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**CFD Simulation of Parabolic Trough Collectors
for Thermal Energy Production in Arid
Conditions of El Oued**

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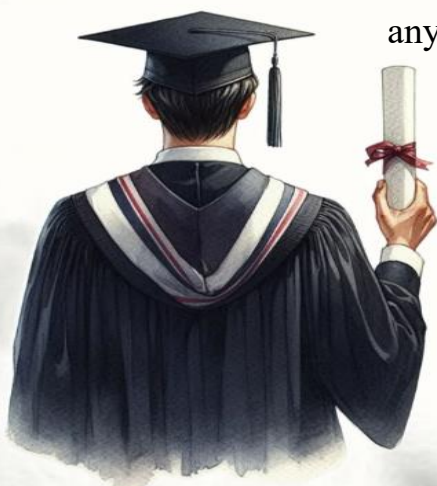
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DEDICATION

In the name of God, the Most Merciful, all praise be to
Allah, the Almighty.

We dedicate this humble work as a sign of respect,
appreciation, and gratitude.

To our beloved fathers, mothers, and brothers, who
supported us both near and far throughout this journey.

To all members of our families for their constant
encouragement and support.

To everyone who, in one way or another, contributed to
the realization and completion of this work.

All of you hold a special place in our hearts



Abstract

الملخص

Abstract

The research gap addressed in this study is the limited site-specific CFD assessment of parabolic trough collectors under the harsh arid conditions of El Oued, where high DNI, wind, temperature, and geometric constraints jointly affect thermal performance. This work aims to evaluate the thermal and hydraulic behavior of a parabolic trough receiver and identify the most influential operating and design parameters. A steady two-dimensional axisymmetric CFD-based numerical model was developed in MATLAB using finite-volume/finite-difference discretization, axial marching, nonlinear heat-loss boundary conditions, and validation through pressure-drop and Nusselt-number comparisons. Under baseline conditions of $\text{DNI} = 900 \text{ W/m}^2$, ambient temperature = 42°C , wind speed = 2 m/s , mass flow rate = 0.04 kg/s , aperture width = 1.5 m , receiver length = 2 m , and inner diameter = 0.025 m , the model predicted about 61.9% thermal efficiency, 1671.73 W useful heat gain, and 45°C outlet temperature. Increasing aperture width from 1 to 3 m raised useful heat from 1121 to 3312 W, while increasing DNI from 500 to 1100 W/m^2 increased it from 937.38 to 2037.71 W. Wind speed from 0.5 to 6 m/s increased heat losses from 89.67 to 189.22 W, and increasing mass flow rate from 0.01 to 0.10 kg/s reduced outlet temperature from 69.99 to 39.21°C but increased pressure drop from 1.47 to 57.16 Pa.

Keyword: Parabolic trough collector; CFD simulation; El Oued; Arid climate; Useful heat gain; Thermal efficiency.

الملخص

تتناول هذه الدراسة محدودية التقييم العددي المحلي باستخدام ديناميكا الموائع الحسابية للمجمعات الشمسية ذات القطع المكافئ تحت الظروف القاسية الجافة لمنطقة الوادي، حيث تؤثر شدة الإشعاع الشمسي المباشر، والرياح، ودرجة الحرارة، والقيود الهندسية معاً في الأداء الحراري. يهدف هذا العمل إلى تقييم السلوك الحراري والهيدروليكي لمستقبل المجمع الشمسي وتحديد أهم معاملات التشغيل والتصميم تأثيراً. تم تطوير نموذج عددي ثنائي الأبعاد متناظر محورياً ومستقر في برنامج MATLAB باستخدام طريقة الحجوم/الفروق المحدودة، والتقدم المحوري، وشروط حدودية غير خطية للفواقد الحرارية، مع التحقق من النموذج من خلال مقارنة هبوط الضغط وعدد نسلت. عند الشروط المرجعية $\text{DNI} = 900 \text{ W/m}^2$ ، ودرجة حرارة محيطية = 42°C ، وسرعة رياح = 2 m/s ، ومعدل تدفق كتلي = 0.04 kg/s ، وعرض فتحة = 1.5 m ، وطول مستقبل = 2 m ، وقطر داخلي = 0.025 m ، تنبأ النموذج بكفاءة حرارية تقارب 61.9%، وحرارة مفيدة قدرها 1671.73 W، ودرجة حرارة خروج قدرها 45°C . أدى رفع عرض الفتحة من 1 إلى 3 m إلى زيادة الحرارة المفيدة من 1121 إلى 3312 W، كما أدى رفع DNI من 500 إلى 1100 W/m^2 إلى زيادتها من 937.38 إلى 2037.71 W. وزادت الفواقد الحرارية من 89.67 إلى 189.22 W عند رفع سرعة الرياح من 0.5 إلى 6 m/s ، بينما أدى رفع معدل التدفق من 0.01 إلى 0.10 kg/s إلى خفض درجة حرارة الخروج من 69.99 إلى 39.21°C ، مع زيادة هبوط الضغط من 1.47 إلى 57.16 Pa.

الكلمات المفتاحية: المجمع الشمسي ذو القطع المكافئ؛ محاكاة CFD؛ الوادي؛ المناخ الجاف؛ الحرارة المفيدة؛ الكفاءة الحرارية.

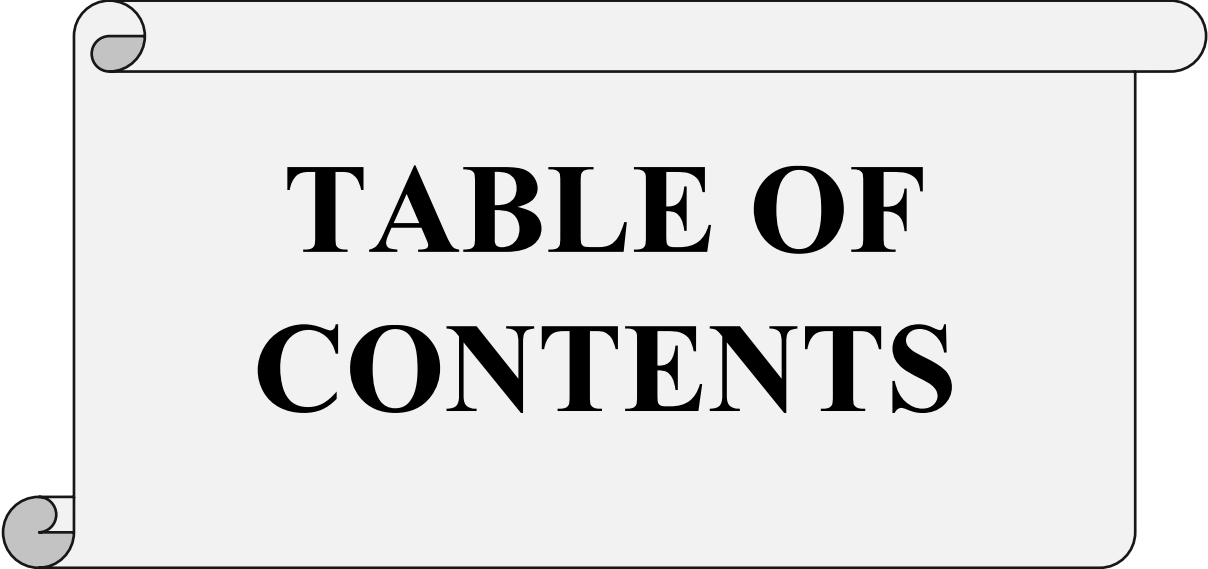


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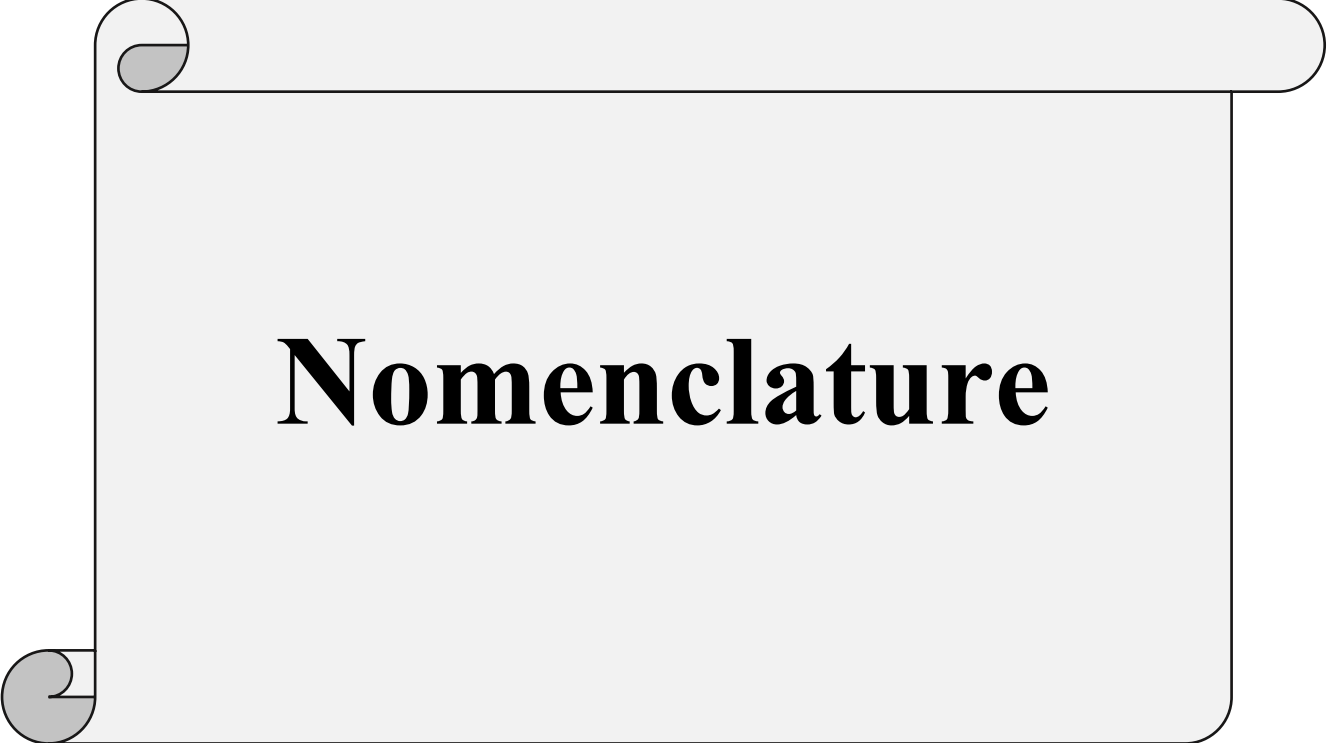
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Nomenclature

Nomenclature

Latin Symbols

Symbol	Description	Unit
A_{ap}	Collector aperture area	m^2
A_f	Internal flow area of the absorber tube	m^2
A_j	Area of the radial control volume j	m^2
A^+	Van Driest damping constant	—
A_q	Axial heat-flux non-uniformity amplitude	—
$a_{conv,j}$	Axial convective coefficient of control volume j	$\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$
a_E, a_W, a_P	East, west, and central finite-volume coefficients	$\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$
B_j	Main diagonal coefficient in the tridiagonal system	—
C_j	Upper diagonal coefficient in the tridiagonal system	—
c_p	Specific heat capacity of the heat-transfer fluid	$\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$
D_i	Inner diameter of the absorber tube	m
D_o	Outer diameter of the absorber tube	m
D_j	Right-hand-side term in the tridiagonal system	—
DNI	Direct normal irradiance	$\text{W} \cdot \text{m}^{-2}$
e_w	Absorber wall thickness	m
f	Darcy friction factor	—
$F(T_{wo})$	Residual function of the nonlinear wall-temperature equation	$\text{W} \cdot \text{m}^{-1}$
G_b	Conductance between absorber wall and near-wall fluid cell	$\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$
$G_{e,j}$	East radial diffusion conductance	$\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$
$G_{w,j}$	West radial diffusion conductance	$\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$
h_i	Internal convective heat-transfer coefficient	$\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$
h_{wind}	External wind-dependent convective heat-transfer coefficient	$\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$
IAM	Incidence angle modifier	—
k_{eff}	Effective radial thermal conductivity	$\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$
k_f	Molecular thermal conductivity of the fluid	$\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$
k_s	Thermal conductivity of the absorber material	$\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$
L	Receiver length	m
LF	Heat-loss fraction	—
$\dot{m}$	Mass flow rate of the heat-transfer fluid	$\text{kg} \cdot \text{s}^{-1}$
N_r	Number of radial control volumes	—
N_x	Number of axial nodes	—
Nu	Local Nusselt number	—
Nu_G	Nusselt number from Gnielinski correlation	—
Nu_{lam}	Laminar reference Nusselt number	—
P_o	External perimeter of the absorber tube	m
Pe	Peclet number	—
Pr	Prandtl number	—
Pr_t	Turbulent Prandtl number	—

q'_{abs}	Local absorbed heat rate per unit receiver length	$W \cdot m^{-1}$
$q'_{abs,mean}$	Mean absorbed heat rate per unit receiver length	$W \cdot m^{-1}$
q'_{conv}	Convective heat loss per unit length	$W \cdot m^{-1}$
q'_{fluid}	Heat transferred from the absorber wall to the fluid per unit length	$W \cdot m^{-1}$
q'_{loss}	Total external heat loss per unit length	$W \cdot m^{-1}$
q'_{rad}	Radiative heat loss per unit length	$W \cdot m^{-1}$
Q_{abs}	Total absorbed solar heat by the receiver	W
Q_{fluid}	Integrated heat transferred to the fluid	W
Q_{loss}	Integrated external heat loss	W
Q_{solar}	Incident solar power on the collector aperture	W
Q_u	Useful heat gain	W
r	Radial coordinate	m
$r_{e,j}$	East face radius of radial control volume j	m
r_p	Radial position of near-wall fluid-cell center	m
$r_{w,j}$	West face radius of radial control volume j	m
R_i	Inner radius of the absorber tube	m
R_o	Outer radius of the absorber tube	m
$R_{f,1/2}$	Fluid half-cell thermal resistance	$K \cdot m \cdot W^{-1}$
R_s	Absorber-wall conduction resistance	$K \cdot m \cdot W^{-1}$
Re	Reynolds number	—
$S(x)$	Normalized axial heat-flux shape function	—
$S_0(x)$	Unnormalized axial heat-flux shape function	—
S_p	Linearized wall-source slope	$W \cdot m^{-1} \cdot K^{-1}$
S_u	Linearized wall-source intercept	$W \cdot m^{-1}$
T	Fluid temperature	$^{\circ}C$ or K
T_b	Bulk fluid temperature	$^{\circ}C$
T_{amb}	Ambient temperature	$^{\circ}C$
T_{in}	Inlet fluid temperature	$^{\circ}C$
T_{out}	Outlet bulk fluid temperature	$^{\circ}C$
T_p	Near-wall fluid-cell temperature	$^{\circ}C$
T_{sky}	Effective sky temperature	$^{\circ}C$
T_{wi}	Inner absorber-wall temperature	$^{\circ}C$
T_{wo}	Outer absorber-wall temperature	$^{\circ}C$
$u(r)$	Axial velocity profile	$m \cdot s^{-1}$
U_{avg}	Average fluid velocity	$m \cdot s^{-1}$
u_{τ}	Friction velocity	$m \cdot s^{-1}$
v_{wind}	Wind speed	$m \cdot s^{-1}$
W_{ap}	Collector aperture width	m
x	Axial coordinate	m
y	Distance from the wall	m
y^+	Dimensionless wall coordinate	—

Greek Symbols

Symbol	Description	Unit
α_{abs}	Absorber solar absorptivity	—
α_t	Turbulent thermal diffusivity	$m^2 \cdot s^{-1}$

γ	Transitional-flow blending factor	—
Δp	Pressure drop along the absorber tube	Pa
Δr	Radial step size	m
ΔT	Fluid temperature rise	K or °C
ΔT_{sky}	Sky-temperature offset	°C
Δx	Axial step size	m
δ_{Φ}	Relative variation of a monitored variable during grid-independence analysis	%
ε	Absorber emissivity	—
ε_B	Bulk closure error	—
ε_E	Energy residual	—
ε_{NL}	Nonlinear convergence error	K
ε_{tol}	Nonlinear convergence tolerance	K
ε_{wall}	Wall-temperature convergence tolerance	K
η_{opt}	Optical efficiency	—
η_{rec}	Receiver efficiency	—
η_{th}	Thermal efficiency	—
κ	Von Karman constant	—
μ	Dynamic viscosity of the fluid	Pa·s
ν	Molecular kinematic viscosity	m ² ·s ⁻¹
ν_t	Turbulent eddy viscosity	m ² ·s ⁻¹
ρ	Fluid density	kg·m ⁻³
σ	Stefan–Boltzmann constant	W·m ⁻² ·K ⁻⁴
$\phi(r)$	Velocity-profile shape function	—
ϕ_{lam}	Laminar velocity-profile shape function	—
ϕ_{plug}	Plug-flow velocity-profile shape function	—
ϕ_{tr}	Transitional velocity-profile shape function	—
ϕ_{turb}	Turbulent velocity-profile shape function	—
Φ	Generic monitored performance variable	—
ω	Under-relaxation factor	—

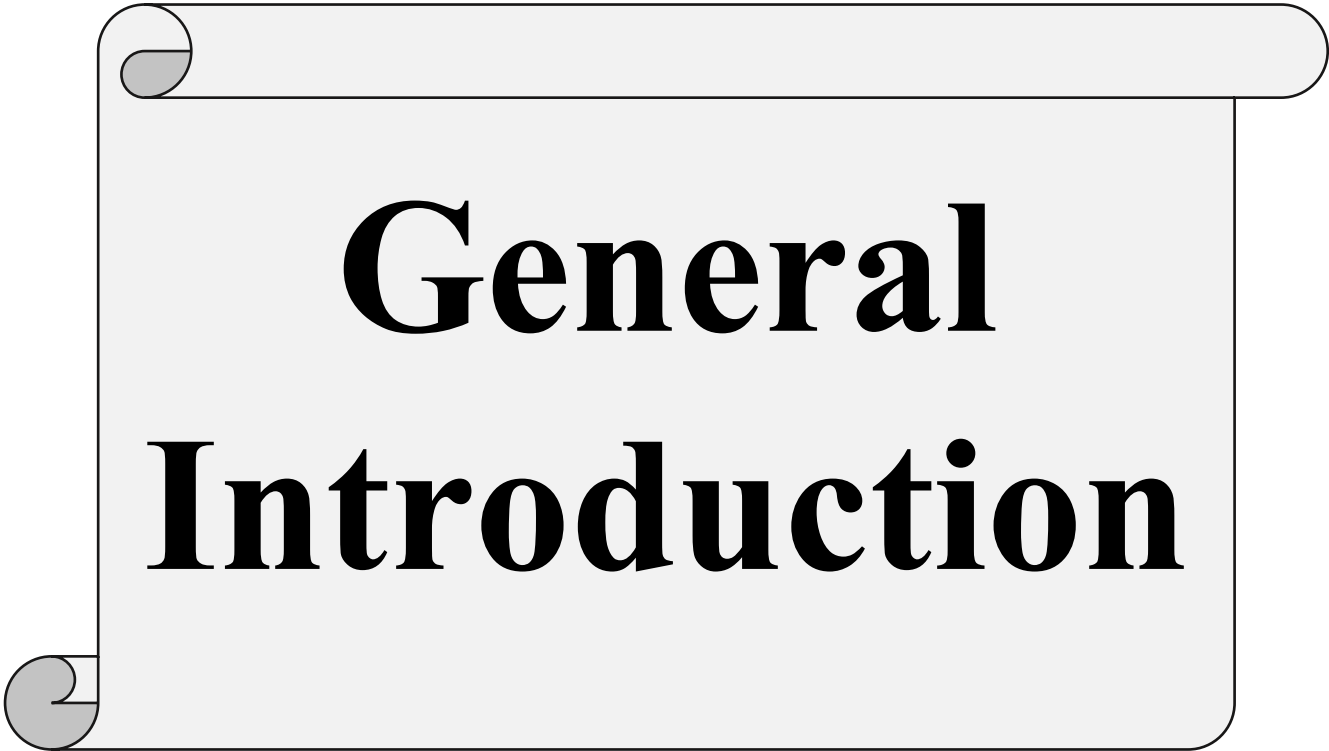
Subscripts and Superscripts

Symbol	Meaning
<i>abs</i>	Absorbed or absorber
<i>amb</i>	Ambient
<i>ap</i>	Aperture
<i>b</i>	Bulk
<i>conv</i>	Convective
<i>e</i>	East face of a control volume
<i>E</i>	East coefficient or energy residual
<i>f</i>	Fluid
<i>fluid</i>	Heat transferred to the fluid
<i>G</i>	Gnielinski correlation
<i>i</i>	Inner, inlet, or axial station depending on context
<i>j</i>	Radial control-volume index
<i>lam</i>	Laminar
<i>loss</i>	Heat loss

<i>mean</i>	Mean value
<i>min, max</i>	Minimum and maximum values
<i>NL</i>	Nonlinear
<i>o</i>	Outer
<i>opt</i>	Optical
<i>P</i>	Near-wall fluid cell
<i>rad</i>	Radiative
<i>rec</i>	Receiver
<i>s</i>	Solid absorber material
<i>sky</i>	Sky
<i>solar</i>	Incident solar radiation
<i>th</i>	Thermal
<i>tol</i>	Tolerance
<i>tr</i>	Transitional
<i>turb</i>	Turbulent
<i>u</i>	Useful
<i>w</i>	Wall or west face
<i>wi</i>	Inner absorber wall
<i>wo</i>	Outer absorber wall
<i>'</i>	Per unit receiver length
<i>*</i>	Iterative estimate or modified coefficient
<i>new, old, sol</i>	New, old, and solved values in nonlinear iteration

Abbreviations

Abbreviation	Meaning
CFD	Computational Fluid Dynamics
CSP	Concentrated Solar Power
DNI	Direct Normal Irradiance
HTF	Heat Transfer Fluid
IAM	Incidence Angle Modifier
PTC	Parabolic Trough Collector
TES	Thermal Energy Storage
TDMA	Tridiagonal Matrix Algorithm
MENA	Middle East and North Africa
NREL	National Renewable Energy Laboratory



