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Faculté Des Sciences Et De La Technologie
Département De Génie Civil, Mécanique et Transport - CMT
Laboratoire de l'Ingénierie Mécanique, Matériaux et Structures

THÈSE

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**Contribution à l'étude des écoulements de nanofluides dans
une unité de captage d'énergie solaire intégrant un MCP**

Présentée par: **DJAAD Fethi**

le: 15/07/2026

Membres du jury	Grade	Institution	Désignation
MOSTEFA Kouider	MCA.	Université de Tissemsilt	Président
KHERRIS Sahraoui	Prof.	Université de Tissemsilt	Encadreur
MEBARKI Brahim	MCA.	Université de Bechar	Co-encadreur
MEKROUSSI Said	Prof.	Université de Tiaret	Examinateur
CHADOULI Rachid	MCA.	Université de Khemis Miliana	Examinateur
BENLEKKAM Mohamed Lamine	MCA.	Université de Tissemsilt	Examinateur



People's Democratic Republic of Algeria
Ministry of Higher Education and Scientific
Research



Tissemsilt university-Ahmed Ben
Yahiya El wancharissi

Faculty of Science and Technology
Department of Civil, Mechanical and Transportation Engineering,
Laboratory of Mechanical Engineering, Materials and Structures

THESIS

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Title

***CONTRIBUTION TO THE STUDY OF NANOFUID
FLOWS IN A SOLAR ENERGY CAPTURE UNIT
INTEGRATING A PCM***

Presented by: **DJAAD Fethi**

On: 15/07/2026

Jury members	Grade	Institution	Designation
MOSTEFA Kouider	MCA.	Tissemsilt university	President
KHERRIS Sahraoui	Prof.	Tissemsilt university	Supervisor
MEBARKI Brahim	MCA.	Bechar university	Co-supervisor
MEKROUSSI Said	Prof.	Tiaret university	Examiner
CHADOULI Rachid	MCA.	Khemis Meliana university	Examiner
BENLEKKAM Mohamed Lamine	MCA.	Tissemsilt university	Examiner

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Dedication

This work is dedicated:

To my beloved parents.

To my family, for their patience, understanding, and constant encouragement.

To my mentors and teachers, who have guided, inspired, and shaped my academic path with their wisdom and support.

To my friends and colleagues.

Djaad Felhi

Abstract

Abstract:

Thermal energy storage using phase change materials (PCMs) has emerged as a promising solution to compensate for the intermittency of solar radiation and to ensure the continuity of energy supply in solar-assisted thermal systems. However, the inherently low thermal conductivity of PCMs still represents a major scientific challenge that limits their thermal performance.

The aim of this thesis is to improve the energetic performance of a solar domestic hot water production system by employing two thermal enhancement strategies: the addition of metallic fins and the incorporation of nanoparticles.

The studied system consists of two interconnected parts. The first part concerns a flat-plate solar collector, for which a mathematical model was developed to evaluate its thermal efficiency as well as the outlet fluid temperature under the climatic conditions of the city of Tissemsilt during four representative days of the year's seasons (July, December, April, and September). Nanoparticles were integrated into the collector in order to enhance its thermal performance. The outlet temperature obtained from this part represents an essential input parameter for the second part of the system, namely a latent thermal energy storage unit based on a triplex-tube heat exchanger.

Within this heat exchanger, two enhancement techniques were simultaneously implemented: the addition of innovative metallic fins to intensify heat transfer, and the direct incorporation of nanoparticles into the RT82 phase change material. Numerical simulations carried out using ANSYS Fluent 18.1 were used to investigate the effect of the adopted strategies on the melting process of the phase change material.

The results reveal a significant improvement in heat transfer, a reduction in melting time, and better thermal uniformity, confirming that the combined use of fins and nanoparticles is an effective approach for enhancing the storage efficiency of low-temperature solar thermal systems.

Keywords:

Thermal energy storage, Phase change materials (PCMs), Flat-plate solar collector, Nanofluids, Modeling, Numerical simulation (CFD).

Résumé:

Le stockage thermique par matériaux à changement de phase (MCP) constitue aujourd'hui une solution prometteuse pour compenser l'intermittence du rayonnement solaire et garantir la continuité de la fourniture énergétique dans les systèmes thermiques assistés par énergie solaire. Toutefois, la faible conductivité thermique intrinsèque des MCP pose toujours un défi scientifique majeur qui limite leur efficacité thermique.

Le but de cette thèse est d'améliorer les performances énergétiques d'un système solaire de production d'eau chaude sanitaire en exploitant deux stratégies d'intensification thermique : l'adjonction d'ailettes métalliques et l'incorporation de nanoparticules.

Le système étudié comporte deux parties interconnectées. La première concerne un capteur solaire plan, pour lequel on a développé un modèle mathématique permettant d'évaluer son rendement ainsi que la température de sortie du fluide sous les conditions climatiques de la ville de Tissemsilt, pour quatre journées représentatives des saisons de l'année (juillet, décembre, avril et septembre). Des nanoparticules y sont intégrées pour renforcer les performances thermiques du capteur. La température ainsi obtenue au niveau de la sortie représente une donnée d'entrée essentielle pour la seconde partie du système, à savoir une unité de stockage thermique latent basée sur un échangeur de chaleur à triple tube. Au sein de cet échangeur, deux actions sont mises en œuvre simultanément: l'ajout d'ailettes métalliques innovantes pour intensifier le transfert thermique, et l'incorporation directe de nanoparticules dans le matériau à changement de phase RT82. Des simulations numériques réalisées sous ANSYS Fluent 18.1 ont permis d'étudier l'effet des stratégies adoptées sur le processus de fusion du matériau à changement de phase.

Les résultats montrent une amélioration notable des échanges de chaleur, une réduction du temps de fusion et une meilleure uniformité thermique, confirmant que la combinaison ailettes-nanoparticules constitue une approche pertinente pour accroître l'efficacité de stockage des systèmes solaires thermiques basse température.

Mots-clés : Stockage thermique, Matériaux à changement de phase (MCP), Capteur solaire plan, Nano-fluides, Modélisation, simulation numérique (CFD).

ملخص:

يُعدّ التخزين الحراري باستخدام مواد تغيير الطور (PCM) اليوم حلاً واعداً للتغلب على التقطع في الإشعاع الشمسي وضمان استمرارية تزويد الطاقة في الأنظمة الحرارية المعتمدة على الطاقة الشمسية. ومع ذلك، فإن انخفاض الموصلية الحرارية الذاتية لمواد تغيير الطور ما يزال يشكل تحدياً علمياً رئيسياً يحدّ من كفاءتها الحرارية.

تهدف هذه الأطروحة إلى تحسين الأداء الطاقوي لنظام شمسي لإنتاج الماء الساخن الصحي من خلال اعتماد استراتيجيتين لتعزيز انتقال الحرارة، تتمثلان في إضافة الزعانف المعدنية وإدماج الجسيمات النانوية.

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الكلمات المفتاحية: التخزين الحراري، مواد تغيير الطور (PCM)، المجمّع الشمسي المستوي، الموائع النانوية، النمذجة، المحاكاة العددية، الحسابات الديناميكية للسوائل.

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Nomenclature

Symbols

C	Constant reflection of the mushy zone	
H	Enthalpy	(J·kg ⁻¹)
h	Sensible enthalpy	(J·kg ⁻¹)
g	Gravity acceleration	(m·s ⁻²)
p	Pressure	(N·m ⁻²)
T	Temperature	(K)
T _∞	Apparent sky temperature	(K)
T _a	Ambient air temperature	(K)
h _{c2}	Convective heat transfer coefficient	(W·m ⁻² ·K ⁻¹)
h _{r2}	Radiative heat transfer coefficient	(W·m ⁻² ·K ⁻¹)
C _p	Specific heat	(J·kg ⁻¹ ·K ⁻¹)
H _{ref}	Reference enthalpy	(J·kg ⁻¹)
T _{ref}	Reference temperature	(K)
Q	Total absorbed solar energy	(W)
Q _a	Energy lost to the environment	(W)
Q _t	Energy required to heat the system to its operating temperature.	(W)
Q _u	Energy stored in the heat transfer fluid	(W)
E _g	Global solar irradiation received by the collector	(W·m ⁻²)

Greek symbols

ρ	Density	(kg·m ⁻³)
ρ _{nf}	Density of nanofluid	(kg·m ⁻³)
C _{p, nf}	Specific heat of nanofluid	(J·K ⁻¹ ·K ⁻¹)
β _{nf}	Thermal expansion coefficient of nanofluid	(K ⁻¹)
μ _{nf}	Dynamic viscosity of nanofluid	(kg·m ⁻¹ ·s ⁻¹)
λ _{nf}	Thermal conductivity of nanofluid	(W·m ⁻¹ ·K ⁻¹)
μ	Dynamic viscosity	(kg·m ⁻¹ ·s ⁻¹)
ν	Kinematic viscosity	(m ² ·s ⁻¹)
β	Thermal expansion coefficient	(K ⁻¹)
γ	Liquid fraction	

γ_s	The solar azimuth angle	
η	Efficiency of the solar collector	
ρ_c	Reflectivity coefficient	
T_c	Transmission coefficient	
α_c	Absorptivity coefficient	
ε_p	Infrared emissivity of the absorber	
ε_c	Infrared emissivity of the glass	
e_v	Glass thickness	(m)
e_i	Insulation thickness.	(m)
λ	Thermal conductivity	(W·m ⁻¹ ·K ⁻¹)
λ_i	Thermal conductivity of the insulation.	(W·m ⁻¹ ·K ⁻¹)
λ_v	Thermal conductivity of glass	(W·m ⁻¹ ·K ⁻¹)
ΔH	Latent heat between solid and liquid phases	(J·kg ⁻¹)
ΔT	Temperature difference	

Subscript

S	Solid
L	Liquid
in	Inlet
ref	Reference

Abbreviation

ES	Energy Storage
HTF	Heat Transfer Fluid
TES	Thermal Energy Storage
PCM	Phase Change Material
PCMs	Phase Change Materials
TTHX	Triplex Tube Heat Exchanger
LTES	Latent Thermal Energy Storage
TST	True Solar Time.
SiO ₂	Silicon dioxide
Al ₂ O ₃	Aluminum oxide
CuO	Copper oxide
Cu	Copper
Q _{latent}	The sensible and latent energy stored

C_{ps}	Specific heat of the material in the solid state
C_{pl}	Specific heat of the material in the liquid state
H_{latent}	Latent heat of fusion

General Introduction

General introduction

1. General Context and Motivation

Over the past decades, global energy consumption has continuously increased due to the combined effects of population growth and sustained industrial development. This growing energy demand has been accompanied by an almost exclusive dependence on fossil fuels (oil, natural gas, and coal), whose reserves are gradually depleting and whose extensive use is responsible for greenhouse gas emissions and atmospheric pollution.

Compared with other economic sectors, the residential sector represents a major contributor to global energy consumption, reaching approximately 1638 Mtoe in 2025, of which more than 55% is still expected to originate from fossil fuels. This energy consumption is mainly associated with space heating, domestic hot water production, air conditioning, and household appliances. Consequently, the residential sector constitutes the third largest emitter of CO₂, with nearly 7.2 Gt of CO₂ emitted during the same year, following the energy sector (electricity and heat production) and the transportation sector. These emissions are expected to continue increasing, reaching nearly 8.5 Gt by 2030 if large-scale energy efficiency measures and renewable energy integration are not significantly strengthened.

In response to these challenges, the scientific community and policymakers agree on the urgent need to profoundly transform current energy systems by promoting the use of renewable energy sources.

Among renewable energies, solar energy occupies a prominent position due to its abundance, inexhaustibility, and worldwide availability. However, its exploitation faces a major limitation: intermittency. Indeed, solar radiation is not continuously available, as it varies according to the time of day, seasons, and weather conditions. This variability necessitates the development of efficient energy storage systems capable of ensuring a stable and continuous energy supply when solar production is insufficient.

In this context, thermal energy storage (TES) technologies have attracted increasing interest. Their role is essential for improving the energy efficiency of solar systems and promoting sustainability in various sectors, particularly in residential applications. Unlike electrical storage systems (batteries), which are relatively mature technologies, thermal storage systems still face significant technological challenges. Generally, thermal storage systems exhibit lower energy density than their electrical counterparts, raising critical questions regarding their design, operational efficiency, and practical implementation.

Among the different thermal storage technologies, latent heat thermal energy storage using phase change materials (PCMs) stands out because of its high energy storage density and its ability to

store heat at a nearly constant temperature. Nevertheless, the intrinsically low thermal conductivity of PCMs remains a major scientific challenge that considerably limits their heat storage and recovery performance. This drawback slows down the melting (charging) and solidification (discharging) processes, thereby compromising the overall efficiency of thermal storage systems.

2. Objectives of the Thesis

Within the framework of this research work, the studied system is a solar thermal system intended for domestic hot water production. It consists of two interconnected main components: a flat-plate solar collector and a latent thermal energy storage unit. The solar collector absorbs solar radiation and heats a heat transfer fluid, while the storage unit, based on a phase change material, stores thermal energy for later use, particularly when solar radiation is insufficient.

The objectives of this thesis are organized around three main aspects:

- **Solar collector modeling:** Development, using Mathcad software, of a mathematical model for the flat-plate solar collector. This model is used to evaluate the collector efficiency and the outlet temperature of the heat transfer fluid under the real climatic conditions of the city of Tissemsilt for four representative days corresponding to the different seasons of the year (July, December, April, and September). Nanoparticles are incorporated into the working fluid circulating within the collector to improve its thermal performance.
- **Modeling and enhancement of the latent thermal energy storage unit:** Design of a latent thermal energy storage unit based on a triplex-tube heat exchanger filled RT82 phase change material. Within this exchanger, two thermal enhancement strategies are simultaneously implemented: the addition of innovative metallic fins and the direct incorporation of nanoparticles into the PCM itself, with the aim of enhancing heat transfer and consequently reducing the melting time.
- **Numerical simulation and validation:** Numerical simulations are carried out using ANSYS Fluent 18 software. These simulations are employed to investigate the influence of thermal enhancement strategies (fins and nanoparticles) on the melting process of the phase change material. The obtained numerical results are then validated through comparison with experimental and numerical studies available in the literature.

3. Methodology and Organization of the Thesis

The manuscript is divided into four chapters following this general introduction. The first chapter presents the general context and background of the study. It is organized into two main sections. The first section is devoted to an overview of the different thermal energy storage technologies, with particular emphasis on latent heat storage using (PCMs). The second section provides a detailed review of PCM applications, particularly in solar systems, energy-efficient

buildings, and latent thermal energy storage systems. Various methods used to improve the melting performance of PCMs are also discussed.

The second chapter presents the state of the art concerning heat transfer enhancement techniques in PCM-based thermal storage systems. Different methods used to improve the thermal conductivity of PCMs are examined, including nanoparticle incorporation, porous materials, multiple PCM techniques, heat pipes, and metallic fins with various geometries. Particular attention is devoted to fins because of their simple design, low cost, and ease of manufacturing. The importance of fin geometry on charging and discharging times is also highlighted.

The third chapter is dedicated to the development of the numerical models. First, the modeling of the solar collection system is presented, including the modeling of solar radiation incident on the flat-plate collector. Secondly, the modeling of the latent thermal energy storage unit is introduced. This unit employs RT82 paraffin wax as the PCM in a triplex-tube heat exchanger configuration incorporating innovative fin designs integrated into a domestic hot water storage tank. Appropriate boundary conditions are applied, and numerical simulations using ANSYS Fluent 18.1 are conducted to analyze the thermal behavior of the system.

The fourth chapter is devoted to the presentation and discussion of the simulation results. The objective is to analyze the influence of fin geometry (length, width, and thickness) on thermal performance. A comparison is established between systems equipped with the newly designed fins and those using conventional uniform fins. The effect of nanoparticles on the thermophysical properties of the PCM is also investigated.

Finally, the conclusions and future perspectives of the research work carried out throughout this thesis are presented at the end of the manuscript.

Chapter I

General Overview on Energy Storage and Phase Change Materials

I.1 Introduction

The increasing reliance on renewable energy resources has highlighted the importance of efficient energy storage technologies capable of overcoming the variability of energy production, particularly in solar energy systems. Thermal Energy Storage (TES) represents a practical solution for storing excess thermal energy and improving the reliability and efficiency of energy conversion systems. Among the available TES technologies, latent heat storage based on Phase Change Materials (PCMs) has attracted considerable interest because of its high energy storage density and its ability to absorb and release heat while maintaining an almost constant temperature during phase transition. This chapter introduces the basic concepts of thermal energy storage and presents the fundamental characteristics of PCMs, including their classification, operating principles, and their applications in latent heat thermal energy storage systems. It also discusses recent methods for enhancing their thermal performance, such as advanced heat transfer techniques and material modifications. Overall, the chapter sets the foundation for understanding PCM-based TES systems and prepares the reader for the detailed analysis presented in the subsequent chapters.

I.2 Energy storage systems

Energy storage devices help maintain daily, weekly, and seasonal equilibrium between the supply and demand of energy. They also contribute to lowering expenses, pollutants, and energy consumption while raising the general effectiveness of energy systems. Energy storage (ES) is an effective solution for storing variable renewable energy [1]. These systems are classified based on the type of storage material into three categories: sensible heat storage, latent heat storage, and thermochemical heat storage.

I.2.1 Sensible heat storage

Sensible heat storage is achieved by varying a storage medium's temperature without altering its phase during the storage operation. The temperature differential between the initial and final conditions, the mass of the storage medium, and its heat capacity all affect how much heat a sensible TES system can hold [2].

The stored sensible heat can be expressed as follows:

$$Q = m \cdot c_p \cdot \Delta T \quad (I.1)$$

where ΔT is the temperature difference between the initial and final conditions, c_p is the storage material's specific heat, and m is its mass.

I.2.2 Latent heat storage

The method known as thermal energy storage (TES) stores thermal energy by heating or cooling a storage medium. This energy can then be employed for power generation and other heating and cooling purposes at a later time. Buildings and industrial operations are two areas where TES systems are used. The energy that a substance absorbs or releases during phase transitions is known as latent heat.

The latent heat storage capacity is generally much higher than the sensible heat change. During the phase change process, as shown in Figure I.1, the temperature remains constant at its so-called phase change temperature [2]

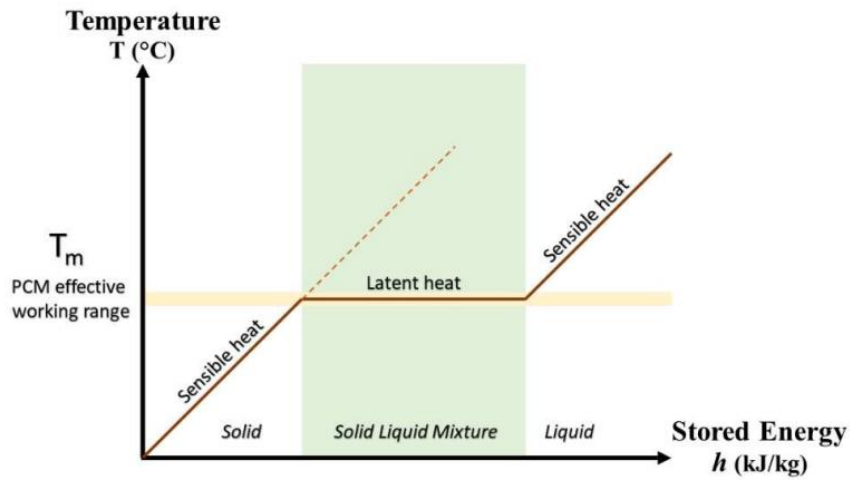


Figure I. 1 - Temperature stored thermal energy during a phase transition

Three different phase changes are possible: solid-solid, solid-liquid, and liquid gas. The solid-liquid phase change, in addition to storing large amounts of heat, is the most used phase change for TES applications because of the small volume change and consequently the small pressure change [2]. The energy density is drastically increased when (PCM) are used. Having a phase change within the temperature range of the storage ($\Delta T = T_2 - T_1$), the stored energy in a PCM can be calculated as follows:

$$Q_{\text{latent}} = \int_{T_1}^{T_{\text{pcm}}} (m \cdot c_{p,s} \cdot dT) + m \cdot H_{\text{latent}} + \int_{T_{\text{pcm}}}^{T_2} (m \cdot c_{p,l} \cdot dT) \quad (\text{I.2})$$

Table I.1 provides a comparative overview of different thermal energy storage approaches, including sensible heat storage systems based on rock beds and water tanks, as well as latent heat storage technologies employing organic and inorganic phase change materials. The comparison highlights the superior energy storage capability of latent heat systems, which can achieve a higher

storage capacity with a significantly smaller volume and lower mass of storage material for the same amount of stored energy.

Furthermore, the results presented in Table I.1 and Figure 1d indicate that inorganic phase change materials, particularly hydrated salts, generally exhibit higher volumetric energy storage densities than many organic materials. This behavior is mainly attributed to their combination of relatively high latent heat of fusion and greater material density, which enhances the amount of thermal energy stored per unit volume [3]

Table I. 1 - Comparison between the different methods of heat storage [4]

Property	Rock	Water	Organic PCM	Inorganic PCM
Density ($\text{kg}\cdot\text{m}^{-3}$)	2240	1000	800	1600
Specific heat ($\text{KJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$)	1	4.18	2	2.0
Latent heat ($\text{KJ}\cdot\text{kg}^{-1}$)	/	/	190	230
Latent heat ($\text{KJ}\cdot\text{m}^{-3}$)	/	/	152	368
Storage mass for 10^6 J (Kg)	67000	16000	5300	4350
Storage volume for 10^6 J (m^3)	30	16	6.6	2.7
Relative storage mass	15	4	1.25	1
Relative storage volume	11	6	2.5	1

I.2.3 Chemical storage method

In this chemical energy storage method, heat is stored using two types of chemical reactions, endothermic chemical reactions and exothermic chemical reactions.

These reactions are carried out under different conditions depending on the specific application. For cooling or heat-absorbing applications, endothermic reactions should be used. In this case, a chemical mixture absorbs heat energy from the surroundings, lowering the surrounding temperature. On the other hand, exothermic reactions are utilized for heating applications, where a chemical mixture reacts to release heat energy, which is then used for heating the surrounding environment [5].

There are three main methods of storing thermal energy in a material: reversible chemical energy, sensible thermal energy, and latent thermal energy, as shown in Figure I.2 Chemical energy is absorbed or released during a chemical reaction in a material, causing changes in the molecular structure. If this process is reversible, it can be used to solar collector and recover energy. An example of this is the splitting of water into its component gases, diatomic oxygen and diatomic hydrogen, and then recombining them back into water. A similar process can be applied to ammonia

in the reversible Haber process, where nitrogen and hydrogen atoms are combined and separated [6].

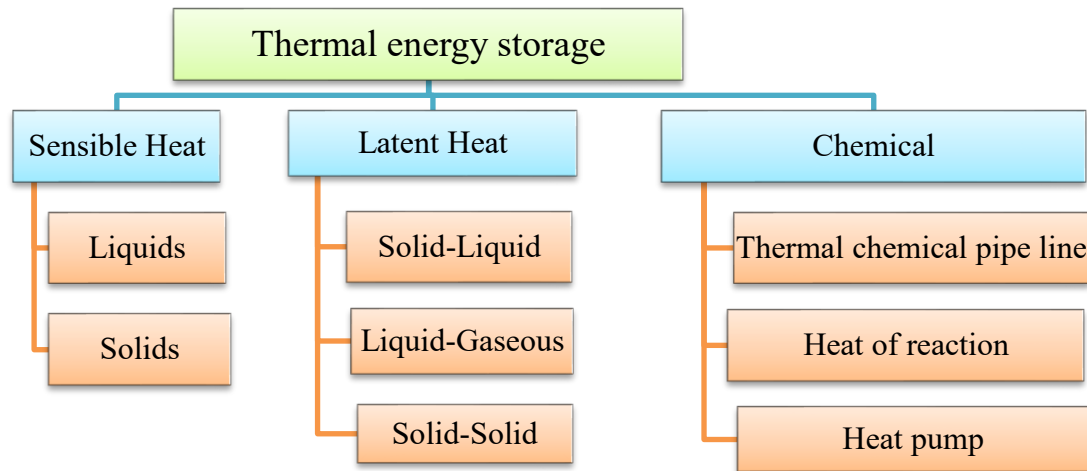


Figure I. 2 - Methods of solar energy storage

I.3 Fundamental principles of phase change materials

Phase Change Materials (PCMs) are employed to store thermal energy by absorbing and releasing large amounts of latent heat during phase transitions at a nearly constant temperature between solid and liquid, as shown in Figure I.3 [3]. PCMs are widely used in applications such as thermal energy storage, temperature regulation, and waste heat recovery, owing to their high energy storage density and their ability to maintain stable temperatures during phase transitions[7]. This process, commonly known as the melting solidification cycle, occurs within a specific temperature range determined by the thermal properties of the selected PCM [7].

