance at different flow rates. The module temperature dropped to 42 °C with 13% electrical and 60% thermal efficiency.

In addition to PVT configurations based on water circulation in pipes, another prevalent PVT configuration is duct flow as shown in Figure 2.35(a). Zubeer et al. [146] used forced water cooling in CPV systems to improve PV module performance. Three experimental setups were tested: stand-alone PV, CPV without cooling, and CPV with cooling. The cooling effect increased CPV electrical performance by 17.7%, harvesting more energy than the ordinary PV system. With the cooling system, PV operating temperature was maintained between 30 and 35%. Pomares-Hernandez et al. [147] and D'Angola et al. [148] conducted numerical studies without solar concentrators. The previous study lowered PV

module temperature to 15.01°C. Eid et al. [165] developed a PVT hybrid system using water flow (WF) and TEC. The TEC was on the back of the module, while the water system was on top. The breakthrough reduced module temperature to 21 °C, increasing efficiency by 11% and daily output by 8%. Khelifa et al. [145] demonstrated how natural fiber can serve as a porous PVT duct medium. The study focused on how the porous flax fiber layer thickness, from 5 mm to 50 mm, affected PVT performance.

Recently, researchers have become interested in jet impingement as a more modern and effective water-based cooling method for PVT systems. Figure 2.36 shows how nozzles direct a high-velocity water jet onto the heated region, targeting the PV panel's back surface. The coolant captures heat in microchannels and releases it as a jet. The direct interaction between the coolant and the heated surface improves cooling efficiency more than traditional methods. Figure 2.36(a) shows Hasan et al.'s [150] low-concentrated PVT (LCPVT) system with microchannel jet cooling. Water jets increase PV module short-circuit current by 28%. Khalaf et al. 's [151] study on water jet-cooled PVT systems found that solar reflectors increased efficiency by 13.4%, 0.7%, and 14%, indicating effective cooling of concentrated PVT systems.

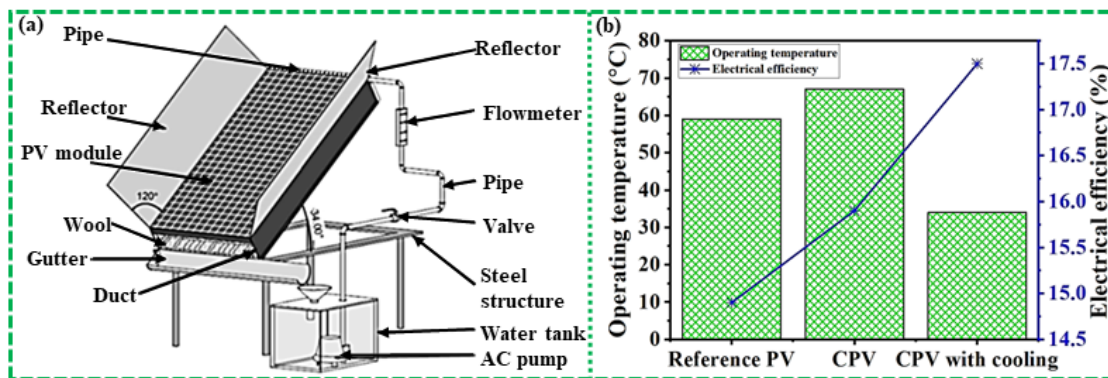


Figure 2.35: CPV with forced cooling [146]

Molavi et al. [152] study how submerged jet impingement cools solar cell fin surfaces. The fluid flow was laminar, and the solar concentration was 1000X throughout the investigation. The hybrid jet-fin configuration outperformed jet impingement cooling, according to the findings. The solar cell with fins was 12 °C cooler than the one without. Javidan and Moghadam examined how nozzle arrangement (multiple or single-orifice) affects hybrid nanofluid-jet impingement PVT cooling efficiency [153]. Multi-orifice nozzles efficiently reduce PV module temperature compared to single-orifice nozzles, with nanofluid density raising average PV temperature by 7.3°C and 6.7°C, respectively.

Forced thermal management of PV modules using nanofluids is not uncommon. Sharifpur et al. [154] found that using single-walled carbon nanotubes (SWCNT) mixed with water can improve PV performance. However, the enhancements are not significantly greater than those achieved with water. Hussein et al. [155] found that forced nanofluid cooling reduced PV temperature by 18°C and enhanced electrical efficiency by about 8%. Asim et al. [156] improved efficiency of a high-concentration solar module by 42% using a

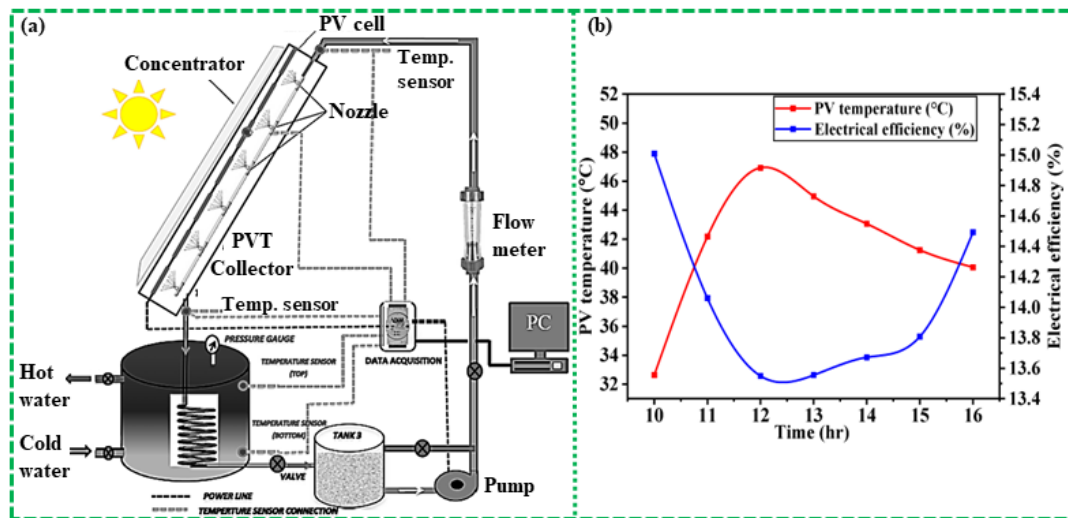


Figure 2.36: (a) Schematic diagram, and (b) Temperature-efficiency dependence of PVT jet cooling [150]

microchannel heatsink and silicon oxide nanoparticles. Islam et al. [157] integrated a transparent PDMS microchannel on concentrated PVT, lowering cell temperature below 110°C.

In addition to raising cell temperature, the buildup of dust is a major factor in the decline of PV efficiency. This issue has been thoroughly researched to find a resolution. One common method involves spraying water onto the top surface of the module, which simultaneously provides active cooling for the PV system. Kabeel et al. [158] designed a hybrid self-cleaning and cooling system for PVT. The PV module undergoes four enhancements, as shown in Figure 2.37(a), which include a curved protective covering, reflective surfaces, continuous water circulation for active cooling, and passive airflow for natural cooling. Comparing conventional and modified PV modules revealed that the modified module reduced the PV surface temperature by 29 °C, as shown in Figure 2.37(b). Consequently, it exhibited a surge of 24.11% and 18.08% in electrical and energy efficiency, respectively, when compared to conventional PV. Sargunanathan et al. [159] developed a rear-surface water PVT cooling system for PV modules, controlling excess water flow and chilling the back surface. The system decreased PV temperature by 34.7°C, 28.7°C, and 18.6°C when both surfaces were cooled. Benato and Stoppato [160] studied the effect of spray angle on a front surface water spray cooling system, finding that nozzles set at 360° and 180° were inefficient. Yusoff et al. [161] studied a hybrid integration of spray cooling and heatsinks at the rear side of PV, enhancing PV power output by 13.3%.

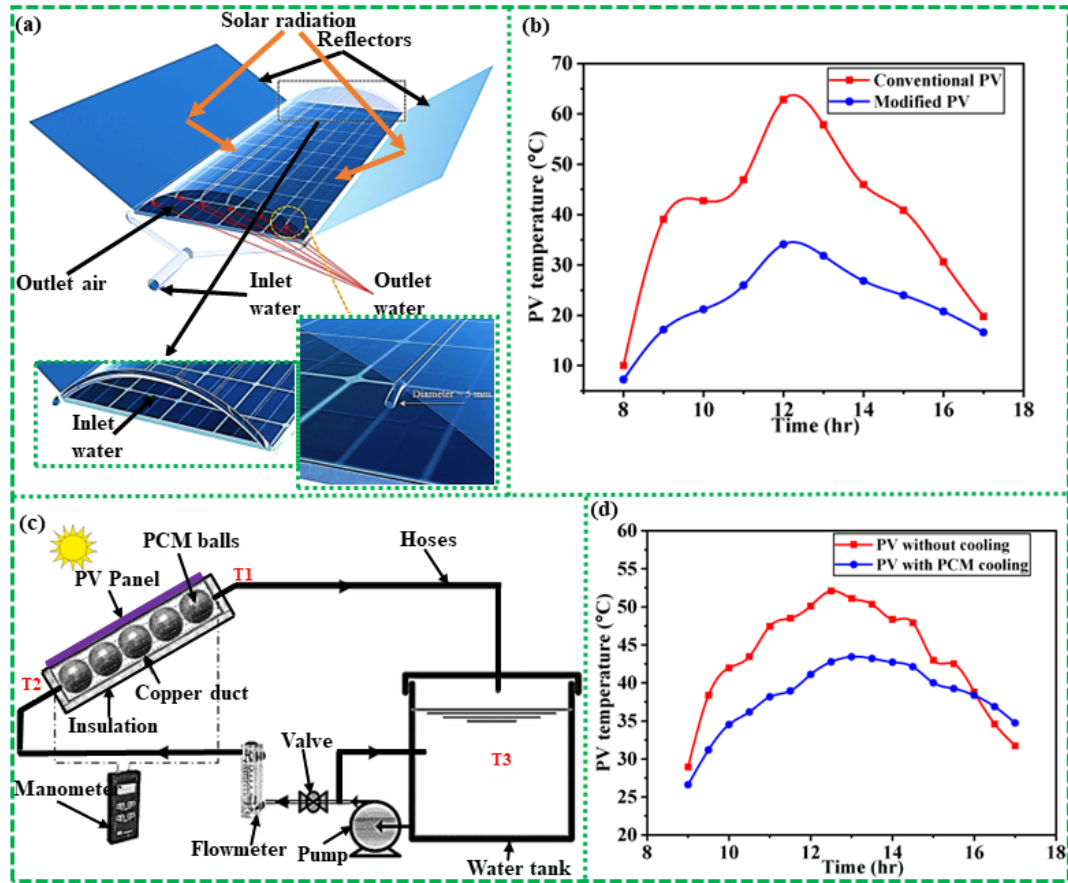


Figure 2.37: (a) Hybrid self-cleaning and active water cooling [158], (b) temperature comparison for conventional (PV-C) and modified (PV-M) module [158], (c) PVT-PCM cooling system [166], and (d) Temperature reduction by [166]

Despite not being a new technology, the PVT system was rarely integrated with PCM. Figure 2.37(c) shows Hamada et al.'s [166] innovative PVT-PCM active cooling system. This system used PCM balls and water-based PVT to improve performance. The PV's rear was connected to a 50-mm copper duct. This wide duct allowed cooling water to flow across the panel's back. Twenty-two 50-mm copper balls filled with 0.88 kg of solid RT35 PCM were placed in the reservoir to absorb heat as they melted. The PVT-PCM system was 74% efficient. PCMs are suitable for thermal management due to their latent heat capacity, but their low thermal conductivity limits their use. In addressing this issue, Dayer

et al. [167] integrated a porous copper foam matrix (CFM) into PCM, creating PVT-PCM-CFM. This resulted in a collector with a 65% thermal efficiency and 13% electrical efficiency under 1000 W/m^2 irradiance and 20°C ambient temperature. Metwally et al. [168] developed an active PV thermal management system using a thermoelectric generator (TEG) and fluid circulation. The system maintains a consistent temperature for 2 hours, reducing the panel's temperature to 38°C during summer. Lopez-Pascual et al. [162] modeled a novel active geothermal PV thermal management system, which efficiently removes extra heat from the PV module using a closed-loop cooling circuit and the underground environment, improving efficiency and power gain by 10.7% and 4.9%, respectively.

2.8.2 Discussion on active cooling using liquid

Figure 2.38 compares the maximum temperature reduction for active water-cooling methods for PV modules to non-cooled modules, as reported by several researchers. Using water flowing via a duct and CPV, Zubeer et al. [146] obtained a considerable temperature reduction of roughly 33°C . This technology demonstrates the usefulness of active water cooling in regulating excessive heat, particularly when paired with concentrated PV systems. In their study, Khalaf et al. [151] demonstrated a 9°C temperature reduction using CPVT integration with PCM. The broad range of approaches employed in active water-cooling is highlighted by various findings. Most observed temperature drops occur between 15 and 30°C . Active water-cooling technologies include duct-flowing water [147], SWCNT nanofluid [154], and PVT with self-cleaning cover [158] can reduce PV temperature by approximately 28°C . Water cooling, regardless of medium or self-cleaning, can reduce PV temperatures, highlighting its applicability in photovoltaic systems. The

efficiency of active water cooling is similar across various technologies, indicating its reliability and flexibility.

Figure 2.39 compares active water cooling versus non-cooled solar modules for electrical efficiency in various studies. Eid et al. [165] increased electrical efficiency by 11%. Their method included photovoltaics, thermoelectric cooling, and water flow. Hattam and Aboaltabooq [143], and Dayer et al. [167] reported less than 1% electricity efficiency gains. Dayer et al. [167] employed PCM and CFM, while Hattam and Aboaltabooq [143] used active water cooling with pipe flow to the module. The findings represent the importance of considering active water-cooling methods and designs to improve electricity efficiency. The different results show that active water-cooling solutions can considerably improve PV system electrical efficiency.

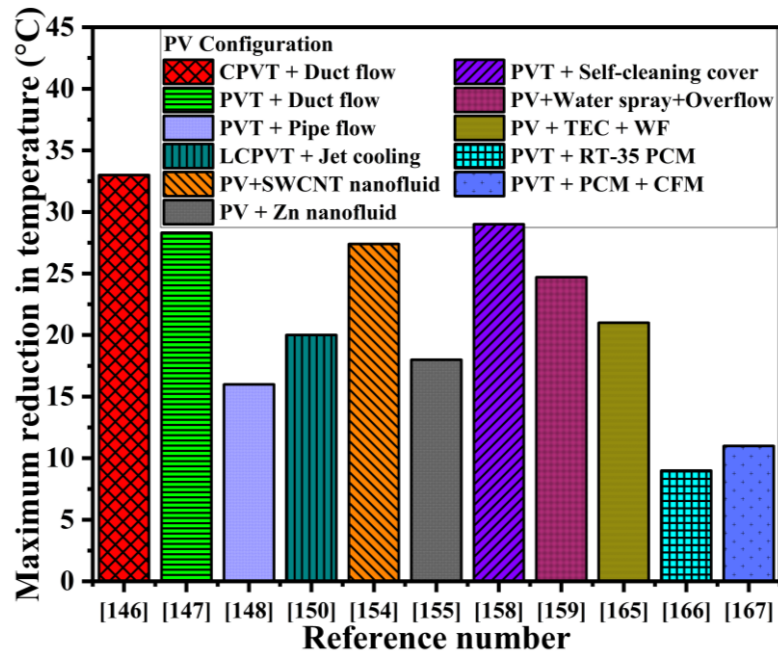


Figure 2.38: Maximum reduction in temperature using active water cooling compared to non-cooled PV module

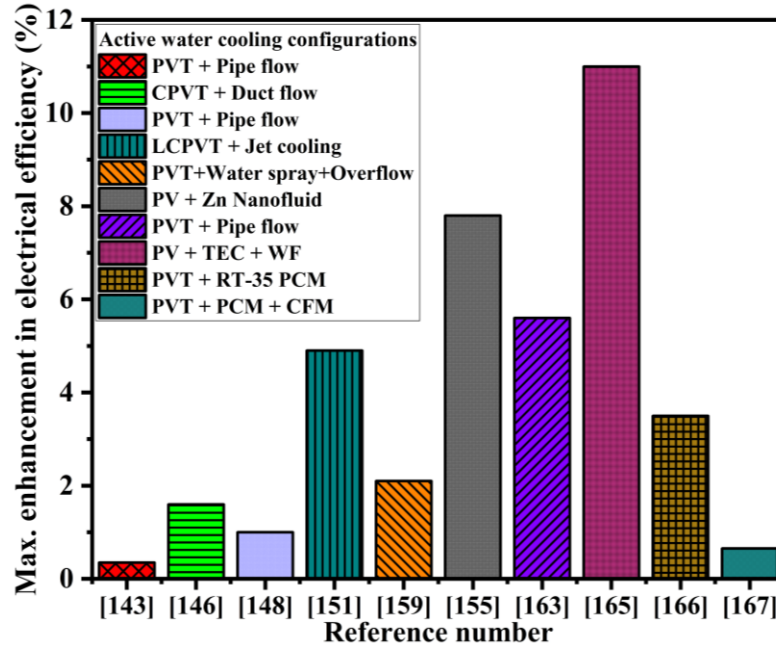


Figure 2.39: Maximum enhancement in electrical efficiency using active water cooling compared to non-cooled PV

The comprehensive literature review presented above highlights the various advancements and challenges in PV thermal management, including passive and active cooling techniques. While numerous studies have focused on optimizing finned heatsinks for air cooling or integrating water-based cooling systems, the approaches have largely been investigated independently. There is a notable lack of studies that explore the simultaneous integration of air-cooled fin heatsinks with water pipe cooling systems for enhanced PV thermal management. Such a dual-cooling approach has the potential of addressing issues of non-uniform temperature distribution and excessive heat under high-irradiance conditions. Despite significant efforts in modeling, analysis, and optimization of PV cooling systems, the literature reveals no standard or unified methodology that integrates these two cooling mechanisms effectively. This gap in the research motivated the present work, which aims to investigate and optimize an innovative hybrid cooling system combining water pipe and air-cooled fin technologies to improve PV efficiency.

Table 2-6: Summary of proposed work by different researchers for cooling PV module using forced air and water cooling

Reference (year)	Cooling medium		Type of PV integration	Nature of work	Maximum T_{PV-ref} ($^{\circ}C$)/ $T_{PV-Air \text{ or } Liquid}$ ($^{\circ}C$)	Key Findings
	Air	Liquid				
Singh et al. [121] (2023)	✓		PV + Internal duct	Numerical	63/50.4	- Internal cooling is more effective than external cooling. - Internal cooling can boost net PV power by 9.6% (5% above external cooling).
Abdrahman et al. [126] (2023)	✓		PVT + Fins	Experimental	NA	17% electrical efficiency was achieved by the proposed design.
Almuwailhi and Zeitoun [122] (2023)	✓		PV + Duct	Experimental	64.5/53	- PV temperature decreased by 11.5 $^{\circ}C$ and efficiency increased by 4.1%. - Forced convection is more efficient than natural convection with evaporation.
Hussien et al. [133] (2023)	✓		1.PV + Blower + Duct 2.PV + DC fans + Duct	Experimental Numerical	1. 62.8/55.6 2. 62.8/51.3	- Small fans are more efficient than blowers. - Maximum T_{PV} reduction of 11.5 $^{\circ}C$ and 7.2 $^{\circ}C$ were recorded, respectively. - Power output improved by 21 and 12%.
Almeshaiei et al. [137] (2023)	✓		PV + Fan + Duct	Experimental	67/47	- The temperature drop increased the relative efficiency by 30% and annual energy production by 12%. - Fan-cooled systems pay back in 2.6 years.
Mojumder et al. [127] (2023)	✓		PV + Fin + Blower	Experimental	59.8/53.1	- Longitudinal fins are more effective than S-shaped fins. - The maximum PV temperature reduced by 12.6%. while the efficiency improved by about 1.1%.
Nabil and Mansour [132] (2022)	✓		PV + Blower + Fin	Experimental	52/45	The maximum PV temperature reduced by 7 $^{\circ}C$. This improved the efficiency by about 1%.
Lebbi et al.[124] (2021)	✓		PV + Blower + Water cleaning	Experimental	46/31	Maximum PV temperature decreased by 15 $^{\circ}C$, and electrical efficiency increased by 5.7%.

Reference (year)	Cooling medium Air Liquid		Type of PV integration	Nature of work	Maximum T_{PV-ref} ($^{\circ}C$)/ $T_{PV-Air \text{ or } Liquid}$ ($^{\circ}C$)	Key Findings
Abushgair [129] (2021)	✓		PV + Heat exchanger + Blower	Experimental	57/44	Maximum PV temperature reduced by 13 $^{\circ}C$ while the efficiency improved by 2%.
Deokar et al. [138] (2021)	✓		PV + Blower + Air dryer	Experimental	71.5/55.4	The air velocity is directly proportional to the electrical efficiency and inversely proportional to the drying time.
Choubineh et al.[123] (2019)	✓		PV + PCM + Blower	Experimental	62/55.6	- The maximum temperature reduction by the PCM was 6.4 $^{\circ}C$. And efficiency increased by 1.29%. - The impact of PCM is significant in the morning and nearly negligible in the afternoon.
Kabeel et al. [125] (2019)	✓		CPV + Blower + Water cleaning	Experimental	71/33	- The solar reflector significantly increased the intensity reaching the solar cell. - Net output electrical power of 178 Wh/day was achieved. - The levelized cost of energy was 0.0722 Wh/day.
Syafiqah et al. [134] (2017)	✓		PV + DC fans + Duct	Numerical	66.2/47.0	- The temperature of the PV system decreases as the speed of the fan increases. - Fan velocity increases energy consumption, reaching a negative value when it exceeds 4.5 m/s.
Ahmed and Nabil [131] (2017)	✓		1.PV + Blower 2.PV + Blower + Fin	Numerical	1. 61/57 2. 61/49	- Fins improve PV cell cooling and lower exit temperature. - Increasing air velocity helps photovoltaic cooling. Low outlet air temperature results from this rise in velocity.
Amelia et al. [135] (2016)	✓		PV + DC fans + Duct	Experimental	54.01/39.86	- The number of DC fans is directly proportional to the efficiency. - Optimum number of DC fans depends on the speed of the fan and PV size.
Hasan et al. [150] (2024)		✓	LCPVT + Jet cooling	Experimental	66.5/46.5	- A temperature rise of around 34 $^{\circ}C$ led to a 2.25% loss in efficiency. - The LCPVT with jet cooling improved the short circuit current and consequently the power output.

Reference (year)	Cooling medium		Type of PV integration	Nature of work	Maximum T_{PV-ref} ($^{\circ}C$)/ $T_{PV-Air \text{ or } Liquid}$ ($^{\circ}C$)	Key Findings
Dwivedi et al. [163] (2023)		✓	PVT + Pipe flow	Experimental	57.2/47.8	- The power output increased by 5.6 W. - The electrical and thermal efficiency improved by 5.6% and 58.8%.
Khelifa et al. [145] (2023)		✓	PVT + Duct flow + Flax fiber	Numerical	--	- Incorporating flax fibers significantly improved the performance of the heatsink. - The Nusselt number and the thermal efficiency improved by 32 and 101%, respectively.
Khalaf et al. [151] (2023)		✓	LCPVT + Jet cooling	Experimental	64/56	- The use of reflectors and jet cooling increased the electrical efficiency of the PV by 4.9%. - Introducing reflectors greatly enhanced the system's overall efficiency by 14% compared to PV with jet cooling.
Hamada et al. [166] (2023)		✓	PVT + RT-35 PCM	Experimental	52/43	- The Peak temperature dropped by 9 C, while the electrical efficiency and power output improved by 3.5% and 2 W. - Adding insulation to the back side of the PVT enhanced heat gain and overall effectiveness.
Dayer et al. [167] (2023)		✓	PVT + PCM +CFM	Numerical	69/58	- CFM lowered PV + PCM system temperature by 11 $^{\circ}C$ and boosted efficiency by 0.65%. - CFM enhanced PCM phase change by 12% due to its improved effective conductivity.
Hattam and Aboaltaboq [143] (2022)		✓	PVT + Pipe flow	Experimental	--	- Increasing the flow rate results in a drop in both PV and outlet water temperature. - The electrical efficiency also improved by 0.35%.
Eid et al. [165] (2022)		✓	PV + TEC + WF	Experimental Numerical	53/32	- The peak PV temperature was reduced by 21 $^{\circ}C$, while the electrical performance improved by 11%. - The water flow rate significantly affects the system efficiency, with pump power accounting for 17% of the total daily energy gain.
Javidan and Moghadam [153]		✓	PV + SiC nanofluid jet cooling	Experimental	--	Nanoparticles decreased the temperature of the photovoltaic system by 6 $^{\circ}C$ in comparison to using pure water for cooling.