General Introduction

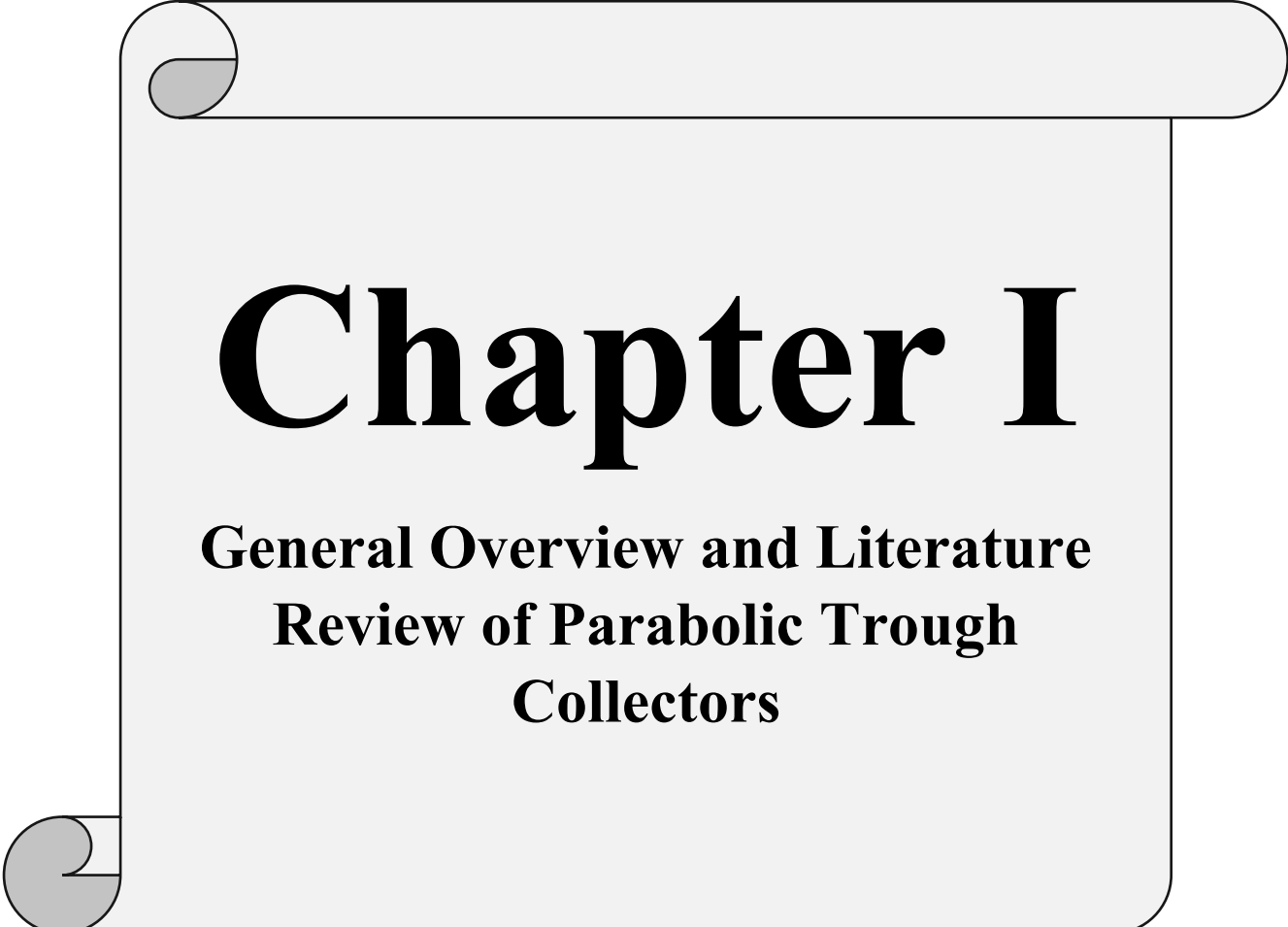
General Introduction

Parabolic trough collectors are among the most mature and widely used concentrated solar power technologies. Their operating principle is based on the concentration of direct normal irradiance by a parabolic reflector onto a receiver tube positioned along the focal line. A heat transfer fluid flows inside the absorber tube and collects the thermal energy, which can then be transferred to a heat exchanger, storage system, or power cycle. The thermal efficiency of this system depends on optical, geometrical, thermal, and hydraulic parameters, including aperture width, receiver diameter, mass flow rate, direct normal irradiance, wind speed, and ambient temperature. Therefore, accurate prediction of the thermal and hydraulic behavior of the receiver is essential for improving the design and operation of parabolic trough collectors under local climatic conditions.

In arid regions such as El Oued, the high availability of direct solar radiation represents a major advantage for concentrating solar technologies. Nevertheless, the harsh desert climate may reduce collector performance through increased convective heat losses, receiver temperature gradients, and operational constraints. Experimental investigations under such conditions can be costly and time-consuming, especially when several design and operating parameters must be analyzed. For this reason, computational fluid dynamics provides a powerful numerical approach for studying heat transfer and fluid flow inside the receiver tube. CFD makes it possible to predict temperature distributions, outlet temperature, useful heat gain, pressure drop, Nusselt number, heat losses, and thermal efficiency with detailed spatial resolution.

The present dissertation, entitled “CFD Simulation of Parabolic Trough Collectors for Thermal Energy Production in Arid Conditions of El Oued,” focuses on the numerical analysis of a parabolic trough receiver operating under representative desert conditions. The main objective is to evaluate the coupled thermal and hydraulic performance of the receiver and to identify the influence of key parameters such as solar irradiance, wind speed, aperture width, receiver inner diameter, and mass flow rate. The numerical model is based on the governing equations of fluid flow and heat transfer, supported by appropriate boundary conditions, heat-loss correlations, and validation using pressure-drop and Nusselt-number comparisons.

This study contributes to a better understanding of parabolic trough collector behavior in arid environments and provides useful guidance for the optimization of solar thermal systems in El Oued. By combining numerical modeling with parametric analysis, the work aims to support the development of efficient, reliable, and locally adapted solar thermal technologies for sustainable energy production in desert regions.



Chapter I

**General Overview and Literature
Review of Parabolic Trough
Collectors**

Chapter I:

General Overview and Literature Review of Parabolic Trough Collectors

1.1 Background and Motivation

Arid and semi-arid regions are becoming a strategic frontier for renewable thermal energy because they combine three structural pressures: very high solar availability, fast-growing cooling demand, and increasing water stress. In North Africa, for example, annual average solar irradiation is about 2,200 kWh/m², while average wind speeds are around 7 m/s, reaching 9.5 m/s in Algeria and Libya. At the same time, the region's growing population and rising living standards are increasing demand for sustainable energy services. These conditions make arid regions particularly suitable for the deployment of solar-based thermal systems for water heating, process heat, cooling, and desalination [1].

The motivation for renewable thermal energy is not limited to electricity generation. In the Arab region, strategic planning documents explicitly identify heating and cooling and water desalination as major untapped fields for renewable energy deployment. This is important because thermal end uses often represent a large and persistent share of final energy demand, especially in hot climates where cooling is essential for buildings, public services, food preservation, and industrial activity. The Pan-Arab Renewable Energy Strategy therefore treats renewable heat, cooling, and desalination not as peripheral applications, but as central components of long-term regional energy transition planning [2].

Water scarcity further strengthens the case for renewable thermal technologies in desert environments. According to IRENA, the Middle East and North Africa accounts for about 38% of global desalination capacity, while regional water demand is expected to rise from 9 billion m³ in 2010 to 13.3 billion m³ by 2030. Yet desalination remains highly energy-intensive and has historically depended on fossil fuels. Renewable thermal energy offers an alternative pathway, especially where solar heat can directly support thermal desalination or where renewable electricity and heat can reduce the fuel burden of water production. In this sense, renewable thermal systems address both the energy crisis and the water crisis simultaneously [3].

Among the available technologies, concentrated solar power (CSP) is especially relevant in arid regions because it can supply not only electricity but also usable heat for industrial processes and desalination. IRENA and IEA-ETSAP note that the joint production of electricity, heat, and desalinated water is of particular interest in arid zones, and that in MENA countries CSP-based desalination could

become economically competitive. This multi-service character gives renewable thermal energy a strategic advantage over single-purpose systems, particularly in remote or infrastructure-constrained desert territories where one integrated platform can serve several basic needs at once [4].

Another major driver is the rapid rise in cooling demand in hot climates. Research from the IEA Solar Heating and Cooling Programme shows that air-conditioning already accounts for nearly 20% of global building electricity demand, and that demand is growing fastest in warm countries. Solar cooling is especially promising in sunbelt regions because cooling demand rises precisely when solar radiation is strongest. The same research also stresses that system design in arid environments must account for boundary conditions such as high temperature, dust, humidity, wind, and limited water availability. This makes renewable thermal energy not only an environmental option, but also a technically adaptive response to the realities of desert climates [5].

Overall, the background to renewable thermal energy in arid regions is rooted in a clear development logic: desert and semi-desert zones possess exceptional solar resources, but also face acute challenges in cooling, water security, and fossil-fuel dependence. The motivation for adopting renewable thermal technologies is therefore both environmental and functional. These systems can reduce emissions, diversify energy supply, improve resilience in remote areas, and support essential services such as cooling and desalination. For arid-region countries, especially in the Arab and North African context, renewable thermal energy is no longer a secondary option; it is increasingly a practical foundation for sustainable regional development.

1.2 Overview of Solar Thermal Technologies

1.2.1 Concentrated Solar Power Technologies

Concentrated Solar Power (CSP) refers to a group of solar thermal technologies that use mirrors or reflective surfaces to concentrate direct normal irradiance onto a receiver, where solar energy is converted into heat and then used to produce steam for electricity generation through a conventional turbine cycle. Unlike solar photovoltaic systems, CSP is based on thermal conversion, which makes it particularly valuable in high-irradiation desert and semi-arid regions where strong sunshine and clear skies are available for most of the year [4,6].

A major technical advantage of CSP is that it can be coupled with thermal energy storage, especially molten-salt storage, allowing electricity generation after sunset or during short periods of cloud cover. This feature gives CSP a degree of dispatchability that distinguishes it from variable solar technologies and makes it relevant for power systems that require greater operational flexibility and peak-load support. The International Energy Agency notes that integrated thermal storage is one of the

most important features of CSP because it improves firmness and grid value, especially in systems with growing shares of variable renewables [6,7].

According to IRENA and ESCWA, CSP technologies are generally classified into four main types: parabolic trough, linear Fresnel reflector, solar tower, and parabolic dish systems. These technologies differ in optical design, concentration ratio, operating temperature, commercial maturity, and storage compatibility. In broad terms, parabolic trough and Fresnel systems concentrate sunlight onto a linear receiver, whereas solar tower and dish systems focus sunlight onto a point receiver, enabling higher temperatures and, in some cases, higher thermodynamic efficiency [4,8].

Parabolic trough (PT) technology is the most mature and historically the most widely deployed CSP option. It uses long, curved mirrors to focus sunlight onto a receiver tube positioned along the focal line. A heat-transfer fluid, often synthetic oil and in some configurations molten salt or steam, absorbs the solar heat and transfers it to a steam generator. IRENA identifies parabolic trough as the dominant commercial CSP technology and notes that its maturity has made it the lowest-risk option for utility-scale deployment, especially in early large-scale projects [4,9].

Linear Fresnel reflector (LFR) systems use long rows of nearly flat or slightly curved mirrors that track the sun and reflect radiation onto a fixed elevated receiver. Their structure is mechanically simpler and potentially cheaper than parabolic trough systems, but they generally operate with lower optical efficiency and, historically, lower commercial maturity. Their main attraction lies in the possibility of reducing structural and manufacturing costs while still delivering medium- to high-temperature solar heat [4,9].

Solar tower (ST) technology, also called central receiver technology, uses a field of heliostats that reflect sunlight onto a receiver mounted on top of a tower. Because this design can achieve much higher concentration ratios and operating temperatures than line-focus systems, it offers significant potential for higher cycle efficiency and improved thermal storage performance. IRENA's comparative assessment states that solar towers are among the most promising CSP technologies for future development, particularly when molten salt is used both as heat-transfer and storage medium, because this configuration can support higher efficiency and longer storage duration [4,9].

Parabolic dish (SD) systems use dish-shaped reflectors to focus sunlight onto a receiver at a single focal point, usually coupled to a Stirling engine or another small-scale power conversion unit. Dish systems can achieve very high temperatures and high conversion efficiencies at small scale, but they have had more limited commercial deployment than parabolic trough or solar tower systems. For that reason, they remain less common in grid-scale planning despite their strong technical performance in niche applications [4].

A practical Arab-region example is Shams 1 in Abu Dhabi, a 100 MW parabolic trough plant developed by Masdar. According to Masdar's official project factsheet, Shams 1 demonstrates how CSP can be deployed under desert conditions while contributing to peak power supply and energy diversification. The project also illustrates the broader strategic value of CSP in arid climates, where high solar resource availability aligns with daytime electricity demand and where dispatchable solar thermal generation can complement national decarbonization efforts [10].



Figure I 1: Concentrated Solar Power Technologies. U.S. Department of Energy [11].

1.2.2 Position of Parabolic Trough Collectors within CSP Systems

Within the broader family of Concentrated Solar Power (CSP) technologies, parabolic trough collectors (PTCs) occupy the position of the most established and commercially mature line-focus technology. CSP systems are commonly divided into two optical groups: line-focusing systems, which include parabolic trough and linear Fresnel collectors, and point-focusing systems, which include solar towers and solar dishes. In this classification, the parabolic trough is the reference line-focus configuration and has historically served as the benchmark technology for utility-scale CSP deployment, especially in regions with strong direct normal irradiance such as the Middle East and North Africa (MENA) [4,12].

Technically, the parabolic trough system is positioned at the solar-field level of a CSP plant as the component responsible for capturing and concentrating solar radiation onto a linear receiver tube. The reflected solar energy heats a working fluid, typically synthetic oil in many commercial plants, which then transfers thermal energy to the steam-generation block. In other words, the parabolic trough is not a separate power plant category outside CSP; rather, it is one of the principal collector-field architectures through which a CSP plant obtains thermal input for the conventional Rankine-cycle power block [4].

Its strategic position inside CSP systems is reinforced by its commercial dominance and operational bankability. IRENA states that parabolic trough technology was the most commercially mature CSP option and accounted for more than 90% of installed CSP capacity at the time of its technology brief, while the Fraunhofer–MENA assessment similarly described trough technology as the most commercial CSP technology, representing 94% of installed CSP plant capacity in the market context it reviewed. This means that, for many years, parabolic trough collectors were effectively the default architecture for early large-scale CSP development and the main route through which investors, utilities, and policymakers gained practical experience with solar thermal electricity generation [4,13].

The position of parabolic trough collectors is especially important in arid-region CSP strategy because they have been repeatedly selected for flagship desert projects. In Abu Dhabi, for example, Masdar’s Shams plant uses 768 tracking parabolic trough collectors and more than 258,000 mirrors; the project also incorporates dry cooling, which is particularly valuable in water-constrained desert conditions. This case illustrates how PTCs function in practice as the core solar-collection subsystem of a CSP plant while also fitting the environmental constraints of hot arid regions. For that reason, parabolic trough technology has held a central place in Arab-region CSP deployment, not only as a mature collector concept, but also as a realistic engineering solution for desert-scale power generation [12, 14].

In comparative terms, parabolic trough collectors can therefore be described as the historical backbone of commercial CSP, situated between simpler but less proven line-focus variants such as linear Fresnel and higher-temperature but more capital-intensive point-focus systems such as solar towers. Their position within CSP systems is defined by four combined attributes: line-focus optical design, proven utility-scale deployment, compatibility with thermal storage, and strong suitability for high-DNI desert environments. For many MENA countries, this combination explains why the parabolic trough has long been treated as the most practical entry point into CSP development [4, 13, 14].

1.3 Operating Principle of Parabolic Trough Collectors

Parabolic trough collectors (PTCs) are line-focus solar thermal devices that operate by concentrating direct solar radiation onto a linear receiver. The parabolic reflector geometry ensures that incoming parallel solar rays are redirected toward the focal line, where the receiver is located. This allows efficient conversion of solar radiation into thermal energy. In CSP classification, parabolic trough systems belong to the line-focus family and typically operate at temperatures ranging from 300 to 550°C depending on the heat-transfer fluid and system design [4].

At the system level, parabolic trough collectors are arranged in large arrays forming the solar field. Each collector is equipped with a single-axis tracking system, which follows the sun’s apparent motion from east to west. This tracking mechanism is essential because trough collectors rely primarily on direct normal irradiance (DNI) rather than diffuse radiation. By maintaining alignment with the sun, the system ensures continuous concentration of solar energy on the receiver throughout the day [15].

The receiver (heat collection element, HCE) is the key component where energy conversion occurs. It consists of a metal absorber tube coated with a selective surface designed for high solar absorptivity and low thermal emissivity. This tube is enclosed within an evacuated glass envelope, which minimizes convective heat losses and protects the coating from environmental degradation. The glass is typically treated with an anti-reflective coating to maximize solar transmission [15,16].

Inside the receiver, a heat-transfer fluid (HTF) flows continuously. As solar radiation is absorbed by the tube, the thermal energy is transferred to the fluid, increasing its temperature. The heated fluid is then transported to the power block, where it exchanges heat to produce steam that drives a turbine connected to an electrical generator. After heat exchange, the cooled fluid is recirculated back to the solar field, forming a closed-loop system [4].

The efficiency of the system also depends on maintaining the vacuum integrity of the receiver and the quality of the selective coating. Degradation of the vacuum or coating can increase thermal losses and reduce system performance. To address this, modern receivers incorporate glass-to-metal seals, expansion bellows, and getter materials that absorb residual gases and maintain vacuum conditions over long operating periods [16].

1.4 Main Components of a Parabolic Trough Collector System

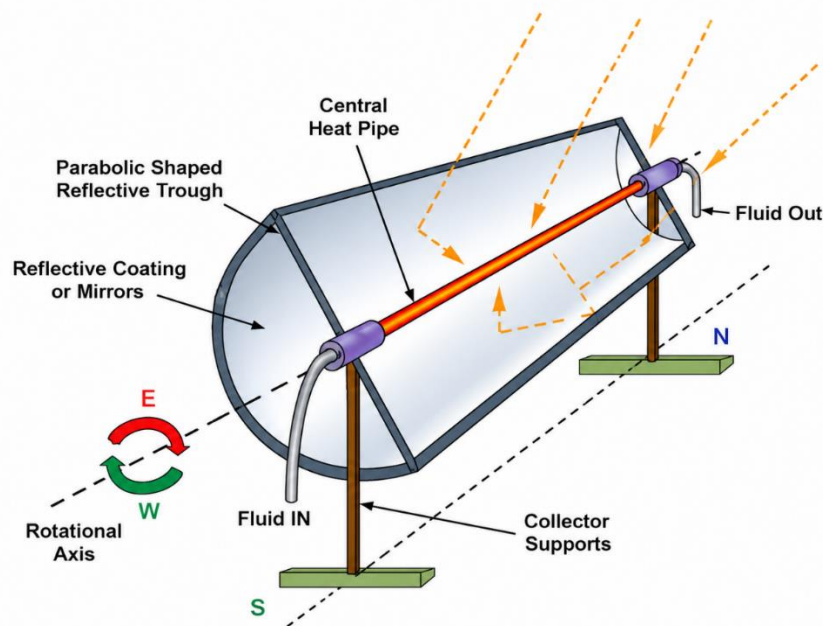


Figure I 2. Main elements of a parabolic trough collector. Source: attached image provided in the prompt.

1.4.1 Reflector

The reflector is the principal optical component of a parabolic trough collector system. Its function is to intercept incoming direct normal irradiance (DNI) and redirect it toward the linear receiver located along the focal line of the parabola. In practical terms, the reflector determines whether the collector can deliver a high solar flux to the absorber tube; for this reason, it is one of the most performance-critical components in the entire system [15,17].

Geometrically, the reflector must maintain a precise parabolic contour. Even small deviations from the intended profile can alter the reflection path and prevent part of the solar radiation from reaching the receiver. This is why optical quality in parabolic trough technology is commonly evaluated through parameters such as slope error, contour deviation, and alignment accuracy. NREL's optical characterization work shows that reflector quality is relevant not only during design and manufacturing, but also during installation and long-term operation, because optical errors directly affect system performance [18].

In commercial trough plants, the reflector is most commonly made from silvered low-iron glass mirrors. NREL's mirror architecture documentation describes a typical parabolic trough mirror as a multilayer structure composed of low-iron glass, a wet-silver reflective layer, a copper back layer, and protective paint layers designed for outdoor service. This architecture is used because low-iron glass improves solar transmittance, silver provides high reflectivity, and the protective back layers improve durability under field exposure [19].



Figure I 3: Parabolic Trough Reflector. (NREL / SkyFuel)

The reflector must satisfy three engineering requirements simultaneously: high optical reflectance, accurate shape retention, and long-term environmental durability. High reflectance is needed to maximize the fraction of solar radiation redirected toward the receiver; accurate shape retention is essential to preserve focus; and durability is necessary because trough mirrors operate under dust, ultraviolet exposure, thermal cycling, humidity, and wind loading for many years. For that reason,

reflector materials are selected not only for optical performance but also for resistance to corrosion, coating degradation, and mechanical deformation [18,19].

1.4.2 Receiver

The receiver is the thermal core of a parabolic trough collector system. Its main function is to absorb the solar radiation reflected by the parabolic mirror and transfer that energy to the heat-transfer fluid (HTF) flowing inside it. In the technical literature, the receiver is often called the heat collection element (HCE), and its performance largely determines the thermal efficiency of the entire collector because it is the component where optical input is converted into useful heat [20,21].

Structurally, a typical parabolic trough receiver consists of a metal absorber tube, usually made of stainless steel, surrounded by a glass envelope with an evacuated annular space between the two layers. The absorber tube is coated with a solar-selective surface that is designed to maximize solar absorptance while minimizing thermal emittance. The surrounding glass envelope acts as a thermal insulator, while the vacuum between the tube and the glass suppresses convective heat losses and protects the selective coating from rapid environmental degradation [15,21].

The operating principle of the receiver is based on a heat-transfer sequence. First, direct normal solar radiation is concentrated by the trough reflector onto the outer surface of the absorber tube. Second, the absorber converts the incident radiation into thermal energy. Third, this heat is conducted through the tube wall and transferred by convection to the HTF circulating inside. The heated fluid then leaves the solar field and delivers thermal energy to the power block, where it is used to generate steam for electricity production or to supply process heat. In this way, the receiver serves as the direct interface between the optical subsystem and the thermodynamic subsystem of the plant [15,20].

A defining feature of the receiver is the use of a solar-selective coating on the absorber tube. This coating is engineered to absorb a very high fraction of incoming solar radiation while emitting as little thermal radiation as possible. Such a property is crucial because the absorber operates at elevated temperatures, and radiative losses would otherwise become significant. NREL's receiver and coating studies emphasize that selective surfaces are one of the key technologies enabling parabolic trough collectors to maintain acceptable efficiency under high-temperature operation [19,22].

The glass envelope also performs several critical functions. It increases solar transmission to the absorber, especially when treated with anti-reflective coatings, and it creates a protective enclosure around the absorber tube. The vacuum annulus between the steel absorber and the glass envelope is particularly important because it sharply reduces convective heat losses. In modern receiver designs, additional components such as glass-to-metal seals, bellows, and getter materials are used to maintain vacuum quality and absorb gases such as hydrogen that may accumulate over time and degrade receiver performance [21,23].