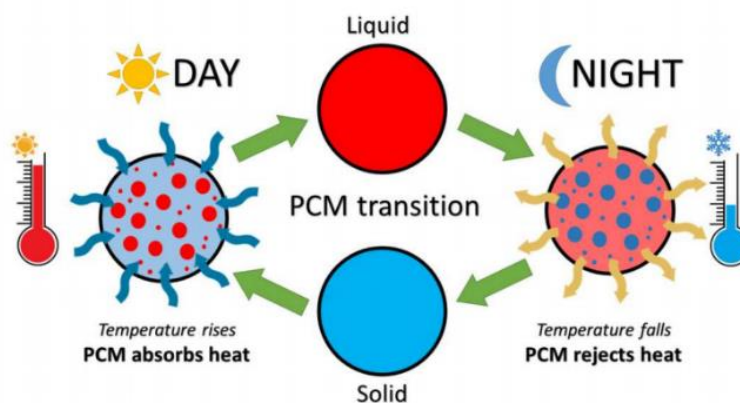


Figure I. 3 - Phase change cycle during temperature variation [8].

I.4 Melting of phase change material

Phase Change Materials absorb and store thermal energy during their phase transition from solid to liquid at a nearly constant temperature, as shown in Figure I.4. The stored latent heat is

subsequently released upon solidification, making PCMs highly suitable for thermal energy storage and temperature regulation systems. These materials are particularly advantageous for applications requiring stable thermal conditions, such as building insulation, cooling technologies, and renewable energy systems [9]. For effective thermal energy storage using PCMs, the materials must possess high latent heat and adequate thermal conductivity. Additionally, they should have a melting point within the desired operating temperature range, undergo congruent melting with minimal supercooling, and exhibit chemical and thermal stability. Other essential characteristics include low cost, non-toxicity, and non-corrosiveness. Over the past four decades, various materials have been investigated for this purpose, including hydrated salts, paraffin waxes, fatty acids, and eutectic mixtures of both organic and inorganic compounds [10].

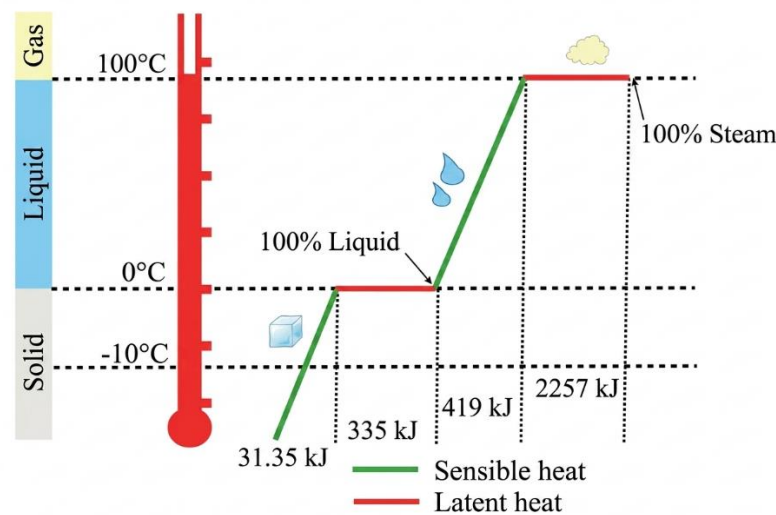


Figure I. 4 - Temperature and energy absorption curve of PCM during melting [11].

I.5 Classification of phase change materials

Phase change materials (PCMs) canister generally exist restricted inside three broad classes located on their synthetic composition: basic, non-carbon, and eutectic combinations, as pictorial in Depiction 1.5. Each class is characterized by distinct physical, chemical, and thermal properties that influence its performance during phase transitions.

In addition, PCMs are defined by specific phase change temperature ranges, which play a critical role in deciding their rightness for particular applications. The election of an appropriate PCM thus depends on various determinants, including the necessary operating hotness, strength depository capacity, warm support, and unity with the destined order. As a result, understanding these classifications is essential for optimizing the effectiveness and dependability of thermaalenérgystorage systems.

Note: PCM are utilized across a wide range of temperatures. For instance, eutectic water salt solutions are commonly used for low temperature storage (below 0 °C), whereas organic and inorganic PCMs are typically employed in medium temperature applications (ranging from 0 to 400 °C).

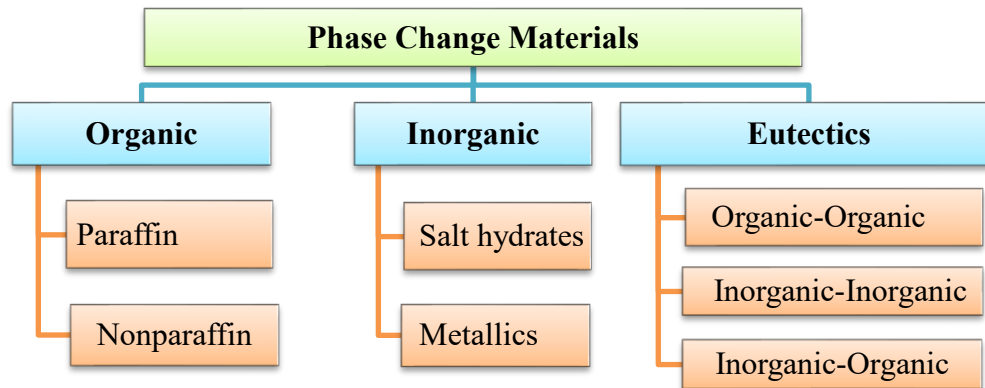


Figure I. 5 - A classification of PCMs based on their material composition

I.5.1 Organic phase change materials

Organic PCM include paraffins, which are composed of hydrocarbon chains of varying lengths, as well as non-paraffin compounds such as high molecular weight sugars, alcohols, and fatty acids. Organic PCMs are generally more expensive than inorganic PCMs, but they offer better chemical stability and corrosion resistance. Compared to inorganic PCM They exhibit relatively high latent heat of fusion, low volume change during phase transition, excellent chemical stability, and corrosion resistance. The stability of the PCM is essential to choose it for a particular application, and this can be achieved by impregnation of PCM into porous material [12].

I.5.1.1 Paraffin type organic compounds

Paraffin waxes are classified as saturated hydrocarbons, consisting of linear chains of n-alkanes with the general formula (C_nH_{2n+2}). The number of carbon atoms (n) typically varies from 14 to 40 depending on the specific type of wax. These materials exhibit a broad range of melting temperatures, generally spanning from about 6°C to 135°C. As the carbon chain length increases (i.e., with higher n values), both the phase transition temperature and the latent heat of fusion tend to rise. For example, a paraffin containing 14 carbon atoms melts at approximately 6°C and has a latent heat of fusion around 230 $\text{kJ}\cdot\text{kg}^{-1}$, whereas a 20 carbon paraffin melts near 38°C with a latent heat close to 283 $\text{kJ}\cdot\text{kg}^{-1}$. Overall, paraffin waxes are known for their relatively high energy storage capacity per unit mass, typically exceeding 200 $\text{kJ}\cdot\text{kg}^{-1}$.

Their main advantages include minimal volume change during phase transitions (a maximum of 10%) and low vapor pressure at the melting point [12].

This enhances the system stability, as well as the number of charging and discharging cycles, without significant degradation of storage capacity. Paraffins are also known for their congruent (or complete) melting and good nucleating properties, resulting in a very small subcooling effect (the effect when the material starts to release heat considerably below the melting temperature) [10].

Table I. 2 - Paraffins' thermophysical properties for possible use as PCM [4].

Material	Melting Temperature (°C)	Melting enthalpy (KJ·kg ⁻¹)	Thermal conductivity (W·m ⁻¹ ·K ⁻¹)	Density(kg·m ⁻³)
n-tetradecane C ₁₄ H ₃₀	6	230	- 0.21(solid)	760 (liquid, 20 °C) -
n-Pentadecane C ₁₅ H ₃₂	10	212	- -	770 (liquid, 20 °C) -
n-Hexadecane C ₁₆ H ₃₄	18	210,238	- 0.21(solid)	760 (liquid, 20 °C) -
n-Heptadecane C ₁₇ H ₃₆	19	240	- -	776 (liquid, 20 °C) -
n-Octadecane C ₁₈ H ₃₈	28	200,245	0,148 (liquid, 40 °C) 0,358 (Solid, 25 °C)	774(liquid, 70 °C) 814 (liquid, 20 °C)
n-Eicosane C ₂₀ H ₄₂	38	283	-	779
n-Triacontane C ₃₀ H ₆₂	66	-	-	775
n-Tetrcontane C ₄₀ H ₈₂	82	-	-	-
n-Pentacontane C ₅₀ H ₁₀₂	95	-	-	779
Polyethylene C _n H _{2n+2} n up to 10 ⁴	110-135	200	-	- 870-940 (Solid, 20°C)

I.5.1.2 Non paraffin type organic compounds

The non-paraffin category of organic Phase Change Materials encompasses a wide range of substances with diverse chemical compositions, including esters, fatty acids, alcohols, and glycols. Among these, fatty acids are of particular interest due to their well-defined chemical structure, typically expressed as CH₃(CH₂)_nCOOH, and their favorable thermal behavior. One of their main advantages lies in their ability to undergo solidification with minimal or no subcooling, which enhances their reliability in thermal storage applications. As indicated in Table 2, fatty acids exhibit latent heat values comparable to those of paraffin-based PCMs. However, their practical use is generally limited to a relatively narrow temperature range, typically between 7.8°C and 64°C [13]. Despite this limitation, they remain well suited for low-temperature thermal-energy-storage

applications. For instance, formic acid, with a melting temperature of approximately 8°C and a latent heat of 247 kJ/kg, is suitable for cold storage applications such as food preservation. Similarly, caprylic acid, with a melting point around 16.3°C and a latent heat of 149 KJ.kg⁻¹, can be effectively used in air conditioning systems[10]. Similar to paraffin-based materials, the relatively low thermal conductivity of fatty acids represents a significant limitation, which may affect the rate of heat transfer during charging and discharging processes. In addition to fatty acids, sugar alcohols also fall within the non-paraffin PCM category. These materials are characterized by a broad melting temperature range, typically between 90°C and 200°C, and exhibit high volumetric latent heat storage capacities exceeding 400 MJ/m³, primarily due to their high density, as presented in Table I.3.

Table I. 3 - Thermo-physical characteristics of fatty acids for PCM applications [10].

Material	Melting		Thermal conductivity (W·m ⁻¹ ·K ⁻¹)	Density(kg·m ⁻³)
	Temperature (°C)	enthalpy (KJ·kg ⁻¹)		
Caprylic acid CH ₃ (CH ₂) ₆ COOH	16	149	0.149 (liquid,38 °C)	901 (liquid,30 °C)
Capric acid CH ₃ (CH ₂) ₈ COOH	32	153	-	981 (Solid ,13 °C)
Lauric acid CH ₃ (CH ₂) ₁₀ COOH	42 - 44	178	0.149 (liquid,40 °C)	886 (liquid,40 °C), 1004 (Solid ,24°C)
Myristic acid CH ₃ (CH ₂) ₁₂ COOH	58	186.204	0.147 (liquid,50 °C)	870 (liquid,50 °C), 1007 (Solid ,24°C)
Palmitic acid CH ₃ (CH ₂) ₁₄ COOH	61.64	185.203	-	861 (liquid,55 °C)
			0,17 (Solid)	990 (Solid ,24°C)
			-	850 (liquid,65 °C)
			-	989 (Solid ,24°C)

Table I. 4 - Properties of sugar alcohols with potential use as a PCM [10]

Material	Melting		Thermal conductivity (W·m ⁻¹ ·K ⁻¹)	Density (kg·m ⁻³)
	Temperature (°C)	enthalpy (KJ·kg ⁻¹)		
Xylitol	94	263	-	-
C ₅ H ₇ (OH) ₅			-	1500 (Solid,20 °C)
D-Sorbitol	97	185	-	-
C ₆ H ₈ (OH) ₆			-	1520 (Solid,20 °C)
Erythritol	120	340	0,326 (liquid,140 °C), 0,733 (Solid,20 °C)	1300 (liquid,140°C)
C ₄ H ₆ (OH) ₄			-	-
D-Mannitol	167	316	-	-
C ₆ H ₈ (OH) ₆			-	1520 (Solid,20 °C)
Galactitol	188	351	-	-
C ₆ H ₈ (OH) ₆			-	1520 (Solid,20 °C)

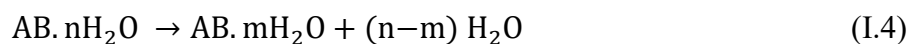
I.5.2 Inorganic phase change materials

Inorganic step change materials (PCMs) are materials usually composed of seasoning hydrates, salts, or ore compounds that enable the capture and release of heat required to raise temperature during tangible renewals, particularly betwixt dependable and liquid phases. These matters are about their native sources and enjoy very high warm generated power, high accuracy, extreme energy depository density, and sharp aspect change temperatures. PCMs are usual in proficient applications needing efficient warm strength storage, to a degree in soler capacity plants, building strength plans, and industrial waste heat improvement.

Inorganic (PCMs) are substances, typically composed of salt hydrates, salts, or metallic compounds, that absorb and release latent heat during phase transitions, primarily between solid and liquid phases. These materials are known for their high thermal conductivity, large energy storage density, and sharp phase change temperatures. Inorganic PCMs are widely used in applications requiring efficient thermal energy storage, such as solar power plants, building energy systems, and industrial waste heat recovery. [3]

I.5.2.1 Salt hydrates

Salt hydrates are the most studied PCMs. They are a mixture of inorganic salts with water forming hydrate crystals ($AB \cdot nH_2O$). At the melting temperature, hydrate crystals lose the water completely, a lower hydrate is formed according to the following formula (anhydrous form)



The melting process of salt hydrates can be either congruent, where the anhydrous salt fully melts in water, or incongruent, where only partial melting occurs due to density differences between the salt and water. Issues such as poor nucleation properties often lead to subcooling effects, which can be mitigated by adding nuucleating ageents or retaIning crystals as nuclei. [14] [10] Phase separation, caused by the settling of salt at the bottom of the container during melting, is a major challenge but can be addressed using gel-forming or thickening agents, mechanical stirring, or encapsulation techniques Studies have directed on improving the warm cycle cohesion of seasoning hydrates, w hich offer benefits such as taller thermal generated power and better energy depository density for one volume distinguished to paraffins.[15].

Table I. 5 - Thermo-physical characteristics of salt hydrates for PCM applications[14].

Material	Melting		Thermal conductivity (W·m ⁻¹ ·K ⁻¹)	Density (kg·m ⁻³)
	Temperature (°C)	Enthalpy (KJ·kg ⁻¹)		
LiClO ₃ 3H ₂ O	8	155	-	1530 (liquid) 1720 (Solid)
KF3H ₂ O	18,5	231	-	1447 (liquid, 20 °C) 1455 (Solid, 18 °C)
CaCl ₂ 6H ₂ O	29,30	171,190	0,540 (liquid, 39 °C) 1,088 (Solid, 23 °C)	1562 (liquid, 32 °C) 1710 (Solid, 25 °C)
LiNO ₃ 3H ₂ O	30	296	-	-
Na ₂ SO ₄ 10H ₂ O	32	354	0,554	1485 (Solid)
Na ₂ SO ₄ 12H ₂ O	35 – 44	280	0,476 (liquid) 0,514 (Solid)	1442 (liquid) 1522 (Solid)
Na ₂ S ₂ O ₃ 5H ₂ O	48 – 55	187,209	-	1670 (liquid) 1750 (Solid)
Na (CH ₃ COO)3H ₂ O	58	226,264	-	1280 (liquid) 1450 (Solid)
Ba (OH) ₂ 8H ₂ O	78	265,280	0,653 (liquid, 86°C) 1,255 (Solid, 23 °C)	1937(liquid, 84 °C) 2180(Solid)
Mg (NO ₃) ₂ 6H ₂ O	89,90	149,163	0,490 (liquid, 95 °C) 0,669 (Solid, 23 °C)	1550(liquid, 94 °C) 1636 (Solid, 25 °C)
MgCl ₂ 6H ₂ O	117	165,169	0,570 (liquid, 120 °C) 0,704 (Solid, 110 °C)	1450(liquid, 120°C) 1569 (Solid, 20 °C)

I.5.2.2 Metallics

For stage change applications, melting metals and ingot eutectics are often utilized materials. They are distinguished by forceful thermal conductivity and a extreme heat of melting per unit capacity. Nevertheless, these materials have only been used as PCMs in a small number of published research. The biggest obstacle to the widespread use of metallic materials seems to be their impractically high weight, despite the fact that their excellent thermal conductivity gives a substantial benefit for thermal-energy-storage requests. Until a briefer version of thermaal-énergie-storage is generated, researchers do not view this technology as a feasible substitute. [13]

I.5.3 Eutectic phase change materials

Eutectics are mixtures of two or more organic, inorganic, or organic-inorganic compounds. The combination of these materials ensures uniform melting and freezing characteristics for all components, without segregation[16]. Unlike paraffins, eutectics are characterized by distinct, sharp melting points. Melting points and latent heat values for several potential organic and inorganic eutectics are provided by Sharma et al. [17], allowing for the design of materials with tailored phase change temperatures suitable for specific applications.

I.6 Properties of phase change materials

The efficiency of (PCMs) in various applications is directly linked to their thermal, physical, kinetic, and chemical properties, as outlined below [19]. To ensure the effectiveness of these materials, the selection of an appropriate PCM must be based on the specific properties of each material, taking into account the boundary conditions and achieving a stable molecular-level balance that aligns with the requirements of the intended application.

I.6.1 Thermo-physical properties

For efficient thermal energy storage, a PCM should satisfy several essential thermophysical and operational requirements. The phase transition temperature must be compatible with the intended application to ensure effective energy storage and recovery. The material should also possess a high energy storage density, allowing a substantial amount of latent heat to be stored within a compact volume. In addition, a relatively high specific heat capacity is desirable to increase sensible heat storage before melting and after solidification. Since the overall performance of latent heat storage systems is strongly influenced by heat transfer, the PCM should exhibit sufficient thermal conductivity in both solid and liquid states to accelerate the charging and discharging processes. Good dimensional stability during phase transition, together with low vapor pressure under operating conditions, is equally important to minimize mechanical stresses and facilitate safe container design. Finally, the material should undergo repeated melting and solidification cycles without phase separation, chemical degradation, or loss of thermal storage capacity, thereby ensuring long-term reliability and durability.

I.6.2 Kinetic properties

- ✓ High heat transfer rates to accelerate charging and discharging processes.
- ✓ Efficient crystallization or melting with minimal delay.

I.6.3 Chemical properties

- ✓ Chemical stability to ensure durability over repeated use cycles.
- ✓ Non-toxicity and environmental safety for sustainable applications.
- ✓ Compatibility with other components in the thermal system.

These properties are essential to achieving optimal performance and efficiency in thermal energy storage systems utilizing PCMs.

I.7 Applications of phase change materials

Phase Change Materials (PCMs) are widely used to enhance thermal energy storage efficiency and regulate temperature across a broad range of applications. By absorbing and releasing latent heat during phase transitions, they contribute significantly to improving energy system performance, increasing building efficiency, enhancing thermal comfort, and protecting temperature-sensitive products. PCMs offer effective and sustainable thermal management solutions that align with the demands of various industrial and engineering sectors.

I.7.1 Application in building

Thermal comfort in the building sector can be significantly enhanced by integrating (PCMs) into walls, ceilings, and floors, as they help stabilize indoor temperatures and reduce the demand for active heating and cooling systems.

In this context, PCMs are often combined with conventional thermal insulation materials within building envelopes to minimize heat transfer. Numerical studies indicate that, in approximately 71.4% of cases, traditional insulation materials outperform PCMs in reducing heat exchange. Moreover, it is generally not recommended to install insulation or PCM layers too close to interior surfaces, as this may negatively affect thermal performance [20].

Among the different components of a building, façades are the most common location for PCM applications due to their greater exposure to solar radiation. Accordingly, a study was carried out to assess the thermal performance of façade systems enhanced with PCMs.

Incorporating (PCMs) into building façades offers a promising approach to enhancing energy efficiency by moderating heat transfer caused by extended exposure to environmental conditions, as illustrated in Figure I.6. To improve durability and prevent leakage, PCMs are typically encapsulated and integrated with compatible polymeric materials.

Ensuring optimal, long-term performance requires selecting PCMs with suitable melting points and verifying their chemical compatibility with the encapsulating agents. As a result, the inclusion of PCMs in façade systems plays a vital role in lowering energy consumption and advancing the thermal sustainability of buildings.



Figure I. 6 - Model of the building's façade incorporating a PCM

Also, applications of (PCMs) in buildings have expanded the scope of research by examining the impact of their placement and effectiveness in different climatic conditions. For example, in a Mediterranean climate, Mark and Kul demonstrated the effectiveness of PCMs in reducing cooling demands and improving the overall energy efficiency of a commercial building, as shown in Figure I.7. As a result, simulation and modeling studies have shown that integrating PCMs into building envelopes can lead to a reduction in heating loads by up to 40% and cooling loads by approximately 15% [21].



Figure I. 7 - Lightweight commercial building model incorporating PCM.

I.7.2 Application in medical

The use of phase change materials (PCMs) in medical applications covers a broad spectrum within the healthcare sector, particularly those involving direct interaction with the human body. Common examples include heat and cold therapies such as heat cushions used in physiotherapy and rheumatism treatment, blankets designed to prevent hypothermia, and cooling packs applied post-

surgery, after accidents, or following intense physical activity. Additionally, devices like incubators, warmers, and coolers play a crucial role in neonatal care by maintaining a stable thermal environment essential for the health and development of newborns. Another important application of PCMs is in enhancing the comfort of orthoses and prostheses, where they contribute to reducing perspiration at the contact interface between the limb and the device [4]. While heat and cold therapy represent a prominent medical use of thermoregulation, neonatal care stands as a highly specialized and distinct field, characterized by unique physiological and technical demands, thereby forming a separate and specialized segment within the healthcare market, as shown in Figure I.8.



Cooling vest

Heat cushions

Therapeutic devices of neonates

Figure I. 8 - Medical applications of heating and cooling using PCM.

I.7.3 Applications in transport and storage containers

In transportation applications and logistics, PCMs are extensively used in cold chain systems to maintain stable temperatures for perishable goods like pharmaceuticals, ensuring their quality during transit. Insulated transport containers often incorporate PCMs, which absorb or release heat during phase changes to keep the internal temperature stable for hours to days. This enables independent temperature control without relying on external cooling or heating, making PCMs especially needed in this field such as airplane cargo bays. Figure I.9 shows a few examples



Figure I. 9 - Packaging systems for storing and transporting heat-sensitive materials: bag, EcoFlex 96, and va-Q-tainer for large volumes.

Transporting food products from production sites to the market using trucks equipped with sheet metal containers filled with PCMs, as shown in Figure I.10 , is an effective way to regulate food temperature during transport. By maintaining a nearly constant temperature for the food throughout transportation, these systems minimize fuel consumption and offer significant advantages in terms of product quality and energy efficiency[22].

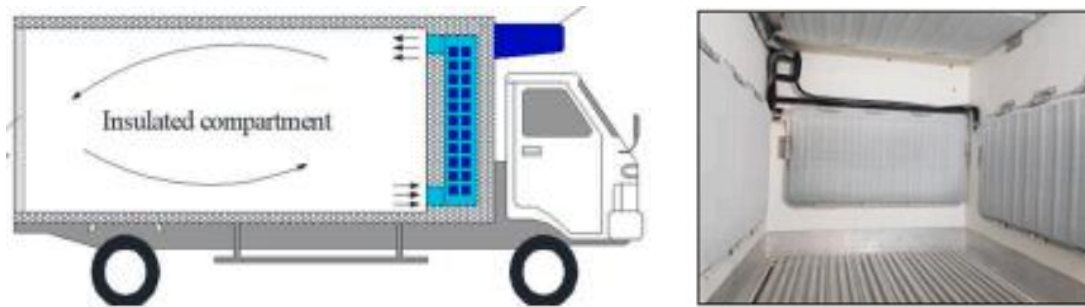


Figure I. 10 - Refrigerated truck containers containing PCM-filled sheet metal for food transportation

I.7.4 Applications of PCMs in Microelectronics

The PCMs having chalcogenide glasses are used as nonvolatile memory devices [23]. They are called nonvolatile because they can retain information even when electric power is not applied . They are also applied for cooling and thermal management of micro sized circuits.

