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جامعة وهران 2 محمد بن أحمد
University Oran 2 Mohamed Ben Ahmed
معهد الصيانة و الأمن الصناعي
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**Numerical Simulation of Parabolic Trough Solar
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Prepared and publicly defended by :

SEKKIOU Sofiane El Hadj and **MEKRI Abdelmonaim**

In front of the Jury composed of :

Full name	Grade	Establishment	Quality
Réda Yahiaoui	MCA	University Oran 2	President
Dar Ramdane Mohamed Zouhir	MCB	University Oran 2	Supervisor
Zineb Hemami	MCA	University Oran 2	Examiner

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معهد الصيانة والأمن الصناعي
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et

MEKRI Abdelmonaim

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Réda Yahiaoui	MCA	Université d'Oran 2	Président
Dar Ramdane Mohamed Zouhir	MCB	Université d'Oran 2	Encadreur
Zineb Hemami	MCA	Université d'Oran 2	Examinatrice

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Dedication

With all pride and gratitude, i dedicate this work to my beloved mother and father, to those who have been the light of my path and my support in every step, to my siblings and friends who were by my side at every moment, and also to the supervising professor Mr Darramdane mohamed Zouhir, who contributed greatly to the completion of this work.

MEKRI Abdelmonaim

Dedication

With deep pride and heartfelt gratitude, i dedicate this work to my dear mother and father, whose unwavering support and guidance have illuminated my journey. I also extend this dedication to my siblings and friends and yp, who stood by me through every moment. A special thanks goes to my supervising professor, Mr. Darramdane Mohamed Zouhir, whose valuable contributions were essential to the completion of this work.

SEKKIOU Sofiane El Hadj

المخلص

تقدم هذه الأطروحة دراسة تحليلية لنظام مجمّع الحوض المكافئ بهدف تقييم أدائه الحراري في ظل ظروف واقعية. وقد تم تطوير نموذج رياضي باستخدام معادلات توازن الطاقة لتقييم الكفاءة البصرية وفقدان الحرارة وتوزيع درجة الحرارة على طول جهاز الاستقبال. تم تنفيذ النموذج باستخدام برنامج MATLAB وتم تطبيقه على دراسة حالة استنادًا إلى هندسة نظام مجمّع الحوض المكافئ المستخدم في محطة حاسي رمل للطاقة الشمسية في الجزائر. لضمان محاكاة واقعية، تم استخدام بيانات الطقس من محطة قمار. تم التحقق من صحة النموذج من خلال المقارنة مع النتائج العددية الحالية، مما يؤكد موثوقيته. استكشف التحليل مؤشرات الأداء الرئيسية مثل درجة حرارة المخرج، الحرارة المفيدة، والكفاءة الحرارية، وبحث في تأثير الطلاءات الانتقائية المختلفة لأنبوب الاستقبال. أظهرت النتائج أن الطلاءات ذات الامتصاصية العالية والانبعاثات المنخفضة تحسن بشكل كبير من كفاءة النظام، والتي تراوحت بين 72% و85%. تسلط هذه الدراسة الضوء على فعالية النمذجة التحليلية كأداة عملية لمحاكاة وتحسين أنظمة التجميع الشمسي بالحوض المكافئ في التطبيقات الحرارية الشمسية.

Abstract

This dissertation presents an analytical study of a Parabolic Trough Collector (PTC) system aimed at evaluating its thermal performance under realistic conditions. A mathematical model was developed using energy balance equations to assess optical efficiency, heat losses, and temperature distribution along the receiver. The model was implemented in MATLAB and applied to a case study based on the geometry of the PTC system used at the Hassi R'Mel solar power plant in Algeria. To ensure realistic simulation, weather data from the Guemar station were used. The model was validated through comparison with existing numerical results, confirming its reliability. The analysis explored key performance indicators such as outlet temperature, useful heat gain, and thermal efficiency, and investigated the impact of different selective absorber coatings. Results showed that coatings with high absorptance and low emittance significantly improve system efficiency, which ranged between 72% and 85%. This study highlights the effectiveness of analytical modeling as a practical tool for simulating and optimizing PTC systems in solar thermal applications.

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LIST OF ABBREVIATIONS

PTC	Parabolic Trough Collector
PCM	Phase Change Material
CSP	Concentrated Solar Power
PVT or PV/T	Photovoltaic/Thermal Collector
BIST	Building-Integrated Solar Thermal
FPC	Flat Plate Collector
ETC	Evacuated Tube Collector
CPC	Compound Parabolic Collector
LFC	Linear Fresnel Collector
HFC	Heliostat Field Collector
PDC	Parabolic Dish Collector
HTF	Heat Transfer Fluid
DSG	Direct Steam Generation
MED	Multi-Effect Distillation
RO	Reverse Osmosis
MWCNT	Multi-Walled Carbon Nanotubes
HP	High-Pressure
LP	Low-Pressure
EES	Engineering Equation Solver
SEGS	Solar Electric Generating System
MATLAB	Mathematical Laboratory
DHC	District Heating and Cooling
HCE	Heat Collector Element

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NOMENCLATURE

A	: Surface area (m^2)
A_{abs}	: Absorber area (m^2)
A_{ap}	: Aperture area (m^2)
a	: Accommodation coefficient
b	: Interaction coefficient
C	: Concentration ration
C_{flux}	: Radiation-flux concentration ratio
C_{geom}	: Geometrical concentration ratio
c_p	: Specific heat at constant pressure of the HTF
D	: Diameter of the absorber (m)
D_2	: Inside diameter of the absorber (m)
D_3	: Outside diameter of the absorber (m)
D_4	: Outside diameter of the glass envelope (m)
D_5	: Outside diameter of the glass envelope (m)
F'	: Collector efficiency factor
F_R	: Heat removal factor
f	: Focal length (m)
g	: Gravitational constant (m^2/s)
h	: Convection heat transfer coefficient ($W/m^2.K$)
h_1	: Convection Heat transfer of HTF at T1 ($W/m^2.K$)
h_{1c}	: Convection heat transfer between absorber and glass envelope ($W/m^2.K$)

h_{1r} : Radiation between absorber and glass envelope ($W/m^2.K$)
 h_{2c} : Convection heat transfer between absorber and glass envelope ($W/m^2.K$)
 h_{1c} : Radiation heat transfer between glass envelope and sky ($W/m^2.K$)
 h_{34} : Convection heat transfer coefficient of annulus gas at T_{34} ($W/m^2.K$)
 h_{56} : Convection heat transfer coefficient of air at T_{56} ($W/m^2.K$)
 h_f : Convection heat transfer of the HTF ($W/m^2.K$)
 I_{abs} : Incident beam solar irradiance on the absorber (W/m^2)
 I_{ap} : Incident beam solar irradiance on the aperture (W/m^2)
 I_b : Beam solar irradiance (W/m^2)
 $K(\theta_i)$: Incident angle modifier
 k : Thermal conductivity of the wall ($W/m.K$)
 k_1 : Thermal conductance of HTF at T_1 ($W/m.K$)
 k_{23} : Thermal conductance of absorber wall at T_{23} ($W/m.K$)
 k_{45} : Thermal conductance of glass envelope at T_{45} ($W/m.K$)
 k_{56} : Thermal conductance of air at T_{56} ($W/m.K$)
 k_f : Thermal conductance of the HTF ($W/m.K$)
 k_{std} : Thermal conductance of the annulus gas at standard temperature and pressure ($W/m.K$)
 L : Collector and absorber length (m)
 $\dot{m}$: Mass flow rate of HTF (Kg/s)
 Nu_{D2} : Nusselt number of HTF based on D_2
 Nu_{D5} : Nusselt number of HTF based on D_5
 P_a : Annulus gas pressure (bar)
 Pr_{56} : Prandtl number for air at T_{56}
 $\dot{q}_{ab}$: Total solar power absorbed by the absorber (W)
 $\dot{q}_{loss}$: Thermal energy losses (W)
 $\dot{q}_u$: Useful thermal energy gain (W)
 $\dot{q}_{cond}$: Conduction heat transfer (W)
 $\dot{q}_{conv}$: Convection heat transfer (W)

$\dot{q}_{rad}$: Radiation heat transfer (W)

$\dot{q}'_{12conv}$: Convection heat transfer rate between the HTF and inside wall of the absorber pipe per unit absorber length (W/m)

$\dot{q}'_{23cond}$: Conduction heat transfer rate through the absorber pipe wall per unit absorber length (W/m)

$\dot{q}'_{34conv}$: Convection heat transfer rate between the outer surface of the absorber pipe to the inner surface of the glass envelope per unit absorber length (W/m)

$\dot{q}'_{34rad}$: Radiation heat transfer rate between the outer surface of the absorber pipe to the inner surface of the glass envelope per unit absorber length (W/m)

$\dot{q}'_{SolAbs}$: Solar irradiation absorption rate into the absorber pipe per unit absorber length (W/m)

$\dot{q}'_{45cond}$: Conduction heat transfer rate through the glass envelope per unit absorber length (W/m)

$\dot{q}'_{56conv}$: Convection heat transfer rate between the outer surface of the glass envelope to the atmosphere per unit absorber length (W/m)

$\dot{q}'_{57rad}$: Radiation heat transfer rate between the outer surface of the glass envelope to the sky per unit absorber length (W/m)

$\dot{q}'_{5SolAbs}$: Solar irradiation absorption rate into the absorber pipe per unit absorber length per unit absorber length (W/m)

$\dot{q}'_{HeatLoss}$: Total heat loss rate from the heat collecting element to the surroundings (W/m)

$\dot{q}''$: Absorbed solar radiation by the absorber (W/m^2)

Ra_{D5} : Rayleigh number of air based on D_5

r : Mirror radius (m)

r_r : Mirror rim radius (m)

T : Radiation surface temperature (K)

T_1 : Mean temperature of the HTF (K)

T_2 : Absorber pipe inner surface temperature (K)

T_{23} : Average absorber wall temperature, $(T_2 + T_3)/2$ (K)

T_3 : Absorber pipe outer surface temperature (K)

T_{34} : Average temperature of annulus gas, $(T_3 + T_4)/2$ (K)

T_4 : Glass envelope inner surface temperature (K)

T_5 : Glass envelope outer surface temperature (K)

T_{56} : Average temperature of air, $(T_5 + T_6)/2$ (K)
 T_6 : Ambient temperature (K)
 T_7 : Sky temperature (K)
 T_∞ : Fluid mean temperature (K)
 T_a : Wall surface temperature (K)
 T_{air} : Air temperature (K)
 T_b : Wall surface temperature (K)
 T_g : Glass envelope temperature (K)
 T_{fi} : Inlet fluid temperature (K)
 T_{fo} : Inlet fluid temperature (K)
 T_r : Absorber temperature (K)
 T_w : Surface plate temperature (K)
 U_l : Overall heat loss coefficient ($W/m^2 \cdot K$)
 W_a : Aperture idth (m)
 X_{end} : End loss factor
 x : Wall thickness (m)
 α_{abs} : Absorber absorptance
 α_{56} : Thermal diffusivity for air at T_{56} (m^2/s)
 β : Volumetric thermal expansion coefficient of air ($1/K$)
 γ : Intercept factor
 γ_g : Ratio of specific heats for the annulus gas
 δ : Molecular diameter of annulus gas (cm)
 ε_3 : Emissivity of absorber selective coating emissivity
 ε_4 : Emissivity of inner surface of glass envelope
 ε_5 : Emissivity of outer surface of glass envelope
 η_0 : Optical efficiency
 η : Thermal efficiency
 θ : Angle between reflected beam and collector axis (degree)
 λ : Mean-free-path between collisions of a molecule (cm)

ν_{56} : Kinematic viscosity of air at T_{56} (m^2/s)

π : Mathematical constant 3.14.

ρ_a : Mirror reflectance

σ : Stefan-Boltzmann constant ($W/m^2.K^4$)

τ_g : Glass envelope transmittance

θ_i : Incidence angle (degree)

θ_m : Half-angle of acceptance (degree)

θ_r : Rim angle (degree)

GENERAL INTRODUCTION

The increasing urgency to transition toward sustainable energy sources has placed renewable technologies at the forefront of global energy strategies. Solar energy, in particular, stands out due to its abundance, availability, and potential to significantly reduce greenhouse gas emissions. Among the various technologies developed to harness solar energy, solar thermal systems play a crucial role, especially in medium- to high-temperature applications. Within this domain, Parabolic Trough Collectors (PTCs) have emerged as one of the most commercially mature and widely adopted solutions for large-scale solar thermal power generation. Their modular design, high optical efficiency, and compatibility with thermal energy storage make them a preferred option for both industrial and utility-scale applications.

This thesis is dedicated to the analytical simulation of a PTC system, with the goal of evaluating its thermal performance under realistic operating conditions. Analytical modeling provides an efficient and accessible alternative to complex numerical simulations, offering valuable insights while reducing computational cost and complexity. The study is structured into four interconnected chapters, each building a step toward the final objective of simulating and understanding the thermal behavior of a PTC through a simplified yet robust approach.

Chapter I presents an overview of recent scientific and technological progress in solar thermal systems, with a focus on Parabolic Trough Collectors (PTCs). It highlights advancements in materials, collector design, and thermal performance, while emphasizing the growing interest in PTCs for large-scale solar applications.

Chapter II establishes the theoretical foundation of solar collectors, detailing their classifications, operating principles, and the specific optical and thermal mechanisms that govern PTC efficiency. This theoretical background is essential for understanding the physical behavior of the system and supports the development of the mathematical model.

In Chapter III, a detailed analytical model of the PTC is developed based on energy balance equations, optical efficiency, and heat loss mechanisms. The model integrates key parameters such as solar irradiance, thermal conductivity, and heat transfer coefficients, and is implemented in MATLAB to simulate system performance. Simplifying assumptions are applied to maintain feasibility while preserving the accuracy needed for meaningful predictions.

Chapter IV applies the developed analytical model to a practical case study to assess the thermal performance of the PTC system. After validating the model through comparison with existing numerical results, the chapter analyzes key parameters such as outlet temperature, useful heat gain, and efficiency. It also explores how different absorber coatings influence thermal behaviors.

CHAPTER ONE

LITERATURE REVIEW

I.1 Introduction

This chapter provides an overview of the scientific and technological progress in solar thermal energy systems, with a particular focus on Parabolic Trough Collectors (PTCs). Recent research on solar collectors has emphasized improvements in design, materials, and thermal performance. Innovations such as hybrid photovoltaic/thermal (PVT) systems, nanofluids, and advanced thermal storage solutions have significantly boosted the efficiency and versatility of solar energy applications. These developments reflect the global push for cleaner energy alternatives and underscore the importance of ongoing research in this domain.

Special attention is given to PTC technology, which remains the most commercially developed and widely used type of concentrating solar collector. Studies in this area have focused on optimizing receiver design, minimizing heat losses, and improving thermal modeling accuracy. While numerical and experimental methods have provided valuable insights, many models remain limited by high computational demands or steady-state assumptions. This dissertation responds to those challenges by developing an analytical simulation model aimed at providing accurate, efficient, and practical performance evaluation of PTC systems. This literature foundation supports the analytical work presented in the following chapters.

I.2 Recent Research on Solar Collectors

Recent research on solar collectors has demonstrated significant advancements in design, materials, and performance, driven by the global need for renewable energy solutions.

Hybrid photovoltaic/thermal (PVT) systems have been a key focus, with flat-plate PVT collectors demonstrating improved efficiency through optimized absorber plate dimensions, fluid flow rates, and PV cell packing factors. Bifacial PVT collectors, where water-submerged panels reduce optical reflection and thermal drift, have enhanced photovoltaic output by 15%, making them highly effective for hybrid energy systems [1].

Material innovations are also prominent, with polymeric materials gaining attention due to their cost-effectiveness and lightweight properties. Biopolymer-based absorbers and nanocomposite films incorporating gold and silver nanoparticles have significantly boosted thermal efficiency while reducing manufacturing costs [2].

Nanofluids such as multi-walled carbon nanotubes (MWCNTs) have emerged as superior heat transfer fluids. Studies reveal that MWCNTs with optimized pH values (3.5, 6.5, and 9.5) achieve higher thermal efficiencies, with 0.4 wt% MWCNT reaching an impressive efficiency of 0.9, outperforming conventional fluids like Al₂O₃-H₂O [3].

Thermal energy storage technologies have advanced, with molten salts and phase change materials (PCMs) enhancing the storage and release of thermal energy, ensuring consistent energy supply during periods of low solar radiation [1].

Concentrating collectors, including heliostat fields, parabolic dishes, and troughs, benefit from innovations in optical optimization, heat loss reduction, and heat recuperation, making them highly efficient for large-scale solar power generation [1].

Solar dryers have become essential for food preservation in developing countries, addressing food security challenges through efficient drying techniques powered by solar energy [3].

Solar power plants have also advanced, utilizing thermal energy storage systems to provide reliable electricity generation even during non-sunny periods. Recent research emphasizes rigorous testing and standards for solar collectors, including rupture, stagnation temperature, thermal shock, and mechanical load tests, ensuring their durability and reliability under various environmental conditions [2].

I.3 Research on Parabolic Trough Collectors (PTC)

Parabolic trough collectors (PTCs) are one of the most widely used concentrated solar power (CSP) technologies, particularly in large-scale thermal power plants. A PTC consists of a parabolic mirror, a receiver tube, a heat transfer fluid (HTF), and a tracking system that follows the Sun to maximize energy capture. These systems are modular and are often deployed in solar power plants, where they operate alongside heat transfer systems, steam turbines, and thermal storage units for continuous energy generation [4].

Research on PTCs has focused on enhancing efficiency and performance, particularly through numerical modeling, experimental validation, and heat transfer optimization. The following studies highlight some of the most significant contributions to this field:

- **Ouagued** [5]. Developed a numerical model to estimate temperature distribution, heat gain, and heat loss in a parabolic trough collector under Algerian climate conditions. Their study compared the performance of different heat transfer fluids (HTFs) and found that oil-based fluids significantly improved thermal efficiency compared to air-based alternatives, making them more suitable for high-efficiency applications.

- **Archer** [6]. Conducted an experimental and numerical study of a PTC used in high-temperature solar cooling and heating applications revealed notable differences between theoretical and experimental outcomes. These discrepancies were mainly attributed to thermal losses at the receiver supports and inaccurate assumptions about the absorber's optical properties, particularly absorptivity. The results underscore the need for enhanced thermal modeling to ensure more precise performance predictions.
- **Gong** [7]. Optimized the design of a high-temperature parabolic trough receiver in China by improving receiver coatings and insulation techniques. Their study demonstrated that these enhancements significantly reduced radiative and convective heat losses, leading to a more efficient system for large-scale solar power generation.
- **Hachicha** [8]. Developed a numerical model that incorporated conduction, convection, and radiation heat transfer mechanisms to predict temperature variations in a PTC receiver. Their findings, validated against experimental data from Sandia National Laboratories, showed that small errors in optical properties and heat transfer coefficients could significantly affect performance predictions.
- **Coccia** [9]. Formulated a mathematical model for evaluating optical, thermal, and global efficiency in a prototype PTC. Their experimental validation revealed that receiver tube emissivity, glass envelope transmittance, and coating absorptance were critical factors influencing overall system efficiency. Their model achieved a prediction error of only 5%, confirming its accuracy.
- **Dudley** [10] Conducted comprehensive experimental evaluations on PTCs named SEGS LS-2, analyzing how coating materials, vacuum insulation, and optical alignment affected efficiency. Their study revealed that vacuum loss in the receiver tube reduced efficiency by over 10%, while cermet coatings outperformed black chrome coatings, providing higher absorptivity and lower heat losses.
- **Forristall** [11] developed a detailed heat transfer model for a parabolic trough receiver using Engineering Equation Solver (EES). His analysis included convective heat transfer correlations, and validation against Sandia National Laboratories' data showed that optimizing fluid velocity and tube geometry could significantly enhance thermal performance.
- **Padilla** [12] Proposed a one-dimensional numerical model to study heat transfer in a PTC receiver. By dividing the receiver tube into multiple axial segments, their model successfully captured temperature variations along the tube length, with an error margin of less than 4% when compared to experimental data.