Reference (year)	Cooling medium		Type of PV integration	Nature of work	Maximum T_{PV-ref} ($^{\circ}C$)/ $T_{PV-Air \text{ or } Liquid}$ ($^{\circ}C$)	Key Findings
(2022)						Higher nanofluid flow rate and concentration can improve electrical and thermal properties but impair hydrodynamics.
Sharifpur et al. [154] (2022)		✓	PV + SWCNT nanofluid	Numerical	68/40.6	- At 1% nanoparticle concentration, the PV temperature dropped by 27 $^{\circ}C$. - The cell's efficiency increased by 49%.
Samad et al. [164] (2021)		✓	PVT + Pipe flow	Numerical	42/29	- The PV temperature reduced to 42 $^{\circ}C$. - The electrical and thermal efficiency improved by 1.11 and 61.2%. - Optimizing the pump reduced electrical inefficiency.
Zubeer et al. [146] (2021)		✓	CPVT + Duct flow	Experimental	67/34	- The operating temperature of the CPV was reduced by 33 $^{\circ}C$. - The electrical efficiency and the power output improved by 1.6% and 20 W. - The payback period of the CPVT is 3.23 years.
Pomares-Hernandez et al. [147] (2021)		✓	PVT + Duct flow	Numerical	67.34/39.03	- The active water cooling reduced the average working temperature of the PV by 15 $^{\circ}C$. - The power output of the PV improved by 23.6 W.
D'Angola et al. [148] (2020)		✓	PVT + Pipe flow	Numerical	37/21	The peak PV temperature was reduced by 16 $^{\circ}C$, while the electrical performance improved by 1%.
Sargunanathan et al. [159] (2020)		✓	PVT + Water spray + Overflow	Experimental	74/39.3	The maximum PV temperature reduced by 34.7 C while the electrical efficiency improved by 2.1%.
Kabeel et al. [158] (2019)		✓	PVT + Concave self-cleaning cover	Experimental	63/34	- The peak PV temperature was reduced by 29 $^{\circ}C$ using this system. - The overall energy efficiency was improved by 18%.
Hussein et al. [155] (2017)		✓	PV + Zn nanofluid	Experimental	76/58	- The peak PV temperature was reduced by 18 $^{\circ}C$, while the electrical performance improved by 7.8%. - The nanoparticles enhanced the PVT electrical performance by 1.3% compared to water cooling alone.

CHAPTER 3

OPTIMIZING FIN HEATSINKS FOR PHOTOVOLTAIC COOLING

3.1 Introduction

Fins, also known as extended surfaces, are commonly employed in heat exchangers to improve heat transfer between the primary surface and the surrounding fluid. There are two main categories of finned heatsinks: plate fin and pin fin heatsink. Plate fins are longitudinal fins that are attached directly to the surface of the heatsink, with the cooling fluid flowing parallel to the fins' longitudinal axis. Pin fins are a collection of tiny, solid cylindrical pins connected to the heatsink. Despite the popularity of pin fins in various applications, rectangular plate fins continue to be the prevailing choice for heatsinks. This is attributed to several significant practical benefits compared to pin fins, such as straightforward design, simplicity in manufacture, and relatively low-pressure. Several studies have been conducted to investigate the potential enhancement of heat transfer rates through the restriction of areas where air can flow smoothly through heatsink channels. Adhikari et al. [9] show a noticeable drop in heat transfer with increasing channel length, which is roughly linear. Thus, concluding that the thermally developing flow regime is more efficient in transferring heat than the developed flow.

The main objective of this chapter is to optimize fin heatsink for PV cooling by employing the concept of thermal boundary layer redevelopment along the rectangular heatsink length by cutting the fin or reducing the fin thickness and replanting it at different spots, thus

allowing for the development of a new boundary layer, as opposed to limiting the heatsink length to maintain developing regime as adopted by Adhikari et al. [169]. Extensive research efforts over the years have been dedicated to the investigation of heat transfer within vertical [170], [171], [172] and horizontal [173], [174], [175] channels containing heated obstructions, leading to a comprehensive body of literature on this subject.

In this chapter, the performance of a rectangular plate-fin heatsink is optimized by changing the configuration and the distribution of the plate-fin heatsink. Three optimizations to the fin type are proposed, namely, Bifurcated longitudinal split fins (BLSF), Variable height gradient fins (VHGF), and Hybrid plate-pin fins (HPPF). Also, the effect of perforation on the heat transfer performance is studied. The major design constraint is maintaining constant volume for all the proposed optimizations while reducing the fin volume in the percentage of the total volume using perforations to determine the optimum fin volume.

3.2 Model description

The fundamental concept underlying this study involves leveraging the impact of the thermal boundary layer development, as depicted in Figure 3.1 to enhance the heat dissipation capabilities of the rectangular plate fin heatsink. At the entrance region, the heat transfer coefficient from the fins to the air increases until the flow reaches a fully developed stage where the heat transfer coefficient becomes constant. Subsequently, in the fully developed region, the heat transfer coefficient decreases when the air flowing in the channel becomes heated to almost the same temperature as the fin surface. The regulation of the thermal boundary layer will primarily be achieved using various fin types and their distribution while maintaining constant fin volume.

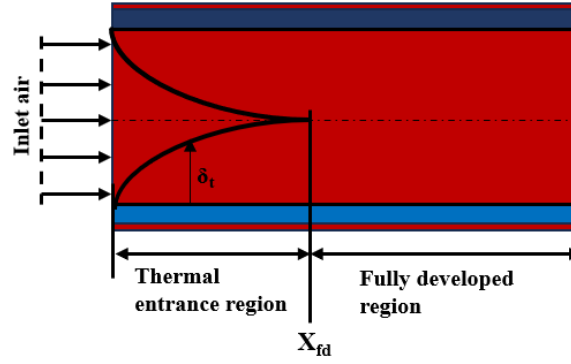


Figure 3.1: Thermal boundary layer development

Four proposed modifications are bifurcated longitudinal split fins (BLSF), variable height gradient fins (VHGF), hybrid plate-pin fins (HPPF), and perforated plate fins (PPF). The overall volume of the fin remains constant at $8 \times 10^{-5} \text{ m}^3$ (80000 mm^3). The choice of this volume is determined by the overall amount of space available to accommodate the heatsink and, consequently, the fins.

3.2.1 Rectangular plate fin (RPF)

Figure 3.2 illustrates the conventional heatsink with rectangular plate fins. The morphing design parameters include the number of fins (n), fin thickness (t_f), and fin spacing (S). However, the dimensions of the fin, including the height (H), base plate thickness (t_b), width (B), and length (L), remain consistent throughout the entire study. Table 3-3 presents the dimensions of these parameters at different fin numbers.

3.2.2 Bifurcated longitudinal split fins (BLSF)

Figure 3.3 depicts the heatsink with a bifurcated longitudinal split fin. The main objective of this design is to control the thermal boundary layer by dividing the plate fin (n_2) at a specific length

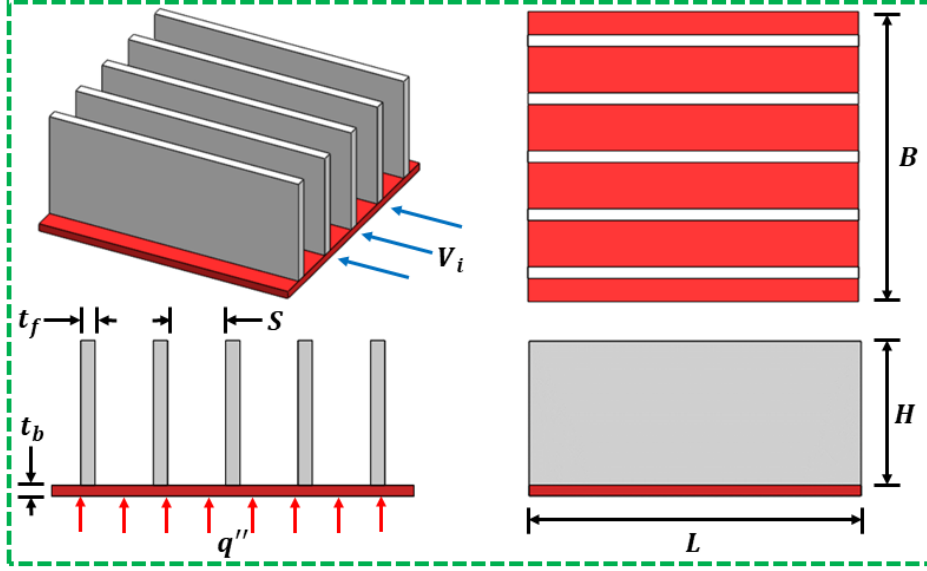


Figure 3.2: Conventional rectangular plate fin

(L_2), splitting the remaining half into two (n_3), and then replanting it to the base plate. The primary design factor is the specific length (L_2) at which the plate fin divides, enabling it to morph in percentage of the overall fin length (L), represented as:

$$L^* = \frac{L_2}{L} \quad (3.1)$$

This optimization is performed on a heatsink with five fins ($n_1 = 3$, $n_2 = 2$, and $n_3 = 4$). Hence, all other parameters remain constant except L_2 and L_3 .

Afterward, we parametrically increase the number of fins in the design with the most optimal bifurcation length (L_2), ranging from 1 to 9. Throughout this parametric analysis, the only variables that remain constant are the thickness (t_b), length (L), and width (B) of the base plate. Additionally, the thicknesses of the fins (t_{f2} and t_{f3}) and the spacing between them (S_1 and S_2) can also vary. Similarly, the major geometric constraint is to maintain constant heatsink volume as the cut length (L_2) is allowed to morph freely in percentage of the total length (L).

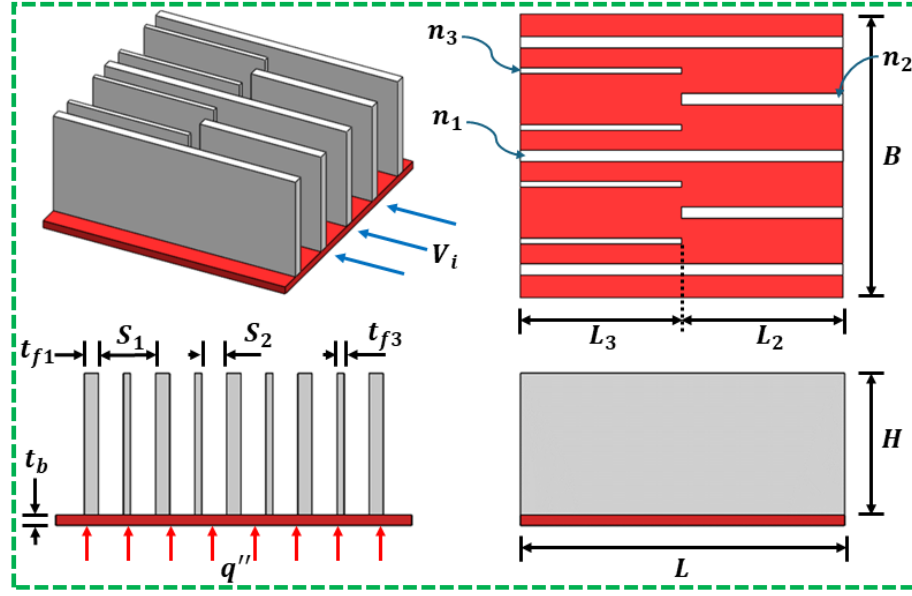


Figure 3.3: Bifurcated longitudinal split fin

3.2.3 Variable height gradient fins (VHGF)

Figure 3.4 shows the diagrammatic description of the variable-height gradient fin. Increasing or decreasing the fin height longitudinally is the primary design goal, which controls the thermal boundary layer. By setting the fin height at the inlet (H_1) and outlet (H_2) as a percentage of the original base case fin height (H), we make sure the sum of H_1 and H_2 is equal to H . Therefore, the optimization parameter is defined as the ratio:

$$H^* = \frac{H_1}{H} \quad (3.2)$$

As we progress from $L = 0$ to $L = 100 \text{ mm}$ in the flow direction, the longitudinal increase in fin height results in a variation of H^* from 0 to 0.5. Furthermore, the optimization parameter H^* exhibits a range of values between 0.5 and 1 as the fin height decreases in the longitudinal direction. It is noteworthy that when $H^* = 0.5$, the heatsink corresponds to the base case (RPF). Also, the optimization study was performed on a heatsink with five fins, thus, all other parameters remain constant except H_1 and H_2 .

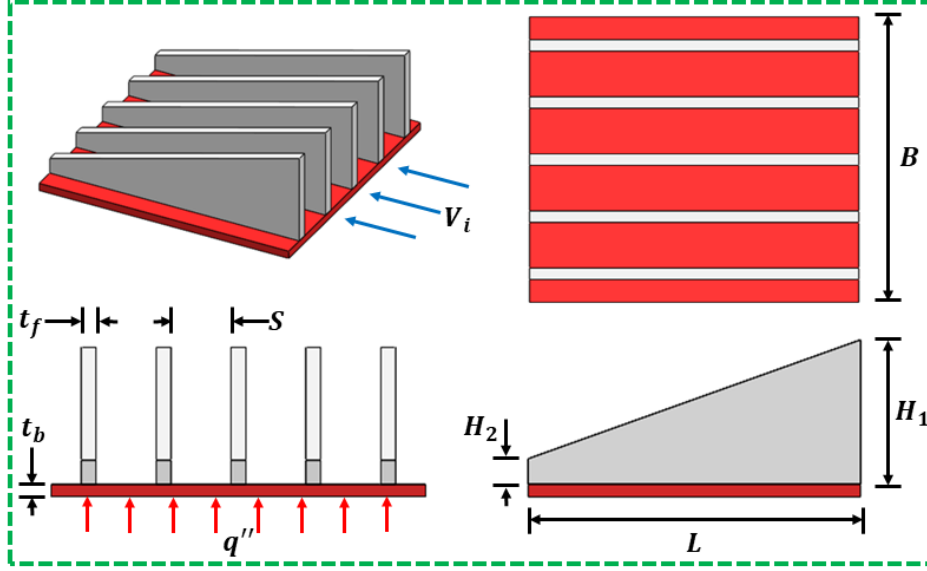


Figure 3.4: Variable height gradient fins

Once we determined the design with the most optimum H^* configuration, we conducted a parametric analysis to examine the impact of the number of fins. We systematically varied the number of fins from 1 to 9. The parametric study produces distinct fin parameters for each case. More precisely, the dimensions of the fin, including the fin thickness (t_f), fin spacing (S), base plate thickness (t_b), length (L), and width (B), are morphing.

3.2.4 Hybrid plate-pin fins (HPPF)

Figure 3.5 represents the arrangement of the hybrid plate-pin fin heatsink. The primary design concept entails decreasing the thickness (t_f) of the plate fin to embed a pin fin onto the plate fin's surface, thereby converting a specific proportion of the plate fin's volume into the pin fin. As the thickness of the plate fin (t_f) decreases, the height of the pin fin (H_p) increases while keeping the pin diameter (d_p) constant. We optimize the reduction in plate fin thickness from 10% to 50% in increments of 10%. The pin fin height (H_p) and plate fin spacing (S) vary in these configurations. Yet, the dimensions of the base plate, including its thickness (t_b), width (B), and length (L), as well

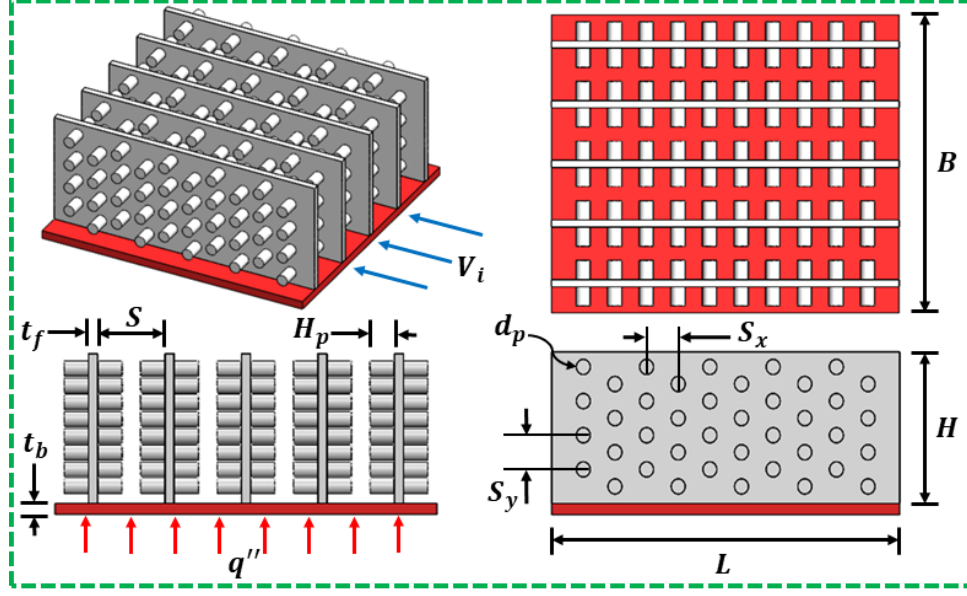


Figure 3.5: Hybrid plate-pin fins

as the height (H) of the plate fins, the number of pin fins, and the spacing between them in both the horizontal (S_x) and vertical (S_y) directions, remain unchanged. In addition, we utilize a staggered array configuration of 4 by 10 pin fins. We conduct a systematic analysis to examine how the number of plate fins impacts the optimal plate-pin configuration that we identify. Modifying the number of plate fins from 1 to 9 results in an alteration in the thickness of each plate fin (t_f).

3.2.5 Perforated plate fins (PPF)

This section focuses on analyzing the impact of perforations on the heat dissipation capabilities of a rectangular plate fin heatsink. Four perforation types are classified according to their shape. The various types of perforations depicted in Figure 3.6 include circular, square, vertical and horizontal rectangular slots. The perforations are coded as P1, P2, P3, and P4, respectively. In this scenario, we decrease the volume of the fin material by a percentage of the total fin volume, ranging from 10% to 60%, with increments of 10%.

Consequently, the design criteria involve balancing the material volume and heat transfer performance of the plate fin. The initial parametric analysis aims to assess the impact of these perforations on a five-finned rectangular plate-fin heatsink under various volume reductions. To accomplish this, we enlarge the diameter of the holes (d_h), the side lengths (a), the length of holes (L_h), and the height of the holes (H_h) for P1, P2, P3, and P4, respectively. Nevertheless, we consistently maintain a fixed number of perforations throughout the process, with 21 perforations in P1 and P2, and 9 perforations in P3 and P4. Table 3-2 presents the different dimensions for these perforations in all scenarios.

Subsequently, we investigate the impact of the optimal perforation (P1) on the number of fins by altering the perforated plate fins from 1 to 9. In this case, the quantity of perforations, the size of each perforation (d_h), the distance between perforations horizontally (S_x) and vertically (S_y) remain unchanged. Table 3-3 additionally displays the dimensions.

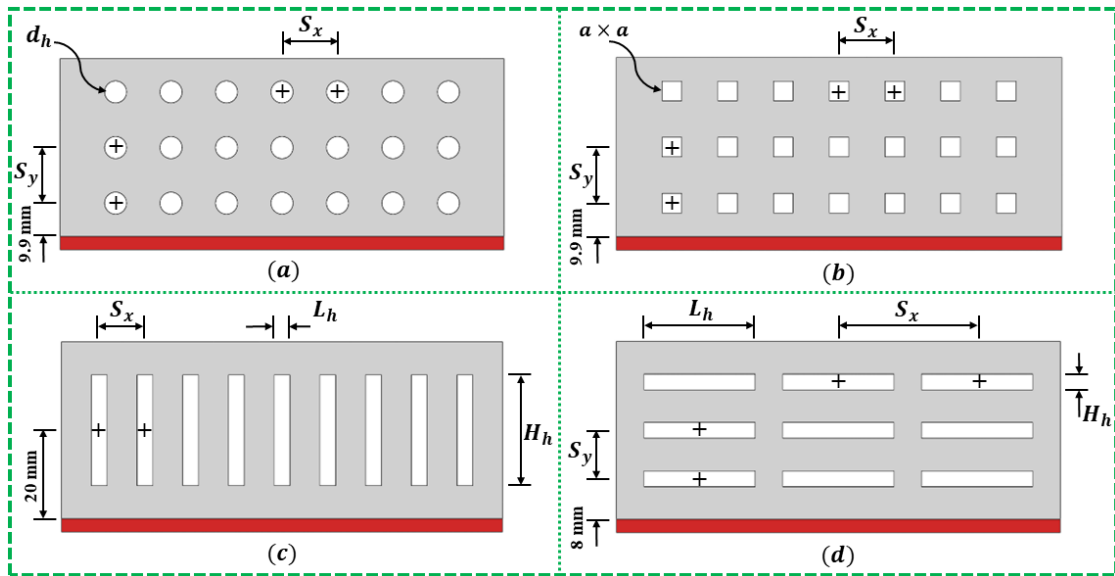


Figure 3.6: Types of perforation (a) circular, (b) rectangular, (c) vertical slots, and (d) horizontal slots

Table 3-1: Dimensions of the optimization scenarios for BLSF, VHGF, and HPPF heatsinks

Type of fin	Number of fins			Thickness of fins (mm)		Spacing (mm)		Optimization	Fin height (mm)	Base length (mm)	Base width (mm)	Total heat transfer area (cm ²)	
BLSF	<i>n</i>	<i>n</i> ₁	<i>n</i> ₂	<i>t</i> _{<i>f</i>2}	<i>t</i> _{<i>f</i>3}	<i>S</i> ₁	<i>S</i> ₂	<i>L</i> [*]	<i>H</i>	<i>L</i>	<i>B</i>	<i>A</i> _{<i>hs</i>}	
	5	3	2	4	2	16	7	0.1	40	100	100	666	
	5	3	2	4	2	16	7	0.2	40	100	100	650	
	5	3	2	4	2	16	7	0.3	40	100	100	634	
	5	3	2	4	2	16	7	0.4	40	100	100	618	
	5	3	2	4	2	16	7	0.5	40	100	100	602	
	5	3	2	4	2	16	7	0.6	40	100	100	586	
	5	3	2	4	2	16	7	0.7	40	100	100	570	
	5	3	2	4	2	16	7	0.8	40	100	100	554	
5	3	2	4	2	16	7	0.9	40	100	100	538		
VHGF	<i>n</i>			<i>t</i> _{<i>f</i>}		<i>S</i>		<i>H</i> [*]	<i>H</i>	<i>L</i>	<i>B</i>	<i>A</i> _{<i>hs</i>}	
	5			4		16.00		0	40	100	100	521.62	
	5			4		16.00		0.1	40	100	100	519.74	
	5			4		16.00		0.2	40	100	100	518.18	
	5			4		16.00		0.3	40	100	100	517	
	5			4		16.00		0.4	40	100	100	516.26	
	5			4		16.00		0.5	40	100	100	516	
	5			4		16.00		0.6	40	100	100	516.26	
	5			4		16.00		0.7	40	100	100	517	
	5			4		16.00		0.8	40	100	100	518.18	
	5			4		16.00		0.9	40	100	100	519.74	
5			4		16.00		1	40	100	100	521.62		
HPPF	<i>n</i>	<i>n</i> _{<i>p</i>}	<i>t</i> _{<i>f</i>}	<i>H</i> _{<i>p</i>}	<i>d</i> _{<i>p</i>}	<i>S</i>	<i>S</i> _{<i>x</i>}	<i>S</i> _{<i>y</i>}	<i>V</i> [*] (%)	<i>H</i>	<i>L</i>	<i>B</i>	<i>A</i> _{<i>hs</i>}
	5	40	3.6	1.6	4	16.4	9	9	10	40	100	100	613
	5	40	3.2	3.2	4	16.8	9	9	20	40	100	100	673
	5	40	2.8	4.8	4	17.2	9	9	30	40	100	100	752
	5	40	2.4	6.4	4	17.6	9	9	40	40	100	100	830
	5	40	2	8	4	18	9	9	50	40	100	100	909

Table 3-2: Dimensions of the optimization scenarios for PPF heatsink

Perforation	Reduction in volume (%)	Number of fins	Number of holes	Control parameter		Spacing (mm)		Total heat transfer area (cm ²)
P1	$V^{\#}$	n	n_h	d_h		S_x	S_y	A_{hs}
	10	5	21	4.92		14.17	12.5	541
	20	5	21	6.96		14.17	12.5	528
	30	5	21	8.53		14.17	12.5	509
	40	5	21	9.85		14.17	12.5	486
	50	5	21	11.01		14.17	12.5	461
	60	5	21	12.06		14.17	12.5	435
P2	$V^{\#}$	n	n_h	a		S_x	S_y	A_{hs}
	10	5	21	4.36		14.17	12.5	549
	20	5	21	6.17		14.17	12.5	540
	30	5	21	7.56		14.17	12.5	523
	40	5	21	8.73		14.17	12.5	503
	50	5	21	9.76		14.17	12.5	480
	60	5	21	10.69		14.17	12.5	456
P3	$V^{\#}$	n	n_h	L_h	H_h	S_x		A_{hs}
	10	5	9	1.78	25	10.82		572
	20	5	9	3.56	25	10.82		539
	30	5	9	5.33	25	10.82		505
	40	5	9	7.11	25	10.82		472
	50	5	9	8.89	25	10.82		438
	60	5	9	10.67	25	10.82		404
P4	$V^{\#}$	n	n_h	L_h	H_h	S_x	S_y	A_{hs}
	10	5	9	25	1.78	31.25	11.97	572
	20	5	9	25	3.56	31.25	11.97	539
	30	5	9	25	5.33	31.25	11.97	505
	40	5	9	25	7.11	31.25	11.97	472
	50	5	9	25	8.89	31.25	11.97	438
	60	5	9	25	10.67	31.25	11.97	404

Table 3-3: Dimensions of the scenarios for the effect of number of fins on the RPF, BLSF, VHGF, HPPF, and PPF

Type of fin	Number of fins			Thickness of fins (mm)		Spacing (mm)		Optimum parameter	Fin height (mm)	Base length (mm)	Base width (mm)	Total heat transfer area (cm ²)	
RPF	n			t_f		S		-	H	L	B	A_{hs}	
	9			2.22		8.89		-	40	100	100	836	
	7			2.86		11.43		-	40	100	100	676	
	5			4.00		16.00		-	40	100	100	516	
	3			6.67		26.67		-	40	100	100	365	
	1			20.00		40.00		-	40	100	100	196	
BLSF	n	n_1	n_2	t_{f2}	t_{f3}	S_1	S_2	L^*	H	L	B	A_{hs}	
	9	5	4	2.22	1.11	8.88	3.88	0.1	40	100	100	1131	
	7	4	3	2.86	1.43	11.49	5.00	0.1	40	100	100	899	
	5	3	2	4.00	2.00	16.00	7.00	0.1	40	100	100	666	
	3	2	1	6.66	3.33	26.67	11.67	0.1	40	100	100	433	
	1	0	1	20.00	10.00	40.00	-	0.1	40	100	100	284	
VHGF	n			t_f		S		H^*	H	L	B	A_{hs}	
	9			2.22		8.89		1	40	100	100	842	
	7			2.86		11.43		1	40	100	100	682	
	5			4.00		16.00		1	40	100	100	522	
	3			6.67		26.67		1	40	100	100	362	
	1			20.00		40.00		1	40	100	100	202	
HPPF	n	n_p	t_f	H_p	d_p	S	S_x	S_y	V^* (%)	H	L	B	A_{hs}
	9	40	1.11	4.42	4.00	10	9	9	50	40	100	100	1227
	7	40	1.43	5.68	4.00	13	9	9	50	40	100	100	1068
	5	40	2.00	8.00	4.00	18	9	9	50	40	100	100	909
	3	40	3.33	13.26	4.00	30	9	9	50	40	100	100	746
	1	40	10.00	39.79	4.00	45	9	9	50	40	100	100	586
PPF	n	n_h	t_f	d_h	S	S_x	S_y	$V^\#$ (%)	H	L	B	A_{hs}	
	9	21	2.22	6.96	8.89	14.17	12.5	20	40	100	100	784	
	7	21	2.86	6.96	11.43	14.17	12.5	20	40	100	100	656	
	5	21	4.00	6.96	16.00	14.17	12.5	20	40	100	100	528	
	3	21	6.67	6.96	26.67	14.17	12.5	20	40	100	100	400	
	1	21	20.00	6.96	40.00	14.17	12.5	20	40	100	100	272	

Table 3-4: Properties of aluminium and air

Property	Unit	Aluminium	Air
Thermal conductivity	$W/(m \cdot K)$	238	0.02887
Heat capacity at constant pressure	$J/(kg \cdot K)$	900	1005.5
Density	kg/m^3	2700	1.204

3.3 Numerical method

The numerical simulations were performed using the commercial CFD software, COMSOL Multiphysics. The simulations were conducted on a three-dimensional computational domain, considering both steady-state and turbulent flow conditions. The simulations consider the fluid flow and heat transfer phenomena occurring within the heatsink, encompassing convection and conduction. However, we have overlooked the effects of radiation since it has less impact on the overall thermal performance of the heatsink [176].