I.7.5 Applications of PCM in Electric Vehicles

Phase change materials (PCMs) have been incorporated into electric vehicle (EV) applications due to their ability to regulate battery temperature. In addition, they help control cabin temperature, thereby reducing energy consumption and extending driving range. Moreover, PCMs and battery thermal management systems (BTMS) work in tandem to enhance thermal regulation in EVs, ensuring optimal performance and safety, as shown in Figure I.11 BTMS maintains the battery temperature within an optimal range and ensures uniform temperature distribution throughout the battery module. It also contributes to extending battery lifespan [13]. Furthermore, BTMS enables battery operation under various environmental conditions with different ambient temperatures.

In this context, several studies have explored the integration of PCMs into BTMS to further improve thermal regulation and battery performance under varying operating conditions. Javani et al. [24], numerically investigated the heat transfer performance of PCMs in battery thermal management for electric vehicles. They reported that the use of PCMs effectively reduced the maximum temperature of the battery, as shown in Figure I.12 .

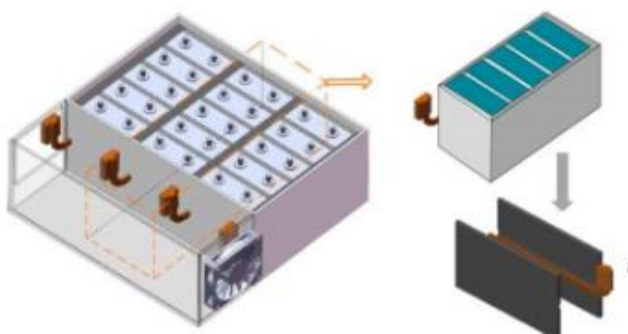


Figure I. 11 - Battery pack with heat pipe-assisted PCM.

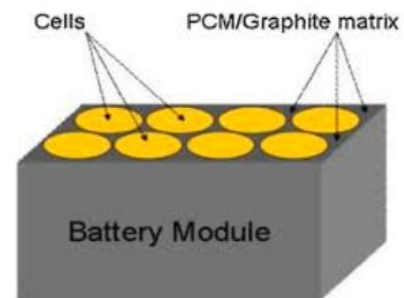


Figure I. 12 - Battery module schematic diagram using PCM/graphite

I.7.6 Applications of PCM in Cold and Thermal Storage

Storage solutions allow keeping goods in the required temperature range while being stored at a given location. To keep the temperature, a good insulation is used to reduce heat exchange with the ambient. In addition, heat or cold can be supplied by a heating or cooling device, by a pre-charged PCM, or by a combination of both. The independent transport containers discussed before use pre-charged PCM heating or cooling device as in transport applications a power connection is usually missing. Nevertheless, they can of course also be used for storage. The focus now is on storage solutions with a field location where a power connection is available or can be made available. The advantage of using PCM to shift or buffer heating or cooling loads is typically an increase in the energetic efficiency, and the option to have a backup in case of power failure.

Two nice examples are a freezer/fridge and a cold room by PLUSS Advanced Technologies Pvt. Ltd. Conventional chest coolers and freezers are available in the variations of 2 to 8 °C (cooling) and -24 to -18 °C (freezing), as shown in Figure I.13 The PCM pouches are integrated along the inner walls. The advantage of the PCM is as backup for the case of power failure, common in countries with low grid stability like India, but also to increase the efficiency of the electric cooler [25].

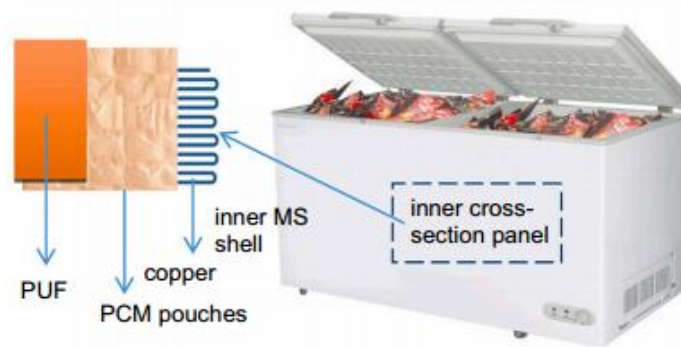


Figure I. 13 - Freezer/cooler

Phase change materials (PCMs) are essential components in latent thermal energy storage systems used in buildings. They are carefully selected based on their thermophysical properties and the specific heat storage requirements of residential structures, enabling energy availability during periods of low solar input, as shown in Figure I.14 To harness latent thermal energy and ensure its availability during nighttime when solar radiation is absent, Bolfazl Nematpour Keshteli et al.[26], Conducted a study on thermal energy utilization in buildings. The study aimed to meet residential energy demands by enhancing storage performance through an improved latent heat melting process of PCMs integrated into a heat exchanger.

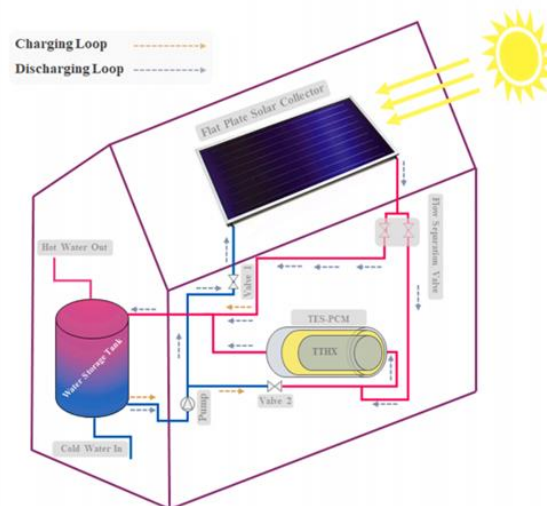


Figure I. 14 - Structural layout of a PCM-integrated solar thermal house water heating system

These diverse applications underscore the importance of PCMs in creating sustainable, efficient, and innovative solutions for everyday challenges, demonstrating their value across a wide range of industries.

I.8 Conclusion

This chapter presented a comprehensive overview of thermal energy storage technologies with particular emphasis on latent heat thermal energy storage systems utilizing PCM. The fundamental principles governing phase transition phenomena, PCM classifications, thermophysical properties, and major engineering applications were thoroughly discussed. The analysis demonstrated that PCMs offer substantial advantages in terms of energy storage density and thermal regulation capabilities.

However, the inherently low thermal conductivity of most PCMs remains a major challenge limiting their thermal performance. Consequently, extensive research efforts have focused on developing various heat transfer enhancement techniques to improve charging and discharging rates. The theoretical and scientific background presented in this chapter establishes the basis for the following chapter, which focuses on a critical literature review of recent studies related to thermal performance enhancement in PCM-based thermal energy storage systems.

Chapter II

Literature Review

II.1 Introduction

Phase Change Materials (PCMs) have gained considerable attention in recent years due to their high potential for thermal energy storage [17]. They can absorb and release large amounts of latent heat during phase transitions, making them suitable for applications such as heat exchangers, building insulation, and solar energy systems including flat-plate collectors and thermal storage tanks [3],[27]. Despite their advantages, the low thermal conductivity of conventional PCMs limits heat transfer performance and slows charging and discharging processes [28].

To address this limitation, various enhancement techniques have been widely investigated, including the use of fins, porous media, heat pipes, and nano-enhanced (NE- PCMs). In particular, integrating PCMs with advanced geometries such as triplex-tube heat exchangers and coupling them with solar collectors has shown significant improvement in thermal efficiency [29], [30]. This chapter reviews recent developments in PCM-based latent heat thermal energy storage systems, with emphasis on heat transfer enhancement methods and nano-enhanced approaches.

II.2 State-of-the-Art Review of Heat Transfer Enhancement Techniques in Phase Change Materials

The low thermal conductivity of (PCMs) represents a significant limitation to their effectiveness in thermal energy storage (TES) applications. This constraint reduces the rate of thermal energy storage and extraction, thereby hindering their performance in latent thermal energy storage systems. To address this issue, various heat transfer enhancement strategies have been developed to improve the thermal performance of PCMs. These techniques include the incorporation of high thermal conductivity nanoparticles, the use of metal foams as conductive matrices, and the integration of fins as extended surfaces. Figure II.1 presents an overview of the main heat transfer enhancement techniques employed in PCM-based thermal energy storage systems.

Improving the performance of latent heat thermal energy storage (LHTES) systems using (PCMs) has been a major focus in recent research due to challenges associated with low thermal conductivity and slow heat transfer rates. Various innovative approaches have been proposed, including geometric modifications of heat exchangers, the use of high-conductivity fins, and the incorporation of nanoparticles to enhance charging and discharging rates.

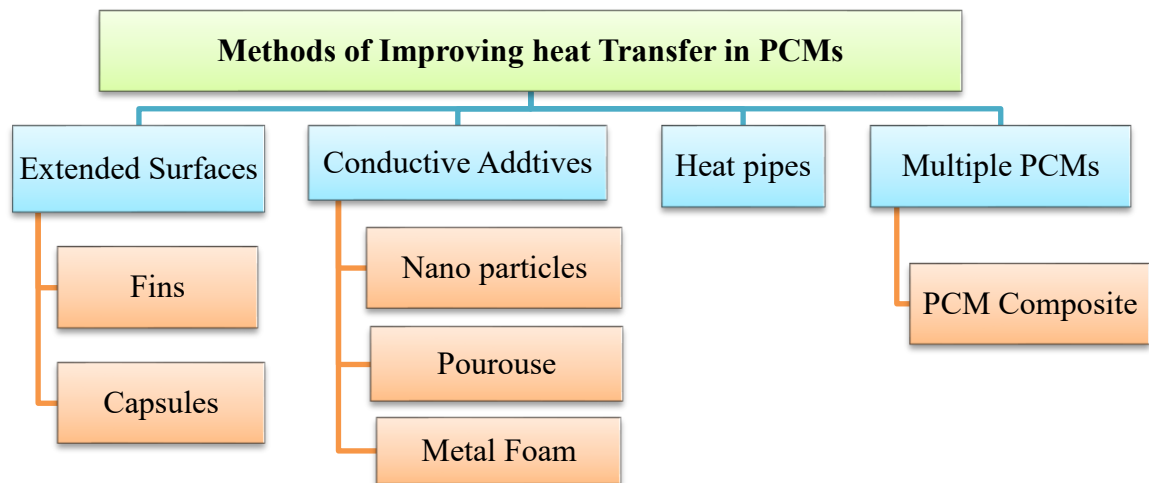


Figure II. 1 - Methods of improving thermal energy storage

II.2.1 Multiple phase change materials

The use of multiple phase change materials (PCMs) with different melting temperatures in latent thermal energy storage systems has attracted considerable attention due to its ability to increase storage capacity, improve heat transfer efficiency, and enlarge the effective operating temperature range. In this context, many researchers have investigated storage configurations employing different PCM arrangements and orientations (vertical, horizontal, and inclined) in order to optimize thermal system performance.

Hongtao Xu et al.[31], performed a numerical study on the melting behavior of triple-layer PCMs in a finned shell-and-tube latent heat thermal energy storage unit. The authors introduced a performance index accounting for the balance between heat transfer enhancement and the reduction in storage capacity caused by the presence of fins. Their findings demonstrated that fins considerably reduced the melting time, while a uniform radial fin distribution provided the best thermal performance under optimal operating conditions.

(PCMs) are widely employed in thermal energy storage applications because of their capability to absorb and release large amounts of heat during phase transition processes. Nevertheless, their inherently low thermal conductivity remains a major drawback that restricts the overall efficiency of thermal storage systems. Consequently, improving the thermal conductivity of PCMs has become one of the principal research directions in the field of thermal energy storage.

Among the most effective enhancement techniques are the addition of fins, the use of metal foams, and the incorporation of nanofluids, all of which have shown significant potential for improving heat transfer characteristics in PCM systems. The integration of PCM technology with

nanotechnology represents an important scientific and industrial challenge that has motivated numerous research studies in recent years. Therefore, a comprehensive bibliographic review is essential to establish a clear understanding of the current developments and to guide future research toward more efficient and optimized thermal storage solutions.

For example, M. Javidan et al. [32], carried out a detailed thermal analysis to examine the influence of wall transparency and thermal radiation on the melting process of paraffin-based PCMs as shown in Figure II.3, II.4. Their results indicated that increasing the applied heat flux significantly intensified convective heat transfer effects. Moreover, considering thermal radiation enhanced the paraffin melting process by approximately 31%, highlighting the importance of combining multiple heat transfer mechanisms to improve PCM thermal performance in energy storage systems.

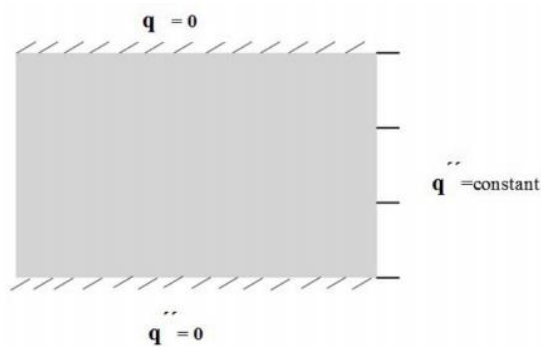


Figure II. 2 - Schematic of the studied cavity with boundary conditions

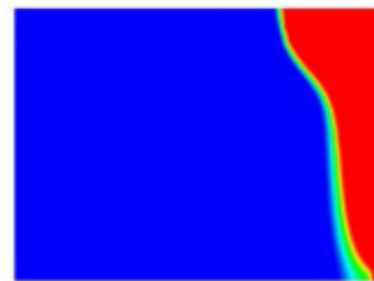


Figure II. 3 - Contours of liquid fraction in PCM -RT24 during melting

II.2.2 Number and arrangement of tubes

Adel Pourakabara et al. [33] studied the melting and solidification of (PCMs) inside a cylindrical vessel with different shell shapes and air chamber arrangements, as shown in Figure II.6 . The results showed that conduction dominated early in the process, while natural convection became significant later. The configuration with two vertically arranged tubes exhibited the highest melting rate, and the configuration with one tube had the lowest. Additionally, incorporating copper foam into the PCM increased the melting and solidification rates by 92% and 94%, respectively, but suppressed natural convection, especially during the melting phase.

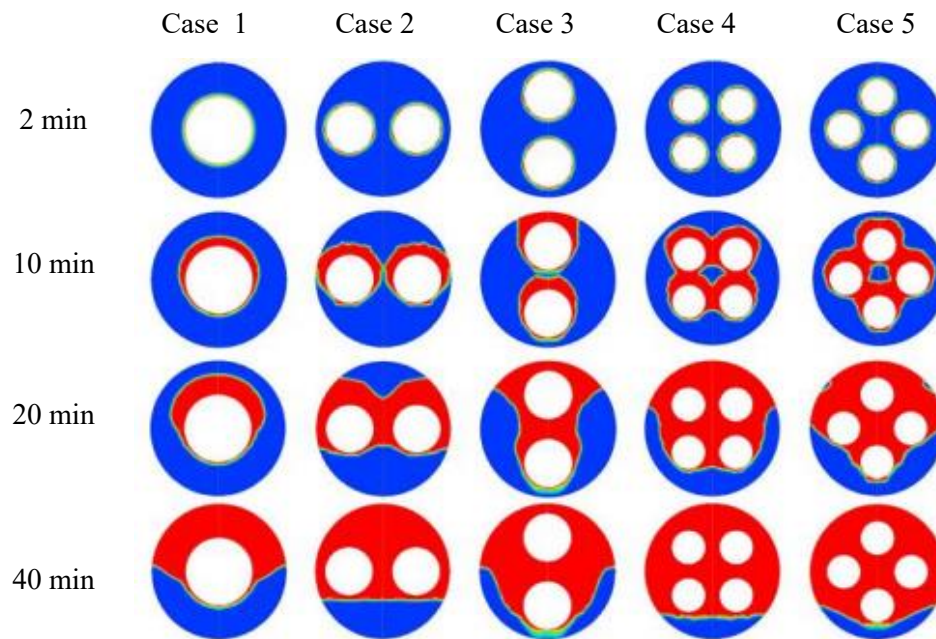


Figure II. 4 - Contours of the liquid fraction at different geometries

A recent study by Soumaya Sokakini et al. [34], Developed a multi-tube LHTES unit for low-temperature industrial waste heat recovery using diamond-shaped tubes with longitudinal fins, unlike previous studies that relied on 2D simulations and square tube designs.

Using erythritol as PCM and Hytherm 600 as HTF, the proposed design achieved 24.5% and 45.5% reductions in melting and solidification times, respectively, compared to a finless case. The study also highlighted the importance of three-dimensional effects, axial temperature gradients, and Reynolds number, which further improved performance by up to 17% and 14%.

Saleh Al Arni et al. [35], numerically studied latent thermal energy storage (LTES) systems assisted by zigzag-shaped tubes as shown in figure II.7 . They demonstrated that modifying the geometry of the tubes, optimizing flow parameters, and incorporating aluminum oxide nanoparticles significantly enhanced the discharge rate and led to a 75% reduction in solidification time compared to baseline straight tubes with pure PCM.

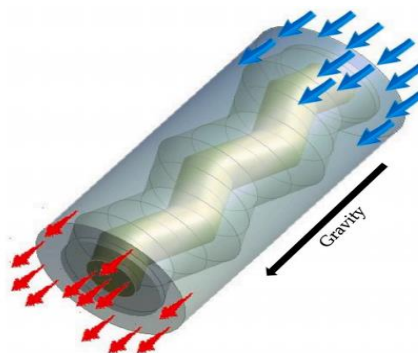


Figure II. 5 - The zigzag-piped heat exchanger schematic for PCM containment.

II.2.3 Enhancement Using Metal Foams and Porous Media

Metal foams and porous structures have been widely recognized as effective techniques for enhancing the thermal conductivity of phase change material (PCM)-based latent heat thermal energy storage systems. Owing to their highly conductive and interconnected porous networks, these materials provide continuous heat transfer pathways that significantly improve heat diffusion within the PCM domain.

Abolfazl Nematpour Keshteli et al. [26] demonstrated that the combination of porous metal foams with nanoparticles leads to a notable enhancement in heat transfer performance in latent heat storage systems. Their results showed significant reductions in charging and discharging times compared with pure PCM systems. However, increasing foam density was found to reduce the effective PCM storage volume and partially suppress natural convection in the molten region.

Similarly, Mohammad Reza Hassani et al. [36], investigated a three-layer PCM configuration integrated with Al6061 metal foam to improve thermal performance. Although the three-layer PCM exhibited a slower initial melting rate compared to a single-layer system, its overall melting behavior was improved. The incorporation of Al6061 metal foam further enhanced the melting process by increasing thermal conductivity and promoting more efficient heat transfer within the storage unit.

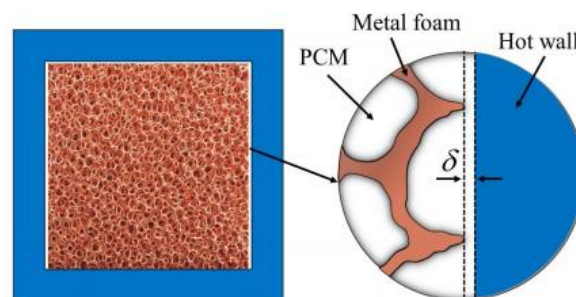


Figure II. 6 - Schematic representation of contact state of PCM composite for TES

In another study, Junlong Zou et al. [37], Showed that embedding copper foam fins (CFF) into encapsulated PCM significantly enhances the performance of latent heat storage systems. Their findings indicated reduced charging and discharging durations, improved energy storage and release capabilities, and enhanced natural convection during melting. Furthermore, optimization of fin geometry and distribution improved thermal performance without significantly decreasing the PCM filling ratio, making the system suitable for solar thermal energy storage and waste heat recovery applications.

Abolfazl Nematpour Keshteli et al. [38], Also studied paraffin-based PCM enhancement in flat-plate solar collectors through a combination of aluminum foam, nanoparticle addition, and

geometric modifications. Their results confirmed that integrating 10 PPI aluminum foam, 5 wt% nanoparticles, and optimized channel geometries significantly improved heat transfer and overall thermal storage efficiency.

Despite the substantial improvements achieved using metal foams and porous media, their practical implementation is still limited by several challenges, including increased manufacturing complexity, higher system cost, and reduced latent heat storage capacity due to the volume occupied by the porous matrix. Therefore, achieving an optimal balance between thermal conductivity enhancement and storage capacity remains a critical design requirement.

II.2.4 Enhancement Using Nanoparticles

Conventional phase change materials exhibit relatively poor thermal conductivity, the addition of nanoparticles has been extensively investigated as a means of enhancing thermal transport, thereby accelerating heat exchange and improving the efficiency of latent heat thermal energy storage systems [39],[28]. Previous investigations demonstrated that metallic and metal-oxide nanoparticles such as Cu, CuO, and Al₂O₃ can significantly reduce melting and solidification times by enhancing conductive heat transfer inside the PCM domain [23] , [40]. Nevertheless, increasing nanoparticle concentration may also increase viscosity and suppress natural convection effects, which may negatively influence the overall thermal performance of the system [41].

Similar to the study conducted by Prashant Saini et al. [42], Investigated Cesaro fin configurations for enhancing PCM melting under various conditions. Their study showed that natural convection improved melting by 26.8%, and optimized fin designs enhanced temperature uniformity and heat transfer. Moreover, combining copper metal foam with nanoparticles (Cu, CuO, Al₂O₃) reduced melting time by 63.4% due to improved thermal conductivity and synergistic effects.

Another interesting approach was proposed by M. Asgari et al. [43], Studied the effects of branch-shaped fins with varying thicknesses and lengths on the thermal conductivity of (PCMs) in a coaxial tube, as shown in Figure II.7. The results indicated that using thinner branch-shaped fins, along with a hybrid Al₂O₃ Cu mixture at a volume ratio of 0.04, significantly accelerated the solidification process. The study also demonstrated that the use of fins, particularly the thinner ones, greatly enhanced the speed of the solidification process.

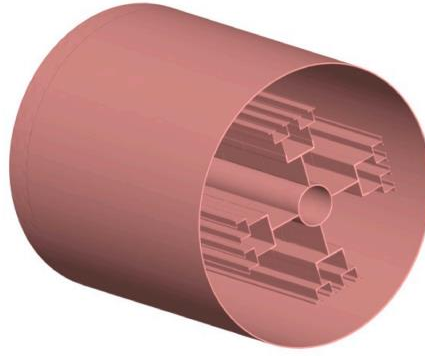


Figure II. 7 - Three-dimensional model of heat exchanger

Furthermore, a detailed study led by Prashant Saini et al. [42], Focused on the design and optimization of Cesaro fins in an LHS system using RT-82 PCM, employing a 2D numerical heat transfer model to analyze the effects of natural convection, fin arrangement, nanoparticles, and copper metal foam. The results revealed that natural convection improved melting rates by 26.8%, while optimized fin structures promoted uniform temperature distribution and enhanced heat transfer. The combined use of porous copper metal foam and nanoparticles (Cu, CuO, Al₂O₃) reduced melting time by 63.4%, attributed to enhanced thermal conductivity and synergistic effects between metal foam porosity and nanoparticle concentration.

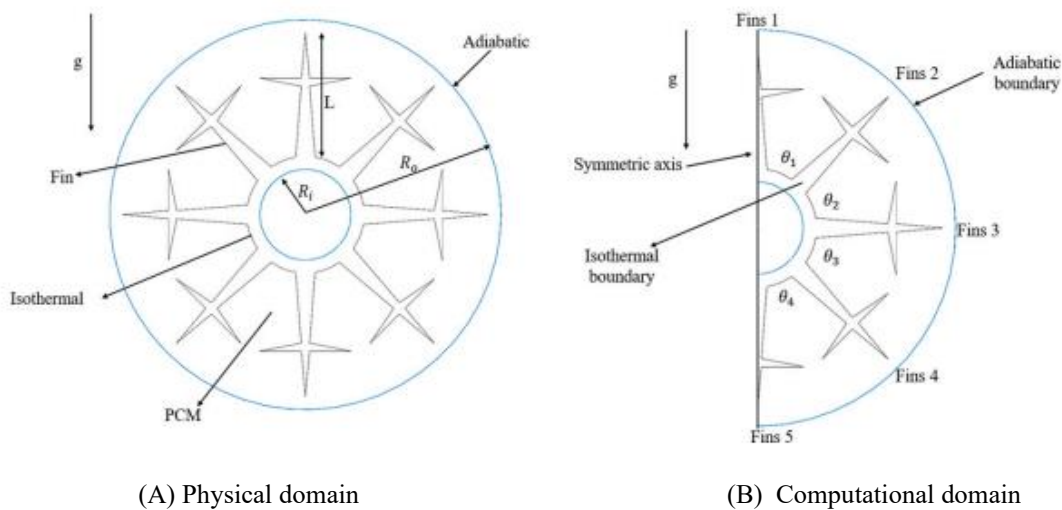


Figure II. 8 - A finned tube physical representation of the latent heat thermal energy storage.