- **Kalogirou** [13] developed a comprehensive thermal model for a PTC receiver, incorporating conduction, convection, and radiation effects. His simulations under various operational conditions demonstrated that receiver tube emissivity and solar absorptance were the most influential parameters affecting thermal efficiency. Validation with real-world CSP plant data confirmed the model's applicability.
- **Yaghoubi** [14] Performed a combined experimental and numerical study on heat loss mechanisms in a 250 kW PTC power plant. Their findings indicated that vacuum degradation increased heat loss by 40%, leading to an efficiency drop of 3-5%. Their study underscored the importance of maintaining vacuum insulation to minimize energy losses.
- **Ghodbane** [62] conducted an optical and thermal performance study on a parabolic trough solar concentrator installed in the Guemar region of El-Oued, southeastern Algeria. SolTrace was used for optical modeling, while MATLAB solved the nonlinear thermal equations using the implicit finite difference method. The system was evaluated over six representative days, and results showed thermal efficiencies exceeding 82%, highlighting the concentrator's effectiveness for high-temperature applications in arid environments.

While PTC research has made significant progress, many studies rely on simplified heat transfer models and primarily consider steady-state conditions, which do not fully capture the dynamic variations in solar radiation, ambient temperature, and operational parameters. To address these limitations, this work develops an analytical model implemented in a computational program to provide a more accurate prediction of temperature distribution, heat loss estimation, and overall performance analysis of PTC systems.

I.4 Objective of the Study

This dissertation contributes to the field of solar thermal engineering by presenting a practical and effective analytical method for simulating the performance of parabolic trough collectors. It bridges the gap between theoretical understanding and practical application, offering a computationally efficient tool to support the design, analysis, and optimization of solar thermal systems. The work aims to develop an analytical model for evaluating the thermal performance of parabolic trough collector systems, contribute to the enhancement of their efficiency through accurate numerical modeling and analysis, and examine the impact of key operating parameters—such as solar radiation intensity, ambient temperature, and heat transfer fluid properties—on overall system performance.

I.5 Conclusion

In summary, this chapter has presented a thorough review of the existing literature on solar collectors, with a particular focus on Parabolic Trough Collectors (PTCs). The discussion began with an overview of solar collector technologies, outlining their evolution, classifications, and areas of application. Recent advancements in solar collector research were highlighted, emphasizing innovations in design, materials, and performance optimization. Special attention was given to PTCs, examining their development, ongoing research efforts, and growing role in large-scale solar thermal applications. The objective of this study was then positioned within the context of the reviewed literature, identifying key knowledge gaps and the need for further investigation. This literature foundation provides the necessary background for the following chapter, which delves into the theoretical concepts underlying the operation and performance of PTC systems.

CHAPTER TWO

SOLAR COLLECTORS AND PARABOLIC TROUGH COLLECTORS

II.1 Introduction

In this chapter, we delve into the theoretical foundations and essential principles underpinning the design and performance of solar thermal systems, with a specific focus on Parabolic Trough Collectors (PTCs). As renewable energy continues to gain prominence in the global energy landscape, solar thermal technologies have emerged as reliable and efficient solutions for harnessing solar energy. PTCs, in particular, have proven to be among the most effective technologies for medium- to high-temperature applications due to their superior optical and thermal efficiency.

II.2 Solar Collectors

Solar thermal collectors are devices used for converting solar radiation into thermal energy, transporting it to a storage device for later use. The system can be characterized by natural or forced circulation. Solar thermal systems are typically used to produce hot water or zone heating, but they can also be used for different purposes [15]. The theoretical limit of energy efficiency of a solar thermal panel is represented by the amount of solar radiation that it can intercept and absorb. Several types of solar thermal collectors are available nowadays for different applications. The architectural integration of solar collectors has led to Building-Integrated Solar Thermal (BIST) systems. figure 2.1. provides the solar systems classification, distinguishing non-concentrating and concentrating systems. Hybrid systems (represented by Photovoltaic/Thermal collectors, called briefly PV/T collectors) are also represented in figure 2.1, non-concentrating and concentrating systems are characterized by different colors, also used in the hybrid system category. Non-concentrating collectors are generally used in residential and commercial buildings for space heating, while concentrating collectors are typically used in solar power plants. Table 2.1 lists the energy applications and the different types of solar collectors used.

Table 2.1 Solar thermal collectors used for different applications [16]

Application	Solar thermal collector type	Circulation
Solar water heating	<ul style="list-style-type: none"> - FPC - ETC - CPC 	Natural/Forced
Space heating and cooling		Forced
Solar refrigeration		
Industrial process heat		
Simultaneous generation of electricity and heat	- PV/T Collector	Forced

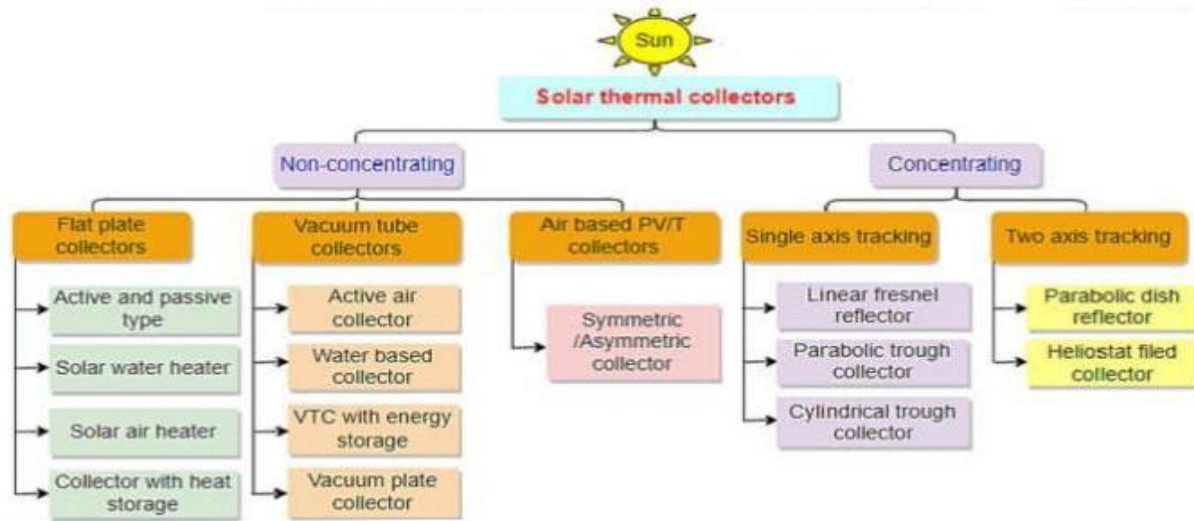


Figure 2.1 Solar thermal collectors classification. [48]

II.3 Types of Solar Collectors

II.3.1 Flat Plate Collectors Using Liquid

Flat-plate collectors are among the most commonly used and studied systems for domestic hot water production [17]. They operate by capturing solar radiation on a dark-colored or selectively coated surface, which then transfers the absorbed heat to a working fluid, usually water. A typical flat-plate collector is composed of several key parts: a black absorber surface that collects solar energy, a transparent cover that permits sunlight to enter while reducing heat loss, a network of fluid channels, a structural frame providing mechanical support, and insulation to minimize thermal losses. To enhance efficiency, the absorber surface is often treated with a selective coating, as shown in figure 2.2.

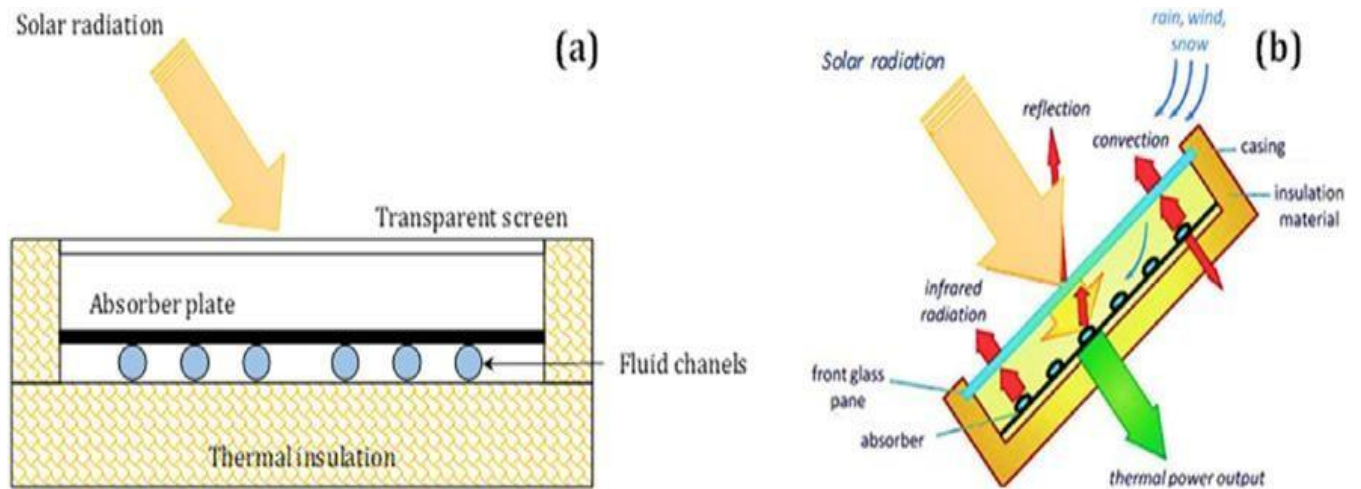


Figure 2.2 Flat-plate thermal collector: (a) basic schema; (b) heat transfer phenomena [18].

II.3.2 Unglazed Collectors

Unglazed solar collectors, illustrated in figure 2.3, are distinguished by the absence of a protective glass cover over the absorber. As these collectors are not insulated, they are mainly used for low-temperature applications, typically below 30°C. They are usually constructed from black plastic materials stabilized against ultraviolet degradation. While the lack of glazing allows for a high rate of solar absorption, it also results in significant heat losses, especially under windy or cold conditions [16]. Consequently, their efficiency is generally lower than that of glazed flat-plate collectors. Unglazed collectors are commonly used for applications like heating outdoor swimming pools.

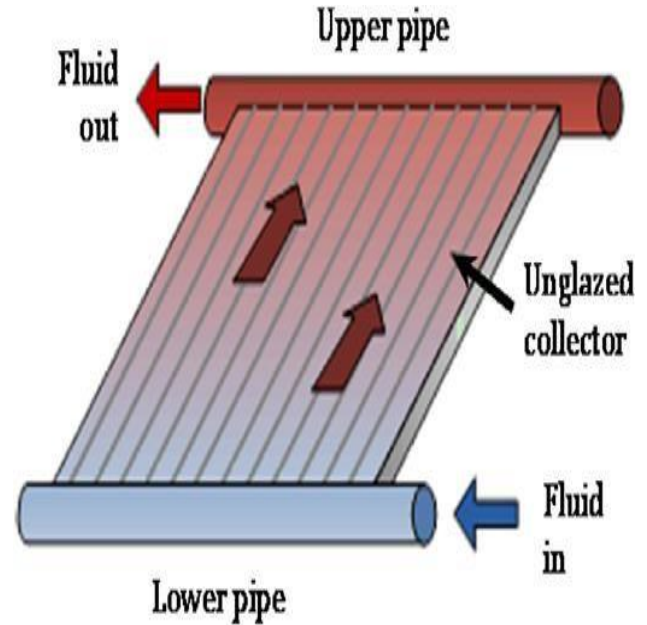


Figure 2.3 Representation of an unglazed solar collector [19]

II.3.3 Evacuated Tube Collectors

Evacuated tube collectors feature a vacuum-sealed space between the absorber and the outer transparent tube, reducing thermal losses. They are particularly suited for medium-temperature applications, typically operating between 60°C and 120°C, depending on ambient conditions [17]. Each tube generally contains a secondary inner tube, coated with a dark selective material to maximize solar absorption, as represented in figure 2.4.

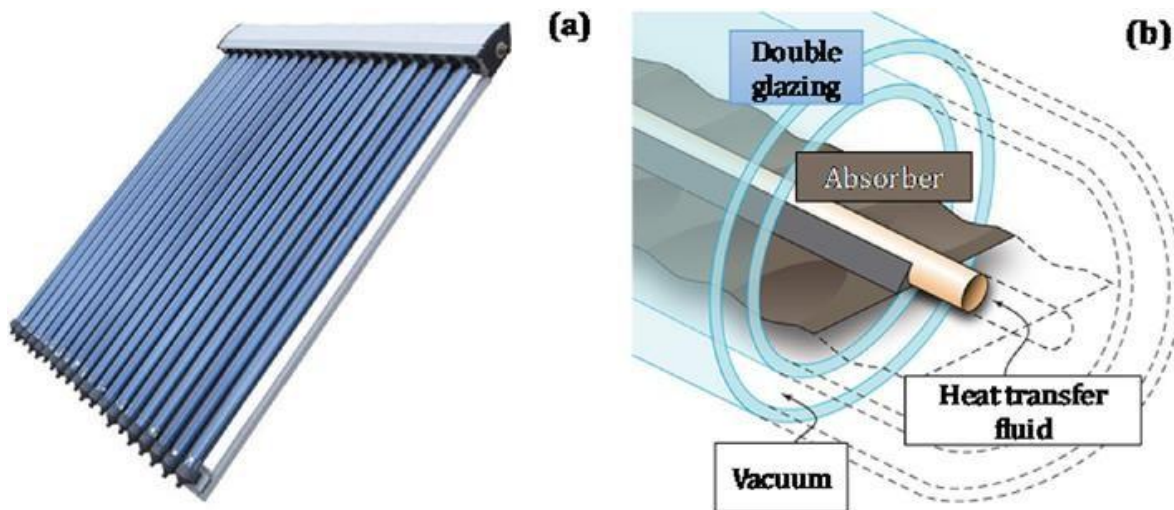


Figure 2.4 (a) Evacuated tube collectors; (b) representation of a concentric tube.

There are three main types of evacuated tube collectors [20][21], depicted in figure 2.5.

Water-in-Glass Collectors: These collectors consist of single-ended water-filled tubes connected to a horizontal storage tank. Each tube comprises two concentric glass layers, with a vacuum between them, and a selective coating on the internal tube's outer surface. Water heats up in the tubes and rises naturally, while cooler water from the tank replaces it, as shown in figure 2.5a.

U-Type Collectors: In this design, the heat transfer fluid directly circulates through a U-shaped metal absorber (commonly copper), placed inside a vacuum tube. These absorbers, often enhanced with fins and coated selectively, are inserted into an evacuated glass tube. The tubes are interconnected as shown in figure 2.5b.

Heat-Pipe Collectors: These systems use a sealed copper heat pipe filled with an evaporative fluid, typically an alcohol-based solution. As the fluid heats up, it evaporates and travels along the pipe, transferring heat to the circulating fluid within the manifold, before condensing and returning. figure 2.5c illustrates this configuration.

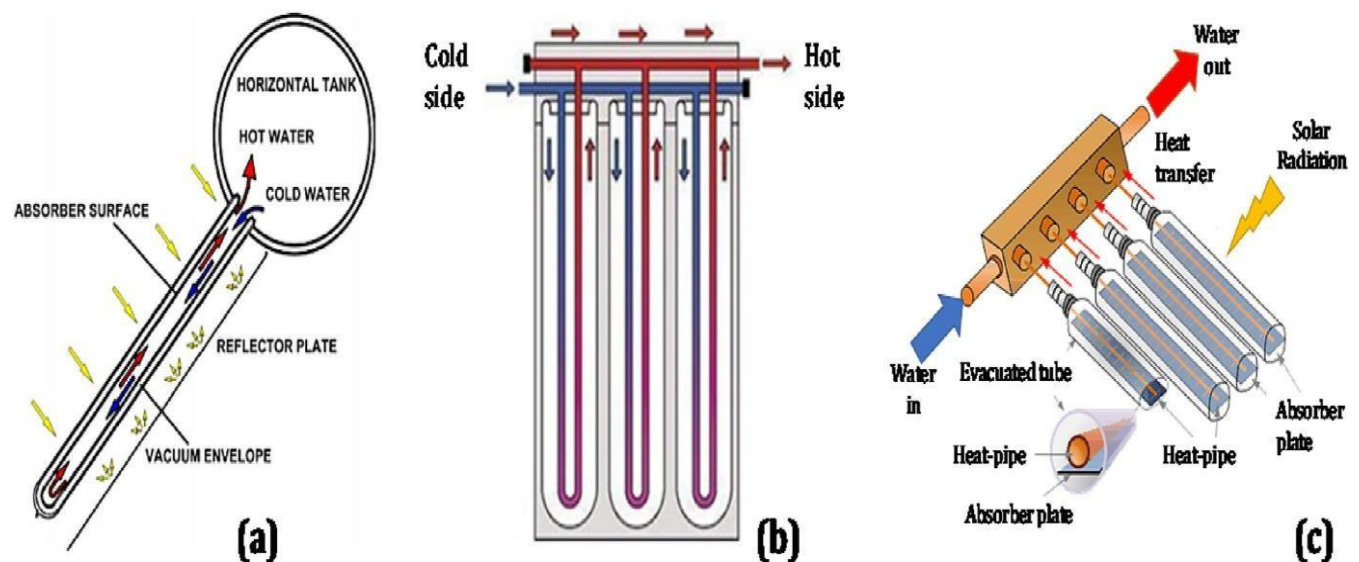


Figure 2.5 Representations of (a) a water-in glass collector [22], (b) of a U-type collector and (c) of a heat-pipe collector [23].

II.3.4 Solar Thermal Collectors Using Air

Solar air collectors utilize air as the working fluid for heat transfer. The heated air can serve domestic heating needs, power gas turbines, or support industrial processes such as drying [24]. Air circulation can occur either between the transparent cover and the absorber or beneath the absorber surface itself. To improve heat transfer, absorbers are often equipped with fins [25]. Structurally, these collectors resemble flat-plate systems, consisting of an absorber plate and a transparent cover secured by a frame. No storage tanks are needed, and the heated air is typically distributed immediately using fans. Some systems incorporate photovoltaic (PV) panels to power the fans, ensuring operation without external electricity supply. A schematic is provided in figure 2.6.