3.3.1 Boundary conditions

Each heatsink model comprised two domains: the air domain, which represented the fluid flow, and the solid domain, which represented the material of the heatsink. Table 3-4 presents the characteristics of air and aluminium that were used in the simulations. Before performing the numerical simulations, suitable boundary conditions were imposed on the computational domain to replicate realistic operating conditions. These boundaries are shown in Figure 3.7 and the condition at each boundary is discussed below.

At the inlet: A velocity vector with magnitudes of 4, 8, and 12 m/s was specified. The temperature of the incoming fluid was maintained at a constant value of 293 K. A turbulent intensity of 0.05 (medium) was specified, and the selection of the turbulent flow model was based on the Reynolds number, which varied from 1.5×10^3 to 4.5×10^3 for the hydraulic

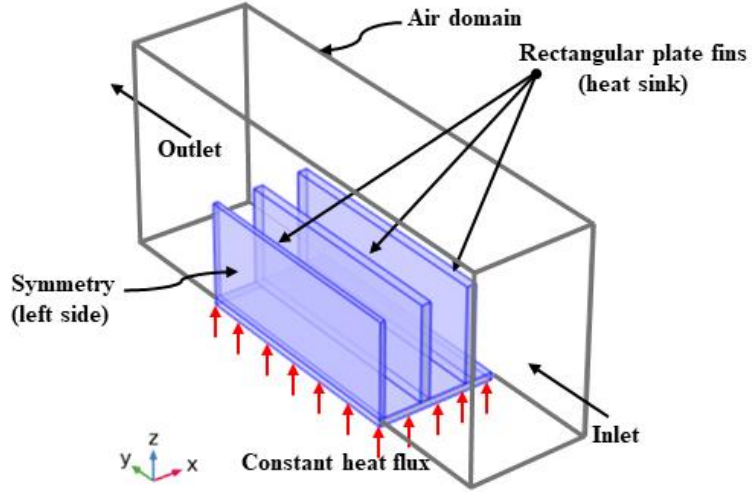


Figure 3.7: Boundary conditions

diameter of the air domain. The heat flux (q) through the inlet boundary was estimated using the Danckwerts condition on the enthalpy while the normal inflow velocity ($\mathbf{u}$) is specified as equation (3.4):

$$-\mathbf{n} \cdot \mathbf{q} = \rho \Delta H \mathbf{u} \cdot \mathbf{n} \quad (3.3)$$

$$\mathbf{u} = -U_o \mathbf{n} \quad (3.4)$$

where ρ is the density, ΔH is the sensible enthalpy, $\mathbf{u}$ is the velocity vector $\mathbf{n}$ is the boundary normal pointing out of the domain, and U_o is the normal inflow speed.

At the outlet: The outlet gauge pressure was adjusted to 0, and the suppress backflow option was enabled to prevent the occurrence of backflow and divergence. The air outlet boundary condition assumes no heat flux through the boundary as shown in equation 3-5 and specifies that the total stress is balanced by the ambient pressure, ensuring a smooth exit of air from the domain as shown in equation (3.6).

$$-\mathbf{n} \cdot \mathbf{q} = 0 \quad (3.5)$$

$$[-p\mathbf{I} + \mathbf{K}]\mathbf{n} = -p_o\mathbf{n} \quad (3.6)$$

where $p\mathbf{I}$ is the pressure tensor, $\mathbf{K}$ is the stress tensor, and p_o is the pressure at the outlet.

At the right and top side: These sides of the computational domain were designated as an adiabatic wall with no-slip conditions, effectively providing thermal insulation as described in equation (3.5).

At the left side: A symmetric boundary condition was imposed on the left side of the channel due to the uniform airflow and symmetry in the fin arrays.

At the bottom: A constant heat flux (q_o) of 5903 W/m² was applied to the bottom wall of the heatsink as described in equation (3.7). However, the lower part of the air domain was thermally insulated (adiabatic) both before and after the heatsink.

$$-\mathbf{n} \cdot \mathbf{q} = q_o \quad (3.7)$$

3.3.2 Mesh independence test

A mesh independence study was performed on the rectangular plate fins (RPF) heatsink with five fins, which was used as the reference case in this study, to ensure the reliability and accuracy of the numerical simulations. The purpose of this test was to verify that the computational domain was sufficiently resolved and that numerical uncertainties were minimized. For each of the heatsinks, a free tetrahedral element mesh with mesh refinement was created. The mesh was specifically designed to concentrate a substantial number of nodes on the heatsink and the immediate air domain that comes into direct contact with the heatsink, to accurately capture the phenomena of conjugate heat transfer.

An investigation was conducted to examine the impact of mesh size on the pressure drop across the heatsink and the temperature difference between the outlet and the inlet, as depicted in Figure 3.8. It can be observed that as the number of elements increases, the pressure drop across the heatsink decreases, while the temperature difference between the outlet and the inlet increases. It was noted that once the number of elements surpassed 6 million, both the pressure drop and the temperature difference remained relatively constant. The percentage change in the pressure drop and temperature difference recorded when the mesh element increased from 6 million to 16 million was 4% and 1.1%, respectively. This suggests that additional mesh refinement had minimal impact on the results. Hence, in this study, it was determined that a mesh consisting of more than 6 million elements is adequate to guarantee precise and dependable simulation outcomes for all the heatsinks investigated.

3.3.3 Model validation

To verify the accuracy of the numerical model, the simulation results were compared to existing experimental and numerical data found in the literature. We conducted a direct

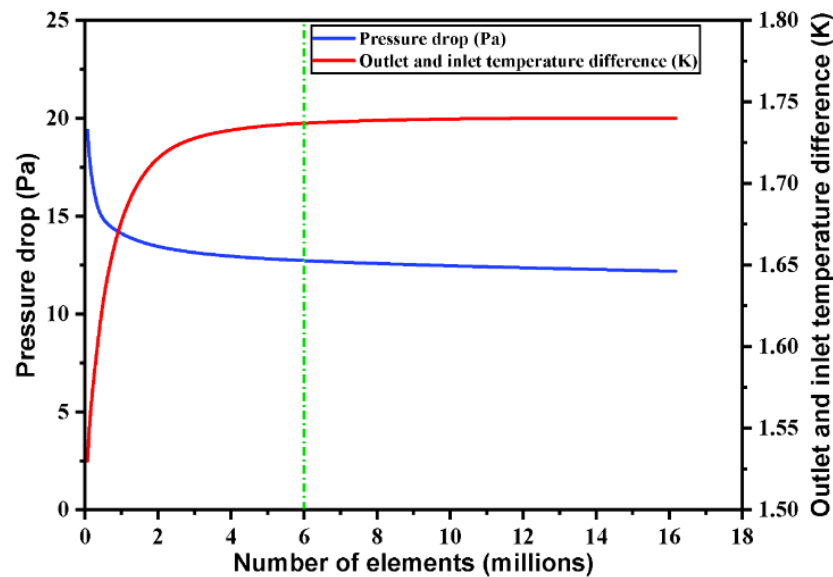


Figure 3.8: Mesh independence test

comparison between the present study and Shaeri's numerical study [177], as well as Johnson's experimental study [178]. The focus of comparison was a heatsink with a rectangular plate and nine fins. The thermal resistance was measured for different Reynolds numbers and the results were compared among the present study, Shaeri's numerical study, and Johnson's experimental study, as depicted in Figure 3.9. We noted a maximum discrepancy of 8% between the present study and Johnson's empirical findings, as well as a maximum deviation of 10% between the current model and Shaeri's computational results. The comparisons demonstrate a strong agreement between the present numerical model and both Shaeri's numerical investigation and Johnson's experimental study, with a maximum error of less than 10% when compared to the experimental findings. The agreement between the current numerical model and the experimental data instills confidence in the precision and dependability of the numerical simulations carried out in this study.

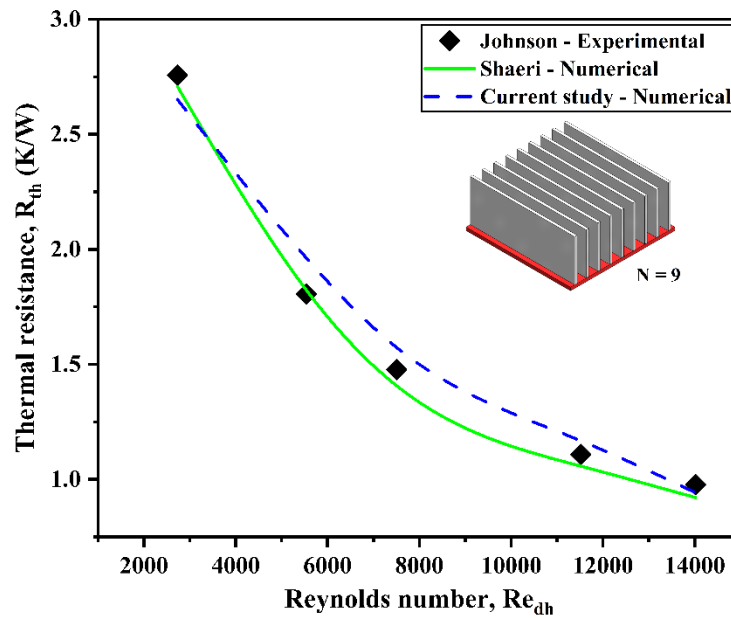


Figure 3.9: Model validation

3.3.4 Governing equation

The governing equations include the conservation principles for energy, mass, and momentum in the air domain, as well as the heat conduction equation for the solid domain [179]. These equations are expressed as follows:

The heat transfer governed by the energy equation is modeled as [180]:

$$\rho C_p \mathbf{u} \cdot \nabla T + \nabla \cdot \mathbf{q} = Q \quad (3.8)$$

$$\mathbf{q} = -k \nabla T \quad (3.9)$$

where C_p is the solid heat capacity at constant pressure, $\mathbf{u}$ is the velocity field defined by the moving mesh, Q (W/m^3) is the heat source, $\mathbf{q}$ is the heat flux vector, k ($W/m \cdot K$) is the thermal conductivity, and ∇T is the temperature gradient. The term $\nabla \cdot \mathbf{q}$ contributes to the heat conduction in solids, $\rho C_p \mathbf{u} \cdot \nabla T$ contributes to the heat transfer in the fluid domain.

The K- ϵ turbulence model is used to simulate the airflow in this study because of its broad applicability for simulating turbulent flows in engineering applications [181]. It is a two-equation model that solves the momentum, continuity and two additional transport equations: one for the turbulent kinetic energy (k) and one for the turbulent dissipation rate (ϵ).

The momentum equation (Navier-Stokes equation for turbulent flow) is given by:

$$\rho(\mathbf{u} \cdot \nabla) \mathbf{u} = \nabla \cdot [-p\mathbf{I} + \mathbf{K}] \quad (3.10)$$

where $\mathbf{K}$ is the stress tensor, defined as:

$$\mathbf{K} = (\mu + \mu_T)(\nabla + (\nabla \mathbf{u})^T) \quad (3.11)$$

The transport equation for turbulent kinetic energy (K) represents the balance between production, diffusion, and dissipation of turbulent energy:

$$\rho(\mathbf{u} \cdot \nabla)K = \nabla \cdot \left[\left(\mu + \frac{\mu_T}{\sigma_K} \right) \nabla K \right] + P_K + \rho \varepsilon \quad (3.12)$$

Here, P_K is the production term, which quantifies the energy transfer from the mean flow to turbulence.

The transport equation for the dissipation rate (ε) governs the rate at which turbulent energy is dissipated into heat at smaller scales:

$$\rho(\mathbf{u} \cdot \nabla)\varepsilon = \nabla \cdot \left[\left(\mu + \frac{\mu_T}{\sigma_\varepsilon} \right) \nabla \varepsilon \right] + C_{\varepsilon 1} \frac{\varepsilon}{K} P_K + C_{\varepsilon 2} \rho \frac{\varepsilon^2}{K} \quad (3.13)$$

The model constants known as closure coefficients are derived from the experiment and are given as; $C_{\varepsilon 1} = 1.44$, $C_{\varepsilon 2} = 1.92$, $\sigma_\varepsilon = 1.3$, $\sigma_K = 1.0$, and $C_\mu = 0.09$. The turbulent viscosity (μ_T) which links the turbulence intensity (K) and dissipation rate (ε) to the effective viscosity is modeled as:

$$\mu_T = \rho C_\mu \frac{K^2}{\varepsilon} \quad (3.14)$$

The production of turbulent kinetic energy (P_K) is calculated using the velocity gradients:

$$P_K = \mu_T [\nabla \mathbf{u} : (\nabla \mathbf{u} + (\nabla \mathbf{u})^T)] \quad (3.15)$$

Finally, the continuity equation ensures mass conservation in the fluid and is given by:

$$\rho \nabla \cdot \mathbf{u} = 0 \quad (3.16)$$

3.3.5 Performance metrics

Various performance metrics are utilized to evaluate the effectiveness of the heatsink. The metrics can be categorized into hydraulic metrics (Pressure drop), thermal metrics (temperature and thermal resistance), and overall system performance combining the effect of metrics.

1. **Pressure drop (ΔP):** The design of heatsinks is significantly influenced by pressure drop, which plays a crucial role in determining the efficiency of heat dissipation. Pressure drop is a phenomenon characterized by the reduction in fluid pressure as it passes a heatsink. It refers to the difference in pressure between the airflow at the inlet and outlet of a test section, specifically a heatsink. It is numerically denoted by [182], [183]:

$$\Delta P = P_{out} - P_{in} \quad (3.17)$$

where P_{out} and P_{in} are the pressure at the outlet and inlet sections in Pa.

2. **Base plate temperature (T_b):** The base plate temperature refers to the temperature of the heatsink's base, which is in direct contact with the heat source. By monitoring T_b , we can assess the effectiveness of the heatsink in dissipating heat and maintaining the temperature of the heat source within acceptable limits. In this study, the base plate temperature is nondimensionalized and denoted by:

$$T^* = \frac{T_b - T_{in}}{T_{in}} \quad (3.18)$$

where T_{in} and T_{out} are the inlet and outlet air temperature (K).

3. **Overall performance (η):** This reveals the overall efficiency of the heatsink with regards to its hydraulic and thermal characteristics. It is defined as the ratio of the Nusselt number (Nu) to the pressure drag coefficient (P_d) denoted by [182], [183]:

$$\eta = \frac{Nu}{P_d} \quad (3.19)$$

The pressure drag coefficient is a dimensionless parameter that characterizes the resistance to fluid flow through the heatsink. It is defined as the ratio of the pressure drop across the heatsink to the dynamic pressure of the fluid flow denoted by [178], [182], [183]:

$$P_d = \frac{\Delta P}{\frac{1}{2} \cdot \rho \cdot v_i^2} \quad (3.20)$$

where ρ : is the density of air in kg/m^3 .

4. **Porosity (ϕ):** The porosity of perforated fins refers to the proportion of void volume in relation to the overall volume of the solid fin material. The metric quantifies the degree of perforation or voids present in the fin material and it is denoted by:

$$\phi = \frac{V_{hole}}{V_{solid}} \quad (3.21)$$

3.4 Results and discussion

This section presents the outcomes derived from the numerical simulations of the different heatsink configurations discussed earlier.

3.4.1 Bifurcated longitudinal split fins (BLSF) heatsink

The Bifurcated Longitudinal Split Fins (BLSF) heatsink optimization examines how bifurcation length (L^*) affects thermal and hydraulic performance. Starting from the inlet,

the fin splits into two branches at the bifurcation length (L^*). This study examines L^* values from 0.1 to 0.9 to find the best configuration. Figure 3.10(a) shows the impact of the length of bifurcation on the temperature of the baseplate. For all bifurcation lengths, baseplate temperature decreases with velocity, indicating improved cooling efficiency with airflow. The temperature rises as L^* increases, with the most notable increase observed at L^* values of 0.8 and 0.9. The ideal bifurcation length is 0.1, with the lowest baseplate temperature measured at around 0.4 at 12 m/s. Positioning the bifurcation near the inlet maximizes heat transfer, improving thermal performance by increasing convective heat transfer surface area.

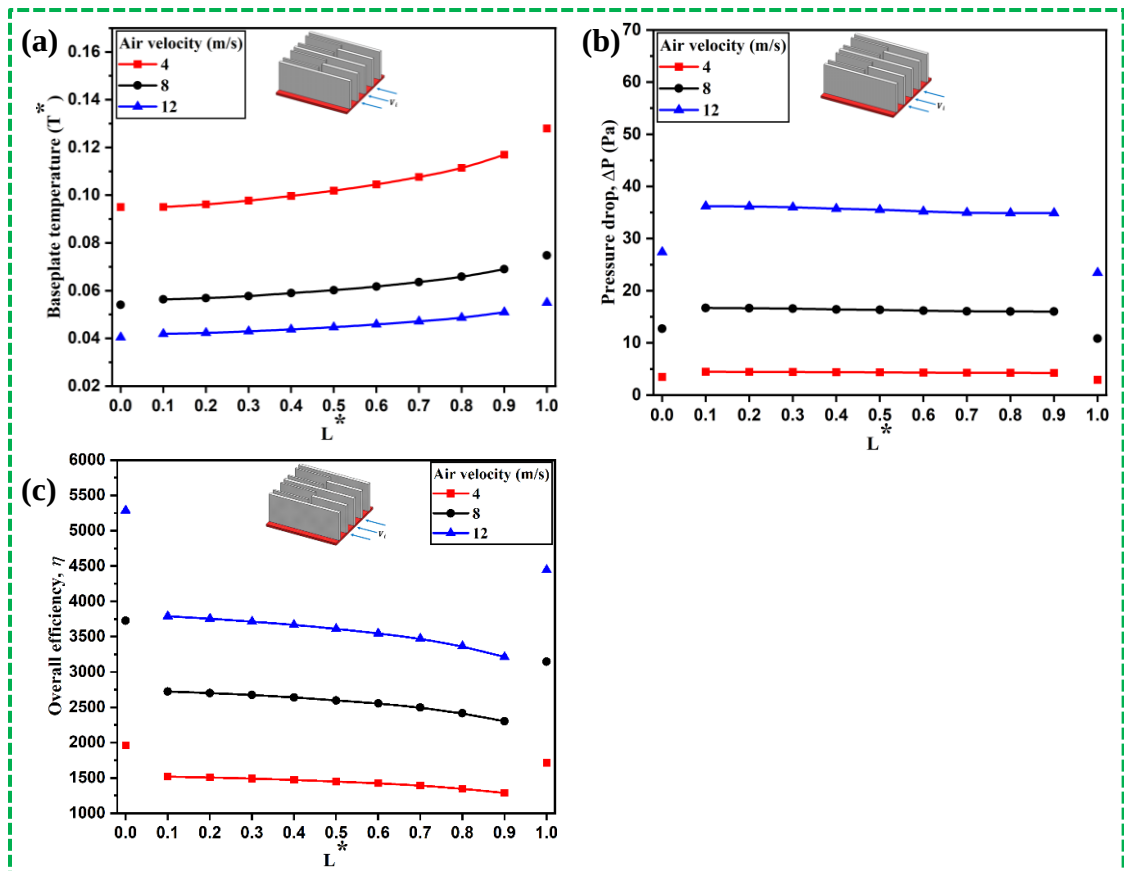


Figure 3.10: Effect of bifurcation length on the (a) baseplate temperature, (b) pressure drop, and (c) system performance of BLSF heatsink

Figure 3.10(b) shows how bifurcation length affects heatsink pressure drop. If L^* increases, pressure drop rises. Because the air has lower momentum at 4 m/s, the fin bifurcation causes minimal resistance, so the pressure drop remains constant as L^* increases. However, pressure drop increases significantly for L^* values of 0.6 and higher at 8 and 12 m/s. As bifurcation length increases, airflow momentum encounters more resistance, resulting in higher pressure losses. The minimum pressure drop is measured as 5 Pa at a flow rate of 4 m/s and a normalized distance of 0.1. Conversely, the maximum pressure drop is recorded as 50 Pa at a flow rate of 12 m/s and a normalized distance of 0.9. Figure 3.10(c) evaluates the overall efficiency of the BLSF heatsink based on the length of the bifurcation. The overall performance metric, encompassing both thermal and hydraulic performance, diminishes as L^* increases. The decline in performance is gradual when L^* is between 0.1 and 0.5, but it becomes more noticeable when L^* exceeds 0.5. The maximum overall performance is approximately 2900 when $L^* = 0.1$ and the speed is 12 m/s, while the minimum performance is around 950 when $L^* = 0.9$ and the speed is 4 m/s. This trend highlights the benefits of shorter bifurcation lengths, which improve the convective heat transfer by increasing the effective surface area while reducing the pressure drop. Based on baseplate temperature, thermal resistance, and performance, the BLSF heatsink's ideal bifurcation length is 0.1. This setup optimizes thermal management and pressure drop. BLSF heatsinks with L^* values of 0.1 will be used in future studies to analyze the effects of morphing fin numbers, as discussed in the following subsection.

3.4.2 Variable height gradient fins (VHGF) heatsink

The VHGF heatsink has been tailored to improve the spread of heat by adjusting the height of the fins along the length of the heatsink. Figure 3.11 illustrates the effect of fin height

gradient (H^*) on the baseplate temperature. As the velocity of the airflow increases, the baseplate temperature decreases due to enhanced convective heat transfer. The temperature increases as the fin height increases along the fin length (H^* from 0.0 to 0.5), indicating that this configuration reduces heat transfer effectiveness, possibly due to reduced heat transfer area when the air is not heated. However, as fin height decreases along fin length (H^* from 0.5 to 1.0), temperature decreases, indicating improved thermal performance. The lowest temperatures are at $H^* = 0.0$ and $H^* = 1.0$, while the highest temperature is at $H^* = 0.5$ (equivalent to the RPF configuration), showing that VHGF configurations improve cooling compared to the base RPF configuration. The lowest baseplate temperature was 0.05 at $H^* = 1.0$ and 12 m/s.

Figure 3.12(a) shows how fin height gradient (H^*) affects heatsink pressure drop. The pressure drop decreases as the fin slope increases (H^* from 0.0 to 0.5), suggesting that taller fins at the inlet initially lower resistance but later increase resistance as airflow encounters more obstruction. From $H^* = 0.5$ to 1.0, where fin height decreases along fin

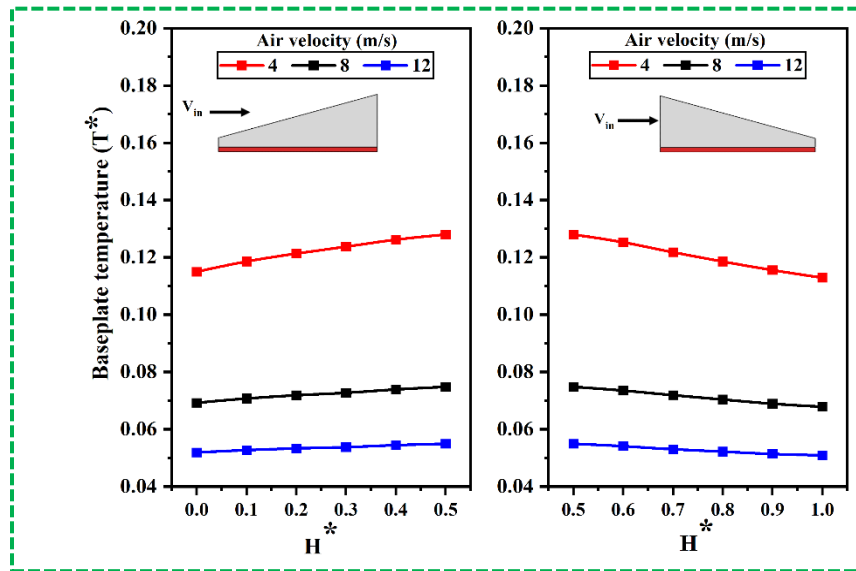


Figure 3.11: Effect of fin height gradient on the baseplate temperature

length, the pressure drop increases, indicating that the outlet configuration with taller fins has higher resistance due to the increased surface area. Despite this, $H^* = 1.0$ has a lower pressure drop than $H^* = 0.0$, showing that a decreasing fin height gradient reduces pressure drop.