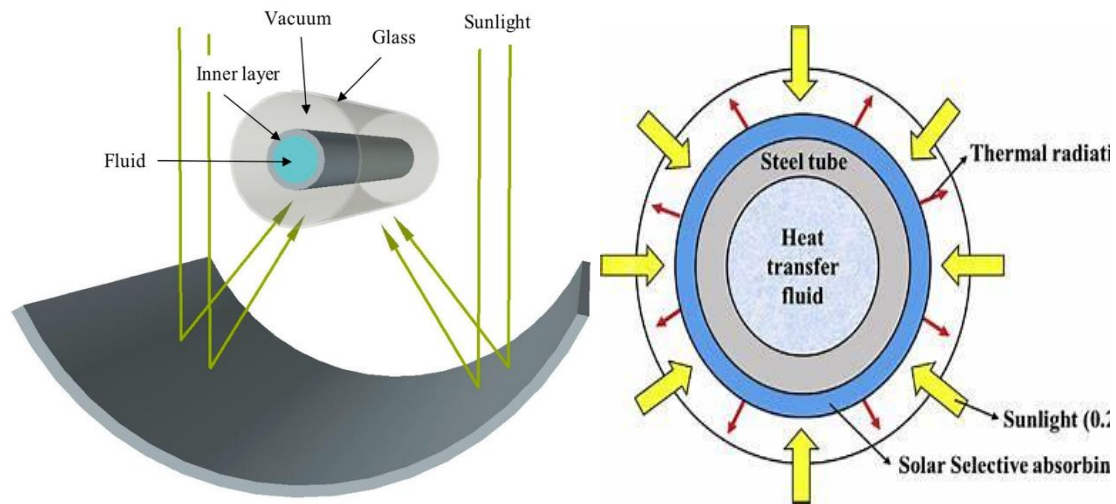


Figure I 4: Receiver Tube (Absorber). (Research on receiver tubes and selective coatings)

1.4.3 Heat Transfer Fluid Loop

The heat transfer fluid (HTF) loop is the subsystem that circulates the working fluid through the solar field and transfers the collected thermal energy to the rest of the plant. In a parabolic trough collector system, solar radiation is first concentrated onto the receiver tube, and the HTF flowing inside that tube absorbs the thermal energy. The hot fluid then exits the solar field and passes to the power block, where its heat is used in heat exchangers to generate high-pressure steam for the turbine-generator system [4,15].

From a hydraulic point of view, the HTF loop is a closed circulation circuit. Pumps drive the fluid through rows of heat collection elements in the solar field, after which the heated fluid is routed either directly to the steam-generation train or, in plants equipped with storage, to the thermal energy storage system before returning to the field. NREL's reference plant description explains that the trough field is arranged in loops composed of multiple solar collector assemblies, and that the HTF is circulated through these interconnected rows under controlled temperature and flow conditions [21].

In conventional parabolic trough plants, the most common HTF has historically been synthetic oil. This fluid is preferred because it has relatively low vapor pressure, an acceptable freezing point for plant operation, and sufficient thermal stability for commercial trough conditions. IRENA notes that most existing parabolic trough systems use synthetic oils that are generally stable up to around 400°C, while newer concepts have explored molten salt as an alternative heat transfer medium at higher temperatures [4,9].

The operational role of the HTF loop is therefore not limited to heat transport; it also sets the thermal regime of the whole solar field. The mass flow rate, inlet temperature, outlet temperature, and pressure of the loop directly influence collector efficiency, receiver heat losses, and power-block

performance. NREL modeling work on parabolic trough receivers shows that the HTF temperature strongly affects the heat-transfer process inside the absorber tube and the magnitude of thermal losses to the surroundings, which means that loop control is essential for stable and efficient plant operation [20].

A key engineering feature of the HTF loop is its integration with the heat-exchanger train. After leaving the solar field, the hot HTF typically transfers energy through a sequence of exchangers such as preheaters, evaporators, and superheaters. The fluid itself usually remains in a separate closed circuit and does not enter the steam turbine directly. The U.S. Department of Energy explains that the HTF is used to generate superheated steam in the power block, after which the cooled HTF is recirculated back through the solar field for reheating [15].

In systems that include thermal energy storage (TES), the HTF loop also acts as the thermal bridge between solar collection and storage charging or discharging. This makes the loop central not only to instantaneous energy conversion, but also to dispatchability. NREL's survey of trough storage concepts notes that both oils and salts can be used in storage-related configurations, although molten salts generally require additional heat tracing or parasitic heating because of their higher freezing points, which introduces extra operational constraints for the loop design [24].

Another important issue in the HTF loop is fluid degradation and chemical stability. In synthetic-oil-based trough plants, high operating temperatures can gradually degrade the HTF and produce by-products such as hydrogen. NREL research has shown that hydrogen generated in the HTF can diffuse through the absorber tube and accumulate in the annulus of the receiver, increasing thermal losses by degrading the vacuum insulation. This means that HTF-loop chemistry has direct consequences for both loop reliability and receiver performance [25].

The design of the HTF loop must also account for freeze protection, pressure containment, pumping power, and thermal expansion. Because the fluid circulates over long distances in a large solar field, the loop requires piping, expansion tanks, pumps, valves, and control systems capable of maintaining safe and continuous operation under variable solar input. NREL's reference plant report emphasizes that the HTF system is one of the principal subsystems of a trough plant and includes the piping network, solar heat collection equipment, and associated circulation components needed to move thermal energy from the field to the power block [21].

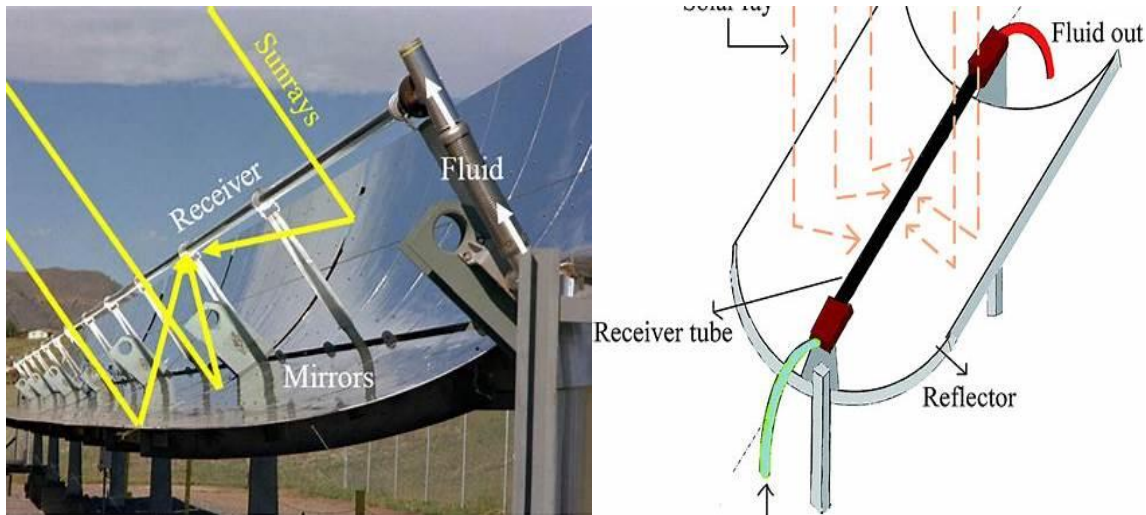


Figure I 5: Heat Transfer Fluid (HTF) Loop. The HTF circulates through the absorber tube and transfers thermal energy to the power block.

1.4.4 Tracking System

The tracking system is the mechanical and control subsystem that continuously orients the parabolic trough so that direct solar radiation remains focused on the linear receiver throughout the day. In utility-scale trough plants, collectors are generally mounted on a single-axis tracking mechanism, most often aligned on a north–south horizontal axis, and they rotate from east to west following the apparent motion of the sun. This tracking function is essential because parabolic trough collectors use direct normal irradiance (DNI), and their thermal output depends on maintaining the correct optical alignment between the reflector aperture and the solar beam [15,26].

From an operational perspective, the tracking system directly affects optical efficiency and therefore plant performance. NREL’s performance guidelines explain that in a linear-axis tracking trough collector, the useful solar input is the beam-radiation component normal to the collector aperture plane, and any deviation from the ideal tracking angle reduces the usable solar resource intercepted by the receiver. More recent NREL-based research also shows that misalignment and tracking error can cause measurable efficiency losses; one cited case reported a 0.6% efficiency loss in a commercial plant due to tracking error and structural twist, while a numerical study found that a 16 mrad tracking error could reduce optical efficiency by 60% under certain conditions [17,26].

Technically, the tracking system combines drive mechanisms, support structure, sensors, and control algorithms. The support structure must transmit tracking torque while preserving the mirror’s parabolic shape, and the control system must keep the collector at the correct angular position under changing solar and environmental conditions. According to recent NREL research, misalignment may result not only from control errors but also from wind loading, gravity, thermal effects, installation inaccuracies, hardware failures, and fatigue, all of which can deform the collector structure or disturb

angular precision. For this reason, the tracking system in a parabolic trough plant is not merely a positioning device; it is a critical performance subsystem that links solar geometry, structural stiffness, and optical reliability [15,17].

1.4.5 Thermal Insulation

In a parabolic trough collector system, thermal insulation is essential because the useful solar heat collected at the receiver must be preserved while the fluid travels through the solar field and associated piping. The most critical insulation is found at the receiver level, where the absorber tube is enclosed inside a glass envelope separated by an evacuated annulus. This arrangement is specifically designed to reduce thermal losses from the hot absorber to the surroundings, while the glass envelope also protects the selective coating and maintains optical transmission. NREL's receiver analysis explains that the glass envelope is anti-reflection treated, the annulus is kept under vacuum to reduce thermal losses, and getter materials are used to absorb hydrogen that would otherwise degrade performance [20].

Thermal insulation is also important beyond the receiver itself, especially in the HTF piping network, where heat must be retained as the fluid moves between collector loops, heat exchangers, and storage units. In large CSP plants, minimizing thermal losses in transport lines is necessary to preserve plant efficiency and reduce parasitic losses. NREL's best-practices study emphasizes that parabolic trough receivers differ from central-receiver heat-exchanger tubes precisely because they use a glass envelope with an evacuated space to reduce thermal losses, underlining that insulation is a defining design feature of trough technology itself [21,27].

1.4.6 Heat Exchanger and Thermal Storage

In a parabolic trough power plant, the heat exchanger is the component that transfers thermal energy from the heat transfer fluid (HTF) circulating in the solar field to the water–steam cycle of the power block. After the synthetic oil or other HTF leaves the solar receiver at elevated temperature, it passes through a steam-generation train composed of heat exchangers such as the preheater, evaporator, and superheater, where its thermal energy is delivered to feedwater without mixing the two fluids directly. This indirect arrangement is a defining feature of conventional trough systems because it allows the solar field and the steam cycle to operate as thermally coupled but hydraulically separate subsystems [15,21].

Thermal energy storage (TES) is integrated into parabolic trough plants to store excess solar heat during periods of strong irradiation and release it later when solar input declines, thereby extending electricity generation beyond daylight hours. According to IRENA, CSP plants can be equipped with heat storage systems that allow electricity generation after sunset or under cloudy conditions, significantly improving capacity factor, dispatchability, and grid integration. In the case of parabolic trough technology, storage may be configured as an indirect two-tank system, where the solar-field HTF

transfers energy through a heat exchanger to a separate storage medium, or as other active storage arrangements depending on plant design [4,24].

Table I 1: Main Components of a Parabolic Trough Collector (PTC) and Their Functions

Component	Function
Parabolic reflective mirror	Reflects and concentrates direct solar radiation onto the receiver tube.
Receiver / absorber tube	Absorbs the concentrated solar radiation and transfers heat to the working fluid.
Heat transfer fluid (HTF)	Carries the absorbed thermal energy toward a heat exchanger, storage system, industrial process, or power cycle.
Glass envelope and vacuum annulus	Reduce convective and radiative heat losses from the absorber tube in advanced receiver designs.
Single-axis tracking system	Rotates the collector so that the reflected rays remain focused on the receiver during the day.
Support structure	Maintains optical alignment and mechanical stability under operational and wind loads.

1.5 Classification of Parabolic Trough Collectors Relevant to This Study

1.5.1 Receiver Types: Bare, Glass-Covered, and Evacuated Receivers

In parabolic trough technology, receiver configurations can be classified into bare receivers, glass-covered non-evacuated receivers, and evacuated receivers, according to the degree of thermal protection provided around the absorber tube. The bare receiver is the simplest configuration: the absorber tube is directly exposed to ambient air, so it experiences the highest convective and radiative heat losses. The glass-covered receiver adds a transparent envelope around the absorber, which reduces direct interaction with ambient air and provides partial thermal protection, but it does not suppress convection as effectively as a vacuum-based design. The evacuated receiver, by contrast, places the absorber tube inside a glass envelope with an evacuated annular space, which is the standard high-performance configuration in commercial parabolic trough systems because it minimizes convective heat loss while preserving optical transmission [20,28].

The thermal distinction among these three receiver types is mainly explained by their different heat-loss mechanisms. For an evacuated receiver, convection between the absorber tube and the inner glass surface is very small as long as the vacuum remains intact, so heat loss is governed mainly by radiation across the annulus and by external losses from the glass envelope to the surroundings. In a glass-covered but non-evacuated receiver, the annular space

still contains air, so natural or forced convection can develop between the absorber and the cover, increasing thermal losses compared with evacuated designs.

This classification is directly relevant to performance-oriented studies because receiver type strongly affects collector efficiency, operating temperature range, and practical application. Commercial parabolic trough power plants generally use evacuated receivers because they offer the lowest thermal losses and the best suitability for high-temperature electricity generation. Glass-covered non-evacuated receivers are often investigated for moderate-temperature solar process heat, where reduced cost may justify somewhat higher heat losses. Bare receivers remain the least thermally efficient option, but they are still useful as a baseline for comparison in experimental work and in simplified low-cost collector designs.

1.5.2 Heat Transfer Fluids: Thermal Oil, Water–Steam, and Molten Salt

In parabolic trough systems, the main heat-transfer-fluid options relevant to this study are thermal oil, water–steam, and molten salt, each of which defines a different operating philosophy and temperature range. In conventional commercial trough plants, thermal oil has historically been the dominant option because it circulates in a closed loop at relatively low vapor pressure and can transfer solar heat from the receiver to the steam generator with proven operational reliability. IRENA notes that most existing parabolic trough plants use synthetic oils as the HTF and that these fluids are generally stable up to about 400°C, which explains their long-standing use in utility-scale trough technology [9,21].

A second option is water–steam, usually discussed as direct steam generation (DSG), in which water is heated directly in the solar field and converted into steam without an intermediate oil loop. This configuration eliminates the need for a primary HTF-to-steam heat exchanger and can simplify the thermodynamic chain, but it also makes the balance of plant more demanding because steam quality, pressure control, startup behavior, and solar-field flow stability must all be managed directly. Recent IEA SHC guidance on direct steam generation shows that the steam drum becomes the core component of the DSG balance of plant, while the solar pump, discharge system, and feedwater control must be designed around saturation conditions and variable solar input, making water–steam technically attractive but operationally more complex than oil-based loops [29].

The third major option is molten salt, which is increasingly important because it allows higher operating temperatures and offers strong advantages for thermal energy storage integration. IRENA reports that new trough plants under demonstration have used molten salt at around 540°C either as heat-transfer fluid or as storage medium, and the agency further notes that molten salt can considerably improve storage performance. However, molten salt also introduces important design constraints because it solidifies at relatively high temperature and therefore requires freeze protection, heat tracing,

and careful thermal management across the loop. For this reason, molten salt is often viewed as a high-performance but more demanding alternative to thermal oil, whereas water–steam remains attractive where direct generation is desired and system operators can manage the added control complexity [4,9].

1.5.3 Tracking Modes and Orientation: Single-Axis, North–South, and East–West

Parabolic trough collectors are fundamentally single-axis tracking systems. The collector rotates about its longitudinal axis so that the angle between the aperture normal and the incoming beam radiation is minimized during the day. NREL’s physical trough model states that this axis may be oriented in any compass direction, but in practice it is usually arranged either in the North–South or East–West direction. This tracking concept is essential because parabolic troughs depend on direct normal irradiance (DNI) rather than diffuse radiation, so accurate solar alignment directly governs optical gain and cosine-loss behavior [28,30].

The choice between North–South and East–West orientation affects the seasonal and annual energy profile of the collector field. According to the U.S. Department of Energy, most parabolic trough systems are oriented either east–west or north–south with single-axis tracking; east–west orientation performs better in winter and tends to provide a more even output over the year, whereas north–south orientation performs better in summer and generally provides higher annual energy output [30].² NREL cost-performance analysis adds an important system-level nuance: although the conventional choice is north–south, an east–west axis can produce a more uniform seasonal generation profile and may become the lower-LCOE option at very high capacity factors, while near-term market conditions with strong summer peak demand still tend to favor north–south orientation [31].

1.5.4 Main Geometrical Parameters: Aperture Width, Rim Angle, Focal Length, and Absorber Diameter

The main geometrical parameters of a parabolic trough collector are aperture width, rim angle, focal length, and absorber diameter, because these variables define both the optical shape of the concentrator and the size of the receiver needed to intercept the reflected solar image. A recent thermodynamic analysis of parabolic trough receivers states explicitly that the most important geometrical parameters in collector design are the rim angle (ϕ_r) focal length (f) and aperture width (a), while the receiver cross-section includes the absorber tube diameter as a defining dimension. The same source presents the parabola in the standard form $y^2 = 4fx$, showing that focal length is the fundamental geometric quantity governing curvature and focus position [32].

These parameters are mathematically linked rather than independent. The same study reports that focal length is related to aperture width and rim angle by the expression $f = a/[4 \tan(\phi_r/2)]$, which means that once any two of these variables are fixed, the third is determined. A second engineering thesis on parabolic trough design gives the complementary form $wa/4f = \tan(\theta_r/2)$, again confirming

that aperture width, focal length, and rim angle are coupled design variables. That thesis also notes that the minimum absorber diameter required to intercept the solar image can be estimated geometrically as $D_{r2} = rr \sin \theta_m$, where rr is the mirror radius at the rim and θ_m is the sun's half-angle. In other words, the absorber diameter is not chosen arbitrarily; it must be consistent with the reflector geometry and solar image size [32,33].

1.6 Arid Conditions of El Oued: Constraints and Expected Effects on PTC Performance

1.6.1 High Direct Normal Irradiance and Temperature Extremes

El Oued offers a highly favorable solar resource for parabolic trough collectors (PTCs) because it combines strong direct normal irradiation with long sunshine duration under a hot, dry Saharan climate. In one Algeria site-comparison study for solar thermal deployment, El Oued was reported with an annual DNI of 2149.5 kWh/m²·year and a mean DNI of 671.7 W/m², values that place it above the commonly cited threshold of 2000 kWh/m²·year used to justify CSP-type projects at selected Algerian sites [34]. This is directly relevant to PTCs because parabolic troughs operate only with beam radiation, tracking the sun so that direct radiation remains concentrated on the receiver tube; accordingly, high DNI in El Oued should increase the useful thermal input available to the collector field and improve the site's suitability for solar concentration technologies [35].

At the same time, El Oued is characterized by marked temperature extremes, which define the desert operating environment in which the collector field must function. Long-term climate data from the El Oued–Guemar airport meteorological station for 1978–2020 show that July is the hottest month, with a recorded monthly maximum temperature of 45.44°C, while the annual average temperature is 26.75°C; the same study reports very low mean annual precipitation of only 67.63 mm, confirming the highly arid character of the region [36].⁴ In practical terms, this means that PTC operation in El Oued must be assessed under sustained hot-weather exposure rather than under mild ambient conditions, and this local climate profile is consistent with the “hot and dry” site-selection rationale used in Algerian solar-thermal feasibility work [34].

For PTC performance, the expected effect of this combination is generally positive on the solar-input side but demanding on the thermal-operational side. High DNI should raise absorbed solar energy and support higher useful heat gain, while ambient temperature remains an explicit input in validated PTC performance models together with wind speed, HTF conditions, and solar insolation because these variables influence collector efficiency, heat gain, and thermal losses [20,37]. NREL receiver modeling further shows that wind and ambient conditions are used to estimate receiver losses, and experimental receiver testing indicates that, for evacuated receivers, heat loss is governed primarily by absorber temperature and is only weakly sensitive to moderate ambient-condition changes, whereas damaged or

non-evacuated receivers are much more vulnerable [20,38]. Therefore, in El Oued, the high-DNI environment is expected to be beneficial for PTC productivity, but the severe desert temperatures still matter because they define the boundary conditions for receiver losses, thermal control, and reliable long-term field operation. [20,38].

1.6.2 Wind Effects on External Convective Heat Losses

In El Oued, wind is an important desert boundary condition because it directly affects the external convective heat losses of parabolic trough receivers. A regional Algerian wind assessment reports that El Oued has an average wind speed above 4.5 m/s, which is high enough to be considered energetically significant at site scale. For parabolic trough collectors, this matters because the receiver ultimately rejects part of its heat to the environment through the outer glass envelope, and the NREL receiver model identifies convection from the glass envelope to the atmosphere as the largest source of heat loss, especially if there is wind [20,39].

The physical reason is straightforward: increasing wind speed raises the external convective heat-transfer coefficient around the glass envelope, which accelerates the removal of heat from the receiver surface to ambient air. In the SAM physical trough model, external heat loss from the envelope to ambient is explicitly represented through a convection term from the outer glass surface to ambient air, alongside radiation to the sky; the model also notes that convection within the annulus is very small for intact evacuated receivers but becomes more important if the vacuum is lost or the envelope is broken. This means that under normal operating conditions wind mainly intensifies the outside convective loss pathway, whereas damaged or non-evacuated receivers become much more vulnerable because additional convection can also develop closer to the absorber tube [28].

Experimental and modeling evidence further shows that the wind effect is real but depends on receiver condition. NREL heat-loss testing for the Schott PTR70 receiver found that, for an intact evacuated receiver, changing wind speed from 0 to 2 m/s altered the outer glass temperature noticeably, while the total heat loss changed only modestly, by about ± 2 W/m at the 250°C case and ± 5 W/m at the 450°C case; the report therefore concludes that evacuated receivers are relatively insensitive to moderate ambient-condition changes. In El Oued, the expected implication is that wind will still increase external convective losses, but the penalty should remain limited as long as the receiver vacuum and glass envelope are intact; by contrast, any degradation of the receiver would make wind a much stronger performance constraint in desert operation [20,28].