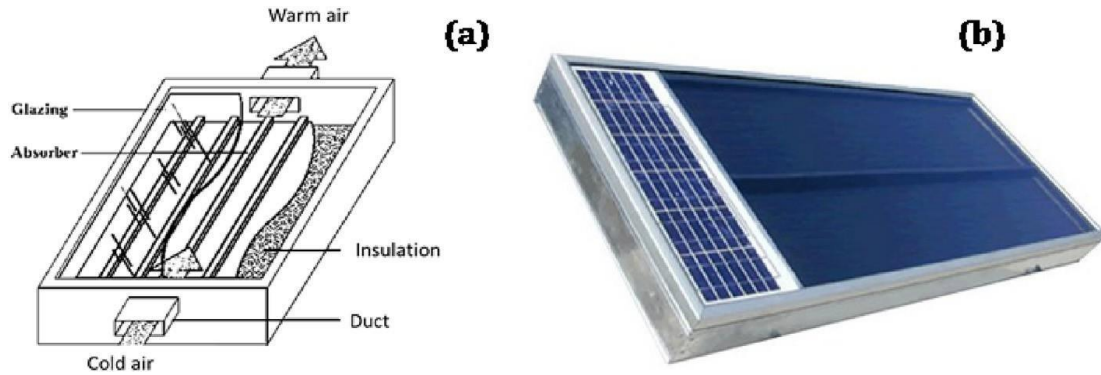


Figure 2.6 Representations of (a) solar thermal collectors using air and (b) a commercial product [25].

II.3.5 Concentrating Solar Collectors

Concentrating solar collectors focus the incident radiation on a smaller area than the receiving one using optical concentrators. The existing types of concentrating solar collectors are described below:

Flat Concentrator Collectors (FPC): These simple systems are flat-plate collectors enhanced with reflective panels attached to their sides, which direct additional sunlight onto the absorber surface (figure 2.7a).

Compound Parabolic Concentrators (CPC): In these collectors, a parabolic reflective sheet beneath each tube redirects sunlight onto the absorber. This design increases the amount of solar energy captured without enlarging the collector's gross area. figure 2.7b illustrates the system.

Parabolic Trough Collectors (PTC): Featuring a parabolic reflector focusing sunlight onto a central receiver tube, these collectors achieve high fluid temperatures between 50°C and 400°C. They require single-axis solar tracking to maintain optimal alignment throughout the day. Typical working fluids include thermal oils, water mixtures, or molten salts (figure 2.7c).

Heliostat Field Collectors (HFC): Comprising an array of mirrors (heliostats) that track the sun and concentrate its energy onto a central receiver, often located atop a tower. The system heats a working fluid, such as molten salt or water, in a closed-loop arrangement (figure 2.7d).

Linear Fresnel Collectors (LFC): These systems use long, flat, or slightly curved mirrors to focus sunlight onto a stationary receiver. The mirrors are arranged horizontally and require simpler tracking mechanisms (figure 2.7e).

Parabolic Dish Collectors (PDC): Shaped like a satellite dish, these collectors focus sunlight onto a receiver positioned at their focal point. They employ dual-axis tracking for precise solar alignment (figure 2.7f).

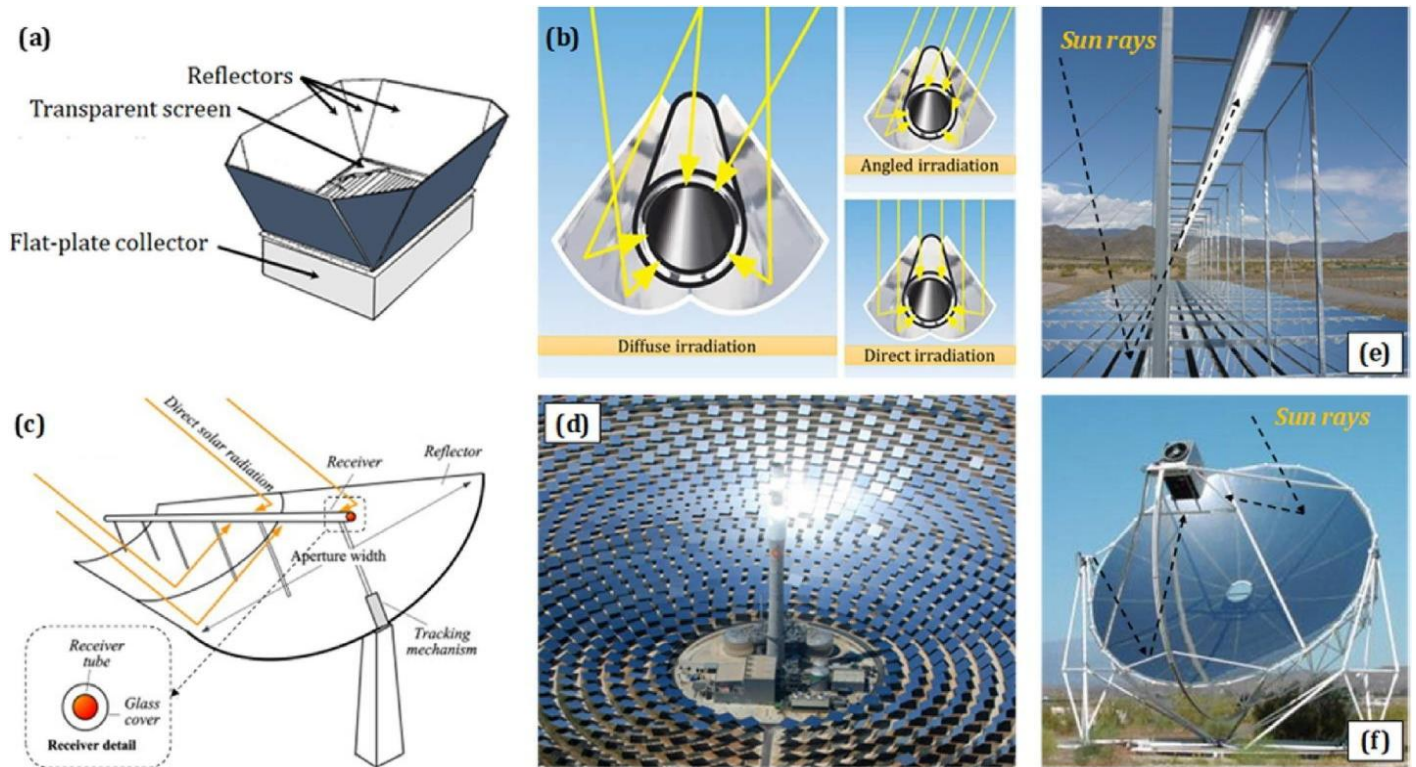


Figure 2.7 Representations of the flat concentrator collector technique (a) and the compound parabolic concentration technique (b); parabolic trough collector [26] (c); actual heliostat field collector [27] (d); actual linear Fresnel collector (e); actual parabolic dish collector (f)

II.3.6 Photovoltaic/Thermal Collectors

Photovoltaic/thermal (PV/T) collectors combine photovoltaic and solar thermal technologies into a single unit. These hybrid systems generate both electricity and heat, enhancing overall solar energy conversion efficiency (figure 2.8). Depending on the design, either air or liquid can be used to remove the heat generated by the photovoltaic cells.

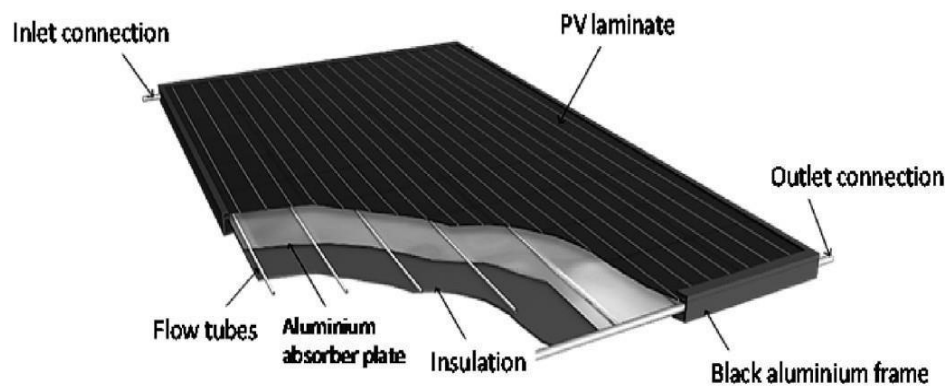


Figure 2.8 Representation of a commercial PV/T panel.

II.4 Parabolic Through Collector (PTC)

A Parabolic Trough Collector (PTC) is a widely used concentrated solar power (CSP) technology, designed to focus direct solar radiation onto a receiver tube containing a heat transfer fluid (HTF). figure 2.9 shows that PTC consists of a parabolic reflector, a receiver tube, a tracking system, and a structural framework, ensuring efficient solar energy conversion [28] [29]. The choice of materials plays a crucial role in PTC performance, with glass mirrors commonly used for their high reflectivity and durability, while copper and aluminum tubes provide excellent thermal conductivity for heat absorption [30].

PTCs are essential in solar power generation, industrial process heating, and desalination, offering low emissions, reduced dependence on fossil fuels, and long-term cost savings. Their high efficiency is further enhanced by technological improvements such as selective coatings, vacuum insulation, and advanced tracking mechanisms, which help reduce optical and thermal losses. Additionally, PTCs play a key role in industrial sectors requiring medium-to-high-temperature heat, such as chemical processing, food production, and water desalination, making them an essential technology for sustainable energy development [28].



Figure 2.9 Parabolic trough collector [44]

II.4.1 Components and Structural Design of a Parabolic Trough Collector

A PTC consists of several essential components as shown in figure 2.10 that work together to efficiently convert solar energy into thermal energy. Each component plays a crucial role in ensuring optimal energy absorption, heat transfer, and overall system efficiency. Additionally, the structural design, including the tracking system and support structure, plays a vital role in enhancing performance and stability.

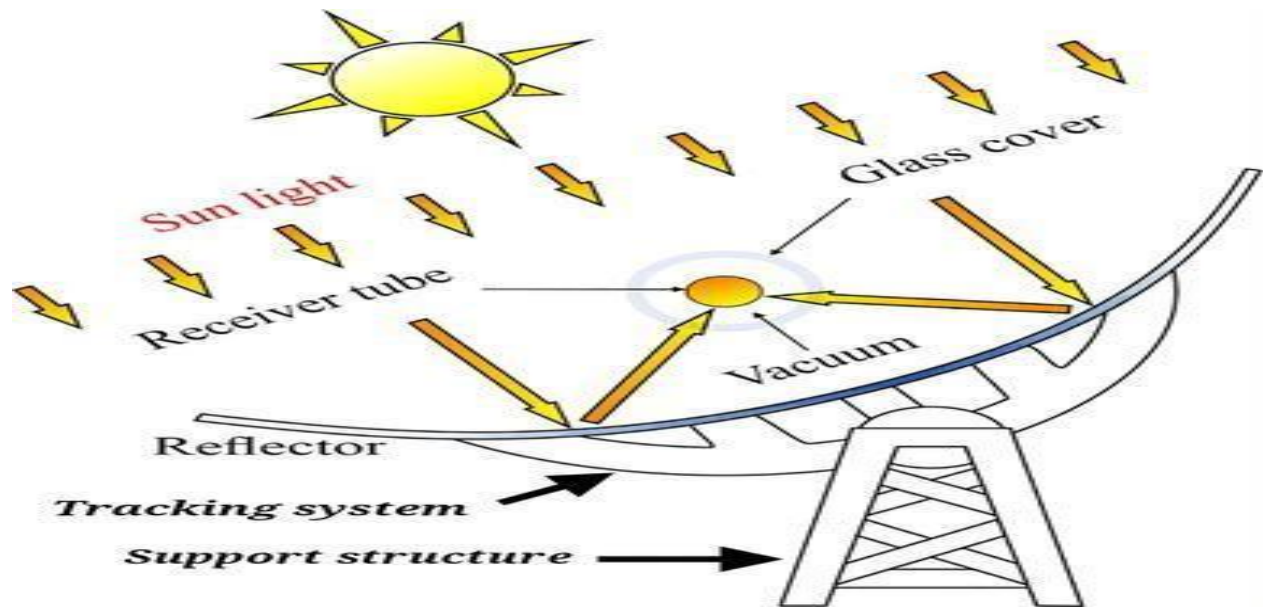


Figure 2.10 Schematic of parabolic trough collector [45]

II.4.1.1 Parabolic Reflector

The parabolic reflector is a curved mirror that focuses sunlight onto the receiver tube positioned at its focal line. It is typically made of glass mirrors coated with silver or aluminum, which offer high reflectivity (90-95%) and durability [29]. Some systems use polished aluminum sheets as a lightweight and cost-effective alternative [28]. The reflector's design is optimized to maximize solar concentration and improve thermal efficiency by directing the maximum amount of sunlight to the receiver [30].

II.4.1.2 Receiver Tube (Absorber)

The receiver tube shown in figure 2.11, also called the absorber tube, is placed at the focal point of the parabolic reflector. It absorbs concentrated solar radiation and transfers the heat to the heat transfer fluid (HTF) flowing inside. The receiver tube is typically made of stainless steel or copper, chosen for their high thermal conductivity and corrosion resistance [16].

To minimize heat loss, the tube is coated with a selective coating such as black chrome or cermet, which enhances solar absorption while reducing infrared radiation losses. These coatings significantly improve efficiency by absorbing more than 90% of incident solar radiation while emitting less than 10% as thermal radiation [31].

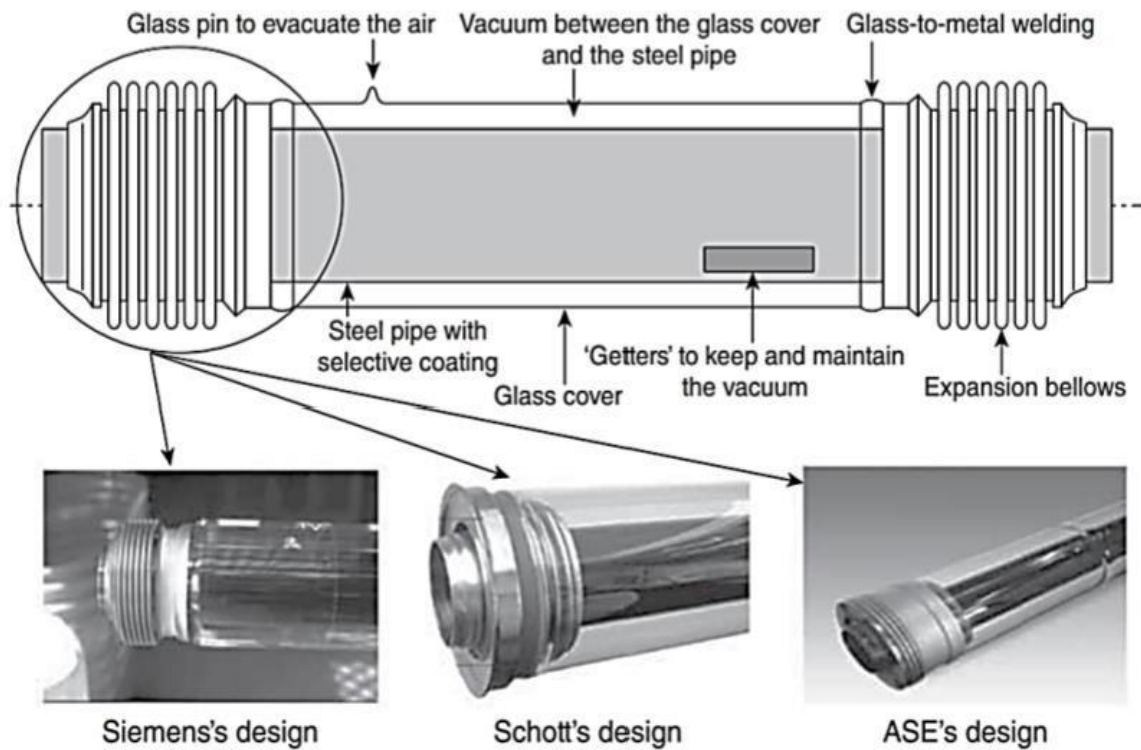


Figure 2.11 Absorber tube receiver [46]

II.4.1.3 Glass Envelope

To further reduce heat loss, the receiver tube is enclosed in a borosilicate glass envelope, creating a vacuum insulation layer around it [11]. This vacuum insulation minimizes convective heat losses and protects the receiver from external environmental factors such as wind and dust [32]. Additionally, the glass must be highly transparent to allow maximum solar radiation to reach the absorber while maintaining mechanical strength and resistance to thermal stress [33].

II.4.1.4 Heat Transfer Fluid (HTF)

The heat transfer fluid (HTF) circulates through the receiver tube, absorbing and transporting the collected heat to the thermal storage or power generation system. Common HTFs include:

- **Synthetic Oils** (e.g., Therminol VP-1): Efficient for high temperatures but degrade over time, requiring regular replacement [12].
- **Molten Salts** (e.g., Sodium-Potassium Nitrate mixtures): Offer excellent thermal stability and heat retention, making them ideal for thermal storage, but they require heating to prevent freezing at low temperatures [34].
- **Pressurized Water/Steam**: Used in direct steam generation (DSG) systems, eliminating the need for intermediate HTFs but requiring higher-pressure components [35].

II.4.1.5 Insulation System

To reduce heat losses, insulation is applied around the receiver tube. Common insulation materials include mineral wool, ceramic fibers, and aerogels, which provide high thermal resistance and durability [38]. These materials are chosen for their low thermal conductivity and high-temperature resistance. Protective cladding made of aluminum or stainless steel is often used to shield the insulation from environmental exposure and mechanical damage [36].

II.4.1.6 Control System

The control system regulates the operation of the PTC, ensuring optimal performance. It includes temperature sensors, flow control valves, and monitoring devices that adjust the heat transfer process, preventing overheating and improving system stability [16].

II.4.1.7 Tracking System

The tracking system is responsible for adjusting the position of the collector to follow the sun's movement throughout the day. This ensures that incoming solar radiation is continuously focused onto the receiver tube, maximizing heat absorption and reducing energy losses.

- **Single-Axis vs. Dual-Axis Tracking:** Most PTC systems use single-axis tracking, which follows the sun's east-west movement, offering a good balance between efficiency and cost. Dual-axis tracking, though more complex, can further enhance energy capture but is rarely used due to increased mechanical and control requirements [31].
- **Impact on Efficiency:** Proper tracking reduces cosine losses, ensuring that the maximum amount of solar radiation is concentrated on the receiver. Studies show that effective tracking can increase energy output by up to 30% compared to fixed systems [16].

To further enhance tracking accuracy, a control system is integrated. This system uses sensors and actuators to dynamically adjust the collector's position, optimizing performance and minimizing tracking errors [38].

II.4.1.8 Support Structure

The support structure provides stability and alignment for the reflector and receiver assembly. While not a direct factor in efficiency, a well-designed support structure prevents misalignment and withstands external forces such as wind and mechanical loads.