Figure 3.12(b) shows how fin height gradient (H^*) affects heatsink performance. Performance increases as H^* increases from 0.0 to 0.7. Performance gradually decreases

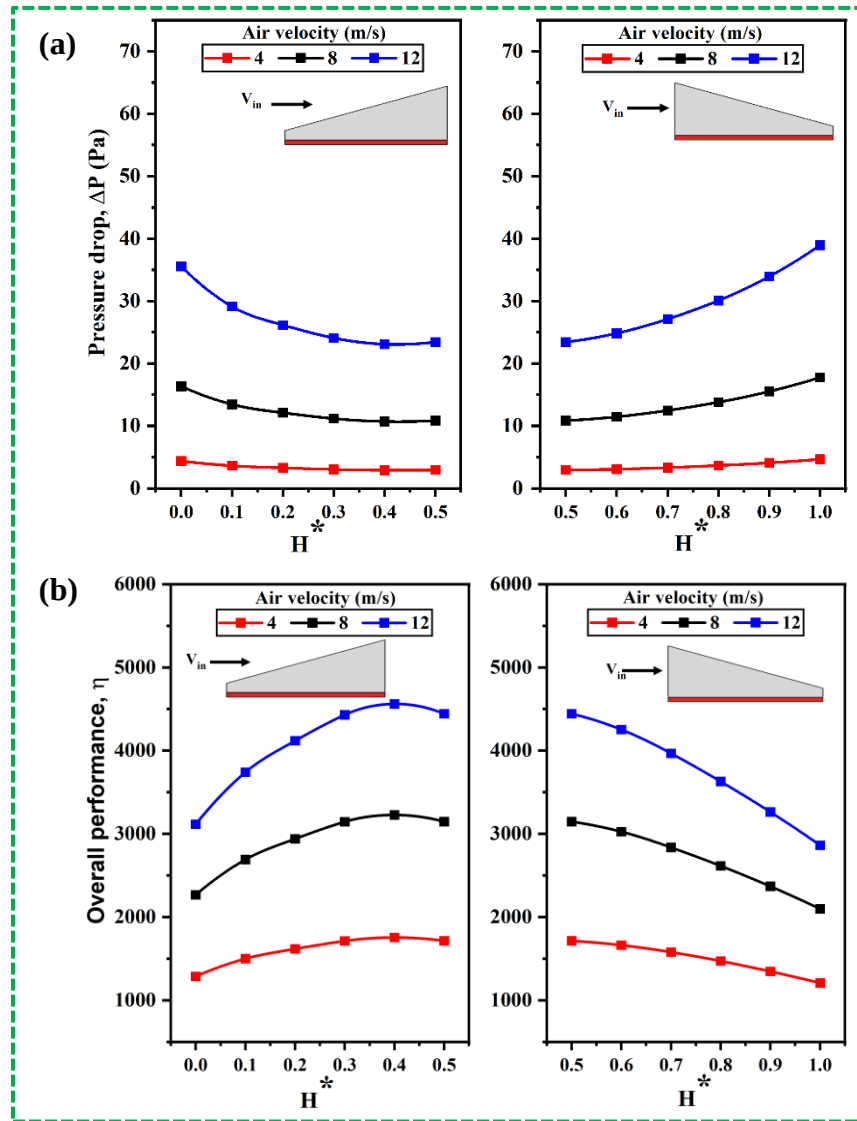


Figure 3.12: Effect of fin height gradient on the (a) pressure drop, and (b) system performance of VHGF heatsink

from $H^* = 0.8$ to 1.0, suggesting that while thermal performance is high, pressure drop at these gradients reduces heatsink effectiveness. Configurations with H^* between 0.0 and 0.4 perform worse than the RPF (base case) heatsink, indicating that a lower inlet fin height is less effective. The optimum overall performance is achieved at $H^* = 0.8$, balancing thermal and hydraulic considerations effectively. Nonetheless, for subsequent studies, a configuration with $H^* = 1.0$ is used, as it provides the best thermal performance, despite a slight compromise in overall performance due to higher pressure drop, emphasizing the prioritization of thermal effectiveness in heatsink design.

3.4.3 Hybrid plate-pin fins (HPPF) heatsink

The HPPF heatsink configuration combines the advantages of both rectangular plate and pin fins, aiming to enhance thermal performance through increased surface area. In Figure 3.13(a), the effect of pin volume on the baseplate temperature is depicted. The graph reveals that for all velocities, an increase in pin volume correlates with a decrease in baseplate temperature. This trend is consistent across varying velocities, with the temperature dropping more significantly as velocity increases. The minimum baseplate temperature recorded was 0.031 at 50% pin volume and a velocity of 12 m/s, while the maximum temperature was 0.107 at 10% pin volume and a velocity of 4 m/s. This indicates that a higher volume of pins enhances heat dissipation due to increased surface area, and reduced plate fin thickness.

In Figure 3.13(b), the impact of pin volume on pressure drop is illustrated. As the pin volume increases from 10% to 50%, the pressure drop also increases. This effect is more pronounced at higher velocities (8 and 12 m/s) compared to 4 m/s. At lower pin volumes (10% to 20%), the increase in velocity has a lesser impact on the pressure drop. This is

because higher pin volumes create greater obstructions to the airflow, leading to increased pressure losses. The maximum pressure drop recorded was 209 Pa at 50% pin volume and 12 m/s, while the minimum pressure drop was 7 Pa at 10% pin volume and 4 m/s. The increased pressure drop at higher pin volumes can be explained by the principles of fluid mechanics, where greater obstruction leads to higher flow resistance. Figure 3.13(c) illustrates the effect of pin volume on the overall performance of the system. Contrary to the trends observed for temperature and thermal resistance, the overall performance decreases as the pin volume increases for all velocities. This decrease is attributed to the significant pressure drop caused by the pin fins, as highlighted in Figure 3.13(b).

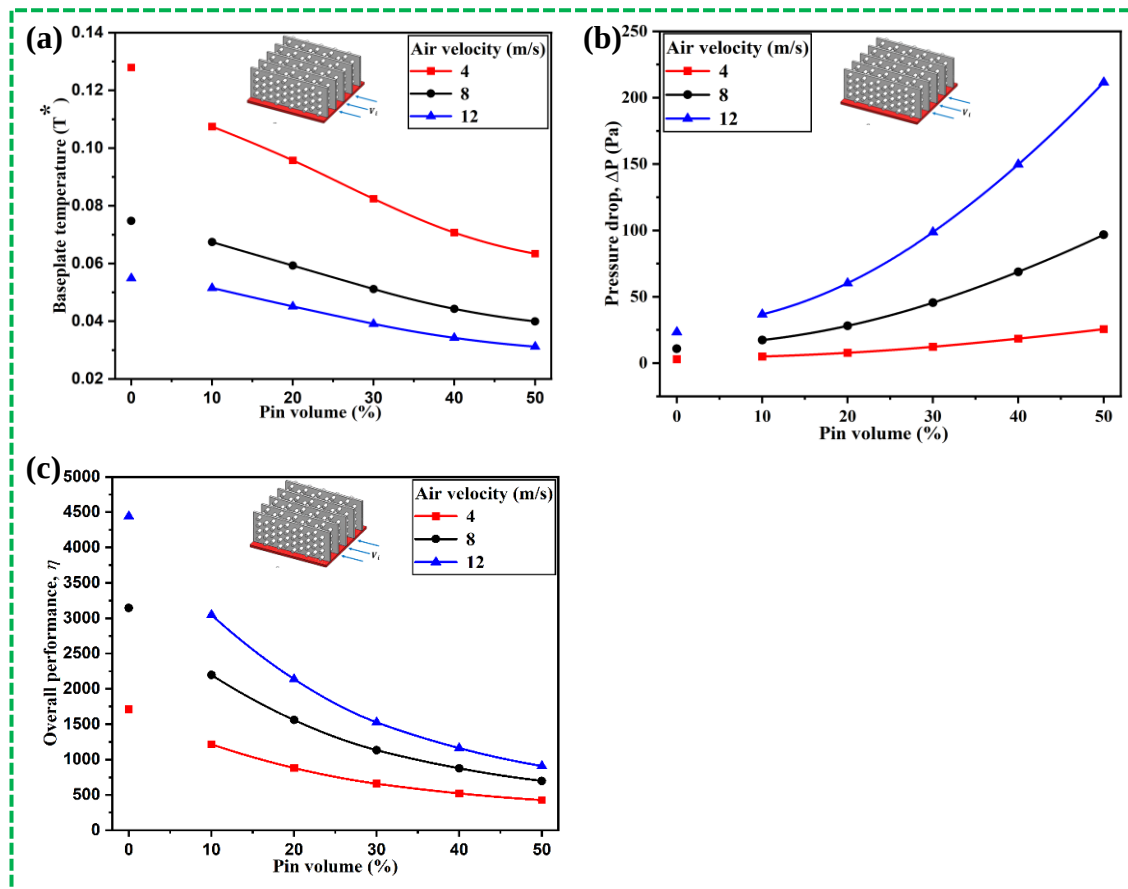


Figure 3.13: Effect of pin volume on the (a) baseplate temperature, (b) pressure drop, and (c) system performance of HPPF heatsink

The maximum system performance of about 2100 was recorded at 10% pin volume and 12 m/s, while the minimum system performance of 430 was recorded at 50% pin volume and 4 m/s. This suggests that while higher pin volumes improve heat dissipation, the resulting increase in pressure drop negatively impacts the overall system performance. In summary, for the HPPF heatsink configuration, an optimal balance must be achieved between heat dissipation and pressure drop to maximize overall performance. Based on these findings, the most effective configuration involves a lower pin volume to minimize pressure drop while still benefiting from the enhanced thermal performance of the hybrid design.

The impact of varying the number of pins on the thermal and hydraulic performance of the HPPF heatsink is systematically analyzed in Figure 3.14. Figure 3.14(a) illustrates the effect of increasing the number of pins from 0 to 50 on the baseplate temperature. As the number of pins increases, there is a significant reduction in the baseplate temperature, highlighting the enhanced heat dissipation capabilities introduced by the pin fins. Notably, the introduction of even a small number of pins (from 0 to 10) leads to a sharp decline in the baseplate temperature, suggesting that the pins substantially augment the heat transfer area and turbulence, thereby improving the cooling performance. However, as the number of pins continues to increase from 10 to 50, the rate of temperature reduction becomes less pronounced. This indicates that while the initial addition of pins dramatically enhances thermal performance, further increases yield diminishing returns. Additionally, the influence of air velocity on temperature reduction diminishes at higher velocities, likely due to the increased convective heat transfer already being close to optimal.

Figure 3.14(b) shows how pin number affects heatsink pressure drop. Results show that pressure drop increases significantly with pin density. More pins block airflow, increasing

fluid flow resistance. Pressure drop increases significantly at higher air velocities due to the pin fins' obstruction and increased flow rate. Pin fins are crucial to hydraulic performance because air velocity has little effect on pressure drop without pins. Finally, Figure 3.14(c) assesses the overall performance of the HPPF heatsink, considering both thermal and hydraulic aspects. The introduction of pins, while beneficial for thermal performance, results in a marked reduction in overall performance due to the increased pressure drop. As the number of pins rises, the overall performance gradually decreases, reflecting the trade-off between improved heat dissipation and the associated hydraulic penalties.

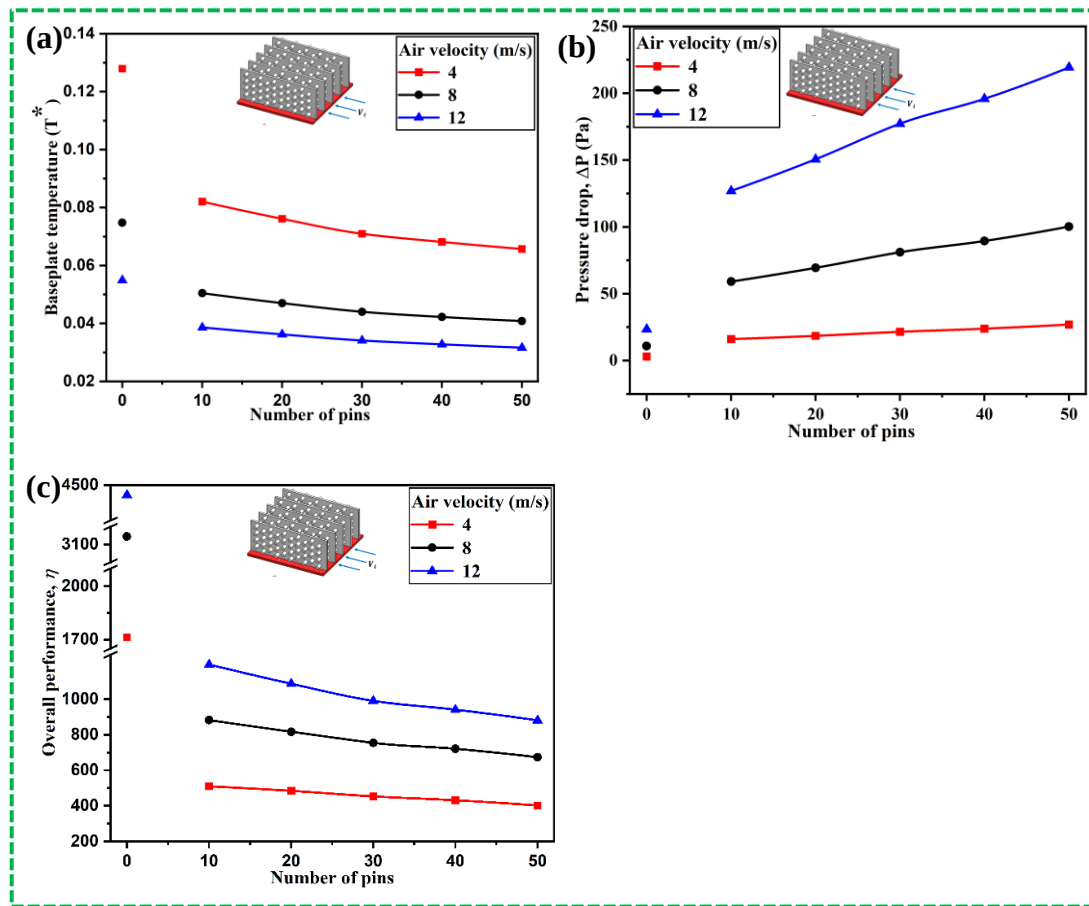


Figure 3.14: Effect of number of pins on the (a) baseplate temperature, (b) pressure drop, and (c) system performance of HPPF heatsink

In conclusion, it is established from Figure 3.14 that the heatsink becomes more thermally efficient as the number of pins increases. However, this improvement in thermal efficiency comes at the cost of decreased hydraulic performance. In this study, we have selected 40 pins as the most efficient configuration, with a greater emphasis placed on optimizing thermal performance.

3.4.4 Perforated plate fin (PPF) heatsink

Figure 3.15(a) illustrates the effect of perforation porosity on the baseplate temperature for various perforation shapes. As porosity increases from 10% to 60%, the baseplate temperature rises for all perforation shapes. Horizontal slots (P4) are the least thermally effective, showing higher baseplate temperatures across the range of porosities. Vertical slots (P3) are highly effective at a lower porosity (10%), recording the minimum baseplate temperature of 0.07. However, above 10% porosity, the thermal performance of P3 reduces significantly, leading to the highest temperatures at 60% porosity. This decline can be attributed to the reduced fin material between vertical slots, which affects heat conduction from the lower fin sections in contact with the baseplate to the upper sections. Circular (P1) and square (P2) perforations maintain the lowest baseplate temperatures at porosities beyond 10%, with negligible differences in thermal performance between them. The minimum baseplate temperature observed was 0.072 at 20% porosity for P1, indicating that perforations can enhance thermal performance up to a certain threshold.

Figure 3.15(b) examines the effect of porosity on pressure drop. As porosity increases, the pressure drop also rises for all perforation shapes. Horizontal slots (P4) consistently exhibit the lowest pressure drop, except at 10% porosity where circular perforations (P1) record the minimum pressure drop. This lower pressure drop is due to its minimal flow

disturbance. Conversely, P2 and P3 experience higher pressure drops, especially at lower (10-20%) and higher (50-60%) porosities. All perforated heatsinks show higher pressure drops compared to the unperforated heatsink, indicating increased flow resistance due to the perforations. Figure 3.15(c) shows the overall performance as a function of perforation porosity. All perforation shapes demonstrate lower overall performance compared to the unperforated heatsink, except for P1 at 10% porosity. Overall performance decreases with increasing porosity, with circular perforations (P1) consistently outperforming other shapes, followed by square (P2), vertical slots (P3), and horizontal slots (P4). The highest overall performance recorded was for P1 at 10% porosity, suggesting that limited perforation can enhance performance by balancing thermal and hydraulic effects.

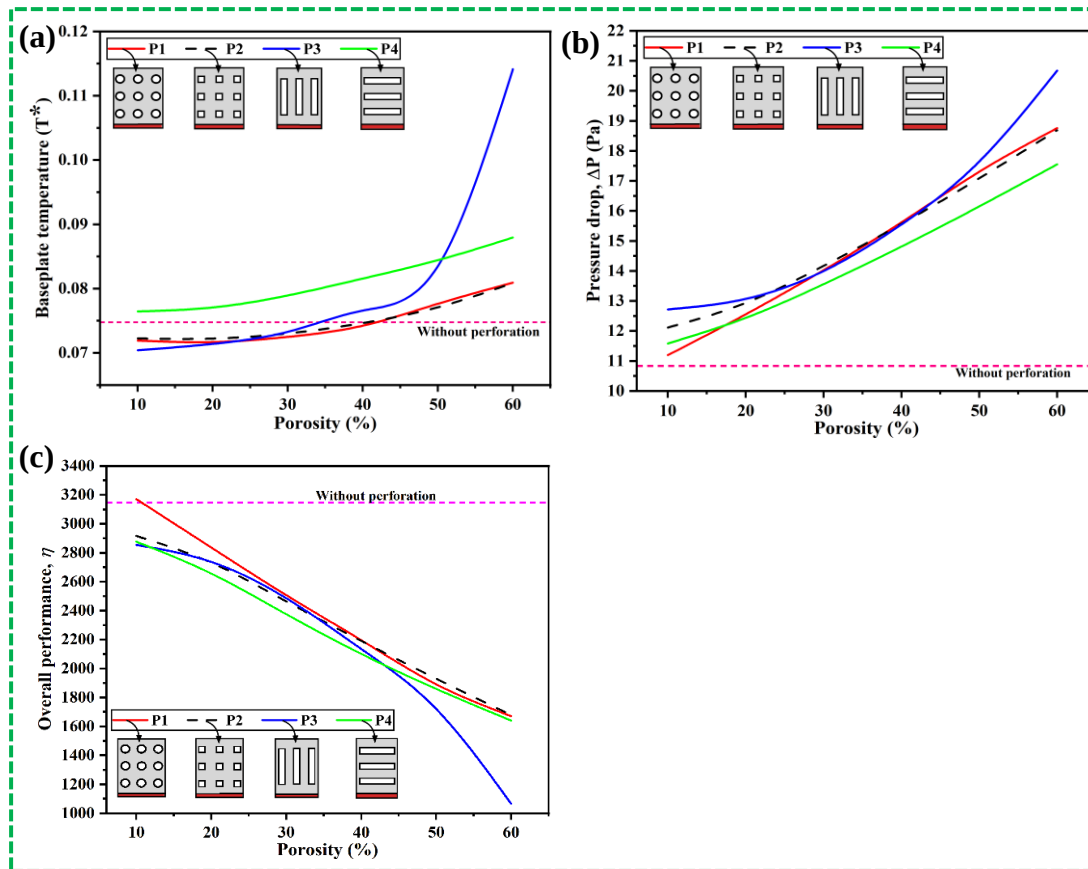


Figure 3.15: Effect of porosity on the (a) baseplate temperature, (b) pressure drop, and (c) system performance of PPF heatsink

In conclusion, the analysis indicates that lower porosities generally offer better thermal performance, though the specific shape of perforations plays a crucial role. Circular perforations at 20% porosity are identified as the most efficient configuration, balancing both thermal and hydraulic performance. This choice avoids the higher temperatures, and pressure drops associated with higher porosities while benefiting from reduced material weight.

3.4.5 Comparison between RPF, BLSF, VHGF, HPPF, and PPF heatsink

This section compares the performance of the five heatsink configurations based on the four-performance metrics as illustrated in Figure 3.16. The comparison is presented at the airflow velocity of 8 m/s. However, the trend remains the same at 4 m/s and 12 m/s. Each configuration's performance is analyzed to identify the optimal design for effective thermal and hydraulic management. The baseplate temperature result is shown in Figure 3.16(a) indicate that the HPPF configuration exhibits the lowest baseplate temperature, followed closely by BLSF and PPF. The enhanced performance of HPPF stems from its hybrid design, which combines plate and pin fins to maximize surface area and introduce turbulence for improved heat transfer. The BLSF configuration achieves a similar reduction in temperature due to its bifurcation mechanism, which promotes thermal boundary layer redevelopment. On the other hand, the RPF and VHGF configurations show higher baseplate temperatures. The uniform geometry of RPF limits its heat transfer efficiency, while VHGF's variable fin height optimizes cooling at the inlet but diminishes performance at the outlet due to reduced fin surface area. Thus, the hybrid and bifurcated designs outperform the conventional plate fins in managing heat dissipation effectively.

The pressure drop analysis reveals significant differences between configurations as shown in Figure 3.16(b). PPF exhibits the lowest pressure drop, attributed to its reduced material volume and streamlined flow paths through perforations. RPF also demonstrates relatively low-pressure drop due to its uniform and unobstructed design. In contrast, HPPF and BLSF exhibit higher pressure drops, driven by increased flow obstructions caused by pin fins and bifurcated fins, respectively. VHGF shows intermediate behavior, as its variable height gradient balances obstruction and flow continuity. The results highlight the trade-offs between enhancing heat transfer and maintaining low hydraulic resistance, with PPF emerging as the most hydraulically efficient design.

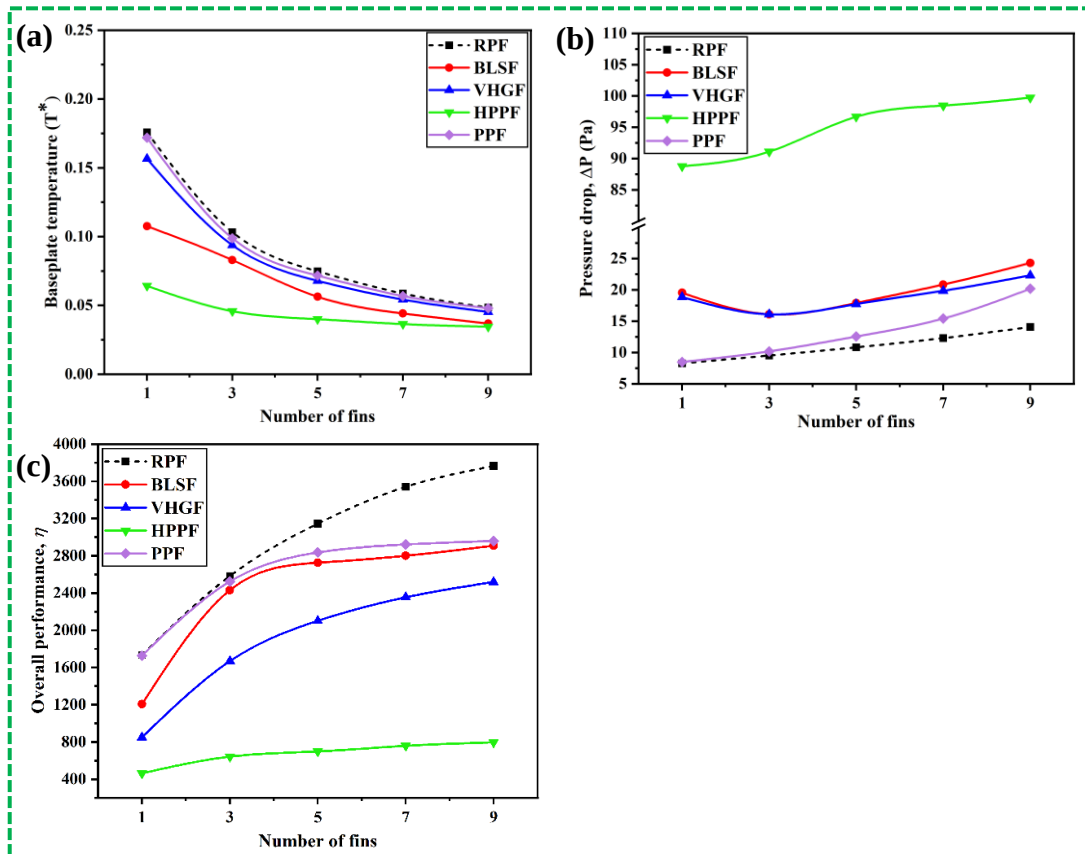


Figure 3.16: Comparison between RPF, BLSF, VHGF, HPPF, and PPF heatsink at 8m/s

The overall performance metric, as shown in Figure 3.16(c), reveals that the RPF configuration achieves the highest overall performance due to its lowest pressure drop, despite its moderate thermal performance. This is followed by the PPF configuration, which benefits from low pressure drop achieved through perforations, providing a balance between hydraulic efficiency and thermal effectiveness. The BLSF configuration ranks third, offering a strong compromise between thermal dissipation and hydraulic resistance, thanks to its bifurcation design that enhances boundary layer redevelopment while maintaining reasonable flow resistance. The VHGF configuration ranks fourth, as its variable-height fins improve local thermal performance but introduce inconsistent hydraulic behavior, limiting its efficiency. Lastly, the HPPF configuration, despite its superior thermal performance, exhibits the lowest overall performance due to the significant pressure drop caused by its complex hybrid structure. These results suggest that RPF and PPF are most suitable for applications requiring low pumping power, while BLSF is ideal for scenarios demanding high heat dissipation with moderate hydraulic constraints. The HPPF configuration, on the other hand, is recommended for systems where maximizing heat transfer is critical, and pumping power is less of a concern.