In addition to passive enhancement techniques such as fins, porous media, and nanoparticles, recent studies have explored active enhancement methods capable of intensifying fluid motion and improving heat transfer mechanisms inside PCM systems. Among these approaches, rotating heat exchangers have attracted considerable attention because rotational motion can strengthen forced convection effects, reduce thermal stratification, and accelerate both charging and discharging processes.

II.2.5 Rotating heat exchangers

Rotational mechanisms have recently attracted increasing attention as active heat transfer enhancement techniques in latent thermal energy storage systems because they intensify fluid motion and improve convective heat transfer within the PCM domain [44], [45]. Several studies reported that rotating heat exchangers significantly reduce charging and discharging times by improving temperature distribution and accelerating phase transition processes [46].

A novel rotary triplex tube with sinusoidal wave-shaped fin design has been proposed as a PCM-based thermal storage unit by Aman Kumar et al. [45]. Compared with a conventional straight fin design. The discharging process of PCM is analyzed at different rotational speeds, as shown in Figure II.9. The thermal behavior of the latent heat storage system was assessed by monitoring the evolution of the liquid fraction, temperature field, and fluid flow patterns during the discharge process. The findings revealed that increasing the rotational speed significantly enhanced the PCM solidification rate, leading to faster heat release. Under stationary conditions or at low rotational speeds, the configuration equipped with straight fins exhibited a higher discharge performance than the sinusoidal-fin design.

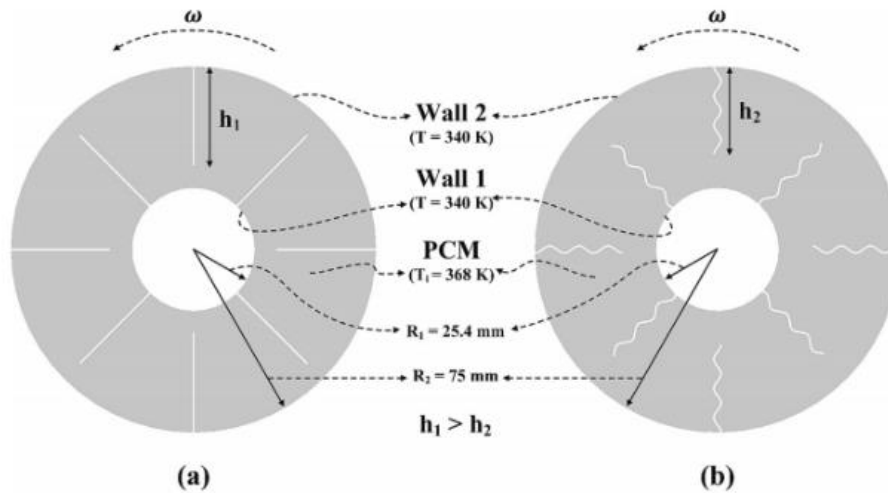


Figure II. 9 - Fin designs in a two-dimensional computational domain: (a) Straight, (b) Wavy.

Xinyu Huang et al.[44], Observed that the solidification time of the system at 1 rpm was reduced by 83.85%, and the heat release rate was 4.98 times higher than that at the no-rotation state, as shown in Figure II.10. These findings demonstrate that the incorporation of a rotational mechanism can effectively reduce the solidification time and increase the heat release rate of the system.

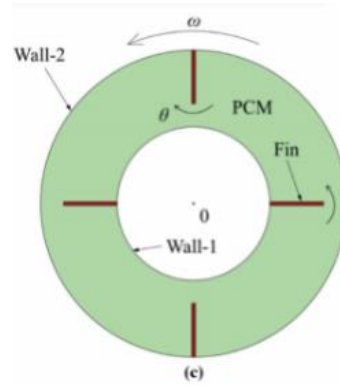


Figure II. 10 - Configurations rotation

Despite the significant thermal enhancement achieved using rotating systems, their practical implementation remains associated with several engineering challenges, including mechanical complexity, increased energy consumption, sealing difficulties, and higher maintenance requirements. Furthermore, excessive rotational speeds may weaken thermal stability and increase operational costs. Consequently, many researchers continue to favor passive enhancement techniques, particularly fin-based configurations, because of their simplicity, reliability, and relatively low manufacturing cost.

Among the various passive enhancement techniques proposed in the literature, fin-based heat transfer enhancement remains one of the most effective and widely adopted solutions for PCM thermal energy storage systems. Fins provide extended conductive pathways that facilitate heat diffusion inside the PCM domain while maintaining relatively simple system configurations. As a result, numerous investigations have focused on optimizing fin geometry, orientation, arrangement, and dimensions in order to maximize thermal performance during both melting and solidification processes.

II.2.6 Heat Transfer Enhancement Using Fins

Fins are among the most effective and widely used passive enhancement techniques in PCM thermal energy storage systems. Their incorporation increases the effective heat transfer area between the heat transfer fluid (HTF) and the phase change material (PCM), thereby improving conductive heat transfer and accelerating both melting and solidification processes. Several studies have demonstrated that fin geometry, orientation, thickness, and arrangement significantly influence the thermal behavior of PCM-based storage systems. Rashid et al. reviewed the role of fins in enhancing solidification within triplex-tube thermal energy storage systems and reported that finned configurations considerably reduced the solidification time compared with unfinned systems.

Similarly, numerous researchers investigated longitudinal and radial fins in cylindrical latent heat storage units. Their findings revealed that radial fins improve temperature uniformity and facilitate heat penetration into remote PCM regions. However, increasing fin density excessively may reduce the effective PCM storage volume and suppress natural convection effects during melting.

In recent years, advanced fin geometries such as Y-shaped, tree-shaped, and T-shaped fins have attracted considerable attention because of their ability to provide improved thermal distribution and larger heat transfer surfaces within the PCM domain. Previous investigations indicated that T-shaped fins significantly accelerate the melting process by enhancing thermal diffusion throughout the storage cavity. Moreover, such advanced structures may strengthen natural convection by guiding the movement of liquid PCM inside the enclosure.

Despite the substantial progress achieved in fin-enhanced PCM systems, most previous investigations have focused primarily on melting behavior, whereas comparatively limited attention has been devoted to solidification characteristics and cyclic thermal performance. In addition, the combined influence of advanced fin geometries and nano-enhanced (NE -PCMs) remains insufficiently explored.

A significant number of studies have also investigated the application of triplex-tube heat exchangers in latent thermal energy storage systems under various operating and geometric configurations. Various fin geometries, materials, and arrangements have been proposed to improve thermal storage efficiency and enhance heat transfer rates during charging and discharging processes. Straight, Y-shaped, W-shaped, V-shaped, and other customized fin configurations have demonstrated promising thermal enhancement capabilities in PCM.

Among these studies, Liang et al. [47], Conducted a comprehensive experimental and numerical investigation to analyze the charging and discharging (C/D) performance of a latent thermal energy storage (LTES) system integrated with a flat micro-heat-pipe array and a metal foam composite structure

A holistic study including both experimental and numerical methods was undertaken by Liang et al. analyse the SCD operation of an LTES system with a flat micro-heat-pipe array and metal foam composite structure.

In order to investigate the effect of fins on enhancing heat transfer in (PCMs) , Sohif et al. [29], Conducted an experimental study using a triplex tube heat exchanger. They evaluated the performance of latent thermal energy storage systems by designing three configurations: inner fins, outer fins, and combined inner-outer fins, as shown in Figure II.11. The results showed that

using a triplex tube heat exchanger with combined inner-outer fins significantly reduced the melting time, achieving a 43.3% faster melting rate compared to the configuration without fins.

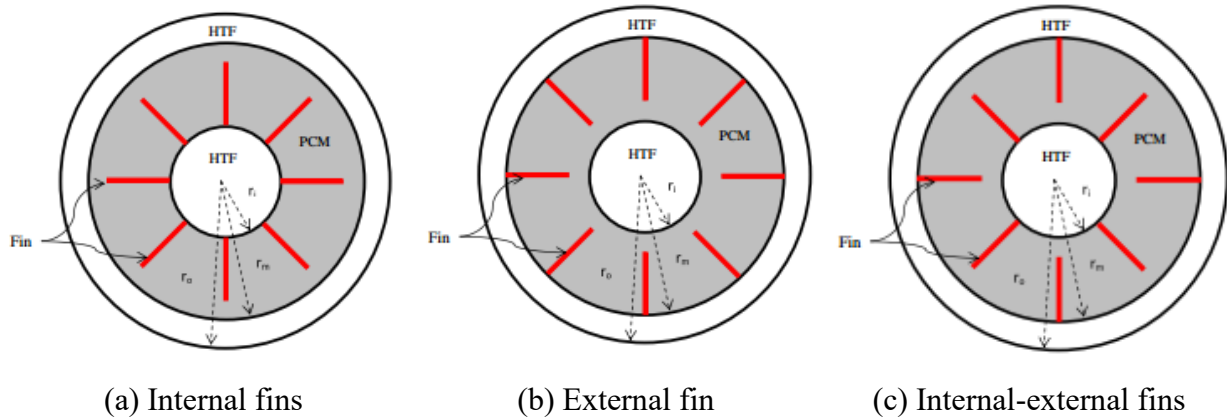


Figure II. 11 - The physical configuration of the TTHX models

Amrita Sharma et al.[17], Evaluated the performance of triplex tube heat exchangers during the solidification of (PCMs) using two fin configurations: attached and detached, as shown in Figure II.12 . The study found that detached fins were more effective in enhancing system performance. Furthermore, the addition of 7% Cu nanoparticles significantly improved the solidification rate and overall thermal efficiency.

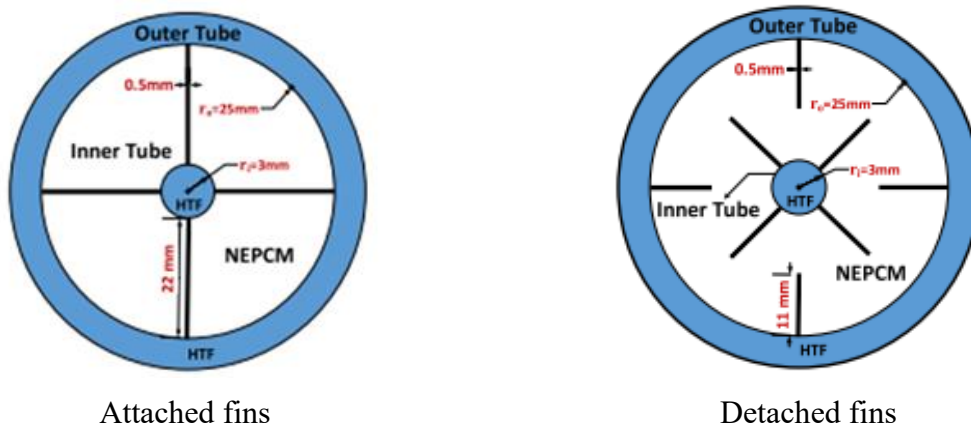


Figure II. 12 - Configuration of triplex tube models with attached fins and detached fins

The same triplex tube heat exchanger model was numerically investigated by Yan et al.[30], who incorporated Y-shaped fins to enhance thermal performance, as shown in Figure II.13 . The study examined various fin configurations and analyzed the influence of the heat transfer fluid (HTF) on the melting behavior of the phase change material (PCM). The results indicated that the melting process progressed through three distinct stages, with solid Y-shaped fins significantly accelerating the melting rate. Moreover, reductions in fin thickness, increases in fin angle, and elevated HTF temperatures were all shown to further enhance the melting performance of the PCM, thereby improving the overall efficiency of the latent thermal energy storage system.

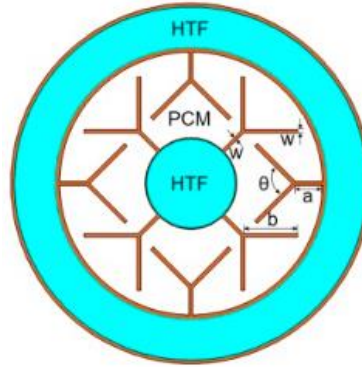
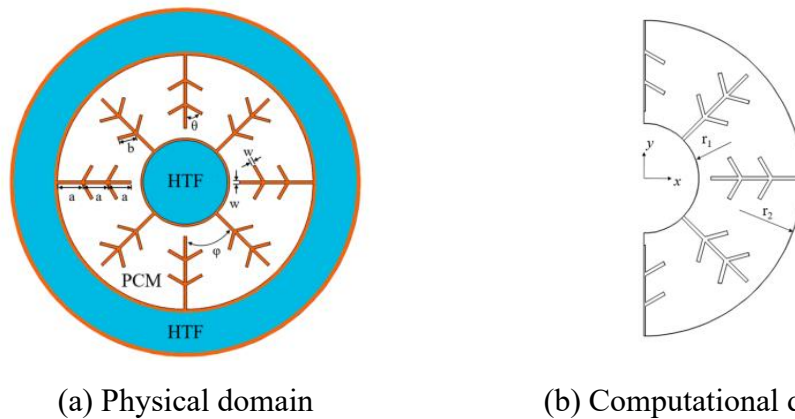


Figure II. 13 - Configuration of triplex tube models with Y-shaped fins

Peiliang Yan et al. [48], Numerically investigated novel leaf vein e-fin structures and different tube arrangements in a three-tube TES system, as shown in Figure II.14 . Using enthalpy and porosity methods, they found that these fin designs significantly reduced PCM melting time. In particular, increasing the fin angle from 30° to 60° in branch vein configurations (with 1–3 branches) reduced total melting time by up to 14.3%.

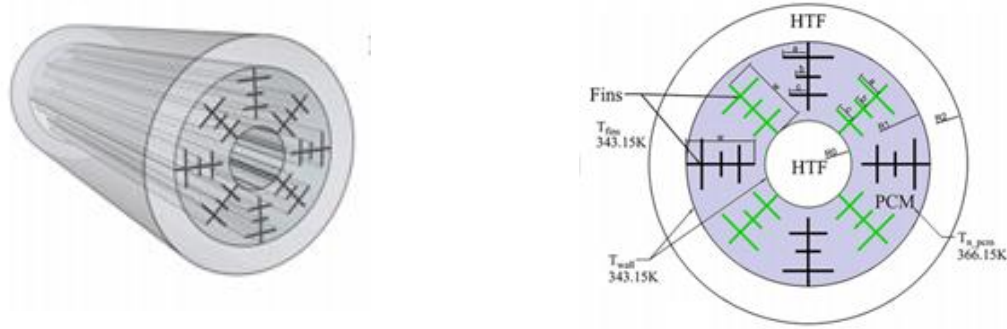


(a) Physical domain

(b) Computational domain

Figure II. 14 - Configuration of a heat exchanger incorporating PCM.

Ji Zhang et al. [49], They proposed a combination of novel branch-structured fins and Al_2O_3 nanoparticles to enhance the performance of a phase change material (PCM) during the solidification process in a triplex-tube heat exchanger, as shown in Figure II.15 . A mathematical model for a two-dimensional structure, comprising PCM (paraffin RT82) and Al_2O_3 nanoparticles, was developed. The model accounts for thermal conduction in metal fins, Brownian motion of nanoparticles, and natural convection in the liquid phase of the PCM and was validated against experimental results. They concluded that the impact of using fins is more significant compared to nanoparticles, and the benefit of incorporating



(a) A three-dimensional physical model

(b) A two-dimensional cross-section domain

Figure II. 15 - Configurations of the heat exchanger with fins: 2D and 3D physical models.

Fan Ren et al. [50], The numerical simulation for the melting and solidification processes of PCM RT 55 in Triplex-tube thermal energy storage (TTES) evaluated the comprehensive thermal performance in a crossed snowflake fin proposed study. The three-casing phase change heat storage unit filed with RT55 is as follows: The distribution of snowflake fins into eight fins is being studied. The analysis of the thermal performance of TTES has shown that when the length of the fin is 11 mm and the width of the fin is 0.8 mm, TTES has the best overall thermal performance. Following optimization, the heat of PCM melting accumulation increased by 3.92 KJ, while the heat of PCM solidification exotherm increased by 5.773 KJ.

Together, these studies highlight the effectiveness of combining multiple enhancement techniques including geometric design optimization, flow parameter control, nanoparticle addition, and metal foam incorporation to overcome the intrinsic limitations of PCMs and significantly improve the thermal performance of LHTES systems.

Arsalan Yawar et al. [46], Conducted a comprehensive numerical study on a triplex-tube latent thermal energy storage (LTES) system under different operating modes, including charging, simultaneous charging–discharging (SCD), and discharging. The study examined the effects of system orientation, heat transfer fluid (HTF) tube placement, flow direction, and flow rate on thermal performance. The results indicated that a horizontal configuration with outer tube charging and inner tube discharging yields superior performance, significantly reducing charging time and enhancing heat transfer. Overall, the study emphasized that proper optimization of flow arrangement and system configuration plays a crucial role in improving the efficiency of LTES systems.

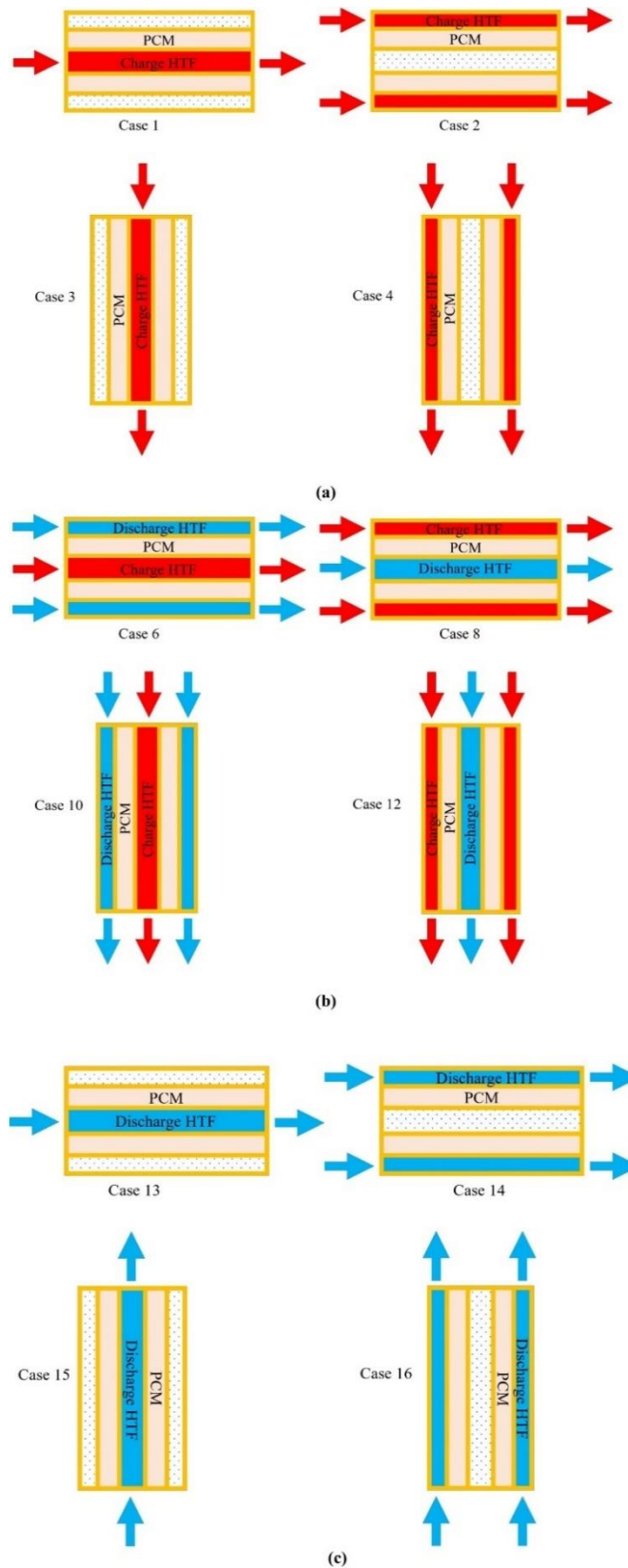


Figure II. 16 - Case of various design configurations

Ibrahim E. Sadiq et al [51], Conducted a study focused on improving the thermal performance and heat transfer of paraffin wax used as a phase change material (PCM) in a triplex tube heat storage system during the discharging process. The study used a combined experimental and numerical approach and investigated the use of hybrid nanoparticles ($\text{Al}_2\text{O}_3/\text{CuO}$) to enhance the thermal conductivity of paraffin and improve system performance. Several nanoparticle volume fractions (0.4%, 0.8%, 1.6%, and 3.2%) were examined, along with the effects of heat transfer fluid (HTF) inlet temperature and mass flow rate. The results showed that decreasing the inlet temperature or increasing the HTF flow rate accelerates the solidification process, with temperature having a stronger influence. The addition of hybrid nanoparticles improved the freezing performance in all cases, with the highest enhancement achieved at 3.2% volume fraction, reducing solidification time by about 23%.

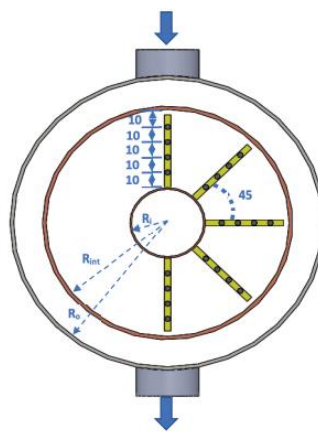


Figure II. 1 - Schematic diagram of thermocouple distribution

Mohamed Boujelbene et al. [52], Highlighted that integrating (PCMs) into thermal energy storage systems significantly enhances energy efficiency across various applications. However, slow PCM melting remains a major limitation. To address this issue, the authors investigated the use of arch-shaped fins in a horizontal triplex-tube storage system. Their findings demonstrated that such geometric enhancements can substantially improve heat transfer, accelerate the melting process, and increase the overall heat storage rate. The study further showed that optimizing fin dimensions and connecting angles plays a critical role in improving system performance, confirming the effectiveness of design modifications in PCM-based thermal storage systems.

Based on previous studies, Mohamed Ahmed Said et al [53], Presented a study aimed at improving the performance of latent heat thermal energy storage (LHTES) systems using (PCM), by developing sunburst-distributed radial fins inside a shell-and-multi-tube configuration to overcome the low thermal conductivity problem.

The results, compared with previous works, showed that this design significantly enhances heat transfer, reduces melting and solidification times, and improves the efficiency of thermal energy storage and recovery.

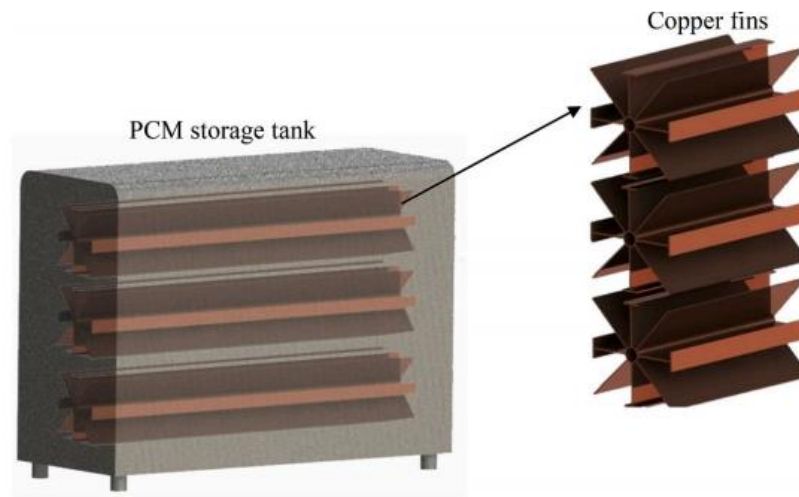


Figure II. 2 - Schematic diagram of proposed PCM system

A comparative review of previous studies indicates that no single enhancement technique can completely overcome the intrinsic limitations of PCMs under all operating conditions. While nanoparticles improve thermal conductivity at the microscopic level, fins and metal foams mainly enhance macroscopic heat diffusion. Rotating systems, on the other hand, intensify convective transport mechanisms but introduce additional mechanical complexity. Therefore, recent research trends increasingly focus on hybrid enhancement techniques that combine multiple approaches simultaneously in order to achieve higher thermal efficiency and shorter charging/discharging times.

