

**INVESTIGATION OF THE PERFORMANCE OF FLAT
TUBE RECEIVER FOR PARABOLIC TROUGH
COLLECTOR**

**BY
MOHAMMED KHALIFA MOHAMED SAADELDIN**

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COLLEGE OF GRADUATE AND INTERDISCIPLINARY STUDIES

This thesis, written by **Mohammed Khalifa Mohamed Saadeldin** under the direction of his thesis advisor and approved by his/her thesis committee, has been presented and accepted by the Dean, College of Graduate and Interdisciplinary Studies, in partial fulfillment of the requirements for the degree of **MASTER OF SCIENCE IN MECHANICAL ENGINEERING**.



Dr. Mohamed A. Antar
Department Chairman



Dr. Suliman S. Al-Homidan
Dean, College of Graduate and
Interdisciplinary Studies

27/5/25

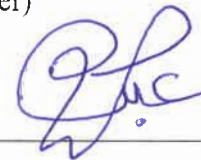
Date



Dr. Shahzada Z. Shuja
(Advisor)



Dr. Syed M. Zubair
(Member)



Dr. Abdulrahman S. AlMerbati
(Member)



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Dedication

***To my beloved Family, my Friends, and my Colleagues
.. with love.***

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LIST OF ABBREVIATIONS

CR	Concentration Ratio
CSP	Concentrated Solar Power
CFD	Computational fluid dynamic
DNI	Direct Normal Irradiance
FEM	Finite Element Method
FVM	Finite Volume Method
GRA	Grey Relational Analysis
HTF	Heat Transfer Fluid
HF	Horizontal flat
<i>LCR</i>	Local Concentration Ratio
MCM	Monte Carlo Method
MCRT	Monte Carlo Ray Tracing
NREL	National Renewable Energy Laboratory
PTC	Parabolic Trough Solar Collector
PEC	Performance evaluation criterion
RMAD	Relative Mean Absolute Deviation
RBCR	Relative Benefit-Cost Ratio
RT	Ray Tracing
SR	Secondary Reflector
S2S	Surface to surface model
VF	Vertical flat

Nomenclature

L	Collector length (m)
W	Aperture width (m)
f	Focal length (m)
Ψ_{rim}	Rim angle (°)
d_{go}	Glass cover outer diameter (m)
d_{gi}	Glass cover inner diameter (m)
d	Base circle diameter (m)
ρ_r	Reflectivity of parabolic reflector
α_a	Absorptivity of the absorber
τ_a	Transmissivity of the glass cover
r_a	The radius of the two half circles of the flat tube geometry (m)
a	The stretching ratio of the flat tube cross-section
C	The length of the flat lines of the flat tube cross-section (m)
$\kappa_1 - \kappa_5$	Randomly generated numbers between 0 and 1
θ	Circumferential angle of the absorber (°)
θ_s	Radial angle of the point within the solar disk (mrad)
φ_s	circumferential angle of the point within the solar disk (mrad)
δ	Sun radial angle ($\delta = 4.65$ mrad)
$\vec{n}_1$	Normal unit vector of the inner glass cover
$\vec{n}_0$	Normal unit vector of parabolic reflector
$\vec{n}$	Normal unit vector
$\vec{r}$	Reflected unit vector
$\vec{\lambda}$	Incident unit vector
$\vec{t}$	Transmitted unit vector
i	Refractive indices ratio
I_i	Local flux density on the i th grid (W/m ²)
I_{eff}	Effective incident irradiance on the aperture (W/m ²)

N_n	Total number of grids
η_0	Optical efficiency (%)
γ	Intercept factor
LCR_{ave}	Average local concentration ratio
LCR_{max}	maximum local concentration ratio
ε	Flux non-uniformity
u, v, w	Velocity components (m/s) in x, y, z directions respectively.
p	Fluid pressure (Pa).
ρ	Fluid density (kg/m ³)
μ_{eff}	Effective dynamic viscosity due to the turbulence (Pa.s)
μ	Dynamic viscosity (Pa.s)
μ_t	Turbulent (eddy) viscosity (Pa.s)
λ_{eff}	Effective thermal conductivity due to turbulence (W/m.K)
λ_t	Turbulent (eddy) conductivity (W/m.K)
λ	Thermal conductivity (W/m.K)
C_p	Heat capacity at constant pressure (J/kg.K)
Φ	Viscous heat dissipation term (W)
k	Turbulent kinetic energy (m ² /s ²)
ε	turbulence dissipation rate (m ² /s ³)
σ_k	Turbulent Prandtl number for kinetic energy
σ_ε	Turbulent Prandtl number for dissipation rate
J	Radiosity (W)
ε_s	Surface emissivity
ρ_d	Diffuse reflectivity
ε_a	Absorber emissivity
σ_{rad}	Boltzmann radiation constant = 1.380649×10^{-23} J/K
h_w	convective heat transfer loss coefficient (W/m ² K)
V_w	Wind speed (m/s)
d_{go}	Outer glass cover diameter (mm)

Re	Reynolds number
v_m	Fluid mean velocity (m/s)
Q_u	Useful heat gain (W)
I_n	Normal solar irradiance (W/m ²)
A_a	Aperture area (m ²)
η_{th}	Overall thermal efficiency
Nu	Nusselt number
D_h	Hydraulic diameter (mm)
h	Convective heat transfer coefficient (W/m ² K)
T_w	Inner absorber wall temperature (K)
T_{mf}	HTF mean temperature (K)
f_r	Friction factor
ΔP	Pressure drop (Pa)
$\dot{V}$	Volumetric flow rate (m ³ /s)
W_p	Pumping power (kW)
λ_R	Price ratio (ratio between thermal energy price and pumping

ABSTRACT

This thesis presents a comprehensive numerical investigation into the optical and thermal-hydraulic performance of parabolic trough collector (PTC) systems equipped with innovative flat tube absorbers. The study explores two novel absorber designs—horizontal and vertical flat tubes—derived by stretching a conventional circular cross-section while maintaining constant cross-sectional area. Optical performance was first evaluated using Monte Carlo Ray Tracing (MCRT) simulations in SolTrace to determine the non-uniform heat flux profiles for different geometries and base circle diameters. Results revealed that horizontal flat tubes can significantly enhance optical efficiency, achieving up to 21.93% improvement for small diameters (10–20 mm), albeit with increased flux non-uniformity. Conversely, vertical flat tubes improved flux uniformity by up to 11.88% but exhibited slight reductions in optical efficiency. Building on the optical analysis, a thermal-hydraulic assessment was conducted using Finite Element Method (FEM)-based simulations, incorporating the non-uniform flux distributions as boundary conditions. —horizontal and vertical flat tube— were evaluated and compared with the conventional circular tube absorbers across three base circle diameters ($d = 15$ mm, $d = 30$ mm, and $d = 70$ mm). The designs were evaluated across a range of Reynolds numbers (20,000–100,000) and heat transfer fluid (HTF) inlet temperatures (150–350 °C). Performance metrics such as Nusselt number, friction factor, thermal efficiency, pumping power, and Relative Benefit-Cost Ratio (RBCR) were analyzed. The horizontal flat tube consistently demonstrated superior overall thermal performance, with efficiency gains reaching up to 9% at $d = 15$ mm, while the vertical flat tube showed moderate improvements at small diameters but was outperformed by the circular design at larger sizes. However, the improved heat transfer of flat tube configurations was accompanied by increased pressure drop, especially at small base diameters, emphasizing the need to balance efficiency gains with hydraulic penalties. Collectively, the findings offer valuable insights into the trade-offs between optical efficiency, thermal performance, and pumping power, and support the potential of geometrically optimized flat tube receivers for next-generation solar thermal applications.

ملخص الرسالة

الاسم الكامل: محمد خليفه محمد سعد الدين

عنوان الرسالة: تقييم الأداء البصري والحراري-الهيدروليكي لمجمعات القطع المكافئ الشمسية المزودة بمستقبلات أنبوبية مسطحة مبتكرة

التخصص: هندسة ميكانيكية

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تقدم هذه الرسالة دراسة عديدة شاملة للأداء البصري والحراري-الهيدروليكي لأنظمة المجمعات القطعية المكافئة المزودة بمستقبلات أنبوبية ذات أنبوب مسطح. تستعرض الدراسة تصميمين جديدين للمستقبل — الأنبوب المسطح الأفقي والأنبوب المسطح الرأسي — تم الحصول عليهما عن طريق تمديد مقطع دائري تقليدي مع الحفاظ على نفس المساحة المقطعية. تم تقييم الأداء البصري أولاً باستخدام محاكاة تتبع الأشعة بطريقة مونت كارلو لتتبع الأشعة (MCRT) عبر برنامج SolTrace لتحديد توزيعات الفيض الحراري غير المنتظمة لمختلف الأشكال الهندسية وأقطار الدائرة الأساسية. أظهرت النتائج أن الأنابيب المسطحة الأفقية يمكن أن تعزز الكفاءة البصرية بشكل ملحوظ، محققة تحسناً يصل إلى 21.93% للأقطار الصغيرة (10–20 مم)، وذلك على حساب زيادة عدم انتظام توزيع الفيض. بالمقابل، حسنت الأنابيب المسطحة الرأسية انتظام الفيض بنسبة تصل إلى 11.88%، لكنها أظهرت انخفاضاً طفيفاً في الكفاءة البصرية.

استناداً إلى التحليل البصري، تم إجراء تقييم حراري-هيدروليكي باستخدام محاكاة تعتمد على طريقة العناصر المحددة، مع استخدام توزيعات الفيض الحراري غير المنتظم كحدود حرارية في النموذج العددي. تم تقييم التصميمات الثلاثة — الأنبوب المسطح الأفقي والرأسي بالإضافة إلى الأنبوب الدائري التقليدي — عبر ثلاثة أقطار أساسية (15 مم، 30 مم، 70 مم)، وعلى مدى من أرقام رينولدز (20,000–100,000) ودرجات حرارة دخول سائل نقل الحرارة تتراوح بين 150 إلى 350 درجة مئوية. تم تحليل مؤشرات الأداء مثل رقم نسلت، معامل الاحتكاك، الكفاءة الحرارية، القدرة المطلوبة للضخ، ونسبة الفائدة للتكلفة. أظهرت النتائج أن الأنبوب المسطح الأفقي حقق أداءً حرارياً متفوقاً باستمرار، مع زيادات في الكفاءة تصل إلى 9% عند قطر 15 مم، بينما أظهر الأنبوب المسطح الرأسي تحسينات معتدلة للأقطار الصغيرة، ولكنه كان أقل أداءً من الأنبوب الدائري التقليدي عند الأقطار الأكبر. ومع ذلك، كان هذا التحسن في انتقال الحرارة مصحوباً بزيادة في الفقد الهيدروليكي (انخفاض الضغط)، خاصةً عند الأقطار الصغيرة، مما يؤكد على أهمية تحقيق توازن بين مكاسب الكفاءة والعقوبات الهيدروليكية.

بشكل عام، تقدم هذه النتائج رؤى قيمة حول التوازن بين الكفاءة البصرية، الأداء الحراري، ومتطلبات القدرة على الضخ، وتدعم إمكانية استخدام المستقبلات المسطحة المحسنة هندسياً في تطبيقات الطاقة الشمسية الحرارية من الجيل القادم.

CHAPTER 1

INTRODUCTION

1.1 Background

Energy is a critical resource for the advancement and prosperity of any nation. With the rising global energy demand, numerous challenges associated with conventional energy sources are becoming increasingly evident. Key concerns include climate change, driven by greenhouse gas emissions from fossil fuel combustion, and the gradual depletion of non-renewable energy sources over time.

As energy needs climb, renewable options offer a key complement able to meet demand growth in a manner replacing traditional fossil fuel dependency. One of the most promising renewable resources for meeting the increased demand for electricity is solar energy. The sun produces a vast amount of radiation, with approximately 1.7×10^{14} kW [1] reaching the earth's surface which would be sufficient to cover the global annual demand if this amount of energy could be harnessed for 84 min [1]. There are very few environmental risks associated with this renewable energy source if it is used effectively [2]. Solar energy can be utilized in many usable energy forms, including thermal power through solar collectors, electricity through Photovoltaic (PV) systems, and chemical energy using fuel cells and photo-electrolysis, making solar energy a highly favorable choice for diverse application domains such as residential energy systems, commercial facilities, and industrial processes. [3–6].

Concentrated Solar Power (CSP) is a notable technology for harnessing solar energy, offering sustainability, thermal storage capability, and substantial economic returns. CSP systems function

by concentrating solar radiation onto small areas, converting it into heat energy across a temperature range of 150–500°C.

Among CSP technologies, the Parabolic Trough Collector (PTC) stands out. PTC systems represent one of the principals CSP technologies presently being deployed commercially on a large scale[7] . PTC systems focus solar irradiance on an absorber tube using a parabolic reflector. The thermal energy is subsequently conveyed to heat transfer fluid (HTF), increasing its temperature. PTC systems operate at 200–400°C with water/steam for power generation [8] but can reach 550–600°C with molten salts [9]. To improve the feasibility of PTC technology, there is a need to reduce the cost either by increasing the overall efficiency or by minimizing the manufacturing costs.

The receiver part in PTC has the most significant effect on the whole performance of PTC. Exploring alternative receiver shapes, beyond the conventional circular tube, offers opportunities for innovation in PTC design. A flat tube configuration, for instance, may simplify manufacturing and integration processes, potentially enhancing assembly and alignment within PTC frameworks. Enhancing the performance of the PTC system means minimizing energy losses throughout the system. The losses associated with PTC consist of optical, cosine, and thermal losses. Optical losses occur due to two main factors: the material attributes of the PTC system, including the mirror's reflectivity, glass cover' transparency, absorber tube's absorptivity, as well as interception losses caused by

geometrical constraints and non-ideal optical parameters, such as installation inaccuracies, surface slope errors, and misalignment [3,10]. Thermal losses in a PTC system play a major role in reducing overall efficiency and mainly occur through three mechanisms: conduction, convection, and radiation. Enhancing the performance of Parabolic PTC systems can be achieved through various techniques, including the use of selective absorber coatings, high-reflectivity mirrors, anti-reflective glass covers, advanced heat transfer fluids, and turbulence promoters to improve internal heat transfer. In addition to these methods, altering the design of the absorber tube offers a highly effective means of improving both optical and thermal efficiency. Geometrical changes can enhance solar radiation interception and optimize heat transfer by influencing the flow dynamics and surface area, and temperature distribution.

1.2 Problem statement

The global shift toward sustainable energy has placed solar thermal technologies—especially parabolic trough collectors (PTCs)—at the center of renewable energy development. Although significant progress has been made by enhancing PTC performance through advanced heat transfer fluids, selective coatings, and internal flow enhancement methods, the geometric optimization of the absorber tube has been given relatively little attention. Conventional circular absorber tubes dominate current PTC designs, yet present limitations in optical interception and thermal-hydraulic performance. Flat tube configurations may offer several theoretical advantages including improved solar flux interception, increased heat transfer surface area, and potentially more uniform flux distribution. However, their integrated optical and thermal-hydraulic performance has not been comprehensively studied. This research is motivated by the need to explore and

validate alternative absorber geometries as means to significantly improve the overall optical and thermal efficiency of PTC systems, thereby contributing to more effective and economically viable solar thermal applications.

1.3 Thesis objectives

This research primarily seeks to investigate the optical and thermal-hydraulic performance of PTC systems featuring innovative flat tube absorber designs, leveraging advanced computational techniques and parametric analyses. This work is directed toward achieving the following objectives:

- Defining a relation to convert from the conventional circular absorber designs to flat tube designs in both vertical and horizontal orientations.
- Implementing MCRT simulations to evaluate the optical performance of PTC systems of the newly introduced configurations.
- Carrying out a parametric analysis to evaluate the influence of geometrical factors including stretching ratio, base circle diameter, aperture width and focal length on the optical performance.
- Conducting numerical analysis by integrating the results of optical analysis with CFD model to examine the thermal-hydraulic behavior of the flat tube designs along with the conventional circular designs.
- Analyzing and comparing the overall performance of the proposed configurations with the conventional designs.

By meeting these objectives, this research seeks to deepen the understanding of how absorber geometry impacts both the optical and thermal-hydraulic performance of PTC

systems. The integrated methodology—combining geometric design, optical simulation, and CFD-based thermal analysis—offers a comprehensive framework for assessing the potential of flat tube configurations. Ultimately, the insights gained from this research are intended to guide future advancements in PTC receiver design, contributing to the advancing the design of more efficient and optimized solar thermal technologies.

1.4 Thesis organization

This thesis provides five main chapters. Chapter 1 presents the background and motivation for the study, outlines the problem statement and defines the research objectives. Chapter 2 provides a review of existing research related to PTC systems, including simulation methods, the impact of geometric modifications on optical performance, heat transfer enhancement techniques, and establishing the context and identifying gaps that the current research aims to address. Chapter 3 details the methodology and results of the optical performance evaluation using the MCRT method, examining the influence of absorber geometry and key design parameters on solar flux distribution and interception efficiency. Chapter 4 analyzes the thermal and hydraulic performance of the proposed absorber geometries using CFD, integrating results from the optical model to assess thermal efficiency, Nusselt number, friction factor, pumping power, and performance enhancement metrics. Finally, Chapter 5 summarizes the major outcomes of the study, presents the conclusions based on the analysis, and proposes directions for future investigations in the field of PTC system design.

CHAPTER 2

LITERATURE REVIEW

2.1 PTC optical modeling methods

The optical performance of a PTC system significantly impacts its overall efficiency, making the study of solar irradiance distribution on the receiver essential for accurate heat transfer calculations. Simplified geometric analysis was a commonly employed approach in investigating optical performance and some fundamental characteristics in the period from the 1970s to the 1980s[11–15]. Daly [16] proposed a two-dimensional backward ray tracing technique to determine the solar flux distribution on absorbers used in parabolic and cylindrical solar collectors by backtracking the rays from the absorber surface to the sun. His technique couldn't consider the variations of flux with aperture length. Jeter[17] initiated the first semi-finite model to estimate the concentrated radiant flux density for concentrators, later extending it to include real-world factors such as non-uniform solar luminance and reflection losses [18].

Subsequent advancements include Grena's [19] three-dimensional ray-tracing algorithm, which incorporates tracking errors and surface imperfections, Khanna et al.'s[20] analytical model to calculate for the axial and circumferential flux distribution on bent absorber tubes. Over the years, the MCRT method has emerged as a robust and flexible method for evaluating the solar irradiance distribution and the optical performance of complex PTC geometries. This method enables precise simulations of light paths, considering various geometrical parameters and optical factors[21–23].

Shuai et al. [24] presented a novel MCRT technique to analyze the optical behavior of parabolic dish systems coupled with cavity receivers to examine the influence of factors like sunshape, surface slope error, and directional characteristics of the focal flux on radiation distribution. Many other researchers further used MCRT to investigate the flux distribution on PTCs[25–27]. Several researchers have combined MCRT with other numerical methods, such as FEM and FVM, to simulate the coupled photo-thermal processes in PTC systems. [22,28–32].

2.2 The impact of geometrical and optical factors on the optical performance of PTC

Numerous studies have explored the influence of geometric and optical parameters on the optical performance of PTCs, primarily through MCRT approach and analytical modeling. Zou et al. [33] employed MCRT to assess the effects of geometric parameters and found that local concentration ratio (LCR) distribution varies significantly, challenging previous simplified models. Islam et al. [34] examined the factors that affect the optical performance using MCRT method. They found that LCR decreased with increasing focal length, while LCR 's uniformity reduced with higher rim angles. Idrissou et al. [35] used MCRT method to evaluate the effect of concentration ratio (CR) on PTC effectiveness and size. They reported that increasing CR by 2.69% is necessary to achieve a 1% improvement in optical efficiency, although these improvements were non-linear and less pronounced at higher CR s. Higher CR values resulted in larger rim angles. Zou et al. [36] examined the effect of non-ideal optical parameters, like sun shape, optical errors, and incident angles on PTCs. Their findings showed a 41.11% decrease in efficiency with increased incident angle from 0-60° and more uniform temperature distribution but lower efficiency with larger

circumsolar ratios. Zou et al. [37] also evaluated combined optical factor effects, finding larger circumsolar ratios and specular errors improved flux uniformity while high reflector quality enhanced efficiency.

Tang et al. [38] introduced a novel secondary reflector (SR) design for parabolic trough concentrators to improve the absorber heat flux uniformity. The design increased optical efficiency and achieved over 90% heat flux uniformity, outperforming existing SR designs. Hou et al. [39] utilized MCRT approach to assess the optical performance of a PTC equipped with flat receiver. Their findings revealed significant changes in *LCR* at the absorber's center with optimum rim angles ranging between 75° – 90° . The study also found that geometric errors notably impact optical performance, particularly the peak *LCR*. Ben Taher et al. [40] introduced a novel design for PTC. The new design analyzed optically using MCRT method. The technique of Grey Relational Analysis (GRA) was employed to enhance radiation interception and uniformity. The suggested layout achieved 92.24% uniformity and 83.01% optical efficiency.

Hu et al. [41] introduced a newly shaped tube with lower and top flat surfaces to address the considerable radiative heat losses at high temperature. They performed simulations to investigate the optical performance and heat loss and thermal efficiency. According to their findings the newly shaped tube, with optimized opening angles, can enhance thermal efficiency over conventional tubes by 0.62% to 4.64%. Furthermore, because of the more uniform heat flow in the new design, thermal stress was decreased. Jebasingh et al. [42] implemented experimental and numerical work on a new elliptical absorber tube with the aim of evaluating thermal performance and pressure drop at varying mass flow rates (0.014, 0.021, 0.028 kg/s). Their results demonstrated that more uniform heat transfer was

observed in the elliptical absorber tube, leading to reduced thermal fatigue, and increased pressure drop compared to the conventional circular tube. Al-Dulaimi et al. [43] examined different receiver geometries including square, triangle, circular, elliptical, and a unique channel design. Results demonstrated that the channel receiver delivered the best optical efficiency of 84%, whereas the elliptical receiver had the lowest at 70%. Conversely, thermal efficiency results indicated the elliptical receiver performed best at 85% while the circular receiver was least effective at 82%.