CHAPTER 4

INTEGRATED FINNED HEATSINK AND WATER COOLING FOR PV THERMAL MANAGEMENT

4.1 Introduction

Efficient thermal management of PV is critical for maintaining high electrical performance and prolonging system lifespan, particularly under high solar irradiance. Integrating air-cooled finned heatsinks with water pipe cooling offers a promising approach to maximize the benefits of both air and water cooling, while simultaneously addressing challenges such as non-uniform temperature distribution and excessive heat accumulation. This chapter investigates the performance of a hybrid cooling system combining finned heatsinks and water cooling techniques, focusing on its ability to enhance PV efficiency, and reduce operating temperatures.

Numerous studies have investigated the cooling processes of PV modules using fins, including the work of Selimefendigil et al. [23], who focused on utilizing fins based on natural convection. Hernandez-Perez et al. [24] examined passive techniques for cooling PV panels using fins with a segmented profile. Similarly, Elbreki et al. [25] used truncated fins to passively regulate the temperature of the PV module. Numerous experiments have investigated the impact of fixed fins on the rear of PV modules. Grubisic-Cabo et al. [33] attached aluminum fins to the rear of the PV module and evaluated its performance under natural conditions. Amber et al. [35] investigated the efficiency of PV modules using two fin-based cooling techniques. The research employed rectangular and threaded pin fins on

the rear of the module. Krstic et al. [41] experimentally incorporated small heatsinks of different geometry on PV panels. Gomaa et al. [42] experimentally investigated the effect of U-shaped fins on the efficiency of grid-tied PV modules under passive cooling. These studies demonstrate that thermal management using fins effectively reduces the operating temperature of PV modules.

On the other hand, liquid heat exchangers are prevalent in the active thermal management of PV modules. In most studies, the heat exchanger is configured with coolant circulating through a pipe fixed beneath the PV module. Maadi et al. [184] integrated ten cooling tubes beneath the PV module, determining that this arrangement yielded optimal outcomes. Zondag et al. [185] conducted a numerical study to evaluate the efficiency of various types of PV/T systems. Siddiqui et al [186] investigated the impact of cooling a concentrated PV with a single serpentine pipe. The solar concentration and cooling channels resulted in significant non-uniformity in irradiance and surface temperature, respectively. Almerbati et al. [187] employed a single cooling pipe configuration that morphs within a square heated plate to devise a cooling system for uniformly heated flat surfaces numerically. Siddiqui et al. [188] developed an innovative water-cooled heat exchanger with channels under the PV to enhance the uniformity of PV surface temperatures. These studies indicate that cooling PV modules using fins and water pipes is feasible, resulting in a substantial decrease in temperature and an enhancement in electrical efficiency. However, the use of water cooling in pipes entails a significant drawback. This leads to temperature non-uniformity in the PV module as shown by Almerbati et al. [187] and Siddiqui et al. [186], hence increasing ohmic losses and diminishing overall power output. The literature review reveals that no existing cooling system integrates a water pipe to lower the overall

operating temperature and fins to alleviate the temperature non-uniformity induced by the cooling pipes. The primary objective of this chapter is to conduct a numerical analysis of a single PV system featuring a water-cooling layer underneath, followed by a layer of fins.

4.2 Model description

In this design, as water flows through the cooling channels positioned directly beneath the PV cells, heat is transferred first to the water-cooling layer and then dissipated through the fins positioned under the water-cooling layer. This layered approach facilitates efficient heat extraction from the PV panel, helping to mitigate temperature build-up, especially under concentrated solar radiation. The combined cooling system not only reduces the average PV cell temperature but also minimizes temperature non-uniformity across the panel, promoting stable PV operation. To optimize the cooling efficiency within these constraints, this study evaluates two primary design parameters: Cooling pipe design, focusing on various pipe arrangements and flow velocities, and fin type and distribution, exploring optimized configurations to facilitate effective heat transfer.

4.2.1 Cooling Pipe Design

The primary objective of the cooling pipe design is to enhance heat transfer while maintaining an equal pipe length of 5.1 m and mass flow rate through the pipe across all configurations. Five pipe configurations were considered, beginning with a conventional serpentine (Configuration 1) and four optimized versions developed for increased cooling effectiveness and temperature uniformity. These optimizations were tailored to non-concentrated and concentrated PV. The following pipe configurations are illustrated in Figure 4.1(a-e).

1. **Configuration 1 (Base Case):** The conventional serpentine design, frequently referenced in the literature (e.g., AlMerbati et al. [187] and Siddiqui et al. [186]), includes 8 bends, and routes the inlet and outlet on opposite sides of the PV module, as shown in Figure 4.1(a).
2. **Configuration 2:** It is optimized for non-concentrated PV and adopts a double-lane serpentine configuration. Here, the serpentine structure spans only half the module's width and then reverses direction to cover the other half. The inlet and outlet are positioned on the same end of the PV panel, achieving a total of 16 bends, as depicted in Figure 4.1(b).
3. **Configuration 3:** Designed for concentrated PV with V-troughs, it addresses the central heating gradient by concentrating the serpentine flow under the central two columns of PV cells, covering the high-irradiance region. Like configuration 1, the flow enters from one side and exits on the opposite, with 20 bends to maintain the 5.1 m length, as seen in Figure 4.1(c).
4. **Configuration 4:** Expanding upon Configuration 3, this design integrates a serpentine flow for the central cells, complemented by two backflow pipes along each outer column. The centered section retains 7 bends, and the side channels, designed for backflow, ensure that the inlet and both outlets are positioned on the same side of the PV panel, as illustrated in Figure 4.1(d).
5. **Configuration 5:** A refinement of Configuration 4 which transforms the side backflow channels into serpentine structures to enhance heat extraction. The central serpentine flow consists of 10 bends, while each outer serpentine return pipe includes 5 bends with a larger radius, as shown in Figure 4.1(e). Each pipe configuration has been strategically

optimized to address distinct thermal management needs for the varying levels of irradiance in non-concentrated and concentrated PV configurations.

4.2.2 Fin Optimization

The cooling performance of the PV was further enhanced by integrating a finned heatsink beneath the water-cooling layer. The rectangular plate fin was selected due to its established effectiveness in PV cooling applications. To optimize the cooling performance while maintaining a constant fin material volume of 16,376 cm³, the fins were tailored

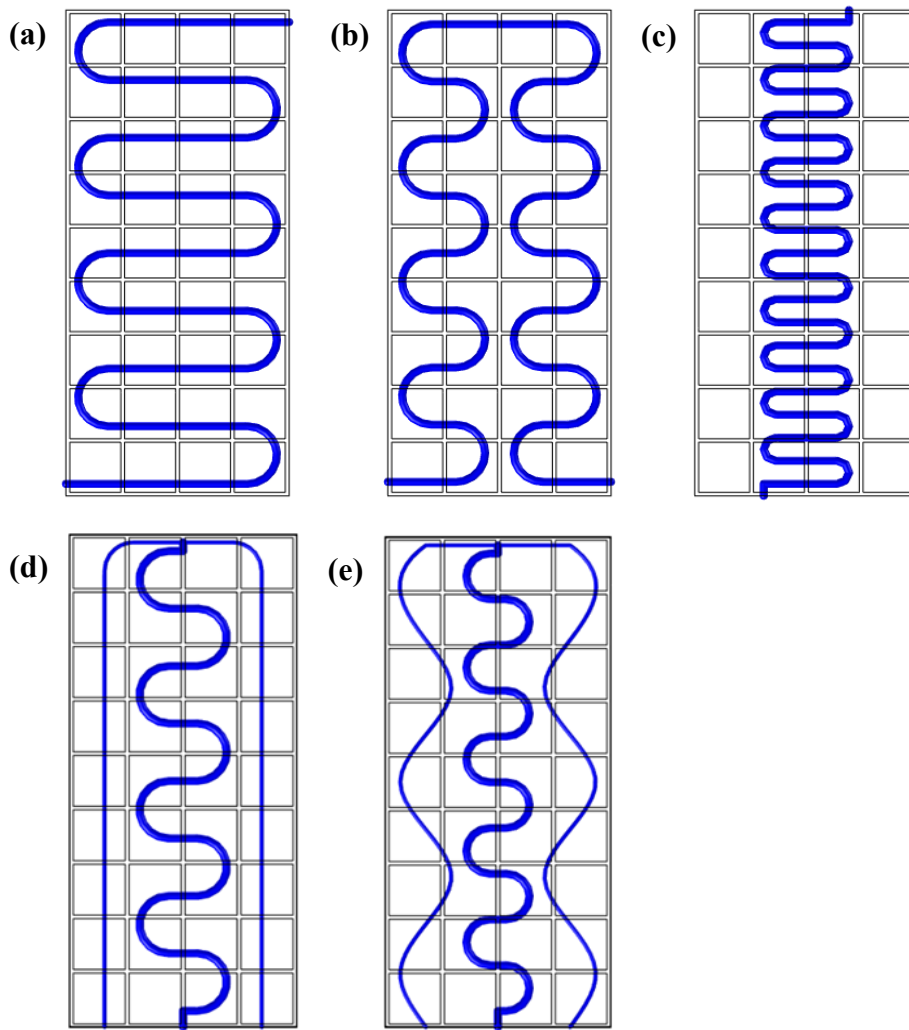


Figure 4.1: Different cooling pipe designs (a) Configuration 1, (b) Configuration 2, (c) Configuration 3, (d) Configuration 4, and (e) Configuration 5

based on localized temperature distributions observed in both concentrated and non-concentrated PV. The bifurcated longitudinal split fin design was adopted for optimization. The concept of fin bifurcation involved bifurcating selected fins and varying fin heights to target regions of higher thermal loads, thereby improving cooling and reducing temperature nonuniformity. Below is a detailed description of the base case and the optimized fin designs:

1. **Base Case Design:** A conventional rectangular plate-fin heatsink with nine uniformly spaced fins of 150 mm height and 10 mm thickness. The fins were attached to a baseplate that houses the water-cooling channel. Experimental studies, such as those by Elbreki et al. and Ekiz et al., provided suggestions on optimum fin dimensions, suggesting heights in the range of 100 – 200 mm for PV cooling. The dimensions of this base case align with experimental findings to ensure consistency in the scale of the fin heatsink. This design serves as the reference point for subsequent optimizations and is shown in Figure 4.2(a).
2. **Optimized Fin 1:** This design applies to non-concentrated PV without water cooling. The fins covering one-third of the PV length, where higher temperatures were observed, are bifurcated to improve cooling efficiency as illustrated in Figure 4.2(b).
3. **Optimized Fin 2:** For concentrated PV without water cooling where the central two-column cells receive higher solar irradiance. To address this, the fins directly beneath it are bifurcated along their full length, while the fins covering the remaining one-third length of the PV are also bifurcated to improve cooling on the hotter sides. This targeted design ensures effective heat extraction and is depicted in Figure 4.2(c).

4. **Optimized Fin 3:** Designed for non-concentrated PV integrated with the champion water-cooling channel (Configuration 2). It targets regions of peak temperature near the cooling channel outlet. Only the fins beneath the six cells in this region are bifurcated as shown in Figure 4.2(d).
5. **Optimized Fin 4:** This design is tailored for concentrated PV with the champion water-cooling channel (Configuration 5). The PV is divided into three sections along its length, each covering three rows of cells; In the first section (near the air inlet), only

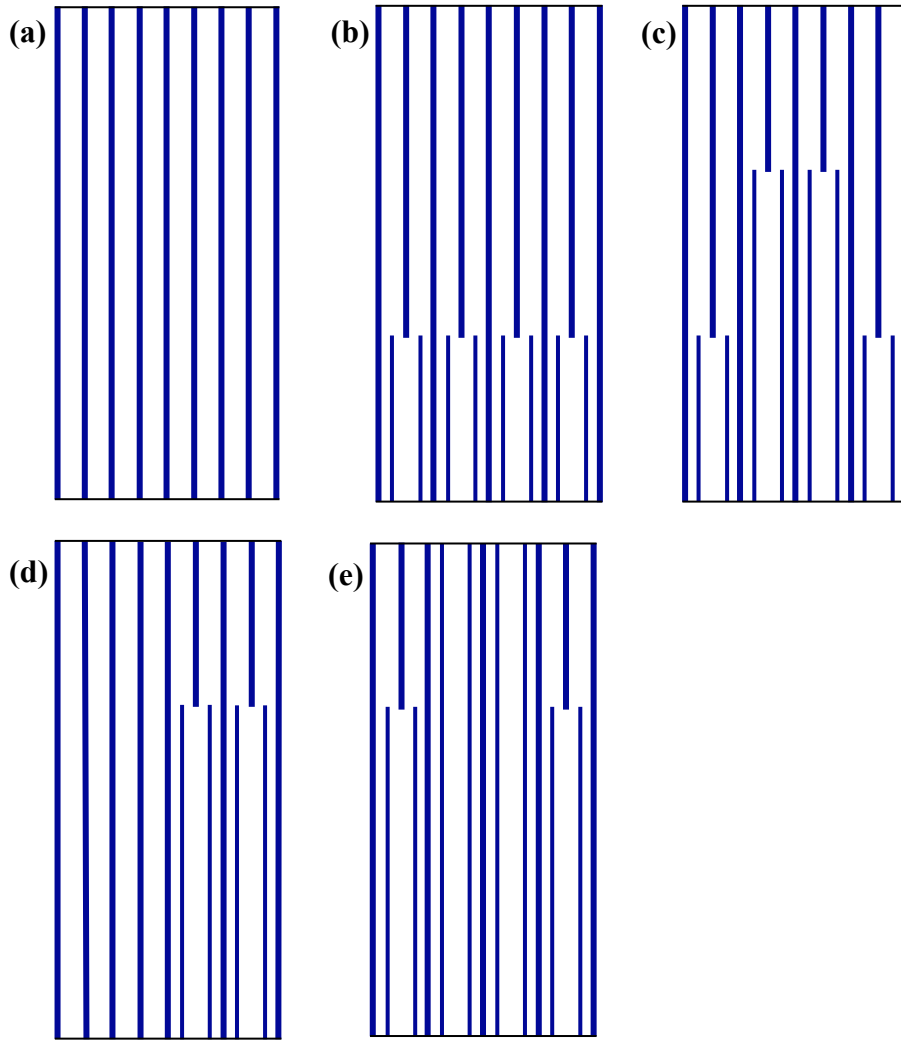


Figure 4.2: Different fin designs (a) Base case, (b) Optimized fin 1, (c) Optimized fin 2, (d) Optimized fin 3, and (e) Optimized fin 4

the middle fins beneath the concentrated cells are bifurcated. In the middle and the last sections, all fins are bifurcated to address higher heat loads. The fin height increases along the PV length, with the lowest height in the first section and the highest in the last. Across the PV width, the middle fin has the greatest height, which decreases symmetrically towards the edges as shown in Figure 4.2(e) and Figure 4.3.

Figure 4.3(a) shows the 3D assembly of the PV, water cooling pipe, and fins, while Figure 4.3(b) shows the 2D side view of the assembly. Table 4-2 provides the properties of the materials in the various layers. The encapsulant and the silicon cells have the same thickness. Heat transfer begins with solar energy absorption by the silicon cells after passing through the tempered glass cover and encapsulant. In standard PV installations, the

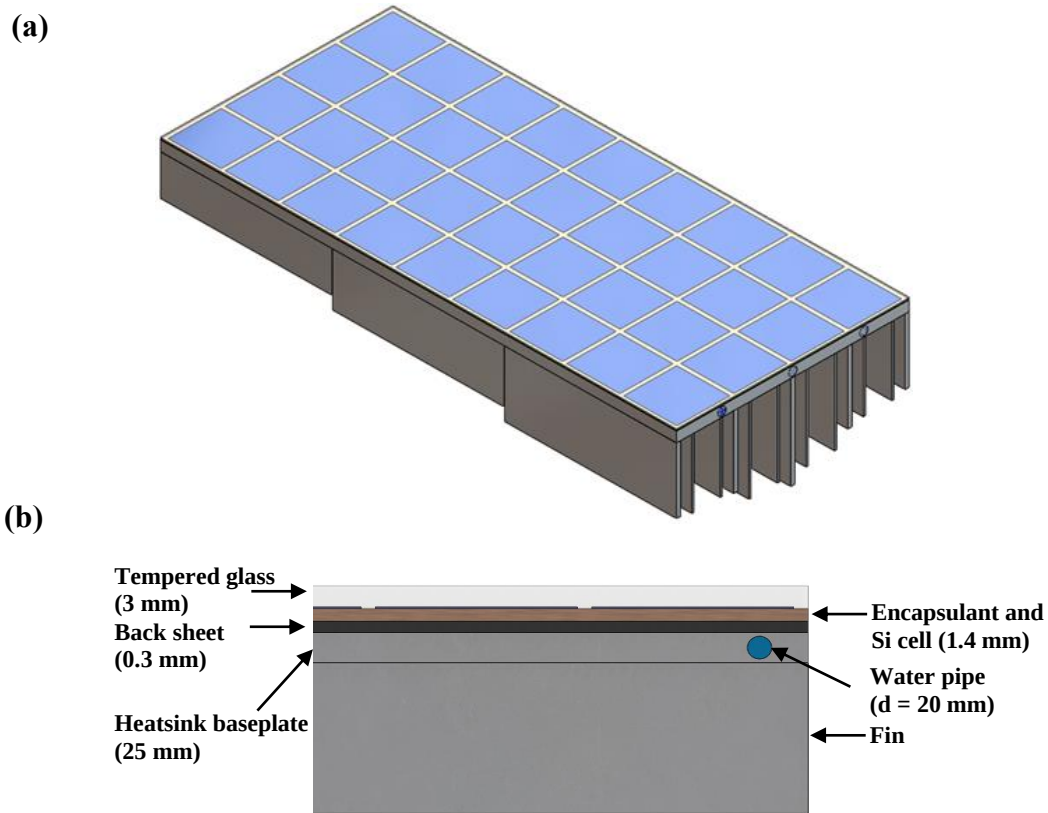


Figure 4.3: (a) 3D assembly of the PV module with water cooling channel and fins, and (b) 2D side view of the assembly showing key components (Not to scale)

back sheet supports the module and dissipates heat directly into the atmosphere. In the modified design, an aluminum baseplate is attached to the back sheet to enhance heat dissipation. This baseplate houses the water cooling channels, as shown in Figure 4.3(b), and provides a mounting surface for the fins.

4.2.3 V-Trough concentrator design

The V-trough concentrator is a non-imaging solar concentrator widely used in PV systems due to its simplicity in fabrication and ease of integration. It operates by concentrating sunlight onto the PV surface through reflection or refraction using flat reflectors or lenses [189]. The design features two flat reflectors slanted at an angle to form a "V" shape, with the PV module positioned between them, as illustrated in Figure 4.4(a). V-trough concentrators are particularly effective for achieving low concentration ratios, typically up to 3X under single reflection, making them suitable for moderate solar concentration applications [189], [190], [191]. The primary design parameters include the length-to-base ratio, reflector length, trough angle, and acceptance angle, which collectively determine the concentrator's performance [186], [192]. In this study, a 2X V-trough concentrator design was adopted based on the ray optics simulations performed by Siddiqui et al [186] for a PV module of comparable dimensions. The simulation optimized these design parameters to enhance light concentration while maintaining uniform distribution on the PV surface. Figure 4.4(b) illustrates the incident solar radiation profile received by the PV in the 2X configuration, highlighting the increased solar intensity on the PV surface achieved through reflection. This design balances effective solar concentration and manageable thermal loads, ensuring compatibility with the integrated cooling systems used in this study.

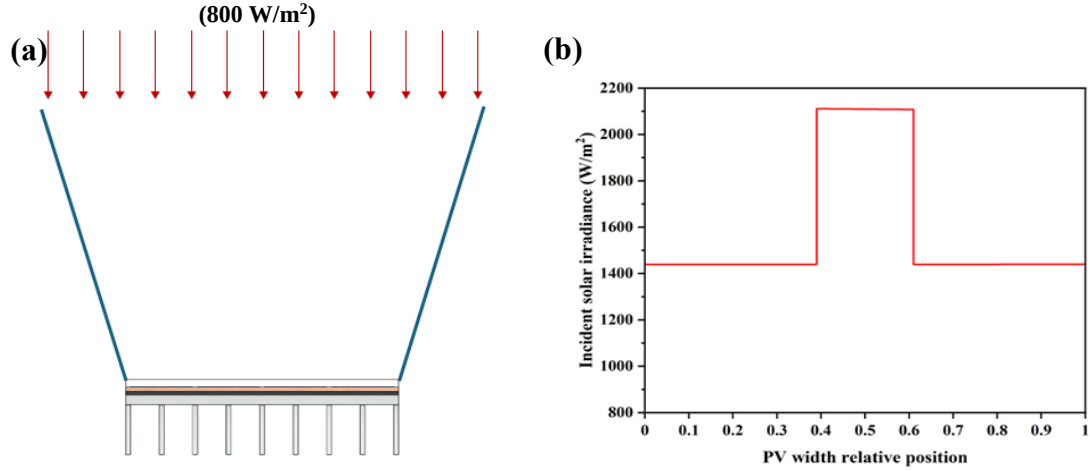


Figure 4.4: (a) Ray optics model for V-Trough concentrator, and (b) Received incident irradiance for 2X V-Trough concentrator [186]

4.3 Numerical method

The numerical method employed in this chapter is similar to the previous chapter which uses COMSOL Multiphysics to solve the governing equations for heat transfer and fluid dynamics. The electrical performance, however, was evaluated using analytical correlations.

4.3.1 Boundary conditions

These boundaries conditions are shown in Figure 4.5 and discussed below.

Air inlet: The velocity and temperature of the incoming air were set as 3 m/s 293 K, respectively. The inlet boundary condition is defined by the Danckwerts condition and the normal inlet velocity as represented by equations (3.3) and (3.4), respectively.

Sides and Bottom boundaries of the air domain: Adiabatic walls with no-slip conditions, ensuring thermal insulation as represented by equation (3.5).

Top boundary of the air domain: The top of the air domain was treated as an adiabatic wall with no-slip conditions represented by equation (3.5), except for the section occupied

by the PV module at the center. The rear side of the PV module aligns with the top boundary, exposing only the underside of the module and the fins (for fin-integrated configurations) to air cooling.

PV module: The PV module was thermally insulated on all four sides. The upper surface was subjected to a constant solar irradiance of 800 W/m^2 represented by equation (4.1). Convective cooling was applied to this surface using air at 3 m/s and 298 K .

$$-\mathbf{n} \cdot \mathbf{q} = q_o \quad (4.1)$$

Water inlet: Velocity ranging from 0.002 m/s to 0.5 m/s represented by equation (4.4) was specified. The flow regime was laminar for velocities between 0.002 m/s and 0.1 m/s and transitioned to turbulent flow for velocities between 0.15 m/s and 0.5 m/s . The inlet water temperature was maintained at 298 K .

$$u_{in} = u_o \quad (4.2)$$

Water outlet: The water exited the cooling channel as a heated outflow at a constant gauge pressure of $101,325 \text{ Pa}$.

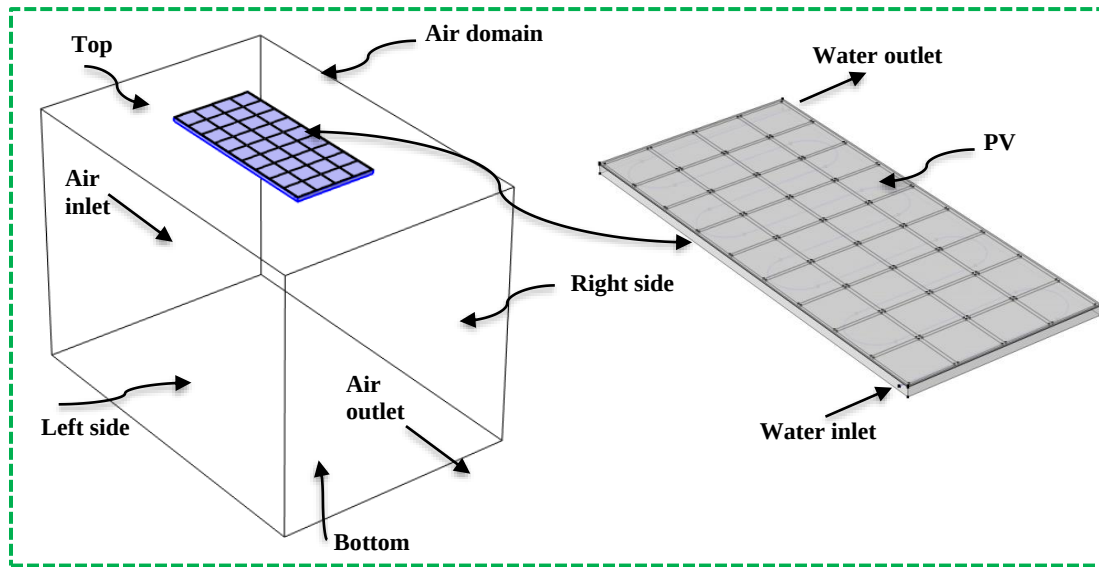


Figure 4.5: Boundary conditions

Table 4-1: Mesh Independent Test

Number of elements	Average PV temperature (K)	Difference (K)
a. PV only		
426,486	339.07	-
672,969	339.20	0.13
1,592,060	339.53	0.33
2,851,203	339.61	0.08
b. PV integrated with water cooling		
116,406	319.79	-
183,486	319.54	0.25
398,329	319.31	0.23
826,058	319.28	0.03
c. PV integrated with water cooling and fins		
201,655	309.53	-
382,793	308.91	0.62
2,214,493	308.86	0.05
3,590,180	308.81	0.05

4.3.2 Mesh independence test

The accuracy of the solution in the finite element method depends upon the size and distribution of the finite elements, or mesh. A finer mesh yields more precise results but necessitates a more memory-intensive simulation that requires additional time to resolve. The ideal mesh minimizes the discrepancy between the approximated and exact solutions while being computationally efficient. A mesh-independent test was conducted to identify the best mesh for the simulation. The influence of mesh size on the level of accuracy of the average PV temperature results is documented in Table 4-1. The solution for PV alone is independent of mesh size when the number of elements exceeds 2.8 million. PV systems coupled with water cooling and fins demonstrate mesh independence at grid element sizes exceeding 400,000 and 2.2 million, respectively. The highest inaccuracy observed at these meshes was below 0.1 K in average PV temperature, confirming that the study was

unaffected by mesh size. The growth rate of the mesh element for the optimum mesh is shown in Figure 4.6. The growth rate is almost above 0.6 except for a few spots, which shows the accuracy of the model with growth rate of 1 most desirable.