- **Material Selection:** The structure is typically made of galvanized steel or aluminum. Steel offers high strength and durability, making it ideal for large installations, while aluminum is used where weight reduction is a priority [28].
- **Structural Stability:** A strong support system ensures that the reflector remains properly aligned, preventing performance degradation due to mechanical shifts [39].

II.4.2 Operating Principle

A Parabolic Trough Collector (PTC) operates by concentrating solar radiation onto a linear receiver, converting sunlight into thermal energy. The parabolic mirrors focus incoming solar radiation onto a single focal line, where the receiver tube is positioned. As the sun moves, the collector's tracking system ensures continuous alignment, maximizing energy capture. The reflective surface, made of silvered glass mirrors or polished aluminum, redirects and amplifies sunlight, increasing the solar flux density at the receiver [29]. The receiver tube, containing a heat transfer fluid (HTF) such as synthetic oil, molten salt, or pressurized water, absorbs this concentrated heat. To enhance efficiency, the receiver is coated with a selective material, which increases solar absorption while minimizing thermal re-radiation losses. Additionally, a vacuum-sealed glass envelope surrounds the absorber tube, providing insulation that significantly reduces convective heat loss [31].

As the HTF (Heat Transfer Fluid) circulates through the receiver tube, it absorbs the concentrated heat, reaching temperatures between 300°C and 400°C. However, thermal losses occur through conduction, convection, and radiation. Conduction losses arise when heat transfers from the absorber tube to the support structure, while convective losses are minimized by the vacuum insulation [11]. Radiative losses, where the receiver emits heat as infrared radiation, are reduced using low-emissivity coatings. Despite these losses, the HTF retains significant thermal energy, which is then transported to a heat exchanger. Here, it produces steam for power generation in a Rankine cycle turbine or supplies high-temperature heat for industrial applications [40]. The efficiency of a PTC depends on optical precision, material properties, and operational conditions, with proper tracking and advanced coatings playing a crucial role in minimizing losses and maximizing energy output.

II.4.3 Applications of Parabolic Trough Collector

Parabolic Trough Collectors (PTCs) are widely used in various sectors due to their ability to efficiently convert solar energy into thermal energy. Their applications span from electricity generation to industrial heating, desalination, and district heating and cooling, making them a key technology in the transition to sustainable energy solutions.

II.4.3.1 Electricity Generation Using Rankine Cycle

PTCs are a key technology CSP plants, where they generate high-temperature steam to drive turbines for electricity production. These systems typically operate at 350°C to 400°C, but advanced designs using molten salts can reach up to 565°C, significantly enhancing efficiency. Using Rankin cycle the generated steam expands through a high-pressure (HP) and low-pressure (LP) turbine, converting thermal energy into mechanical work, which is then transformed into electricity by a generator. A condenser and cooling tower recycle the steam back into liquid form for continuous operation, with some systems incorporating a boiler as a backup heat source, as shown in figure 2.12. Many CSP plants integrate thermal energy storage (TES), enabling electricity generation even after sunset and improving grid reliability. A prime example is the Noor Ouarzazate Solar Complex in Morocco, one of the world's largest CSP plants. The Noor I plant (160 MW) utilizes PTCs with a molten salt storage system, allowing it to generate electricity for 3 hours after sunset. Similar projects include the Andasol CSP plants in Spain, which also use molten salt storage to extend energy generation beyond daylight hours [28] [16].

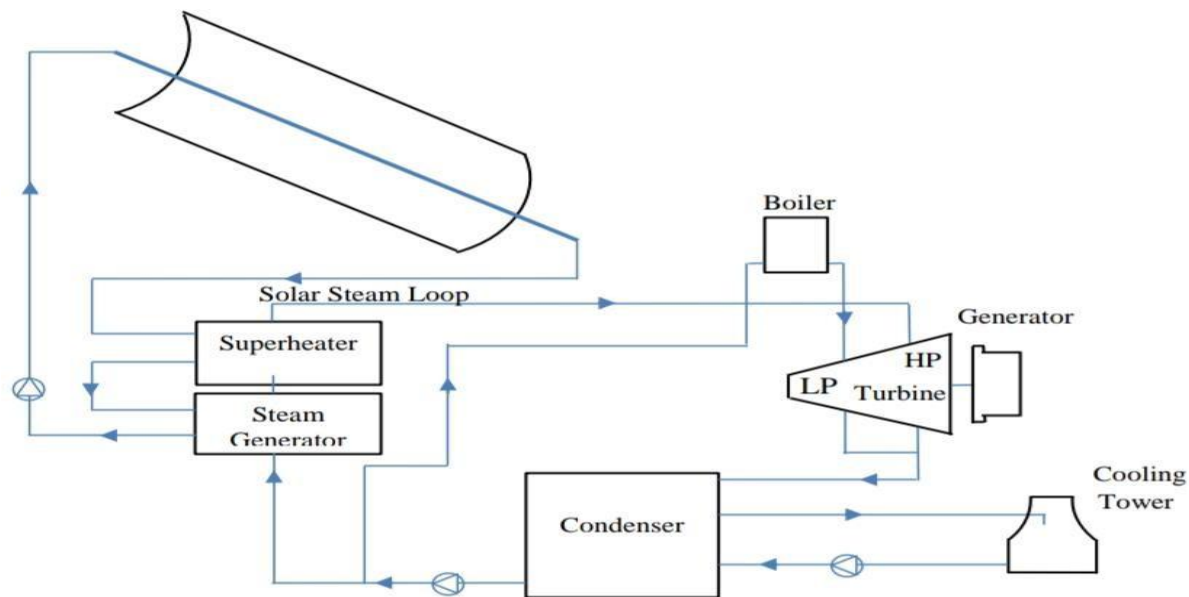


Figure 2.12 Typical schematic of SEGs plants [16]

II.4.3.2 Industrial Process Heat

PTCs are widely used in industrial applications to generate process heat, particularly in sectors requiring low to medium-temperature steam. A study conducted in Southern Spain examined the integration of a parabolic trough solar plant to supply saturated steam at 7 bar for a food processing facility specializing in vegetable preservation. The system consisted of a parabolic trough solar field, thermally stratified energy storage, and a steam generator (unfired boiler), with a pre-heating heat exchanger included to improve efficiency. The solar plant was designed to integrate with the existing steam distribution network, simplifying the incorporation of solar energy into industrial processes. Located in Almería, Spain, the facility processes tomatoes and other vegetables, producing products such as tomato sauces, jams, and roasted peppers. The annual heat demand was 148 MWh, with 64% utilized for processing and 36% accounting for thermal losses.

The study assessed key operational parameters, such as the solar field outlet temperature and steam generator return temperature, demonstrating that PTCs can enhance energy efficiency and reduce reliance on fossil fuels in food processing industries [41]. Figure 2.13 illustrates a typical PTC integrated in an industrial process.

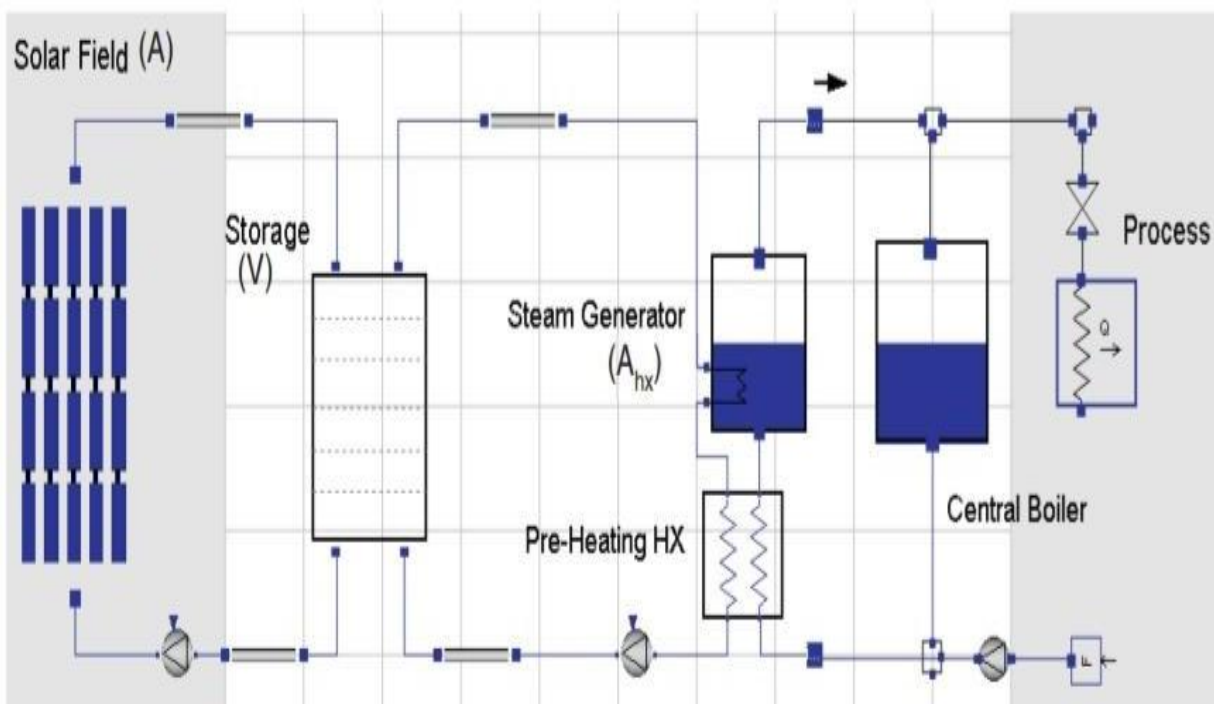


Figure 2.13 Diagram of the parabolic trough solar plant and industrial process [41]

II.4.3.3 Desalination

PTCs play a crucial role in solar-powered desalination, particularly in arid and coastal regions where freshwater is scarce. They provide the necessary thermal energy for Multi-Effect Distillation (MED) and Reverse Osmosis (RO) processes, improving efficiency and reducing energy consumption. In MED, solar heat is used to evaporate seawater in multiple stages, while in RO, PTCs preheat the feedwater, reducing the electricity required for membrane filtration.

As shown in figure 2.14, a PTC field supplies thermal energy to a Multiple-Effect Boiling evaporator, which operates at 70–100°C. The system includes a flash vessel and a backup boiler to ensure continuous operation. Seawater is first filtered before entering the system, where it undergoes evaporation and condensation in multiple stages, producing fresh water while discharging concentrated brine back to the sea. [42] Kalogirou (1998) highlights that solar-assisted Multiple-Effect Boiling is one of the most efficient desalination methods due to its low energy consumption and simple seawater treatment requirements [28].

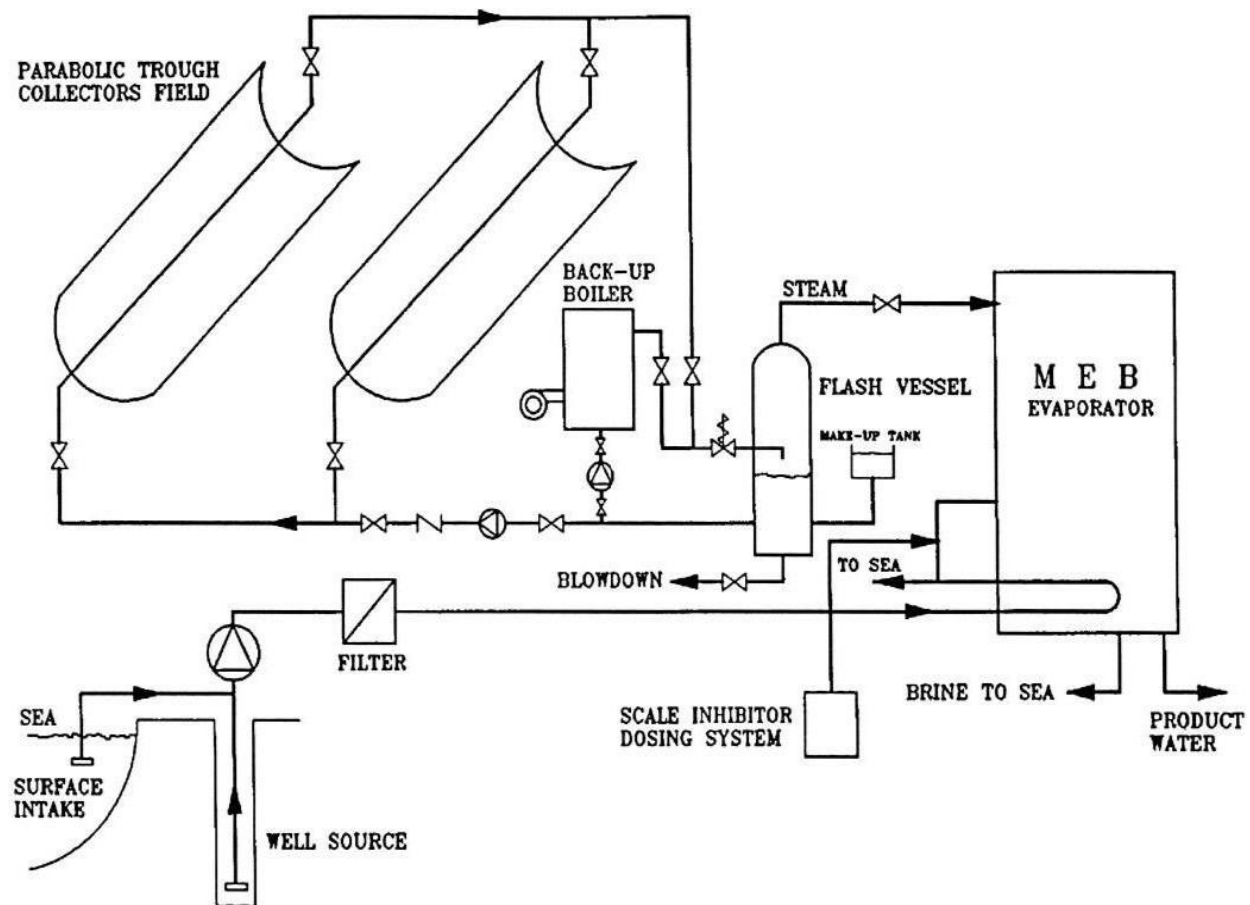


Figure 2.14 Desalination system arrangement [42]

II.4.3.4 District Heating and Cooling

Parabolic Trough Collectors (PTCs) are integral components of district heating and cooling (DHC) networks, providing centralized thermal energy for applications such as space heating, domestic hot water, and industrial processes. The solar heat collected by PTCs is typically stored in Thermal Energy Storage (TES) systems, like molten salts or pressurized water, ensuring a continuous heat supply even during periods without sunlight. In district cooling, PTCs can power absorption chillers to produce cooling energy for air conditioning and refrigeration.

A notable example is the Zhabuye 40 MW Parabolic Trough CSP Project in Tibet, China, which utilizes a parabolic trough collector field covering 621,000 m². The system includes thermal storage capable of providing 16 hours of nominal output after sunset, ensuring a stable heat supply in extreme winter conditions. The project shown in figure 2.15 demonstrates the effectiveness of solar thermal technology in reducing reliance on conventional heating fuels while ensuring a sustainable and continuous energy supply [43].



Figure 2.15 Euro Trough for CSP in Tibet [47]

II.5 Conclusion

In summary, this chapter has provided a comprehensive theoretical foundation for understanding solar collectors, with particular emphasis on Parabolic Trough Collectors (PTCs). We began with the classification and operational principles of various solar collector technologies, including flat plate, unglazed, and evacuated tube collectors, categorized according to their temperature operating ranges and working fluids. Among these, PTCs stand out for their ability to achieve high thermal efficiency and elevated working temperatures, making them ideal for industrial process heat and power generation. The design and operation of PTCs were explored in detail, covering essential components. Furthermore, we discussed the optical and thermal principles governing their performance and examined real-world applications across various sectors. This foundation sets the stage for the next chapter, which will focus on the mathematical modeling and analytical resolution of the PTC's thermal behavior, aiming to evaluate and optimize its performance based on fundamental energy balance equations.

CHAPTER THREE

**MATHEMATICAL AND THERMAL
MODELING OF PARABOLIC TROUGH
COLLECTORS**

III.1 Introduction

Theoretical modeling is a fundamental approach for analyzing the performance of parabolic trough collector (PTC) systems. Before implementation or simulation, it is essential to establish a solid mathematical framework that captures the key thermal and optical behaviors of the system. This chapter focuses on developing such a model by formulating equations that describe heat losses, optical efficiency, and the overall energy balance of the PTC. The model is based on a set of simplifications and assumptions that make the analysis both feasible and relevant, providing a foundation for predicting system performance under different operating conditions.

III.2 Heat Transfer Mechanism

Heat is the energy form which can be transfer between two systems as a result of different temperature. Generally, there are three mechanisms of heat transfer, radiation, conduction, and convection. Actually, the heat energy transfer from higher temperature surface to lower temperature one and the process is stopped when the two surface reach to the same temperature (equilibrium).

III.2.1 Conduction Heat Transfer

Heat transfer by conduction is the transfer of thermal energy through a material without large-scale movement, driven by a temperature gradient and governed by Fourier's law, in the one-dimensional steady-state case [49]

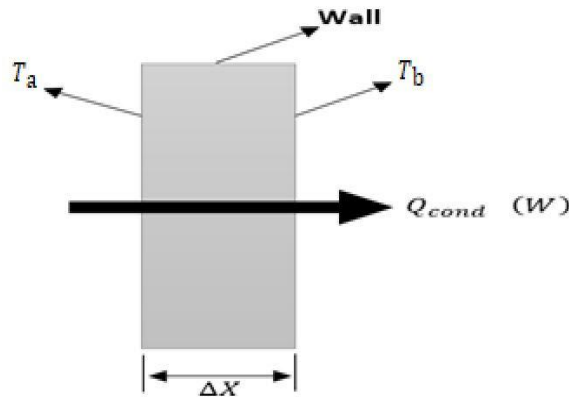


Figure 3.1 Conduction heat transfer across wall

$$\dot{q}_{cond} = -\frac{kA}{\Delta x} (T_b - T_a) \quad (III.1)$$

III.2.2 Convection Heat Transfer

Convection is the heat transfer between the body surface and adjacent motion gas or liquid. When the fluid (liquid or gas) motion is natural, the convection is called free (natural) convection. On the other hand, forced convection means that the fluid is forced to flow by external forced like wind, pump, or fan [49].

$$\dot{q}_{conv} = h(T_w - T_{\infty}) \quad (III.2)$$

III.2.3 Radiation Heat Transfer

Radiation heat transfer (thermal radiation) is the energy emitted from the different body and it is very fast equal the speed of light, each body emit thermal radiation when it has temperature above absolute zero. The consideration of thermodynamics show that the maximum radiation rate that can be emit from body surface is expressed as Stefan-Boltzmann law [49]

$$\dot{q}_{rad} = \sigma AT^4 \quad (III.3)$$

III.3 Modeling the Parabolic Trough Collector

The modeling of a Parabolic Trough Collector aims to analyze and predict its performance by focusing on two main aspects: the optical behavior of the concentrator and the thermal performance of the receiver tube. The optical model addresses how solar radiation is reflected and concentrated onto the absorber tube, while the thermal model focuses on the heat transfer processes and temperature distribution within the working fluid.