2.3 Thermal performance enhancing techniques

There are different approaches adopted by researchers to enhance the thermal efficiency of PTC systems. These strategies include applying advanced working fluids such as nanofluids [44], geometric modifications to the absorber [45], and the integration of turbulators in various configurations [46,47], and integrating porous materials [48,49]. Okonkwo et al. [50] numerically investigated the impact of different HTFs on the thermal efficiency of the LS-2 PTC. Among the three nanofluids analyzed, Al_2O_3 -Therminol VP-1 nanofluid achieved the greatest efficiency increase, reaching 0.22%, followed by CuO-Therminol VP-1 and Fe_3O_4 -Therminol VP-1, which improved efficiency by 0.18% and 0.15%, respectively. Bellos et al. [51] used EES-based model to examine the impact of various nanoparticles (CuO , Cu , Fe_2O_3 , SiO_2 , TiO_2 , and Al_2O_3) in Syltherm 800 under varying flow rates (50–300 L/min), inlet temperatures (300–650 K), and particle concentrations (up to 6%). Results showed that Cu-based nanofluid exhibited the highest efficiency improvement, followed by CuO , Fe_2O_3 , TiO_2 , Al_2O_3 , and SiO_2 . Allouhi et al. [52] proposed a mathematical formulation to evaluate the impact of different nanofluids (Al_2O_3 -based, CuO -based, and TiO_2 -based) on PTC performance under real operating

conditions. Results showed that nanoparticles enhance convective heat transfer, and 5% volumetric concentration of Al_2O_3 in Therminol VP-1 increased daily thermal energy gain by 1.46%. Bellos et al. [53] examined using mono and hybrid nanofluids with Syltherm 800 as the base fluid. Results showed improved thermal efficiency by up to 1.8% when applying hybrid nanofluids, outperforming mono nanofluids (0.7% increase) due to enhanced heat transfer. They used a validated EES-based thermal model under turbulence conditions. Mashhadian et al. [54] experimentally evaluated the impact of hybrid nanofluids (Al_2O_3 + MWCNTs in water) on the thermal performance with Three volume concentrations (0.01%, 0.02%, and 0.04%). At 0.04% hybrid nanofluid showed enhanced thermal efficiency by 21% in comparison to water with minimal pressure drop increases.

Many studies have centered around improving the thermal performance of PTCs by incorporating porous materials or adding turbulators with different designs such as discs, twisted tapes, and internal fins to increase flow turbulence. Reddy et al. [55] implemented numerical analysis in ANSYS FLUENT to investigate the influence of using porous inserts in PTC receiver. They examined the effects of fin porosity, aspect ratio, and thickness and under varying heat flux conditions. Results showed that porous inserts enhanced heat transfer by 17.5%, with a pressure drop of 2 kPa. Too et al. [56] explored the use of helically coiled wires, twisted tape inserts, and dimpled tube, and porous foam for pressurized air tubular absorbers under different operating conditions. The findings revealed that tubes with deeper inner-wall protrusions improved heat transfer performance while simultaneously reducing pressure losses compared to plain tubes and other heat transfer enhancement techniques. Yilmaz et al. [57] utilized numerical simulations to explore the use of wire coil inserts with various dimensions to boost the thermal behavior

of a PTC system. The study demonstrated that the heat transfer performance was enhanced by as much as 183%, accompanied by a thermal efficiency gain ranging from 0.4% to 1.4% at flow rates below 13 m³/h. Xiangtao et al. [58] did a numerical investigation on incorporating pin fin arrays within a PTC receiver to boost thermal performance. The findings revealed a 9% rise in the Nusselt number and a 12% improvement in the system's overall efficiency. However, the study also found that pin fin arrays are not suitable at high Reynolds numbers as they negatively affect overall efficiency.

Another common approach discussed in the literature is reshaping the absorber geometry to boost heat transfer and overall efficiency. Bellos et al. [59] introduced an absorber tube featuring sinusoidal dimpled geometry aimed at increasing the surface area of heat transfer and promoting flow-induced turbulence in PTC. The system was designed in SolidWorks and simulated using its integrated flow simulation module. Results showed that this geometry modification increased collector efficiency by 4.55%. Bitam et al. [60] performed three-dimensional simulations to examine the thermal behavior of a novel S-shaped/ sinusoidal tube for PTC. Their findings indicated that the sinusoidal design increases heat transfer by 45%–63%, with a maximum performance evaluation criterion (PEC) of 135%, while keeping thermal stresses lower by maintaining a circumferential temperature difference below 35K. Kaood et al. [61] introduced a conical solar receiver as an innovative method to enhance the thermal-hydraulic efficiency of PTCs. Simulations using ANSYS Fluent and TracePro analyzed diameter ratios (1.25–2), Reynolds numbers (20,000–100,000), and inlet temperatures (150°C–350°C). Findings revealed a 7%–65% heat transfer improvement and enhanced thermal and exergy efficiencies compared to a straight absorber. However, reducing the outlet diameter increased the friction coefficient.

CHAPTER 3

OPTICAL ANALYSIS OF PTC WITH FLAT TUBE

RECEIVER

3.1 Introduction

PTC systems focus solar irradiance on an absorber tube using a parabolic reflector. The heat is then transferred to an HTF, increasing its temperature. PTC systems operate at 200–400°C with water/steam for power generation [3] but can reach 550–600°C with molten salts [4]. To improve the feasibility of PTC technology, there is a need to reduce the cost either by increasing the overall efficiency or by minimizing the manufacturing costs.

While numerous studies have examined the optical behavior of PTC systems with conventional circular receivers and a few other configurations, the optical performance of PTC systems incorporating a flat-tube receiver has not yet been examined in published studies. Recent advancements in metal forming, 3D printing, and precision machining have made it significantly easier to manufacture absorbers with diverse shapes, including flat tubes, while maintaining consistent quality and durability.

In the current study, the optical behavior of a novel flat tube receiver, generated by stretching a circular shape in both horizontal and vertical orientations, was analyzed in detail using the MCRT method. In addition, the effect of different geometric factors - such as focal length, aperture width, and base circle diameter—on the optical behavior were further analyzed and discussed. Key findings from this work offer crucial understanding into how receiver shape and geometrical parameters influence the optical behavior of the

PTC system, thereby contributing valuable insights into optimizing the design of PTCs during preliminary design stages.

3.2 System model and methodology

The primary components of a traditional PTC are reflective parabolic mirror and a receiver assembly. The receiver consists of circular tube absorber and a glass cover surrounds the absorber. The solar irradiance comes from the sun, hits the reflector, and then is reflected to be focused onto the receiver.

In this work the optical behavior of PTC with horizontally and vertically oriented flat tube receiver was investigated. The flat tube's cross-section was generated by transforming the original circular cross-section of the absorber into a flat configuration using varying stretching ratios while fixing the cross-sectional area. The study explored how these geometric alterations influence the distribution of solar flux and the optical efficiency of the system, and the impact of different geometrical parameters was analyzed and discussed. The simulations were conducted using SolTrace software which is free software provided by National Renewable Energy Laboratory (NREL).

3.2.1 Physical model

Figure 1 (a) provides a 3D schematic of the PTC system equipped with a flat tube receiver, highlighting critical geometrical parameters. These parameters include the aperture width (W), glass cover outer and inner diameter (d_{go} , d_{gi}), the rim angle (ψ_{rim}), and the focal length (f), and the solar disk is assumed to have a radial angle (δ) of 4.65 mrad. The assembly consists of a metallic absorber surface treated with a low-emissivity, high-absorption coating and sealed under a glass enclosure. The annulus is evacuated to mitigate

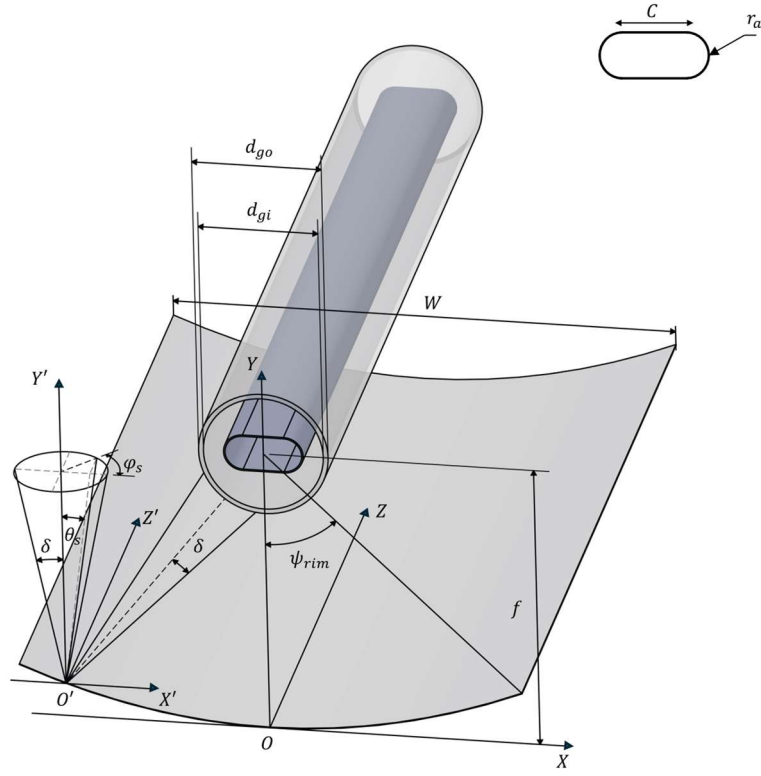
thermal losses. Figure 1 (b) and (c) represent 2D diagrams of horizontal and vertical oriented flat tubes respectively. In this study the LS-2 PTC module was adapted as a base module to simulate optical behavior. The optical and geometrical parameters of the adopted module are presented in Table 1.

The flat tube cross section can be viewed as a stretched circle with four parts, two half circles and two flat lines. The transformation of the circular shape to a flat tube shape, depicted for both horizontal and vertical orientations with different stretching ratios are shown in Figure 2. Eqs (**Error! Reference source not found.**) and (**Error! Reference source not found.**) were used to relate the circular cross section with the flat tube cross section.

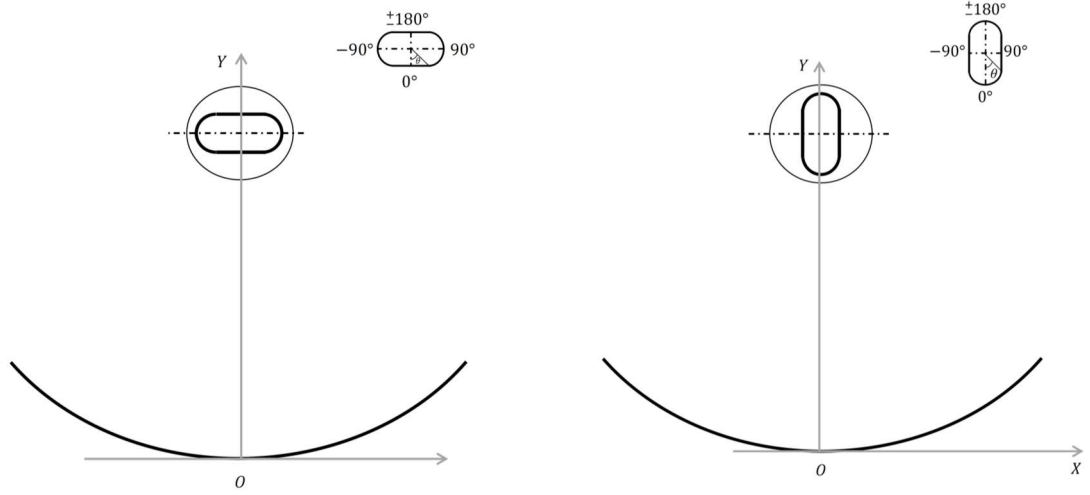
If the base circle diameter is d , and the cross-section area is fixed, then:

$$r_a = \frac{d}{2} \sqrt{\frac{\pi}{\pi + 4a}} \quad (1)$$

$$a = \frac{C}{2r_a} \quad (2)$$



(a) 3D diagram of PTC equipped with flat tube receiver



(b) cross-sectional view of PTC equipped with horizontal tube

(b) cross-sectional view of PTC equipped with vertical tube

Figure 1 Schematic diagram of a PTC module with flat tube receiver

Table 1 Geometrical and physical properties of LS-2 PTC module

Collector length (L_a)	7.8 m
Collector aperture width (W)	5 m
Focal length (f)	1.84 m
Glass cover outer diameter (d_{go})	0.115 m
Glass cover inner diameter (d_{gi})	0.109 m
Circular absorber diameter (d)	0.07 m
Mirror reflectivity (ρ_r)	0.93
Absorber absorptivity (α_a)	0.96
Glass cover transmittance (τ_a)	0.95
Glass cover refractive index	1.5

Where r_a is the radius of the two half circles, a is the stretching ratio, C is the distance shown in Figure 1.

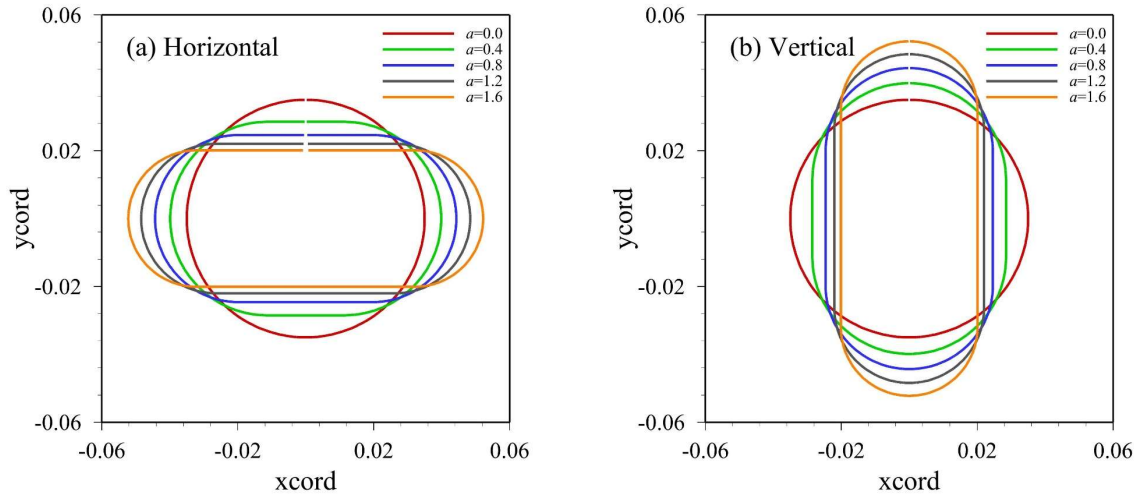


Figure 2 The absorber geometry with different stretching ratios (a) horizontal stretching (b) vertical stretching

3.2.2 Monte Carlo Ray Tracing (MCRT) model

MCRT is an advanced computational technique that combines the Monte Carlo Method (MCM), which utilizes random sampling and statistical analysis, with Ray Tracing (RT), which operates on the principles of geometric optics. This integration allows MCRT to model complex light interactions in optical systems with high precision.

Initially, the MCRT procedure begins by assigning initial spatial location and direction for each ray above the aperture plane based on a probability distribution function. The optical behaviors at each interface are then specified using a sequence of random numbers that are distributed evenly. Based on these computations, all rays are traced within the PTC system. After that, the intersections are determined and counted, and the *LCR* profile is determined using the number of ray intersections with each segment on the absorber surface.

As the central aim of this study is to investigate flat tube geometry, several assumptions have been adopted in optical modelling. These assumptions are as follows: (1) the solar disk was assumed to have a uniform brightness (pillbox distribution). (2) the effect of optical errors like slope and installation errors were ignored. (3) The receiver is perfectly aligned with the focal position along its length. (4) Solar radiation was assumed to be incident at an angle of zero degrees with an intensity of 1000 W/m^2 .

As can be seen from Figure 1 there are two coordinate systems, the global OXYZ coordinate system representing the whole system, and the O'X'Y'Z' coordinate system associated with the incident ray. The initial hitting point (x_0, y_0, z_0) is given by Eq.(3).

$$\begin{cases} x_0 = W\kappa_1 - W/2 \\ y_0 = \frac{x_0^2}{4f} \\ z_0 = L_a\kappa_2 \end{cases} \quad (3)$$

κ_1 and κ_2 are randomly generated numbers with values between 0 and 1, L_a is the absorber length, W is the aperture width of the parabola.

The initial position of the ray within the solar disc is determined randomly in the coordinate system (O'X'Y'Z') by Eq. (4).

$$\begin{cases} x' = l' \sin \theta_s \cos \varphi_s \\ y' = l' \cos \theta_s \\ z' = l' \sin \theta_s \sin \varphi_s \end{cases} \quad (4)$$

l' represents the distance (PO), θ_s and φ_s are radial angle and circumferential angle, and they are determined using Eqs. (5) and (6) respectively.

$$\theta_s = \arctan \left(\sqrt{\kappa_3 \tan \delta} \right) \quad (5)$$

$$\varphi_s = 2\pi\kappa_4 \quad (6)$$

κ_3 and κ_4 are randomly generated numbers with values between 0 and 1, δ is the sun radial angle (a value of 4.65 mrad is taken [17]). The incident vector in (O'X'Y'Z') can be represented by Eq. (7).

$$\vec{S}' = (\sin \theta_s \cos \varphi_s, -\cos \theta_s, \sin \theta_s \sin \varphi_s) \quad (7)$$

Since it is known that the coordinates transformation doesn't affect the incident unit vector so the unit vector in (OXYZ) system will be the same, which is given by Eq. (8).

$$\vec{S} = \vec{S'} \quad (8)$$

The parabolic reflector equation can be expressed as Eq. (9).

$$x^2 = 4fy \quad (9)$$

The normal unit vectors, of the inner glass cover and the parabolic reflector are $\vec{n}_1$ and $\vec{n}_0$ are calculated by Eqs. (10) and (11) respectively.

$$\vec{n}_1 = \left(-\frac{2x}{d_{gi}}, -\frac{2(y-f)}{d_{gi}}, 0 \right) \quad (10)$$

$$\vec{n}_0 = \left(-\frac{x}{\sqrt{x^2+4f^2}}, \frac{2f}{\sqrt{x^2+4f^2}}, 0 \right) \quad (11)$$

The reflected unit vector ($\vec{r}$) can be found using Eq. (12).

$$\vec{r} = \vec{\lambda} - 2(\vec{\lambda} \cdot \vec{n})\vec{n} \quad (12)$$

Where $\vec{\lambda}$ is the incident vector and $\vec{n}$ is the normal unit vector of the surface. The transmitted unit vector can be calculated by Eqs. (13) and (14).

$$\vec{t} = \sqrt{1-i^2[1-(\vec{n} \cdot \vec{\lambda})^2]}\vec{n} + i[\vec{\lambda} - (\vec{n} \cdot \vec{\lambda})\vec{n}] \quad (13)$$

$$i = \frac{\text{refractive index of medium1}}{\text{refractive index of medium2}} \quad (14)$$

$\vec{t}$ is the transmitted unit vector and i is the refractive indices ratio.

The algorithms that govern the MCRT process begin by determining a random point of incidence within the aperture, using randomly generated numbers (κ_1, κ_2). These values are generated using a pseudo-random number generator that produces numbers within the interval [0,1], allowing for statistically accurate sampling of the spatial and angular distributions [62]. Subsequently, the starting position of each ray within the sun disc above the aperture plane is assigned based on additional generated random values (κ_3, κ_4) ensuring accurate representation of the sun's angular spread. When a ray intersects with a surface, optical behavior occurs. This optical behavior is evaluated using randomly generated numbers. For instance, when a ray encounters the reflector's surface, a random value κ_5 is produced and checked against the reflectivity of the reflector (ρ_r). If κ_5 is less than the reflectivity, the ray is reflected; otherwise, it is absorbed and abandoned from further tracing. The algorithm of MCRT logic is shown in Figure 3.