4.3.3 Model validation

To verify the accuracy of the numerical model, the simulation results were compared to existing numerical data by Siddiqui et al. [186]. The focus of the comparison was a PV module with water cooling in a serpentine pipe for both non-concentrated and concentrated irradiance. The concentrator is a 2X V-trough designed by Siddiqui et al. [186].

The operating conditions and materials properties for this validation are presented in Table 4-2. Figure 4.7(a) and Figure 4.7(b) shows the average PV cell temperature measured for different water inlet velocities and number of bends in the serpentine pipe for non-concentrated and concentrated PV, respectively. The comparisons demonstrate a reasonable agreement between the present numerical model and Siddiqui's, with a maximum error of less than 2%. Hence, it is concluded that the proposed numerical model is accurate enough to simulate the thermal performance of PVT integrated with fins.

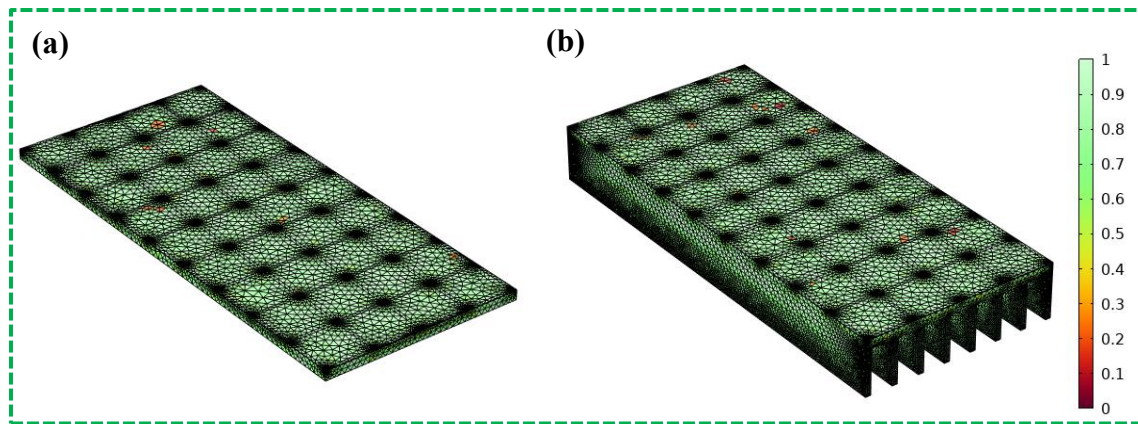


Figure 4.6: The growth rate for the mesh used in the present model for (a) PV/Water, and (b) PV/Water/Fins

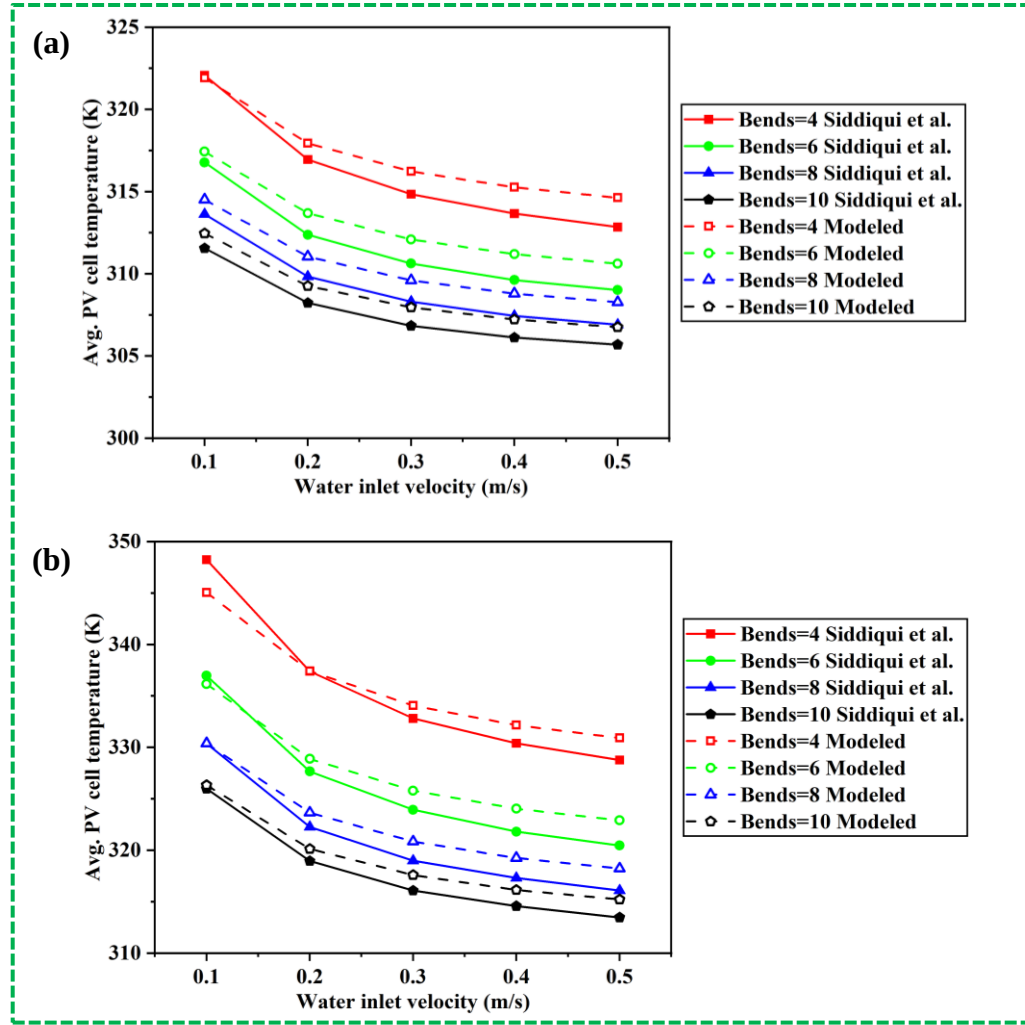


Figure 4.7: Validation of the thermal model for (a) Non-concentrated PV and (b) Concentrated PV at 800 W/m²

4.3.4 Governing equation

The governing equations for airflow and heat transfer in the air and PV domains are identical to those described in Chapter 3. These equations include the momentum equation, continuity equation, and the transport equations for turbulent kinetic energy (K) and dissipation rate (ϵ), which were solved using the K - ϵ turbulence model. For brevity, they are not repeated here. In addition to these, the governing equations specific to the water domain are solved to model the heat transfer and fluid flow within the cooling pipes. These are detailed as follows:

Table 4-2: Material properties, boundary conditions and PV characteristics used for the model

Layer	Properties	Value	Unit
Tempered Glass	Thickness	3	mm
	Density	2210	Kg/m ³
	Thermal conductivity	1.4	W/m.K
	Heat capacitance	730	J/kg.K
Encapsulant (EVA)	Thickness	1.4	mm
	Density	960	Kg/m ³
	Thermal conductivity	0.35	W/m.K
	Heat capacity	2090	J/kg.K
Silicon cells	Thickness	1.4	mm
	Density	2329	Kg/m ³
	Thermal conductivity	130	W/m.K
	Heat capacity	700	J/kg.K
Aluminium	Thickness (validation)	3	mm
	Thickness (Model)	25	mm
	Density	2700	Kg/m ³
	Thermal conductivity	238	W/m.K
	Heat capacity	900	J/kg.K
Thermal grease (validation)	Width	14	mm
	Density	2600	Kg/m ³
	Thermal conductivity	3	W/m.K
	Heat capacity	1200	J/kg.K
Water tube	Diameter	20	mm
Boundary conditions (Validation)	Glass top surface	Convective heat loss to the ambient at 298 K and 3 m/s air velocity.	
	Glass top surface	800 W/m ² solar irradiance as general inward heat flux	
	Water velocity	0.002 to 0.5 m/s ($40 \leq Re \leq 10,000$)	
	All other boundaries	Insulated	
PV (Canadian Solar CS5C-90M)	PV length	1.213	m
	PV width	0.547	m
	Cells area	0.125×0.125	m ²
	Number of cells	36 (9×4)	
	Max. power at STC	90	W
	Max voltage at STC	18	V
	Max. current at STC	4.99	A
	Power temperature coefficient	0.0045	1/K

The Nonisothermal Pipe Flow module is used to compute the temperature, velocity, and pressure fields in the water pipes. It approximates the pipe flow profile by 1D assumptions in lines. The energy equation (heat transfer) in this domain is solved by [180]:

$$\rho AC_p u_e \cdot \nabla_t T = \nabla_t \cdot (Ak \nabla_t T) + Q_{wall} \quad (4.3)$$

where: $\rho AC_p u_e \cdot \nabla_t T$: is the convection term, representing the transport of heat by water as it flows through the pipe. $\nabla_t \cdot (AK \nabla_t T)$: is the thermal conduction term, representing heat transfer due to temperature gradients in the water. Q_{wall} : Heat transfer through the pipe walls from the aluminium plate. ρ : density of water. A : cross-sectional area of pipe. C_p : specific heat capacity of water. u_e : average velocity of water in the pipe. $\nabla_t T$: change in the temperature along the pipe. k : is the thermal conductivity of water

To evaluate the wall heat transfer from the aluminum plate to the water inside the pipe, the following equations are employed. The wall heat transfer (Q_{wall}) is determined by:

$$Q_{wall} = (hZ)_{eff} \cdot (T_{ext} - T) \quad (4.4)$$

where $(hZ)_{eff}$ represents the effective heat transfer coefficient, T_{ext} is the external surface temperature of the wall (aluminum plate), and T is the bulk temperature of the cooling water inside the pipe. The effective heat transfer coefficient accounts for the combined thermal resistance due to convection and conduction across multiple layers, expressed as equation (4.5) for circular pipes:

$$(hZ)_{eff} = \frac{1}{\frac{1}{Z_o h_{int}} + \frac{1}{Z_N h_{ext}} + \sum \left(\frac{\ln \left(\frac{r_n}{r_{n-1}} \right)}{2\pi k_{wall,n}} \right)} \quad (4.5)$$

where: Z_o and Z_N are the respective dimensions or scaling factors at the internal and external surfaces, ensuring proper alignment of the heat transfer resistance terms. h_{int} and h_{ext} are the internal and external convective heat transfer coefficients, respectively. r_n and $r_n - 1$ denote the radial positions of consecutive layers in the wall structure. $k_{wall,n}$ is the thermal conductivity of the n-th layer of the wall material.

The internal and external convective heat transfer coefficients are evaluated by the equation:

$$h = Nu \frac{k}{d_h} \quad (4.6)$$

where the Nusselt number is the maximum between the laminar and turbulent Nusselt number depending on the flow condition inside the pipe. For laminar flow, based on the thermal film theory, the Nusselt number is analytically approximated as:

$$Nu_{lam} = 3.66 \quad (4.7)$$

While for turbulent flow, the Nusselt number is calculated as:

$$Nu_{turb} = \frac{\left(f_D/8\right)(Re - 1000)Pr}{1 + 12.7\sqrt{f_D/8}(Pr^{2/3} - 1)} \quad (4.8)$$

The momentum equation (fluid flow) in this domain governs the pressure and velocity of water along the pipe and is solved by:

$$0 = -\nabla_t P \cdot e_t - \frac{1}{2} f_D \frac{\rho}{d_h} |u|u \quad (4.9)$$

where: $\nabla_t P \cdot e_t$: represents the pressure gradient driving the water flow.

$\frac{1}{2} f_D \frac{\rho}{d_h} |u|u$: is the frictional resistance term accounting for the energy loss due to

friction between the water and the pipe walls. f_D : is the Darcy friction factor. d_h :

is the hydraulic diameter

The continuity equation (mass conservation) in this domain is solved by:

$$\nabla_t \cdot (A \rho u_e) = 0 \quad (4.10)$$

4.3.5 Performance metrics

To evaluate the thermal and electrical performance of the PV system under various cooling configurations, five performance metrics were defined and computed as follows:

1. **Average PV cell temperature (T_{avg}):** The average cell temperature of the PV module was calculated to assess the effectiveness of heat dissipation. Performance is best for lower average cell temperature.
2. **Dimensionless temperature non-uniformity (ζ):** PV cell temperature difference ($^{\circ}\text{C}$) non-dimensionalized by the ambient temperature ($^{\circ}\text{C}$) was used to measure the deviation of surface temperatures across the PV module, which impacts electrical efficiency by causing a current mismatch between cells [193]. It is given by:

$$\zeta = \frac{T_{max} - T_{min}}{T_{amb}} \quad (4.11)$$

3. **Power output (P_{PV}):** The actual power production of the PV module as a function of the PV-rated power (P_R) in W, PV derating factor (D_F), cell temperature (T_C) in

K, reference temperature (T_{ref}) K, reference solar irradiance (G_{ref}) in W/m² and the power temperature coefficient (τ_p) in 1/K is given by [194], [195]:

$$P_{PV} = P_R \times D_F \times \left(\frac{G_T}{G_{ref}} \right) \times [1 + \tau_p \times (T_c - T_{ref})] \quad (4.12)$$

4. Electrical efficiency (η_e): The electrical performance of the solar PV can be assessed using solar cell power conversion efficiency which is defined as the ratio of the actual power output and the incident power input on the PV surface as follows [196], [197]:

$$\eta_e = \frac{P_{PV}}{G_T \times A} \cdot 100\% \quad (4.13)$$

where P_{PV} , G_T , and A are the actual PV power output (W), Solar radiation incident on the PV array (W/m²), and the PV surface area (m²), respectively.

4.4 Results and discussion

This section establishes the results obtained from the numerical simulations of the various photovoltaic cooling configurations previously discussed. The analysis includes thermal, electrical, and overall performance, providing a thorough evaluation of each design's effectiveness in heat dissipation. The outcomes are contrasted and analyzed to highlight the advantages and limitations of each design.

4.4.1 Thermal performance of non-concentrated and concentrated PV

without cooling

Figure 4.8(a) and (b) displays the mean cell temperature of uncooled non-concentrated and

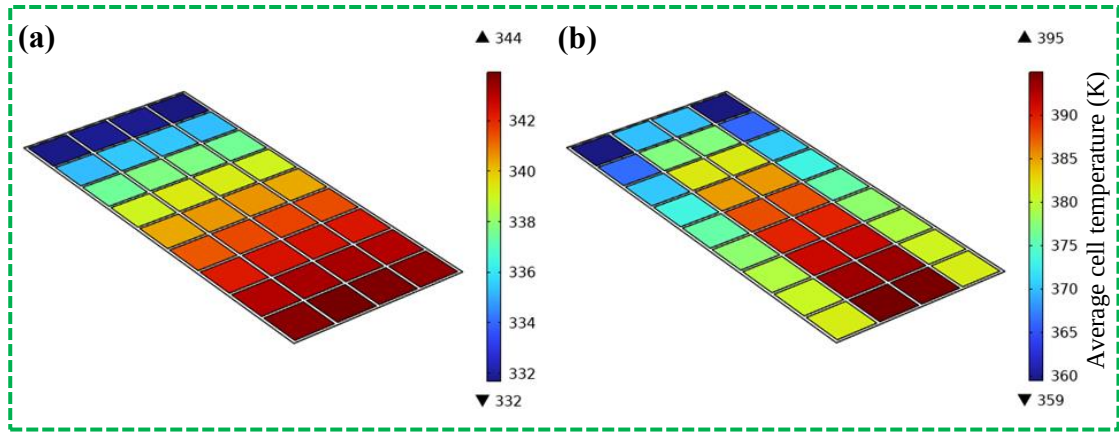


Figure 4.8: Average cell temperature of (a) non-concentrated and (b) 2X V-Trough concentrated PV at 800 W/m^2

V-trough concentrated PV at 800 W/m^2 solar irradiation, 3 m/s ambient air velocity, and 298 K ambient temperature. The non-concentrated PV functions at a peak average temperature of 344 K . The airflow direction causes the cell temperature to rise from a minimum of 332 K at the air entrance to 344 K at the outflow, resulting in a temperature difference of 12 K . In contrast, the V-trough concentrated PV depicted in Figure 4.8(b) functions at a maximum average cell temperature of 395 K . The temperature distribution indicates that the middle cells, characterized by larger solar concentrations, exhibit the highest temperatures, whilst the outer rows, have reduced temperatures. Similarly to the non-concentrated PV, the temperature gradient escalates from the air inlet section towards the outlet. The temperature non-uniformity in this instance is 36 K . The average cell temperature of the concentrated PV is 51 K greater, and the surface non-uniformity is 24 K greater than that of the non-concentrated PV.

4.4.2 Thermal design of optimum cooling pipe

Figure 4.9 and Figure 4.10 illustrates the results of integrating the five pipe layouts outlined in Section 2. Figure 4.9(a) and Figure 4.9(b) illustrate the effect of water cooling at a flow

velocity of 0.01 m/s on non-concentrated PV. The conventional serpentine pipe decreased the PV temperature from 344 K to 322 K, as illustrated in Figure 4.9(a). The temperature non-uniformity in the cells reaches a maximum of 7 K, which may adversely affect the electrical performance of the PV module. The significant non-uniformity arises from the counterflow between water cooling and ambient air cooling. The cells at the water inlet exhibit the lowest temperature, while the temperature of the cells rises as the water progresses toward the outflow, accumulating additional heat. This indicates that the impact of water cooling is more significant than that of air cooling. To mitigate this non-uniformity, we designed the serpentine pipe to traverse the PV module twice, positioning both the pipe inlet and outlet on the same side while ensuring equal pipe length, as illustrated in Figure 4.9(b). This design not only reduced the average cell temperature by a comparable degree but also decreased the surface non-uniformity to 4 K. This configuration relocates the hotspot from the end side to the right side of the module, directing it towards the new outlet. Consequently, we have chosen this pipe configuration as the champion for the forthcoming designs of water cooling in non-concentrated PV.

Figure 4.10 illustrates the thermal design of water cooling integrated into a V-trough concentrated PV. The conventional serpentine pipe shown in Figure 4.10(a) decreased the average PV cell temperature from 395 K to 345 K, exhibiting a surface non-uniformity of 13 K. Similar to non-concentrated PV, the temperature gradient ranges from low at the water entrance to high at the outflow section. Due to the V-trough concentration, the cells in the central two rows display a higher temperature than the exterior two rows. Figure 4.10(b) illustrates the first pipe optimization for concentrated PV systems. We focus the pipe in the central area of higher solar concentration by decreasing the serpentine width

and augmenting the number of bends from 8 to 20 while ensuring an equal pipe length. This also decreased the average cell temperature by 50 K and the non-uniformity to 13 K. To enhance the serpentine pipe configuration, we devised the water to reverse flow toward the entrance once the concentrated section of the PV was cooled, as illustrated in

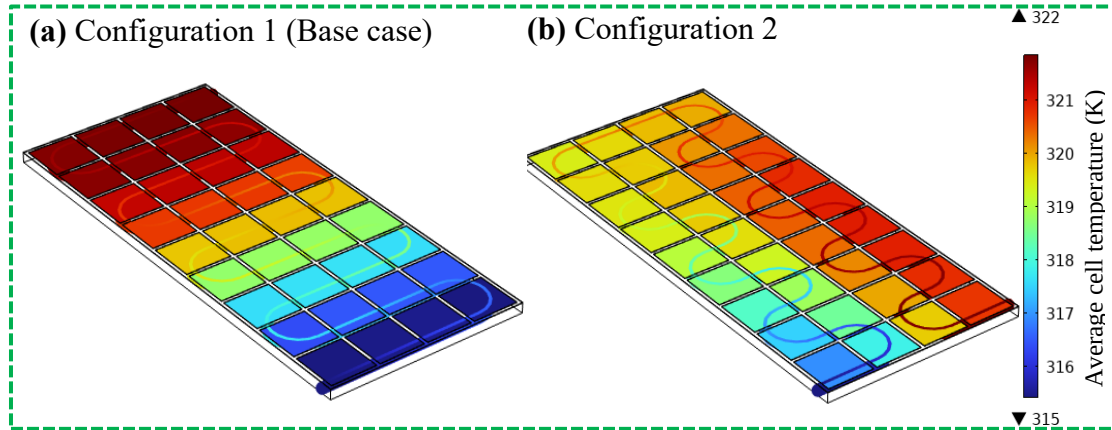


Figure 4.9: Cooling pipe configurations for non-concentrated PV

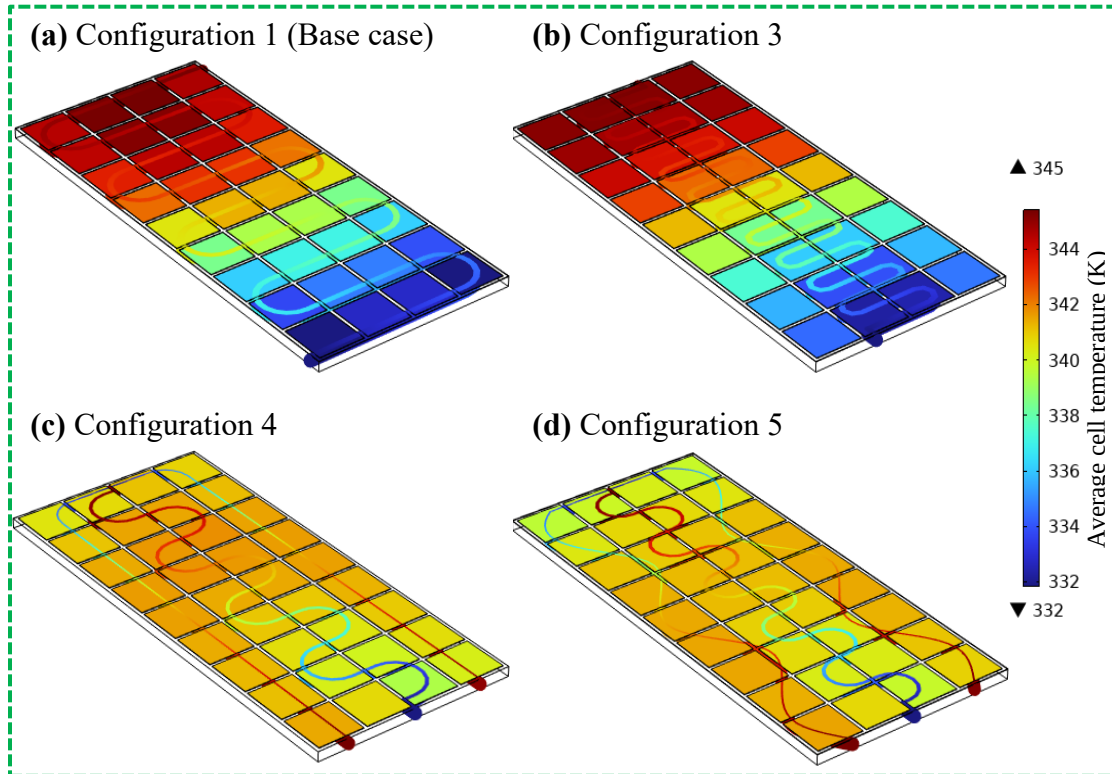


Figure 4.10: Cooling pipe configurations for concentrated PV

Figure 4.10(c). This decreased the average PV temperature by an extra 2 K and enhanced the temperature uniformity to 4 K. Finally, the backflow pipe in configuration 4 is constructed in a large-radius serpentine shape, as illustrated in Figure 4.10(d). This not only decreased the PV temperature by the same degree as Configuration 4 but also diminished the temperature non-uniformity to merely 3 K. Consequently, we designate Configuration 5 as the champion pipe design among the configurations studied.

4.4.3 Thermal design of optimum fins

This section presents the design of an optimum fin for cooling standalone PV, along with PV coupled with water cooling. Its optimization entails arranging higher fin density in high-temperature areas using longitudinal bifurcation and increase in fin height. Figure 4.11 illustrates the optimal fin designs for PV and water-cooled PV systems for non-concentrated PV under conditions of 800 W/m^2 solar irradiation, 3 m/s ambient air velocity, 298 K ambient temperature, and 0.01 m/s inlet water velocity. Figure 4.11(a) illustrates the PV cooled using the base case fins which decreased the peak average cell temperature from 344 K to 311 K , yielding a surface non-uniformity of 2 K . Figure 4.11(b) illustrates the optimization of the fins by bifurcating them beneath the three rows at the outlet section. This decreased the average cell temperature by 2 K and enhanced the non-uniformity to below 1 K . Figure 4.11(c) illustrates the non-concentrated PV with water cooling at 0.01 m/s incorporated with the base case fin heatsink which decreased the temperature from 321 K to 308 K . We improved the heatsink by bifurcating the fins beneath the final three rows and column cells, as depicted in Figure 4.11(d). This results in a negligible reduction in the average cell temperature compared to the base case heatsink. However, we reduced the temperature nonuniformity to less than 1 K .