II.3 Triplex Tube Heat Exchanger

A Triplex Tube Heat Exchanger consists of three concentric tubes arranged one inside the other. In PCM applications, the Phase Change Material is typically placed in the annular space between the inner and middle tubes, while hot and cold fluids flow through the innermost and outermost tubes. This design allows effective heat transfer between the fluids and the PCM, enhancing thermal energy storage and temperature regulation. The triplex tube heat exchanger is favored for its compactness and high heat transfer efficiency, making it well-suited for systems requiring reliable thermal management.

II.4 Research Gap in Thermal Performance Enhancement of PCM Systems

Despite the considerable progress achieved in enhancing heat transfer in phase change material (PCM)-based thermal energy storage systems, several research gaps still remain, particularly regarding the combined use of fins and nanoparticle additives under identical operating conditions, as most previous studies have investigated these enhancement techniques separately; furthermore, the influence of fin geometry, arrangement, and design parameters on both melting and solidification behavior is still not fully understood, while the majority of existing works have focused mainly on the charging (melting) process with limited attention to discharging performance and cyclic thermal behavior, in addition to unresolved issues related to nanoparticle agglomeration, increased viscosity, and their effect on natural convection within the PCM domain, which all indicate the need for further comprehensive numerical and experimental investigations to optimize the coupled use of fins and nanoparticles in order to achieve higher heat transfer rates, improved thermal response, and enhanced energy storage efficiency.

II.5 Conclusion

In this section, the review of studies of PCM is performed, including the manufacturing methods, the thermal performance enhancement, and the application in the thermal energy storage system. It is found that the shaped fins as the thermal enhancer plays an important role in the thermal performance enhancement of PCM.

In the previous literature, many researchers have investigated PCM in triplex tube used. The T-Shaped fins in triplex tube heat exchange affect the enhancement of PCM melting. The present work numerically investigates heat transfer in PCM

Chapter III

Mathematical Modeling and Numerical Simulati

III.1 Introduction:

In this chapter, describes the modeling of the studied system is presented. The system consists of two main components: the solar collector unit, represented by a flat plate solar collector, and the thermal energy storage unit, represented by a storage tank. Particular attention is devoted to the modeling of solar radiation, since it constitutes a fundamental aspect of this study. Subsequently, the various thermal interactions occurring within the solar collector are analyzed. The main objective is to develop a mathematical model capable of predicting the outlet temperature of the collector under steady-state conditions for different nanofluids (CuO, Al₂O₃, and TiO₂) during representative days throughout the year in Tissemsilt city. The incorporation of nanoparticles into the working fluid is proposed in order to improve the thermal performance of the system. Finally, a detailed description of the thermal energy storage model using Paraffin RT82 as a phase change material (PCM) is also presented.

This section is organized into three main parts:

- **Modeling of Solar Radiation:**

This part focuses on the evaluation and modeling of solar radiation, which determines the amount of solar energy available to the system. The model considers geographical Location and seasonal variations specific to Tissemsilt.

- **Modeling of the Flat Plate Solar Collector:**

In this section, the thermal behavior of the flat plate solar collector is investigated in order to describe the solar to thermal energy conversion process. The analysis takes into account the thermophysical properties of different nanofluids (CuO, Al₂O₃, and TiO₂), as well as the collector efficiency and its thermal performance under varying climatic conditions.

- **Numerical Simulation of the Thermal Storage System Using Paraffin RT82:**

This part presents the modeling of the thermal energy storage system using Paraffin RT82 as a phase change material for both case of proposed fin designs , specifying geometric features, and boundary conditions. The storage and release processes of thermal energy are analyzed through the study of a domestic hot water storage tank coupled with the solar collector.

III.2 System description of the solar energy unit

The solar energy unit is a fully integrated system developed to harness solar energy for the purpose of thermal power generation Figure. III.1.

- ✓ The proposed solar thermal system is composed of three interconnected subsystems that operate together to capture, transfer, and store thermal energy. The first component is a flat-plate solar collector, which is responsible for converting incident solar radiation into useful thermal energy through the circulation of a heat transfer fluid. The second element is a circulation pump, whose function is to maintain the fluid flow between the collector and the storage unit, thereby improving heat transport and ensuring continuous operation of the system.
- ✓ The third component is a latent heat thermal energy storage unit based on a triplex-tube heat exchanger. This storage device employs phase change materials (PCMs) to accumulate thermal energy during periods of solar availability and release it when solar input is insufficient, such as during nighttime operation or low-radiation conditions. The integration of PCM-based storage enhances the reliability of the system by providing a more stable and continuous thermal energy supply for different applications.

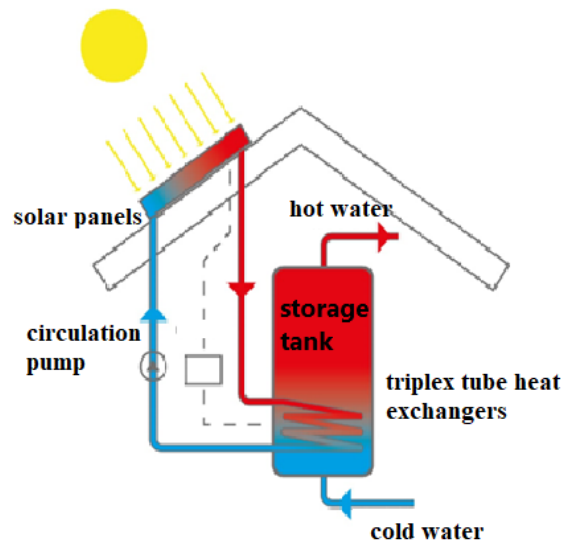


Figure III. 1 - Presentation of the physical system of flat collector and storage tank.

III.3 Modeling of solar radiation

III.3.1 Geographical Parameters

III.3.1.1 Longitude (λ)

The longitude of the place λ corresponds to the angle between the plane of the meridian passing through this place and the plane of the meridian maintained as origin. The meridian (origin point 0) was chosen as the plane passing through the Greenwich observatory. According to the convention, a sign (+) is assigned to the longitude to the east of this longitude, and a sign (-) to the longitude to the west [54].

III. 3.1.2 Latitude (ϕ)

Latitude ϕ is a characteristic property of any location that is widely employed in solar applications. A location's latitude is its angular position with respect to the equator. It ranges from 0 to 90 degrees in the north hemisphere and from 0 to -90 degrees in the south [6].

III. 3.2 Astronomical Parameters

III. 3.2.1 Declination angle (δ)

The declination angle, designated as δ , shows the bent distance of the Sun from the Earth's tropical plane calculated at the local summit. This angle changes with little or no deviation on account of the tilt of the Earth's capable of rotating shaft and allure at fixed intervals motion about the Sun. The declination angle ranges from 23.45° to $+23.45^\circ$, place beneficial principles display that the Sun is northward of the globe. If the Earth were not tilted, this angle would wait nonstop at 0° during the whole of the period [6].

Certain days of the year are considered astronomically significant. These include the summer and winter solstices, as well as the spring and autumn equinoxes. The hot season of the year happens about 21 June and complements to the maximum event of dawn in the Northern Hemisphere. In contrast, the winter solstice occurs around 21 December and corresponds to the minimum duration of daylight. During this period, the Sun reaches its most southern declination, approximately -23.45° [6], [55].

The equinoxes happen about 21 March (spring equinox) and 21 September (season between summer and winter equinox). On these dates, neverending are nearly equal existing, each enduring about 12

Hours general. The migratory conduct illustrated above applies to the Northern Hemisphere, while in the Southern Hemisphere the opposite happens[55].

Figure III.2 illustrates the annual variation of solar declination and its relationship to these key astronomical events [55].

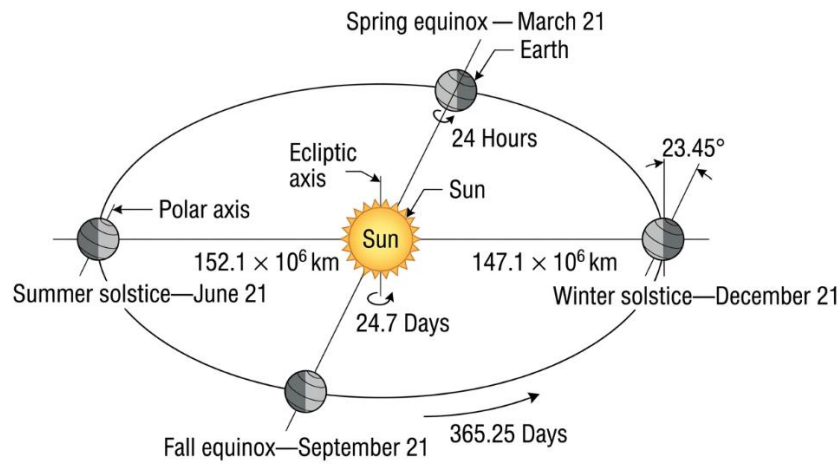


Figure III. 2 - Earth's orbit around the sun over the course of a year.

In Figure III. 3 a drawing of the alternative of refusal in strengths over the old ge is likely. Similar is Table III.1 appearance the advantage of refusal in the average era of each temporal length of event or entity's existence [5].

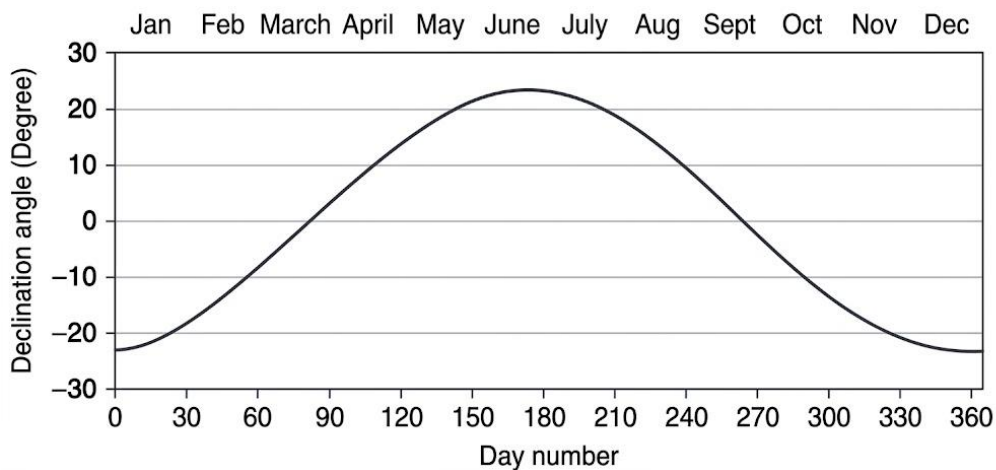


Figure III. 3 - Declination variation over the year [5].

Table III. 1 - Representative day number and corresponding solar declination angle for each month

MONTH	DAY NUMBER	AVERAGE DAY OF THE MONTH		
		Date	n	δ [deg]
January	I	17	17	-20.92
February	31+ i	16	47	-12.95
march	59+ i	16	75	-2.42
April	90+ i	15	105	-9.41
May	120+ i	15	135	18.79
June	151+ i	11	162	23.09
July	181+ i	17	198	21.18
August	212+ i	16	228	13.45
September	243+ i	15	258	2.22
October	273+ i	15	288	-9.60
November	304+ i	14	318	-18.91
December	334+ i	10	344	-23.05

The equation mostly used for engineering purposes is:

$$\delta = 23.45 \sin(360 \times (284 + n)/365) \quad (\text{III.1})$$

In the above equations we normally use noninteger values for n, as the maximum daily declination is 0.4° and B is obtained from equation III.1.

III. 3.2.2 Hour Angle (ω)

The ending angle, ω , is the bent flow of the light part of 24 hours either Korean or west from the solar noonday, for that reason the earth's excite allure point about that entity revolves. At solar the middle of a day the entertainment industry angle is 0° and 14° - a displacement of 15° is equivalent to 1 importance. Morning hours are thought-out negative, while sundown hours are advantageous. The stage angle possibly on the way to breadth from equating:[5].

$$\omega = 15(TSV - 12) \quad (\text{III.2})$$

Slope

The surface of a solar hobbyist is mainly slanted alternatively level so that increase the amount of occurrence solar dissemination. The tilt angle, usually designated by β , is delineated as the angle 'tween the plane of the connoisseur and the level surface.

Its value ranges from 0° (horizontal surface) to 180° , where angles greater than 90° correspond to downward-facing surfaces. Appropriate selection of the tilt angle, often based on the local latitude,

helps maintain the collector surface as nearly perpendicular as possible to incoming solar radiation over a given period [6].

III. 3.2.2 Azimuth Angle

The angle between the local meridian and the normal of the surface on the horizontal plane is called the surface azimuth angle, abbreviated as γ . It can assume values ranging from -180° to 180° , with 0° representing the south. Eastward is designated as negative, while westward is designated as positive. [5],[6].

III. 3.2.3 Sunrise and Sunset Hours Angles (ω_s)

It is sometimes useful to know the sunrise and sunset hours. At the moment that sun rises or sets the altitude angle is 0° and the zenith angle is 90° . So, the sunset hour angle, ω_s , can be estimated from equation III.3 :

$$\cos(\omega_s) = -\frac{\sin(\phi) \times \sin(\delta)}{\cos(\phi) \times \cos(\delta)} = \tan(\phi) \times \tan(\delta) \quad (\text{III.3})$$

With a change of the list duplicate equating we take the rise of star above skyline importance angle. Because at solar top the profit of ending angle is 0° and knowledgeable that each displacement of 15° is had connection with 1 ending, critical moment of dawn and sunset maybe deliberate.

III. 3.2.3 Day Length (N)

Day length it is even more powerful to knowledge the dawn hours, N, significance only to reproduce the dawn or sunset occasion following 2 Equation 16 gives the number of dawn hours:

The daylight hours, N, having only to multiply the sunrise or sunset time with 2.

Equation III.4 gives the number of daylight hours:

$$N = \frac{2}{15} \times \cos^{-1}(-\tan \phi \times \tan \delta) \quad (\text{III.4})$$

In Figure III.6 a nomogram for determining the period distance and twilight obtain sight respect to scope, decay and season is likely. An instance is likely apart from the character for any following $\phi = 39^\circ$ and $\delta = 20^\circ$. These standard bring about dusk stage of 7:08 pm for time off and 4:52 pm for cold. Moreover, the age distance in sabbatical is 14.28 hours and in cold 9.72 hours [6].

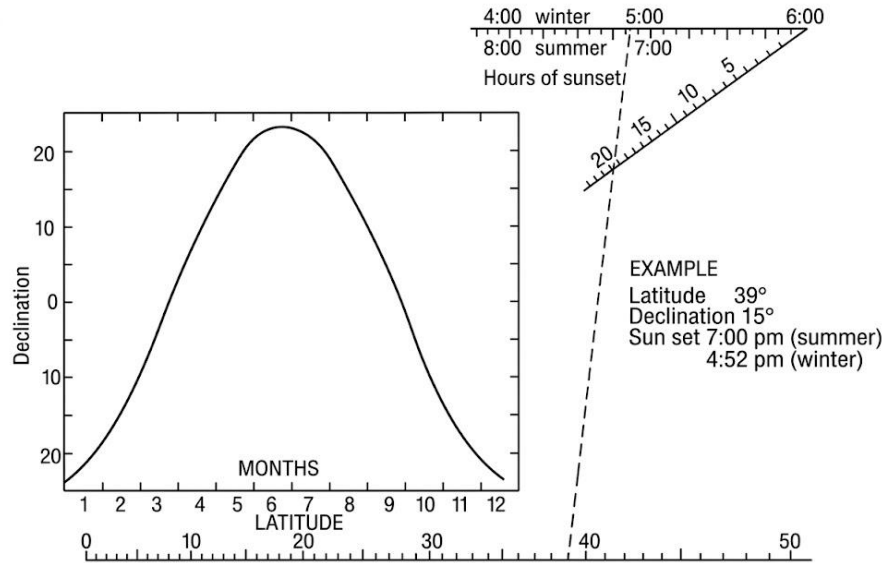


Figure III. 4 - Nomogram estimating day length and sunset hour

III.3.3 Time Parameters

III.3.3.1 True Solar Time (TST)

The True Solar Time (TST) at any given instant can be determined using the following equation:

$$\text{TST} = 12 \pm \frac{\omega_0}{15} \quad (\text{III.5})$$

ω : En degree,

Si $\omega_0 = 0 \dots \text{TST} = 12\text{h.00}$,

Si $\omega_0 < 0 \dots$ The morning,

Si $\omega_0 > 0 \dots$ The afternoon.

III.3.3.2 Mean Solar Time (MST).

It is given by the relation:

$$\text{MST} = \text{TST} - \text{Et} \quad (\text{III.6})$$

Et: the equation of time, which corrects the TST with respect to the MST

$$\text{Et} = 9.87 \sin 2\beta - 7.35 \cos \beta - 1.5 \sin \beta \quad (\text{III.7})$$

$$\beta = \frac{360}{365} (n - 81) \quad (\text{III.8})$$

III.3.3.3 Universal Time (UT)

It is the mean civil time of the Greenwich Meridian (prime meridian), also called GMT (Greenwich Mean Time).

$$TU = MST \pm \frac{L}{15} \quad (III.9)$$

Where: L is the longitude. $L > 0$ for cities located to the East of Greenwich. $L < 0$ for cities located to the West of Greenwich. $TU = MST$ for the Greenwich Meridian.

III.3.3.4 Legal Time (LT)

$$LT = TU + C \quad (III.10)$$

C: the time difference between the Greenwich Meridian and the considered location.

C= 1 hour for Algeria .

$$LT = TST + C + \left(\frac{Et + 4L}{60} \right) \quad (III.11)$$

III. 3.4 Solar angle

III. 3.4.1 Solar Altitude Angle (α_s)

The solar altitude angle, also known as the elevation angle, α_s , is the supplemental angle to the zenith angle, as illustrated in Figure III.4. Solar altitude angle, then, is the angle created by the sun's rays and the horizontal. The altitude angle is zero degrees at dawn and dusk and ninety degrees at solar noon, reaching its highest value at that time,[6]. Equation 8 is utilized to estimate the solar altitude angle, while equation 7 illustrates the link between the zenith and altitude angles:

$$\theta_z + \alpha_s = \frac{\pi}{2} = 90^\circ \quad (III.12)$$

$$\sin(\alpha_s) = \cos(\delta) \times \cos(\phi) \times \cos(\omega) + \sin(\phi) \times \sin(\delta) \quad (III.13)$$

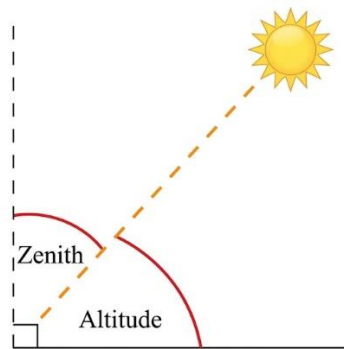


Figure III. 5 - Altitude angle [18]

III. 3.4.2 Solar Azimuth Angle (γ_s)

The solar azimuth angle, meant accompanying γ_s , is the bent dislocation of direct dissemination on a level plane from on west side when facing north. Eastward displacement is negative, and situated or toward the west is beneficial. It can catch principles middle from two points -180° and 180° [6],

The solar azimuth angle takes the value of the timing angle signal, and the use of equation III. 5 can be described for its calculation:

$$\gamma_s = \sin(\omega) \left| \cos^{-1} \left(\frac{\cos \theta_z \times \sin \varphi - \sin \delta}{\sin \theta_z \times \cos \varphi} \right) \right| \quad (\text{III.14})$$

In the abandoned constituent Figure 8 miscellaneous angles that have happened elucidated in this place passage are proved for a fought surface.

In the right constituent Figure III.5 a plan view of solar azimuth angle is likely.

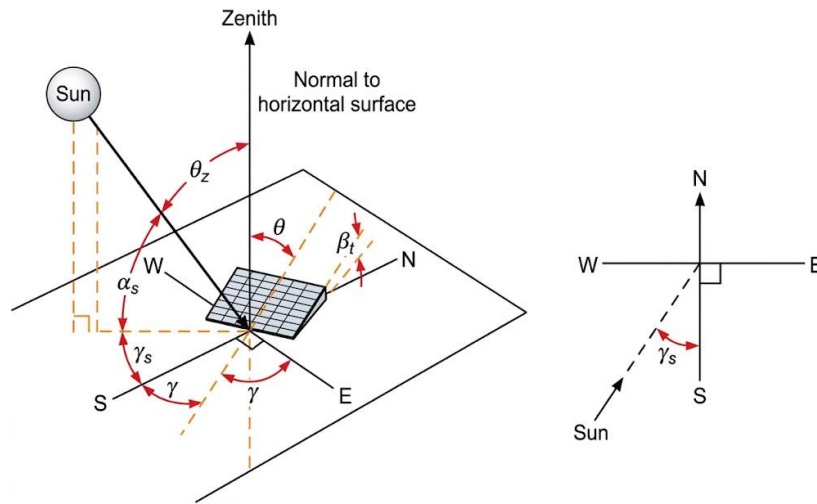


Figure III. 6 - Surface tilt angles and corresponding solar azimuth in plan view [6]

III. 3.4.3 Zenith Angle (θ_z)

The zenith angle (θ_z), is the solar angle that forms when the sun's beams intersect with the vertical. In the case of beam radiation, the angle of incidence on a flat surface remains constant.[5],[6].

III.3.4.4 Incidence Angle (θ)

The angle of occurrence, θ , is the angle named for one sane of a surface and the beam fallout occurrence on it. As pronounced before, for level surfaces the occurrence angle is the unchanging accompanying the top angle [5], [6]. A set of equatings are handy, pertaining the angle of occurrence accompanying the former noticed angles.

The general equations are III.15 and III.16:

$$\begin{aligned} \cos(\theta) = & \sin(\delta) \times \sin(\phi) \times \cos(\beta) - \sin(\delta) \times \cos(\phi) \times \sin(\beta) \times \cos(\gamma) \\ & + \cos(\delta) \times \cos(\phi) \times \cos(\beta) \times \cos(\omega) \\ & + \cos(\delta) \times \sin(\phi) \times \sin(\beta) \times \cos(\gamma) \times \cos(\omega) + \cos(\delta) \times \sin(\beta) \times \sin(\gamma) \times \sin(\omega) \end{aligned} \quad (\text{III.15})$$

$$\cos(\theta) = \cos(\theta_z) \times \cos(\beta) \mp \sin(\theta_z) \times \sin(\beta) \times \cos(\gamma_z - \gamma) \quad (\text{III.16})$$

It bear be famous that the angle of occurrence maybe above 90o what in equating 2.10 the moment angle endure be 'tween dawn and sundown. Also, any of streamlined cases can happen. I n the case of established leaned surfaces either towards south or northward, intention that the sur face azimuth angle is 0o or 180o individually, the last term of equating III.17 maybe excluded. This is very ordinary to occur implant plate hobbyists [6].

For horizontal surfaces, where zenith angle and angle of incidence are the same and the slope angle is 0°, equation III.17 becomes:

$$\cos(\theta_z) = \cos(\phi) \times \cos(\delta) \times \cos(\omega) + \sin(\phi) \times \sin(\delta) \quad (\text{III.17})$$

For vertical surfaces, where the slope is 90° we have:

$$\begin{aligned} \cos(\theta) = & -\sin(\delta) \times \cos(\phi) \times \cos(\gamma) \\ & + \cos(\delta) \times \sin(\phi) \times \cos(\gamma) \times \cos(\omega) \times \cos(\delta) \times \cos(\gamma) \times \cos(\omega) \end{aligned} \quad (\text{III.18})$$

Lastly, for the extreme case of solar noon and a south facing tilted surface in the northern hemisphere we have:

$\theta_{\text{noon}} = \phi - \delta - \beta $	(III.19)
--	----------

III.4 Modeling of the Flat Solar Collector

Let Q be the solar energy absorbed by a collector. A portion of this energy is stored in the heat transfer fluid, which is the useful energy Qu. This energy can either be used directly (air heating) or transferred to a storage system (hot water tank). A solar collector is subject to thermal losses. A portion of the absorbed energy is transferred to the environment as thermal energy, primarily through convection and radiation. These losses will be denoted as Qa.

Finally, if we focus on a system's temperature ramp-up period (e.g., at startup), we must consider the energy absorbed by the collector to reach its operating temperature. This energy, denoted Qt, depends on the thermal inertia of the system [5], [56].