3.3 Optical analysis

The outer surface of the absorber is discretized into multiple segments as shown in Figure 4. the number of hitting rays into each segment is counted from MCRT procedure. Then local flux intensity is determined as the product of the ray count and the power of each ray as Eq.(15).

$$I_i = \text{power of one ray} \times \text{number of hitting rays} \quad (15)$$

The power per one ray is determined by dividing the total incident on the aperture area power by the total simulated number of rays as Eq.(16).

$$power\ of\ one\ ray = \frac{DNI \cdot A_a}{total\ number\ of\ rays} \quad (16)$$

DNI is selected to be 1000 W/m^2 in the current study. A_a represents the aperture area, which is the projected surface area through which solar radiation is collected and directed towards the absorber, and it is expressed by Eq.(17).

$$A_a = W \cdot L_a \quad (17)$$

The key parameter for assessing optical performance is the LCR . It quantifies the level of concentrated flux in a particular segment (I_i) relative to the effective solar irradiance reaching the

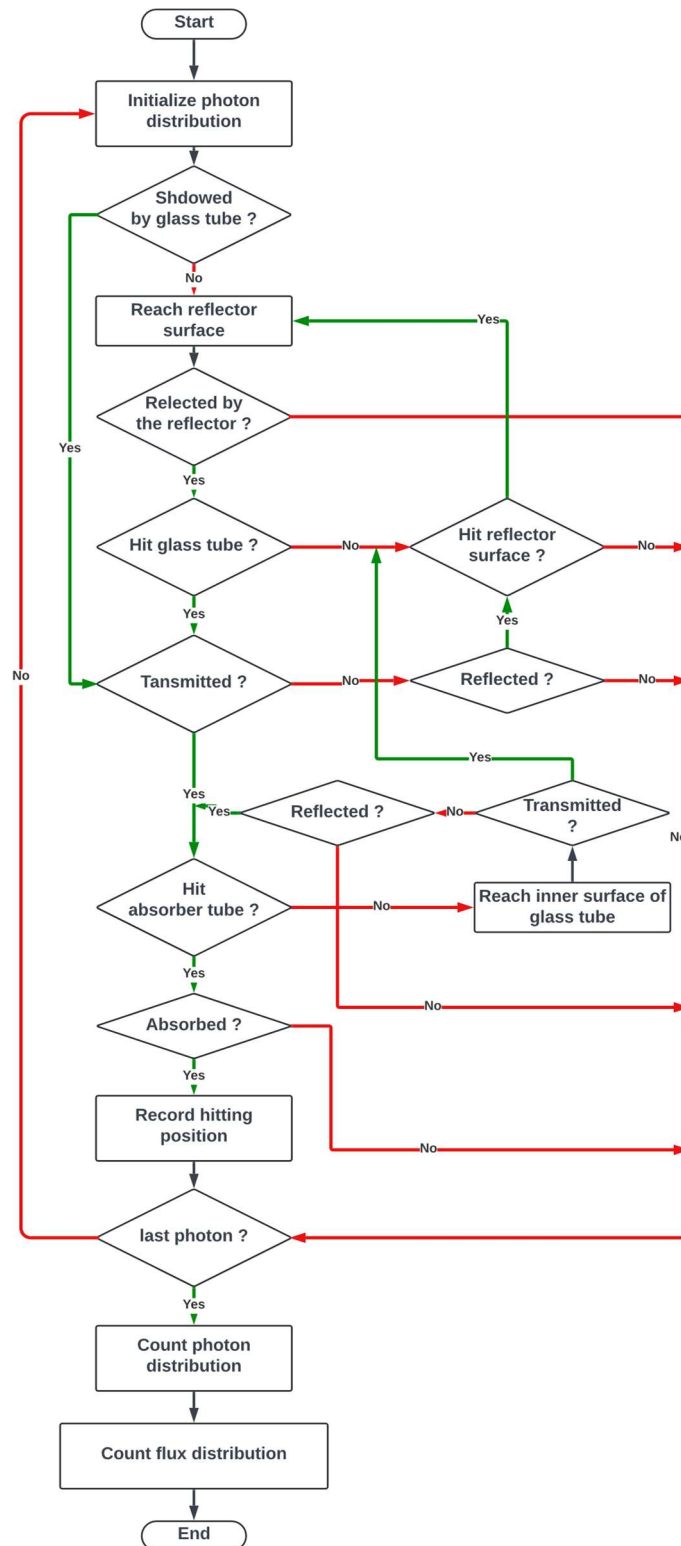


Figure 3 Process flow of the MCRT technique for computing the flux distribution.

aperture (I_{eff}) and it is calculated by Eq. (18).

$$LCR = \frac{I_i}{I_{eff}} \quad (18)$$

In the present study, the effective solar irradiance (I_{eff}) is equivalent to the DNI , with a value of 1000 W/m².

Optical efficiency (η_o) refers to the ratio of total energy absorbed to the total incident radiation, and it is expressed by Eq. (19).

$$\eta_o = \frac{\sum_{i=1}^{N_p} I_i \cdot A_i}{I_{eff} \cdot A_a} \quad (19)$$

A_i is the area of each segment, and it expressed by Eq.(20).

$$A_i = x_i \cdot L_a \quad (20)$$

x_i is the segment angular length along the circumference as shown in Figure 4. The circumference of the absorber is discretized into 360 segments based on a circular cross-section, where each segment corresponds to a 1-degree angular division. For flat tube cross-section, the segmentation is adapted to provide an equivalent division, ensuring a consistent resolution across different geometries.

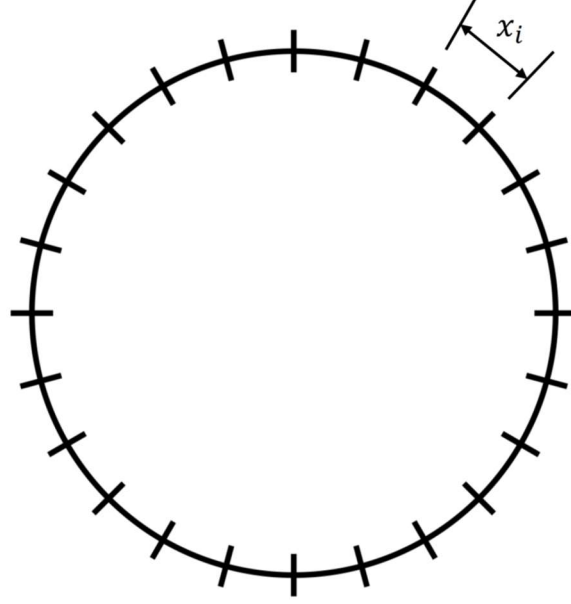


Figure 4 Segmentation of the circular absorber's circumference into discrete sections

The intercept factor (γ) is another highly significant parameter. It represents the fraction of radiation captured by the absorber relative to the total irradiation incident on the aperture area. The key difference between the intercept factor and optical efficiency is that the intercept factor quantifies the fraction of rays that reach the absorber while considering only geometric and optical errors affecting ray interception, regardless of reflectivity, transmissivity, and absorptivity. It specifically measures the absorber's ability to intercept all the rays within the PTC system. In contrast, optical efficiency incorporates these additional losses, providing a broader assessment of system performance. The intercept factor (γ) is determined using Eq. (21).

$$\gamma = \frac{\eta_o}{\rho_r \tau_g \alpha_a} \quad (21)$$

To further examine the features of the energy flux profile, both the average energy concentration (LCR_{ave}) and the distribution non-uniformity (ε) are analyzed, the LCR_{ave} is evaluated using Eq. (22).

$$LCR_{ave} = \frac{\sum_{i=1}^{N_n} LCR_i}{N_n} \quad (22)$$

The non-uniformity (ε) of the flux is calculated using Relative Mean Absolute Deviation (RMAD) approach as shown in Eq.(23) [33,63].

$$\varepsilon = \frac{\sum_{i=1}^{N_n} |LCR_i - LCR_{ave}|}{N_n \cdot LCR_{ave}} \quad (23)$$

3.4 Model validation

3.4.1 Suitable number of rays and grid

In general, the accuracy of MCRT is significantly influenced by both the number of rays traced and the grid size[33,64]. Increasing the number of rays leads to more accurate results as it increases statistical reliability by increasing the number of samples. However, this also increases computational capability and time. The choice of grid size affects spatial resolution; finer grids can capture more detailed features but may introduce numerical errors if not adequately sampled with sufficient rays. According to Zhao et al.[64] for a circular tube with 180 grid divisions were optimal when the number of rays was below 10^8 . In this current work, a ray count of 3.5×10^7 and a grid count of 360 were adopted for accuracy and operational considerations.

3.4.2 A comparison between an in-house Matlab Code and SolTrace

For validation purposes, A MATLAB-based model was initially implemented to simulate the optical behavior of the PTC without considering the refraction effect of the glass cover. The code incorporated factors such as sunshape, reflectivity, transmissivity and absorber geometry. However, complexities related to the glass cover required the use of SolTrace for further simulations. The results obtained using Matlab were compared with that of SolTrace and showed excellent agreement as illustrated in Figure 5.

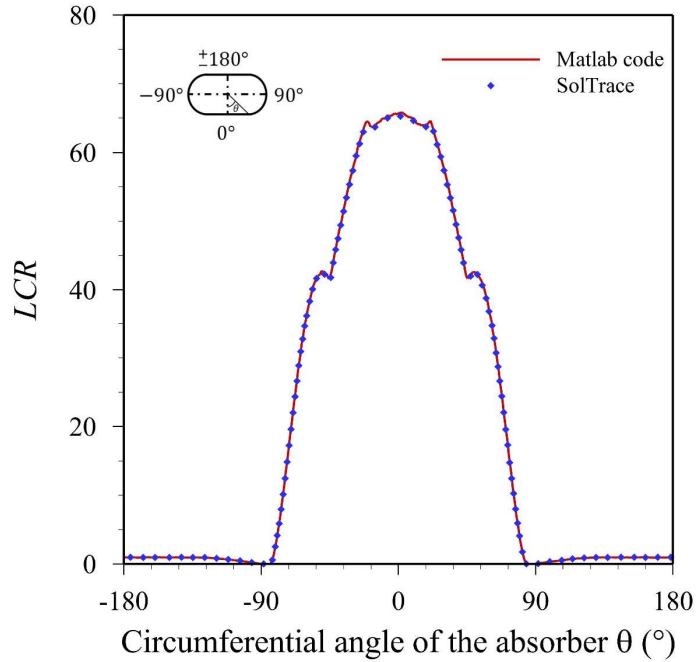


Figure 5 A comparison between the results obtained from Matlab code and SolTrace results for horizontal flat tube with $d = 0.07$ m, $a = 1$, $W = 5$ m, and $f = 1.84$ m.

3.4.3 Model Validation

As no previous optical analysis has been conducted for a PTC with a flat tube absorber, the model was initially developed to simulate a circular absorber. The LCR distribution results obtained from the simulation were then compared with Jeter's[17] analytical model

findings, demonstrating very good agreement, as shown in Figure 6, thereby validating the model's accuracy. The model used for the validation considers ideal optics with a sun radial angle of 7.5 mrad, geometric concentration ratio of 20, and rim angle of 90° [17].

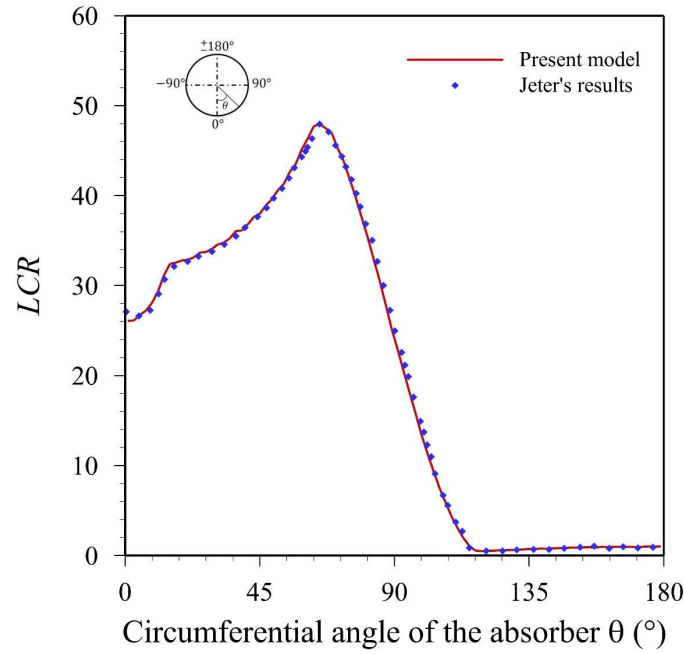


Figure 6 Results comparison of the present model and Jeter's findings[17].

3.5 Results and discussion

The current section provides an evaluation of the main performance metrics of the current study, including local concentration ratio (LCR) profile, optical efficiency, intercept factor and flux non-uniformity. The effect of stretching ratio on these metrics is initially presented and discussed for both horizontal and vertical stretching at the base receiver diameter of the original SEGS LS-2 module (70 mm). Furthermore, the impact of various geometrical parameters including aperture width, focal length, and base circle diameter is discussed for horizontal and vertical flat tube configurations considering a stretching ratio of 1 in all cases.

Another study to assess the effect of stretching ratio at small base circle diameters on the optical behavior of horizontal flat tube configuration is presented. This study aims to examine the transition from circular tube to flat tube geometry at small base circle diameters. The results of these studies highlight the differences in optical behavior between the conventional circular configuration and flat tube configurations, demonstrating the effect of horizontal and vertical stretching. The results also emphasize how geometric transformations influence flux distribution on the absorber surface. These insights provide a robust foundation for further investigation into optical-to-thermal energy conversion and optimizing the design and efficiency of PTC systems.

The simulation conducted for the flat tube in this study considered two scenarios based on the relation between the flat tube shape and the original circular shape: one assuming constant cross-sectional area as mentioned before and the other assuming constant perimeter. The results of the LCR distribution for different stretching ratios, as illustrated in Figure 7, demonstrate nearly identical behavior for both scenarios, suggesting that either assumption provides comparable outcomes under the investigated conditions. Therefore, the present work focuses on the constant cross-sectional area scenario. The reason behind using a constant cross-sectional area as the basis choice lies in the fact that maintaining a constant cross-sectional area ensures a constant flow rate, making it a reliable foundation for evaluating thermal performance.

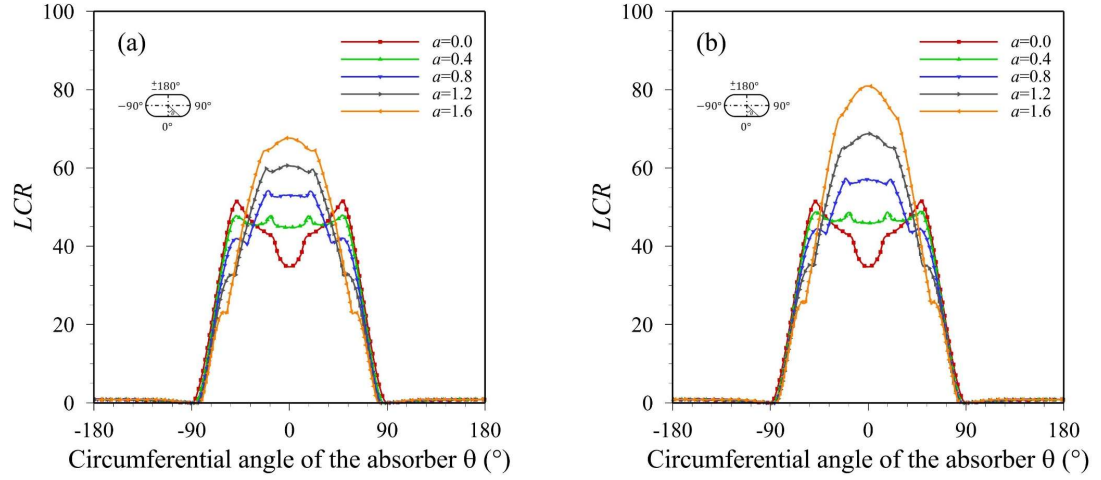


Figure 7 LCR distribution versus absorber angle for horizontal flat tube with different stretching ratios at $W = 5$ m, $d = 70$ mm and $f = 1.84$ m assuming (a) constant cross-sectional area with the base circle (b) constant perimeter with the base circle.

3.5.1 Effect of stretching ratio

Figure 8 (a) and (b) present the LCR distribution as a function of the absorber angle for different stretching ratios (a) ranging from 0 to 1.6 for a horizontal and vertical flat tube respectively. It is evident that as a increases, the span of the flat sections expands, while the extent of the curved regions diminishes. The flux in the lower flat surface increases as a progresses from a circular shape ($a = 0$) to more elongated flat tube geometries ($a > 0$). In contrast, the flux in the curved regions decreases with increasing a . The flux on the upper flat surface remains approximately constant, with an LCR value of around 1. A similar behavior is noticed in the vertical tube case, in this configuration, the increase in flux within the flat regions is less pronounced compared to the horizontal case. In the vertical orientation, the LCR values demonstrate an overall reduction as a increases.

The underlying reason for the observed behavior is the redistribution of concentrated energy across the absorber surface as the geometry is progressively stretched. As a

increases, the absorber undergoes a transformation from an initially circular shape to a more elongated, flattened geometry. This transformation results in enlarging the surface area of the flat regions, thereby extending their flux-absorbing capacity while reducing the span of the curved regions, which continues to be subjected to concentrated beam radiation. The increase in flux within the flat regions can be attributed to their proximity to the focal point of the parabolic reflector. As the absorber flattens, the flat surfaces approach the focal point, leading to higher concentrations of solar radiation. Conversely, in the curved regions, the reduced curvature and increased distance from the focal point result in a lower concentration of reflected rays, hence reducing the flux in these areas. In horizontal orientation the lower part which is subjected to high concentration is flat. This configuration results in higher LCR values compared to the vertical tube, where a curved region occupies the bottom region. The upper part consistently maintains an LCR value of approximately 1, as it predominantly receives direct normal solar radiation.

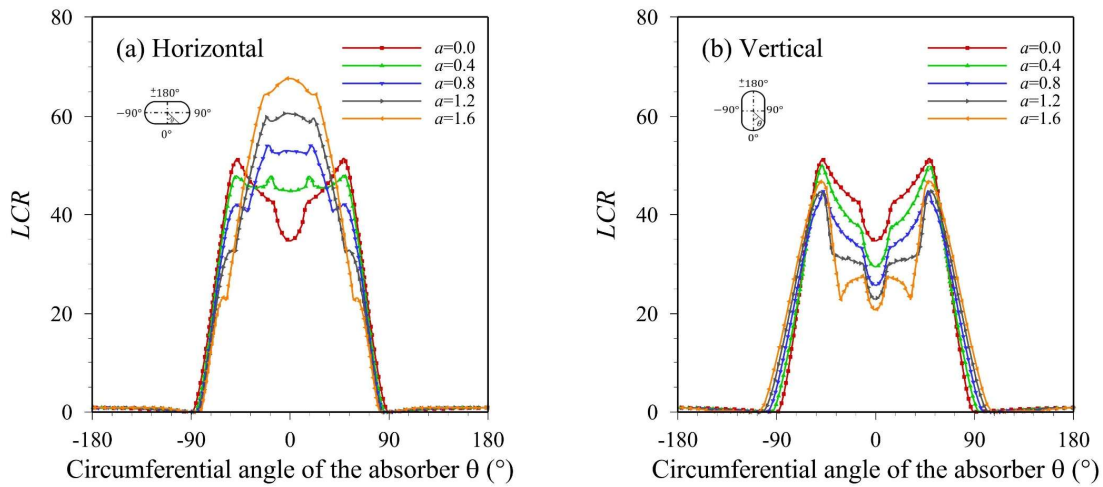


Figure 8 Effects of stretching ratio on LCR distribution at $W = 5$ m, $d = 70$ mm and $f = 1.84$ m for (a) Horizontal tube (b) Vertical tube

Figure 9 illustrate the variations of the optical efficiency (η_0) and intercept factor (γ) with a for horizontal and vertical configurations. Since the intercept factor (γ) is essentially the optical efficiency (η_0) divided by a constant, as illustrated in Eq.(21), it can be represented on the same figure by utilizing a secondary y-axis. This is achieved by multiplying the optical efficiency axis by a factor, axis by factor, as shown in Eq.(21). It can be observed that both η_0 and γ remain approximately constant for both the horizontal and vertical configurations across the range of stretching factors with only very slight variations. For the horizontal tube configuration, a very slight increasing trend is evident, with η_0 and γ rising from 77.69% and 0.916, respectively, for the fully circular case ($a = 0$) to peak values of 78.21% and 0.922 at $a = 1.6$. Conversely, the vertical tube configuration exhibits a minor decreasing trend, with η_0 and γ decreasing from their initial values at $a = 0$ to 77.13% and 0.909, respectively, at $a = 1.6$.

The observed trends are primarily attributed to changes in absorber geometry and their impact on the interaction between solar radiation and the absorber surface. As a increases, the upper surface of the absorber in the horizontal configuration becomes wider, allowing for increased interception of direct solar radiation. This increased interception minimizes the optical losses associated with non-shadowed rays, thereby slightly improving η_0 and γ . In contrast, for the vertical tube configuration, the upper surface becomes narrower with increasing a , resulting in fewer intercepted rays and a slight reduction in η_0 and γ . In other words, if the absorber were a flat plate completely shading the reflector from incoming solar radiation, its efficiency would be largely determined by the absorptivity of the plate. The differences in performance between the two configurations can also be attributed to the behavior of the horizontal gap between the absorber and the glass cover. In the vertical

configuration, this gap increases with a , whereas in the horizontal configuration, the gap decreases. An increased gap in the vertical configuration leads to a higher probability of rays escaping from the PTC system, thereby increasing energy losses and contributing to the reduction in η_0 and γ .

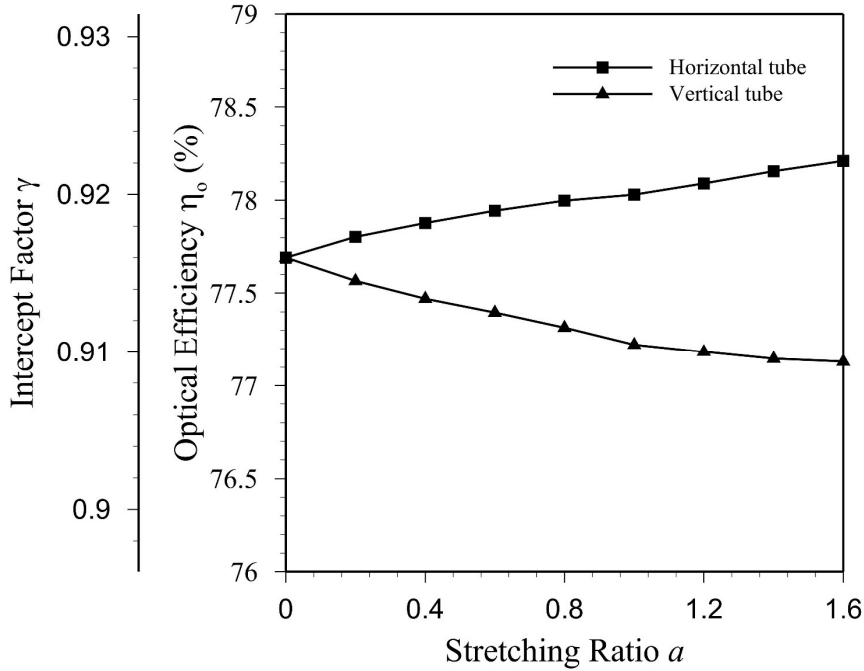


Figure 9 Effects of stretching ratio on optical efficiency and intercept factor at $W = 5\text{m}$, $d = 70\text{ mm}$ and $f = 1.84\text{ m}$ for horizontal and vertical flat tubes

Figure 10 (a) and (b) illustrate the variations of the maximum and the average local concentration ratio (LCR_{max} , LCR_{ave}) and the flux non-uniformity (ε) with a for horizontal and vertical configurations respectively. In both orientations, the LCR_{ave} registers a slight decrease as a increases. For the horizontal orientation, ε shows an increasing trend, rising from 1.0854 at $a = 0$ to 1.1703 at $a = 1.6$. In contrast, the vertical orientation exhibits a decreasing trend in ε , dropping from 1.0854 at $a = 0$ to 0.9565 at $a = 1.6$. The LCR_{max} displays both decreasing and increasing trends in both orientations, with overall values being consistently higher in the horizontal configuration compared to the vertical one.