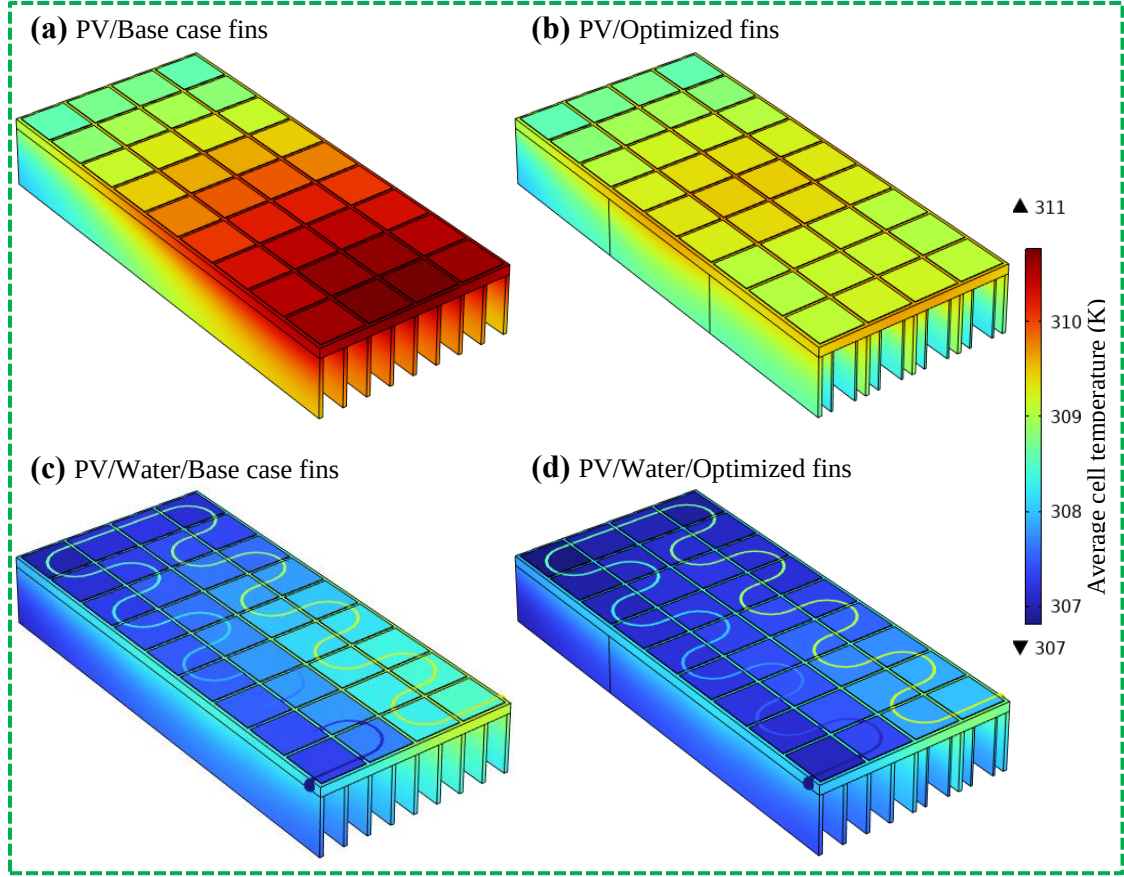


Figure 4.11: Non-concentrated (a) PV with base case fins, (b) PV with optimized fins, (c) PV/Water with base case fins, and (d) PV/Water with optimized fins

The concentrated PV is also incorporated with base case and optimized fin heatsink as shown in Figure 4.12. The base case fins decreased the average cell temperature from 395 K to 324 K, despite a 6 K nonuniformity as shown in Figure 4.12(a). The redesigned bifurcated fin heatsinks further decreased these to 321 K and 3 K, respectively as shown in Figure 4.12(b). Figure 4.12(c) and Figure 4.12(d) illustrate the concentrated PV with water cooling, fitted with the base case fins and the optimized fins, respectively. The base case fins lowered the temperature of the concentrated PV with optimal water cooling at 0.01 m/s from 342 K to 318 K. The surface nonuniformity measures approximately 3 K, with hotspots situated in the central rows and at the airflow outlet. The improved heatsink

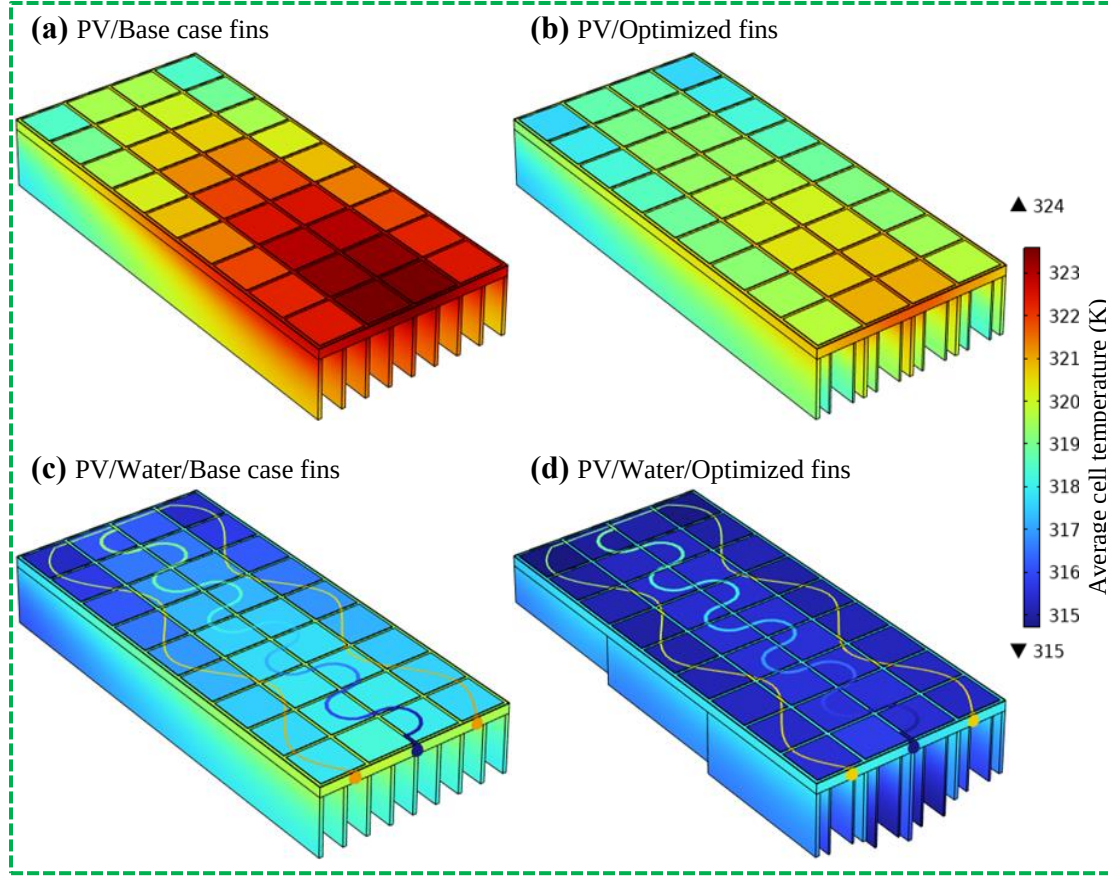


Figure 4.12: Concentrated (a) PV with base case fins, (b) PV with optimized fins, (c) PV/Water with base case fins, and (d) PV/Water with optimized fins

features split fins at the airflow exit section. The height of the fins is augmented by 0.2% of the base case fin, while concurrently being diminished by the same amount at the inlet section to maintain uniform fin volume. This design decreased the module's temperature by 2 K to 316 K and decreased the non-uniformity to below 2 K. This illustrates that the integration of fins substantially lowered the temperature nonuniformity to the minimal achievable level.

4.4.4 Effect of water flow rate on the thermal performance of PV

In water-based PV cooling, an elevated water flow rate enhances the heat transfer coefficient, and Abdul-Ganiyu et al. [200] have demonstrated that photovoltaic operating

temperature and efficiency are dependent on the flow rate. However, the literature presents diverse recommendations for the optimal flow rate, predominantly cited between 0.002 and 0.03 kg/s [201], [202], [203], [204]. Researchers have examined increased flow rates, specifically 0.06 kg/s, and advocate for even greater values ranging from 0.09 to 0.15 kg/s for concentrated PV, taking into account the module's capacity to attain significantly high temperatures [186]. This section examines the impact of altering the water flow rate from 0.006 kg/s (0.002 m/s) to 0.15 kg/s (0.5 m/s) on the thermal performance of the PV system integrated with a water cooling and fin heatsink, aiming to determine the optimal flow rate for each configuration. The flow transitions from laminar ($Re \leq 2300$) to turbulent ($Re > 2300$) at a flow rate of 0.15 m/s.

For non-concentrated PV shown in Figure 4.13(a), both PV with water and PV with water integrated fin cooling show reduced PV temperatures as the flow rate increases. The effect of fins decreases with higher flow rates, marked by three key points (A, B, and C). At Point A (0.002 m/s), fins reduce the PV temperature by 24 K due to low water-cooling capacity; where fins alone match the cooling capacity of water/fin, making combined use redundant. At Point B (0.067 m/s), PV cooled by only fins equals PV cooled by water, while adding fins to PV/Water decreases temperature by an additional 5 K, suggesting a trade-off based on cost. At Point C (0.3 m/s), fins add no thermal benefit to PV/Water. Hence, for non-concentrated PV system, PV/fin is recommended instead of low flow rate water cooling, PV/Water/Fin at intermediate rates (0.02-0.08 m/s), and PV/Water alone at high flow rates (0.1-0.5 m/s).

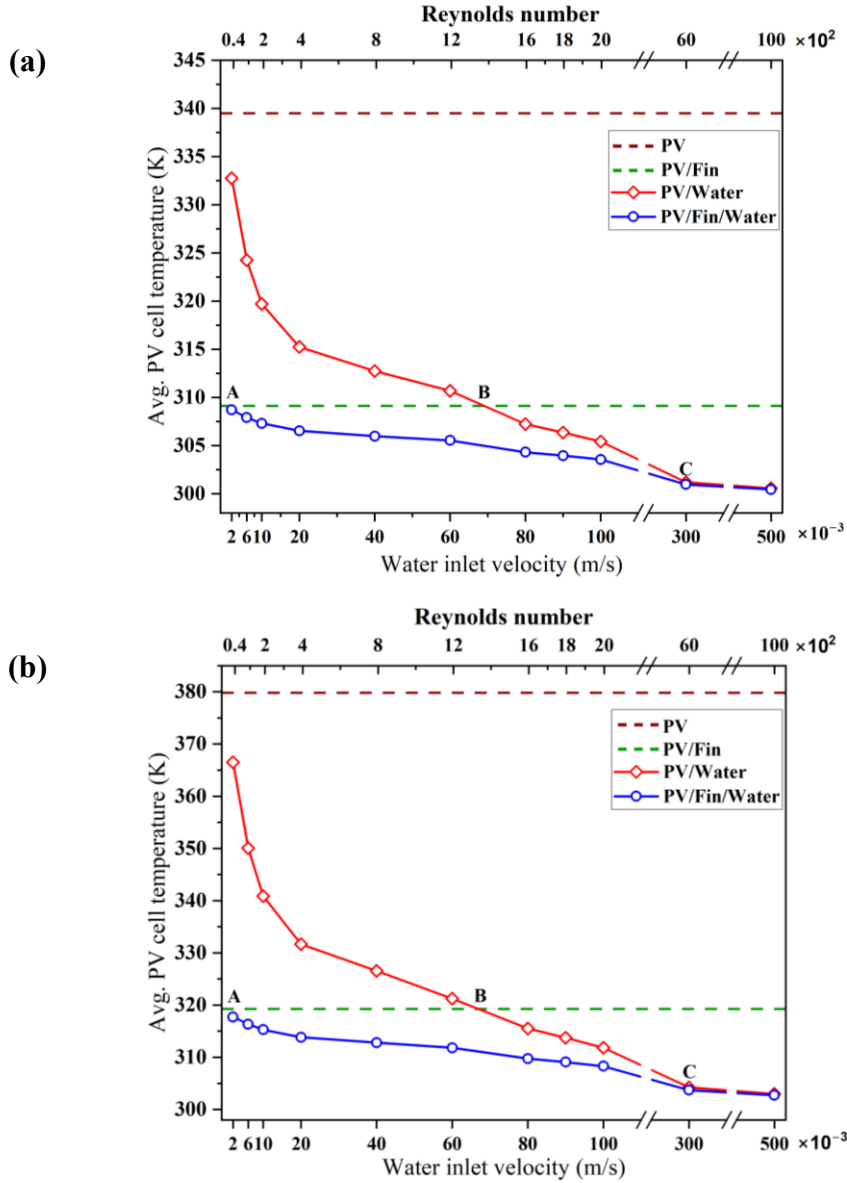


Figure 4.13: Effect of water flow rate on the performance of (a) Non-concentrated, and (b) Concentrated PV integrated with water cooling and fins, $V_{\text{air}} = 3 \text{ m/s}$

In Figure 4.13(b), the trend for concentrated PV mirrors that of non-concentrated PV, exhibiting reduced fin effectiveness at elevated flow rates owing to thermal dynamics. At Point A (0.002 m/s), the performance of PV cooled alone with fins is equivalent to that of PV/Water/Fin. The water-cooling capacity is constrained by the low flow rate, which limits convective heat transfer, rendering the fins the principal cooling mechanism and leading to

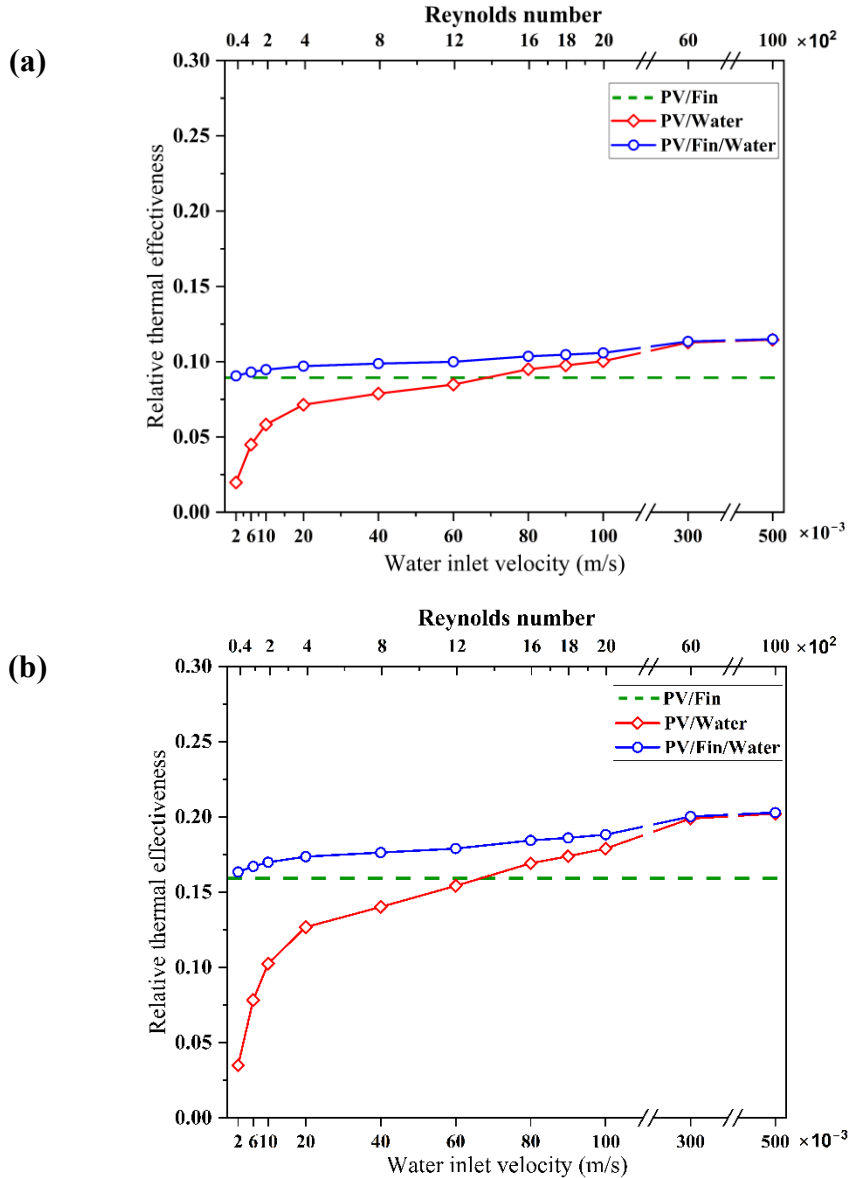


Figure 4.14: Thermal effectiveness of the various configurations relative to the standalone PV for (a) non-concentrated and (b) concentrated solar irradiance

a significant temperature reduction of 48 K upon their incorporation. At Point B (0.067 m/s), when the flow rate increases, the improved convective cooling of PV/Water equals that of PV/Fin, while the addition of fins to PV/Water further enhances the heat transfer area, resulting in a temperature reduction of an additional 8 K. At Point C (0.3 m/s), the water flow rate is sufficiently high to dominate heat dissipation, rendering the additional surface area from fins ineffective, as PV operating temperature nears the room temperature.

Hence, for both concentrated and non-concentrated PV, PV/Fin is most effective at low flow rates instead of PV water, PV/Water/Fin at moderate flow rates, and PV/Water alone at high flow rates. These findings can further be clarified by plotting the thermal effectiveness of the three advanced cooling configurations (PV/Fin, PV/Water, and PV/Fin/Water) relative to the standalone PV system as shown in Figure 4.14. For both concentrated and non-concentrated PV, at low flow rate regime, the results suggest the use of just PV/Fin because the effectiveness of the water cooling is negligible. At moderate flow rate regime, PV/Water can function with effectiveness equivalent to PV/Fin, and PV/Fin/Water configuration is even better. At high flow rate regime, the effect of fin becomes negligible as the thermal load is on the high velocity water cooling layer between the PV and the fins. And hence, PV/Water configuration can be said to be thermally effective at this regime. To conduct a comprehensive analysis, three flow rates: 0.01 m/s (low), 0.067 m/s (moderate), and 0.3 m/s (high) are employed to rigorously assess the performance of each configuration using the outlined performance metrics.

4.4.5 Effect of ambient temperature on the thermal performance of PV

Figure 4.15 illustrates the effect of ambient temperature on the thermal performance of the various configurations. As ambient temperature rises, PV cell efficiency declines due to increased operating temperatures. For both concentrated and non-concentrated PV shown in Figure 4.15(a) and Figure 4.15(b), the average PV temperature rises with ambient temperature, a trend particularly pronounced in uncooled PV and PV with fin cooling, where air-based cooling depends heavily on ambient conditions. In contrast, PV systems cooled with water exhibit a more gradual temperature increase, as convective water cooling effectively mitigates PV heating even at higher ambient temperatures.

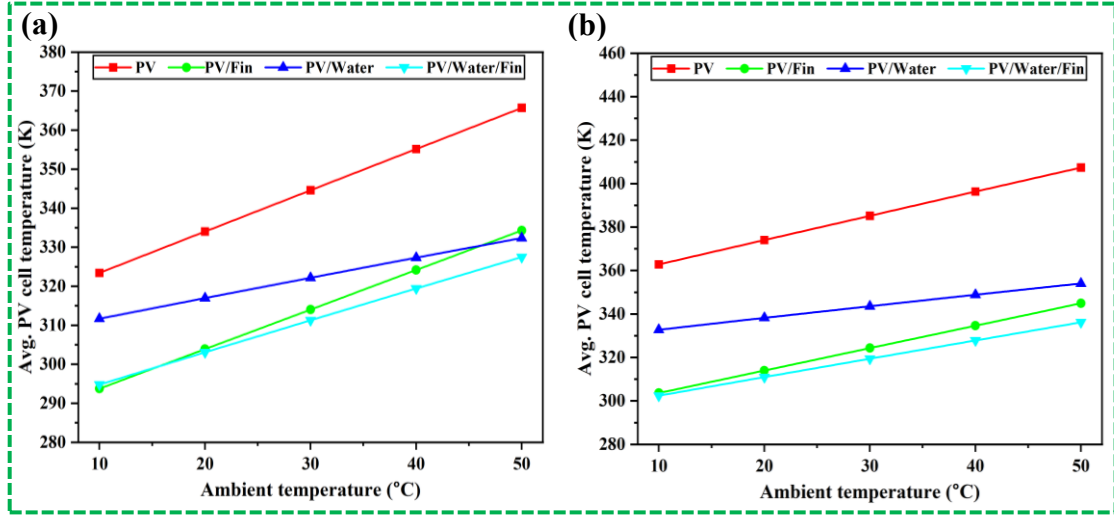


Figure 4.15: Effect of ambient temperature on the performance of (a) Non-concentrated, and (b) Concentrated PV integrated with fins and water cooling at 0.01 m/s

4.4.6 Electrical performance of individual PV cells across different cooling configurations

Figure 4.16 presents the electrical efficiency of individual PV cells across different cooling configurations and standalone PV, calculated using Equation (4.13) for non-concentrated PV. For standalone PV in Figure 4.16(a), the maximum efficiency (11.6%) is observed in the cells near the air inlet, where convective cooling is highest. Efficiency declines progressively toward the outlet, reaching a minimum of 10.9%, due to reduced convective cooling and increased cell temperature. Similar trends in the PV with fins shown in Figure 4.16(b) are observed; however, fin-induced cooling raises the efficiency range to 13% at the inlet, with a slight drop to 12.9% at the outlet. For PV with champion water cooling shown in Figure 4.16(c), cell efficiency follows the temperature gradient along the water flow direction, peaking at 12.5% near the cooler air inlet and gradually dropping to 12.2% toward the outlet. In the combined PV/Water/Fin configuration shown in Figure 4.16(d), cell efficiency is nearly uniform across all cells, between 13% and 13.1%, owing to

minimal temperature non-uniformity (under 1 K) across the PV surface. This highlights the effectiveness of fins in enhancing cooling uniformity.

The cell-based electrical efficiency distribution in concentrated PV for the different cases are shown in Figure 4.17. In a standalone PV with air cooling only, the cells in the outer columns, characterized by lower concentration, attained a maximum cell efficiency of 9.8% as depicted in Figure 4.17(a). cell efficiency declines to a minimum of 7.6% in the two middle columns receiving the maximum solar concentration. The concentrated PV cooled with fins, as illustrated in Figure 4.17(b), exhibits a similar trend. However, owing to the cooling effects of the fins, the observed maximum and minimum efficiencies are 12.4% and 12.2%, respectively, making the cells almost at uniform efficiency.

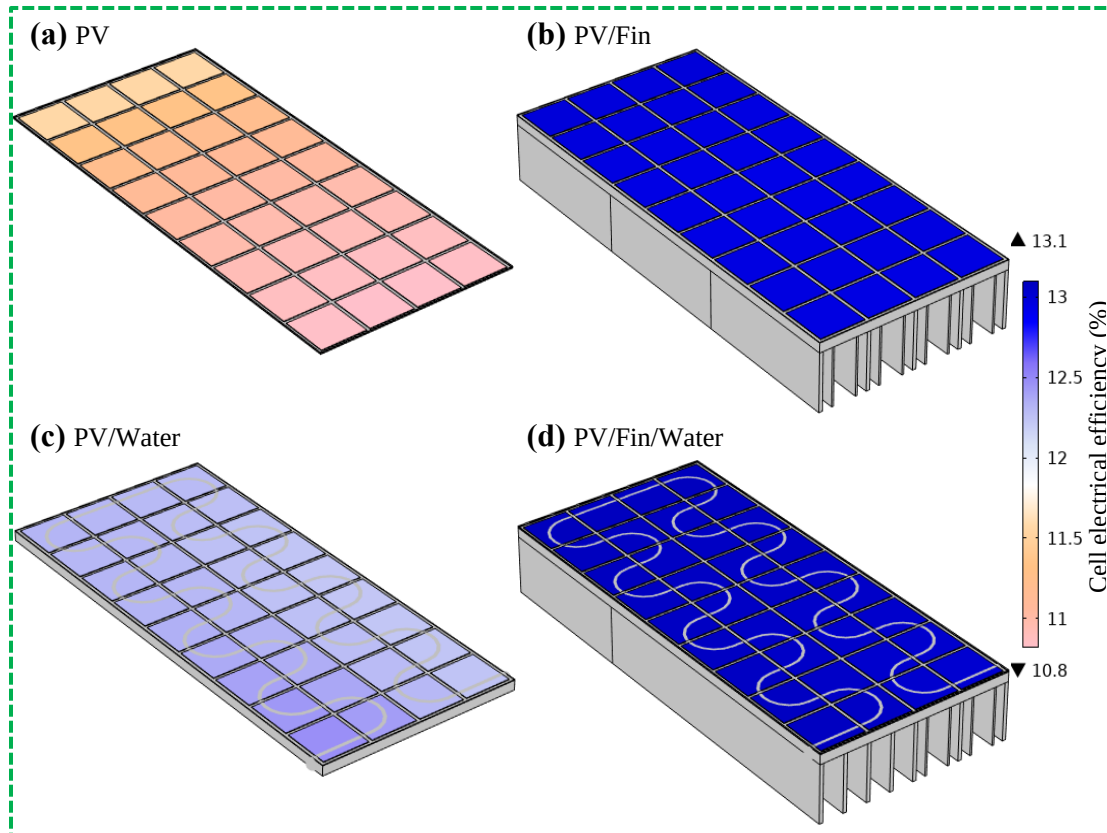


Figure 4.16: Cell electrical efficiency of a non-concentrated (a) PV, (b) PV/Fin, (c) PV/Water, and (d) PV/Fin/Water

Figure 4.17(c) similarly shows the distribution of cell efficiency for the concentrated PV system with the champion water cooling where electrical efficiency reaches its peak at the lowest temperature, air inlet region, around 11.1%. The temperature decreases progressively in the direction of water flow until it attains a minimum of 10.9% at the water outlet section. Ultimately, in the PV/Water/Fin arrangement depicted in Figure 4.17(d), a consistent cell efficiency of 12.6% is achieved owing to the homogeneous temperature distribution across the cells.