1.6.3 Dust and Soiling Effects on Performance and Maintenance

In an arid Saharan environment such as El Oued, dust and soiling are expected to be major operational constraints for parabolic trough collectors because CSP systems rely on highly reflective mirrors and optically clean receiver surfaces. A widely cited techno-economic assessment defines

soiling as the deposition of contaminants on the mirrors and tubes of concentrated solar power systems and states that it can cause a drastic reduction of power generation, especially in high-insolation arid and semi-arid climates. For parabolic troughs, the main immediate effect is a reduction in mirror cleanliness and solar-weighted reflectance, which lowers the amount of direct radiation redirected toward the receiver and therefore reduces useful thermal output [40,41].

The performance penalty is not limited to mirrors alone. DLR research on parabolic trough receiver tubes shows that soiling of the glass envelope can also significantly reduce transmittance and therefore lower overall collector efficiency. The same study notes that although mirror soiling generally has the larger optical impact, the contribution of receiver-tube soiling cannot be neglected when a complete assessment of trough performance is required; measured tube-soiling rates of about 0.1–0.3% per day were reported in the investigated cases. This means that, under dusty desert conditions, both the reflector field and the receiver envelope can progressively lose optical performance between cleaning events, making soiling a dual-surface problem rather than a mirror-only issue [42].

These effects have direct implications for maintenance strategy. SolarPACES guidelines state that soiling in operational CSP fields is spatially non-uniform and site-dependent, and that specular reflectance below 60% is not uncommon in the field, which is why cleanliness monitoring is essential for reliable performance assessment and cleaning decisions [41]. DLR operational-analysis work further notes that CSP plants are typically located in arid regions with high dust loads, that mirrors must be cleaned frequently to maintain high reflectivity, and that improving cleanliness prediction can make cleaning schedules easier to optimize and reduce unnecessary washing. For El Oued, this implies that dust and sand deposition are likely to increase both performance losses and operation-and-maintenance requirements, especially through more frequent mirror washing, cleanliness monitoring, and site-specific maintenance planning under scarce-water desert conditions [40,43].

1.7 Literature Review on PTC Performance and CFD Studies

1.7.1 Experimental Studies: Main Findings and Limitations

Experimental studies on parabolic trough collectors (PTCs) consistently show that real performance depends on the combined behavior of the optical system, receiver, tracking, and heat-transfer loop, rather than on geometry alone. At the receiver level, NREL heat-loss testing showed that the newer UVAC3 receiver had a heat loss of 310 W/m at 400°C, compared with 380 W/m for the earlier UVAC2, corresponding to a 20% reduction in heat loss; the same report stresses that lower receiver heat loss directly increases the heat available to the heat-transfer fluid and improves collector/receiver efficiency [44]. At the collector-system level, experimental testing of a medium-scale stainless-steel PTC for hot-water production found an efficiency-curve intercept of 0.72987 and slope of 0.2358, a time constant of 75 s, and water production exceeding 226 L/day at temperatures $\geq 50^\circ\text{C}$; these results

indicate that properly built trough collectors can achieve stable and practically useful thermal output under field conditions [45].

A second major finding from experimental work is that PTC performance is highly sensitive to operating conditions and component quality. NREL's receiver-performance research makes clear that experimental evaluation must include both optical efficiency and heat-loss behavior, because heat losses alone do not fully describe useful heat gain in service [44]. Likewise, Forristall's validated receiver study, which compared model results with experimental receiver data, identifies HTF inlet temperature and flow rate, solar insolation, wind speed, and ambient temperature as essential inputs governing collector efficiency, outlet temperature, heat gain, and thermal losses [20].

The main limitations of the experimental literature are equally clear. First, many studies are conducted on small-scale, site-specific, or prototype systems, which limits direct extrapolation to utility-scale CSP fields. Second, test results are often reported under short-duration or quasi-steady conditions, even though real plants operate under fluctuating solar input and nonuniform field conditions. Third, NREL's acceptance-test guideline explicitly notes that no dedicated ASME performance test code yet exists for utility-scale parabolic trough solar fields, so detailed test procedures still need to be defined project by project; this reduces comparability across studies and partly explains why the literature mixes different boundaries, measurement methods, and reporting conventions [26].

1.7.2 Numerical and CFD Studies: Conjugate Heat Transfer, Radiation, and Turbulence Models

Numerical and CFD studies on parabolic trough collectors (PTCs) have progressively moved from simplified one-dimensional thermal balances toward conjugate heat transfer (CHT) formulations that simultaneously resolve heat conduction in the solid walls, radiation exchange at the receiver surfaces, and forced convection inside the heat-transfer-fluid passage. A foundational NREL receiver model already framed the problem as a coupled thermal system by treating the absorber tube, glass envelope, annulus, and external environment within one integrated heat-transfer formulation, with outputs including collector efficiency, outlet temperature, heat gain, and optical and thermal losses [20]. More recent literature reviews confirm that numerical work now dominates this part of the field: a 2024 review reported that 62.23% of parabolic-trough heat-loss studies were "general studies" devoted to calculating or predicting heat loss, which reflects the central role of numerical modeling in present PTC research [46].

1.7.3 Effects of Wind Speed, Ambient Temperature, Solar Irradiation, and Mass Flow Rate

The literature consistently shows that solar irradiation is the primary driver of parabolic trough collector (PTC) performance because it determines the amount of beam energy available for

concentration on the receiver. NREL's performance-testing guidance for utility-scale trough systems states that measured thermal output varies mainly with the aperture-normal irradiance (ANI) input to the solar field and with the time of day and season, while the accepted efficiency definition is based on delivered thermal power divided by ANI times aperture area. In practical terms, higher direct solar irradiation increases the absorbed solar input, raises useful heat gain, and generally improves thermal efficiency, provided that the rest of the system operates within its design limits [26].

By contrast, wind speed and ambient temperature mainly affect PTC performance through the thermal-loss side of the receiver. Forristall's validated NREL receiver model identifies solar insolation, wind speed, ambient temperature, HTF inlet temperature, and HTF flow rate as key input variables for predicting collector efficiency, outlet temperature, and heat gain, and it shows that convection from the glass envelope to ambient air is a major loss mechanism, especially under windy conditions. The SAM physical trough model likewise treats ambient temperature as a control and performance variable in solar-field operation, while receiver heat-loss calculations explicitly depend on surrounding thermal conditions. Thus, stronger wind generally increases external convective losses, whereas ambient temperature changes the thermal gradient between the receiver and the environment and therefore influences overall heat rejection from the collector [20,28].

The effect of mass flow rate is more operational than climatic, but it is equally important because it governs the thermal regime inside the receiver tube and the ability of the solar field to maintain the target outlet temperature. The SAM trough model states that HTF mass flow rate is continuously adjusted so that the loop outlet temperature reaches the design value whenever possible, and it provides explicit minimum and maximum allowable flow ranges tied to receiver diameter and fluid properties. Because HTF temperature is itself a function of mass flow rate, changing the flow rate alters internal heat transfer, outlet temperature, and the balance between absorbed energy and thermal loss. For this reason, the literature treats mass flow rate as a key control variable: increasing it generally improves heat removal and stabilizes outlet conditions, but it also changes hydraulic demand and may require defocusing or flow-limit management at very high solar input [28].

1.7.4 Optimization Approaches: Tilt Angle, Alignment, and Geometrical Parameters

Optimization studies on parabolic trough collectors (PTCs) generally treat tilt angle and orientation as optical-yield variables rather than simple mounting choices. In tracked troughs, the collector operates through single-axis rotation, and the tracking axis is usually arranged either north–south or east–west. NREL's SAM manual states that the axis may be oriented in any compass direction, though it typically aligns in either the North–South or East–West direction, while the U.S. Department of Energy reports that east–west orientation gives more uniform yearly output and better winter performance, whereas north–south orientation usually gives better summer performance and higher annual output. At system level, NREL's cost study adds that east–west orientation can become attractive

when a flatter seasonal generation profile is desired, while north–south remains the conventional choice for maximizing annual production in many markets [28,30,31].

A second optimization line concerns alignment accuracy, because even a well-designed trough loses performance when the reflector, absorber tube, or tracking angle deviates from the ideal optical position. NREL developed dedicated tools to measure absorber alignment and combined slope/alignment errors, explicitly noting that maintaining high optical performance requires controlling these geometric imperfections. More recent NREL work on operating CSP plants shows that tracking error reduces optical performance, that wind loading can induce trough-angle deviation, and that at peak tracking error the optical performance of observed rows can drop dramatically. This means that alignment optimization is not only a manufacturing issue, but also an operational one involving reflector slope quality, absorber positioning, structural stiffness, and control precision under real wind conditions [47,48].

A third optimization approach focuses on geometrical parameters, especially aperture width, focal length, rim angle, and absorber diameter, because these control concentration ratio, interception margin, shading, and cost. A geometry-based analysis reports that increasing focal length reduces collector width and area, while reducing focal length enlarges the allowable deviation margin and improves tolerance to tracking inaccuracy; however, this comes at the expense of larger aperture area and therefore higher cost. The same study also shows that increasing absorber diameter improves interception robustness but increases shadowing on the reflector surface. Complementing this, DLR-based literature defines the collector geometry through parameters such as aperture width, focal length, and rim angle, confirming that geometrical optimization is essentially a trade-off between optical concentration, error tolerance, and economic feasibility rather than a one-variable maximization problem [49,50].

1.8 Research Gap

Although the parabolic trough collector (PTC) literature is already extensive, the evidence base is still uneven. A recent review of heat-loss studies shows that 62.23% of parabolic-trough studies are still concentrated on general heat-loss calculation, whereas only 14.89% focus on geometry and coating materials and only 7.98% examine different heat-transfer fluids as a loss-mitigation strategy. This distribution suggests that the field has produced many thermal evaluations, but far fewer integrated studies that simultaneously optimize geometry, receiver configuration, working fluid, and real operating conditions in a unified framework [46]. In parallel, local Algerian work carried out specifically for El Oued confirms the solar potential of the site, but the published study identified through the literature is still a numerical investigation for a single selected day, using water as the HTF and reporting that optical and thermal efficiency exceeded 75%. That is useful as a first indication, but it is not sufficient to establish the long-term thermo-hydraulic behavior of PTCs under the full range of Saharan operating conditions expected in El Oued.

A second gap concerns the incomplete treatment of arid-environment degradation factors in mainstream PTC modeling. Existing studies clearly show that soiling is a major yield-loss mechanism in dry climates: in Morocco, CSP mirror soiling averaged $-1.18\%/day$, about 5 to 6 times the PV rate measured under the same semi-arid conditions, and the authors explicitly note that many studies examine PV and CSP soiling separately rather than under identical monitored conditions [51]. At the same time, recent work on parabolic-trough receiver tubes demonstrates that dust on the transparent receiver cover can reduce transmittance from above 93% for a clean condition to below 79% at heavy soiling levels, directly affecting optical input to the absorber and therefore plant efficiency [52]. These findings show that dust and soiling are not marginal maintenance issues but central performance variables in desert operation. Yet there is still little evidence of a site-specific, Saharan, PTC-focused study for El Oued that combines mirror soiling, receiver-cover contamination, and maintenance implications in one performance-oriented framework.

A third gap lies in the connection between advanced CFD modeling and real desert deployment conditions. The literature already includes sophisticated conjugate-heat-transfer, radiation, and turbulence modeling, but these studies are still dominated by receiver-scale investigations and enhancement concepts rather than locally validated plant-relevant scenarios. This is consistent with the broader review result that most publications remain at the level of heat-loss estimation rather than integrated operational optimization [46].

1.9 Objectives and Research Questions

The literature indicates that parabolic trough collector (PTC) research is already mature in some areas, but still fragmented in others. NREL's best-practices review states that parabolic trough technology is mature, yet continued improvement is still needed in optical accuracy, receiver performance, operating strategy, and plant-level integration, especially under site-specific environmental conditions [27]. At the same time, the detailed SAM physical trough model shows that PTC performance depends on the interaction of several coupled variables, including solar-field optics, receiver heat loss, HTF flow rate, piping, storage, ambient conditions, and control strategy [28]. In the specific case of El Oued, the available local numerical study focused on a small PTC under the climatic conditions of a single day, with water as the heat-transfer fluid, which confirms local solar potential but does not provide a complete framework for design-oriented evaluation in Saharan operation [53]. On this basis, the first objective of the present study is to develop a more integrated thermo-optical and thermo-hydraulic assessment of PTC behavior under El Oued conditions, with particular attention to receiver performance, operating parameters, and desert-environment constraints.

Accordingly, the central research questions arise directly from the variables that the literature identifies as most influential. Since NREL's validated receiver model treats solar insolation, wind speed, ambient temperature, HTF inlet temperature, and HTF flow rate as key inputs governing collector

efficiency, outlet temperature, and heat gain [20], the present work asks: How do the climatic conditions of El Oued affect the thermal and hydraulic performance of a parabolic trough collector? Which parameters most strongly influence useful heat gain and thermal efficiency under these desert conditions? How do receiver configuration, geometrical design, and operating flow conditions alter heat loss and overall collector performance? These questions are also justified by the current review literature, which shows that a large share of PTC studies still concentrates on general heat-loss estimation rather than on integrated optimization of geometry, receiver type, and operating conditions in one coherent framework [46].

1.10 Scope, Assumptions, and Limitations of the Study

The scope of the present study is limited to the thermo-optical and thermo-hydraulic behavior of a parabolic trough collector (PTC) under the arid climatic conditions of El Oued, with emphasis on the receiver, heat-transfer-fluid loop, and the influence of the main environmental and operating variables. This scope is consistent with the engineering logic of the SAM Physical Trough Model, which was developed to analyze parabolic-trough performance from system geometry and physical properties, while resolving the major subsystems of the plant such as the solar field, power cycle, thermal storage, and control interface [28]. It is also consistent with the local desert-oriented literature, where the El Oued case has already been investigated as a small PTC under real climatic conditions, using water as the heat-transfer fluid and focusing on thermal behavior rather than on full plant economics or long-term field operation [54]. Accordingly, the present study is concerned with collector-level performance prediction and parameter sensitivity under Saharan conditions, rather than with the complete techno-economic design of a commercial CSP plant.

The main assumptions of the study follow the standard simplifications widely used in validated PTC models. Forristall's receiver model states that one- and two-dimensional energy balances are employed depending on the application, with the one-dimensional formulation being acceptable for short receivers and comparative design studies, whereas the two-dimensional formulation becomes necessary for longer receivers where axial temperature variation becomes important [20]. The same report identifies the principal model inputs as collector and receiver geometry, optical properties, HTF properties, HTF inlet temperature and flow rate, solar insolation, wind speed, and ambient temperature, which means that PTC analysis is normally performed under specified boundary conditions and with prescribed material and operating properties rather than under fully unconstrained real-world variability [20].³ In the same spirit, the SAM manual explicitly notes that the physical trough model is only one possible model formulation, that many other valid formulations exist, and that physical and empirical models answer different questions and must be used within their inherent constraints [28].

The principal limitations therefore arise from model idealization, data boundaries, and scale. The Forristall report explicitly states that the modeling assumptions and limitations are discussed, and its

purpose is to predict receiver performance under defined conditions rather than to reproduce every source of field uncertainty or degradation [20]. Likewise, the SAM manual warns that a complex engineering system such as parabolic trough technology creates “an opportunity for misunderstanding or misuse” and emphasizes that the selected formulation does not necessarily represent the best possible one, which means that simulation results must be interpreted as engineering approximations rather than exact replicas of long-term plant behavior [28].

1.11 Organization of the Thesis

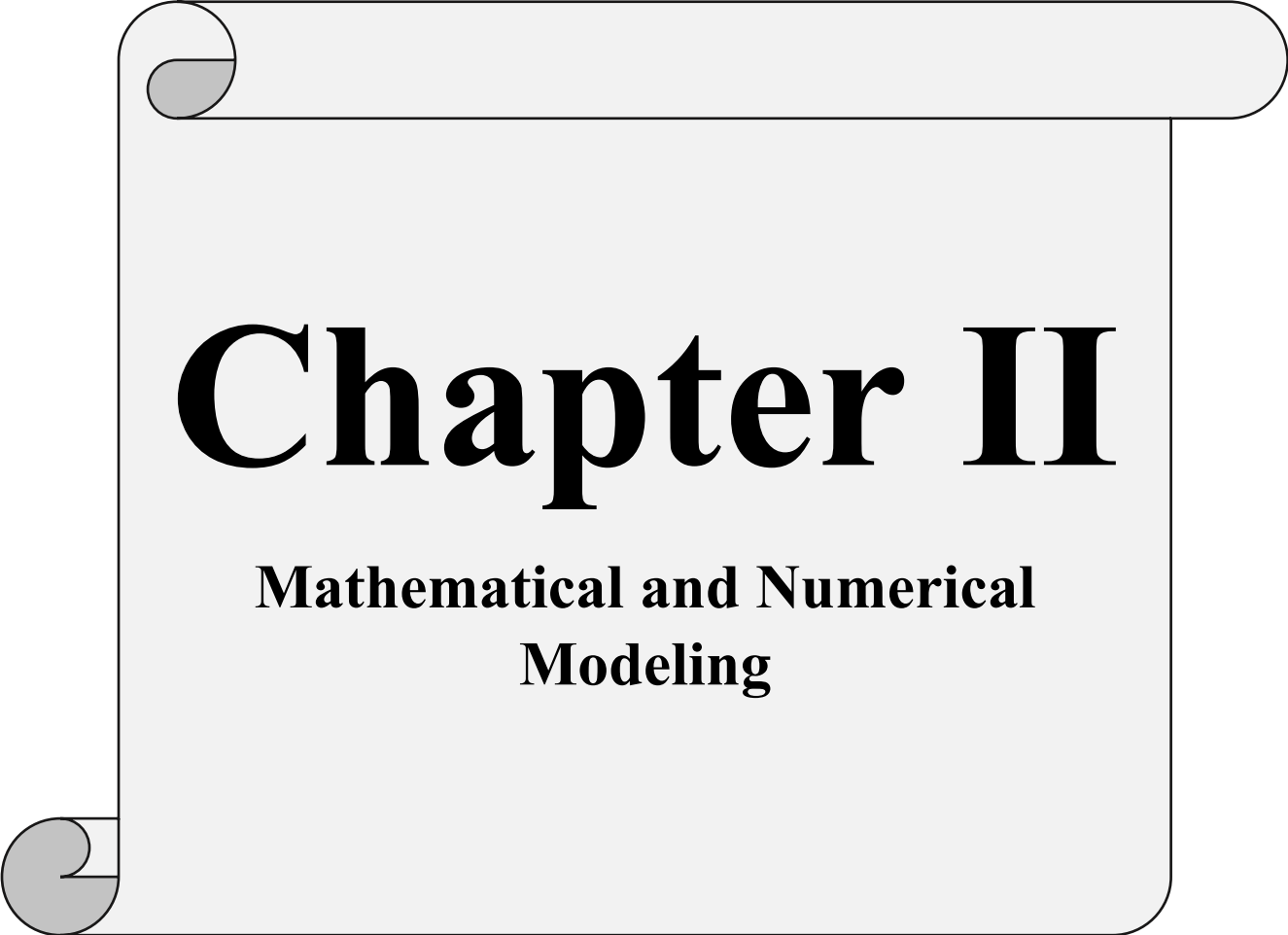
The organization of this thesis follows a conventional scientific structure in which the early chapters establish the problem, the scholarly context, and the methodological framework before moving to analysis and interpretation. University writing guidance commonly presents the thesis as a sequence that begins with an introduction, followed by a literature-based background, then a methods chapter, and subsequently the chapters devoted to results, discussion, and conclusion. In the present work, Chapter 1 introduces the research background, the problem statement, the objectives, the research questions, the scope, and the specific relevance of parabolic trough collectors under the arid conditions of El Oued. Chapter 2 is devoted to the theoretical and technical background of parabolic trough collectors, including CSP classification, operating principles, major components, performance indicators, and the main environmental constraints affecting operation in desert climates. This arrangement is consistent with standard thesis guidance, which requires the introduction to explain the research problem and significance, while the literature-based chapter provides the scholarly context and identifies the research gap that justifies the study [55,56].

The middle part of the thesis contains the analytical core of the study. Chapter 3 presents the methodology, including the modeling assumptions, governing equations, thermophysical properties, boundary conditions, and the numerical or CFD procedures adopted to evaluate heat transfer, thermal losses, and hydraulic behavior. Where relevant, it also describes the geometry of the collector, the receiver configuration, and the climatic inputs representing El Oued conditions. Chapter 4 presents the results in a structured form, while Chapter 5 discusses those results in relation to the research questions, previous literature, and the expected practical implications for PTC operation in arid environments. This sequencing reflects established academic practice, according to which the reporting and discussion chapters constitute the central body of the thesis, present the data and analysis, and connect the findings back to the research questions, methods, and theoretical framework outlined earlier in the document [55,56].

1.12 Conclusion

This chapter has shown that the parabolic trough collector (PTC) remains the most mature and best-documented linear concentrating solar technology, with performance governed by the

coupled effects of optical efficiency, receiver heat loss, heat-transfer-fluid circulation, tracking accuracy, and local climatic conditions. The reviewed literature also makes clear that useful heat gain and thermal efficiency are not controlled by solar irradiation alone, but by the interaction between reflector quality, receiver configuration, operating flow conditions, and external losses by convection, radiation, and conduction, while hydraulic losses and soiling further influence practical plant output and maintenance requirements. In the specific context of El Oued, the combination of high direct normal irradiance, high ambient temperatures, wind exposure, and dust-prone desert conditions confirms that the site is promising for solar concentration, but also technically demanding for long-term PTC operation. Therefore, the chapter establishes the scientific basis for the present study by showing that a site-specific thermo-optical and thermo-hydraulic assessment is necessary to understand how PTC systems can be adapted, optimized, and reliably operated under Saharan conditions, especially where current literature still lacks integrated analyses combining receiver behavior, hydraulic performance, optical degradation, and local environmental constraints.



Chapter II

Mathematical and Numerical Modeling

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Mathematical and Numerical Modeling

2.1 Introduction

The thermal performance of a parabolic trough collector depends on the interaction between solar radiation concentration, absorber-wall heat transfer, internal fluid heating, and thermal losses to the surrounding environment. In this work, the absorber receiver tube is modeled using a steady two-dimensional axisymmetric numerical formulation. The model is based on the MATLAB codes developed for validation and parametric study, where the unknown field is the fluid temperature $T(x, r)$ inside the receiver tube. The codes solve a CFD-like axisymmetric energy equation using axial upwind convection, implicit radial diffusion, nonlinear wall heat-loss boundary conditions, and prescribed fully developed velocity profiles.

The purpose of this chapter is to present the mathematical equations and numerical procedure used to simulate the parabolic trough collector receiver under the arid climatic conditions of El Oued, Algeria. The model predicts the two-dimensional temperature field, outlet bulk temperature, useful heat gain, absorber wall temperature, heat losses, thermal efficiency, receiver efficiency, pressure drop, and dimensionless flow parameters.