LCR_{ave} decreases slightly in both cases by 13.1%. This decrease in LCR_{ave} can be attributed to the reduction in the angular span of the concentrated beam that results from the stretching. The behavior of LCR_{max} is more complex, showing an initial decrease followed by an increase. Initially, the reduction in LCR_{max} can be explained by the fact that the flat part of the absorber does not approach the focal point closely enough to achieve a concentration higher than that of the remaining sections. However, beyond a certain value of a , the flux in the flat region begins to increase, which contributes to an increase in the maximum flux. In the horizontal orientation, the subsequent increase in LCR_{max} is more pronounced because the flat surface is positioned at the bottom, corresponding to regions with high concentration. In contrast, in the vertical orientation, the lower part does not benefit from the same level of concentrated radiation, leading to a less pronounced increase. The increasing trend of ε for the horizontal configuration is primarily due to the significant increase in peak LCR values at the bottom region of the absorber, which results in a higher level of non-uniform flux distribution. Conversely, in the vertical configuration, non-uniformity decreases as the distance between the bottom region and the focal point increases with a , thereby resulting in less concentration and less non-uniformity.

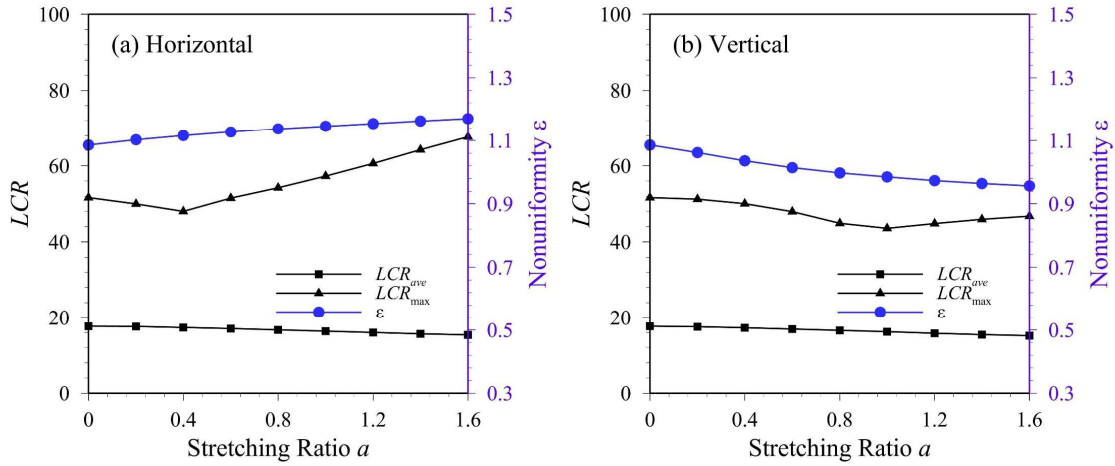


Figure 10 Variations of LCR_{max} , LCR_{ave} , and ϵ with the stretching ratio at $W = 5$ m, $d = 70$ mm and $f = 1.84$ m for (a) Horizontal tube (b) Vertical tube

3.5.2 Effect of aperture width

Figure 11 (a) and (b) depict the LCR distribution for various values of aperture width (W) ranging from 3 to 14 for horizontal and vertical tube configurations, respectively. In this study the stretching ratio is fixed at 1, and the other parameters of the PTC module held constant. It is observed that there exists a region in which the LCR values remain constant across the different aperture widths. The angular span of this constant region is 20.5° for the horizontal tube and 30.5° for the vertical tube. In both horizontal and vertical configurations, the region subjected to direct insolation decreases as W increases, while the remaining regions exhibit an increase. Moreover, the maximum LCR values are higher in the vertical tube compared to the horizontal tube.

The presence of a constant LCR region can be explained by the findings of [33], who demonstrated theoretically that for a circular absorber, there is a specific calculated value (abscissa) that defines a region of the parabolic reflector which contributes exclusively to

concentrating flux at the bottom of the absorber. This phenomenon results in a consistent LCR across a fixed angular span, regardless of changes in aperture width.

The reduction in the direct insolation region with increasing W is mainly linked to the increase in the angular span of the concentrated beam, which grows as W increases. This expanded span of concentrated radiation reduces the relative contribution from direct insolation. In the vertical tube configuration, the LCR values are generally higher due to the increased span causing greater concentration in the flat regions that are closer to the focal point of the reflector, as previously discussed. The closer proximity of these regions to the focal point enhances their ability to absorb concentrated flux, thereby increasing the overall LCR in these sections.

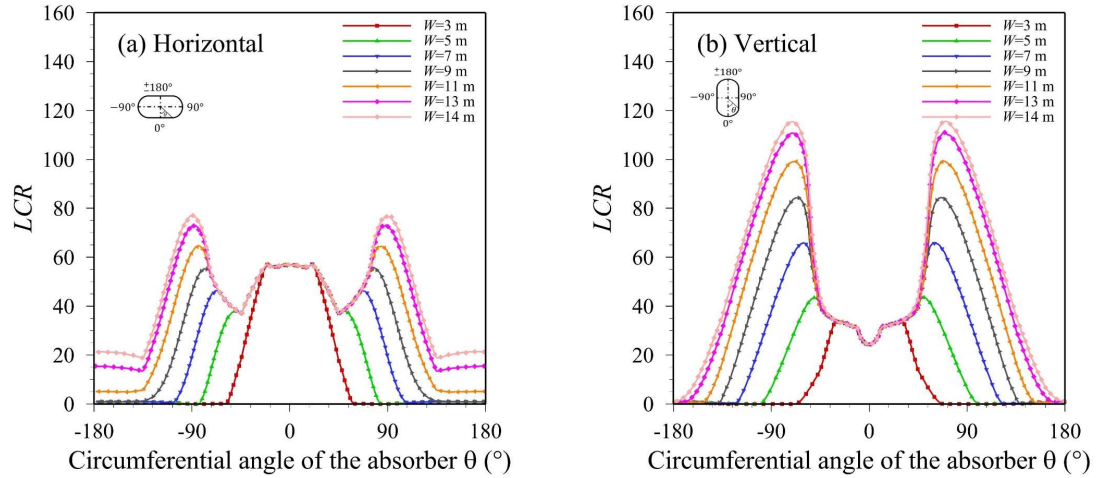


Figure 11 Effects of aperture width on LCR distribution at $d = 70$ mm, $f = 1.84$ m, and $a = 1$ for (a) Horizontal tube (b) Vertical tube

Figure 12 illustrate the variations of η_0 and γ with W for horizontal and vertical configurations. It is observed that the variations in η_0 and γ with W are minimal for the horizontal tube, showing a slight increase between 3 and 5 m, followed by a declining trend. In contrast, the variation is more pronounced for the vertical tube, where η_0 and γ

increase by approximately 1% as W grows from 3 to 9 m, after which they remain nearly constant.

One contributing factor to this behavior is the span of the direct normal radiation region, which increases optical losses with increasing width. Another factor is the scattering effect of the glass cover. In the vertical tube configuration, the improvement can be attributed to a reduced scattering effect as W increases, which may not be as effective in the horizontal configuration.

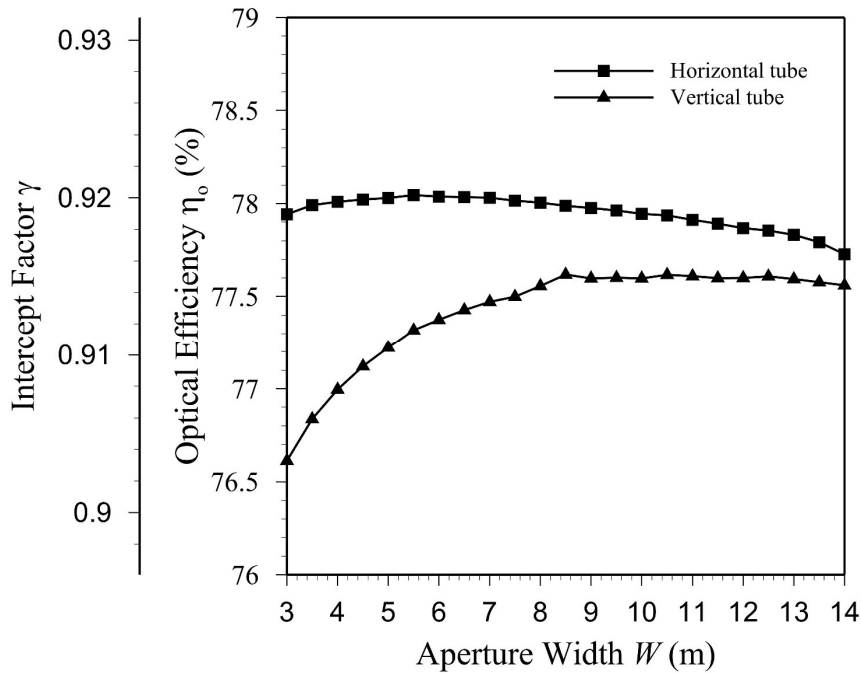


Figure 12 Effects of aperture width on optical efficiency and intercept factor at $d = 70$ mm, $f = 1.84$ m, and $a = 1$ for horizontal and vertical flat tubes

Figure 13 (a) and (b) represent the variations of LCR_{max} , LCR_{ave} , and ϵ with W for horizontal and vertical configurations, respectively. From the figures LCR_{ave} increases with W in both cases, whereas ϵ shows a significant decrease with increasing W . LCR_{max} recorded nearly constant value in horizontal tube in the range between 3 and 9, and after that it shows a

considerable linear increase. LCR_{max} for vertical tube shows a dramatic increase with increasing W compared to the horizontal tube. The increase in LCR_{ave} is mainly due to the increase in concentrated flux because of the added reflective surface with increasing W . The behavior of epsilon can be attributed to the increase in the angle span of concentrated beam which will cause less non-uniformity. The constant value of LCR_{max} in horizontal tube in the range between 3 and 9 aperture width is primarily because LCR_{max} is in the lower flat part within the constant region of LCR , and further increase happens because of the growth in concentration. in contrast, the vertical tube which has maximum values of LCR out in the flat surfaces outside the constant LCR region.

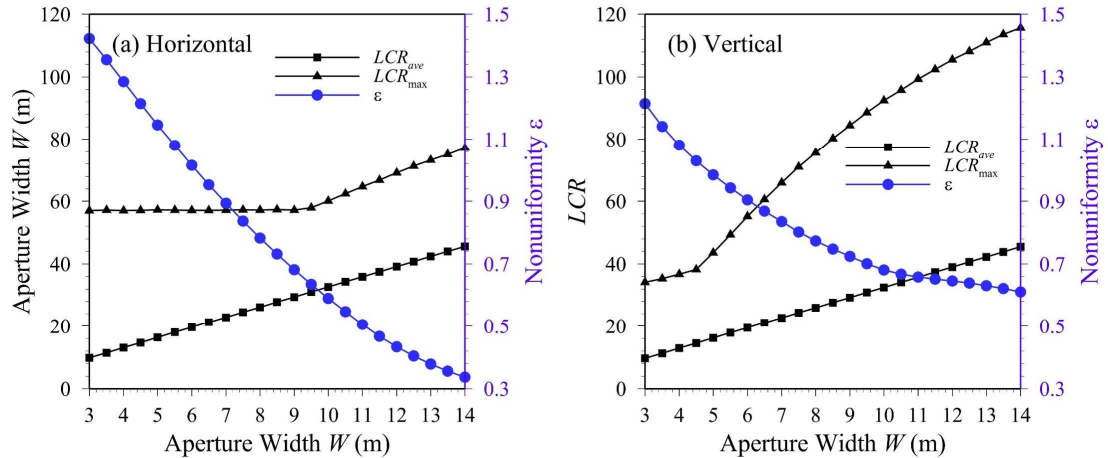


Figure 13 Variations of LCR_{max} , LCR_{ave} , and ϵ with aperture width at $d = 70$ mm, $f = 1.84$ m, and $a = 1$ for (a) Horizontal tube (b) Vertical tube

3.5.3 Effect of focal length

Figure 14 (a) and (b) illustrate the LCR distribution with various focal lengths (f) ranging from 0.2 to 3.2 for horizontal and vertical configurations, respectively. For both horizontal and vertical tubes, the flux in the bottom region of the absorber increases as f increases. At $f = 0.2$, a region of nearly zero concentration is observed for both cases, with no area

receiving direct insolation. The direct insolation region starts to emerge and expands as the focal length exceeds $f = 0.2$. For the horizontal tube, the LCR distribution shows only a single peak when $f > 1.4$, whereas for all focal length values, the vertical tube exhibits two peaks in the LCR distribution.

These phenomena can be attributed to changes in the PTC configuration as f increases. With increasing f , the configuration causes concentrated solar rays to converge more directly towards the bottom of the absorber. A larger focal length results in a narrower angular span of concentrated rays, leading to a more focused concentration of energy towards the lower portion of the absorber. This effect explains the increase in flux at the bottom region and the differences in LCR peak behavior between the horizontal and vertical configurations.

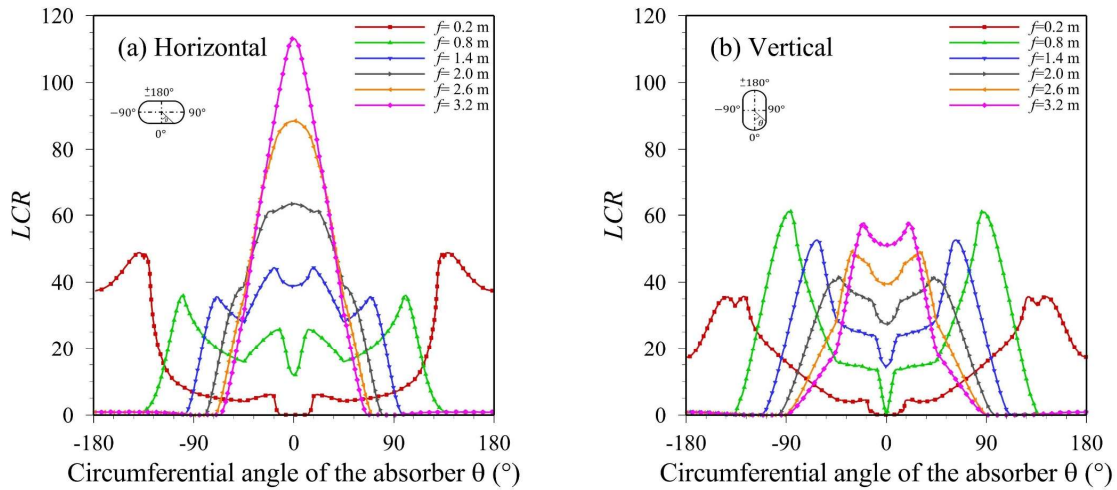


Figure 14 Effects of focal length on LCR distribution at $d = 70$ mm, $W = 5$ m, and $a = 1$ for (a) Horizontal tube (b) vertical tube

Figure 15 depict the variations of η_0 and γ as a function of f for horizontal and vertical configurations, respectively. It is observed that both η_0 and γ show a slight increase between

$f = 0.2$ and $f = 0.8$, followed by a minimal decreasing trend. Additionally, the vertical tube configuration registers higher values for η_0 and γ compared to the horizontal tube.

This behavior can be attributed to a phenomenon known as the "rays escaping effect." At both very low and very high focal lengths, the PTC system geometry reduces the effective area of the absorber exposed to reflected solar rays from the parabolic mirror. This occurs because the absorber fails to fully intercept the entire reflected light cone, resulting in some rays missing the absorber and escaping. This leads to increased optical losses, as fewer rays are concentrated on the absorber surface, thereby reducing the system's overall optical efficiency.

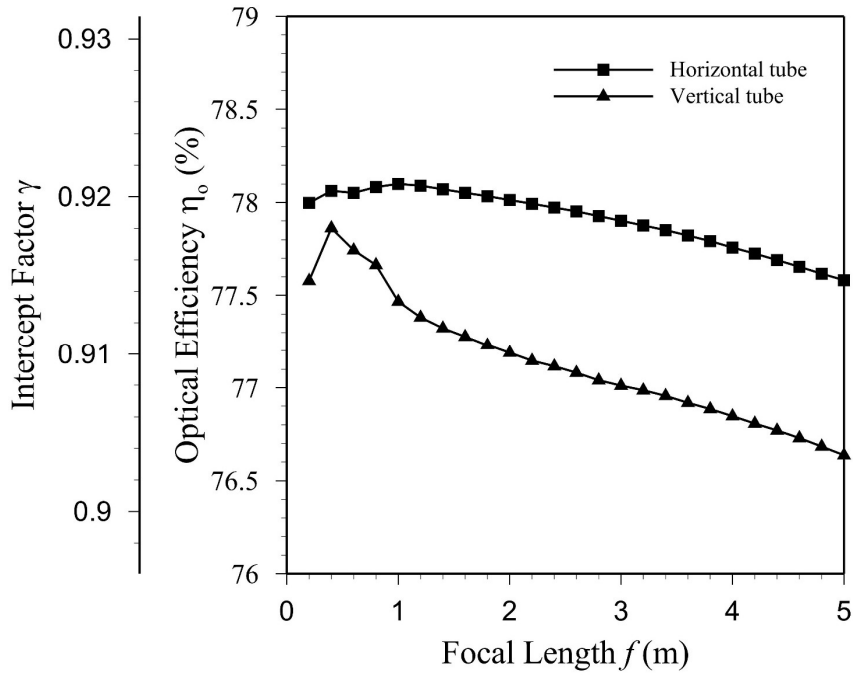


Figure 15 Effects of focal length on optical efficiency and intercept factor at $d = 7\text{mm}$, $W = 5\text{ m}$, and $a = 1$ for horizontal and vertical flat tubes

Figure 16 (a) and (b) depict LCR_{max} , LCR_{ave} , and ε as a function of f for horizontal and vertical configurations respectively. LCR_{ave} remains almost constant for both orientations

because of the nearly constant efficiency. LCR_{ave} remains nearly constant for both horizontal and vertical orientations due to the almost constant efficiency within the range of focal lengths. For the horizontal tube, LCR_{max} initially decreases, followed by a sharp increase, peaking at $f = 4$, after which it stabilizes. In contrast, LCR_{max} for the vertical tube first increases, then decreases, and finally rises again as f increases. ε in the horizontal tube initially decreases, then sharply increases to a peak value of 1.4 at $f = 3.4$, followed by a slight decrease. In the vertical tube, ε shows a consistent increasing trend with f .

The behavior of these metrics can be attributed to the effective span of the concentrated beam. In the horizontal tube, when $f < 0.6$, the span of the concentrated beam increases with f , leading to reduced non-uniformity. Afterward, the span decreases and becomes approximately constant at $f = 4$, resulting in minimal changes in ε . This behavior is not observed in the vertical tube, where the span of the concentrated beam decreases throughout the range, as noticed in Figure 14.

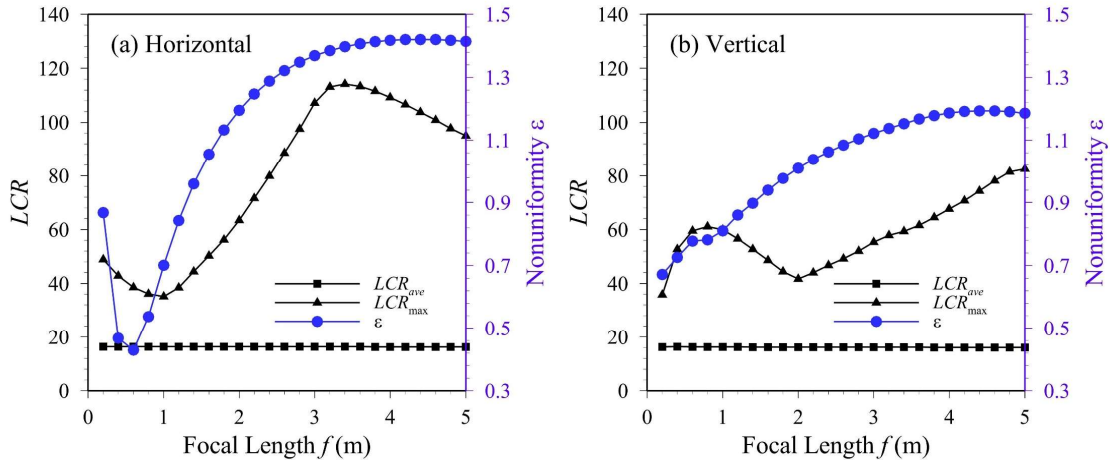


Figure 16 Variations of LCR_{max} , LCR_{ave} , and ε with focal length at $d = 70$ mm, $W = 5$ m, and $a = 1$ for (a) Horizontal tube (b) vertical tube

3.5.4 Effect of base circle diameter

Figure 17 (a) and (b) illustrate the effect of varying the base circle diameter (d) on the LCR distribution for horizontal and vertical configurations, respectively. It is observed that the flux in the bottom region increases as d increases in both horizontal and vertical tube configurations. Additionally, the area of direct insolation increases with increasing d .

The increase in flux in the bottom region can be attributed to the closer proximity of the absorber's bottom surface to the focal point as the diameter decreases. At smaller diameters, the bottom of the absorber can intercept concentrated solar rays from all points along the parabolic reflector, resulting in a single LCR peak at the absorber's bottom position. However, as the absorber diameter exceeds this threshold, not all points on the absorber can receive radiation from every point on the reflector, leading to a more distributed or multi-peak LCR pattern, as described by [33]. This phenomenon manifests more rapidly in the flat regions (horizontal tube) compared to the curved regions (vertical tube).