In summary, the electrical efficiency of individual PV cells varies significantly by cooling configuration. In non-concentrated PV, standalone systems exhibit temperature-driven efficiency drops from inlet to outlet, while adding fins or water cooling reduces these

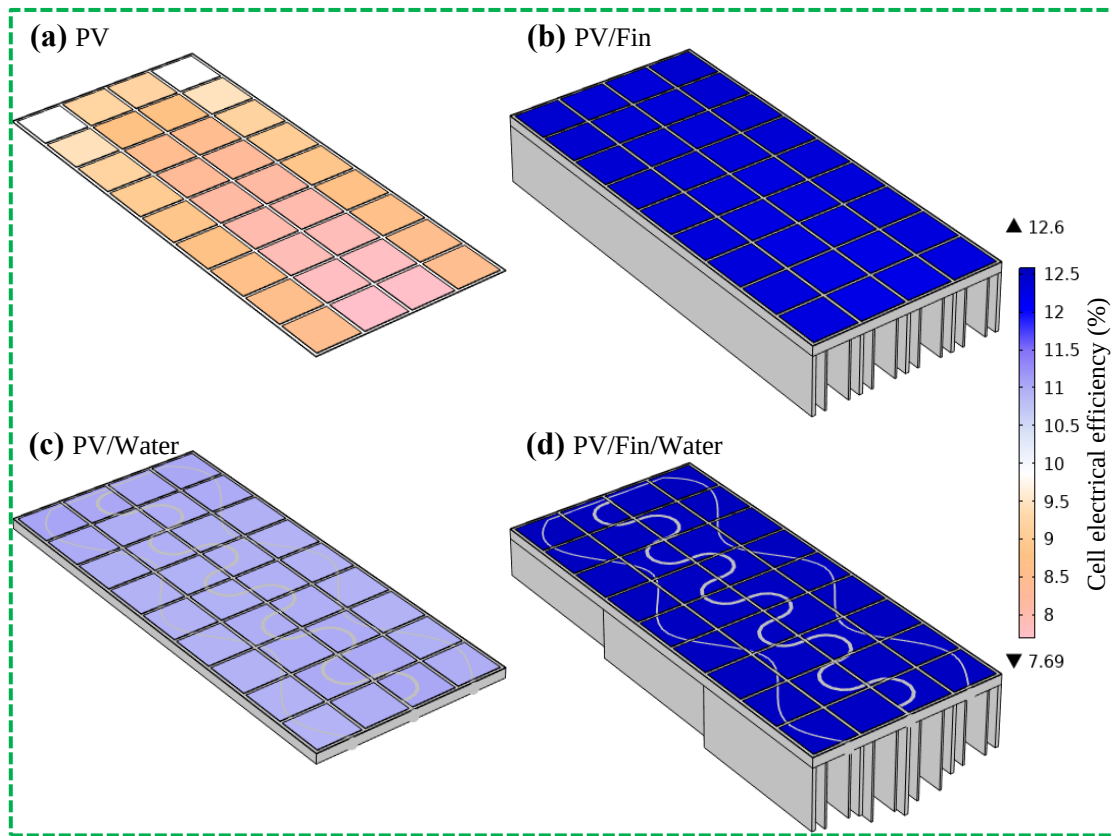


Figure 4.17: Cell electrical efficiency of a concentrated (a) PV, (b) PV/Fin, (c) PV/Water, and (d) PV/Fin/Water

variations and raises efficiency. The PV/Water/Fin combination achieves near-uniform efficiency across cells due to minimized temperature differences. For concentrated PV, efficiency peaks in outer cells and declines centrally where solar concentration is highest, but fins and water cooling improve uniformity. The PV/Water/Fin setup again shows the most consistent performance. A detailed efficiency comparison for these configurations follows in the next section.

4.4.7 Comparison of the four cooling cases (PV, PV/Fin, PV/Water, and combined PV/Fin/Water) based on the performance metrics

Thermal management in PV systems is pivotal in maximizing PV efficiency and lifespan. This section compares the air-cooled standalone PV panel with three advanced cooling cases; PV integrated with fins (PV/Fins), PV with water cooling (PV/Water), and PV with a combined fin and water-cooling (PV/Fin/Water). These configurations are evaluated in terms of their ability to manage cell temperature, reduce thermal gradients between cells, and ultimately enhance power output and electrical efficiency. By examining these variations, this analysis aims to identify the most effective cooling strategy under different water flow rates, offering insights into the benefits of thermal management for high-performance PV applications.

Figure 4.18 and Figure 4.19 presents the comparison of PV cooling configurations based on the five performance metrics for non-concentrated and concentrated PV. Figure 4.18(a) shows the effect of cooling configurations on the average PV cell temperature. The air-cooled PV maintains a high average temperature of 339 K, with significant improvements recorded by integrating advanced cooling configurations. At 0.01 m/s, PV/Water is less effective (319 K) than both PV/Fin and PV/Fin/Water, which reach lower temperatures of

309 K and 307 K, respectively. The combined cooling of PV/Fin/Water offers consistently enhanced cooling, especially at intermediate velocity (0.067 m/s), where it achieves 305 K, surpassing both PV/Fin (309 K) and PV/Water (309 K). At the highest flow rate (0.3 m/s), PV/Water reaches 301.8 K, showing that water cooling alone becomes highly effective at greater velocities. The temperature non-uniformity between the cells is calculated using the dimensionless metric given in equation (4.11). Figure 4.18(b) shows the temperature non-uniformity across cooling configurations for standalone PV, exhibiting a high non-uniformity of 0.53. At the lowest velocity, the PV/Water configuration's non-uniformity remains relatively high at 0.19 compared to PV/Fin (0.04) and PV/Fin/Water (0.06), illustrating that fins help in achieving uniform cell temperature. As velocity increases to 0.067 m/s and 0.3 m/s, the nonuniformity in the PV/Water and PV/Fin/Water remains almost constant at 0.07 and 0.05, respectively. These correspond to less than 2 K of temperature non-uniformity.

Figure 4.18(c) illustrates the power output across various cooling configurations. The PV/Fin/Water configuration demonstrates better performance across all inlet velocities compared to PV/Fin and PV/Water. Also, the power output rises with increased water inlet velocity, with PV/Fin/Water reaching the highest value of 71 W at 0.3 m/s, followed closely by PV/Water at 70.9 W, both significantly exceeding the standalone PV baseline of 58.6 W. This enhancement is attributed to effective heat dissipation, preventing temperature-induced degradation of power output as observed in the standalone PV's lower power output of 91 W. The use of fins and water synergistically ensures both rapid convective cooling and lower thermal resistance across the panel. Figure 4.18(d) assess electrical efficiency for non-concentrated PV cooling configurations. Since electrical

efficiency is a direct function of the power output, a similar trend is recorded. The air cooled PV has electrical efficiency of 11.1%, while all cooling configurations enhance efficiency due to reduced cell temperature. The PV/Fin configuration has an electrical efficiency of 12.9% while the PV/Water and PV/Fin/Water configurations have maximum efficiency of 13.4% at 0.03 m/s.

The corresponding analysis under concentrated conditions is presented in Figure 4.19. Similar trends in average cell temperature to those observed under non-concentrated conditions are recorded, as shown in Figure 4.19(b), although the temperatures are generally higher due to increased solar radiation. The standalone air-cooled PV reaches 379.8 K, while PV/Fin and PV/Fin/Water exhibit significantly lower average temperatures, particularly at intermediate and high flow velocities. PV/Fin/Water demonstrates superior cooling performance, achieving 310.97 K at intermediate flow and 304.47 K at 0.3 m/s. The PV/Water configuration achieves 305.27 K, indicating sufficient performance at high flow rates without the inclusion of fins. Figure 4.19(b) also illustrates greater non-uniformity in temperature distribution under concentrated conditions. Uncooled PV records a maximum non-uniformity of nearly 1.9, while PV/Fin exhibits the second highest at 0.2. For PV/Water and PV/Fin/Water, non-uniformity remains nearly constant as flow rates increase, with minimum values of 0.06 and 0.05 recorded at 0.3 m/s, respectively.

Figure 4.19(c) highlights the power output trends under concentrated conditions, where the PV/Fin/Water configuration achieves the highest power output of 140 W at 0.3 m/s, demonstrating its efficient cooling capability. Similar to the non-concentrated case, power output increases with flow rate for systems incorporating water cooling. The standalone PV generates the lowest power, 91 W, due to its elevated operating temperature, as shown

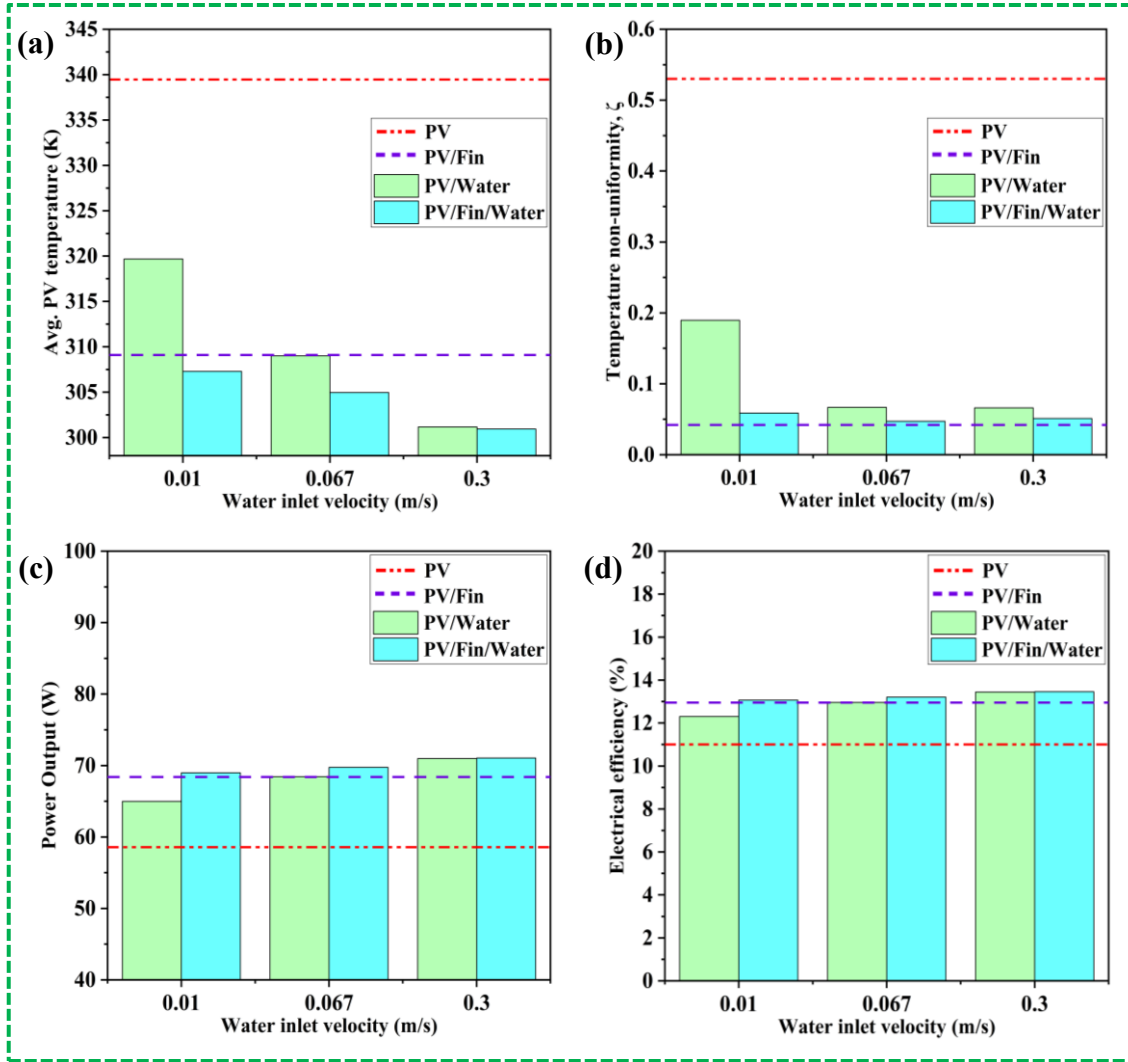


Figure 4.18: Comparison of the non-concentrated PV cooling cases based on (a) average cell temperature, (b) temperature nonuniformity, (c) power output, (d) electrical efficiency

in Figure 4.19(a). Efficiency trends are shown in Figure 4.19(d). Concentrated conditions result in reduced efficiency for the standalone air-cooled PV, which stabilizes at 8.6%.

Advanced cooling methods, however, significantly enhance efficiency. PV/Fin/Water achieves the highest efficiency of 12.6% at 0.01 m/s and further improves to 13.2% at 0.067 m/s, outperforming PV/Fin (12.3%) and PV/Water (12.4%). At the highest velocity, PV/Water closely matches PV/Fin/Water, reaching a peak efficiency of 13.4%. These results underscore the consistent performance of the PV/Fin/Water configuration in

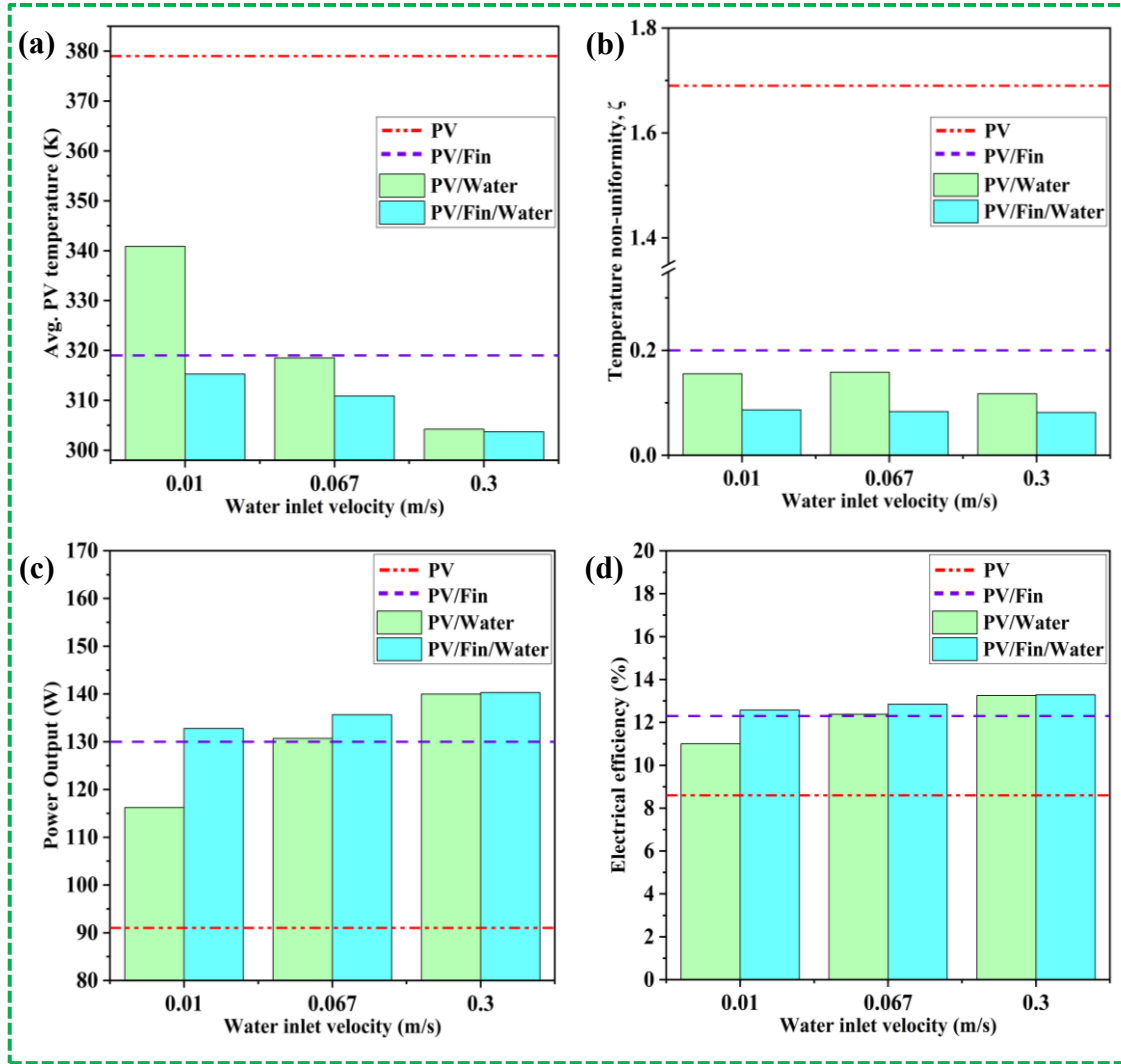


Figure 4.19: Comparison of the concentrated PV cooling cases based on (a) average cell temperature, (b) temperature nonuniformity, (c) power output, (d) electrical efficiency

maximizing power output and efficiency under concentrated conditions, effectively mitigating thermal resistance and maintaining PV performance across operational regimes. In summary, the comparative analysis presented in Table 4-3 highlights the superior performance of the PV/Fin/Water system across all metrics under both concentrated and non-concentrated solar conditions. This configuration consistently ranked first in all individual metrics, achieving a 53.6% increase in power output under concentrated conditions at high water flow rates (0.3 m/s). The PV/Water system demonstrated strong

performance as a standalone active cooling solution, ranking second in most metrics, and is particularly effective at high flow velocities where fins become less impactful. Conversely, the PV/Fin system showed excellent capability in reducing temperature non-uniformity but ranked third in metrics such as power output and efficiency, reflecting the trade-offs of the finned heatsink.

Table 4-3: Ranking of the different cases based on the performance metrics

Condition	Velocity (m/s)	Cases	Decrease in Avg. Temp. (%)	Decrease in non-uniformity (%)	Increase in power output (%)	Increase in efficiency (%)	Rank Avg. Temp	Rank Non-uniformity	Rank Power Output	Rank Efficiency
Non-concentrated	0.01	PV/Fin	8.9	92.1	16.8	16.8	2	1	2	2
		PV/Water	5.8	64.3	10.9	10.9	3	3	3	3
		PV/Fin/Water	9.5	88.9	17.8	17.8	1	2	1	1
	0.067	PV/Fin	8.9	92.1	16.8	16.8	3	1	3	3
		PV/Water	9.0	87.4	16.9	16.9	2	3	2	2
		PV/Fin/Water	10.2	91.1	19.1	19.1	1	2	1	1
	0.3	PV/Fin	8.9	92.1	16.8	16.8	3	1	3	3
		PV/Water	11.3	87.6	21.2	21.2	2	3	2	2
		PV/Fin/Water	11.3	90.4	21.3	21.3	1	2	1	1
Concentrated	0.01	PV/Fin	15.9	88.2	43.1	43.1	2	3	2	2
		PV/Water	10.2	90.8	27.6	27.6	3	2	3	3
		PV/Fin/Water	17.0	94.9	45.9	45.9	1	1	1	1
	0.067	PV/Fin	15.9	88.2	43.1	43.1	2	3	2	2
		PV/Water	15.9	90.6	43.1	43.1	3	2	3	3
		PV/Fin/Water	18.1	95.1	49.0	49.0	1	1	1	1
	0.3	PV/Fin	15.9	88.2	43.1	43.1	3	3	3	3
		PV/Water	19.6	93.1	53.0	53.0	2	2	2	2
		PV/Fin/Water	19.8	95.2	53.6	53.6	1	1	1	1

CHAPTER 5

CONCLUSION & RECOMMENDATIONS

5.1 Summary

This thesis presents a comprehensive investigation into advanced thermal management techniques for photovoltaic (PV) systems, addressing the critical challenge of temperature-induced performance degradation. The research spans three key areas: (1) a detailed review of the cooling technologies for PV systems, (2) the design and optimization of innovative finned heatsinks for air-cooled PV systems, and (3) the integration of air-cooled fins with water cooling systems for enhanced thermal management. Using numerical analysis, this work provides novel insights into improving PV performance by integrating finned heatsink and water cooling under concentrated and non-concentrated solar irradiance.

The literature review in Chapter 2 synthesized recent advancements in PV cooling methods, categorized into passive and active techniques. It highlighted research gaps, including the absence of integrated air and water cooling solutions for PV systems, motivating the subsequent studies in this thesis. In Chapter 3, advanced fin configurations such as bifurcated longitudinal split fins, variable height gradient fins, and hybrid plate-pin fins were optimized for thermal and hydraulic performance. Chapter 4 expanded on this by integrating water channels into finned heatsink designs, achieving significant improvements in thermal and electrical performance.

The research demonstrated that innovative fin configurations can significantly reduce temperature non-uniformity and thermal resistance, while integrated air-water cooling

systems provide enhanced PV operating temperature and efficiency. This work establishes a robust foundation for future research in PV thermal management, offering scalable solutions for high-performance applications.

5.2 Conclusion

This thesis concludes that advanced cooling configurations are essential for improving PV performance, particularly under high-irradiance conditions. Key findings include:

1. Performance of Finned Heatsinks (Chapter 3):

- The hybrid perforated plate fin configuration achieved the lowest baseplate temperature, reducing it by up to **68%** compared to the baseline rectangular plate fins.
- The bifurcated longitudinal split fin configuration demonstrated a strong balance of thermal and hydraulic performance, reducing temperature by **39%** while maintaining manageable pressure drops.
- Pressure drops, a critical hydraulic parameter, were highest for the hybrid plate-pin fin integrated system due to its complex geometry.
- The rectangular plate fin design, while hydraulically efficient, exhibited the lowest thermal performance.

2. Integrated Air and Water Cooling Systems (Chapter 4)

- **Thermal Performance:**

The PV/Fin/Water configuration significantly reduced the average PV cell temperature compared to standalone PV and single cooling methods. In concentrated PV systems, the integrated configuration achieved a temperature

reduction of up to **91 K**, representing a **75% reduction** relative to standalone PV systems. In non-concentrated systems, the reduction was approximately **36 K**, equivalent to a **60% improvement** over standalone configurations.

- **Temperature Uniformity:**

The bifurcated fin design effectively minimized temperature non-uniformity across the PV surface. For both concentrated and non-concentrated conditions, the temperature difference was reduced to less than **1 K**, ensuring consistent cell performance and mitigating localized hot spots. This uniformity improvement was critical for enhancing overall PV reliability and efficiency.

- **Power Output:**

The combined cooling configurations substantially enhanced power output. Under concentrated irradiance, the power output increased from **91 W** in standalone systems to **140 W** with the optimized PV/Fin/Water design, marking an improvement of **54%**. Similarly, for non-concentrated conditions, the output increased from **58.6 W** to **71 W**, representing a **21% enhancement**.

- **Electrical Efficiency:**

The integrated cooling system significantly improved electrical efficiency by mitigating thermal losses. Under concentrated solar irradiance, efficiency increased from **8.6%** in standalone PV systems to **13.4%**, reflecting a **56% improvement**. Under non-concentrated conditions, efficiency rose from **11.1%** to **13.4%**, a **21% increase**.

- **Flow Rate Optimization:**

The study identified an optimal water flow rate range of **0.02–0.08 m/s**, where the combined cooling method achieved superior thermal performance without excessive energy consumption. This balance highlights the importance of optimizing hydraulic parameters to minimize pumping power requirements while maximizing cooling efficiency.

5.3 Limitations

The findings in this study provides a valuable insight into the potential of integrated PV/Fin/Water configuration, however, several limitations must be acknowledged:

1. Economic Analysis

This study does not provide a detailed economic analysis. Key aspects such as the cost of materials, the, operational costs of the cooling system, and the levelized cost of energy have not been assessed. Incorporating these factors would provide a more comprehensive evaluation of the system's economic viability in real-world applications.

2. Irradiance Assumptions

The simulations and analyses are conducted under a single solar irradiance value of 800 W/m^2 for both concentrated and non-concentrated PV. This fixed irradiance does not account for the variability in solar radiation over time or across different geographic locations, which may influence the performance and feasibility of the cooling system.

3. Cooling System Design

The design of the cooling system, including the pipe layout and fin configuration, is based on the resulting temperature distribution across the photovoltaic surface. While this approach ensures thermal optimization, it does not consider alternative

configurations that might improve manufacturability, cost, or efficiency under varying operational conditions.

4. Limited Environmental Conditions

The study does not explore the effects of varying wind speeds, or other environmental factors that can significantly influence the thermal and electrical performance of the integrated PV system.

5.4 Recommendations for future work

The advancements presented in this thesis lead to the following key recommendations:

1. Economic Analysis:

- Perform a comprehensive simulation using hourly solar irradiance data over a year to estimate the annual energy production for various configurations (PV, PV/Fin, PV/Water, and PV/Fin/Water).
- Calculate the Levelized Cost of Energy (LCOE) for each configuration and compare their economic viability under different climatic and operational conditions.

2. Experimental Validation:

- Conduct experimental studies to validate the CFD simulation results, focusing on temperature reduction, pressure drops, and electrical efficiency under real-world conditions.

3. Energy Optimization:

- Develop strategies to minimize pumping power requirements in hybrid cooling systems without compromising thermal performance.

- Investigate the use of renewable-driven pumps (e.g., solar-powered pumps) to enhance the sustainability of hybrid systems.

4. Material Innovations:

- Explore the use of advanced materials with higher thermal conductivity, such as graphene-enhanced composites, for fin and water channel designs.
- Investigate corrosion-resistant materials to improve the durability of water-based cooling systems.

5. Advanced Modeling and Control:

- Incorporate dynamic weather data into simulations to better understand the performance of cooling systems under fluctuating environmental conditions.
- Develop control algorithms for hybrid cooling systems to dynamically adjust flow rates and fan speeds for optimal performance.

6. Use of artificial intelligence (AI) and machine learning (ML) for optimization

- AI/ML models can be trained on simulation data or experimental results to identify optimal pipe layouts and fin configurations that minimize temperature gradients and maximize cooling efficiency.

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