The approach adopted in this work is intermediate between a simplified one-dimensional energy balance and a full three-dimensional CFD simulation. Unlike a one-dimensional model, the present formulation resolves the radial temperature distribution inside the working fluid. However, unlike a full CFD model, the momentum equation is not solved directly; instead, the axial velocity profile is prescribed according to the flow regime. This makes the model computationally efficient and suitable for validation, grid-independence analysis, and extensive parametric investigations.

2.2 Physical Description of the Modeled Receiver

The modeled system is the absorber tube of a parabolic trough collector. Solar radiation is reflected by the parabolic mirror and concentrated onto the receiver tube. The absorber converts the concentrated solar radiation into heat, which is transferred through the absorber wall to the heat-transfer fluid flowing inside the tube.

The receiver is assumed to be a straight cylindrical tube of length L , inner diameter D_i , outer diameter D_o , inner radius R_i , and outer radius R_o . The heat-transfer fluid enters the tube at an inlet temperature T_{in} and mass flow rate $\dot{m}$. The surrounding environment is characterized by the ambient temperature T_{amb} , wind speed v_{wind} , and effective sky temperature T_{sky} .

The internal fluid domain is solved explicitly, while the absorber wall is represented through an equivalent cylindrical thermal resistance. The external heat loss from the absorber is modeled as the sum of convective loss to ambient air and radiative loss to the sky.

The receiver inner radius is defined as

$$R_i = \frac{D_i}{2} \quad (1)$$

The receiver outer radius is

$$R_o = \frac{D_o}{2} \quad (2)$$

The absorber wall thickness is

$$e_w = R_o - R_i \quad (3)$$

or equivalently,

$$e_w = \frac{D_o - D_i}{2} \quad (4)$$

The collector aperture area is

$$A_{ap} = W_{ap}L \quad (5)$$

where W_{ap} is the aperture width of the parabolic trough collector.

The internal flow area of the absorber tube is

$$A_f = \pi R_i^2 \quad (6)$$

or, in terms of the inner diameter,

$$A_f = \frac{\pi D_i^2}{4} \quad (7)$$

The external perimeter of the absorber tube per unit length is

$$P_o = 2\pi R_o \quad (8)$$

These geometrical relations are used throughout the model to calculate the flow velocity, absorbed solar heat, heat-transfer area, and external thermal losses.

2.3 Modeling Assumptions

The mathematical model is based on the following assumptions:

1. The receiver operates under steady-state conditions.
2. The fluid flow is incompressible and single-phase.
3. The thermophysical properties of the fluid are constant for each simulation case.
4. The absorber material properties are constant.
5. The flow inside the receiver is hydrodynamically fully developed.
6. The axial velocity profile is prescribed according to the flow regime.
7. Axial heat conduction in the fluid is negligible compared with axial convection.
8. Heat transfer inside the receiver is axisymmetric.
9. Circumferential non-uniformity of the concentrated solar flux is not explicitly resolved.
10. The absorber wall is modeled using an equivalent cylindrical conduction resistance.
11. External heat losses occur by convection to ambient air and radiation to the sky.
12. The glass envelope, vacuum annulus, optical tracking errors, dust deposition, selective-coating degradation, and thermo-mechanical stresses are not explicitly modeled.

These assumptions allow the dominant thermal and hydraulic behavior of the receiver to be represented with good computational efficiency.

2.4 Solar Radiation and Absorbed Heat Model

The useful solar resource for a parabolic trough collector is the direct normal irradiance DNI , because only beam radiation can be concentrated by the parabolic reflector. The solar radiation intercepted by the aperture is converted into absorbed heat at the receiver by accounting for optical efficiency, absorber absorptivity, and incidence angle modifier.

The mean absorbed heat rate per unit receiver length is

$$q'_{abs,mean} = DNI W_{ap} \eta_{opt} \alpha_{abs} IAM \quad (9)$$

where $q'_{abs,mean}$ is the mean absorbed heat rate per unit length, η_{opt} is the optical efficiency, α_{abs} is the absorber solar absorptivity, and IAM is the incidence angle modifier.

For a uniform absorbed-flux model, the axial heat-flux shape function is

$$S(x) = 1 \quad (10)$$

For a non-uniform cosine-centered distribution, the unnormalized shape function is

$$S_0(x) = 1 + A_q \cos \left[2\pi \left(\frac{x}{L} - \frac{1}{2} \right) \right] \quad (11)$$

where A_q is the axial flux non-uniformity amplitude.

To keep the total absorbed heat unchanged, the shape function is normalized as

$$S(x) = \frac{S_0(x)}{\bar{S}_0} \quad (12)$$

where $\bar{S}_0$ is the mean value of $S_0(x)$ along the receiver length.

The local absorbed heat rate per unit length is therefore

$$q'_{abs}(x) = q'_{abs,mean} S(x) \quad (13)$$

The total absorbed heat by the receiver is

$$Q_{abs} = \int_0^L q'_{abs}(x) dx \quad (14)$$

The total incident solar power over the collector aperture is

$$Q_{solar} = DNI W_{ap} L \quad (15)$$

2.5 Governing Energy Equation

The thermal field inside the receiver is described by the steady axisymmetric energy equation in cylindrical coordinates. The unknown is the fluid temperature $T(x, r)$, where x is the axial coordinate and r is the radial coordinate.

The governing equation is

$$\rho c_p u(r) \frac{\partial T}{\partial x} = \frac{1}{r} \frac{\partial}{\partial r} \left[r k_{eff}(r) \frac{\partial T}{\partial r} \right] \quad (16)$$

where ρ is the fluid density, c_p is the specific heat capacity, $u(r)$ is the axial velocity profile, and $k_{eff}(r)$ is the effective radial thermal conductivity.

The left-hand side of Eq. (16) represents heat transport by axial convection. The right-hand side represents radial heat diffusion inside the fluid. Because axial conduction is neglected, the equation has a marching character in the axial direction.

The inlet boundary condition is

$$T(0, r) = T_{in} \quad (17)$$

At the tube centerline, the symmetry condition is

$$\frac{\partial T}{\partial r} \big|_{r=0} = 0 \quad (18)$$

At the wall, the heat transferred to the fluid is determined from the balance between absorbed solar heat and external heat losses:

$$q'_{fluid}(x) = q'_{abs}(x) - q'_{loss}(T_{wo}) \quad (19)$$

where T_{wo} is the outer absorber-wall temperature.

2.6 Flow Model and Dimensionless Numbers

The average velocity of the fluid is calculated from the imposed mass flow rate:

$$U_{avg} = \frac{\dot{m}}{\rho A_f} \quad (20)$$

The Reynolds number is

$$Re = \frac{\rho U_{avg} D_i}{\mu} \quad (21)$$

where μ is the dynamic viscosity of the fluid.

The Prandtl number is

$$Pr = \frac{c_p \mu}{k_f} \quad (22)$$

where k_f is the molecular thermal conductivity of the fluid.

The Peclet number is

$$Pe = Re Pr \quad (23)$$

The flow regime is identified from the Reynolds number. Laminar flow is assumed when

$$Re < 2300 \quad (24)$$

Transitional flow is considered when

$$2300 \leq Re < 4000 \quad (25)$$

Turbulent flow is considered when

$$Re \geq 4000 \quad (26)$$

2.7 Velocity Profile Formulation

Because the momentum equations are not solved directly, the axial velocity profile is prescribed according to the flow regime.

For fully developed laminar flow in a circular tube, the parabolic Poiseuille profile is used:

$$\phi_{lam}(r) = 2 \left[1 - \left(\frac{r}{R_i} \right)^2 \right] \quad (27)$$

For turbulent flow, a one-seventh power-law velocity profile is used:

$$\phi_{turb}(r) = \left(1 - \frac{r}{R_i} \right)^{1/7} \quad (28)$$

For plug flow, the velocity profile is

$$\phi_{plug}(r) = 1 \quad (29)$$

For transitional flow, a linear blending factor is defined as

$$\gamma = \frac{Re - 2300}{4000 - 2300} \quad (30)$$

with

$$0 \leq \gamma \leq 1 \quad (31)$$

The transitional velocity shape is then

$$\phi_{tr}(r) = (1 - \gamma)\phi_{lam}(r) + \gamma\phi_{turb}(r) \quad (32)$$

After selecting the appropriate shape function, the velocity profile is normalized to conserve the imposed mass flow rate:

$$u(r) = U_{avg} \frac{\phi(r)}{\langle \phi \rangle_A} \quad (33)$$

where $\langle \phi \rangle_A$ is the area-weighted average of the shape function:

$$\langle \phi \rangle_A = \frac{\int_0^{R_i} \phi(r) 2\pi r dr}{\int_0^{R_i} 2\pi r dr} \quad (34)$$

In the numerical implementation, the discrete area-weighted average is

$$\langle \phi \rangle_A = \frac{\sum_{j=1}^{N_r} \phi_j A_j}{\sum_{j=1}^{N_r} A_j} \quad (35)$$

This normalization ensures exact conservation of the prescribed mass flow rate on the numerical grid.

2.8 Effective Radial Thermal Conductivity

For laminar flow, radial heat transfer occurs only by molecular conduction:

$$k_{eff}(r) = k_f \quad (36)$$

For turbulent flow, radial heat transfer is enhanced by turbulent mixing. The effective thermal conductivity is expressed as

$$k_{eff}(r) = k_f + \rho c_p \alpha_t(r) \quad (37)$$

where $\alpha_t(r)$ is the turbulent thermal diffusivity.

The turbulent thermal diffusivity is related to the turbulent eddy viscosity by

$$\alpha_t(r) = \frac{\nu_t(r)}{Pr_t} \quad (38)$$

where Pr_t is the turbulent Prandtl number.

The molecular kinematic viscosity is

$$\nu = \frac{\mu}{\rho} \quad (39)$$

The friction velocity is

$$u_\tau = U_{avg} \sqrt{\frac{f}{8}} \quad (40)$$

The distance from the wall is

$$y = R_i - r \quad (41)$$

The dimensionless wall coordinate is

$$y^+ = \frac{y u_\tau}{\nu} \quad (42)$$

The turbulent eddy viscosity is calculated using a damped mixing-length expression:

$$v_t(r) = \kappa u_\tau y \left(1 - \frac{y}{R_i}\right) \left[1 - \exp\left(-\frac{y^+}{A^+}\right)\right]^2 \quad (43)$$

where κ is the von Karman constant and A^+ is the Van Driest damping constant.

This expression increases the radial thermal diffusivity in turbulent flow while forcing turbulent diffusion to vanish near the wall.

2.9 Absorber Wall Thermal Resistance

The absorber wall is not meshed as a separate solid domain. Instead, heat transfer between the outer absorber surface and the near-wall fluid cell is modeled using a series thermal resistance.

The radial position of the near-wall fluid cell center is

$$r_p = R_i - \frac{\Delta r}{2} \quad (44)$$

The fluid half-cell thermal resistance is

$$R_{f,1/2} = \frac{\ln\left(\frac{R_i}{r_p}\right)}{2\pi k_f} \quad (45)$$

The absorber-wall conduction resistance is

$$R_s = \frac{\ln\left(\frac{R_o}{R_i}\right)}{2\pi k_s} \quad (46)$$

where k_s is the thermal conductivity of the absorber material.

The total conductance between the outer absorber wall and the near-wall fluid cell is

$$G_b = \frac{1}{R_{f,1/2} + R_s} \quad (47)$$

The heat transferred from the wall to the fluid per unit receiver length is

$$q'_{fluid} = G_b(T_{wo} - T_p) \quad (48)$$

where T_p is the near-wall fluid-cell temperature.

The inner absorber-wall temperature is reconstructed as

$$T_{wi} = T_p + q'_{fluid} R_{f,1/2} \quad (49)$$

This resistance formulation allows the wall conduction effect to be included without increasing the number of unknowns in the numerical system.

2.10 External Heat-Loss Model

The external heat loss from the absorber tube is modeled as the sum of convective and radiative losses:

$$q'_{loss} = q'_{conv} + q'_{rad} \quad (50)$$

The convective heat loss per unit length is

$$q'_{conv} = h_{wind} P_o (T_{wo} - T_{amb}) \quad (51)$$

The wind-dependent convective heat-transfer coefficient is approximated by

$$h_{wind} = 5.7 + 3.8v_{wind} \quad (52)$$

The radiative heat loss per unit length is

$$q'_{rad} = \varepsilon \sigma P_o \left[(T_{wo} + 273.15)^4 - (T_{sky} + 273.15)^4 \right] \quad (53)$$

where ε is the absorber emissivity and σ is the Stefan-Boltzmann constant.

The effective sky temperature is calculated as

$$T_{sky} = T_{amb} - \Delta T_{sky} \quad (54)$$

The complete nonlinear wall energy balance is

$$q'_{abs}(x) = q'_{fluid}(x) + q'_{loss}(T_{wo}) \quad (55)$$

Substituting the wall-to-fluid heat transfer and loss terms gives

$$q'_{abs}(x) = G_b(T_{wo} - T_p) + h_{wind} P_o (T_{wo} - T_{amb}) + \varepsilon \sigma P_o \left[(T_{wo} + 273.15)^4 - (T_{sky} + 273.15)^4 \right] \quad (56)$$

Because Eq. (56) contains the fourth power of the wall temperature, the wall boundary condition is nonlinear.

2.11 Solution of the Nonlinear Wall Boundary Condition

For a known near-wall fluid temperature T_p , the unknown outer wall temperature T_{wo} is obtained by solving the residual equation

$$F(T_{wo}) = G_b(T_{wo} - T_p) + q'_{loss}(T_{wo}) - q'_{abs}(x) = 0 \quad (57)$$

The Newton iteration is written as

$$T_{wo}^{(m+1)} = T_{wo}^{(m)} - \frac{F(T_{wo}^{(m)})}{\left. \frac{dF}{dT_{wo}} \right|_{T_{wo}^{(m)}}} \quad (58)$$

The derivative of the residual is

$$\frac{dF}{dT_{wo}} = G_b + h_{wind}P_o + 4\varepsilon\sigma P_o(T_{wo}+273.15)^3 \quad (59)$$

The wall-temperature iteration is stopped when

$$|T_{wo}^{(m+1)} - T_{wo}^{(m)}| < \varepsilon_{wall} \quad (60)$$

where ε_{wall} is the prescribed wall convergence tolerance.

2.12 Numerical Grid

The receiver is discretized using N_x axial nodes and N_r radial control volumes.

The axial step size is

$$\Delta x = \frac{L}{N_x - 1} \quad (61)$$

The radial step size is

$$\Delta r = \frac{R_i}{N_r} \quad (62)$$

The radial cell centers are located at

$$r_j = \left(j - \frac{1}{2}\right) \Delta r, j = 1, 2, \dots, N_r \quad (63)$$

The west face of radial control volume j is

$$r_{w,j} = (j - 1) \Delta r \quad (64)$$

The east face of radial control volume j is

$$r_{e,j} = j \Delta r \quad (65)$$

The area of each annular radial control volume is

$$A_j = \pi(r_{e,j}^2 - r_{w,j}^2) \quad (66)$$

The west radial diffusion conductance is

$$G_{w,j} = \frac{k_{w,j} 2\pi r_{w,j}}{\Delta r} \quad (67)$$

The east radial diffusion conductance is

$$G_{e,j} = \frac{k_{e,j} 2\pi r_{e,j}}{\Delta r} \quad (68)$$

At the tube centerline, $r = 0$, the west-face conductance is zero. Therefore, the symmetry condition is automatically satisfied.

2.13 Finite-Volume Discretization of the Energy Equation

The governing equation is discretized using axial upwind convection and implicit radial diffusion. For radial control volume j , the axial convective coefficient is

$$a_{conv,j} = \frac{\rho c_p u_j A_j}{\Delta x} \quad (69)$$

For an interior radial control volume, the discretized energy equation at axial station i is

$$a_{P,j} T_j^i = a_{W,j} T_{j-1}^i + a_{E,j} T_{j+1}^i + a_{conv,j} T_j^{i-1} \quad (70)$$

where T_j^i is the temperature at radial cell j and axial station i .

The west diffusion coefficient is

$$a_{W,j} = G_{w,j} \quad (71)$$

The east diffusion coefficient is

$$a_{E,j} = G_{e,j} \quad (72)$$

The central coefficient is

$$a_{P,j} = a_{conv,j} + a_{W,j} + a_{E,j} \quad (73)$$

At the centerline cell,

$$a_{W,1} = 0 \quad (74)$$

At the near-wall cell, the external heat addition from the absorber is added as a wall source term.

2.14 Linearization of the Wall Heat Source

The heat transferred from the absorber wall to the near-wall fluid cell is nonlinear because it depends on the wall heat balance. To include this heat source in the algebraic system, it is linearized as

$$q'_{fluid}(T_P) \approx S_u + S_p T_P \quad (75)$$

The source slope is

$$S_p = \frac{dq'_{fluid}}{dT_P} \quad (76)$$

The source intercept is

$$S_u = q'_{fluid}(T_P^*) - S_p T_P^* \quad (77)$$

where T_P^* is the current iterative estimate of the near-wall fluid temperature.

The modified wall-cell central coefficient becomes

$$a_{P,N_r}^{modified} = a_{P,N_r} - S_p \quad (78)$$

The modified right-hand-side term becomes

$$b_{N_r}^{modified} = b_{N_r} + S_u \quad (79)$$

This procedure keeps the algebraic system tridiagonal while accounting for the nonlinear wall heat balance.

2.15 Tridiagonal Algebraic System

At each axial station, the radial discretization produces a tridiagonal system:

$$A_j T_{j-1}^i + B_j T_j^i + C_j T_{j+1}^i = D_j, j = 1, 2, \dots, N_r \quad (80)$$

where A_j , B_j , and C_j are the lower, main, and upper diagonal coefficients, respectively, and D_j is the right-hand-side term.

The system is solved using the Thomas algorithm. The forward elimination begins with

$$C_1^* = \frac{C_1}{B_1} \quad (81)$$

$$D_1^* = \frac{D_1}{B_1} \quad (82)$$

For $j = 2, 3, \dots, N_r$, the modified coefficients are calculated as

$$C_j^* = \frac{C_j}{B_j - A_j C_{j-1}^*} \quad (83)$$

$$D_j^* = \frac{D_j - A_j D_{j-1}^*}{B_j - A_j C_{j-1}^*} \quad (84)$$

The back substitution starts from the wall-side radial cell:

$$T_{N_r}^i = D_{N_r}^* \quad (85)$$

Then, for $j = N_r - 1, N_r - 2, \dots, 1$,

$$T_j^i = D_j^* - C_j^* T_{j+1}^i \quad (86)$$

This method is computationally efficient because the number of operations increases linearly with the number of radial cells.

2.16 Axial Marching and Nonlinear Iteration

The solution begins at the inlet, where the temperature profile is known from Eq. (17). The solution then proceeds from one axial station to the next. At each axial station, nonlinear iterations are performed because the absorber wall temperature and heat losses depend on the unknown near-wall fluid temperature.

The under-relaxed temperature update is

$$T_j^{new} = \omega T_j^{sol} + (1 - \omega) T_j^{old} \quad (87)$$

where ω is the relaxation factor.

The nonlinear convergence error is calculated as

$$\varepsilon_{NL} = \max_{1 \leq j \leq N_r} |T_j^{new} - T_j^{old}| \quad (88)$$

The nonlinear solution is considered converged when

$$\varepsilon_{NL} < \varepsilon_{tol} \quad (89)$$

where ε_{tol} is the prescribed nonlinear convergence tolerance.

The numerical algorithm can be summarized as follows:

1. Define the receiver geometry, optical properties, fluid properties, operating conditions, environmental conditions, and numerical parameters.
2. Generate the axial and radial grids.
3. Calculate the flow area, average velocity, Reynolds number, Prandtl number, and Peclet number.
4. Select and normalize the velocity profile.
5. Calculate the absorbed heat distribution along the receiver.
6. Compute the wall thermal resistance and boundary conductance.
7. Initialize the fluid temperature at the inlet.

8. March axially from inlet to outlet.
9. At each axial station, solve the nonlinear wall heat balance.
10. Linearize the wall heat source.
11. Assemble the tridiagonal system.
12. Solve the radial system using the Thomas algorithm.
13. Apply under-relaxation.
14. Check nonlinear convergence.
15. Store the converged temperature profile.
16. Post-process the thermal and hydraulic quantities.

2.17 Bulk Temperature and Useful Heat Gain

The bulk fluid temperature is calculated as a velocity-weighted average over the receiver cross-section:

$$T_b(x) = \frac{\int_0^{R_i} \rho c_p u(r) T(x, r) 2\pi r dr}{\int_0^{R_i} \rho c_p u(r) 2\pi r dr} \quad (90)$$

In discrete form,

$$T_b(x_i) = \frac{\sum_{j=1}^{N_r} \rho c_p u_j T_j^i A_j}{\sum_{j=1}^{N_r} \rho c_p u_j A_j} \quad (91)$$

The outlet bulk temperature is

$$T_{out} = T_b(L) \quad (92)$$

The fluid temperature rise is

$$\Delta T = T_{out} - T_{in} \quad (93)$$

The useful heat gain is

$$Q_u = \dot{m} c_p (T_{out} - T_{in}) \quad (94)$$

The integrated heat transferred to the fluid is

$$Q_{fluid} = \int_0^L q'_{fluid}(x) dx \quad (95)$$

The integrated heat loss is

$$Q_{loss} = \int_0^L q'_{loss}(x) dx \quad (96)$$

The global receiver energy balance is

$$Q_{abs} = Q_{fluid} + Q_{loss} \quad (97)$$

2.18 Thermal Efficiency and Receiver Efficiency

The thermal efficiency of the collector is defined as the ratio of useful heat gain to the incident solar power on the aperture:

$$\eta_{th} = \frac{Q_u}{Q_{solar}} \quad (98)$$

The receiver efficiency is defined as the ratio of useful heat gain to absorbed heat:

$$\eta_{rec} = \frac{Q_u}{Q_{abs}} \quad (99)$$

The heat-loss fraction is

$$LF = \frac{Q_{loss}}{Q_{abs}} \quad (100)$$

The energy residual is used to evaluate global heat-balance accuracy:

$$\varepsilon_E = \frac{Q_{abs} - Q_{loss} - Q_{fluid}}{\max(|Q_{abs}|, 1)} \quad (101)$$

The bulk closure error is

$$\varepsilon_B = \frac{Q_{fluid} - Q_u}{\max(|Q_{fluid}|, 1)} \quad (102)$$

Small values of ε_E and ε_B indicate that the numerical solution satisfies energy conservation.

2.19 Pressure Drop Calculation

The pressure drop along the absorber tube is calculated using the Darcy-Weisbach equation:

$$\Delta p = f \frac{L}{D_i} \frac{\rho U_{avg}^2}{2} \quad (103)$$

For laminar flow, the Darcy friction factor is

$$f = \frac{64}{Re} \quad (104)$$

For turbulent flow, the Blasius correlation is used:

$$f = \frac{0.3164}{Re^{0.25}} \quad (105)$$

For transitional flow, the friction factor is interpolated between laminar and turbulent values:

$$f = (1 - \gamma) \frac{64}{Re} + \gamma \frac{0.3164}{Re^{0.25}} \quad (106)$$

The pressure drop is important because increasing the mass flow rate may improve heat extraction but also increases the hydraulic penalty.