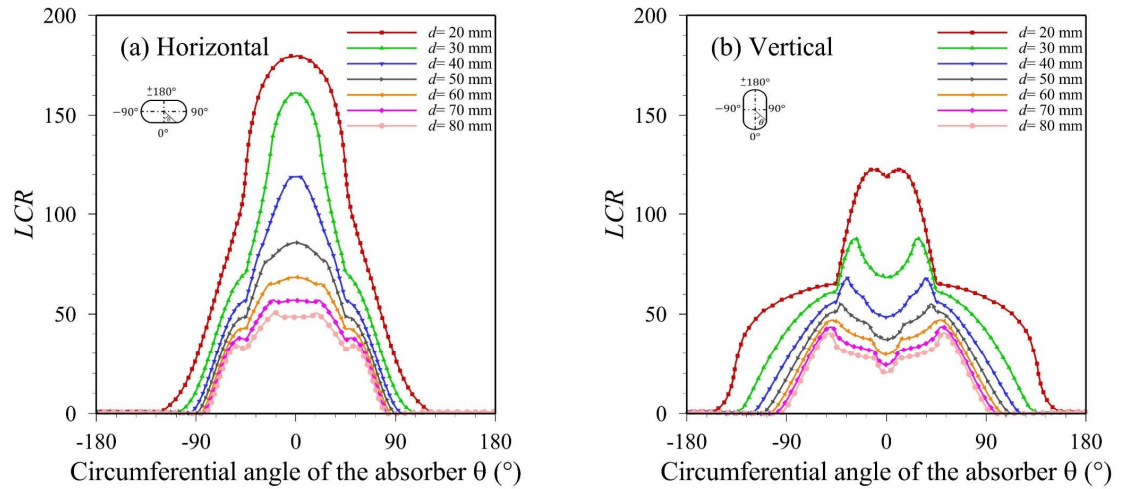


Figure 17 Effects of base circle diameter on LCR distribution at $W=5$ m, $f=1.84$ m, and $a=1$ for (a) Horizontal tube (b) vertical tube

Figure 18 present the variations of η_0 and γ with the d for horizontal and vertical configurations. It is observed that there is a gradual increase in η_0 and γ as d exceeds 30 mm. Two main factors can be used to explain this behavior. The other reason is that more direct insolation will be intercepted with the absorber as the size increases which will reduce the optical losses as mentioned before. at small diameter ($d = 20$ mm) we can notice a significant decrease in the η_0 as d goes less than 25 mm with a value of vertical tube higher than the horizontal tube. This is mainly due to the rays escaping effect at small absorber size compared to light cone, and this appears more in horizontal tube.

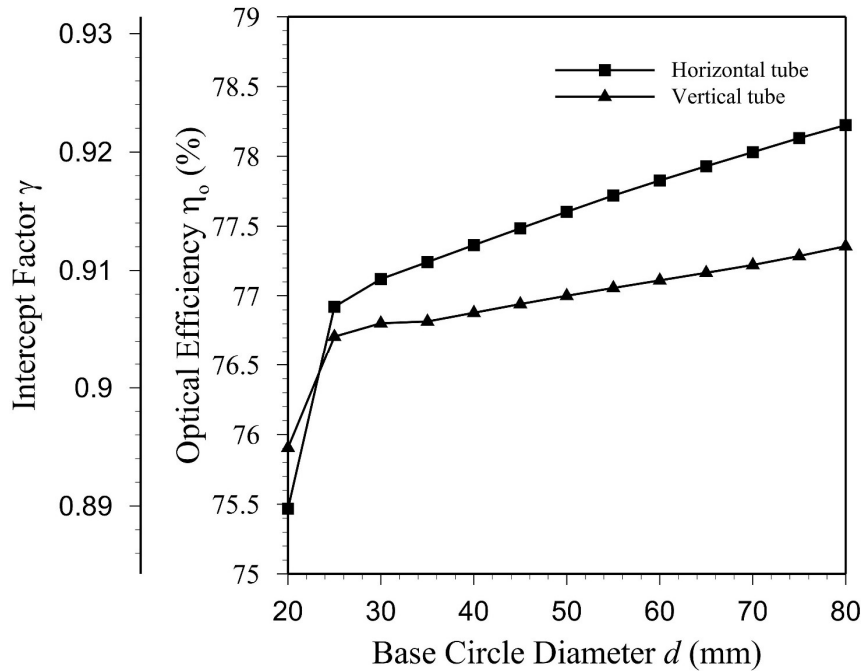


Figure 18 Effects of base circle diameter on optical efficiency and intercept factor at $W = 5$ m, $f = 1.84$ m, and $a = 1$ for horizontal and vertical flat tubes

Figure 19 (a) and (b) illustrate the variations of LCR_{max} , LCR_{ave} , and ε with the d for horizontal and vertical configurations, respectively. It is found that both LCR_{max} and LCR_{ave} decrease with increasing d , whereas ε shows an increase trend. This behavior can be

explained by the decrease in concentrated angle span with the increase in d which will cause higher non-uniformity and less average and maximum flux.

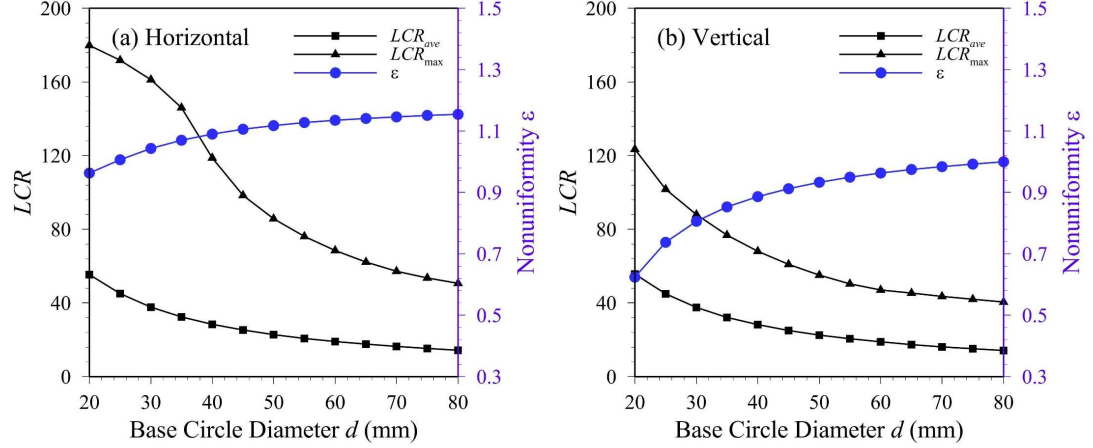


Figure 19 Variations of LCR_{max} , LCR_{ave} , and ε with base circle diameter at $W = 5$ m, $f = 1.84$ m, and $a = 1$ for (a) Horizontal tube (b) vertical tube

Effect of stretching ratio at small base circle diameters on the optical performance of the horizontal flat tube configuration

This section focuses on analyzing the effect of stretching ratio at smaller base circle diameters on the performance of the horizontal flat tube, which has demonstrated superior performance compared to the vertical flat tube.

Figure 20 shows LCR distribution for base circle diameters (d) of 10 mm, 15 mm, 20 mm, and 25 mm, with different stretching ratios (a) ranging from 0 to 4 for a horizontal flat tube. It is evident that at $d=10$ mm the maximum LCR is nearly constant across all a values, with its maximum value of approximately 190. However, as d increases the maximum LCR varies more significantly with the increase in a as can be clearly seen from the figure. This behavior can be attributed to the proximity of the absorber's lower surface to the focal point. This proximity is minimally influenced by changes in a at small d values. However, as d

value decreases the change in this closeness becomes more pronounced, leading to greater differences in the maximum LCR . It is noticeable that in all the d values shown the circular tube ($a=0$) has only one peak LCR value in contrast to the behavior at $d = 70$ mm as shown in Figure 8. This stems from that at small diameters the lower surface of the absorber is capable of capturing reflected rays originating from any location along the aperture, as previously discussed..

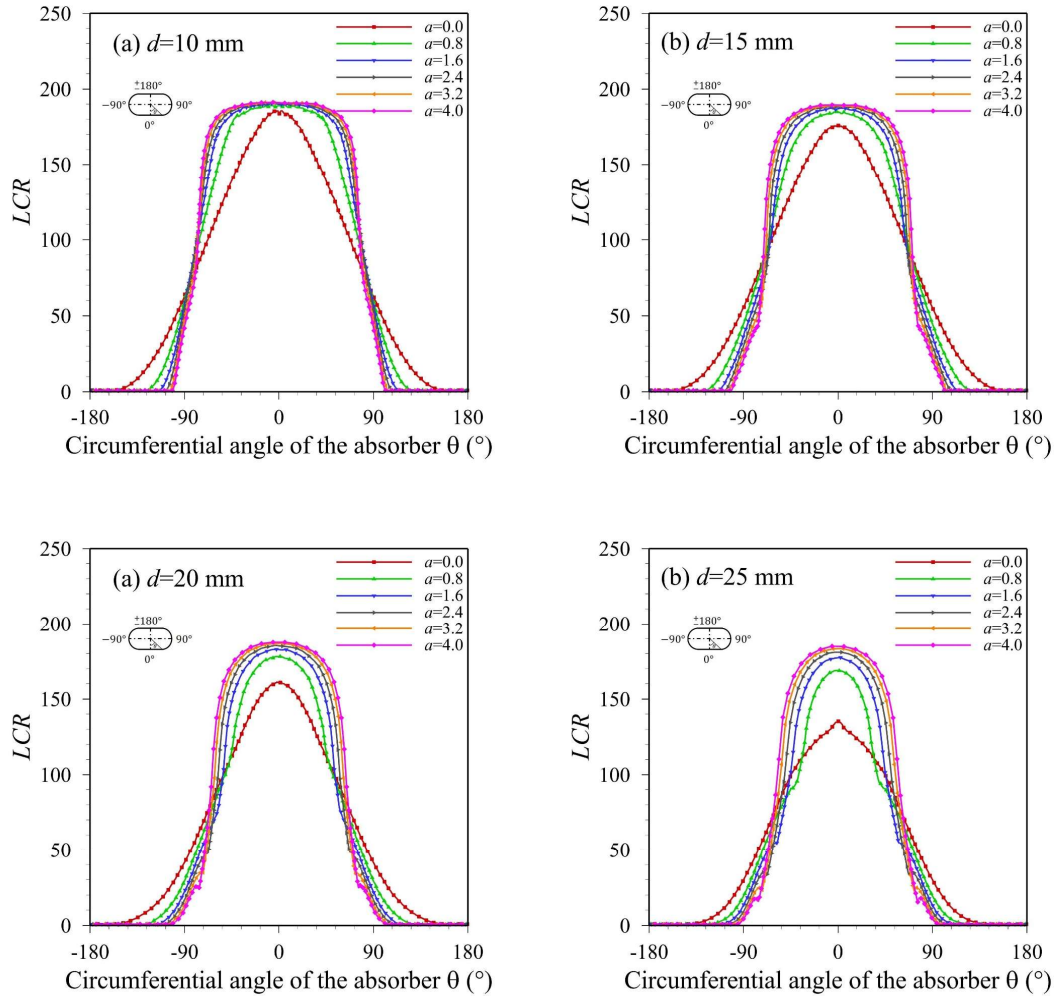


Figure 20 Effects of stretching ratio on the LCR distribution at $W = 5$ m, and $f = 1.84$ m for the horizontal flat tube at small base circle diameters of 10 mm, 15 mm, 20 mm, and 25 mm

Figure 21 shows the effect stretching ratio (a) on η_0 and γ for horizontal flat tube at different base circle diameters (d). It is obvious from the figure that at small (d) values η_0 and γ increased significantly with a . However, after $d = 20$ mm, the changes in η_0 and γ become negligible. The maximum enhancements in η_0 occur at $a = 6$, with values of 21.93%, 9.14%, and 1.66% for $d = 10$ mm, 15mm, and 20 mm respectively. This improvement is primarily attributed to the absorber's inability to intercept all reflected rays from the mirror at smaller d values and lower a . As a increases, the absorber transitions from a circular to a flat tube geometry, enabling it to intercept a greater proportion of the reflected rays. Beyond a certain d value, the original circle's relative size becomes sufficient to capture all reflected radiation and so the stretching of the geometry would not affect efficiency significantly. That means at small flow area, adopting a flat tube geometry is better to maximize the gain of energy.

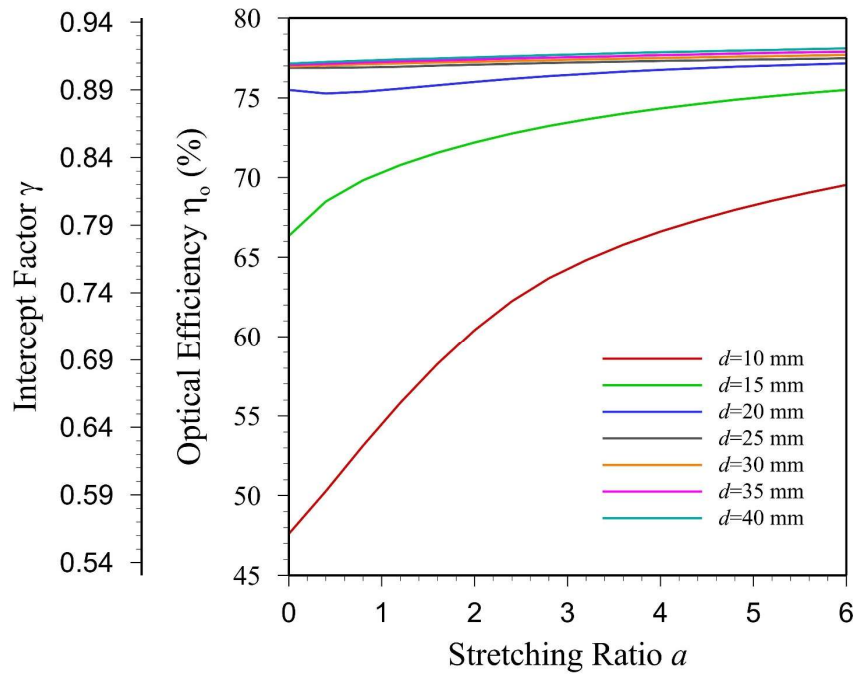


Figure 21 Effects of stretching ratio on optical efficiency and intercept factor at $W = 5$ m, and $f = 1.84$ m for the horizontal flat tube at small base circle diameters

Figure 22 illustrates the variation of the non-uniformity (ε) with the stretching ratio (a) at small base circle diameters (d). As expected, ε increases as a with higher values observed as d increases. Again, as mentioned before, this behavior is primarily due to the decrease in concentrated angle span which will lead to higher non-uniformity.

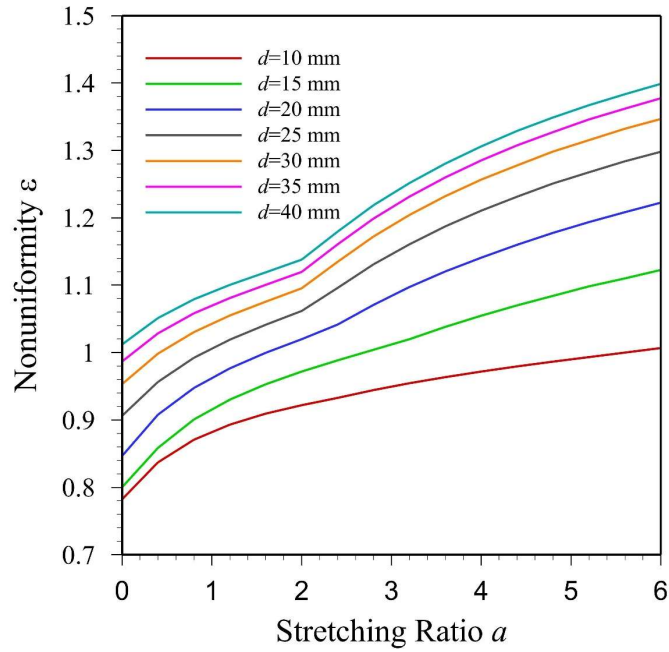


Figure 22 Variations of non-uniformity (ε) with stretching ratio at $W = 5$ m, and $f = 1.84$ m for the horizontal flat tube at small base circle diameters

CHAPTER 4

THERMAL-HYDRAULIC PERFORMANCE EVALUATION OF FLAT TUBE RECEIVERS

4.1 Introduction

Most previous research have primarily focused on enhancing the performance of PTC systems through internal modifications—such as the integration of turbulators, porous media, or extended surfaces or by utilizing advanced heat transfer fluids like nanofluids. In contrast, very limited attention has been given to modifying the external geometry of the absorber tube itself. A Recent reported work by [65] has demonstrated that incorporating flat tube absorber designs can improve the PTC optical efficiency by increasing the interception factor. However, the optical-to-thermal conversion performance of flat tube absorbers in PTC systems has not been previously explored in literature. The goal of this study is to numerically evaluate the thermal-hydraulic performance of PTC systems featuring innovative flat tube absorbers as well as the conventional circular tube under actual non-uniform solar flux conditions. The flat tube cross-sections are generated by applying a stretching ratio to the original circular base diameter in both horizontal and vertical directions. This study includes testing the thermal-hydraulic behavior for various absorber bas circle diameters by evaluating the friction factor, Nusselt number, overall thermal efficiency, and pumping power requirements over a Reynolds number range of 20,000–100,000 and HTF inlet temperatures of 150–350 °C.

4.2 Materials and methods

4.2.1 Physical model

This study investigates the performance of PTCs incorporated with new flat tube absorbers and compares them with the conventional circular absorbers. The 3D representation of the PTC equipped with flat tube receiver is illustrated in Figure 23. The system's design adopted in the current paper is based on the SEGS LS-2 PTC, as outlined in Table 1.

The flat tubes are created by reshaping the circular cross-section into a flatter form using different values of the stretching ratio (a), while keeping the cross-sectional area the same. The study considers three base circle diameters (d) of 15 mm, 30 mm, and 70 mm. Two flat-tube configurations are investigated: horizontally oriented and vertically oriented flat tubes as shown in Figure 24. For diameters of 15 mm and 30 mm, the original circular tubes were reshaped using a stretching ratio of 5 in both orientations. For the 70 mm diameter, a lower stretching of 1.5 was applied. The stretching ratio values for each base diameter were selected to ensure that the modified flat cross-sections remain entirely within the fixed diameter of the glass cover with a reasonable clearance. The relationship between the circular and flat tube cross-sections is established by assuming constant cross-sectional area. If the base circle diameter is d , the geometric transformation can be defined as follows:

$$r_a = \frac{d}{2} \sqrt{\frac{\pi}{\pi + 4a}} \quad (24)$$

$$a = \frac{C}{2r_a} \quad (25)$$

Where r_a represents the radius of the two semicircles, a denotes the stretching ratio, C corresponds to the distance shown in Figure 1.

Figure 24 illustrates the cross-sectional shapes of the different absorber designs tested in this study. The illustrations clearly compare the circular absorber with those that have been stretched horizontally and vertically. All configurations were created by assuming fixed cross-sectional area based on the original circular diameter (d).

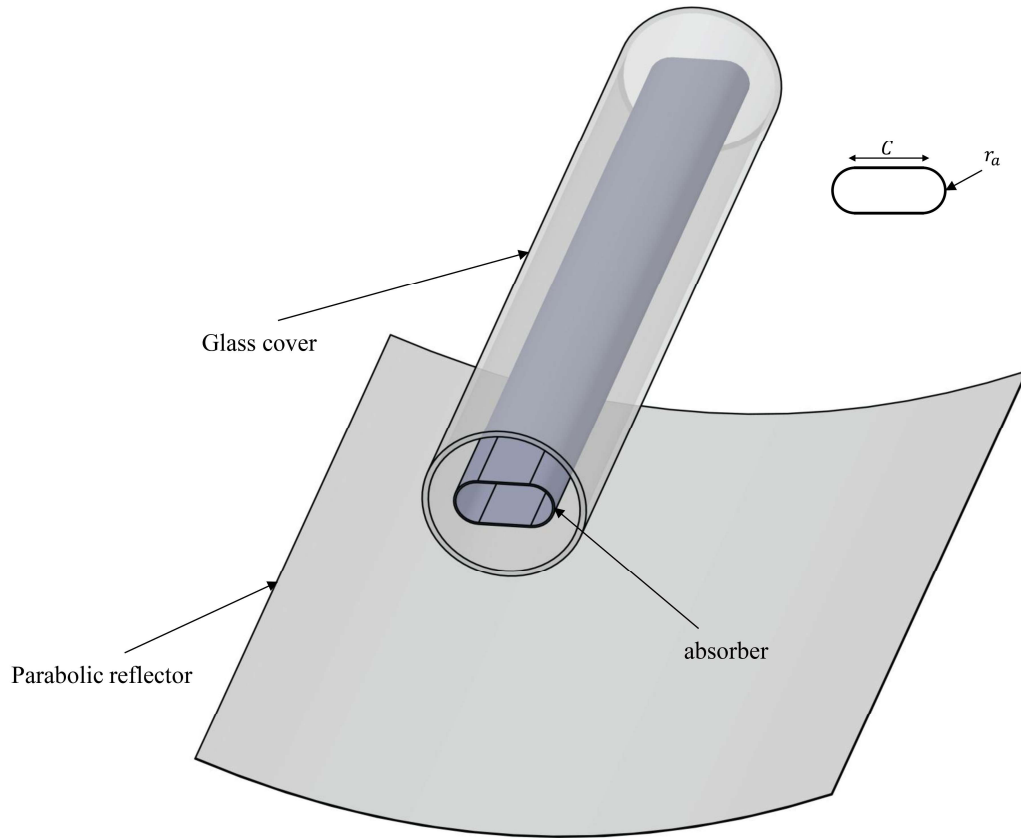


Figure 23 3D schematic of a PTC equipped with flat tube absorber

Table 2 Physical specifications of SEGS LS-2 PTC module

Specification	Value
Aperture width	5000 mm
Aperture length	7800 mm
Focal length	1840 mm
Glass cover outer diameter	115 mm
Glass cover inner diameter	109 mm
Circular absorber diameter	70 mm
Reflector reflectivity	0.93
Absorber's selective coating absorptivity	0.96
Glass transmittance	0.95
Glass refractive index	1.5
Glass cover emissivity	0.86
Glass cover material	Borosilicate glass
Absorber material	Stainless steel

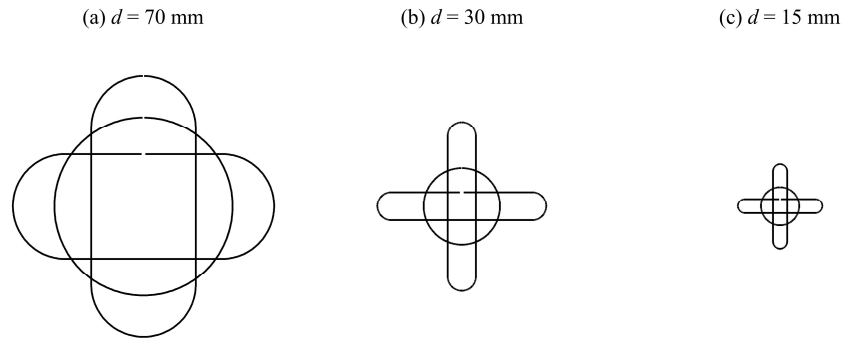


Figure 24 cross-sectional view of the tested absorber designs including circular, horizontal flat tube, and vertical flat tube at base circle diameters (a) 70 mm (b) 30 mm (c) 15 mm

4.2.2 Optical modeling

In this study, a detailed 3D optical simulation was carried out for both conventional and flat tube receiver configurations to determine the heat flux profiles, which serve as boundary conditions for subsequent thermal analysis. The geometrical factors employed in the optical analysis are presented in

Table 3. The simulation was implemented using SolTrace software, which employs the MCRT method to model the interaction of sunlight with various components of the system. MCRT is a widely used statistical method for accurately simulating optical behavior in concentrating solar power systems. It works by generating and tracking thousands of rays, every ray is initialized with a specific energy magnitude, and its path is determined by both the light source's distribution and how it interacts with different optical components—whether it reflects, refracts, scatters, gets absorbed, or diffracts. These interactions depend on the physical characteristics of the surfaces the ray encounters. In this simulation, radiation was modeled as striking the mirror surface at a perpendicular angle, considering a sun radial angle of 4.65 milliradians to account for the sun's non-point source nature [17]. The simulated number of rays were taken 3.5×10^7 for accuracy and computational considerations.