To make the modeling process manageable and computationally efficient, several simplifications and assumptions are adopted:

- The PTC is assumed to operate under steady-state conditions.
- There is uniform heat flux over the surface of the absorber tube.
- No contact thermal losses occur between the absorber and its surrounding components.
- The solar radiation is considered uniform and perpendicular to the collector aperture.
- Optical errors such as tracking deviations and mirror imperfections are either neglected or included in an average optical efficiency factor.
- The heat transfer is modeled as one-dimensional along the axis of the absorber tube; radial and circumferential effects are neglected.

- Thermophysical and optical properties of materials are assumed constant with respect to temperature.
- Heat losses are modeled using an overall heat loss coefficient, which combines convection and radiation effects.
- Ambient conditions (e.g., temperature, wind) are represented using average values rather than time-dependent profiles.

III.3.1 Mathematical Model

III.3.1.1 Optical Study

The optical performance of a PTC is strongly influenced by its geometric design. A cross-sectional view, as illustrated in figure 3.2, highlights key geometric parameters such as the rim angle θ_r , focal length f , mirror radius r , and receiver diameter D .

When solar radiation strikes the reflector at the rim, it forms a rim angle θ_r . For optimal alignment, the receiver must intercept the entire solar image. The required receiver diameter is given by [50]:

$$D = 2r_r \sin(\theta_m) \quad (III.4)$$

The mirror radius r can be calculated using the focal length f and the angle θ , which is the angle between the reflected beam and the collector axis [50]

$$r = \frac{2f}{1 + \cos(\theta)} \quad (III.5)$$

At the rim angle θ_r , the corresponding radius becomes:

$$r_r = \frac{2f}{1 + \cos(\theta_r)} \quad (III.6)$$

The radiation beam of figure 3.2 is incident at the rim of the concentrator. The angle θ_r , made by the reflected beam radiation with the center line, is called rim angle [51]:

$$\theta_r = \tan^{-1} \left[\frac{8 \left(\frac{f}{W_a} \right)}{16 \left(\frac{f}{W_a} \right)^2 - 1} \right] = \sin^{-1} \left(\frac{f}{W_a} \right) \quad (III.7)$$

The aperture width W_a , which is the opening width of the parabola, is also related to the focal length and rim angle [52]:

$$W_a = 4f \tan\left(\frac{\theta_r}{2}\right) \quad (III.8)$$

The concentration ratio is a critical parameter that defines the capability of the system to focus solar radiation onto the receiver. It directly impacts the system's temperature and thermal performance. There are two main types [53]:

- **Radiation-Flux Concentration Ratio C_{flux}**

$$C_{flux} = \frac{I_{abs}}{I_{ap}} \quad (III.9)$$

$$I_{ap} = I_b K(\theta_i) \quad (III.10)$$

This ratio represents the energy flux density and indicates the magnification of solar radiation.

- **Geometrical Concentration Ratio C_{geom}**

$$C_{geom} = \frac{A_{ap}}{A_{abs}} \quad (III.11)$$

This ratio is a purely geometrical characteristic of the system and does not vary with solar conditions or time of day. It is widely used in practical engineering applications due to its simplicity and stability.

An additional approximation for the effective concentration ratio in parabolic trough systems is given by [52]:

$$C = \frac{W_a L}{\pi D L} = \frac{W_a}{\pi D} \quad (III.12)$$

The concentration ratio C is related to θ_r can also be defined as [52]:

$$C = \frac{\sin \theta_r}{\pi \sin \theta_m} \quad (III.13)$$

In general, the higher the concentration ratio—whether radiation-flux or geometrical—the more capable the system is at focusing sunlight onto the absorber. This leads to increased thermal energy collected and higher operating temperatures

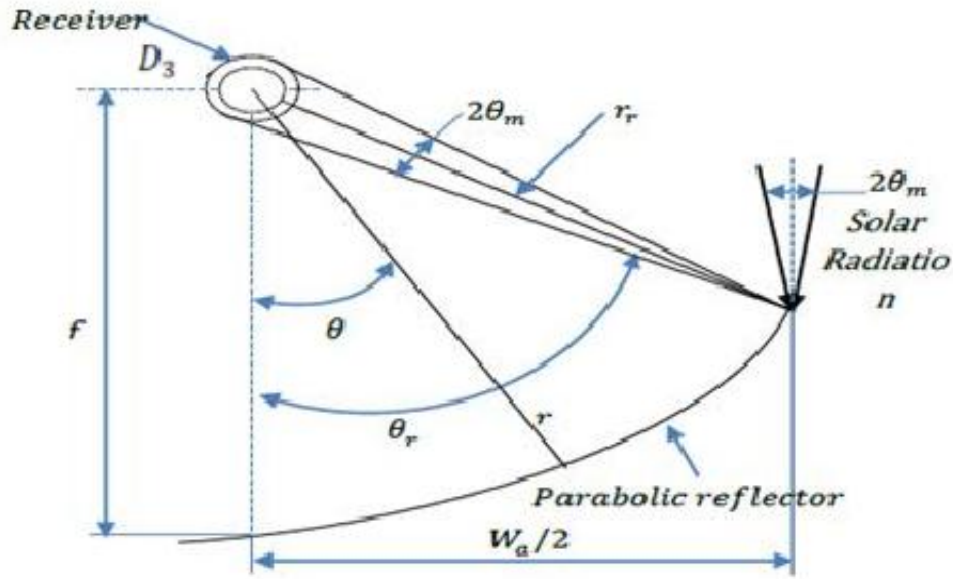


Figure 3.2 A cross section of the receiver and the reflector showing different angles [28]

III.3.1.2 Optical Efficiency

The optical efficiency of a parabolic trough solar collector is defined as the ratio of the solar energy absorbed by the receiver tube to the total solar energy incident on the collector aperture. It encompasses all optical losses, including mirror reflectivity losses, tracking errors, and absorption losses within the collector system. Optical efficiency is a key performance parameter, as it determines the effective solar input available for thermal conversion.

$$\eta_0 = \frac{\dot{q}''}{I_{ap}} \quad (III.14)$$

This efficiency is influenced by several optical parameters:

- **Mirror Reflectance ρ_a** : the fraction of solar radiation reflected by the mirror surface.
- **Glass Envelope Transmittance τ_g** : the portion of radiation transmitted through the glass envelope surrounding the absorber.
- **Receiver Absorptance α_{abs}** : the portion of transmitted radiation absorbed by the receiver surface.
- **Intercept Factor γ** : the fraction of reflected rays that successfully reach the receiver.
- **Incidence Angle Modifier (θ_i)**: compensates for the variation in performance caused by changes in the angle of solar incidence.
- **End Loss Factor X_{end}** : accounts for radiation energy lost at the ends of the receiver tube.

III.3.1.3 Incidence Angle Modifier

The incidence angle modifier $K(\theta_i)$ compensates for the reduction in collected energy as the solar incidence angle θ_i deviates from normal. As the angle increases, the alignment of reflected rays with the receiver becomes less accurate, resulting in optical losses. The modifier is expressed as [54]:

$$K(\theta_i) = \cos(\theta_i) - 0.00384(\theta_i) - 0.000143(\theta_i)^2 \quad (III. 15)$$

III.3.1.4 End Losses

End losses occur when part of the reflected radiation misses the ends of the receiver tube, especially at higher incidence angles. The fraction of radiation not lost at the ends is given by [55]:

$$X_{end} = 1 - \left(\frac{f}{L}\right) \tan(\theta_i) \quad (III. 16)$$

A visual representation of end losses, illustrating the geometry and energy loss, is provided in figure 3.3.

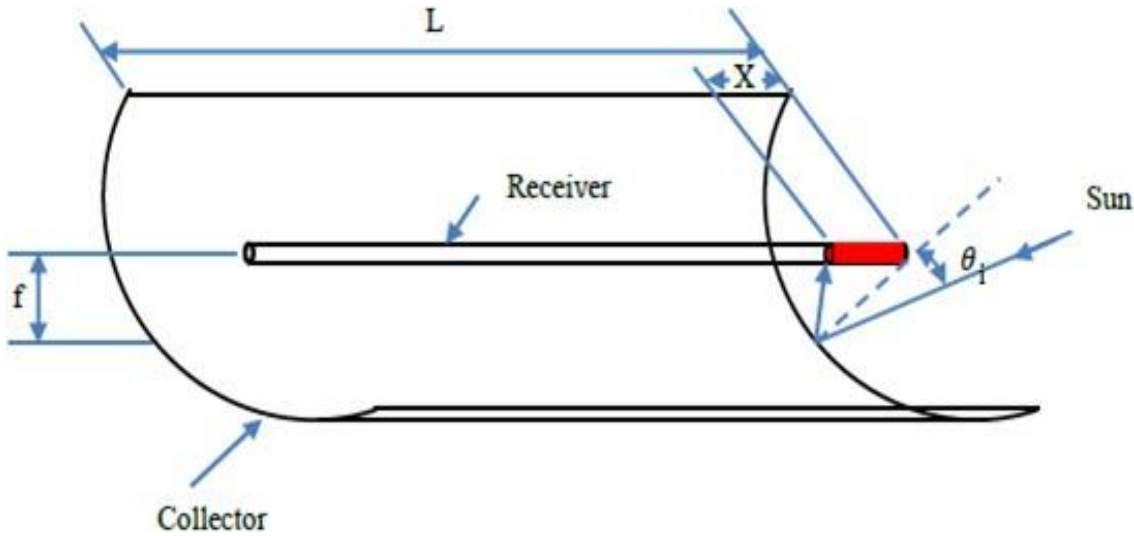


Figure 3.3 End Losses [28]

III.3.1.5 Intercept Factor

The intercept factor γ represents the fraction of reflected rays that successfully reach the receiver. It accounts for real-world imperfections such as:

- Tracking and alignment errors
- Surface inaccuracies of the mirror
- Dust and dirt accumulation
- Structural deformations

With all of the modifiers taken into account, the absorbed radiation, $\dot{q}''$ in Eq (III. 14) is the actual amount of radiation on the receiver is calculated by:

$$\dot{q}'' = I_b(\rho_a \tau_g \alpha_{abs} \gamma)K(\theta_i)X_{end} \quad (III. 17)$$

Thus, the optical efficiency η_0 becomes:

$$\eta_0 = (\rho_a \tau_g \alpha_{abs} \gamma)K(\theta_i)X_{end} \quad (III. 18)$$

This formulation now incorporates both the intensity of concentrated solar radiation and the concentration ratio, allowing for a precise evaluation of the collector's optical performance under various operating conditions [58].

III.3.2 Thermal Analysis

The HCE performance model is based on an energy balance between the heat transfer fluid (HTF) and the surrounding environment, incorporating all necessary equations and correlations to predict the various terms in the balance. These terms depend on factors such as the collector design, HCE condition, optical characteristics, and ambient conditions. Figure 3.4.a depicts the one-dimensional steady-state energy balance across an HCE cross-section and figure 3.4.b presents the associated thermal resistance. For simplicity, the incoming solar energy and optical losses are excluded from the resistance model. Optical losses result from imperfections in the mirrors, tracking inaccuracies, shading, and the cleanliness of both the mirrors and the HCE.

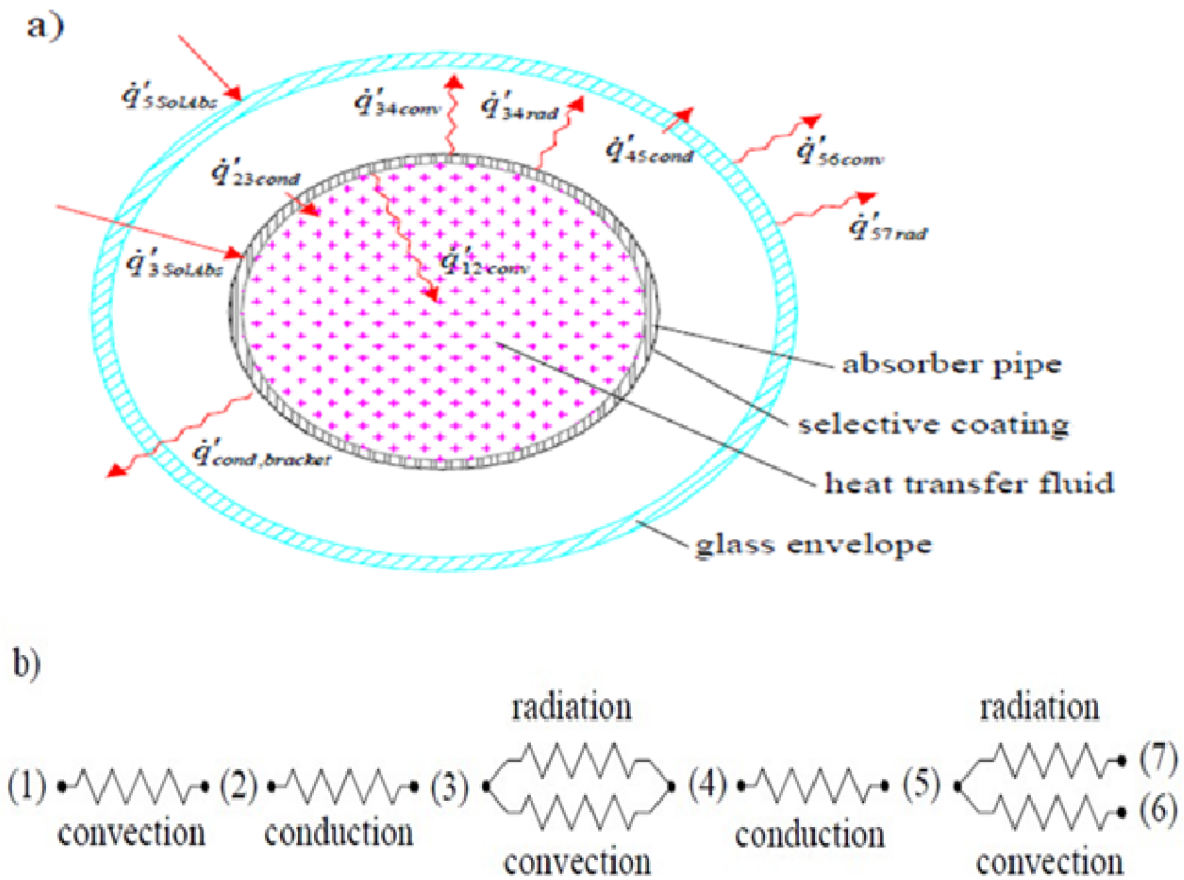


Figure 3.4 a) One-dimensional steady-state energy balance and **b)** Thermal resistance model for a cross-section of an HCE

With

- (1) Heat transfer fluid
- (2) Absorber inner surface
- (3) Absorber outer surface
- (4) Glass envelope inner surface
- (5) Glass envelope outer surface
- (6) Surrounding air
- (7) Sky

The effective incoming solar energy (solar energy minus optical losses) is absorbed by the glass envelope ($\dot{q}'_{5SolAbs}$) and absorber selective coating ($\dot{q}'_{3SolAbs}$). Some energy that is absorbed into the selective coating is conducted through the absorber ($\dot{q}'_{12conv}$) and transferred to the HTF by convection ($\dot{q}'_{23cond}$); remaining energy is transmitted back to the glass envelope by convection ($\dot{q}'_{34conv}$) and radiation ($\dot{q}'_{34rad}$). The energy from the radiation and convection then passes through the glass envelope by conduction ($\dot{q}'_{45cond}$) and is lost to the environment by convection ($\dot{q}'_{56conv}$) and radiation ($\dot{q}'_{57rad}$).

The model assumes all temperatures, heat fluxes, and thermodynamic properties are uniform around the circumference of the HCE. Also, all flux directions shown in figure 3.4.a are positive. With the help of figure 3.4.b, the energy balance equations are determined by conserving energy at each surface of the HCE cross-section.

$$\dot{q}'_{12conv} = \dot{q}'_{23cond} \quad (III.19.a)$$

$$\dot{q}'_{3SolAbs} = \dot{q}'_{23cond} + \dot{q}'_{34conv} + \dot{q}'_{34rad} \quad (III.19.b)$$

$$\dot{q}'_{34conv} + \dot{q}'_{34rad} = \dot{q}'_{45cond} \quad (III.19.c)$$

$$\dot{q}'_{45cond} + \dot{q}'_{5SolAbs} = \dot{q}'_{56conv} + \dot{q}'_{57rad} \quad (III.19.d)$$

$$\dot{q}'_{HeatLoss} = \dot{q}'_{56conv} + \dot{q}'_{57rad} \quad (III.19.e)$$

Note that the solar absorptance ($\dot{q}'_{3SolAbs}$) and ($\dot{q}'_{5SolAbs}$) are treated as heat flux terms, this simplifies the solar absorption terms and makes the heat conduction through the absorber pipe and glass envelope linear.

All the terms in Equations (III.19) are defined in Table 3.1 Dotted variables indicate rates and the prime indicates per unit length of receiver.

Table 3.1 Heat flux definitions

Heat Flux (W/m)	Heat transfer mode	Heat transfer path	
		From	To
$\dot{q}'_{12conv}$	Convection	Inner absorber pipe surface	Heat transfer fluid
$\dot{q}'_{23cond}$	Conduction	Outer absorber pipe surface	Inner absorber pipe surface
$\dot{q}'_{3SolAbs}$	Solar irradiation absorption	Incident solar irradiation	Outer absorber pipe surface
$\dot{q}'_{34conv}$	Convection	Outer absorber pipe surface	Inner glass envelope surface
$\dot{q}'_{34rad}$	Radiation	Outer absorber pipe surface	Inner glass envelope surface
$\dot{q}'_{45cond}$	Conduction	Inner glass envelope surface	Outer glass envelope surface
$\dot{q}'_{5SolAbs}$	Solar irradiation absorption	Incident solar irradiation	Outer glass envelope surface
$\dot{q}'_{56conv}$	Convection	Outer glass envelope surface	Ambient
$\dot{q}'_{57rad}$	Radiation	Outer glass envelope surface	Sky
$\dot{q}'_{HeatLoss}$	Convection and Radiation	Heat collecting element	Ambient and Sky
*Per unit aperture length			

III.3.2.1 Convection Heat Transfer Between The HTF And The Absorber

The heat transfer from the absorber's inner surface to the heat transfer fluid (HTF) follows Newton's law of cooling, expressed as:

$$\dot{q}'_{12conv} = h_1 D_2 \pi (T_2 - T_1) \quad (III.20)$$

$$h_1 = Nu_{D2} \left(\frac{k_f}{D_2} \right) \quad (III.21)$$

III.3.2.2 Conduction Heat Transfer Through The Absorber Wall

The conduction of heat through the absorber wall is modeled by Fourier's law for a hollow cylinder:

$$\dot{q}'_{23cond} = \frac{2\pi k_{23}(T_2 - T_3)}{\ln\left(\frac{D_3}{D_2}\right)} \quad (III.22)$$

III.3.2.3 Heat Transfer in The Annulus

Both convection and radiation heat transfer take place between the absorber and the glass envelope. The nature of convection depends on the pressure within the at low pressures ($< \sim 0.0013$ bar), heat transfer is dominated by molecular conduction, whereas at higher pressures ($> \sim 0.0013$ bar), free convection becomes the primary mechanism. Radiation heat transfer results from the temperature difference between the outer surface of the absorber and the inner surface of the glass envelope.