2.20 Nusselt Number Evaluation

The internal heat-transfer coefficient is calculated from the wall-to-fluid heat transfer:

$$h_i(x) = \frac{q'_{fluid}(x)}{\pi D_i [T_{wi}(x) - T_b(x)]} \quad (107)$$

The local Nusselt number is

$$Nu(x) = \frac{h_i(x) D_i}{k_f} \quad (108)$$

For fully developed laminar flow with constant heat flux, the reference Nusselt number is

$$Nu_{lam} = 4.364 \quad (109)$$

For turbulent flow, the Gnielinski correlation is used as a reference:

$$Nu_G = \frac{\left(\frac{f}{8}\right) (Re - 1000) Pr}{1 + 12.7 \left(\frac{f}{8}\right)^{1/2} (Pr^{2/3} - 1)} \quad (110)$$

The comparison between the numerical Nusselt number and reference correlations is used as part of the thermal validation.

2.21 Validation Methodology

The numerical model is validated through several complementary tests. Each test verifies a specific part of the mathematical and numerical formulation.

The first validation concerns the velocity profile. The prescribed velocity distribution is compared with the analytical laminar or turbulent profile, and the mass-flow conservation is checked.

The second validation concerns pressure drop. The pressure drop predicted by the model is compared with the Darcy-Weisbach equation and standard friction-factor correlations.

The third validation concerns energy conservation. The absorbed heat, heat transferred to the fluid, and heat lost to the environment must satisfy the global energy balance given by Eq. (97).

The fourth validation is the zero-loss analytical test. When heat losses are disabled, the absorbed solar heat must be completely transferred to the fluid. The analytical outlet temperature is

$$T_{out,analytical} = T_{in} + \frac{Q_{abs}}{\dot{m}c_p} \quad (111)$$

The corresponding useful heat gain is

$$Q_{u,analytical} = Q_{abs} \quad (112)$$

The outlet-temperature error is

$$Error_{T_{out}} = \left| \frac{T_{out,numerical} - T_{out,analytical}}{T_{out,analytical}} \right| \times 100 \quad (113)$$

The useful-heat error is

$$Error_{Q_u} = \left| \frac{Q_{u,numerical} - Q_{u,analytical}}{Q_{u,analytical}} \right| \times 100 \quad (114)$$

The fifth validation concerns the Nusselt number, where the numerical heat-transfer prediction is compared with standard laminar and turbulent correlations.

The sixth validation concerns physical trends. The model must predict that increasing direct normal irradiance increases useful heat gain, increasing wind speed increases thermal losses, increasing absorber emissivity increases radiative losses, and increasing mass flow rate decreases temperature rise while increasing pressure drop.

2.22 Grid-Independence Study

A grid-independence study is performed to verify that the solution is not significantly affected by the number of axial and radial cells. Several grids are tested by increasing N_x and N_r . For each grid, the main monitored variables are outlet temperature, useful heat gain, thermal efficiency, maximum absorber wall temperature, pressure drop, energy residual, and bulk closure error.

The relative variation of a monitored variable Φ between two successive grids is calculated as

$$\delta_\Phi = \left| \frac{\Phi_{fine} - \Phi_{coarse}}{\max(|\Phi_{fine}|, 1)} \right| \times 100 \quad (115)$$

A grid is considered independent when further refinement causes negligible changes in the main output variables. In the present work, the baseline grid of approximately 240 to

250 axial nodes and 75 to 80 radial control volumes provides a good balance between numerical accuracy and computational cost.

2.23 Parametric Study Methodology

After validation and grid selection, the model is used to conduct a parametric study of the parabolic trough receiver under El Oued climatic conditions. The parametric study follows the one-factor-at-a-time method. In this method, one parameter is varied while all other parameters are kept at their baseline values.

The climatic parameters include direct normal irradiance, ambient temperature, wind speed, and sky-temperature offset. The operating parameters include mass flow rate and inlet temperature. The geometrical parameters include receiver length, aperture width, inner diameter, and wall thickness. The optical and absorber parameters include optical efficiency, absorber absorptivity, absorber emissivity, incidence angle modifier, and flux non-uniformity amplitude. The material and fluid parameters include absorber thermal conductivity and working-fluid type.

For any performance indicator Φ , the normalized sensitivity is defined as

$$S_{\Phi} = \frac{\Phi_{max} - \Phi_{min}}{|\Phi_{baseline}|} \times 100 \quad (116)$$

This sensitivity index is used to rank the influence of each parameter on the thermal and hydraulic performance of the collector.

In addition to the one-factor-at-a-time study, selected two-parameter interaction maps are considered. These maps evaluate the coupled effects of important parameter pairs, such as direct normal irradiance and mass flow rate, ambient temperature and wind speed, inner diameter and mass flow rate, receiver length and aperture width, and absorber emissivity and wind speed.

2.24 Main Model Outputs

The numerical model provides the following outputs:

The two-dimensional temperature field inside the receiver is

$$T = T(x, r) \quad (117)$$

The axial bulk temperature distribution is

$$T_b = T_b(x) \quad (118)$$

The outer absorber-wall temperature distribution is

$$T_{wo} = T_{wo}(x) \quad (119)$$

The inner absorber-wall temperature distribution is

$$T_{wi} = T_{wi}(x) \quad (120)$$

The distributed absorbed heat, heat loss, and heat transferred to the fluid are

$$q'_{abs} = q'_{abs}(x) \quad (121)$$

$$q'_{loss} = q'_{loss}(x) \quad (122)$$

$$q'_{fluid} = q'_{fluid}(x) \quad (123)$$

The global outputs are the outlet temperature, useful heat gain, thermal efficiency, receiver efficiency, pressure drop, Reynolds number, Prandtl number, Peclet number, energy residual, and bulk closure error.

2.25 Limitations of the Model

Although the present model captures the main thermal and hydraulic behavior of the receiver, it has some limitations. The model is steady-state and therefore does not describe transient effects such as start-up, shut-down, cloud passage, or daily solar variations. The velocity field is prescribed and not obtained from a direct solution of the momentum equations. Fluid properties are assumed constant for each case. Circumferential variation of the concentrated solar flux is not explicitly represented. The glass envelope, vacuum annulus, selective coating degradation, dust deposition, mirror optical errors, and structural deformation are not included.

Despite these limitations, the model is appropriate for receiver-level thermal analysis because it resolves the radial temperature field, accounts for nonlinear external losses, predicts pressure drop, verifies energy conservation, and enables systematic parametric investigation.

2.26 Conclusion

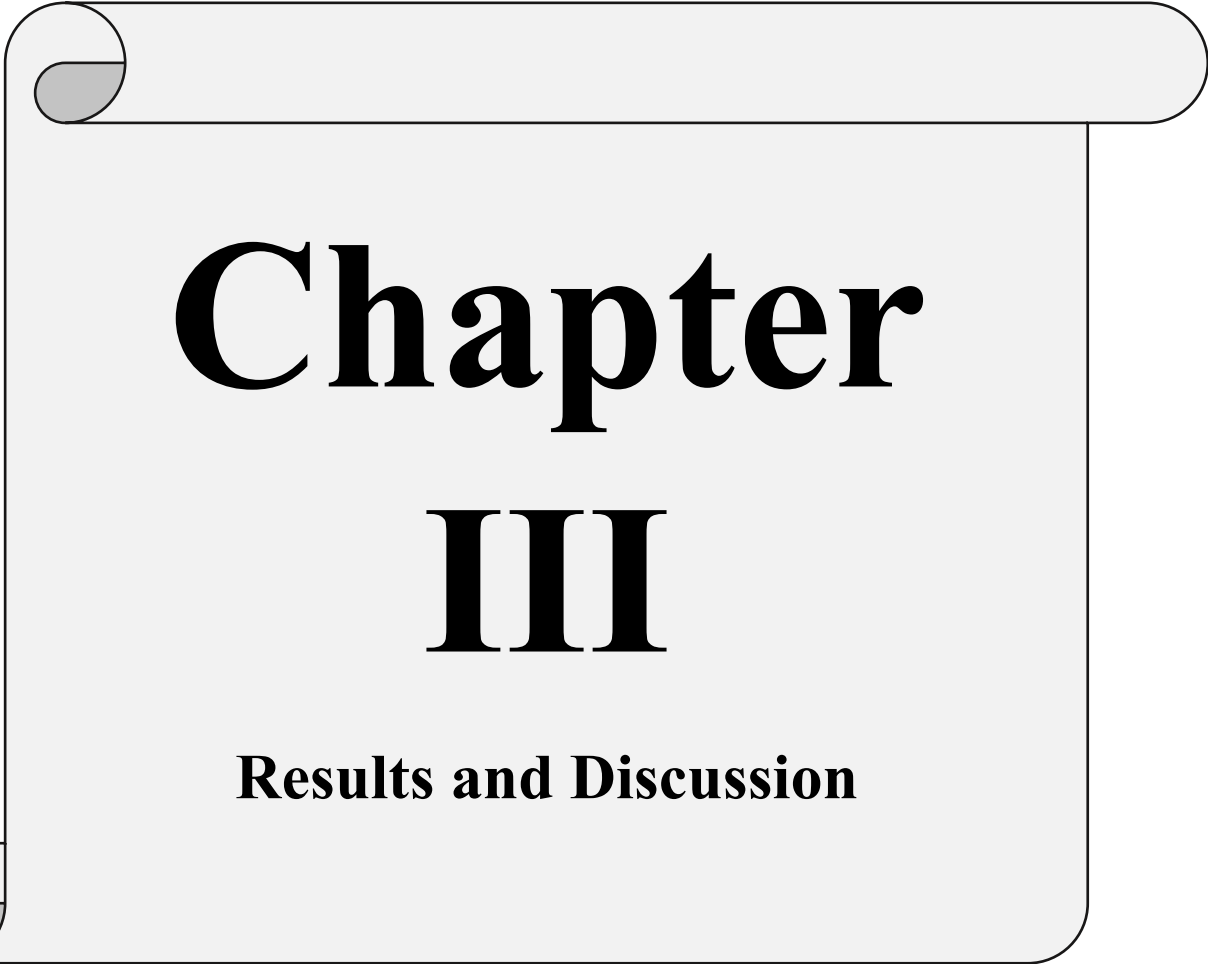
This chapter presented the mathematical and numerical model used to simulate the absorber receiver of a parabolic trough collector. The receiver was modeled as a steady two-dimensional axisymmetric domain in which the heat-transfer-fluid temperature is governed by axial convection and radial diffusion.

The solar input was formulated as an absorbed heat rate per unit receiver length based on direct normal irradiance, aperture width, optical efficiency, absorber absorptivity, and incidence angle modifier. The flow model used fully developed velocity profiles for laminar, transitional, turbulent, and plug-flow regimes. The effective radial thermal conductivity included molecular conduction and, when required, turbulent eddy diffusivity.

The absorber wall was represented by an equivalent cylindrical resistance. The external wall boundary condition was formulated as a nonlinear heat balance including heat transferred to the fluid, convective loss to ambient air, and radiative loss to the sky. The governing equation was discretized using axial upwind convection and implicit radial diffusion. At each axial station, the resulting tridiagonal algebraic system was solved using the Thomas algorithm, while nonlinear wall coupling was handled iteratively with under-relaxation.

The chapter also defined the post-processing equations for bulk temperature, outlet temperature, useful heat gain, absorbed heat, heat loss, thermal efficiency, receiver efficiency, pressure drop, Nusselt number, energy residual, and bulk closure error. The validation methodology, grid-independence procedure, and parametric-study formulation were also presented.

The mathematical and numerical model developed in this chapter provides the foundation for the simulation results and performance analysis of the parabolic trough collector under the arid climatic conditions of El Oued.



Chapter

III

Results and Discussion

Chapter III:

Results and Discussion

3.1 Introduction

This chapter presents and discusses the numerical results obtained from the two-dimensional axisymmetric model of the parabolic trough collector receiver. The objective is to evaluate the reliability of the developed MATLAB solver and to analyze the thermal and hydraulic behavior of the receiver under the reference climatic conditions of El Oued, Algeria.

The numerical model solves the steady energy equation inside the absorber tube using an axisymmetric finite-volume/finite-difference formulation. The solution procedure includes axial upwind convection, implicit radial diffusion, nonlinear absorber heat losses, prescribed fully developed velocity profiles, and optional turbulent radial diffusivity. The validation code also includes velocity-profile validation, hydrodynamic pressure-drop validation, energy conservation validation, zero-loss analytical validation, Nusselt-number validation, and physical-trend validation.

The first part of this chapter discusses the baseline thermal fields and the principal validation results. These results are essential because they demonstrate whether the numerical model can reproduce physically consistent temperature distributions, hydraulic pressure losses, and convective heat-transfer trends before being used for wider parametric analysis.

3.2 Baseline Thermal Behavior of the Receiver

3.2.1 Two-Dimensional Temperature Field

Figure III.1 presents the two-dimensional temperature field $T(x, r)$ inside the receiver tube. The horizontal direction represents the axial coordinate x , while the vertical direction represents the radial position r . The temperature field varies from the inlet temperature of approximately 35°C to a maximum fluid temperature of about 60.93°C near the heated wall region.

The figure shows a clear axial increase in fluid temperature as the flow moves from the inlet to the outlet. This behavior is physically expected because the heat-transfer fluid continuously absorbs energy from the absorber wall along the receiver length. At the inlet, the temperature is almost uniform over the radial direction because the boundary condition imposes

a uniform inlet temperature. As the fluid advances downstream, a radial temperature gradient develops due to heat transfer from the wall toward the tube core.

The highest temperatures are located close to the tube wall, while the lowest temperatures remain closer to the tube centerline. This confirms that heat enters the fluid from the absorber wall and diffuses radially inward. The formation of this radial gradient is important because it proves that the model is not behaving as a simple one-dimensional plug-flow model; instead, it resolves the internal thermal stratification of the receiver fluid.

The temperature contours are smooth and continuous, with no numerical oscillations or nonphysical discontinuities. This indicates that the axial marching scheme and the implicit radial diffusion treatment are numerically stable. The smoothness of the solution also confirms that the nonlinear wall boundary condition has been correctly coupled with the internal fluid energy equation.

A noticeable feature of the temperature field is that the strongest heating occurs in the downstream and near-wall regions. This is explained by the combined influence of continuous solar heat absorption and finite radial heat diffusion. The wall receives concentrated solar energy, while the fluid core is heated more gradually. Therefore, the near-wall fluid temperature remains higher than the core temperature throughout most of the receiver length.

Overall, Figure III.1 demonstrates that the model correctly captures the main physics of receiver heating: axial temperature rise, radial heat diffusion, wall-dominated heat input, and progressive thermal development of the internal flow.

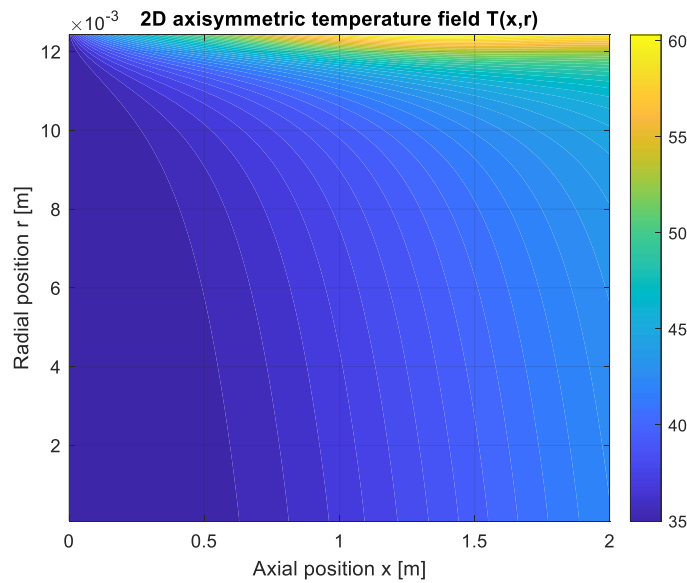


Figure III.1. Two-dimensional axisymmetric temperature field inside the receiver.

3.2.2 Axial Temperature Distributions

Figure III.2 shows the axial evolution of three important temperatures: the bulk fluid temperature, the inner absorber-wall temperature, and the outer absorber-wall temperature. The bulk fluid temperature increases from 35°C at the inlet to approximately 45.28°C at the outlet. This corresponds to a temperature rise of about 10.28°C , confirming that the receiver effectively transfers absorbed solar energy to the working fluid.

The inner absorber-wall temperature increases from approximately 36.20°C near the inlet and reaches a maximum value of about 62.48°C . The outer absorber-wall temperature increases from approximately 37.50°C and reaches a maximum value of about 64.20°C . The outer wall temperature is higher than the inner wall temperature because solar heat is absorbed at the external side of the absorber and must pass through the tube wall before reaching the fluid. This temperature difference represents the conductive resistance of the absorber wall and the thermal resistance between the wall and the near-wall fluid cell.

The wall temperatures are significantly higher than the bulk fluid temperature. This difference is expected in concentrated solar receivers because the absorbed solar heat is imposed at the wall, while the bulk fluid temperature represents an average over the whole flow cross-section. The difference between wall and bulk temperature is a key parameter because it directly affects heat losses to the environment. Higher wall temperature increases both convective and radiative losses, especially the radiative component because it depends on the fourth power of the absolute wall temperature.

The maximum wall temperatures do not occur exactly at the outlet. The outer wall temperature reaches its maximum around $\approx 1.30\text{ m}$, while the inner wall temperature reaches its maximum around $x \approx 1.33\text{ m}$. This behavior is consistent with the cosine-centered absorbed-flux distribution used in the baseline case. In this flux model, the absorbed heat is stronger near the central region of the receiver, producing a local wall-temperature peak before the outlet.

The bulk temperature, in contrast, increases monotonically along the receiver length. This is because the bulk fluid temperature represents the accumulated effect of heat addition. Even if the local heat input decreases slightly near the outlet, the fluid continues to gain energy, so the bulk temperature remains increasing.

This figure is important because it confirms that the model correctly separates three thermal levels: the outer absorber wall, the inner absorber wall, and the bulk fluid. The ordering

$T_{wo} > T_{wi} > T_b$ is physically consistent and confirms the correct direction of heat transfer from the external absorber surface toward the internal fluid.

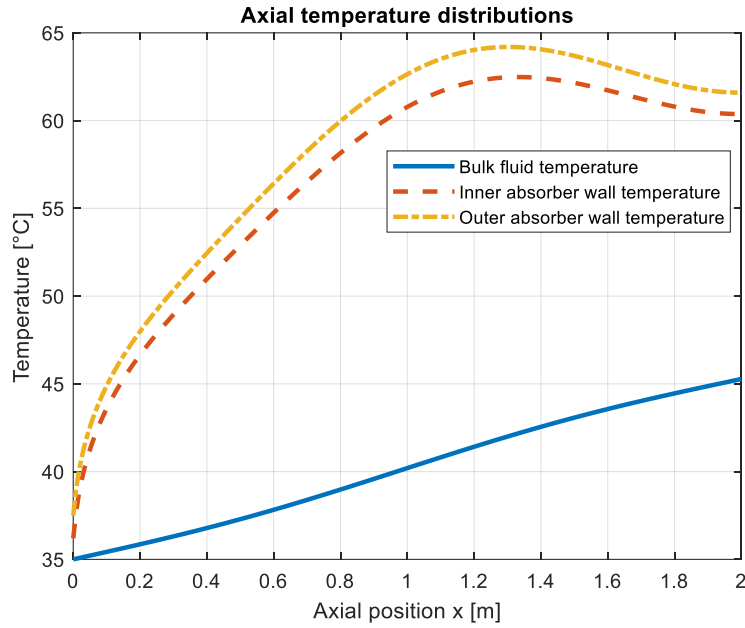


Figure III.2. Axial distributions of bulk fluid temperature, inner wall temperature, and outer wall temperature.

3.2.3 Radial Temperature Profiles

Figure III.3 presents the radial temperature profiles at three axial positions: the inlet, the middle of the receiver, and the outlet. At the inlet, the temperature is uniform and equal to 35°C , which confirms the correct application of the inlet boundary condition.

At the middle section, the temperature increases from approximately 37.06°C near the centerline to approximately 59.02°C near the wall. This large radial difference indicates that the wall receives intense heat input and that radial diffusion requires a finite distance to distribute this heat toward the core region.

At the outlet section, the temperature varies from approximately 42.56°C near the centerline to approximately 59.25°C near the wall. Compared with the middle section, the core temperature is significantly higher at the outlet. This is expected because the fluid core has had more axial distance to receive heat by radial diffusion from the wall-heated region.

The wall-side temperature at the outlet remains close to the wall-side temperature at the middle section. This indicates that the wall region reaches a thermally developed behavior more rapidly than the core region. The core temperature continues to rise downstream, while the near-

wall temperature is controlled more directly by the local balance between absorbed heat, heat transferred to the fluid, and heat lost to the environment.

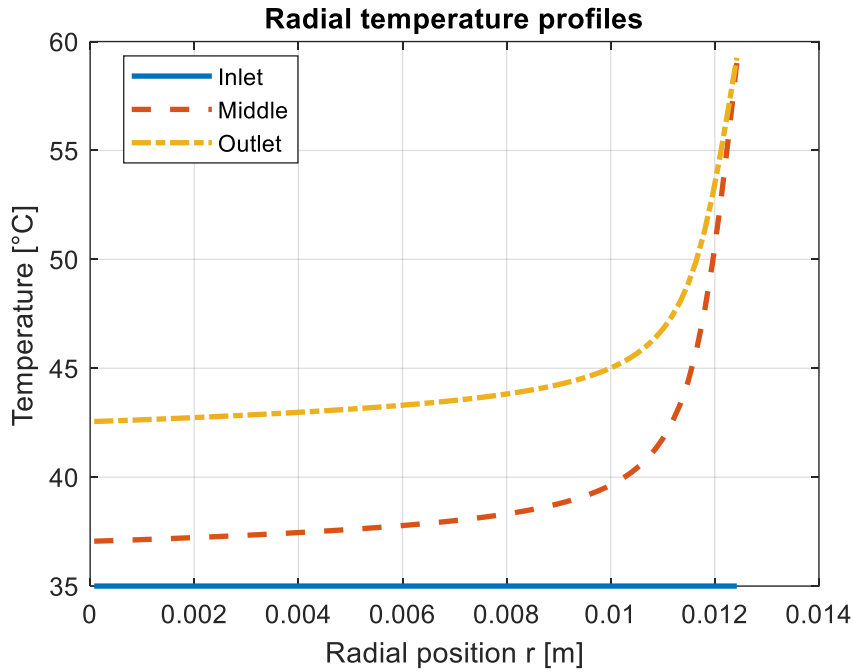


Figure III.3. Radial temperature profiles at the inlet, middle, and outlet of the receiver.