Table 3 Absorbers parameters used in the optical analysis

Configuration	Base circle diameter (d)	Stretching ratio (a)
Conventional circular tube	15 mm	0
	30 mm	0
	70 mm	0
Horizontal flat tube	15 mm	5
	30 mm	5
	70 mm	1.5
Vertical flat tube	15 mm	5
	30 mm	5
	70 mm	1.5

The flux distribution profiles from the optical simulations are shown in Figure 25. Each absorber circumference was divided into 360 angular segments to capture the irradiance variation.

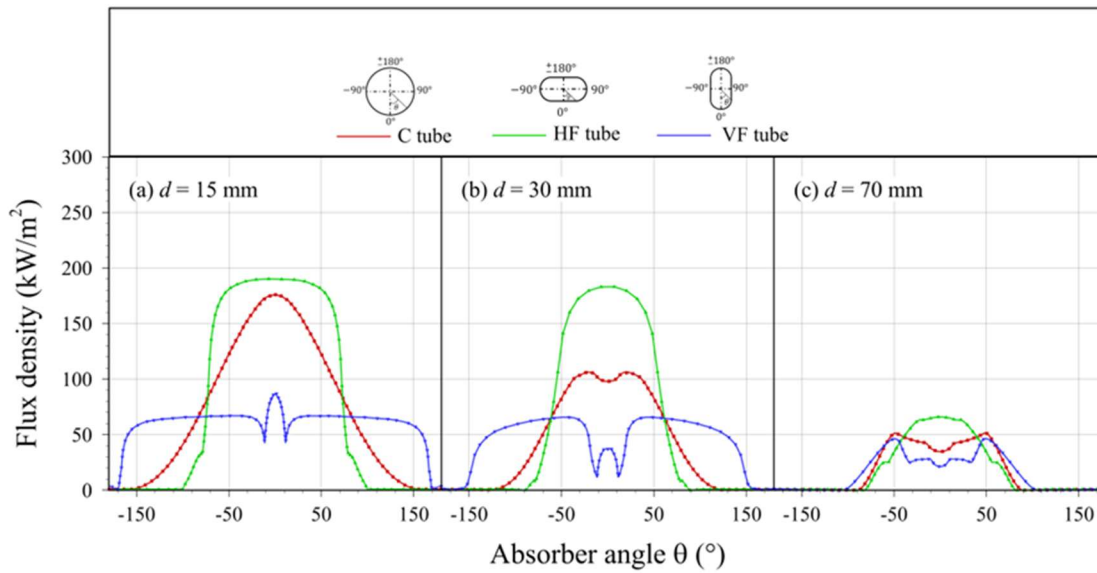


Figure 25 Flux distribution obtained by MCRT for the tested configurations including C tube, HF tube, and VF tube at (a) $d = 15$ mm (b) $d = 30$ mm (c) $d = 70$ mm.

4.2.3 Governing equations

In a PTC system, the radiation is absorbed by an HTF flowing within the absorber tube. The flux profile is obtained through the MCRT method as previously described. A turbulent, incompressible, and steady-state flow is assumed for the HTF inside the absorber. with the velocity considered to be unidirectional along the z-axis. The physical behavior of the system is governed by the foundational equations describing mass, momentum, and energy balance.

Mass conservation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \quad (26)$$

x-momentum

$$\rho \left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} \right) = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left(\mu_{\text{eff}} \frac{\partial u}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu_{\text{eff}} \frac{\partial u}{\partial y} \right) + \frac{\partial}{\partial z} \left(\mu_{\text{eff}} \frac{\partial u}{\partial z} \right) \quad (27)$$

y-momentum

$$\rho \left(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} \right) = -\frac{\partial p}{\partial y} + \left(\mu_{\text{eff}} \frac{\partial v}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu_{\text{eff}} \frac{\partial v}{\partial y} \right) + \frac{\partial}{\partial z} \left(\mu_{\text{eff}} \frac{\partial v}{\partial z} \right) \quad (28)$$

z-momentum

$$\rho \left(u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} \right) = -\frac{\partial p}{\partial z} + \left(\mu_{\text{eff}} \frac{\partial w}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu_{\text{eff}} \frac{\partial w}{\partial y} \right) + \frac{\partial}{\partial z} \left(\mu_{\text{eff}} \frac{\partial w}{\partial z} \right) \quad (29)$$

Energy conservation

$$\rho C_p \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} \right) = \frac{\partial}{\partial x} \left(\lambda_{\text{eff}} \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(\lambda_{\text{eff}} \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left(\lambda_{\text{eff}} \frac{\partial T}{\partial z} \right) + \Phi \quad (30)$$

$$\Phi = \mu \left[2 \left(\frac{\partial u}{\partial x} \right)^2 + 2 \left(\frac{\partial v}{\partial y} \right)^2 + 2 \left(\frac{\partial w}{\partial z} \right)^2 + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 + \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \right)^2 \right] \quad (31)$$

Where u , v , and w are the time-averaged x, y, z velocity components respectively, p is the time-averaged pressure, T is the time-averaged local temperature, Φ the viscous dissipation term, ρ and C_p are the density and specific heat respectively. μ_{eff} , and λ_{eff} are the effective dynamic viscosity and thermal conductivity respectively. They are expressed as

$$\mu_{\text{eff}} = \mu + \mu_t \quad (32)$$

$$\lambda_{\text{eff}} = \lambda + \lambda_t \quad (33)$$

Here μ and λ are the dynamic viscosity and thermal conductivity respectively. μ_t and λ_t are turbulent (eddy) viscosity and conductivity, respectively. they are given by

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (34)$$

$$\lambda_t = \frac{\mu_t C_p}{Pr_t} \quad (35)$$

Where k and ε represent the turbulent kinetic energy and its dissipation rate, respectively, C_μ is a model-dependent constant ($C_\mu = 0.09$), Pr_t is turbulent Prandtl number (typically ≈ 0.85 – 1.0). In this study, the standard $k - \varepsilon$ turbulence model was used to simulate the effects of turbulence on the fluid flow. Recognized as one of the standard models frequently utilized in engineering analysis, the standard $k - \varepsilon$ turbulence model has been commonly

adopted since it was first developed by Launder and Spalding [66]. This model incorporates two additional transport equations: one governing turbulent kinetic energy k and the other describing its rate of dissipation ε . The two added equations are as follows

$$\rho \left(u \frac{\partial k}{\partial x} + v \frac{\partial k}{\partial y} + w \frac{\partial k}{\partial z} \right) = \frac{\partial}{\partial x} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x} \right] + \frac{\partial}{\partial y} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial y} \right] + \frac{\partial}{\partial z} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial z} \right] + P_k - \rho \varepsilon \quad (36)$$

$$\rho \left(u \frac{\partial \varepsilon}{\partial x} + v \frac{\partial \varepsilon}{\partial y} + w \frac{\partial \varepsilon}{\partial z} \right) = \frac{\partial}{\partial x} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x} \right] + \frac{\partial}{\partial y} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial y} \right] + \frac{\partial}{\partial z} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial z} \right] + C_{\varepsilon 1} \frac{\varepsilon}{k} P_k - C_{\varepsilon 2} \rho \frac{\varepsilon^2}{k} \quad (37)$$

In the above equations $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.3$, $C_{\varepsilon 1} = 1.44$, and $C_{\varepsilon 2} = 1.92$ are the model constants. P_k is the production term which quantifies the generated kinetic energy introduced due to mean flow. P_k is denoted as

$$P_k = \mu_t \left[2 \left(\frac{\partial u}{\partial x} \right)^2 + 2 \left(\frac{\partial v}{\partial y} \right)^2 + 2 \left(\frac{\partial w}{\partial z} \right)^2 + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 + \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \right)^2 \right] \quad (38)$$

4.2.4 Surface to surface radiation model

To simulate the radiative losses from the absorber to the glass cover, Surface-to-Surface (S2S) radiation model was utilized. This model operates under the primary assumption that the medium in between the surfaces doesn't participate in the radiation process-meaning that it does not absorb, emit, or scatter radiation. Additionally, the surfaces involved are considered opaque, implying that radiation cannot transmit through them. In the S2S model, part of the radiation is diffusively reflected from the surface, and another part is emitted. The sum of the two parts is called the radiosity (J) and it is represented by the following expression [67]

$$J = \varepsilon_s \sigma_{rad} T^4 + \rho_d G \quad (39)$$

Where J is the radiosity, which is the total diffuse radiative energy emitted and reflected from the surface. G is the total incoming radiation flux from the surroundings, ρ_d is the diffuse reflectivity, and ε_s is the surface emissivity. The process involves computing the view factors between the participating surfaces to accurately capture the radiative interactions within the enclosure.

4.2.5 Boundary conditions

To model the thermal behavior of the receiver under realistic operating scenarios, suitable boundary conditions were defined on both solid and fluid regions of the receiver in thermal simulations. The non-uniform flux distributions derived from the optical simulations were applied to the outer surface of the absorbers.

A uniform velocity inlet condition was set on the fluid side based on Reynolds numbers ranging from 20,000 to 80,000, increasing in steps of 15,000, allowing the analysis of different flow regimes. The inlet temperature of the HTF varied between 150 °C and 350 °C, representing typical conditions found in CSP systems. At the tube's exit, a pressure outlet boundary condition was applied, and no-slip conditions were used on all wall surfaces to accurately capture the flow behavior within the system.

In this study, thermal losses were modeled in two stages: first, between the absorber surface and the inner surface of the glass cover, and subsequently from the outer surface of the glass cover to the surrounding environment. Due to the assumed high vacuum (0.013 Pa) [68], the losses from the absorber to the glass cover were considered to occur only by radiation, as convection and conduction were negligible under such conditions. To model

this radiative exchange, the S2S radiation model was employed, and the glass cover is treated as a gray surface in this radiation spectrum. The absorber emissivity was calculated using the following expression [69].

$$\varepsilon_a = 0.000327 T - 0.065971 \quad (40)$$

Where T is the temperature in K. The losses from the outer surface of the glass cover were assumed to occur by both radiative and convective heat transfer mechanisms. The radiation between the outer glass cover and the ambient environment was treated as a heat exchange between the glass surface and a large enclosing surface, the sky. The sky temperature was estimated using a correlation that links it to the ambient air temperature as [70].

$$T_{sky} = 0.0552 T_{amb}^{1.5} \quad (41)$$

The ambient temperature (T_{amb}) was taken to be 300 K. The radiation losses were calculated using Stefan–Boltzmann equation [71]

$$E_{rad,loss} = \varepsilon_g \sigma_{rad} A (T_{amb}^4 - T_{sky}^4) \quad (42)$$

Here, ε_g represents the glass cover emissivity, and σ_{rad} denotes the Stefan–Boltzmann constant. The coefficient of the convective heat transfer between the ambient and the outer glass cover was estimated by [3].

$$h_w = 4V_w^{0.58} d_{go}^{-0.42} \quad (43)$$

Here, V_w denotes the wind speed, which was taken 2 m/s in this study, and d_{go} refers to the outer diameter of the glass cover.

To reduce the computations a symmetrical boundary condition is applied along the y-z plane by considering a half-domain model as shown in Figure 26. This condition is justified by the system's physical and geometric symmetry—as the flux profile is symmetric, and the fluid flow is expected to mirror on either side.

Syltherm-800 was chosen as the HTF for this research, a high-performance, silicone-based fluid engineered for high-temperature applications. The thermophysical properties of HTF vary as a function of temperature. and are determined by the following expressions [3].

$$\rho = -6.0616 \times 10^{-4} T^2 - 4.1535 \times 10^{-1} T + 1.1057 \times 10^3 \quad (44)$$

$$\mu = 6.6720 \times 10^{-13} T^4 - 1.5660 \times 10^{-9} T^3 + 1.3882 \times 10^{-6} T^2 - 5.5412 \times 10^{-4} T + 8.4866 \times 10^{-2} \quad (45)$$

$$c_p = 1.7080 T + 1.1078 \times 10^3 \quad (46)$$

$$\lambda = -5.7534 \times 10^{-10} T^2 - 1.8752 \times 10^{-4} T + 1.9002 \times 10^{-1} \quad (47)$$

4.2.6 Thermal-hydraulic performance parameters

As stated before, simulations were carried out for both conventional and flat tube configurations by varying the Re from 20,000 to 80,000 and setting the HTF inlet temperature at 150 °C, 250 °C, and 350 °C. the definition of Re number is as follows

$$Re = \frac{\rho v_m D_h}{\mu} \quad (48)$$

Here, D_h is the hydraulic diameter (m), v_m is the mean flow velocity (m/s). The overall thermal efficiency indicates the proportion of absorbed thermal energy gained by the HTF, given by

$$\eta_{th} = \frac{Q_u}{I_n A_a} \quad (49)$$

Where Q_u denotes the useful heat gain (W), I_n represents the incident irradiance on the parabola aperture area (1000 W/m² was taken), and A_a is the parabola aperture area (m²).

The useful heat gain is given by

$$Q_u = I_n \eta_o A_a - Q_{loss} \quad (50)$$

Where η_o is the optical efficiency achieved from the optical analysis, and Q_{loss} corresponds to thermal losses from the system. Q_{loss} can be obtained either by calculating the losses from the absorber due to the radiation, or the combined radiative and convective losses from the glass cover as follows

$$Q_{loss} = \int_{A_{abs}} q_{rad,abs} dA = \int_{A_{glass}} (q_{rad,glass} + q_{conv,glass}) dA \quad (51)$$

$q_{rad,abs}$, $q_{rad,glass}$, and $q_{conv,glass}$ are flux quantities obtained from the numerical solution. The thermal flow behavior within the system is quantified by the mean Nusselt number, calculated along the absorber inner surface.

$$Nu = \frac{h D_h}{k} \quad (52)$$

Where h corresponds to the average convective heat transfer coefficient (W/m² K), and it is calculated by

$$h = \frac{Q_u}{A_i (T_w - T_{mf})} \quad (53)$$

Here A_i is the absorber's inner surface area (m^2), T_w and T_{mf} are the absorber's inner wall temperature and the HTF mean temperature in (K) respectively. $\frac{T_i+T_o}{2}$ and the wall temperature can be calculated as follows

$$T_w = \frac{1}{A_i} \int_{A_i} T dA \quad (54)$$

The friction factor (f_r) is another important parameter in evaluating the hydraulic performance, and it represents the dimensionless pressure drop due to wall shear stress. It is calculated by

$$f_r = \frac{2\Delta P D_h}{\rho L v_m^2} \quad (55)$$

Where ΔP denotes the pressure change between inlet and outlet (Pa), L stands for the length of the pipe (m). Colebrook correlation [72] for smooth pipe is also utilized to calculate Darcy friction factor and compare it with the numerical solutions as follows

$$\frac{1}{\sqrt{f_r}} = -2.0 \log \left(\frac{2.51}{\text{Re} \sqrt{f_r}} \right) \quad (56)$$

The required pumping power to overcome friction losses is given by

$$W_p = \dot{V} \Delta P \quad (57)$$

Where W_p is the pumping power in (W) and $\dot{V}$ stands for volumetric flow rate (m^3/s).

4.2.7 Numerical modeling

The numerical method used in this study to solve the governing equations is the Finite Element Method (FEM). FEM offers a structured approach by breaking down the computational domain into smaller subdomains, or elements. The Galerkin method is initially employed to discretize the conservation equations and integrating the boundary conditions. The velocity, pressure, and temperature are then discretized over the domain, then substituted into the governing equations. The Galerkin weighted residual method is employed to minimize discretization errors by enforcing the residuals to be orthogonal to the selected interpolation functions. After that the Gauss-divergence theorem is applied and the discretized conservation equations can be rewritten in matrix form. The solution convergence is considered when the relative error in each variable from one iteration to the next falls below a specified threshold, certainly when $|\psi_{j+1} - \psi_j| \leq 10^{-4}$, where j stands for the iteration number, and ψ is a variable representing the flow parameters like u , v , w , p , and T .

4.3 Mesh sensitivity analysis and model validation

A custom user-defined meshes for both circular and flat tube configurations were generated using COMSOL Multiphysics 6.2, with the domain discretized primarily using quadrilateral elements. Only one half of the domain was taken for the numerical simulation due to the symmetrical condition. The model consists of solid domains, which include the absorber wall and the glass cover, and a fluid domain representing the internal volume of the absorber where HTF flows. Figure 26 illustrate the generated mesh for different geometry at base circle diameter of 70 mm.

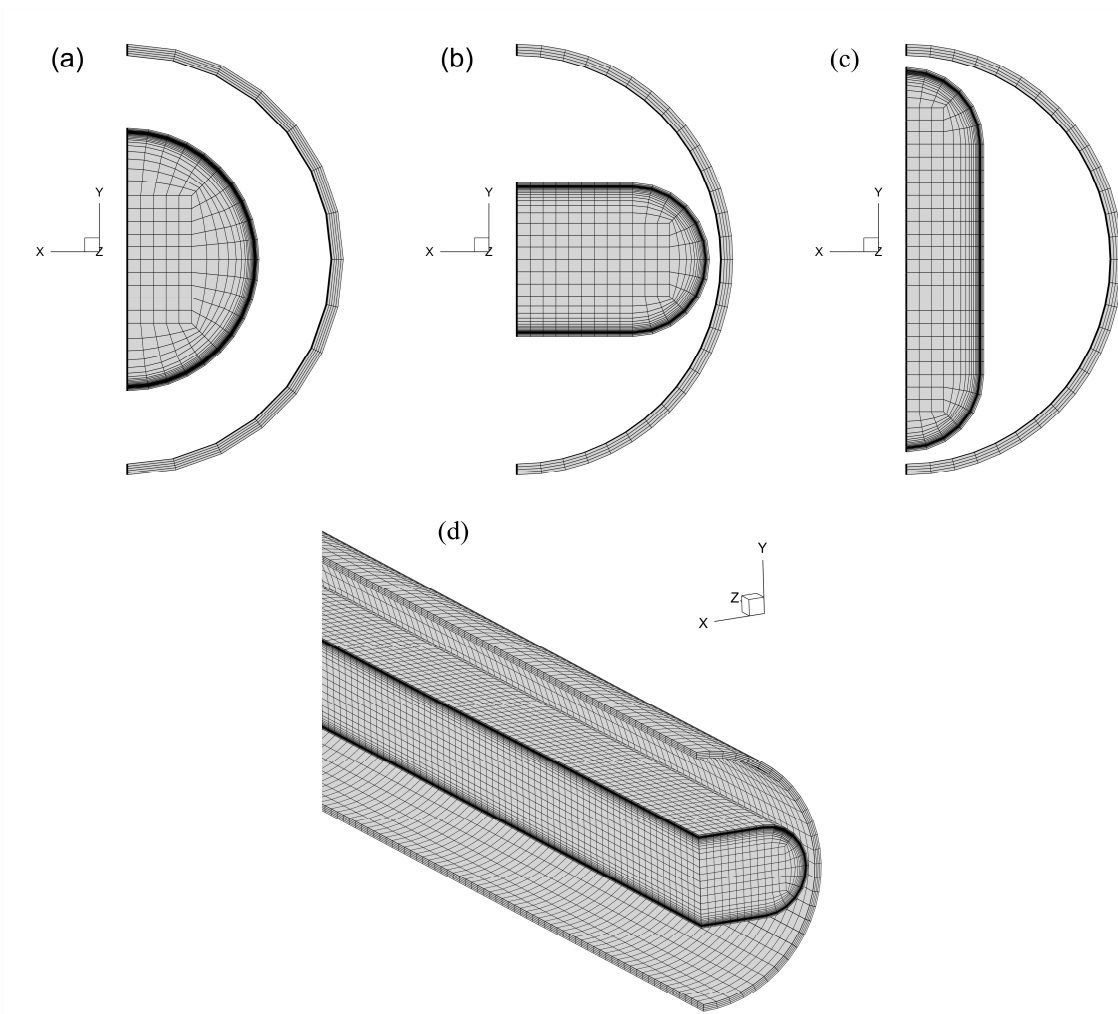


Figure 26 A representative mesh structure generated using COMSOL Multiphysics 6.2 for 70 mm base circle diameter. Cross-sectional views of (a) circular tube (b) horizontal flat tube (c) vertical flat tube. (d) isometric view of the horizontal flat tube

To ensure accurate solution a mesh sensitivity studies were carried out for the circular and flat tube configurations. The horizontal flat tube was adopted to represent the flat tube configuration in the mesh sensitivity studies. The studies were performed for various mesh densities ranging from coarse to very fine, while keeping the operating conditions constant—a base circle diameter of 70 mm, Reynolds number of 50,000 and a heat transfer fluid (HTF) inlet temperature of 250 °C. the parameters that selected to compare the different cases are Nu and f_r . Based on the tested cases the optimum grid numbers were

selected to be 523600 for the circular tube and 681600 for the flat tube. The corresponding average errors compared to the very fine mesh results were 0.23% and 0.022%, respectively, as summarized in Table 4.