Therefore, the energy balance can be expressed as:

$$Q = Q_u + Q_a + Q_t \quad (\text{III.20})$$

Let Eg be the global solar irradiation received by the collector. The average efficiency of this irradiation over a period Δt can be written as[5]:

$$\bar{\eta} = \frac{\int_0^{\Delta t} Q_u dt}{\int_0^{\Delta t} E_g \times dt} \quad (\text{III.21})$$

The sky radiates at the temperature T∞ , which is given by the following empirical formula:

$$T_{\infty} = 0.0552 \times T_a^{1.5} \quad (\text{III.22})$$

The solar power absorbed by the collector can be expressed as:

$$Q = Eg \times \tau_c \times \alpha_p \quad (\text{III.23})$$

$$\rho_c + \alpha_c + \tau_c = 1 \quad (\text{III.24})$$

The heat flux Q_a lost to the environment can be determined using an electrical analogy [57], as shown in Figure III.7 .

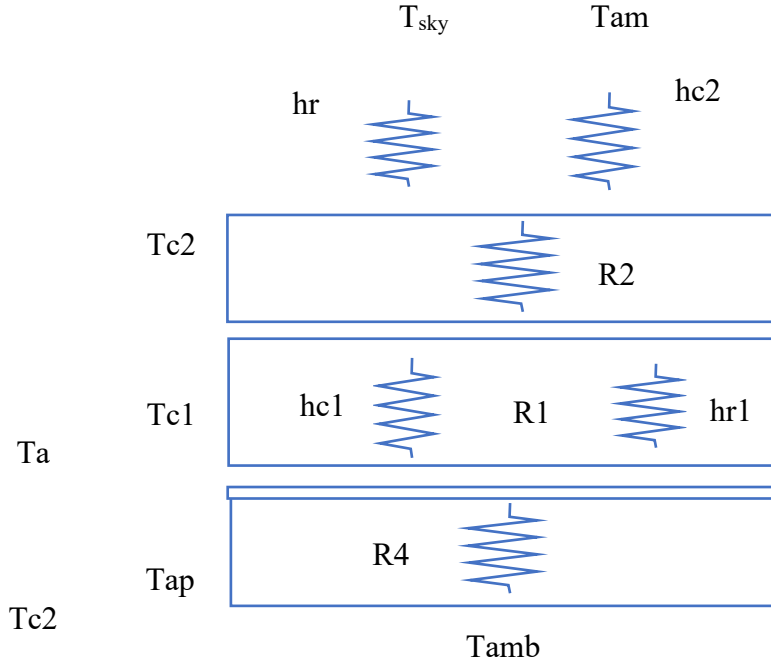


Figure III. 7 - Electrical analogy for determining the heat flux lost to the environment

Calculation of the thermal resistance R_1 between the absorber plate at T_a and the glass at T_c : First, the natural convection heat transfer coefficient between the plate and the glass is calculated.

$$h_{c1} = \frac{Nu \times \lambda_a}{b} \quad (\text{III.25})$$

The first calculating of the Grashof number, a dimensionless number characteristic of the natural convection phenomenon:

$$Gr = \frac{g}{T} \times \frac{(T_p - T_c) \times b^3}{\nu_a^2} \quad (\text{III.26})$$

The average temperature of the air confined between the absorber and the glass is represented by the following expression:

$$\bar{T} = \frac{T_p + T_c}{2} \quad (\text{III.27})$$

Arbitrarily chose the temperature of the glass. The thermodynamic fluid properties are interpolated from the air air property table . Nu is the Nusselt number. For an inclined surface, it is given by:

$$Nu = 1 + 1.44 \left[1 - \frac{1708}{Ra \cos(a)} \right] \left(1 - \frac{1708(\sin(1.8a))^{\frac{1}{8}}}{Ra \cos(a)} \right) + \left[\left(\frac{Ra \cos(a)}{5830} \right)^{\frac{1}{3}} - 1 \right] \quad (III.28)$$

Calculate the Rayleigh number:

$$Ra = Gr \times Pr \quad (III.29)$$

$$Pr = 7.57 \times 10^{-5}(T_f) + 0.0242 \quad (III.30)$$

The thermal exchange coefficient by radiation:

$$h_{r1} = \frac{\sigma(T_p^2 + T_c^2)(T_p + T_c)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} - 1} \quad (III.31)$$

T_p and T_c are, respectively, the temperatures of the absorber and the transparent cover.

$$R_1 = \frac{1}{h_{c1} + h_{r1}} \quad (III.32)$$

Calculation of the thermal resistance R_2 In the glass, there is thermal conduction

$$R_2 = \frac{e_v}{\lambda_v} \quad (III.33)$$

Calculation of the thermal resistance R_3 between the glass at T_c and the exterior T_a :

$$R_3 = \frac{1}{h_{r2} + h_{c2}} \quad (III.34)$$

The same approach by calculating, respectively:

$$h_{c2} = 5,7 + 3,8w \quad (III.35)$$

$$T_{\infty} = 0.0552T_a^{1,5} \quad (III.36)$$

$$Q_{r2} = h_{r2} \times (T_c - T_a) \quad (III.37)$$

From which h_{r2} is derived:

$$h_{r2} = \frac{\varepsilon_c \sigma (T_c^4 - T_{\infty}^4)}{T_c - T_a} \quad (III.38)$$

From which the thermal resistance towards the front of the collector R_{av} can be expressed as:

$$R_{av} = R_1 + R_2 + R_3 \quad (III.39)$$

Thermal resistance created by the insulation R'_1 :

$$R'_1 = \frac{e_i}{\lambda_i} \quad (III.40)$$

Thermal resistance R'_2 due to convection between the insulation and the ambient air. Generally, this resistance can be neglected; consequently, R_{ar} is expressed as:

$$R_{ar} = R'_1 + R'_2 \quad (\text{III.41})$$

The electrical analogy allows us to write:

$$\frac{1}{R} = \frac{1}{R_{av}} + \frac{1}{R_{ar}} = \frac{1}{R_1 + R_2} + \frac{1}{R'_1 + R'_2} \quad (\text{III.42})$$

And by using thermal conductance's:

$$K = K_{av} + K_{ar} \quad (\text{III.43})$$

The thermal conductance at the front face of the collector is written as:

$$K_{av} = \frac{1}{R_1 + R_2} \quad (\text{III.44})$$

The thermal conductance at the rear face of the collector is written as:

$$K_{ar} = \frac{1}{R'_1 + R'_2} \approx \frac{1}{R'_1} \quad (\text{III.45})$$

It remains to verify the choice for the temperature T_c of the glass. For this, we can write that, in steady-state conditions, the sum of the thermal fluxes exchanged by the glass is zero:

$$(h_{c1} + h_{r1})(T_c - T_p) + (h_{c2} + h_{r2})(T_c - T_a) = 0 \quad (\text{III.46})$$

Or, by using thermal resistances:

$$\frac{T_c - T_p}{R_1} + \frac{T_c - T_a}{R_2} = 0 \quad (\text{III.47})$$

And consequently:

$$T_c = \frac{R_1 \times T_a + R_2 \times T_p}{R_1 + R_2} \quad (\text{III.48})$$

The thermal flux exchanged with the environment at T_a is written as:

$$Q_a = K \times (T_p - T_a) \quad (\text{III.49})$$

The instantaneous efficiency is calculated as follows:

$$\eta = \frac{Q_u}{E_g} = \frac{Q - Q_a}{E_g} = (\tau_c \times \alpha_p) - \frac{Q_a}{E_g} \quad (\text{III.50})$$

Finally, the instantaneous efficiency in steady state of the flat plate collector and the optical efficiency of the collector are calculated [57].

$$\eta = \eta_0 - K \frac{T_p - T_a}{E_g} \quad (\text{III.51})$$

III.4.1 Nanofluid Thermophysical Properties

To effectively utilize nanoparticles in PCM for thermal energy storage systems, it is essential to select particles based on specific criteria, including high thermal conductivity, excellent stability, low density, and cost-effectiveness. nanoparticles were selected are detailed in Table 4 The thermophysical properties of nanofluids are determined by the correlations of Khanafer et al.[58], The physical properties of the pure fluid (water) and those of the nanoparticles (Cu, Al₂O₃ and TiO₂) are given in Table III.2 at temperature T 25°C.

Table III. 2 - Thermophysical properties of base fluid and nanofluids

Property	Base fluid (water)	Nanoparticle (Al ₂ O ₃)	Nanoparticle (cu)	Nanoparticle (TiO ₂)	Nanoparticle (CuO)
C _p (J·kg ⁻¹ ·K ⁻¹)	4179	765	385	686.2	535.6
ρ (kg·m ⁻³)	997.1	3970	8933	4250	5600
λ (W·m ⁻¹ ·K ⁻¹)	0.613	40	400	8.9538	20
μ (kg·m ⁻¹ ·S ⁻¹)	8,91×10 ⁻⁴	/	/	/	/
β (K ⁻¹)	21 ×10 ⁻⁵	0.85 ×10 ⁻⁵	1.67×10 ⁻⁵	0.9 ×10 ⁻⁵	1.74×10 ⁻⁵

In the following, all thermophysical properties of the nanofluids used are:

The density of the nanofluid is given by the following relation [59],[60] :

$$\rho_{nf} = \left(\frac{m}{V} \right)_{nf} = \frac{m_f + m_p}{V_f + V_p} = \frac{\rho_f V_f + \rho_p V_p}{V_f + V_p} = (1 - \varphi)\rho_f + \varphi\rho_p \quad (\text{III.52})$$

$$\varphi = \frac{V_p}{V_f + V_p} \quad (\text{III.53})$$

The specific heat Cp_{nf} of the nanofluid is determined by assuming thermal equilibrium between the nanoparticles and the pure fluid (water):

$$Cp_{nf} = \frac{(1 - \varphi)\rho_f Cp_f + \varphi\rho_p Cp_p}{\rho_{nf}} \quad (\text{III.54})$$

Some authors prefer to use a simpler expression given by:

$$Cp_{nf} = (1 - \varphi)Cp_f + \varphi Cp_p \quad (\text{III.55})$$

The coefficient of thermal expansion of the nanofluid β_{nf} is determined by the expression:

$$\beta_{nf} = \frac{(1 - \varphi)\rho_f \beta_f + \varphi\rho_p \beta_p}{\rho_{nf}} \quad (\text{III.56})$$

Some prefer to use a simpler expression given by:

$$\beta_{nf} = (1 - \varphi)\beta_f + \varphi\beta_p \quad (\text{III.57})$$

The viscosity model used is based on the volume fraction [26]. Einstein and determined the viscosity of a nanofluid μ_{nf} consisting of a suspension of spherical solid particles, with a volume fraction less than 0.05 ($\phi < 0.05$), using the following relation:

$$\mu_{nf} = (1 + 2.5\phi)\mu_f \quad (\text{III.58})$$

This equation is valid for a dilute suspension of rigid spheres in a viscous liquid. Several equations have been developed for suspensions with higher concentrations. Brinkman extended Einstein's equation to concentrated suspensions by establishing the following correlation:

$$\mu_{nf} = \frac{\mu_f}{(1 - \phi)^{2.5}} \quad (\text{III.59})$$

The thermal conductivity of the nanofluid is determined from [61]:

$$\lambda_{NEPCM} = \frac{\lambda_s + (S - 1)\lambda_{PCM} - (S - 1)\phi(\lambda_{PCM} - \lambda_s)}{\lambda_s + (S - 1)\lambda_{PCM} + \phi(\lambda_{PCM} - \lambda_s)} \times \lambda_{PCM} \quad (\text{III.60})$$

The thermal diffusivity of the nanofluid is given by the expression:

$$\alpha_{nf} = \frac{k_{nf}}{(\rho C p)_{nf}} \quad (\text{III.61})$$

III.4.2 Determination of the Collector Output Temperature

It seems more reasonable to determine the efficiency of the collector based not on the temperature T_p of the absorber, but rather as a function of the temperature T_{fe} of the fluid at the inlet of the collector [57], as shown in Figure III.8. The absorbing surface is considered isothermal at T_p . In reality, the plate is subjected to a non-uniform temperature field, constant in steady state

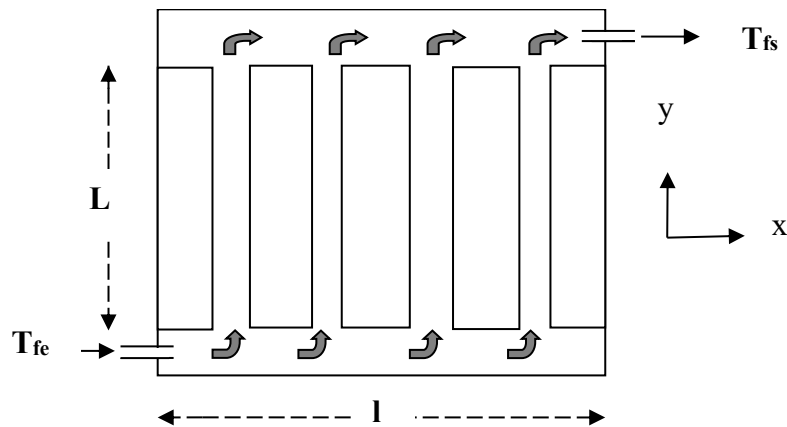


Figure III. 8 - Flat-Plate solar collector

In the first approximation, we can neglect the temperature variations along the y and z directions Figure III.8, and we can then write:

$$\frac{\partial T \vec{\epsilon}_p}{\partial y} = \frac{\partial T_p}{\partial z} = 0 \quad (\text{III.62})$$

This means considering that the temperature of the plate depends only on x . However, we will assume it is constant for the part located above a water pipe, over a length De see Figure III.9. Therefore, the temperature of the plate is assumed to vary only over the length $2w$, which corresponds to the interval separating two pipes[57]. To determine the temperature variation, we write the heat equation for an element of the plate with thickness δ , width dx , and length L located at position x between two pipes, as shown in Figure III.9.

$$\rho_p \cdot c_p \frac{\partial T_p}{\partial t} = \lambda_p \cdot \Delta T_p + \frac{\dot{Q}_u}{\delta} = 0 \quad (\text{III.63})$$

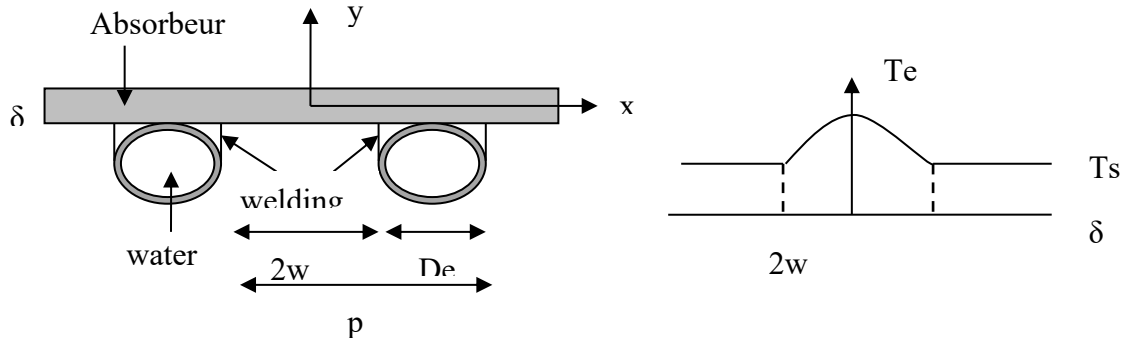


Figure III. 9 - Cross-section of the collector and temperature evolution

Temperature of the plate as a function of x .

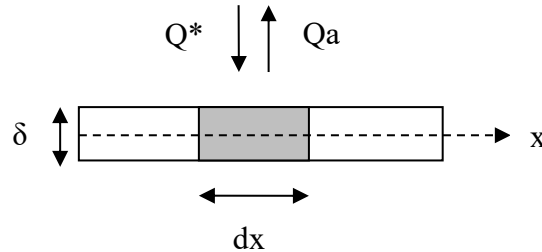


Figure III. 10 - Diagram used for calculating the thermal balance of a plaque element with length dx .

In steady state $Q_u = Q^* - Q_a$.

In this equation, λ_p represents the thermal conductivity of the material constituting the absorber plate.

In steady-state conditions, we have:

$$\frac{\partial T \vec{\epsilon}_p}{\partial t} = 0 \quad (\text{III.64})$$

But also, since $Q_t = 0$ $Q_u = Q^* - Q_a$

Since the temperature T_p is independent of the variables y and z , the Laplacian of the temperature is written as:

$$\Delta T_p = \frac{d^2 T_p}{dx^2} \quad (\text{III.65})$$

On the other hand, the thermal flux lost to the environment is written as:

$$Q_a = K(T_p - T_a) \quad (\text{III.66})$$

Where K is the overall conductance of the collector, which we assume, in the first approximation, to be constant.

The heat equation then becomes

$$\frac{d^2 T_p}{dx^2} = \frac{K}{\lambda_p \delta} \left[T_p - \left(T_a + \frac{\dot{Q}}{K} \right) \right] \quad (\text{III.67})$$

The solution of this second-order differential equation requires the knowledge of two boundary conditions[56].

For reasons of symmetry, we have:

$$\left(\frac{dT_p}{dx} \right)_{x=0} = 0 \quad (\text{III.68})$$

On the other hand, it is assumed that the temperature T_s of the weld is equal to the local temperature T_p , which is assumed to be constant over the length De corresponding to the outer diameter of the pipe, as shown in Figure III.11 .

This can be written $T_p = T_s$ For $x = w$

By setting: $m^2 = \frac{K}{\lambda_p \delta}$

The solution is written as:

$$T_p(x) = T_a + \frac{\dot{Q}^*}{K} + \left[T_s - \left(T_a + \frac{\dot{Q}^*}{K} \right) \right] \frac{ch(m.x)}{ch(m.w)} \quad (\text{III.69})$$

Le terme $ch(m.w)$ représentant la fonction cosinus hyperbolique

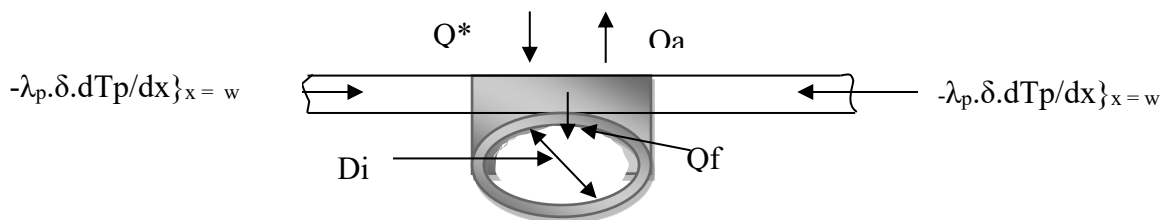


Figure III. 11 - Increase in heat transfer by thermal conduction

Now, let's determine the thermal power $\dot{Q}_f$ absorbed by the heat transfer fluid per meter of pipe length. This power is equal to the solar input $\dot{Q}^*$ reduced by the thermal losses $\dot{Q}_a$ and increased by the contributions from thermal conduction. The steady-state energy balance[56], equation III.70. is written as :

$$\dot{Q}_f = \left[\dot{Q}^* - \dot{Q}_a \right] D_e - 2\lambda_p \delta \frac{dT_p}{dx} \Big|_{x=w} \quad (\text{III.70})$$

Let's assume:

$$\eta_w = \frac{th(m.w)}{m.w} \quad (\text{III.71})$$

Où th représente la fonction tangente hyperbolique. On obtient alors :

$$\dot{Q}_f = \left[\dot{Q}^* \times (-K)(T_s - T_a) \right] (D_e + 2w\eta_w) \quad (\text{III.72})$$

Let's express this quantity in terms of the temperature T_f of the fluid, rather than the temperature T_s of the weld.

Note:

Let x be the first point of contact of the pipe: $x = \pi \cdot D_i$, where D_i is the inner diameter of the pipe.

Let h_c be the convective heat transfer coefficient between the wall and the fluid[56].

We then obtain:

$$\dot{Q}_f = \left[\dot{Q}^* - K(T_f - T_a) \right] pF \quad (\text{III.73})$$

With: the distance between pipes[56].

$$F = \frac{1}{\frac{p}{(D_e + 2.w.\eta_w)} + \frac{p.K}{x.h_c}} \quad (\text{III.74})$$

Where: $p = D_e + 2.w$

This factor corresponds to the ratio of the thermal resistance of the absorber plate to the ambient air over the thermal resistance of the fluid to the ambient air.

Now, let us consider an elementary volume of fluid, as shown in Figure III.12 . The heat balance equation in steady-state conditions is expressed as:

$$\dot{m}C_p dT_f = \dot{Q}_f dy \quad (\text{III.75})$$

Where m is the mass flow rate of the heat transfer fluid (in kg/s per pipe) and C_p is the specific heat capacity of the fluid (in $\text{KJ} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$).

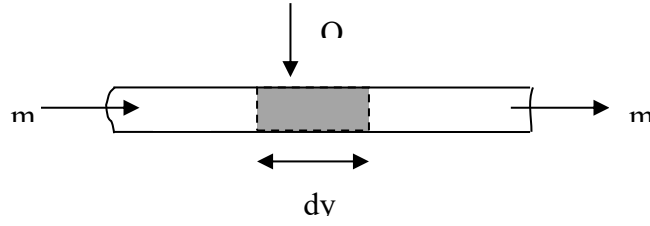


Figure 12 - Diagram used to establish the thermal balance of an elementary fluid volume

By substituting Q_f with its expression in terms of T_f , we obtain:

$$\frac{dT_f}{T_f - T_a - \frac{\dot{Q}^*}{K}} = -\frac{pFK}{\dot{m}Cp} dy \quad (\text{III.76})$$

The boundary conditions are expressed as: $T_f = T_{fe}$ for $y=0$ and $T_f = T_{fs}$ for $y = L$.

$$\dot{M} = \frac{\dot{m}}{pL} \quad (\text{III.77})$$

The mass flow rate per unit area of the collector ($\text{kg/s} \cdot \text{m}^2$).

$$\frac{T_{fs} - T_a - \frac{\dot{Q}^*}{K}}{T_{fe} - T_a - \frac{\dot{Q}^*}{K}} = \exp\left(-\frac{FK}{\dot{M}Cp}\right) \quad (\text{III.78})$$

Expression that allows determining the outlet temperature T_{fs} of the heat transfer fluid as a function of its inlet temperature T_{fe} .

The power supplied to the fluid per square meter of the collector is expressed as:

$$\dot{Q}_u = \dot{M}Cp(T_{fs} - T_{fe}) \quad (\text{III.79})$$

$$Ft = \frac{\dot{M}Cp}{K} \left[1 - \exp\left(\frac{-FK}{\dot{M}Cp}\right) \right] \quad (\text{III.80})$$

Referred to as the thermal transfer factor. This factor represents the ratio of the actual heat flux received by the fluid to the heat flux that would be recovered if the absorbing surface were isothermal at T_{fe} . This factor will approach unity in cases of high flow rates and collector operation at low temperatures (swimming pool collectors). Thus, we have:

$$\dot{Q}_u = Ft \left[\dot{Q}^* - K(T_{fe} - T_a) \right] \quad (\text{III.81})$$

Hence, the instantaneous efficiency:

$$\eta = Ft \left(v^* - K \frac{T_{fe} - T_a}{E_G} \right) \quad (\text{III.82})$$

Finally, we can calculate the outlet temperature of HTF from the expression for useful energy.

$$T_{fs} = T_{fe} + \frac{Q_u}{\dot{m}C_p} \quad (\text{III.83})$$

III.5 Modeling of the storage system

III.5.1 Physical model

In this section, we will present the modeling of storage unite system triplex tube heat exchanger, as shown in Figure 23 with the geometric parameters given in Table III.3 [62].

The (PCM) utilized is paraffin RT-82, is filled between the inner and outside tubes. A circulation of heat transfer fluid (HTF). In this design, where the phase change material is present, T-shaped fins and new fins are attached to the walls in alternation as shown in Figure III.12, b.