The radial profiles confirm that the receiver cannot be accurately represented by a single uniform cross-sectional temperature if local thermal behavior is important. The wall-to-core temperature gradient is substantial, particularly in the middle of the receiver. This justifies the use of a two-dimensional axisymmetric model rather than a purely one-dimensional energy balance.

From a numerical viewpoint, the profiles are smooth and monotonic from the centerline to the wall. This confirms that the radial discretization is stable and that the centerline symmetry condition is properly satisfied. No artificial oscillations are observed near the wall, despite the nonlinear heat-flux boundary condition.

3.3 Hydrodynamic Validation

3.3.1 Pressure-Drop Validation

Figure III.4 compares the pressure drop predicted by the numerical solver with the reference pressure-drop correlation over a wide range of Reynolds numbers. The Reynolds number varies from approximately 500 to 80 000, covering laminar and turbulent regimes.

The solver prediction and the reference correlation are superimposed over the entire Reynolds-number range. This excellent agreement indicates that the implemented Darcy-

Weisbach pressure-drop calculation and the selected friction-factor correlations are correctly integrated into the model.

At low Reynolds numbers, the pressure drop is very small. For example, at $Re \approx 500$, the pressure drop is approximately 1.01 Pa . At $Re \approx 1000$, it increases to approximately 2.02 Pa , and at $Re \approx 2000$, it reaches approximately 4.03 Pa . This nearly linear increase in the laminar region is consistent with the Hagen-Poiseuille relation, where the friction factor is inversely proportional to Reynolds number but the pressure drop remains proportional to velocity.

In the turbulent region, the pressure drop increases much more rapidly. At $Re \approx 12\,000$, the pressure drop is approximately 137.20 Pa . At $Re \approx 40\,000$, it increases to approximately 1128.22 Pa , and at $Re \approx 80\,000$, it reaches approximately 3794.87 Pa . This strong increase is physically expected because turbulent pressure losses scale strongly with flow velocity.

The exact overlap between the solver and reference curves confirms that the hydraulic part of the model is reliable. This is important because pressure drop is one of the main constraints in parabolic trough receiver operation. A higher mass flow rate may improve heat extraction and reduce wall temperature, but it also increases pumping power requirements. Therefore, accurate pressure-drop prediction is necessary for evaluating the thermal-hydraulic compromise of the receiver.

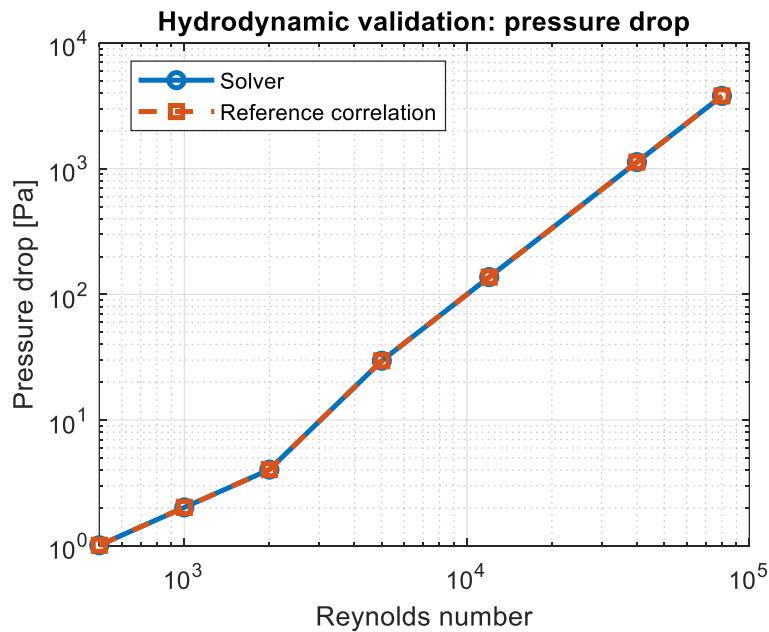


Figure III.4. Hydrodynamic validation of pressure drop as a function of Reynolds number.

The hydrodynamic validation also supports the reliability of the prescribed velocity-profile approach. Since the thermal solution depends on velocity through the axial convection term, accurate hydraulic behavior is necessary for credible prediction of outlet temperature and heat transfer.

3.4 Thermal Validation Using the Nusselt Number

3.4.1 Nusselt-Number Comparison

Figure III.5 presents the comparison between the numerical Nusselt number and the reference Nusselt number over different Reynolds numbers. The Nusselt number is a dimensionless measure of convective heat-transfer intensity. A higher Nusselt number indicates stronger heat transfer between the absorber wall and the fluid.

For laminar cases, the numerical Nusselt number is very close to the reference value. At $Re \approx 1000$, the numerical value is approximately 4.37, while the reference value is 4.364. The corresponding error is only about 0.23%. At $Re \approx 2000$, the numerical value is approximately 4.50, compared with the same laminar reference value of 4.364, giving an error of about 3.20%. These results confirm that the model correctly approaches the fully developed laminar constant-heat-flux behavior.

For turbulent cases, the Nusselt number increases significantly with Reynolds number. At $e \approx 12\,000$, the numerical Nusselt number is approximately 77.15, compared with a reference value of approximately 81.68. The error is about 5.54%. At $e \approx 40\,000$, the numerical value is approximately 212.31, while the reference value is approximately 233.07, with an error of about 8.91%. These differences remain acceptable for an engineering model using a prescribed turbulent velocity profile and an eddy-diffusivity closure.

At the highest Reynolds number, $e \approx 80\,000$, the numerical Nusselt number is approximately 323.98, while the reference value is approximately 421.37. The error increases to about 23.11%. This larger deviation is not unexpected because the numerical model uses a simplified turbulent representation based on a prescribed velocity profile and an approximate eddy thermal diffusivity. In contrast, the reference correlation is an empirical turbulent pipe-flow correlation that implicitly includes a wide range of experimentally observed turbulence effects.

Despite this deviation at very high Reynolds number, the numerical model correctly captures the main physical trend: the Nusselt number increases strongly with Reynolds number. This means that higher flow rates enhance convective heat transfer, reduce the thermal

resistance between the wall and the fluid, and improve the ability of the fluid to remove heat from the absorber wall.

The Nusselt-number validation therefore confirms that the thermal part of the model is physically consistent. The model is especially accurate in the laminar and moderate turbulent regions, while the high-Reynolds-number region should be interpreted with appropriate caution.

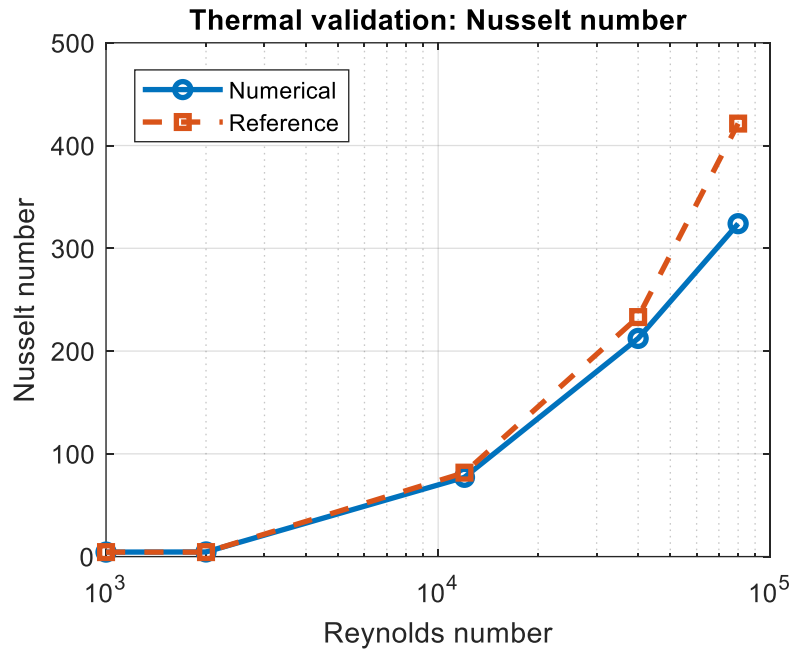


Figure III.5. Thermal validation of the Nusselt number against reference correlations.

3.5 General Discussion of the Validation Results

The validation results demonstrate that the numerical model provides a consistent representation of the receiver thermal and hydraulic behavior. The two-dimensional temperature field confirms that the solver captures axial heating and radial thermal gradients. The axial temperature profiles show the expected hierarchy between outer wall, inner wall, and bulk fluid temperatures. The radial profiles confirm that heat is transferred from the wall toward the tube core and that the inlet boundary condition is correctly imposed.

The hydrodynamic validation shows excellent agreement between the solver and the reference pressure-drop correlation. This confirms the correct implementation of the Reynolds-number-based friction factor and the Darcy-Weisbach pressure-drop equation. The agreement is important because hydraulic losses directly affect the evaluation of pumping requirements and operating feasibility.

The Nusselt-number validation confirms that the convective heat-transfer behavior is physically reasonable. The numerical results reproduce the laminar reference value with very

small error and follow the increasing turbulent trend with Reynolds number. The higher deviation at $Re \approx 80\,000$ is acceptable within the context of the simplified turbulent closure and does not invalidate the model for the receiver-level parametric analysis.

Overall, the validation confirms that the numerical solver can be used with confidence to analyze the parabolic trough collector receiver under El Oued operating conditions. The model conserves the main physical behavior of the system, predicts stable thermal fields, reproduces hydraulic pressure drop accurately, and provides realistic convective heat-transfer trends.

3.6 Parametric Study Results and Discussion

After validating the numerical model, a parametric study was performed to evaluate the influence of the main operating, geometrical, climatic, and numerical parameters on the performance of the parabolic trough collector receiver. The parametric campaign was based on a one-factor-at-a-time approach, in which one parameter was varied while the remaining parameters were kept fixed at their baseline values. The baseline conditions used in the MATLAB model correspond to El Oued-type operating conditions, with $DNI = 900\text{ W/m}^2$, $T_{amb} = 42^\circ\text{C}$, wind speed of 2 m/s , inlet temperature of 35°C , mass flow rate of 0.04 kg/s , receiver length of 2 m , aperture width of 1.5 m , and inner receiver diameter of 0.025 m . The numerical campaign also included grid-independence analysis, OFAT parametric results, sensitivity analysis, and selected two-parameter interaction maps.

The objective of this section is to discuss the physical meaning of the obtained trends and to identify the parameters that most strongly influence useful heat gain, thermal efficiency, outlet temperature, heat losses, and pressure drop.

3.6.1 Grid-Independence Analysis Based on Thermal Efficiency

Figure III.6 presents the variation of thermal efficiency with the number of axial nodes N_x . The investigated grids range from $N_x = 80$ to $N_x = 320$, with the corresponding radial refinement also increased in the grid-independence study.

The thermal efficiency decreases very slightly from approximately 62.013% at $N_x = 80$ to approximately 61.903% at $N_x = 320$. The total variation over the whole grid-refinement range is only about 0.11 percentage points. More importantly, the difference between the grids $N_x = 240$ and $N_x = 320$ is very small, decreasing from about 61.916% to 61.903%. This represents a relative variation of approximately 0.02%, which is negligible for engineering interpretation.

This result confirms that the numerical solution is practically grid independent. The small decrease in thermal efficiency with mesh refinement can be attributed to a more accurate resolution of the radial temperature gradient and wall heat-transfer behavior. Coarser grids may slightly overestimate the heat transferred to the fluid because they do not capture the near-wall thermal gradient with the same accuracy as finer grids.

The convergence of the thermal efficiency indicates that the selected baseline grid is sufficiently accurate. Therefore, the grid of approximately 240 axial nodes and 75 radial cells provides a good compromise between numerical accuracy and computational cost. This is important because the parametric study requires many repeated simulations, and using an unnecessarily fine mesh would increase computation time without significantly improving the reliability of the results.

In conclusion, Figure III.6 confirms that the numerical results discussed in the following sections are not affected by grid dependency.

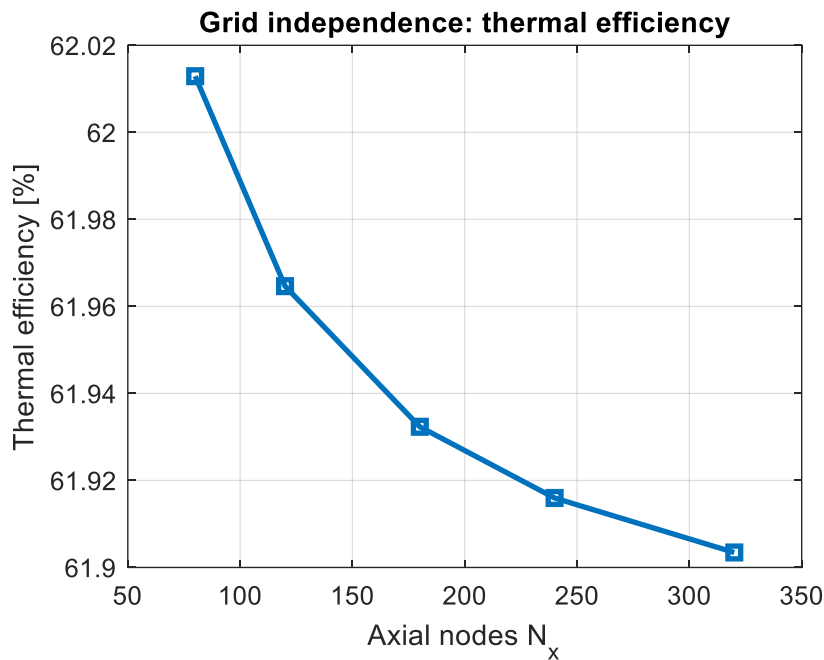


Figure III.6. Grid independence: thermal efficiency as a function of axial nodes.

3.6.2 Effect of Aperture Width on Useful Heat Gain

Figure III.7 shows the effect of the collector aperture width W_{ap} on the useful heat gain Q_u . The aperture width was varied from 1.0 m to 3.0 m. The useful heat gain increases from approximately 1121 W at $W_{ap} = 1.0$ m to approximately 3312 W at $W_{ap} = 3.0$ m.

This increase is physically expected because the aperture width directly determines the amount of solar radiation intercepted by the collector. When the aperture width increases, the aperture area increases, and therefore the incident solar power available for concentration on the absorber also increases. Since the absorbed heat rate is proportional to $DNI \times W_{ap}$, a larger aperture width produces a higher absorbed heat input and consequently a higher useful heat gain.

The increase in useful heat is almost linear. For example, increasing the aperture width from 1.0 m to 1.5 m increases useful heat from about 1121 W to 1672 W. Increasing it further to 2.0 m, 2.5 m, and 3.0 m raises the useful heat to about 2220 W, 2767 W, and 3312 W, respectively.

However, the useful heat gain does not increase perfectly proportionally. A slight deviation from perfect linearity is observed at larger aperture widths. This is mainly due to the increase in absorber wall temperature when more solar power is concentrated on the receiver. A higher wall temperature increases convective and radiative thermal losses to the environment. Therefore, although a wider aperture increases the absorbed solar power, part of the additional energy is lost before being transferred to the fluid.

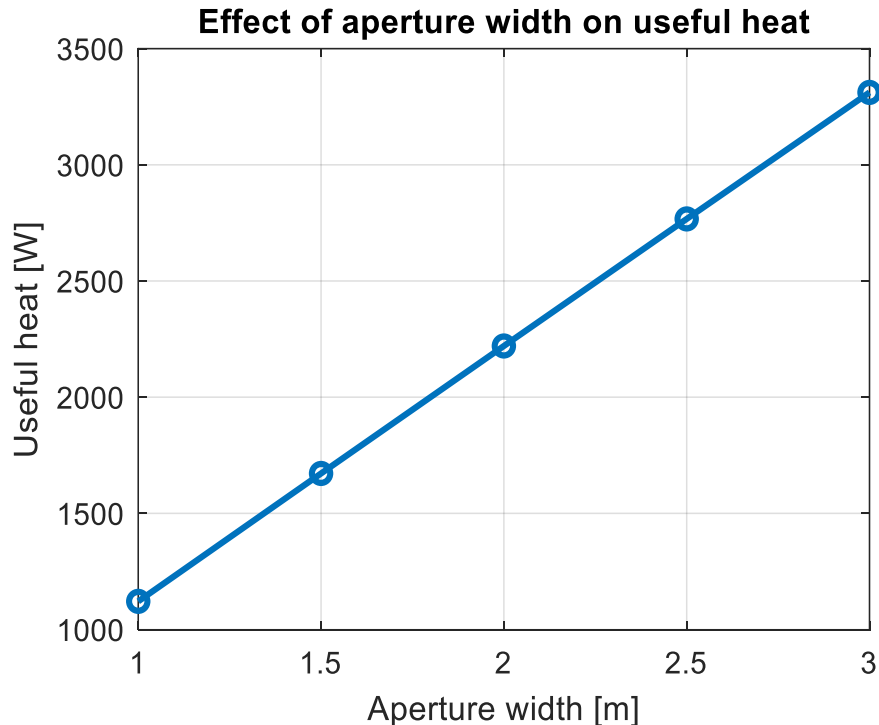


Figure III.7. Effect of aperture width on useful heat gain.

From an engineering viewpoint, the aperture width is one of the most influential geometrical parameters for thermal production. Increasing the aperture width is an effective

way to increase useful heat gain, but it must be accompanied by adequate receiver cooling and acceptable wall-temperature control. Otherwise, excessive aperture enlargement may increase thermal losses and thermal stress risk.

3.6.3 Effect of Wind Speed on Thermal Loss

Figure III.8 illustrates the influence of wind speed on the thermal loss from the absorber receiver. The wind speed was varied from 0.5 m/s to 6 m/s. The total thermal loss increases from approximately 89.67 W at 0.5 m/s to approximately 189.22 W at 6 m/s.

This result confirms the strong role of wind speed in external convective losses. In the numerical model, the wind convective heat-transfer coefficient increases with wind speed. Therefore, increasing wind speed intensifies heat exchange between the hot absorber surface and the surrounding ambient air. As a result, a larger fraction of the absorbed solar energy is rejected to the environment instead of being transferred to the heat-transfer fluid.

The increase in thermal loss is significant. When wind speed increases from 0.5 m/s to 2 m/s, the thermal loss increases from about 89.67 W to about 118.24 W. When wind speed reaches 6 m/s, the loss becomes about 189.22 W. This corresponds to more than a doubling of thermal loss over the tested range.

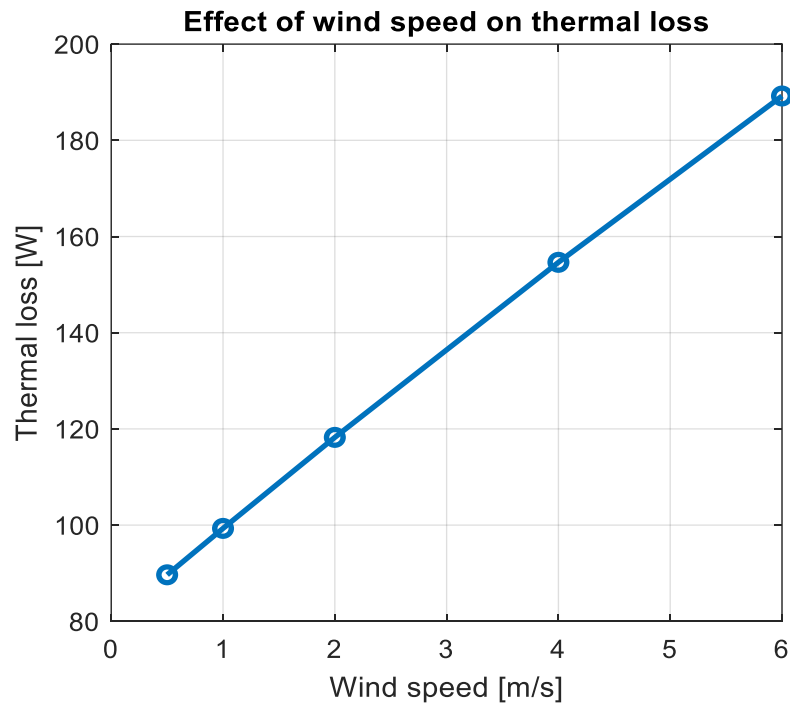


Figure III.8. Effect of wind speed on thermal loss.

The curve also shows that the effect of wind speed remains important even at moderate operating temperatures. This is particularly relevant for arid regions such as El Oued, where high solar radiation is often accompanied by wind and dust transport. Under windy conditions, the receiver loses more heat by forced convection, which reduces the useful heat delivered to the fluid and can lower the overall thermal efficiency.

From a design perspective, this result highlights the importance of reducing external convective losses. Possible engineering solutions include improving receiver insulation, using evacuated glass envelopes, protecting the receiver from direct wind exposure, and maintaining selective coatings with low thermal emissivity. Wind effects must therefore be included when predicting real collector performance under desert conditions.

3.6.4 Effect of Receiver Inner Diameter on Pressure Drop

Figure III.9 presents the effect of the receiver inner diameter D_i on the pressure drop Δp . The inner diameter was varied from 0.015 m to 0.040 m. The pressure drop decreases sharply from approximately 130.15 Pa at $D_i = 0.015$ m to approximately 0.90 Pa at $D_i = 0.040$ m.

This strong reduction is physically expected because the pressure drop in internal pipe flow is highly sensitive to tube diameter. For a fixed mass flow rate, increasing the diameter increases the flow area and reduces the average fluid velocity. Since pressure drop depends strongly on velocity and diameter, even a moderate increase in diameter can produce a large decrease in hydraulic resistance.

The numerical results show that increasing the inner diameter from 0.015 m to 0.020 m reduces the pressure drop from about 130.15 Pa to 29.17 Pa. Increasing it to the baseline value 0.025 m reduces the pressure drop further to about 7.89 Pa. At 0.032 m, the pressure drop becomes about 2.19 Pa, and at 0.040 m, it reaches less than 1 Pa.

This result demonstrates that the receiver diameter is one of the dominant parameters controlling hydraulic performance. A small-diameter receiver may improve fluid velocity and internal heat-transfer coefficient, but it creates a high pressure-drop penalty. Conversely, a larger diameter reduces pressure drop and pumping requirements, but it may reduce internal convective heat transfer and increase receiver surface area.

Therefore, the receiver diameter must be selected as a compromise between thermal and hydraulic requirements. The optimal diameter is not necessarily the largest diameter, because thermal performance and wall-temperature distribution must also be considered. However, from the hydraulic point of view, larger diameters clearly reduce pressure loss and improve pumping feasibility.