Table 4 Mesh sensitivity analysis

Configuration	Case #.	Number of elements	Nu	f_r	% $ΔNu$	% $Δf_r$
Circular tube	1	60928	711.54	0.0212	3.54 %	2.43 %
	2	146880	721.23	0.0212	2.23 %	2.34 %
	3	293760	732.82	0.0216	0.6 %	0.69 %
	4	523600	736.17	0.0217	0.2 %	0.26 %
	5	750720	737.69	0.0217	Reference	Reference
Flat tube	1	212000	758.06	0.0214	0.92%	0.79 %
	2	276000	761.73	0.0216	0.44%	0.07%
	3	347200	763.01	0.0216	0.27%	0.15%
	4	681600	765.09	0.0216	0.003%	0.04%
	5	1006000	765.07	0.0216	Reference	Reference

Since no one conducted study on flat tube design, the numerical model was validated for the circular tube design using experimental data reported by [73], with a focus on thermal efficiency. The operating conditions for the validation cases are outlined in Table 5. As illustrated in Figure 27, the numerical findings closely match the experimental data, showing good agreement with an average deviation of just 0.51%.

Table 5 Operating conditions of the experimentally tested cases as reported by [73]

Case #	DNI (W/m ²)	Wind speed (m/s)	Ambient temperature (°C)	HTF inlet temperature (°C)
1	968.2	3.7	22.4	151
2	933.7	2.6	21.2	102.2
3	982.3	2.5	24.3	197.5
4	909.5	3.3	26.2	250.7

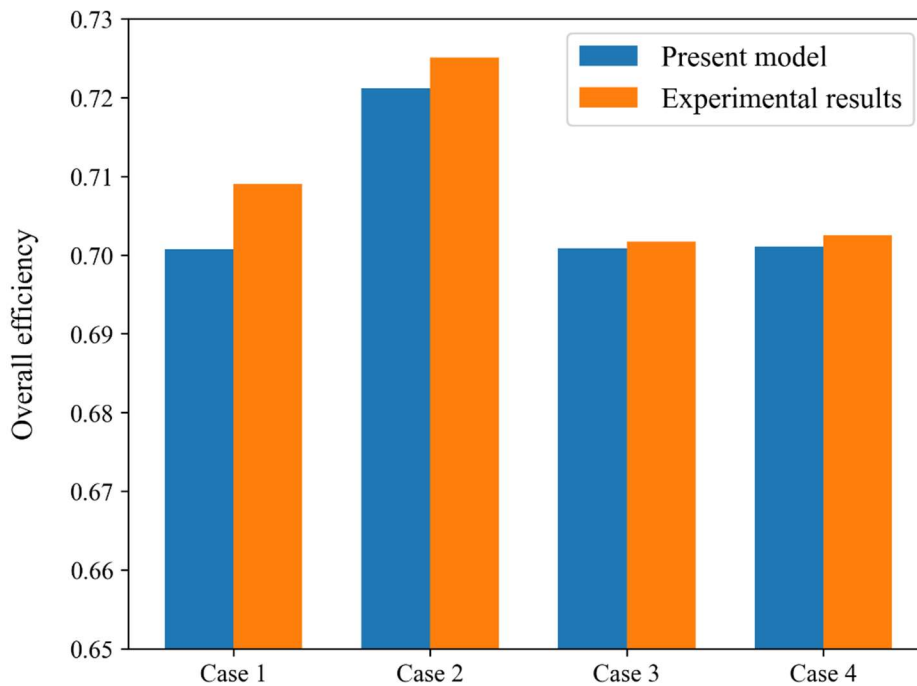


Figure 27 Comparison between the numerical results and the experimental results of [73] for the different validation cases

4.4 Results and discussion

This section presents an in-depth evaluation of the thermal and hydraulic performance of PTC systems equipped with three absorber geometries: a conventional circular tube, a horizontal flat tube and a vertical flat tube. The results presented in this section are derived

from coupled optical and thermal simulations performed for the three tested geometries for three different base circle diameters (15 mm, 30 mm, and 70 mm). To evaluate performance under realistic operating conditions, simulations were carried out at three heat transfer fluid (HTF) inlet temperatures (150 °C, 250 °C, and 350 °C), while varying the Reynolds number from 20,000 to 80,000 in increments of 15,000. Key performance metrics—including Nusselt number, friction factor, thermal efficiency, and pumping power—are used to evaluate and compare each absorber configuration. Furthermore, the Relative Benefit-Cost Ratio (RBCR) was evaluated for all cases under three different price ratio scenarios (0.1, 1, and 10), providing a comprehensive assessment of the performance relative to economic cost.

For clarity and simplification, abbreviations are used throughout the analysis: C tube refers to the conventional circular absorber, while HF tube and VF tube represent the horizontal and vertical flat tube absorbers, respectively.

4.4.1 Friction factor

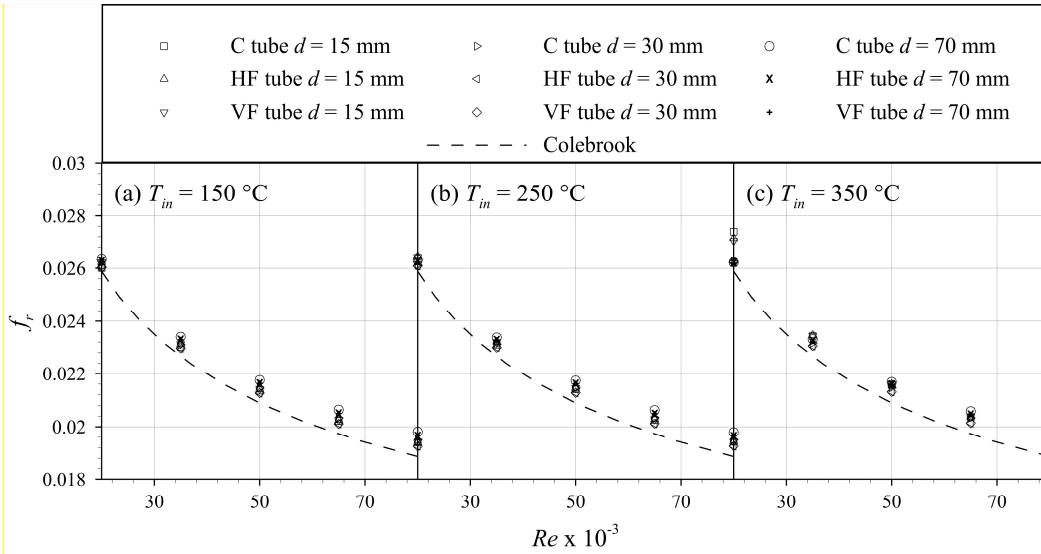


Figure 28 Variation of friction factor with Reynolds number for all tested geometries at (a) $T_{in} = 150$ °C (b) $T_{in} = 250$ °C (c) $T_{in} = 350$ °C

Figure 28 shows the variations of friction factors (f_r) with Re number for all the tested geometries at different HTF inlet temperatures. Based on the findings, it can be clearly noticed an inverse relationship between f_r and Re number, which is consistent with classical fluid dynamics for turbulent flow in smooth pipes. As Re number increases from 20000 to 80000, f_r decreases across all tested configurations. Furthermore, the results indicate clearly that the fluid inlet temperature has almost no effect on f_r . This observed trend indicates the dominating effect of inertial forces over viscous forces at higher Reynolds numbers, reducing the relative resistance along the pipe walls.

The dashed line represents the Colebrook correlation, a standard for turbulent flow in smooth pipes. The close agreement with the simulation results confirms turbulent conditions. Additionally, the f_r values observed across different receiver shapes were found to be nearly identical, indicating that the cross-sectional shape has negligible effect on hydraulic performance, at least within the tested Re range. Minor deviations from the

Colebrook curve may be attributed to factors such as flow structures, or numerical uncertainty. However, the overall trend remains consistent

4.4.2 Pumping power

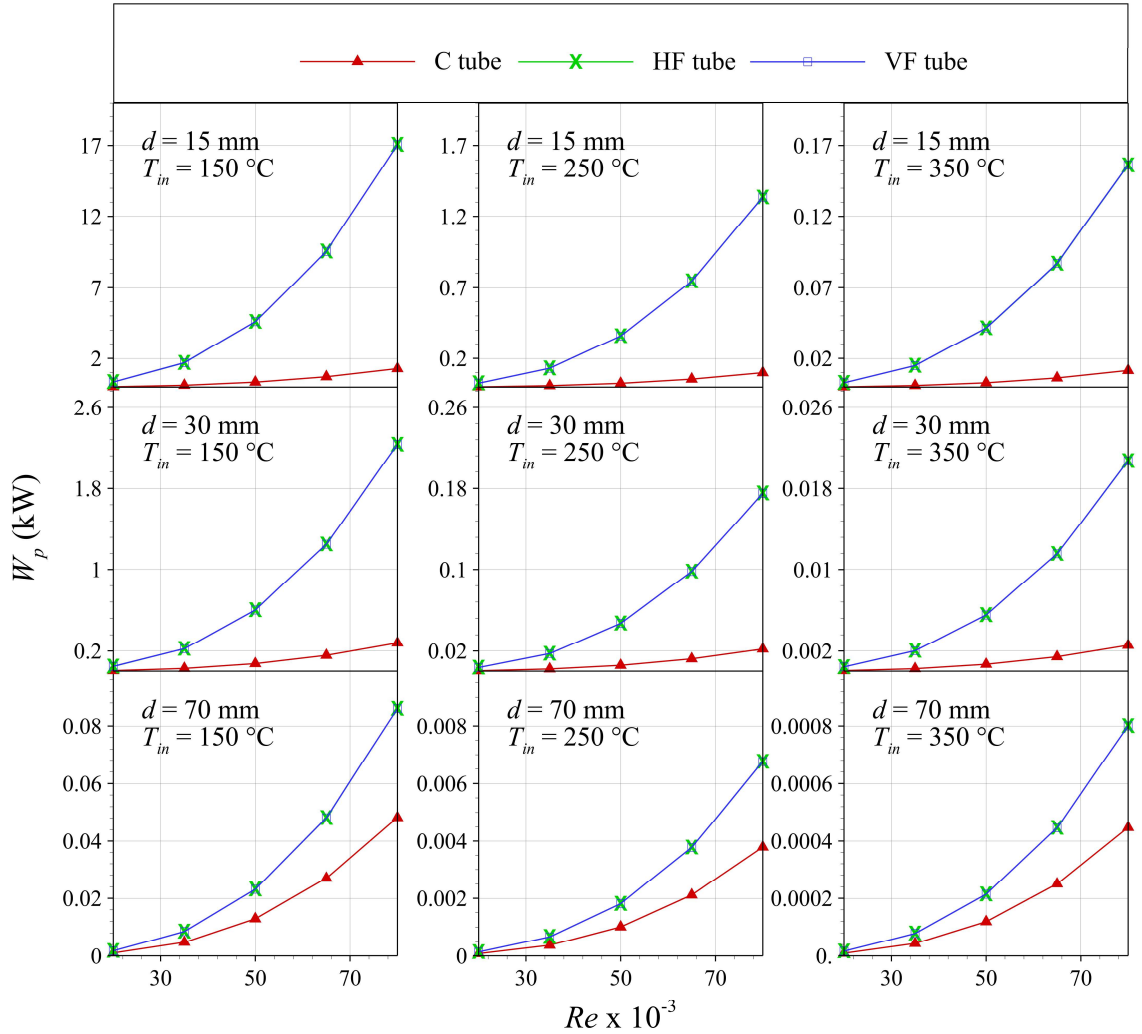


Figure 29 Variation of pumping power W_p with Re for CT, HF tube, and VF tube at ($d = 15$ mm, $d = 30$ mm, and $d = 70$ mm).

The results obtained for the required pumping power (W_p) are illustrated in Figure 29. It is obvious that W_p increases with Re across all the cases, which is expected due to the increased flow velocity at higher Re , leading to increased pressure losses and, consequently, higher pumping power requirements. Furthermore. The increase in HTF

inlet temperature from 150 °C to 350 °C results in significant reduction in Pumping power. For example, at $d = 30$ mm and $Re = 50000$, W_p decreased for both HFT and VFT from 0.603 kW at $T_{in} = 150$ °C to 0.047 kW at $T_{in} = 250$ °C and to 0.0055 kW at $T_{in} = 350$ °C all these values evaluated. This behavior occurs due to the change in the HTF properties with temperature, specifically due to the decrease in fluid density and viscosity at higher temperatures, which leads to lower pressure losses for a given Re .

The findings of Figure 29 indicates that W_p required by HF tube and VF tube designs are much higher in comparison to CT configuration, with HF tube and VF tube exhibiting identical W_p values. Although the friction factors are the same for all configurations, these findings are primarily due to the smaller flow cross-sectional area of the flat tube designs as wall thickness is fixed at 1 mm for all configurations, and only the outer absorber area is fixed. A smaller cross-sectional area leads to lower hydraulic diameter and thus high velocity at the same Re , both contribute to increased pressure drop across the tube.

When it comes to the effect of the base circle diameter, Figure 29 clearly shows that shows that the required pumping power (W_p) decreases significantly as the diameter increases. For example, the VF tube and HF tube with 15 mm base diameters require approximately 0.35 kW at $Re = 50000$ and $T_{in} = 250$ °C, compared to 0.047 kW at $d = 30$ mm and just 0.002 kW at $d = 70$ mm. This reduction is primarily attributed to the increase in hydraulic diameter with larger base diameters, which lowers flow resistance. Moreover, the gap between flat tube designs and circular design decreases significantly at $d = 70$ mm. This is because the difference in hydraulic diameter between the flat tube designs and the circular tube becomes relatively small at high base diameters.

4.4.3 Nusselt number

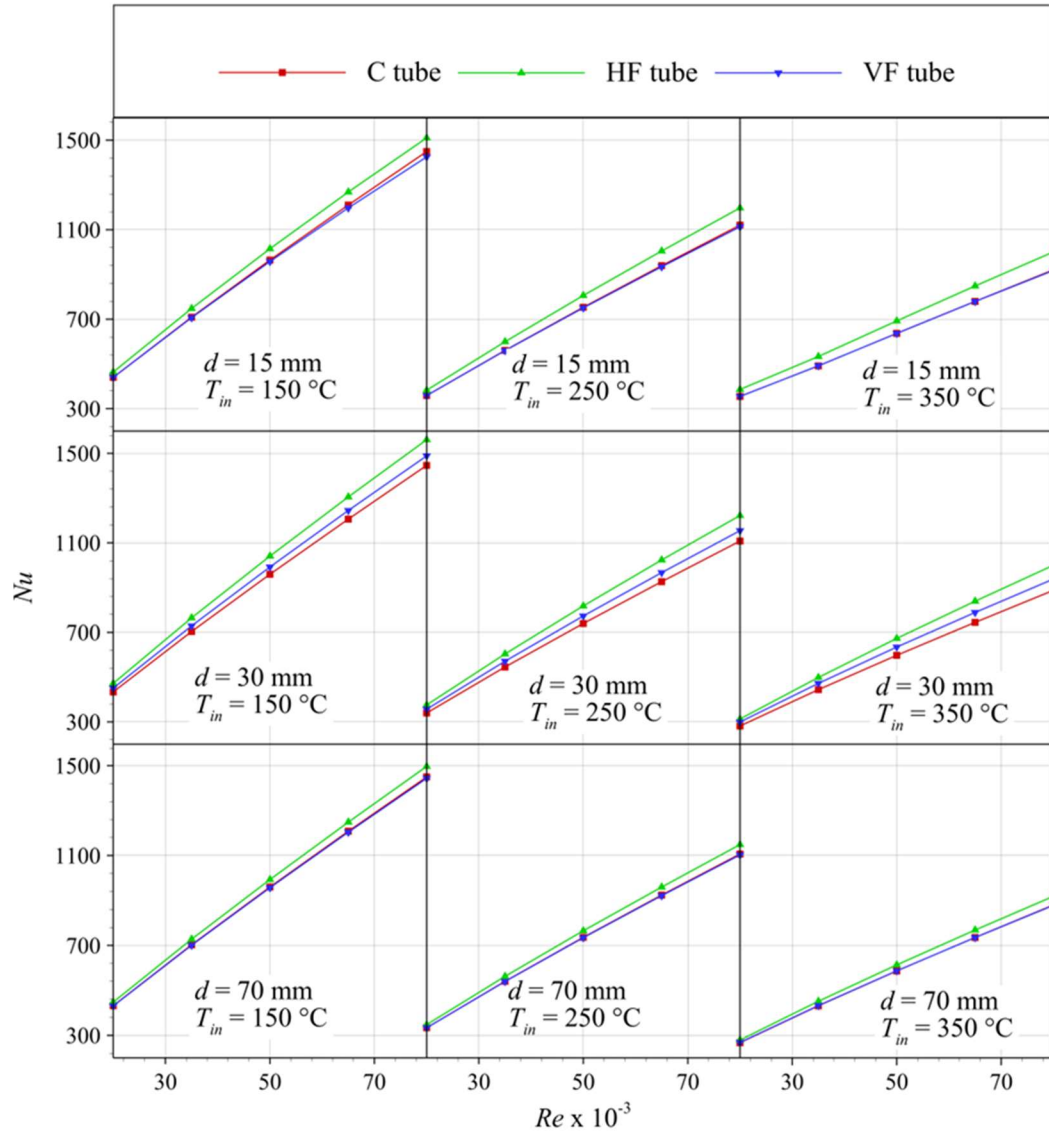


Figure 30 Variation of average Nu with Re for CT, HF tube, and VF tube at ($d=15$ mm, $d=30$ mm, and $d=70$ mm).

Figure 30 depicts the results obtained for average Nu number as a function of Re and HTF inlet temperature. The results indicate a directly proportional relationship between Nu and Re across all cases, which aligns with convective heat transfer theory, where higher Re intensifies turbulence and mixing near the absorber surface, thereby improving heat transfer. increasing the HTF inlet temperature tends to reduce Nu . For instance, HF tube

shows a reduction in Nu from 1040.6 at $T_{in} = 150\text{ }^{\circ}\text{C}$ to 818.16 at $T_{in} = 150\text{ }^{\circ}\text{C}$ and 672.68 at $T_{in} = 150\text{ }^{\circ}\text{C}$, all evaluated at $Re = 50000$ and $d = 15\text{ mm}$. The observed trend is mainly because of the temperature dependent properties of HTF. As the fluid temperature increases Prandtl number (Pr) decreases due to reduced viscosity. A lower Pr indicates that the thermal boundary layer develops more slowly relative to the momentum boundary layer, which in turn lowers the effectiveness of convective heat transfer.

The results presented in Figure 30 indicates superior performance of HF tube in all cases, particularly at smaller d values. For example, at $d = 15\text{ mm}$ and $Re = 50000$, HF tube achieves a higher Nu of 805.85, compared to 753.35 for the C tube and 636.29 for the VF tube. The enhanced convective heat transfer capability of HF tube configuration can be attributed to the lower temperature values of the fluid, resulting from a higher mass flow rate at the same Reynolds number. This reduction in local fluid temperature leads to an increase in the Prandtl number, which in turn enhances convective heat transfer effectiveness.

According to the findings from Figure 30 C tube and VF tube exhibit non-consistent behavior across different base diameters. At $d = 15\text{ mm}$ CT shows slightly higher Nu values compared to VF tube overall. However, at $d = 30\text{ mm}$ VF tube outperforms C tube, and they have almost same Nu values at $d = 70\text{ mm}$. These observations could also be attributed to the changes in HTF temperature distribution which is influenced by the flux profile.