III.3.2.3.1 Vacuum in Annulus

When the HCE annulus is under vacuum conditions (pressure less than approximately 1 to 0,0013 bar), heat transfer between the absorber and the glass envelope takes place through free- molecular convection.

With

$$\dot{q}'_{34conv} = \pi D_3 h_{34}(T_3 - T_4) \quad (III.23)$$

$$h_{34} = \frac{k_{std}}{\left(\frac{D_3}{2\ln\left(\frac{D_4}{D_3}\right)}\right) + b\lambda\left(\frac{D_3}{D_4} + 1\right)} \quad (III.24)$$

$$b = \frac{(2 - a)(9\gamma_g - 5)}{2a(\gamma_g + 1)} \quad (III.25)$$

$$\lambda = \frac{2,331E(-20)(T_{34} + 273,15)}{(P_a\delta^2)} \quad (III.26)$$

III.3.2.3.2 Radiation Heat Transfer Across the Annulus

Radiative exchange between the absorber and the inner surface of the glass envelope is determined by:

$$\dot{q}'_{34rad} = \frac{\sigma \pi D_3 (T_3^4 - T_4^4)}{\left(\frac{1}{\varepsilon_3} + \frac{(1 - \varepsilon_4) D_3}{(\varepsilon_4 D_4)} \right)} \quad (III. 27)$$

III.3.2.4 Conduction Heat Transfer Through The Glass Envelope

The conduction through the glass envelope is modeled using the same equation as for the absorber wall (Section III.3.2.2). The anti-reflective coating is assumed to add no thermal resistance or affect the glass emissivity, as it is formed by chemical etching without adding material.

$$\dot{q}'_{45cond} = \frac{2\pi k_{45} (T_4 - T_5)}{\ln \left(\frac{D_5}{D_4} \right)} \quad (III. 28)$$

III.3.2.5 Heat Transfer from The Glass Envelope to The Environment

Heat is transferred from the glass envelope to the surrounding atmosphere through both convection and radiation. Convection can be either natural or forced, depending on the presence of wind. Radiation losses occur as a result of the temperature difference between the glass envelope and the sky.

III.3.2.5.1 Convection to The Ambient

Convection to the Ambient Heat convection from the outer glass surface to the surrounding air is a significant loss, especially under windy conditions. It is described by :

$$\dot{q}'_{56com} = h_{56} \pi D_5 (T_5 - T_6) \quad (III. 29)$$

$$h_{56} = Nu_{D5} \left(\frac{k_{56}}{D_5} \right) \quad (III. 30)$$

The Nusselt number depends also on whether the convection heat transfer is natural (no wind) or forced (with wind).

III.3.2.5.1.1 No Wind Case

Since there is no wind in our model, the heat transfer from the glass envelope to the surrounding environment occurs by natural convection. In this case, the Nusselt number, which characterizes the convective heat transfer, is determined using the correlation developed by Churchill and Chu [59].

$$Nu_{D5} = \left\{ 0,6 + \frac{0,387 Ra_{D5}^{\frac{1}{6}}}{\left[1 + \left(\frac{0,559}{Pr_{56}} \right)^{\frac{9}{16}} \right]^{\frac{9}{27}}} \right\}^2 \quad (III.31)$$

$$Ra_{D5} = \frac{g\beta(T_5 - T_6)D_5^3}{(\alpha_{56}v_{56})} \quad (III.32)$$

$$\beta = \frac{1}{T_{56}} \quad (III.33)$$

$$Pr_{56} = \frac{v_{56}}{\alpha_{56}} \quad (III.34)$$

III.3.2.6 Radiation to The Sky

The glass envelope also radiates heat toward the sky, following:

$$\dot{q}'_{57rad} = \sigma D_5 \pi \varepsilon_5 (T_5^4 - T_7^4) \quad (III.35)$$

III.3.2.7 Solar Irradiation Absorption in The Glass Envelope

To simplify the modeling of solar absorption, it is treated as a heat flux at both the glass envelope and the absorber. In reality, solar absorption is a volumetric heat generation process that depends on the glass envelope's thickness. However, because the glass envelope is relatively thin and its solar absorption coefficient is low, this simplification introduces only a minor error. The correlation used to represent solar irradiation within the glass envelope is given by:

$$\dot{q}'_{5SolAbs} = I_b A_{ap} \rho_a \gamma K(\theta_i) X_{end} \quad (III.36)$$

III.3.2.8 Solar Irradiation Absorption in The Absorber

Since the absorber captures solar energy primarily near its surface, it is considered as a heat flux. Therefore, the solar absorption within the absorber is described by the following equation:

$$\dot{q}'_{3Solabs} = \dot{q}'_{5Solabs} \tau_g \alpha_{abs} \quad (III. 37)$$

III.3.2.9 Thermal Efficiency

The thermal efficiency of a PTC is defined as the ratio of the useful heat gain of the HTF, $\dot{q}_u$, to the solar energy intercepted by the collector aperture area and is given by:

$$\eta = \frac{\dot{m} c_p (T_{fo} - T_{fi})}{I_{ap} A_{ap}} \quad (III. 38)$$

Another approach to formulating the energy balance of the heat collector element (HCE) is presented below. While the method shown previously in Equations (III. 19. a – III. 19. d) detailed each heat flux individually, the energy balance can be summarized into a more general and compact form, which simplifies the modeling while preserving the physical meaning.

The general energy balance of the HCE can be expressed as:

$$\dot{q}_{ab} = \dot{q}_u + \dot{q}_{loss} \quad (III. 39)$$

In addition, by substituting the individual heat transfer terms derived previously in Equations (III. 19. a – III. 19. d), a more detailed energy balance is obtained:

$$\dot{q}'_{3.SolAbs} + \dot{q}'_{5SolAbs} = \dot{q}'_{12conv} + \dot{q}'_{HeatLoss} \quad (III. 40)$$

This detailed form reflects the specific contributions of solar absorption at both the absorber and glass envelope surfaces, and the corresponding thermal losses through convection and radiation to the environment.

The $\dot{q}_{loss}$ can be rewritten in terms of an overall loss coefficient, U_L by considering the expression:

$$\dot{q}_{loss} = U_L A_r (T_r - T_{air}) \quad (III. 41)$$

Figure 3.5 shows the simplified thermal resistance network, representing the dominant heat transfer paths while excluding less significant conduction effects to reduce complexity and enhance clarity.

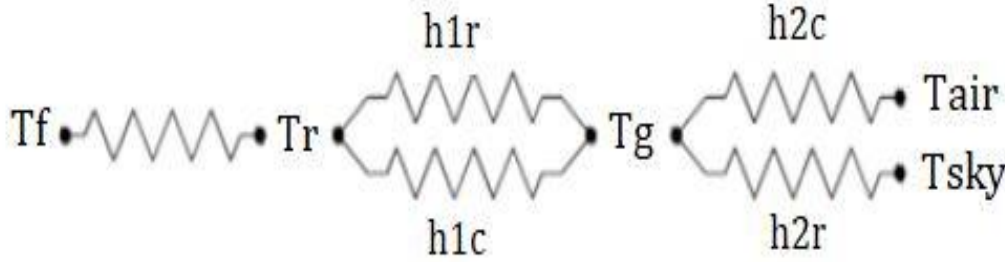


Figure 3.5 Simplified thermal resistance network

A representative expression for the total heat loss coefficient, combining all major convective and radiative losses in the system, is given by:

$$U_L = h_{1c} + h_{1r} + h_{2c} + h_{2r} \quad (III.41)$$

To determine the useful heat gain, the following equation is used:

$$\dot{q}_u = \dot{q}_{ab} - \dot{q}_{loss} \quad (III.42)$$

$$\dot{q}_u = F_R [I_b (\rho_a \tau_g \alpha_{abs} \gamma K(\theta_i) X_{end}) A_{ap} - U_L A_r (T_r - T_{air})] \quad (III.43)$$

F_R The heat removal factor represents the ratio of the actual useful energy gain that would result if the collector-absorbing surface had been at the local fluid temperature [13].

$$F_R = \frac{\dot{m} c_p}{A_r U_L} \left[1 - \exp \left(- \frac{A_r U_L F'}{\dot{m} c_p} \right) \right] \quad (III.44)$$

F' is the collector efficiency factor, which was presented by Kalogirou (2009) [50] as follows (Kalogirou 2012) [13] :

$$F' = \frac{\frac{1}{U_L}}{\frac{1}{U_L} + \frac{D_3}{h_f D_2} + \left(\frac{D_3}{2k_f} \ln \frac{D_3}{D_2} \right)} \quad (III.45)$$

The thermal efficiency of the collector can now be re-written from Eq. (III. 38) and Eq. (III. 42) as follow:

$$\eta = F_R \left[\eta_0 - \frac{U_L}{C} \left(\frac{T_{fi} - T_{air}}{I_b} \right) \right] \quad (III.46)$$

III.4 Computational Program Algorithm

This section describes the algorithmic implementation of the derived analytical PTC model, transforming the theoretical framework into an automated computational process for efficient performance evaluation and parametric analysis. The structured algorithm systematically processes input parameters including collector geometry, material properties, and operating conditions. It executes the analytical optical model to determine flux distribution and intercept factors, while the thermal module solves the energy balance equations for the absorber tube and heat transfer fluid. Developed in MATLAB, this computational approach enables rapid simulation across a range of operating scenarios while maintaining the mathematical rigor of the analytical solution.

The proposed methodology employs a steady-state approach, assuming constant and uniform conditions across all collector components while disregarding transient thermal effects. Although the theoretical model in Section III.3 includes all modes of heat transfer between system components, certain assumptions are applied in the computational implementation to simplify the analysis. Specifically, axial and radial heat conduction within both the absorber tube and the glass envelope are neglected. This simplification is justified by the relatively small conduction areas involved, which contribute negligibly to the overall heat transfer balance. Additionally, the model assumes average temperatures between the inner and outer surfaces of both the receiver and the glass cover, allowing the use of mean thermal properties and simplifying the solution of energy balance equations.

To initiate the thermal analysis, the algorithm assigns initial estimates for the absorber and glass cover temperatures. These are based on the inlet fluid temperature: the absorber is set $T_r = T_{fi,i=0} + 20^\circ\text{C}$, and the glass cover to $T_g = T_{fi,i=0} + 10^\circ\text{C}$. These assumed values enable the initial calculation of thermal parameters including the overall heat loss coefficient U_L , the heat removal factor F_R , and the useful heat gain $\dot{q}_u$.

To solve the thermal behavior of the collector system, the program follows a step-by-step numerical routine. After assigning initial temperature values to the absorber and glass cover based on the inlet fluid conditions, the model begins calculating the thermal parameters. At this stage, the simulation does not assume that the initial estimates are correct. Instead, it uses them as a starting point and refines them through successive recalculations. During each simulation cycle, the model updates the absorber temperature by solving the energy balance equations using the current thermal and environmental inputs.

After each update, it evaluates how much the temperature has changed compared to the previous step $|(T_r)_{i+1} - (T_r)_i| < 0.01^\circ\text{C}$. If the difference is larger than a predefined tolerance, the program repeats the calculations using the newly estimated temperature. This loop continues until the change becomes small enough to meet the convergence criteria, indicating that a stable solution has been found. Only then does the program proceed to analyze the next set of conditions or parameters.

The computational model primarily outputs the outlet temperature of the heat transfer fluid, thermal efficiency, and useful heat gain. In addition to these core metrics, the model also provides detailed information on the temperatures of the absorber and glass cover, as well as the total thermal losses. These results offer a comprehensive understanding of the collector's thermal performance. The program is designed to be adaptable for evaluating different collector types and can be extended to incorporate additional physical effects as needed.

III.4.1 Step-By-Step Numerical Methodology

The numerical solution procedure follows a systematic nine-step methodology:

1. Input Parameters Initialization

The algorithm begins by processing three categories of input parameters:

- Meteorological conditions (solar irradiance, ambient temperature)
- Material thermo-physical properties (fluid viscosity, conductivity, mass flow rate)
- Optical characteristics (transmittance-absorptance product, glass and absorber emissivities)
- Geometric specifications (aperture width, tube diameters, collector length)

2. Temperature Initialization

Initial temperature estimates are assigned to system components:

- Glass cover temperature: $T_g = T_{fi_i=0} + 10^\circ\text{C}$
- Absorber temperature: $T_r = T_{fi_i=0} + 20^\circ\text{C}$

3. Thermal Loss Calculation

The model computes heat transfer coefficients by convection h_{1c} and h_{2c} , and radiation h_{1r} , h_{2r} .

4. Cover Temperature Validation

An iterative verification process:

- a) Recalculates cover temperature using energy balance
- b) Compares with previous estimate: $\Delta T = |(T_g)_{i+1} - (T_g)_i|$
- c) Repeats calculation if $\Delta T > 0.01^\circ\text{C}$ (convergence threshold)
- d) Proceeds upon achieving thermal equilibrium

5. Collector Performance Parameters

Calculates the energy heat loss $\dot{q}_{loss}$, heat removal factor F_R , and efficiency factor F' .

6. Absorber Validation

Repeats the convergence process for the absorber:

- a) Solves updated absorber temperature
- b) Checks $\Delta T = |(T_r)_{i+1} - (T_r)_i|$
- c) Iterates until $\Delta T \leq 0.01^\circ\text{C}$

7. Performance Output Calculation

Determines system performance metrics:

- Heat transfer fluid outlet temperature T_{fo}
- Instantaneous collector efficiency η
- Useful heat gain $\dot{q}_u$

8. Time-Step Progression

Advances the simulation to subsequent time steps, repeating the entire calculation sequence

9. Results Export

Finalizes the computation by:

- Recording temperature profiles
- Outputting efficiency and heat gain data

The algorithm maintains strict convergence criteria ($\Delta T \leq 0.01^\circ\text{C}$) throughout all iterative processes, ensuring solution accuracy while preserving computational efficiency. This structured approach enables comprehensive parametric analysis while remaining faithful to the steady-state assumptions.

III.4.2 Flowchart of the computational algorithm

The different steps mentioned Section III.4.1 are represented in the following flowchart:

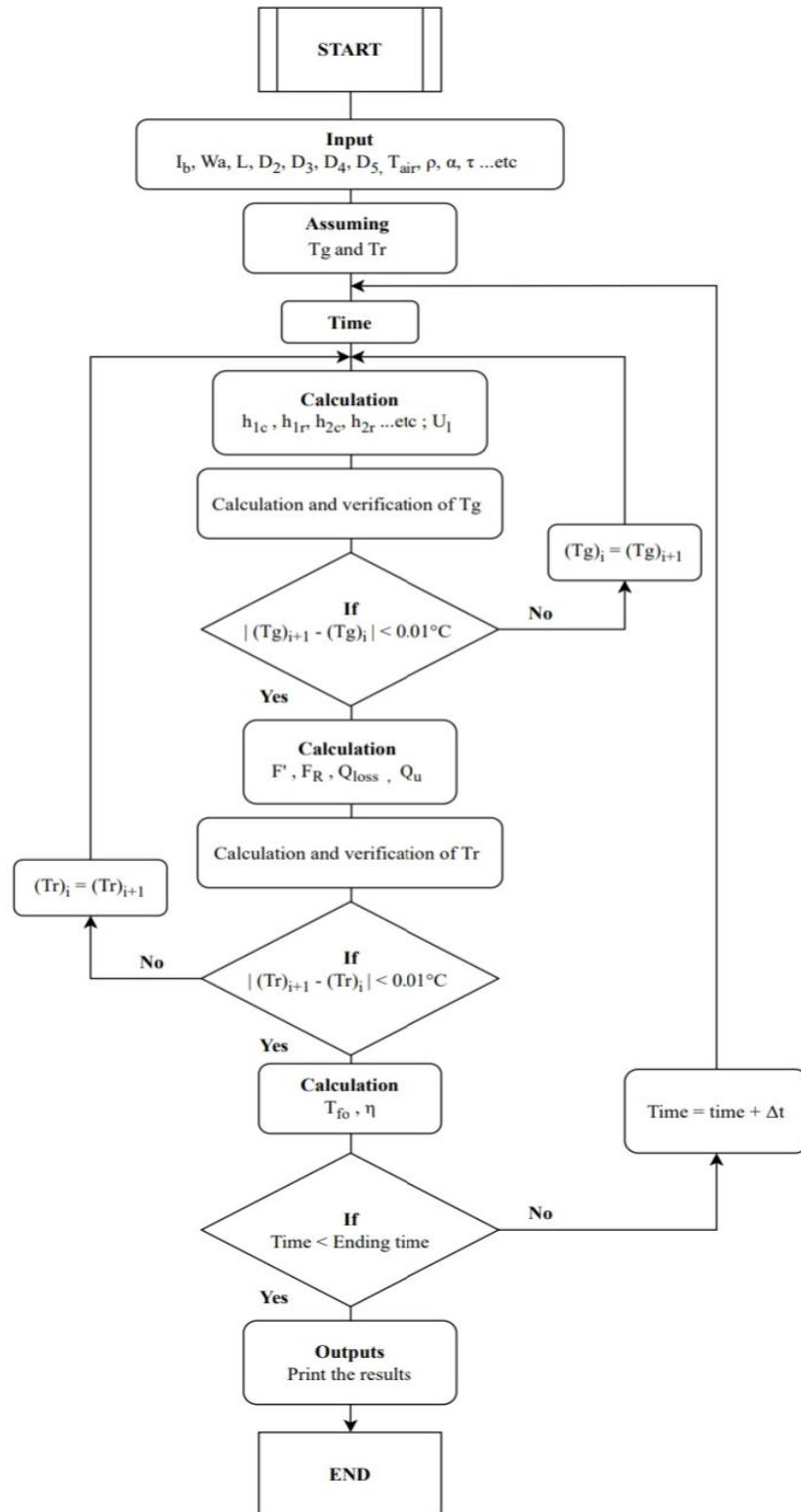


Figure 3.6 Flowchart of the PTC computational algorithm.

III.5 Conclusion

A comprehensive theoretical model of the parabolic trough collector (PTC) system has been developed in this chapter. The governing optical and thermal equations were formulated and structured into a computational algorithm, which was implemented using MATLAB to simulate the thermal performance of the collector. The simulation results provide meaningful insights into the behavior and efficiency of the system and will serve as a basis for further analysis in the following chapter.

CHAPTER FOUR

RESULTS AND DISCUSSION

IV.1 Introduction

This chapter presents and discusses the results obtained from the analytical model developed for the parabolic trough collector. It begins with a description of the geometrical configuration, optical system, and operating conditions used in the simulation. The model is then validated through comparison with existing numerical results to ensure its reliability. Following validation, the chapter examines the influence of different selective absorber coatings on the thermal performance of the collector. Key performance indicators—such as outlet fluid temperature, useful heat gain, and thermal efficiency—are analyzed to highlight the impact of coating properties.