The effect of dimensional and physical parameters of T shaped fins the PCM is studied, namely: thicknesses (ea) an (eb), length (la) and (lb) and temperature of the heat transfer fluid Table III.5

Table III. 3 - Dimensions of the Triplex Tube

Tube	Inter radius (mm)	Outer radius (mm)	Thickness (mm)
Inner	25,4	26,6	1,2
Intermediate	75	77	2
Outer	98	100	2

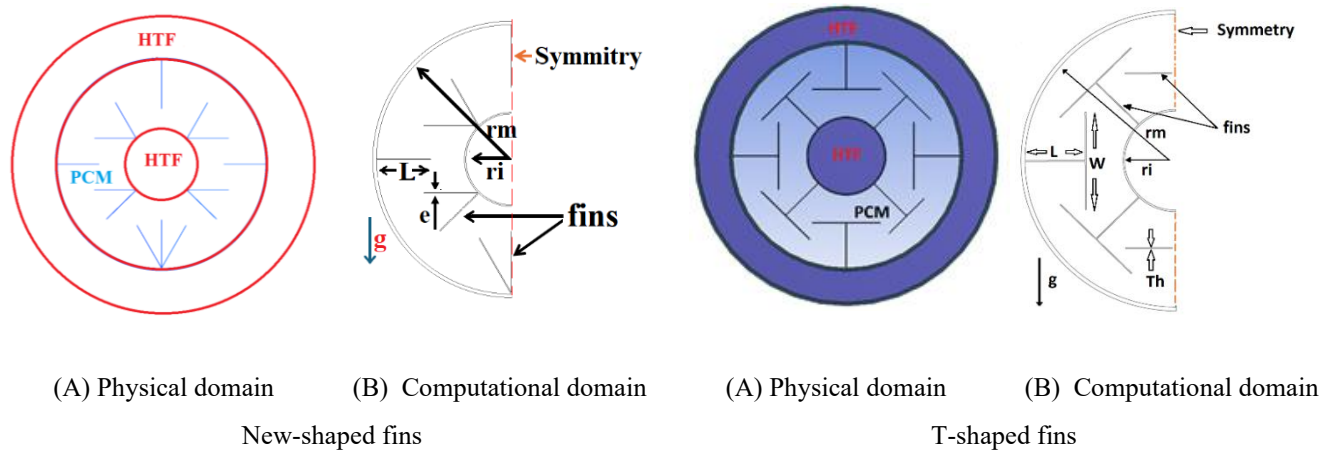


Figure III. 12 - Configuration model triplex tube

In this study, the (PCM) utilized is paraffin RT-82, selected for its favorable thermal properties and performance characteristics. The heat exchanger, specifically a triplex tube configuration, is constructed from copper, chosen for its excellent thermal conductivity and durability [29]. In addition to RT-82, other PCMs were as well as considered for comparative analysis, namely RT-50 [63] and RT-60 [64]. These materials were selected to assess their relative performance and effectiveness in different thermal management scenarios.

The thermophysical properties of these three PCMs RT-82, RT-50, and RT-60 are detailed in Table III.4, providing crucial information such as melting points, latent heat, and thermal conductivity, which are essential for understanding their behaviors in the heat exchanger system.

Table III. 4 - Thermophysical properties of three paraffin waxes (PCMs) and copper

Propriety	Unit	RT 50	RT60	RT82	Copper
Solid density	ρ_s (kg·m ⁻³)		880	950	8978
Liquid density	ρ_l (kg·m ⁻³)	830	770	770	/
Specific heat of PCM	C_{pl} (J·kg ⁻¹ K ⁻¹)	2400	2000	2000	381
Latent heat of fusion	L (J·kg ⁻¹)	179000		176000	
Liquid temperature	T _m (K)	331.15		350.15	/
Solidus temperature	T _s (K)	325.15		358.15	
Thermal conductivity	K (W·m ⁻¹ ·K ⁻¹)	0.2	0,2	0.2	387.6
Thermal expansion coefficient	β (l·K ⁻¹)	0.0006		0.001	/
Viscosity dynamics	(kg·m ⁻¹ ·s ⁻¹)	0.005		0.03499	/

Furthermore, the study explores various geometrical configurations for the T-shaped fins and new fins used in the heat exchanger. The detailed geometric parameters of these T-shaped fins, including their thickness (Th), length (L), width (w), and the operational fluid temperature for heat transfer, are specified in Table III.5. and new shaped are specified in Table III.6 These parameters are critical as they directly influence the heat transfer efficiency and overall performance of the system. By examining different configurations, the study aims to optimize the design for improved thermal management and effectiveness in the heat exchanger application

Table III. 5 - The cases of numerical simulation with T-shaped fins

CASES	LENGTH (mm)		WIDTH (mm)	THICKNESS (mm)	T _{HTF} (K)
	INNER	OUTER			
01	10	10	33	0,5	363
02	20	20	33	0.5	363
03	30	30	33	0.5	363
04	40	40	33	0.5	363
05	30	30	40	0.5	363
06	30	30	45	0.5	363
07	30	20	33	0.5	363
08	20	30	33	0.5	363
06	30	30	45	0.5	363
11	30	30	45	01	363
12	30	30	45	02	363

Table III. 6 - The cases of numerical simulation with the new shaped fins

CASES	LENGTH (mm)		THICKNESS (mm)	T _{HTF} (K)
	INNER	OUTER		
Case (A)	30	30	0.5	363
Case (B)	35	35	0.5	363
Case (C)	40	40	0.5	363
Case (D)	45	45	0.5	363

III.5.2 Numerical model

Problem description

A comprehensive 2D model was meticulously developed to simulate the process using ANSYS FLUENT, ensuring a detailed and accurate representation of the physical phenomena. The simulation operates in two dimensions, providing a balance between computational efficiency and precision. The SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) algorithm was employed to manage the coupling between pressure and velocity fields, a critical aspect of fluid dynamics simulations. For the pressure correction equation, the PRESTO! (Pressure Staggering Option) scheme was selected, known for its ability to enhance accuracy in pressure distribution. To further refine the simulation, a second-order scheme was applied to correct both momentum and energy equations, improving the fidelity of the results. Additionally, solution controls were carefully calibrated, with under-relaxation factors set at 0.3 for pressure, 0.2 for momentum, 0.9 for the liquid fraction update, and 1 for energy. These settings were chosen to ensure numerical stability and to facilitate convergence during the iterative solution process. The convergence criteria were meticulously monitored, with each time step rigorously checked to ensure that the solution met the required thresholds. A maximum of 150 iterations per time step was allowed, which was sufficient to guarantee convergence. This rigorous approach assures the validity and reliability of the numerical procedure, making it robust for further computations and capable of providing accurate predictions for the system under study.

III.5.3 Assumptions and Conditions

In order to write the equations governing the physical problem to be addressed, it is necessary to make a certain number of hypotheses.

- The thermophysical properties of the PCM are constant
- The problem is two-dimensional and transient.
- Conduction in the solid and liquid phases of the PCM is taken into account, particularly at the first moments. And even when a large part of the solid is in liquid form, conduction remains of the

order of magnitude of convection.

- Transfer by radiation is neglected.
- The flow is considered laminar, incompressible and natural convection is taken into account through the Boussinesq hypothesis.
- The liquid phase is Newtonian and incompressible.
- Given the heat transfer rates at the vertical walls, viscous dissipation is negligible.
- The PCM is pure. In reality, this condition is difficult to meet in commercial materials. This is, for the moment, a typical problem used to validate a numerical formulation.
- The MCP is homogeneous and isotropic.
- The expansion of the MCP is neglected and the phase change is isothermal.

III.5.4 Governing Equations

The design model used to simulate the melting of the PCM-RT-82 phase-change material by flow through these incompressible laminar layers incorporates a transient physical model to illustrate the dynamic behavior of the solid-to-liquid phase change. This model is based on the fundamental principles of conservation of mass and momentum, as well as energy conservation. Viscous dissipation is assumed to be negligible, simplifying the analysis by eliminating viscosity effects. The analysis considers the constant thermophysical properties of the molten phase-change material, as shown in Table 2, including viscosity, thermal conductivity, and specific heat. The effect of natural convection during the melting period is taken into account when applying the Bosinski approximation. This approximation applies when there are fluctuations in the buoyancy density; otherwise, the effect is neglected. The density change is expressed as follows [65]:

$$\rho = \rho_m (1 - \beta(T - T_m)) \quad (\text{III.84})$$

$$\rho_m = (\rho_s + \rho_l)/2$$

The equations utilized for this situation are continuity, momentum, and energy ones expressed as:
Continuity:

$$\partial_t(\rho) + \partial_i(\rho u_i) = 0 \quad (\text{III.85})$$

Momentum:

$$\partial_t(\rho u_i) + \partial_j(\rho u_i u_j) = \mu \partial_{jj} u_i - \partial_i p + \rho g_j + s_i \quad (\text{III.86})$$

Where S_i , the source term for the porosity function defined by [65], Is defined as:

$$S_i = c (1 - \gamma)^2 \frac{U_i}{\gamma^3 + \epsilon} \quad (\text{III.87})$$

The equations used in this study are analogous to Carman-Kozeny's equations for flow in porous media. In this context, the constant C represents the characteristic of the mushy zone's morphology. For this study, a value of C=105 was used [66]. Additionally, $\varepsilon = 0.001$ is employed to avoid division by zero issues.

Energy:

$$\partial_t(\rho h) + \partial_t(\rho \Delta H) + \partial_i(\rho u_i h) = \partial_i(k \partial_i T) \quad (\text{III.88})$$

The bellow equation defines enthalpy H as:

$$H = h + \Delta H \quad (\text{III.39})$$

Where sensible enthalpy h is defined as:

$$h = h_{ref} + \int_{T_{ref}}^T C_p \Delta T \quad (\text{III.90})$$

The liquid fraction of the PCM, γ , is generated during the phase change between the solid and liquid states when the temperature is between $T_L > T > T_S$, and can be expressed as:

$$\gamma = \Delta H / 2 \quad (\text{III.91})$$

$$\gamma = \begin{cases} 0 & \text{if } T < T_S \\ 1 & \text{if } T > T_L \\ \frac{(T - T_S)}{(T_L - T_S)} & \text{if } T_L > T > T_S \end{cases} \quad (\text{III.92})$$

The thermophysical properties of the nano-enhanced phase change material (NEPCM) were calculated using correlations derived from Maxwell's model [61]. The corresponding governing equations are presented as follows:

Boussinesq approximation was adopted to calculate the change in PCM density as a function of temperature in the liquid density given by:

$$\rho = \rho_0 [1 - \beta(T - T_0)] \quad (\text{III.93})$$

The corresponding relationship for buoyancy forces in the momentum equation is given by:

$$-\rho g = \rho_0 g [\beta(T - T_m) - 1] \quad (\text{III.94})$$

Where ρ_0 is the reference density at melting temperature T_m and β is the thermal expansion.

The density of NEPCM can be computed as:

$$\rho_{NEPCM} = \rho_{PCM} \left[(1 - \phi) + \phi \left(\frac{\rho_s}{\rho_{PCM}} \right) \right] \quad (\text{III.95})$$

Where ϕ is the volume fraction of nanoparticles.

The subscripts np, n, and p stand for NEPCM, nanoparticles, and PCM, respectively.

The heat capacity of the NEPCM is expressed as:

$$\rho C p_{NEPCM} = \rho C p_{PCM} \left[(1 - \phi) + \phi \left(\frac{\rho C p_s}{\rho C p_{PCM}} \right) \right] \quad (\text{III.96})$$

The effective dynamic viscosity of the NEPCM can be obtained using the Brinkmann model [31]:

$$\mu_{NEPCM} = \frac{\mu_{PCM}}{(1 - \phi)^{2.5}} \quad (\text{III.97})$$

The effective thermal conductivity of NEPCM can be computed using the Hamilton-Crosser model based on the Maxwell theory [60]:

$$\lambda_{NEPCM} = \frac{\lambda_s + (S - 1)\lambda_{PCM} - (S - 1)\phi(\lambda_{PCM} - \lambda_s)}{\lambda_s + (S - 1)\lambda_{PCM} + \phi(\lambda_{PCM} - \lambda_s)} \times \lambda_{PCM} \quad (\text{III.98})$$

Where S is the particle shape factor.

III.5.5 Selection of Nanofluids

The selection of nanoparticles is a critical step in the design of high-performance nanofluids for thermal energy applications. Their effectiveness depends not only on their intrinsic thermophysical properties but also on their ability to remain uniformly dispersed within the base fluid throughout the operating period. Consequently, nanoparticles should combine high thermal conductivity with low density, good chemical and physical stability, and reasonable cost to ensure both efficient heat transfer and practical applicability.

Considering these requirements, aluminum oxide (Al_2O_3), copper oxide (CuO), and titanium dioxide (TiO_2) were chosen as the dispersed solid particles in the present investigation. These nanoparticles have been extensively employed in thermal engineering because of their favorable thermal characteristics and compatibility with water-based nanofluids. The effective thermophysical properties required for the numerical simulations were evaluated using the predictive correlations developed by Khanafer et al. [58], which account for the influence of nanoparticle concentration on the behavior of the nanofluid. The fundamental properties of the selected nanoparticles are summarized in Table III.7.

Table III. 7 - Thermophysical properties of nanofluids

Property	Nanoparticle (Al_2O_3)	Nanoparticle (CuO)	Nanoparticle (TiO_2)
C_p ($\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$)	765	535.6	686.2
ρ ($\text{kg} \cdot \text{m}^{-3}$)	3970	5600	4250
k ($\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$)	40	20	8.9538
β (K^{-1})	0.85×10^{-5}	1.74×10^{-5}	0.9×10^{-5}

III.5.6 Boundary conditions

the beginning of the simulation, (PCM) is considered to be entirely in the solid phase with a uniform initial temperature of 300 K. Melting is initiated by maintaining the wall in direct contact with the heat transfer fluid (HTF) at a constant temperature of 363 K throughout the simulation. Since the thermal response of the storage unit is strongly affected by the imposed thermal conditions, suitable boundary conditions were assigned to the triplex-tube heat exchanger and the embedded fins. These conditions were selected to evaluate their influence on heat transfer enhancement and the melting characteristics of the PCM. The boundary conditions adopted in the present numerical model are summarized as follows:

$$r = ri \rightarrow T = T_{HTF} \quad (III.99)$$

$$r = rm \rightarrow \frac{\partial T}{\partial r} = 0 \quad (III.100)$$

The boundary conditions adopted exterior:

$$r = ri \rightarrow \frac{\partial T}{\partial r} = 0 \quad (III.101)$$

$$r = rm \rightarrow T = T_{HTF} \quad (III.102)$$

Heating both sides method:

$$r = ri \rightarrow T = T_{HTF} \quad (III.103)$$

$$r = rm \rightarrow T = T_{HTF} \quad (III.104)$$

Initial temperature for all models:

$$t = 0 \rightarrow T = T_{Initial} \quad (III.105)$$

III.6 Conclusion

Based on the previous literature review, it can be concluded that combining advanced fin geometries with nanoparticle-enhanced PCMs represents a promising solution to accelerate the melting process. Consequently, the next chapter presents the mathematical and numerical modeling of the proposed solar thermal energy storage system.

The fourth case, which yielded positive results in terms of fusion time and heat distribution, allowed us to calculate the outlet temperature of the collector. This calculation concerns the connection between the collection section and the storage section.

In this chapter, the mathematical and numerical models developed or utilized to study and analyze the phenomena related to the research problem were presented. The governing equations

describing the physical system were outlined, along with the assumptions made to simplify and analyze the problem. Additionally, the numerical methodologies employed, such as the finite element method or other relevant approaches, were discussed.

The models were validated by comparing the simulation results with experimental data (if available) or previous studies, demonstrating their ability to accurately represent the physical behavior of the problem. These models serve as essential tools for understanding the studied phenomenon and analyzing the impact of various parameters, paving the way to achieving the study's objectives and future recommendations.

This analysis sets the stage for the next chapter, where the results will be presented, and the models will be used to investigate different scenarios and achieve the desired outcomes of the research.

Chapter IV

Results and Discussion

IV.1 Introduction

This section presents the results and analysis of the study related to the solar radiation system and the thermal energy storage system. Regarding the solar system, the flat plate solar collector model was validated through comparison with experimental results, allowing the accuracy of the adopted model to be evaluated. The results of solar radiation and the outlet fluid temperature of the collector were also presented and analyzed. In addition, the effect of nanoparticles on the thermophysical properties of the heat transfer fluid and on the performance of the solar collector was investigated.

As for the thermal energy storage system, Grid independence and time-step sensitivity analyses were carried out, together with model validation procedures, in order to ensure the accuracy and reliability of the simulation for all parameters affecting the melting process of the phase change material (PCM). Subsequently, a (CFD) model was developed to investigate the melting behavior of RT82 under the considered configuration.

Furthermore, the effects of various physical and operating parameters on the melting behavior of the PCM were analyzed with the aim of improving the thermal performance of the heat exchanger and the efficiency of thermal energy storage. The obtained results were compared with data available in the scientific literature for both steady-state and transient systems in order to validate the adopted numerical models.

In addition, the influence of different innovative fin configurations and operating conditions on the evolution of the melting process and heat transfer enhancement was examined. Based on the finite volume method, an enthalpy-temperature model was employed to study the thermal characteristics of the PCM under different heating conditions and for various material types. These characteristics include the evolution of the solid-liquid interface, liquid fraction, velocity field, total melting time, and average thermal energy storage rate.

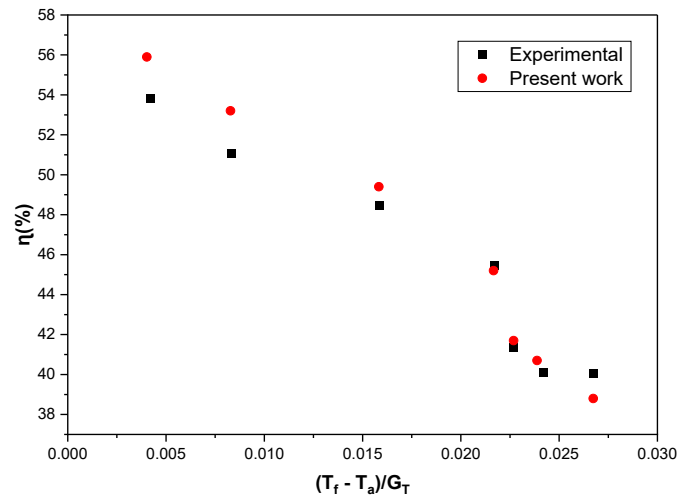
IV.2 Validation of the Flat plate Solar collector system

The present model of the flat-plate solar thermal collector was validated by comparing its predictions with the experimental and theoretical results reported by Saini et al. [41]. A comparative analysis was conducted using seven different test cases. Two key performance parameters were evaluated: the outlet temperature and the thermal efficiency of the collector. As shown in Table IV.1, the obtained results are in good agreement with the reference data, demonstrating the accuracy and reliability of the proposed model for predicting the thermal performance of the flat-plate solar collector.

Table IV. 1 - Validation model of flat plate solar collector with experimental results [67]

Case	E_g ($W \cdot m^{-2}$)	T_{am} ($^{\circ}C$)	T_{in} ($^{\circ}C$)	T_{out} ($^{\circ}C$)		Efficiency (%)	
				Our work	EXP	Our work	EXP
1	958.66	34.31	38.17	45.046	41.54	0.579	0.5385
2	990.06	33.15	48.82	54.906	52.14	0.494	0.4844
3	913.51	34.57	54.36	59.511	57.14	0.452	0.4552
4	933.63	22.39	47.35	51.855	49.80	0.388	0.4000
5	939.42	30.66	51.97	56.852	54.12	0.417	0.4124
6	887.81	22.82	44.02	48.452	46.44	0.407	0.4026
7	969.43	15.06	23.09	29.208	26.39	0.532	0.5112

Figure IV.1 illustrates a comparison between the numerical results obtained in the present work and the experimental data reported by Saini et al. [41] , for the flat-plate solar collector, showing good agreement between the two sets of results.

**Figure IV. 1** - Validation of the flat-plate solar collector model against experimental results

IV.3 Validation of the storage unit system

The numerical model developed in this study was designed to simulate the melting process of phase change material RT-82 in a triplex-tube heat exchanger used for latent heat thermal energy storage. The thermal energy storage unit was validated by comparing the average temperature distribution predicted by the present model with the experimental and numerical results reported by Sohif Mat et al. [29]. The validation was performed using the same physical configuration, thermophysical properties, metal properties, and boundary conditions adopted in the reference study. The validation results for the triplex-tube heat exchanger show a close agreement between the present numerical predictions and the experimental and numerical data reported in the literature, as illustrated in Figure IV.2. In particular, the temperature distribution profiles and the evolution of

the liquid fraction confirm the reliability of the proposed numerical approach and demonstrate its accuracy in predicting the thermal behavior of the storage system. Therefore, the validation results confirm that the developed model can accurately simulate the heat transfer and phase change phenomena within the thermal energy storage unit.

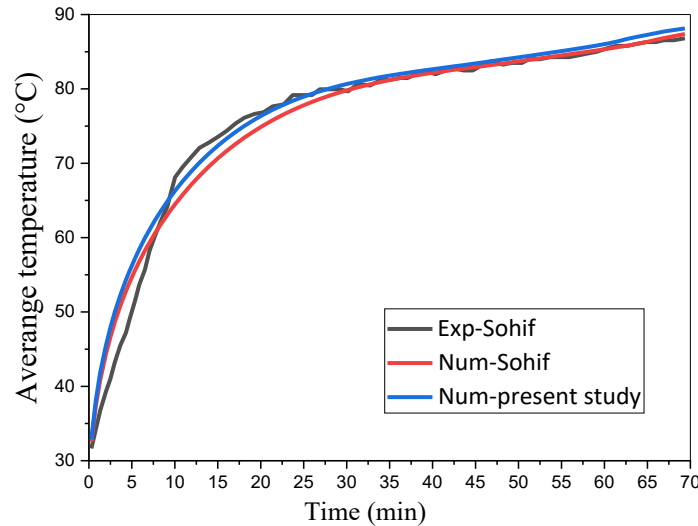


Figure IV. 2 - Validation with experimental and numerical results by Sohif Mat.

IV.4.1 Effect of different nanofluids on the thermal performance of solar collector

To evaluate the impact of different nanofluid types on collector performance, we calculated the thermophysical properties of the nanofluids, as shown in the Table IV.2, and examined three materials: copper (Cu), aluminum oxide (Al_2O_3), and titanium dioxide (TiO_2). The results showed that TiO_2 outperformed the other simulated materials. Therefore, the results for TiO_2 will be presented in the following sections.

Table IV. 2 - Thermophysical properties of nanofluids

Property	Base fluid (water)	Nanofluid (Al_2O_3 + H_2O)	Nanofluid (Cu + H_2O)	Nanofluid (TiO_2 + H_2O)
C_p ($\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$)	4179	$3,805 \times 10^3$	$3,356 \times 10^3$	$3,772 \times 10^3$
ρ ($\text{kg} \cdot \text{m}^{-3}$)	997.1	$1,086 \times 10^3$	$1,235 \times 10^3$	$1,095 \times 10^3$
k ($\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$)	0.613	0,688	0,691	0,676
μ ($\text{kg} \cdot \text{m}^{-1} \cdot \text{s}^{-1}$)	$8,91 \times 10^{-4}$	$9,578 \times 10^{-4}$	$9,578 \times 10^{-4}$	$9,578 \times 10^{-4}$
β ($1 \cdot \text{K}^{-1}$)	21×10^{-5}	$2,042 \times 10^{-4}$	$2,042 \times 10^{-4}$	$2,04 \times 10^{-4}$

Figure IV.4 illustrates the influence of nanofluid type on the thermal performance of the flat-plate solar collector. Three nanoparticles CuO, Al_2O_3 , and TiO_2 were evaluated and compared with the base fluid. The results indicate that the Al_2O_3 /water nanofluid achieves the highest collector efficiency, followed by TiO_2 /water and CuO/water, while the base fluid exhibits the lowest

performance. This enhancement is mainly attributed to the superior thermophysical properties of the nanofluids, particularly their higher thermal conductivity, which promotes more effective heat transfer between the absorber plate and the heat transfer fluid, thereby increasing the collector's thermal efficiency.

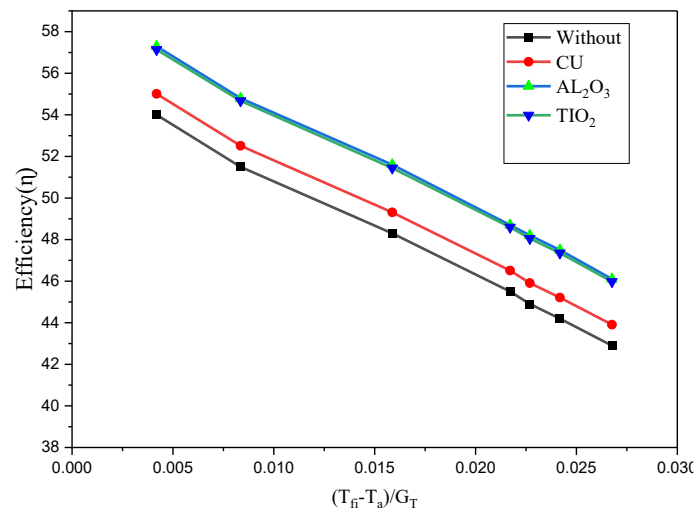


Figure IV. 3 - Efficiency of solar collector without and with nanoparticles.