4.4.4 Temperature distribution

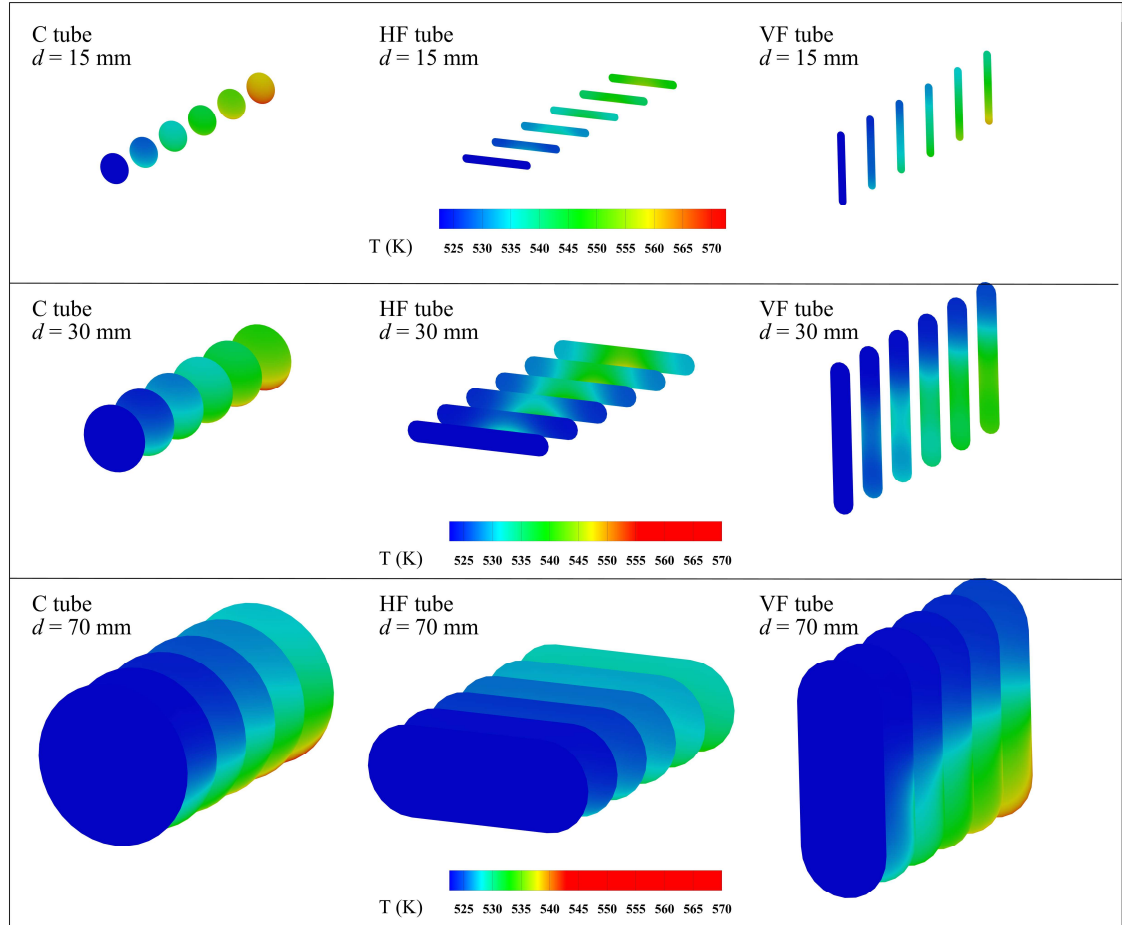


Figure 31 Temperature distribution of HTF at different cross-sections for inlet temperature of 523.15 °C, and $Re = 50000$

Figure 31 presents the fluid temperature distribution at various cross-sections along the flow direction within the absorber. The figure clearly illustrates that the fluid temperature increases along the tube length, with higher temperatures near the bottom wall due to the higher flux concentration in that region. It's important to note that absolute temperature values across different configurations aren't directly comparable, as each operates with a different mass flow rate to maintain a fixed Reynolds number. Since Re depends on hydraulic diameter and velocity, variations in cross-sectional area between geometries cause significant differences in flow rate, affecting the resulting temperature fields.

4.4.5 Overall thermal efficiency

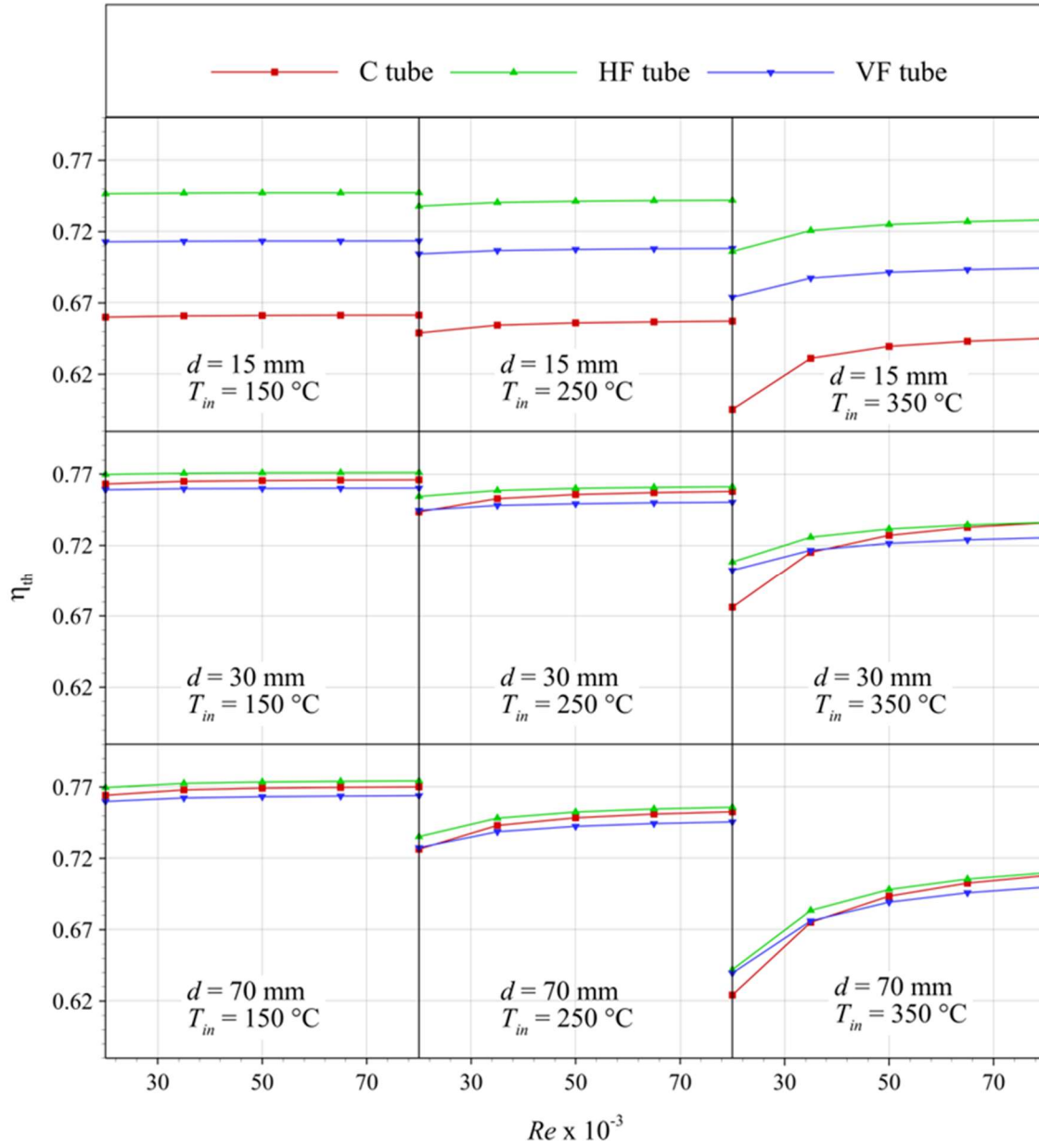


Figure 32 Variation of η_{th} with Re for C tube, HF tube, and VF tube at ($d=15$ mm, $d=30$ mm, and $d=70$ mm).

Figure 32 presents the variations of η_{th} with Re for different d values at T_{in} of 150 °C, 250 °C, and 350 °C. It is noticed that the system maintains consistent behavior, where an increase in Re leads to a rising trend in η_{th} , while a lower inlet temperature corresponds to

a reduction in η_{th} . The findings indicate that horizontal flat tube demonstrates a notable improvement in η_{th} compared to the circular and vertical flat tube across all cases. Meanwhile, the vertical flat tube outperforms the circular tube at $d = 15$ mm over the entire Re range. Additionally, it surpasses the circular tube at $Re = 20000$ for $d = 30$ mm and $d = 70$ mm, specifically at $T_{in} = 250$ °C and $T_{in} = 350$ °C. The horizontal flat tube demonstrates an average enhancement in η_{th} of approximately 9% at $d = 15$ mm, 0.5% at $d = 30$ mm, and 0.3% at $d = 70$ mm, while the vertical flat tube improved η_{th} by 5% in average at $d = 15$ mm. The findings indicate that the improvement in η_{th} is more pronounced at small base circle diameters. The observed enhancement can be attributed to two main reasons: (1) the increase in optical efficiency, which is most significant at small base circle diameters, and (2) the improved heat transfer in the horizontal flat tube. The vertical flat tube shows lower η_{th} values compared to the circular tube at $d = 30$ mm and $d = 70$ mm mainly due to the decreased optical efficiency.

4.4.6 Performance based on heat transfer-pumping power

To comprehensively evaluate the trade-off between thermal enhancement and hydraulic penalty, a new performance parameter is introduced: the Relative Benefit-Cost Ratio (RBCR). This dimensionless factor is defined as the ratio between the benefit from additional heat gain offered by the flat tube geometry and the additional cost associated with its increased pumping power, both compared to the conventional configuration. The RBCR is defined as:

$$RBCR = \frac{Q_{flat\ tube} - Q_{circular}}{(W_p)_{flat\ tube} - (W_p)_{circular}} \times \lambda_R \quad (58)$$

Here λ_R represents the price ratio between thermal energy and pumping power $\lambda_R = \Gamma_1/\Gamma_2$.

Mathematically, values of RBCR greater than unity indicate a net performance enhancement—where the thermal gain justifies or exceeds the hydraulic penalty—while values below unity denote a disadvantage.

To account for varying economic priorities, three distinct scenarios are considered, each defined by a different price ratio between heat gain and pumping power cost. These scenarios simulate systems where the thermal energy is:

- Scenario A ($\lambda_R = 0.1$): Pumping power is 10 times more costly than heat gain.
- Scenario B ($\lambda_R = 1$): Equal pricing weight is assigned to both heat gain and pumping power.
- Scenario C ($\lambda_R = 10$): Heat gain price is 10 times more valuable than the cost of pumping.

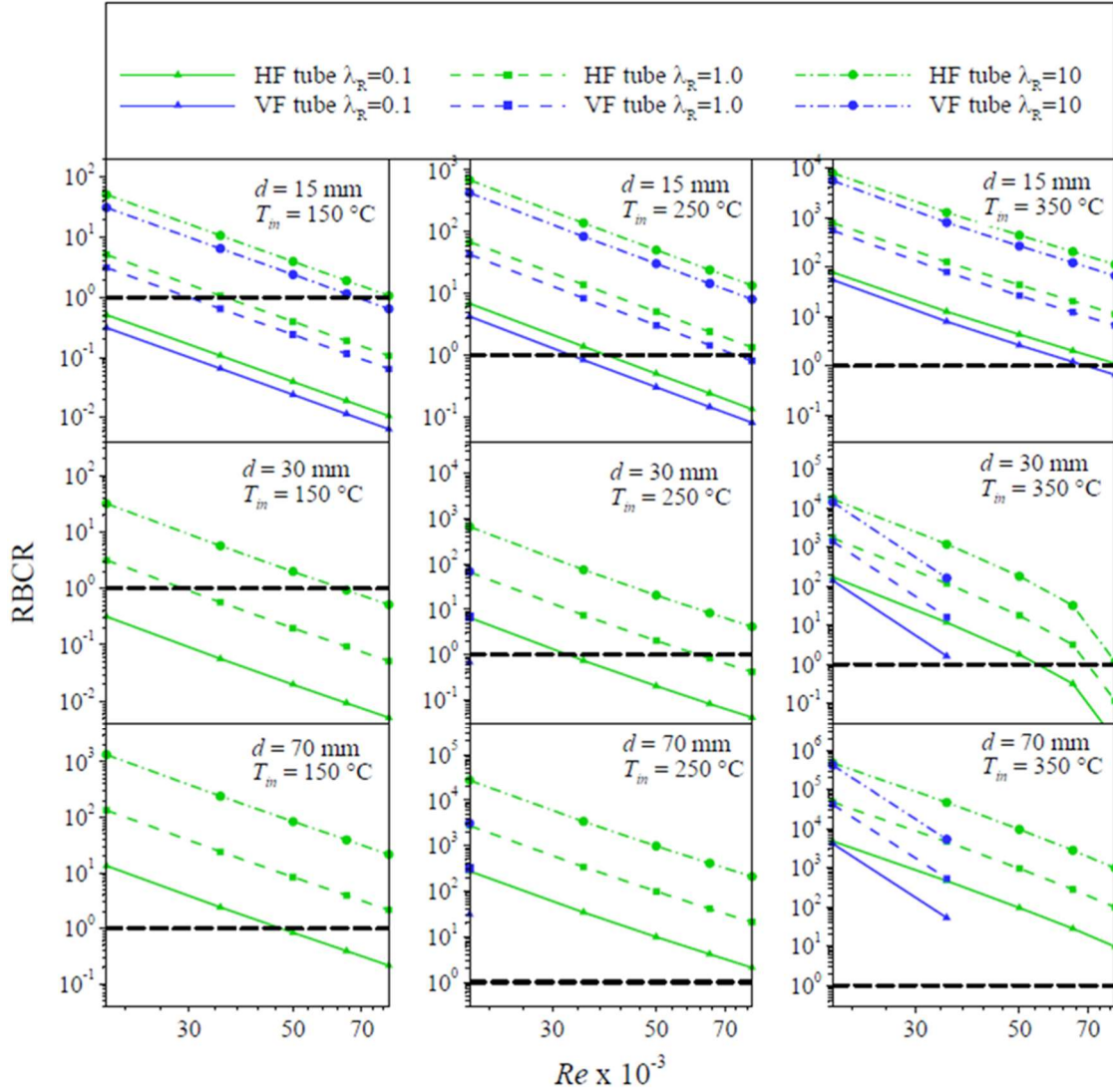


Figure 33 Variation of Relative Benefit-Cost Ratio (RBCR) with Reynolds number at ($d=15$ mm, $d=30$ mm, and $d=70$ mm) under various thermal-to-pumping price ratios ($\lambda_R = 0.1, 1, 10$)

Figure 33 illustrates the variation of the RBCR as a function of Reynolds number (Re) for different base diameters at different inlet temperatures of 150 °C, 250 °C, and 350 °C. Black dashed lines are included at $RBCR = 1$, serving as a threshold to distinguish favorable regions ($RBCR > 1$), from unfavorable regions ($RBCR < 1$). It is important to note that the plot employs logarithmic scaling on both axes. Consequently, any cases resulting in negative RBCR values are omitted from the graph, as they cannot be

represented on a long scale. At higher temperatures, horizontal (HF) and vertical (VF) consistently exhibit RBCR values greater than one across nearly all λ_R values, with few exceptions. This is primarily due to the substantial increase in thermal gain at higher temperatures, which offsets the pressure losses associated with these geometries.

It is notable that VF tubes often yield negative RBCR values, which appear as missing curves in the figure. This occurs because the circular design performs thermally better under these flow conditions. Additionally, in some low Re cases, the VF tube shows high RBCR values, attributed to enhanced heat transfer performance under low turbulence flow conditions. However, this advantage is generally limited to narrow operating ranges and diminishes at higher diameters or Re.

It can be seen from Figure 33 that the worst performance is observed at a price ratio of $\lambda_R = 0.1$ particularly at lower temperatures and smaller base diameters. At such low-price ratios, design optimization should prioritize minimizing pressure drop rather than maximizing heat gain. Furthermore, for price ratios of $\lambda_R = 1$ and $\lambda_R = 10$, the HF tube shows favorable performance in most cases. This suggests that the horizontal flat geometry becomes a cost-effective enhancement when the thermal benefit is equally or more important than pumping power.

Also, it is observed that largest diameter case $d = 70$ mm the HF tube consistently shows favorable RBCR in most scenarios. This is due to its ability to deliver high thermal performance while maintaining a comparable pressure drop to that of the circular geometry.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

5.1 Summary

This thesis presents a comprehensive investigation of parabolic trough collector (PTC) systems incorporating innovative flat tube absorber designs. The flat tube cross-section was generated from a conventional circular cross-section by defining a stretching ratio parameter. Two absorber designs were introduced depending on the direction of the circular cross-section stretching: the horizontal flat tube (HF tube) and vertical flat tube (VF tube). To evaluate the overall performance of these modified geometries the research is divided into two primary analysis: optical performance investigation and thermal-hydraulic performance investigation. In Chapter 3, a detailed analysis of the optical performance investigation has been presented. The MCRT method was used to investigate the effect of altering the conventional circular absorber to horizontal and vertical flat tubes with various stretching ratios along with other geometrical factors like the aperture width, focal length, and the base circular diameters. The investigation focused on the resulting flux distribution and optical efficiency.

Chapter 4 presented the thermal-hydraulic performance evaluation, which was numerically performed using three-dimensional finite element simulations. These simulations incorporated the flux profiles obtained from the optical analysis to ensure realistic and accurate boundary conditions. The study compared the heat transfer characteristics, friction factors, pumping power demands, and overall thermal efficiency of the horizontal flat tube

(HFT) and vertical flat tube (VFT) against the conventional circular tube across various inlet temperatures and Reynolds numbers.

The research demonstrated that the new absorber shapes can significantly enhance the overall performance of PTC systems, offering novel insights into the value of geometry-based design enhancements for advancing solar thermal technologies and highlight absorber shape optimization as a practical, cost-effective approach to improving PTC system efficiency.

5.2 Conclusion

The thesis concludes that flat tube absorbers can significantly enhance the performance of PTC systems. The main outcomes of this thesis can be concluded as follows

1. Optical performance (Chapter 3)

- Converting the absorber cross-section from circular to flat tube significantly altered the local concentration ratio (LCR) profiles. The horizontal stretching increased the maximum LCR values as the stretching ratio increased, while vertical stretching exhibited the opposite trend.
- At $d=70$ mm, horizontal flat tube achieved a slight optical efficiency improvement over circular absorbers, with a maximum increase of 0.52% at $a=1.6$, but also exhibited higher non-uniformity, increasing by 7.8%.
- Vertical flat tube absorbers showed a slight optical efficiency decrease at $d=70$ mm, with a maximum reduction of 0.56% at $a=1.6$, but significantly improved flux uniformity by reducing non-uniformity by 11.88%.

- The horizontal flat tube generally exhibited superior optical efficiency due to lower ray escape, while the vertical flat tube provided better flux uniformity, which could help mitigate localized thermal stress.
- At smaller base circular diameters, horizontal flat tubes showed notable optical efficiency improvements compared to circular absorbers; for example, 21.93% increase at $d = 10$ mm, 9.14% increase at $d = 15$ mm, and 1.66% increase at $d = 20$ mm. These results suggest that horizontal flat tubes are particularly advantageous for designs with smaller flow cross-sectional areas.
- While horizontal flat tubes improve optical efficiency, they come at the cost of increased irradiance non-uniformity, which may lead to localized thermal stress and hotspots affecting material durability and heat transfer.

2. Thermal-hydraulic performance (Chapter 4)

- All the configurations tested showed almost the same values of friction factors across all the operating conditions, indicating the cross-section of the absorber has almost no impact on dimensionless hydraulic performance.
- HF tube design demonstrated higher Nu compared to C tube and VF tube particularly at $d = 30$ mm. The highest Nu value is achieved by HF tube at $d = 30$ mm, $Re = 80000$, and $T_{in} = 150$ °C exceeding the values for the C tube and VF tube by 7.93% and 4.85%, respectively. Moreover, VF tube gave close Nu values with C tube in all cases except at $d = 15$ mm, where VF tube outperformed slightly. Overall, Nu was found to be

positively affected by Reynolds' number and negatively affected by increasing inlet temperature.

- HF tube achieved the highest overall thermal efficiency under all operating conditions. The average enhancements in η_{th} achieved by HF tube recorded 9% at $d = 15$ mm, 0.5% at $d = 30$ mm, and 0.3% at $d = 70$ mm compared to the conventional C tube. Furthermore, VF tube achieved improved overall thermal efficiency by an average of 5% compared to C tube at ($d = 15$ mm). However, at $d = 30$ mm and $d = 70$ mm, C tube outperformed VFT by showing slightly higher efficiencies by 0.5 % in average.
- The optical efficiency had the major contribution in overall efficiency at $d = 15$ mm, since it achieved more values in flat tube designs compared to the circular design because of the increased rays interception.
- The HF and VF tube designs exhibited higher pumping power requirements compared to the conventional circular (C) tube, despite having similar friction factors. This is primarily attributed to differences in hydraulic diameter, which influence the flow velocity at equivalent Reynolds numbers. Furthermore, the HF and VF configurations demonstrated nearly identical pumping power across all cases.
- The RBCR analysis confirms that the horizontal flat tube geometry offers superior thermal-economic performance across a wide range of operating conditions. This is especially evident at price ratios of $\lambda_R = 1$ and $\lambda_R = 10$, where the value of thermal gain price is equal to or significantly greater than the cost of pumping power.

5.3 Recommendations for future work

Building on the findings of this research, the following directions are recommended for future research:

- Conduct experimental studies to validate the numerical results and evaluate the practical feasibility of flat tube absorber designs under real-world operating conditions
- Extend the current work to include transient simulations that account for diurnal and seasonal variations in solar irradiance, enabling a more realistic assessment of performance over time.
- Extend the work to include the effect of optical and structural constraints such as installation errors, tracking mechanisms, thermal stresses, mechanical stability, and manufacturing limitations, to ensure the practical viability of the proposed designs.
- Perform coupled exergetic, environmental, or economic analyses to support comprehensive system-level optimization and decision-making.
- Explore the combination of flat tube absorbers with thermal energy storage systems to evaluate long-term system efficiency.
- Investigate the integration of other thermal enhancement techniques such as using advanced working fluids and incorporating turbulators.

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Vitae

Name : Mohammed Khalifa Mohamed Saadeldin

Nationality : Sudan

Date of Birth : 22/10/1998

Email : mkhalifauofk1998@gmail.com

Address : Khartoum, Sudan

Academic Background : **M.Sc. in Mechanical Engineering**
May 2025
King Fahd University of Petroleum and Minerals,
Dhahran- 31261, Saudi Arabia
B.Sc. in Mechanical Engineering
December 2021
University of Khartoum,
Althawra, Omdurman, P.O. Box: 11121, Khartoum, Sudan.

Research

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