IV.2 Geometrical Configuration and Operating Conditions

The geometrical configuration of the PTC is modeled based on the specifications of the Hassi R'Mel solar field. The receiver tube dimensions used in the simulation reflect those employed in the actual plant, ensuring the accuracy of the thermal and optical modeling. These dimensions play a crucial role in defining the concentration ratio, heat absorption, and thermal losses within the collector system. The geometrical characteristics of a single collector segment are presented in Table 4.1.

Table 4.1 Geometrical dimensions of the PTC

Geometrical dimensions	Value (m)
Mirror length (L)	12.27
Mirror width (W_a)	5.75
Internal diameter of the absorber (D_2)	0.034
External diameter of the absorber (D_3)	0.039
Internal diameter of the glass (D_4)	0.109
External diameter of the glass (D_5)	0.115

The optical system of the parabolic trough collector consists primarily of a reflective surface and a single absorber tube positioned along the focal line. The reflective surface is modeled as a continuous parabolic mirror that focuses incoming solar radiation onto the absorber to maximize energy capture.

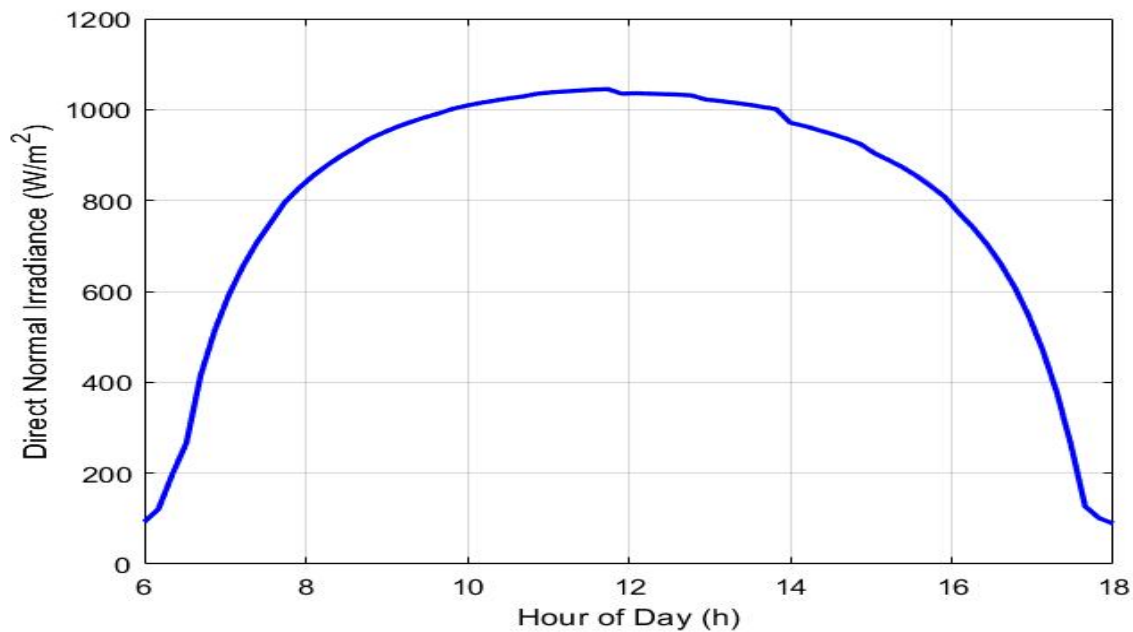
It is assumed that the optical properties—such as reflectivity and absorptivity—are uniform across the entire reflective surface. Additionally, solar tracking is considered to be highly accurate, maintaining the collector aperture perpendicular to the incident solar rays at all times. The key optical parameters used in the simulation are presented in Table 4.2.

Table 4.2 Optical properties of the PTC

Optical Property	Value
Mirror reflectivity (ρ_a)	0.92
Glass envelope transmissivity (τ_g)	0.945
Absorber tube absorptivity (α_{abs})	0.94
Emissivity of the glass envelope (ε_g)	0.935
Emissivity of the absorber tube (ε_{abs})	0.12

The weather data used in this study came from Guemar station, located El Oued Province, Algeria. The analysis focuses on June 11, 2014, a clear summer day ideal for evaluating thermal performance.

The simulation was driven by hourly measurements of direct normal irradiance (DNI) and ambient air temperature recorded at this location. In figure 4.1, the DNI exhibits a characteristic diurnal profile, peaking during midday hours, which directly influences the thermal output of the collector.

**Figure 4.1** Hourly measurements of direct normal irradiance (DNI) [61]

Similarly, figure 4.2 presents the hourly variation of ambient air temperature on the same day, which plays a critical role in estimating thermal losses and overall system efficiency.

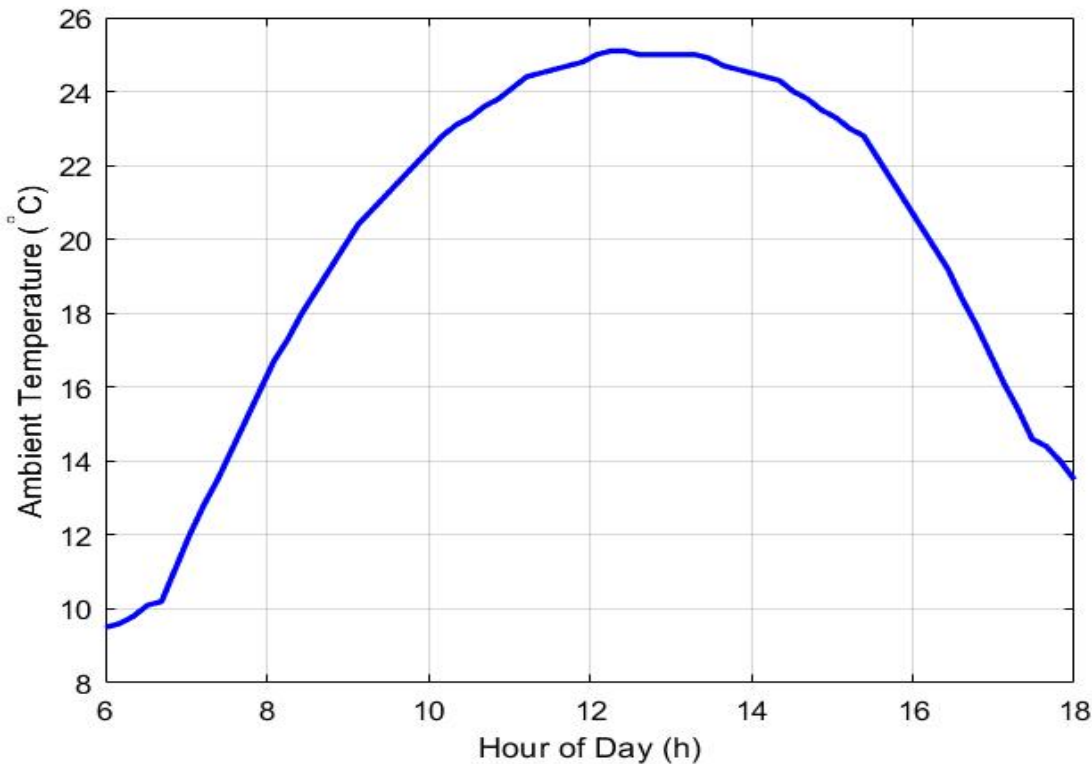


Figure 4.2 Hourly variation of ambient air temperature [61]

Tap water is used as the heat transfer fluid in this simulation due to its high specific heat capacity, environmental friendliness, and wide availability. The average mass flow rate of water circulating through the absorber tube is set at 0.06 kg/s. The inlet temperature of the fluid is assumed to be slightly above the ambient temperature, varying between 20 °C and 30 °C throughout the day, reflecting typical operating conditions. To simplify the numerical calculations, the thermo-physical properties of the fluid—such as density, specific heat, and thermal conductivity—are considered constant throughout the simulation, rather than being temperature-dependent.

IV.3 Validation of The Model

To support the validity of the present simulation, analytical thermal performance results are compared with numerical results obtained using the finite difference method, as reported by Ghodbane (2016) [61]. This comparison focuses on outlet fluid temperature, useful heat gain, and thermal efficiency, demonstrating the consistency between the current model and findings from established literature.

The outlet fluid temperature serves as a fundamental parameter for evaluating the thermal performance of a parabolic trough collector, as it reflects the system's ability to absorb and

transfer solar energy to the heat transfer fluid. figure 4.3 shows the hourly variation of outlet fluid temperature, the outlet temperature in the present simulation increases rapidly after sunrise, reaching a peak of just over 225 °C around 12:00. Meanwhile, the results reported by Ghodbane (2016) exhibit a slower thermal response, with a delayed peak of approximately 215 °C occurring around 15:00.

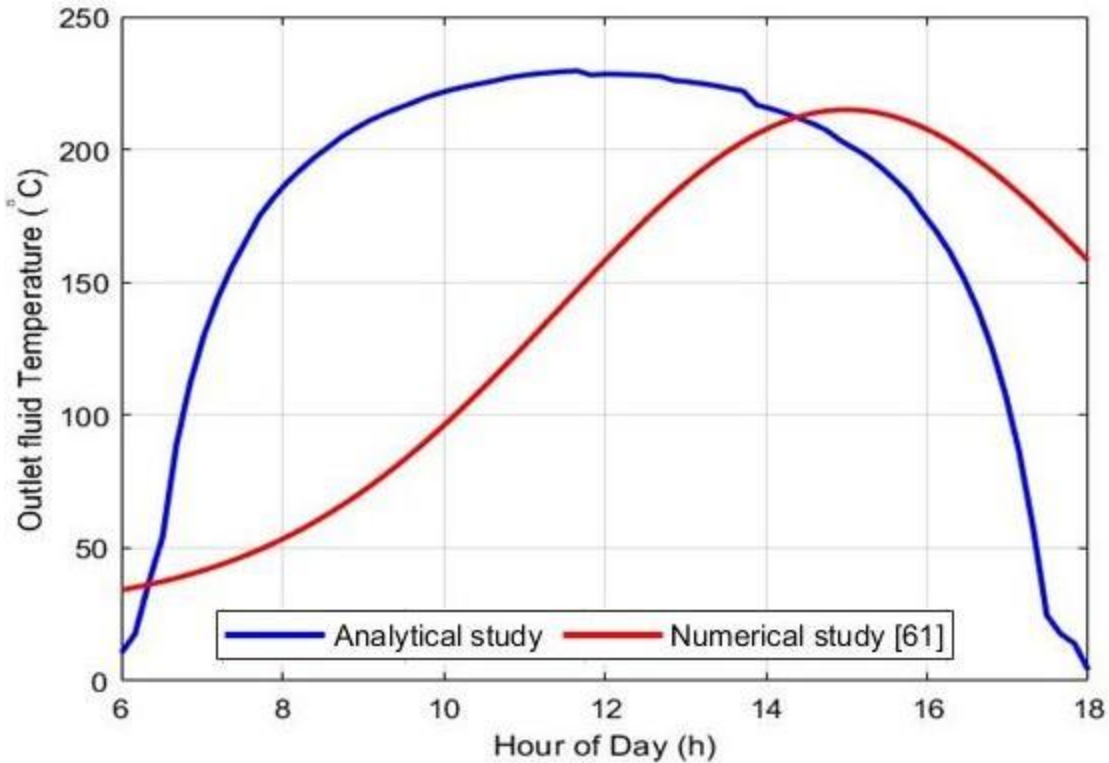


Figure 4.3 Hourly variation of outlet fluid temperature

Figure 4.4 illustrates the relative error between the outlet fluid temperatures obtained from the analytical and numerical studies throughout the day. The error is significantly high in the early morning, peaking at approximately 255% around 7:00. It then decreases gradually, dropping below 100% around 10:30 and reaching a minimum of 0% at approximately 14:00. After this point, the error begins to rise again, reaching about 90% by 18:00. The elevated error observed during the early and late hours of the day could be attributed to differences in model assumptions.

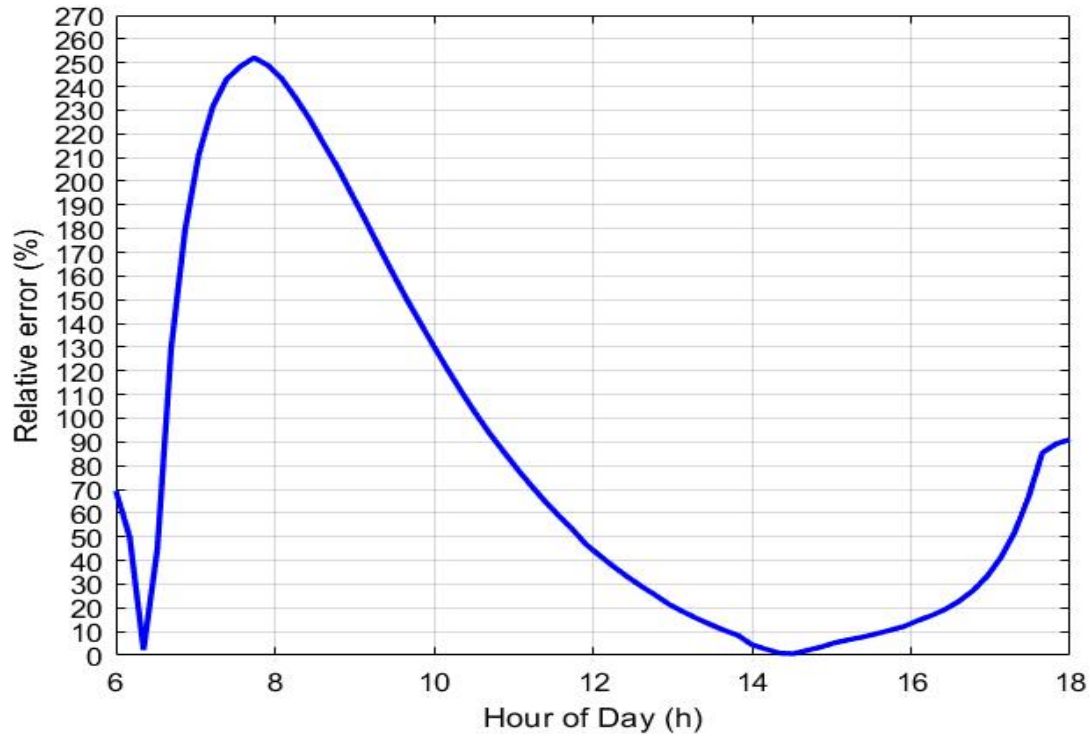


Figure 4.4 Relative error between analytical study and numerical study.

The thermal behavior of the absorber tube, outlet fluid, and glass cover are presented in figure 4.5. The absorber temperature exhibits the highest values throughout the day, peaking at approximately 265 °C around solar noon, as it directly receives concentrated solar radiation from the reflector. The outlet fluid temperature follows closely, indicating effective thermal energy transfer from the absorber to the working fluid via convection. The relatively small temperature difference between the absorber and the fluid suggests good thermal contact and a well-optimized heat exchange mechanism. In contrast, the glass cover maintains a significantly lower temperature profile, with a maximum of around 110 °C, due to its indirect exposure to solar irradiance and its role as a thermal barrier that reduces convective losses.

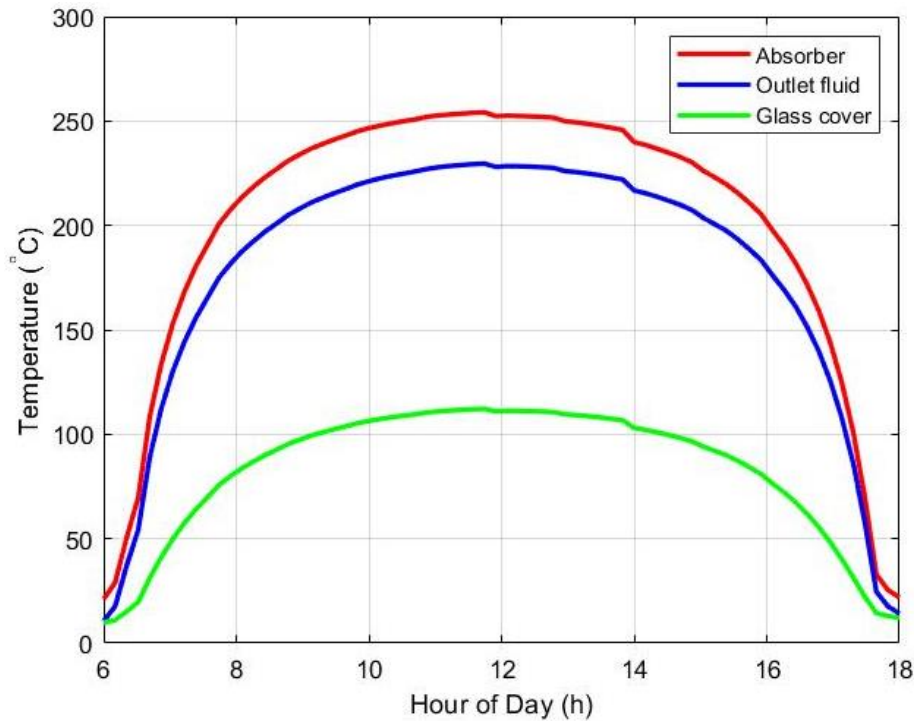


Figure 4.5 Hourly variation of absorber, outlet fluid and glass cover temperatures

Useful heat gain is a critical parameter in evaluating the thermal performance as it directly quantifies the amount of absorbed solar energy effectively transferred to the working fluid.

Figure 4.6 presents the hourly variation of useful heat gain. The analytical model gives higher values during the day, with a peak of about 730 W around noon, while the numerical result peaks at around 630 W. The sharper peak in the numerical result suggests that the model may have more thermal resistance, which limits the heat transfer rate compared to the analytical model.

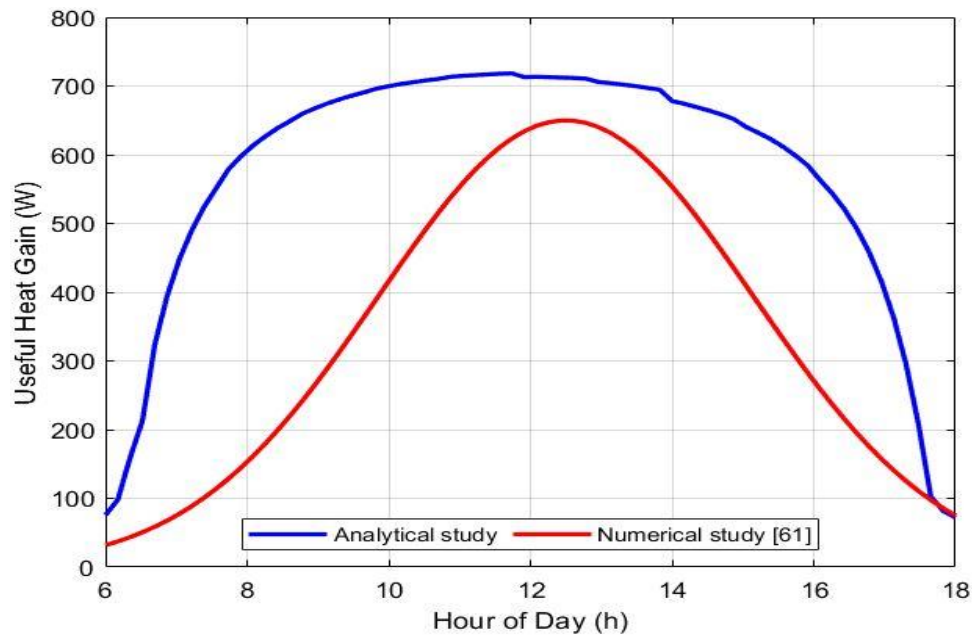


Figure 4.6 Hourly variation of useful heat gain

As for the thermal efficiency, which shows how well solar energy is turned into useful heat.