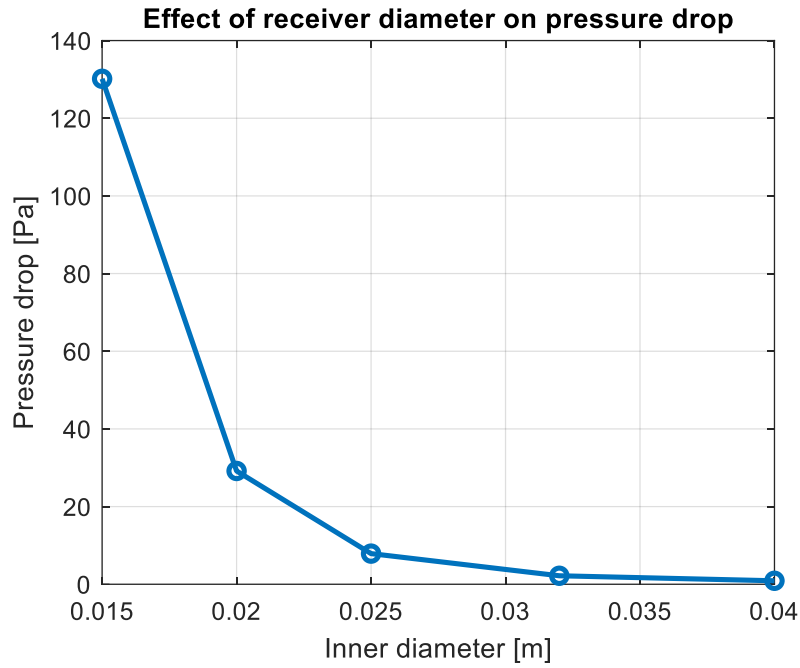


Figure III.9. Effect of receiver inner diameter on pressure drop.

3.6.5 Effect of Mass Flow Rate on Pressure Drop

Figure III.10 shows the influence of mass flow rate on pressure drop. The mass flow rate was varied from 0.01 kg/s to 0.10 kg/s . The pressure drop increases from approximately 1.47 Pa at 0.01 kg/s to approximately 57.16 Pa at 0.10 kg/s .

The trend is strongly increasing because the mass flow rate directly controls the average fluid velocity inside the absorber tube. Increasing the mass flow rate increases the Reynolds number and the dynamic pressure of the flow. As a result, the frictional pressure loss along the receiver becomes higher.

At low mass flow rates, the increase in pressure drop is moderate. For example, increasing the mass flow rate from 0.01 kg/s to 0.02 kg/s raises the pressure drop from about 1.47 Pa to 2.94 Pa . At the baseline mass flow rate of 0.04 kg/s , the pressure drop is approximately 7.89 Pa .

At higher mass flow rates, the increase becomes much stronger. The pressure drop reaches about 23.38 Pa at 0.06 kg/s , 38.68 Pa at 0.08 kg/s , and 57.16 Pa at 0.10 kg/s . This nonlinear increase is related to the transition toward stronger inertial and turbulent effects, where pressure drop rises more rapidly with velocity.

This result is important because mass flow rate is a key control variable in solar thermal systems. Increasing mass flow rate can improve heat extraction and reduce absorber wall

temperature, but it also increases pumping power. Therefore, the best operating mass flow rate must be selected by balancing thermal benefit against hydraulic cost.

The baseline value of 0.04 kg/s appears to provide a reasonable compromise because it gives moderate pressure drop while still allowing significant heat removal from the receiver.

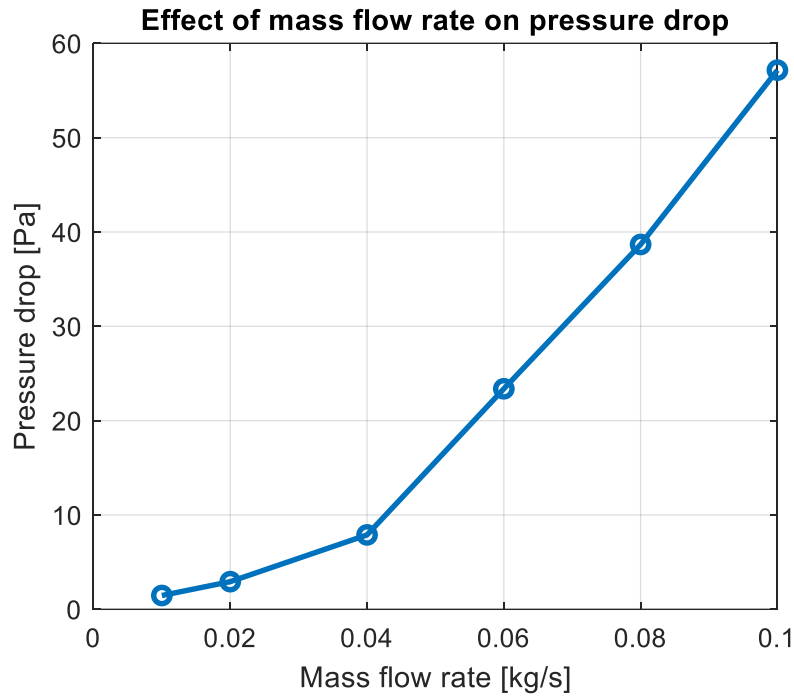


Figure III.10. *Effect of mass flow rate on pressure drop.*

3.6.6 Effect of Mass Flow Rate on Outlet Temperature

Figure III.11 presents the influence of mass flow rate on the outlet bulk temperature. The outlet temperature decreases as the mass flow rate increases. At 0.01 kg/s , the outlet temperature is approximately 69.99°C . At 0.02 kg/s , it decreases to about 53.45°C . At the baseline mass flow rate of 0.04 kg/s , the outlet temperature is approximately 45.00°C . At 0.10 kg/s , it decreases further to about 39.21°C .

This trend is physically consistent with the energy balance of the flowing fluid. For a given absorbed solar power, the temperature rise is inversely related to the mass flow rate. When the mass flow rate is low, the same amount of heat is distributed over a smaller amount of fluid, producing a higher outlet temperature. When the mass flow rate is high, the heat is distributed over a larger amount of fluid, so the temperature rise becomes smaller.

The temperature rise above the inlet temperature decreases significantly over the tested range. At 0.01 kg/s , the temperature rise is about 35°C . At 0.04 kg/s , it is approximately 10°C . At 0.10 kg/s , it is only about 4.2°C . This shows that low flow rates are favorable when

the objective is to obtain a high outlet temperature, whereas high flow rates are favorable when the objective is to improve cooling and reduce wall temperature.

However, a high outlet temperature does not necessarily mean better overall performance. At very low mass flow rate, the residence time is longer, but the absorber wall temperature may become higher, increasing thermal losses and thermal stress risk. At very high mass flow rate, the outlet temperature becomes lower, but the heat extraction process is more stable and the wall temperature is reduced. Therefore, the choice of mass flow rate depends on the required application: hot-water production may favor higher outlet temperature, while receiver protection and thermal efficiency may favor larger flow rates.

When Figures III.10 and III.11 are considered together, they show the fundamental thermal-hydraulic compromise. Increasing mass flow rate lowers outlet temperature but increases pressure drop. Therefore, the mass flow rate should not be selected from outlet temperature alone. It must be optimized using useful heat gain, pressure drop, wall temperature, and efficiency together.

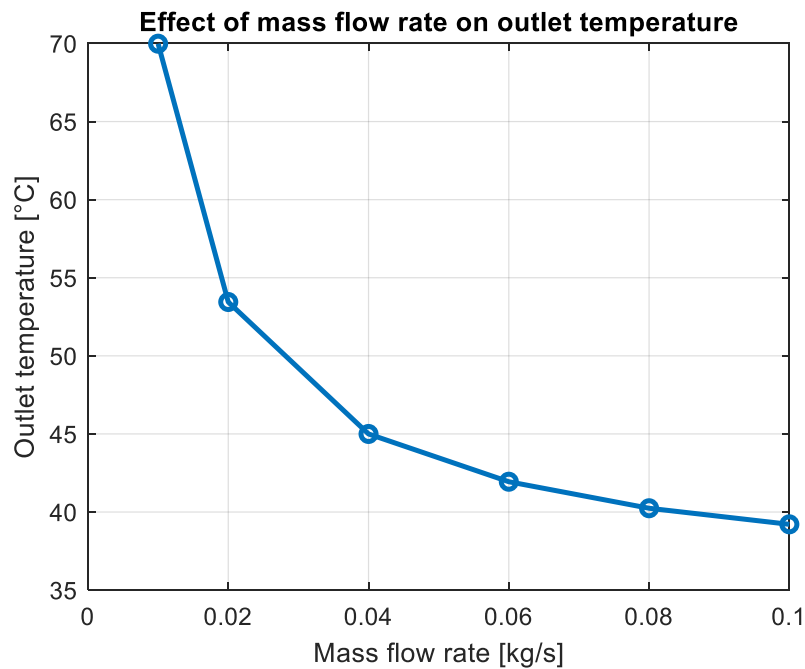


Figure III.11. Effect of mass flow rate on outlet temperature.

3.6.7 Effect of Direct Normal Irradiance on Useful Heat Gain

Figure III.12 shows the effect of direct normal irradiance DNI on useful heat gain. The DNI was varied from 500 W/m^2 to 1100 W/m^2 . The useful heat gain increases from approximately 937.38 W at 500 W/m^2 to approximately 2037.71 W at 1100 W/m^2 .

The trend is almost linear because the absorbed solar heat is directly proportional to the incident direct normal irradiance. Higher DNI means that more beam solar radiation is available for concentration by the parabolic reflector. Consequently, the absorber receives more heat, and the fluid extracts a larger useful thermal power.

At $DNI = 650 \text{ W/m}^2$, the useful heat gain is approximately 1213.12 W . At $DNI = 800 \text{ W/m}^2$, it increases to about 1488.44 W . At the baseline condition $DNI = 900 \text{ W/m}^2$, the useful heat gain is approximately 1671.73 W . When the irradiance reaches 1100 W/m^2 , the useful heat gain exceeds 2 kW .

The slight deviation from perfect linearity is caused by thermal losses. As DNI increases, the absorber wall temperature also increases. This raises convective and radiative heat losses, especially radiation because it depends on the fourth power of the absolute wall temperature. Therefore, although higher solar irradiance increases useful heat, a small part of the additional absorbed energy is lost to the environment.

This result confirms that solar irradiance is one of the most important climatic parameters affecting parabolic trough collector performance. Under the arid conditions of El Oued, where high solar radiation is available, the collector has strong potential for thermal energy production. However, high irradiance should be accompanied by adequate mass flow rate and receiver thermal management to avoid excessive wall temperatures.

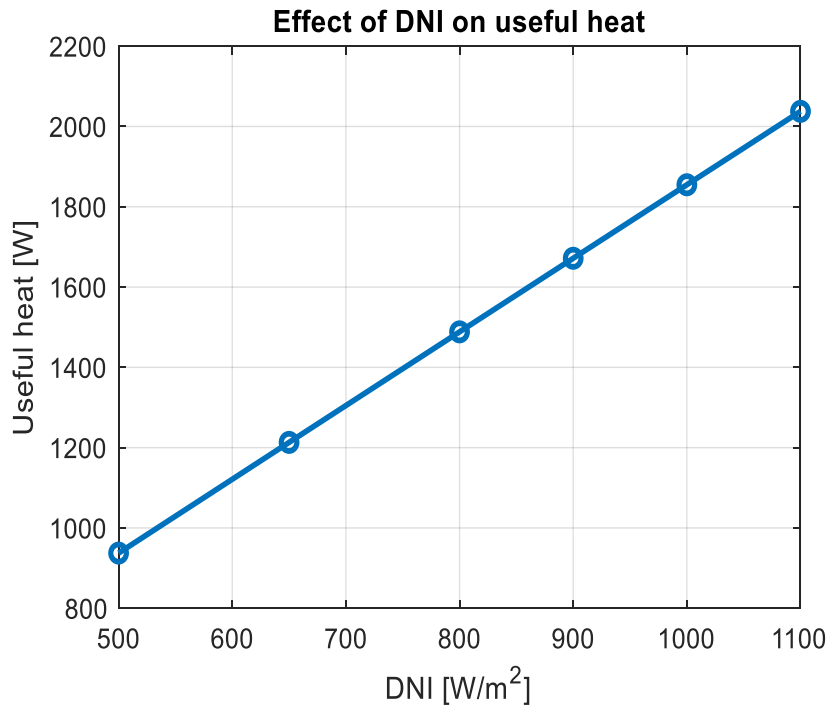


Figure III.12. Effect of direct normal irradiance on useful heat gain.

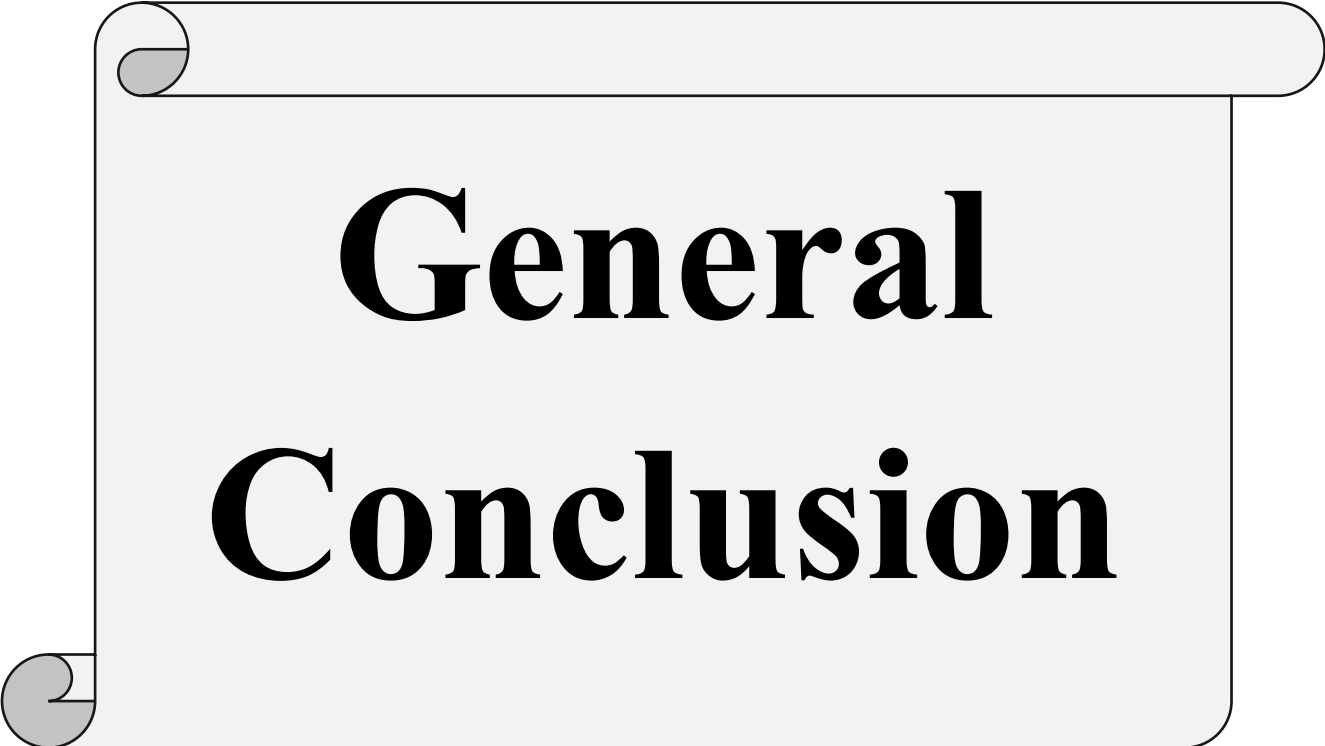
3.7 Comparative Interpretation of the Parametric Results

The parametric results show that different parameters affect the receiver in different ways. The aperture width and direct normal irradiance mainly control the solar energy input. Increasing either parameter increases the useful heat gain because more solar power is intercepted and absorbed by the receiver.

Wind speed, in contrast, affects the loss side of the energy balance. Increasing wind speed does not increase the available solar energy; instead, it increases convective heat rejection from the absorber to the environment. This reduces the fraction of absorbed heat that can be transferred to the fluid.

The receiver inner diameter and mass flow rate mainly control the hydraulic behavior. Increasing the inner diameter strongly reduces pressure drop, while increasing mass flow rate strongly increases pressure drop. Therefore, hydraulic optimization requires careful selection of both diameter and flow rate.

The mass flow rate has a dual effect. It reduces outlet temperature because the same heat input is distributed over a larger mass of fluid. At the same time, it improves heat removal from the absorber and can reduce wall temperature. However, it also increases pressure drop. This confirms that mass flow rate is not only a thermal variable but also a hydraulic control parameter.



General Conclusion

General Conclusion

This dissertation presented a numerical investigation of a parabolic trough collector receiver for thermal energy production under the arid climatic conditions of El Oued, Algeria. The main objective was to evaluate the thermal and hydraulic behavior of the receiver using a CFD-based numerical model, with particular attention to temperature distribution, useful heat gain, heat losses, outlet temperature, pressure drop, and thermal efficiency. The study was motivated by the high solar potential of El Oued and by the need to adapt concentrated solar thermal technologies to harsh desert environments characterized by high direct normal irradiance, elevated ambient temperature, wind exposure, and dust-prone conditions.

The developed model allowed the prediction of the two-dimensional temperature field inside the receiver and the evaluation of the influence of the most important geometrical, operating, and climatic parameters. The grid-independence study confirmed the reliability of the numerical solution. The thermal efficiency varied only from about 62.013% to 61.903% when the axial grid was refined from 80 to 320 nodes, with a negligible variation of about 0.02% between 240 and 320 axial nodes. Therefore, the selected grid of approximately 240 axial nodes and 75 radial cells was considered sufficiently accurate while maintaining a reasonable computational cost.

Under the baseline operating conditions representative of El Oued, namely $\text{DNI} = 900 \text{ W/m}^2$, ambient temperature = $42 \text{ }^\circ\text{C}$, wind speed = 2 m/s , inlet temperature = $35 \text{ }^\circ\text{C}$, mass flow rate = 0.04 kg/s , receiver length = 2 m , aperture width = 1.5 m , and inner receiver diameter = 0.025 m , the model predicted a thermal efficiency of approximately 61.9%, a useful heat gain of about 1671.73 W , an outlet temperature of approximately $45.00 \text{ }^\circ\text{C}$, and a pressure drop of about 7.89 Pa . These values confirm that the parabolic trough receiver can provide useful thermal output under desert conditions, provided that the receiver geometry and flow conditions are properly selected.

The parametric study showed that aperture width and direct normal irradiance are the dominant parameters controlling the useful heat gain. When the aperture width increased from 1.0 m to 3.0 m , the useful heat gain increased almost linearly from about 1121 W to 3312 W . Similarly, increasing DNI from 500 W/m^2 to 1100 W/m^2 increased the useful heat gain from

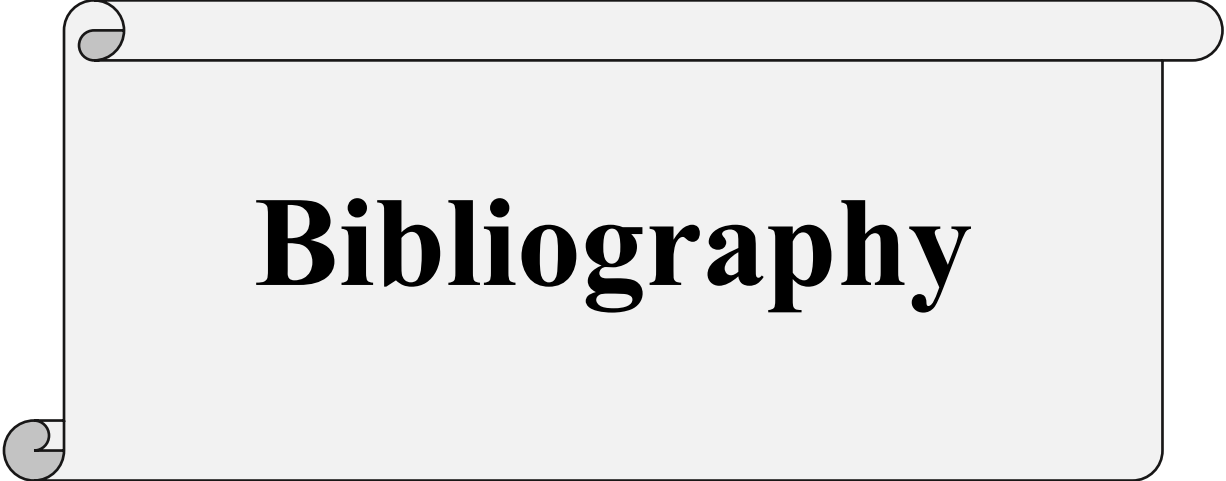
approximately 937.38 W to 2037.71 W. These results confirm that El Oued's high solar resource is highly favorable for parabolic trough collector applications.

Wind speed had a negative effect on receiver performance because it increased external convective heat losses. When wind speed increased from 0.5 m/s to 6 m/s, total thermal losses increased from about 89.67 W to 189.22 W, more than doubling over the studied range. This result demonstrates the importance of accounting for wind effects in the design and operation of PTC systems in arid regions, especially where strong wind may also transport dust and accelerate optical degradation.

The hydraulic analysis confirmed that receiver diameter and mass flow rate strongly affect pressure drop. Increasing the inner receiver diameter from 0.015 m to 0.040 m reduced the pressure drop from about 130.15 Pa to less than 1 Pa, showing that larger diameters reduce pumping requirements. Conversely, increasing the mass flow rate from 0.01 kg/s to 0.10 kg/s increased pressure drop from 1.47 Pa to 57.16 Pa. At the same time, the outlet temperature decreased from 69.99 °C to 39.21 °C over the same mass-flow range. Therefore, mass flow rate represents a key thermal-hydraulic compromise: low flow rates increase outlet temperature but may raise wall temperature and heat losses, whereas high flow rates improve heat removal but increase pumping power and reduce outlet temperature.

Overall, the results demonstrate that the performance of a parabolic trough collector in El Oued is governed by the coupled interaction of solar input, receiver geometry, external heat losses, and fluid-flow conditions. The best operating strategy requires balancing high useful heat gain, acceptable outlet temperature, low pressure drop, and limited heat losses. The baseline configuration studied in this work provides a reasonable compromise between thermal performance and hydraulic behavior.

Finally, this study confirms the technical potential of parabolic trough collectors for thermal energy production in the arid region of El Oued. The developed CFD-based model provides a useful tool for preliminary design, performance prediction, and parametric optimization. Future work should extend the model by including transient daily operation, variable thermophysical properties, dust and soiling effects, optical errors, glass-envelope behavior, and experimental validation under real outdoor conditions. Such improvements would allow a more complete assessment of PTC performance and support the development of efficient solar thermal systems adapted to Saharan environments.



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