Figure 4.7 illustrates a slight decrease in the present model between 9:00 and 13:00, reaching a minimum of about 0.73 around 12:00, before rising again in the afternoon. In contrast, the numerical model remains nearly constant at around 0.83 throughout the entire period.

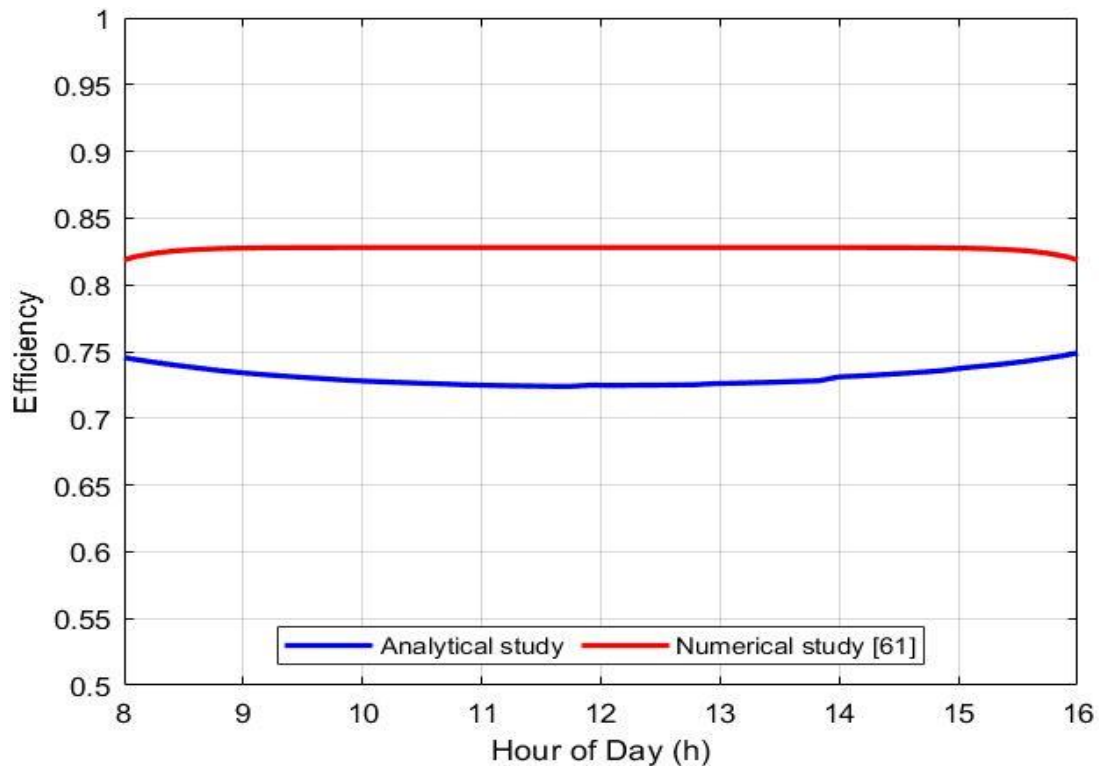


Figure 4.7 Hourly variation of thermal efficiency

While both analytical and numerical models provide valuable insights into the thermal performance of the parabolic trough collector, they should be compared with experimental results to verify their accuracy and reliability. Experimental validation is essential to confirm the predictions and account for real-world factors not fully captured in simulations. Overall, the present analytical model shows consistent trends in temperature, useful heat gain, and efficiency, indicating its potential for further development and optimization of solar thermal systems.

IV.4 Influence of Absorber Coating on Thermal Performance

The selective coating applied to the absorber tube significantly affects the thermal performance of parabolic trough collectors by maximizing solar absorptance and minimizing thermal emittance. This section analyzes the impact of five different selective coatings, along with a black surface as a non-selective reference, on the variation of outlet fluid temperature, useful heat gain, and thermal efficiency throughout the day. The comparison highlights how coating properties influence heat absorption and overall system efficiency..

Table 4.3 presents the detailed optical properties of the selective coatings considered in this study, including envelope transmittance, coating absorptance, and coating emittance at the representative operating temperature of 100 °C.

Table 4.3 Optical properties for each selective coating

Selective Coating	Envelope Transmittance	Coating Absorptance	Coating Emittance 100°C
Luz Cermet	0.935	0.92	0.06
Solel UVAC Cermet proposed a	0.97	0.98	0.04
Solel UVAC Cermet proposed b	0.97	0.955	0.02
Solel UVAC Cermet avg	0.965	0.955	0.076
Black surface	0.96	0.95	0.95

The variation in outlet fluid temperature across different selective coatings, presented in figure 4.8.

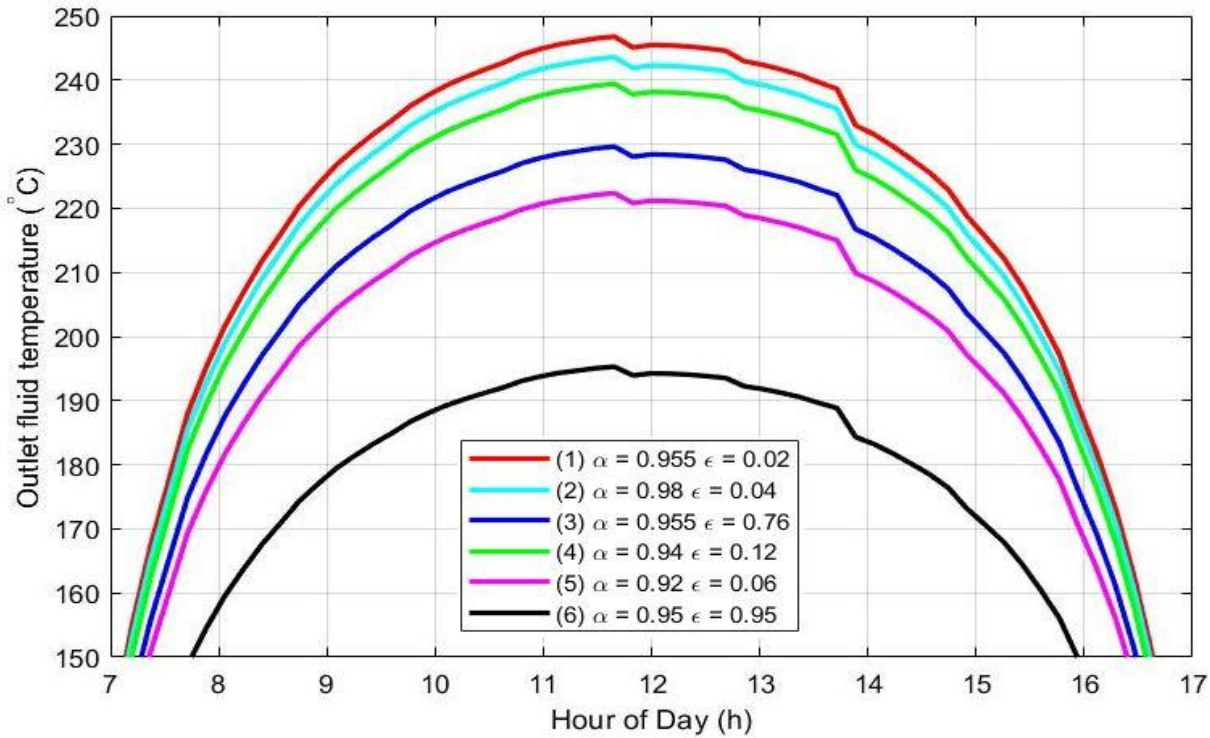


Figure 4.8 Hourly variation of outlet fluid temperature for different selective coatings

Coating (1), Proposed b, and coating (2), Proposed a, show the highest outlet temperatures, reaching up to 250 °C, due to their high absorptance and very low emittance, with Proposed b having the lowest at 0.02. Coating (4), Hassi R'Mel, reaches about 235 °C with moderate properties, while coating (5), Luz Cermet, peaks near 225 °C due to its higher emittance of around 0.15. Coating (6), the black surface, shows the lowest performance, with temperatures below 200 °C because of its very high emittance of 0.95. As transmittance is almost the same across all coatings, temperature variation is mainly influenced by differences in absorptance and emittance.

Figure 4.9 shows the variation of useful heat gain across different selective coatings reflects their influence on the thermal energy transferred to the working fluid.

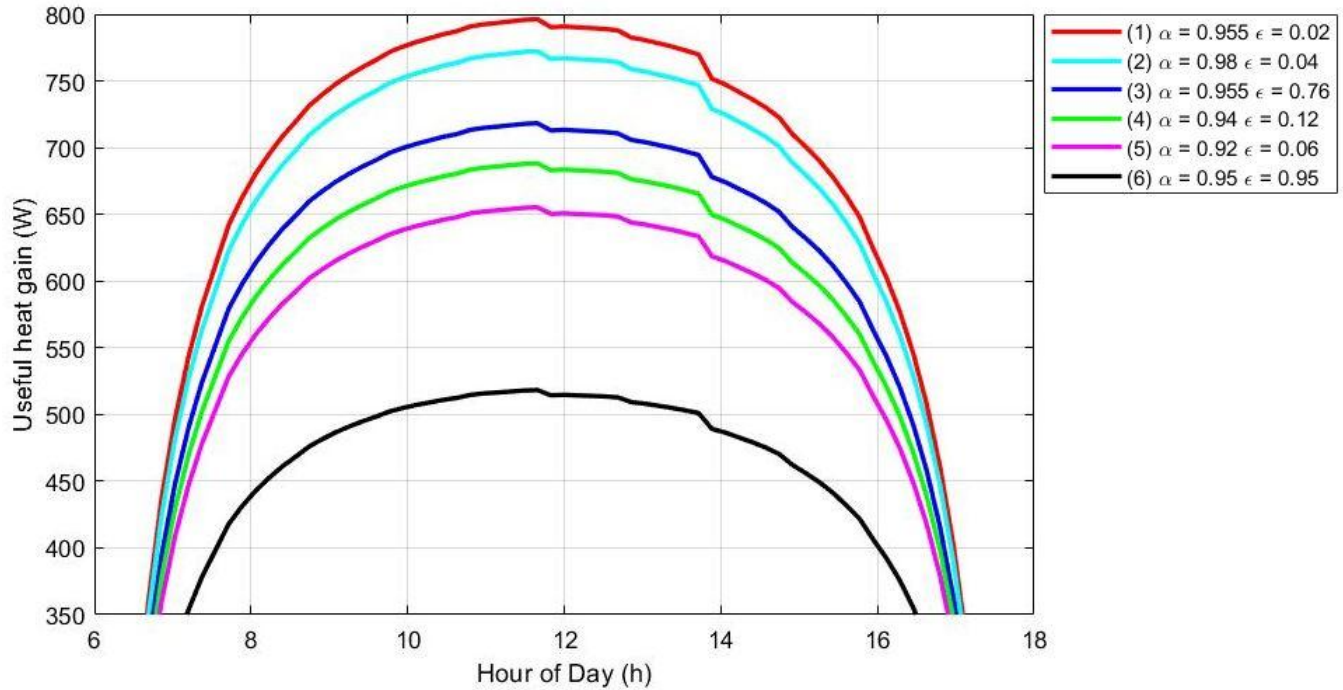


Figure 4.9 Hourly variation of useful heat gain for different selective coatings

Useful heat gain varies significantly with the type of selective coating. Coating (1), Proposed b, achieves the highest gain, reaching about 770 W, due to its high absorptance and very low emittance. Coatings (2) and (3) also perform well, staying above 700 W. Hassi R'Mel (4) and Luz Cermet (5) show moderate results, while the black surface (6) performs the worst, staying below 500 W due to its high emittance and greater thermal losses.

Figure 4.10 shows the evolution of thermal efficiency for different selective coatings. The thermal efficiency varies noticeably depending on the selective coating applied. Coating (1), Proposed b, consistently achieves the highest efficiency throughout the day, staying close to 85%, thanks to its low emittance of 0.02. Coating (2), Proposed a, follows with slightly lower efficiency due to its higher emittance. Coatings (3) and (4), representing the average and Hassi R'Mel respectively, show moderate performance, while Luz Cermet (5) records visibly lower efficiency because of its

higher thermal emittance around 0.15. The black surface (6) performs the worst, with efficiency dropping below 55%, confirming the significant impact of high emittance on thermal losses.

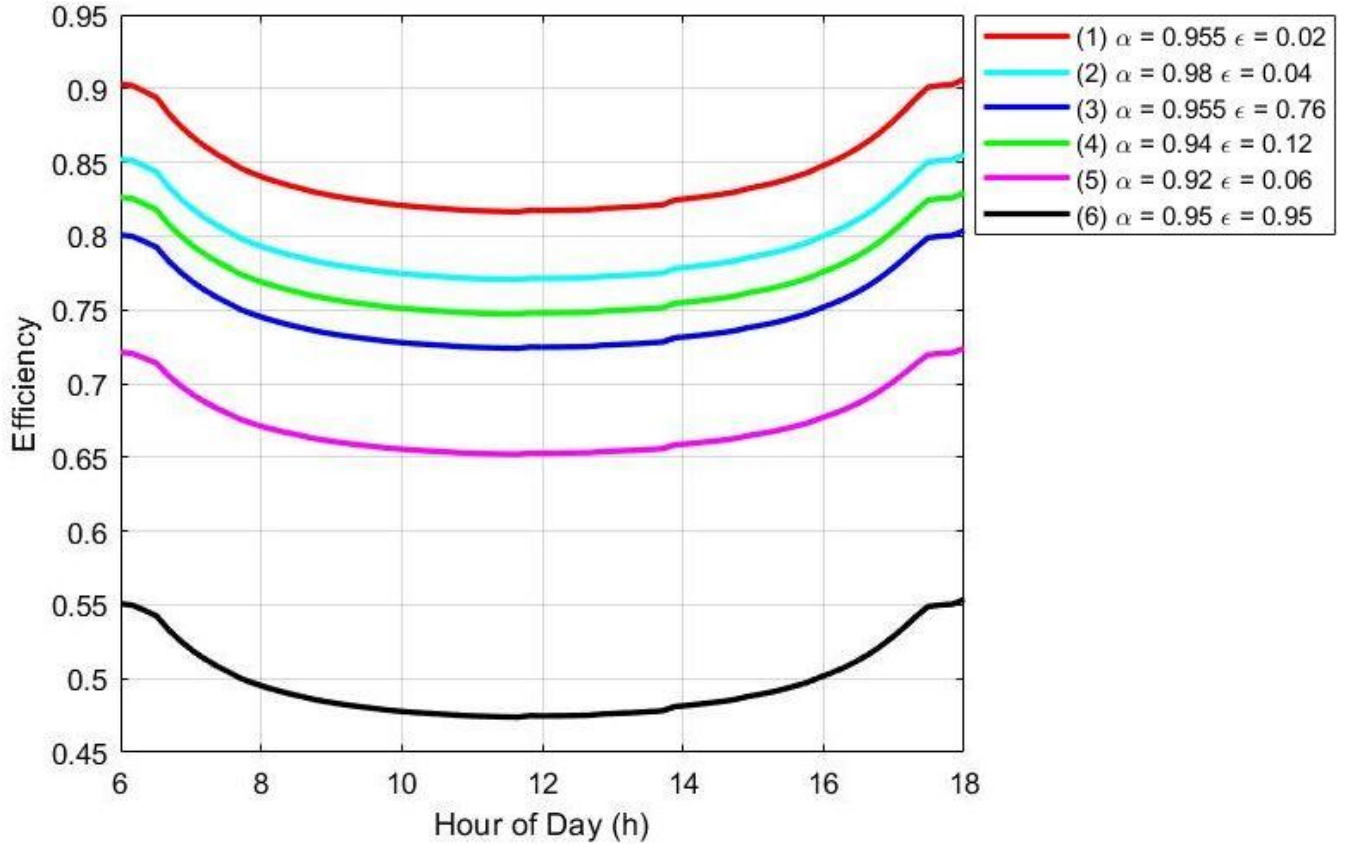


Figure 4.10 Hourly variation of thermal efficiency for different selective coatings

In short, selective coatings greatly influence the thermal performance of the collector. Coatings with high absorptance and low emittance show superior results in outlet temperature, useful heat gain, and efficiency, while high-emittance surfaces like the black coating perform the worst.

IV.4 Conclusion

In this chapter, the thermal behavior of the parabolic trough collector was analyzed through an analytical model. After validating the model, the influence of different selective coatings on the system's performance was examined. The results showed that the choice of absorber coating has a noticeable impact on temperature, heat gain, and efficiency. This highlights the importance of coating selection in improving the overall performance of solar thermal systems.

GENERAL CONCLUSION

As the world moves toward cleaner and more sustainable energy solutions, solar energy continues to gain importance due to its availability, environmental benefits, and potential for large-scale application. In this context, solar thermal systems have proven particularly valuable for medium- and high-temperature energy needs. Among these, Parabolic Trough Collectors (PTCs) stand out as a mature and widely adopted technology for harnessing solar power efficiently. Their effectiveness, combined with the ability to integrate thermal energy storage, makes them a strong candidate for supporting the global energy transition.

This dissertation focused on evaluating the thermal performance of a PTC system through an analytical modeling approach. The aim was to develop a simple, reliable model capable of simulating system behavior under realistic operating conditions, while maintaining computational efficiency. Each chapter contributed to building this understanding.

Chapter I reviewed the current state of solar thermal technologies, focusing on parabolic trough collectors (PTCs). It highlighted recent advancements in design, materials, and system performance, and emphasized the need for accurate yet simple modeling methods.

Chapter II provided the theoretical background of solar collectors, detailing their classifications, working principles, and the key optical and thermal mechanisms that influence the performance of PTC systems.

Chapter III developed an analytical model based on energy balance equations, integrating essential optical and thermal parameters. The model was implemented in MATLAB to simulate the behavior of the system under different conditions.

Chapter IV validated the analytical model by comparing its results with existing numerical results, confirming its reliability. It also assessed the impact of different absorber coatings, showing that materials with high absorptance and low emittance led to improved temperatures, heat gain, and efficiency.

In conclusion, this dissertation demonstrated that analytical modeling is a practical and efficient method for analyzing and optimizing the performance of PTC systems. The findings contribute valuable insights for future design improvements and support the broader goal of advancing solar thermal energy technologies.

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