IV.4.2 Effect of solar radiation on the outlet temperature of the solar collector

Figure IV.5 illustrates the outlet temperature of the solar collector as a function of solar time (throughout the day) for the simulated typical days. For all four typical days, it was observed that the temperature follows the daily solar radiation pattern. Additionally, it was noted that the inclusion of nanoparticles significantly increases the solar collector's temperature, particularly at noon.

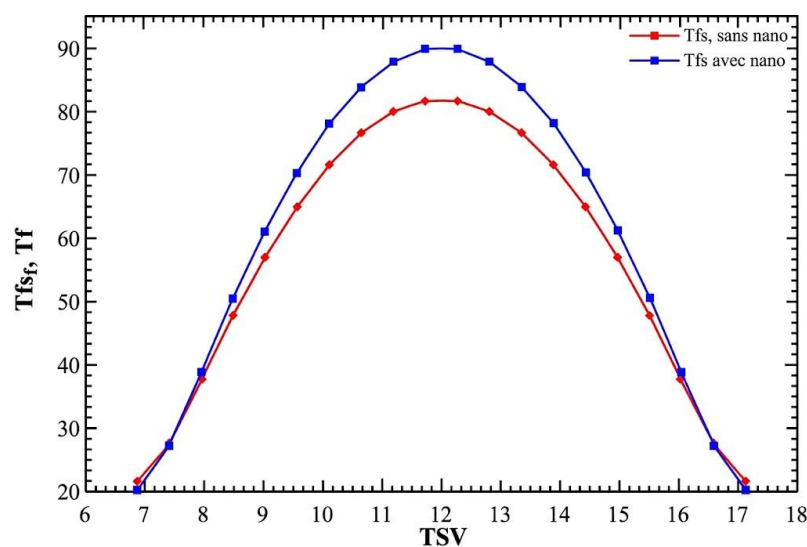


Figure IV. 4 - Outlet temperature of the collector with and without nanoparticles

IV.5 Results of numerical simulation for thermal energy storage

IV.5.1 Time step and grid independency tests

To ensure that the numerical predictions are independent of the computational mesh, a grid independence study was performed prior to the main simulations. The objective of this procedure was to determine a mesh density capable of producing stable and accurate numerical results while avoiding unnecessary computational cost. Several mesh configurations with different numbers of computational cells were generated and compared, as illustrated in Figure IV.6 .

The selected meshes contained approximately 11,799, 13,000, and 17,692 elements. For each mesh, the numerical solution was monitored and compared to evaluate the influence of mesh refinement on the predicted thermal characteristics. The results indicate that refining the mesh beyond 17,692 elements produces only negligible variations in the calculated quantities, demonstrating that the solution has reached mesh independence. Consequently, the mesh consisting of 17,692 elements was adopted for all subsequent simulations because it provides an appropriate compromise between numerical accuracy, solution stability, and computational time.

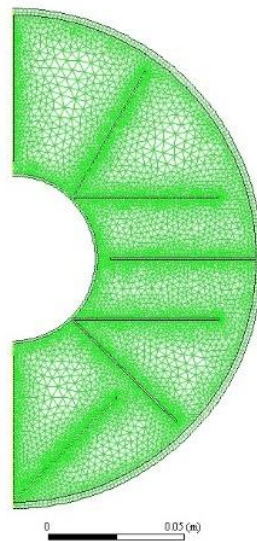


Figure IV. 5 - Mesh Distribution Model

A time-step independence analysis was conducted to verify that the transient numerical solution was not influenced by the selected temporal discretization. Choosing an appropriate time step is essential for accurately capturing the evolution of the melting process while maintaining reasonable computational requirements. An excessively large time step may overlook rapid thermal variations and reduce numerical accuracy, whereas an excessively small value significantly increases the computational time without providing a proportional improvement in the solution.

To identify the optimum temporal resolution, three time-step sizes of 1.0, 0.5, and 0.3 s were evaluated, as illustrated in Figure IV.7. The predicted thermal response obtained with each time step was compared to determine the sensitivity of the numerical solution to temporal refinement. The results indicate that reducing the time step from 1.0 s to 0.3 s improves the accuracy of the transient solution, while further refinement would produce only marginal changes at the expense of additional computational cost. Therefore, a time step of 0.3 s was selected for all subsequent simulations, as it provides a satisfactory balance between numerical accuracy, solution stability, and computational efficiency.

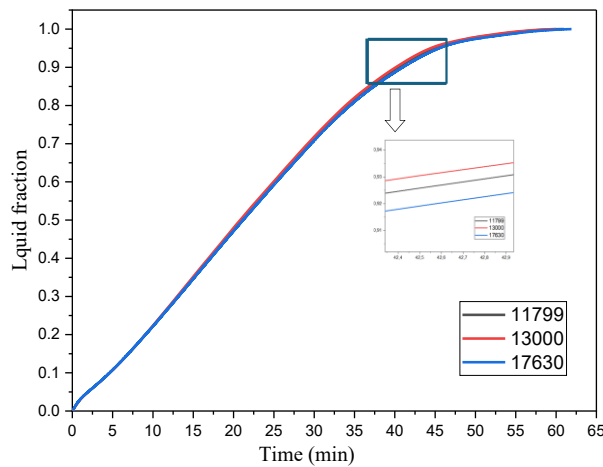


Figure IV. 6 - Mesh size independency test

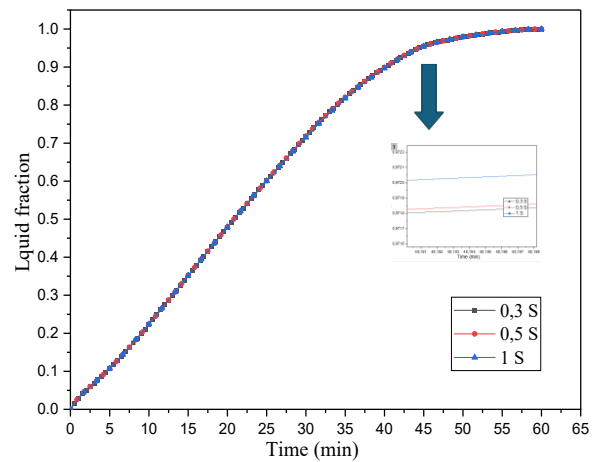


Figure IV. 7 - Time Step Sensitivity Test

IV.5.2 CFD simulation with fin's configuration

The numerical simulation focuses on enhancing the melting process of (PCMs) through computational fluid dynamics (CFD) analysis. Two new configurations of triplex-tube heat exchangers are investigated. The first configuration incorporates T-shaped fin designs, while the second employs newly proposed shaped-fins with different geometrical shapes.

These configurations are used to evaluate the influence of key geometric parameters, including fin length, width, and thickness, in order to identify the most efficient fin design for improving heat transfer performance. The simulation results are presented in terms of temperature distribution contours and liquid fraction evolution for both configurations.

Furthermore, to assess the effects of the heat transfer fluid and different types of (PCMs), the numerical results are presented graphically and analyzed in detail. This comparison provides a clear understanding of the thermal behavior and the effectiveness of each configuration in enhancing the melting process.

IV.5.2.1 T-shaped fin geometric properties

IV.5.2.1 The effect of T-shaped fin length on the melting process

Figure IV.9 illustrates the effect of the T-shaped fin length (L) on the liquid fraction and average temperature distribution during the melting process. The simulations were conducted at a constant HTF temperature of 363 K and a fixed fin width of 33 mm, while the investigated fin lengths were 10, 20, 30, and 40 mm. The results indicate that the fin length has a significant influence on the melting behavior and thermal performance of the PCM system. Among all investigated cases, the configuration with a fin length of 30 mm (Case 3) exhibited the shortest melting time of 71 min, indicating the highest heat transfer enhancement. In addition, this case achieved the highest average temperature distribution, which accelerated the melting process and improved the thermal energy storage rate. Although increasing the fin length generally enhances heat conduction, excessive fin length does not necessarily improve the melting performance due to the possible reduction in natural convection effects within the PCM domain. Therefore, the T-shaped fin length of 30 mm can be considered the optimal configuration for the present study.

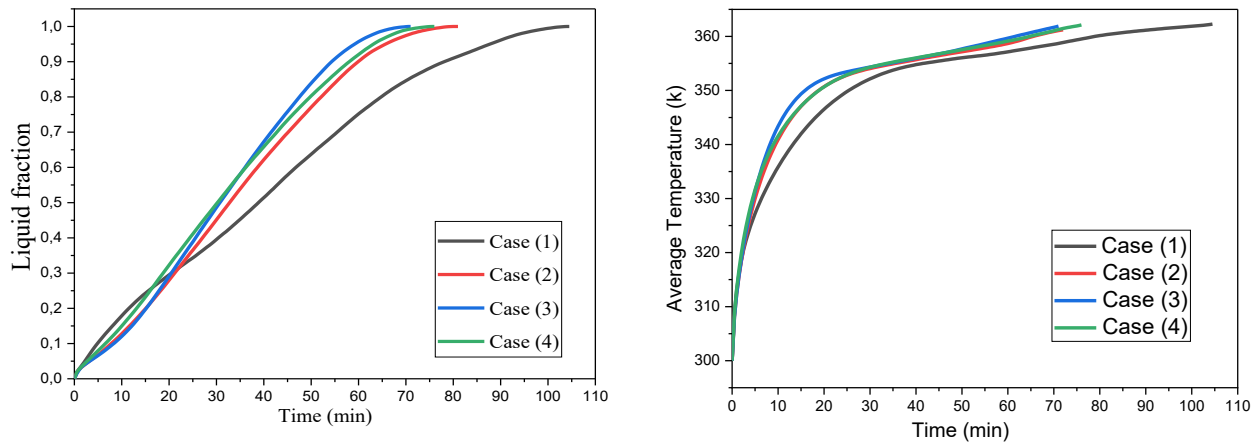


Figure IV. 8 - Variation of liquid fraction and average temperature of melting time function for different fin lengths at 33mm widths

IV.5.2.2 The T-shaped fin width's effect of on the liquid fraction

Figure IV.10 illustrates the effect of the T-shaped fin width (w) on the liquid fraction and average temperature distribution during the melting process. The simulations were performed at a constant HTF temperature of 363 K and a fixed fin length of 30 mm, while the investigated fin widths were 33, 40, and 45 mm. The results demonstrate that increasing the fin width significantly reduces the melting time of the PCM. Among all investigated configurations, the fin width of 45 mm (Case 6) achieved the shortest melting time of 33.5 min, indicating the best thermal performance. Furthermore, this case exhibited the highest average temperature distribution, which

accelerated the melting process. The enhancement in thermal behavior can be attributed to the increase in the heat transfer surface area between the fin and the PCM, resulting in improved heat conduction and faster thermal energy propagation within the storage unit. Therefore, the T-shaped fin width of 45 mm can be considered the optimal configuration for enhancing the melting rate and reducing the charging time in the present study.

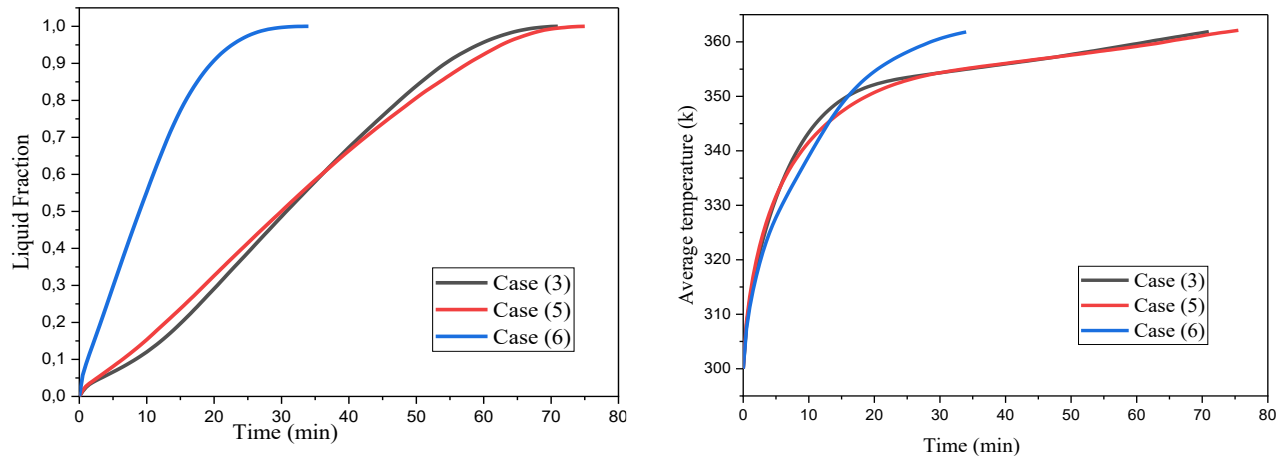


Figure IV. 9 - Variations in liquid fraction and the average temperature with melting time for fins of different widths fin at a length of 30 mm.

Table IV. 3 - Melting time as a function of different fin lengths at 33mm widths

Case	Case 1	Case 2	Case 3	Case 4
Width (mm)	33	33	33	33
Length (mm)	10	20	30	40
Melting time (min)	104,5	74,5	71	76

IV.5.2.3 The effect of T-shaped fin thickness on the liquid fraction

Figure IV.11 illustrates the effect of fin thickness on the PCM melting time for three different thickness values (0.5, 1, and 2 mm) while maintaining the optimal fin length and width configurations at an HTF temperature of 363 K. The results indicate that the melting time decreases as the fin thickness increases, revealing an inverse relationship between fin thickness and melting duration. The numerical simulation results of the liquid fraction evolution demonstrate that fin thickness has a significant influence on the melting performance of the PCM system. Increasing the fin thickness enhances heat conduction through the fin structure, which improves heat transfer from the HTF to the PCM and accelerates the melting process. Consequently, thicker fins contribute to reducing the charging time and improving the thermal performance of the storage unit.

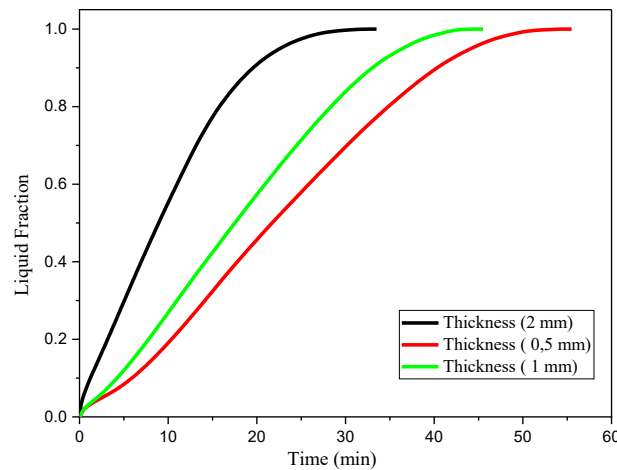


Figure IV. 10 - Effect of T-shaped fin thickness on PCM melting behavior

IV.5.2.4 Contours of length t-shaped fin improvement of heat transfer

The contours of the effect fin length on the liquid fraction melting PCM in the triplex tube heat exchanger. With the heating internal-external method the fin's lengths of (10, 20, 30, and This study demonstrates the effect of fin lengths, chosen based on the phase change material (PCM) melting ratio variable, in the calculation of a three-tube heat exchanger. Based on the internal-external heating method[29], the selected fin lengths (10, 20, 30, and 40 mm) were analyzed. Simulations showed different melting times (15, 30, 45, and 60 minutes), as illustrated in Figure IV.11. It is observed that the liquid initially appeared on both the inner and outer surfaces of the tube, including the fins. Furthermore, a clear average rotation was observed at the bottom of the tube. The liquid increased over time, developing at the top of the tube after 45 minutes. At 71 minutes, the annular spaces were covered with varying melting ratios. The 30 mm finned heat exchanger exhibited the best performance compared to the other lengths. Table IV shows the melting times for each case.

Figure IV.12 shows the liquid fraction contours for fins of different widths at various time intervals (15, 30, 45, and 60 minutes). In 15 min, a few liquid fractions were observed on internal and external surfaces and fins. After 30 minutes, the liquid fraction increased rate, and circles appeared clearly in the low part of PCM on case 3 and case 4. As for case 6, The PCM has been completely melted. The annular spaces are covered with a liquid fraction in case 3 and case 4 for 45 minutes, and the PCM has completely melted in both cases at 60 minutes. The contours of the liquid fraction illustrate the critical role of supply in the melting process, with the 45mm fins notably influencing the overall melting time. The numerical model's predictions for melting duration, as detailed in Table IV.4, further underscore the impact of fin size on the efficiency of the phase change process.

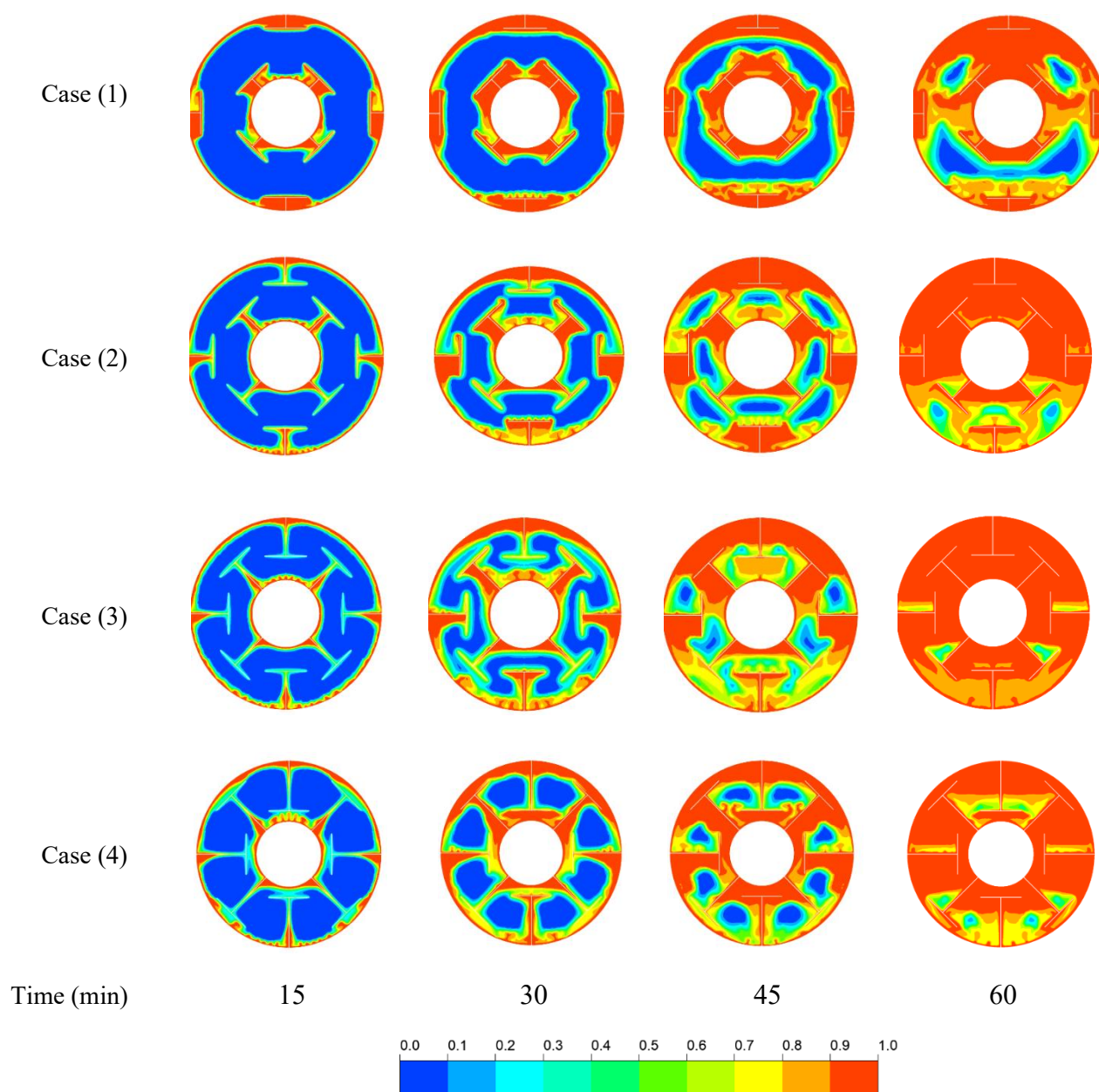


Figure IV. 11 - The liquid fraction's contours with variation of length and fixed width

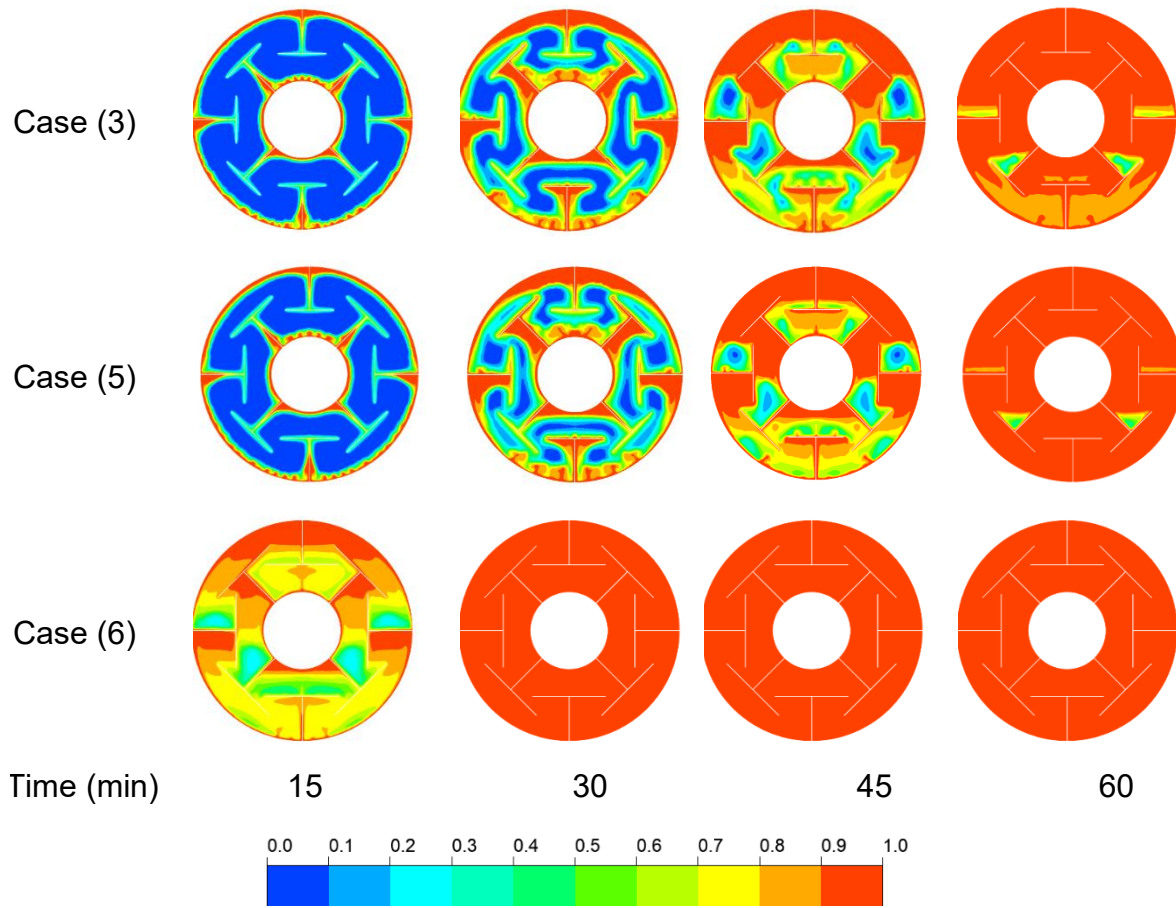


Figure IV. 12 - Contours of liquid fraction with variation of width and a fixed length

Table IV. 4 - Melting time for various widths, with a fixed length of 30 mm.

Case	Length (mm)	Width (mm)	Melting time (min)
03	30	33	71
05	30	40	75
06	30	45	33,5

IV.5.2.5 Contours of distributions of the average temperature

Figure IV.14 shows the melting PCM's temperature average contours with various length cases. In all cases, heat transfer to internal and external surfaces and fins at 15 min. The average temperature increased over time in the presence of fins of 30 mm in length compared to other lengths at a time 30 min. the annular spaces are covered with convection on the case (3), compared to other cases at 60 minutes. Using fins 30 mm in length, thermal equilibrium in all parts of the PCM since a part of the PCM. The fin's length distributes heat more evenly, assisting in the PCM's melting. After then, the temperature took 60 minutes to reach thermal equilibrium.

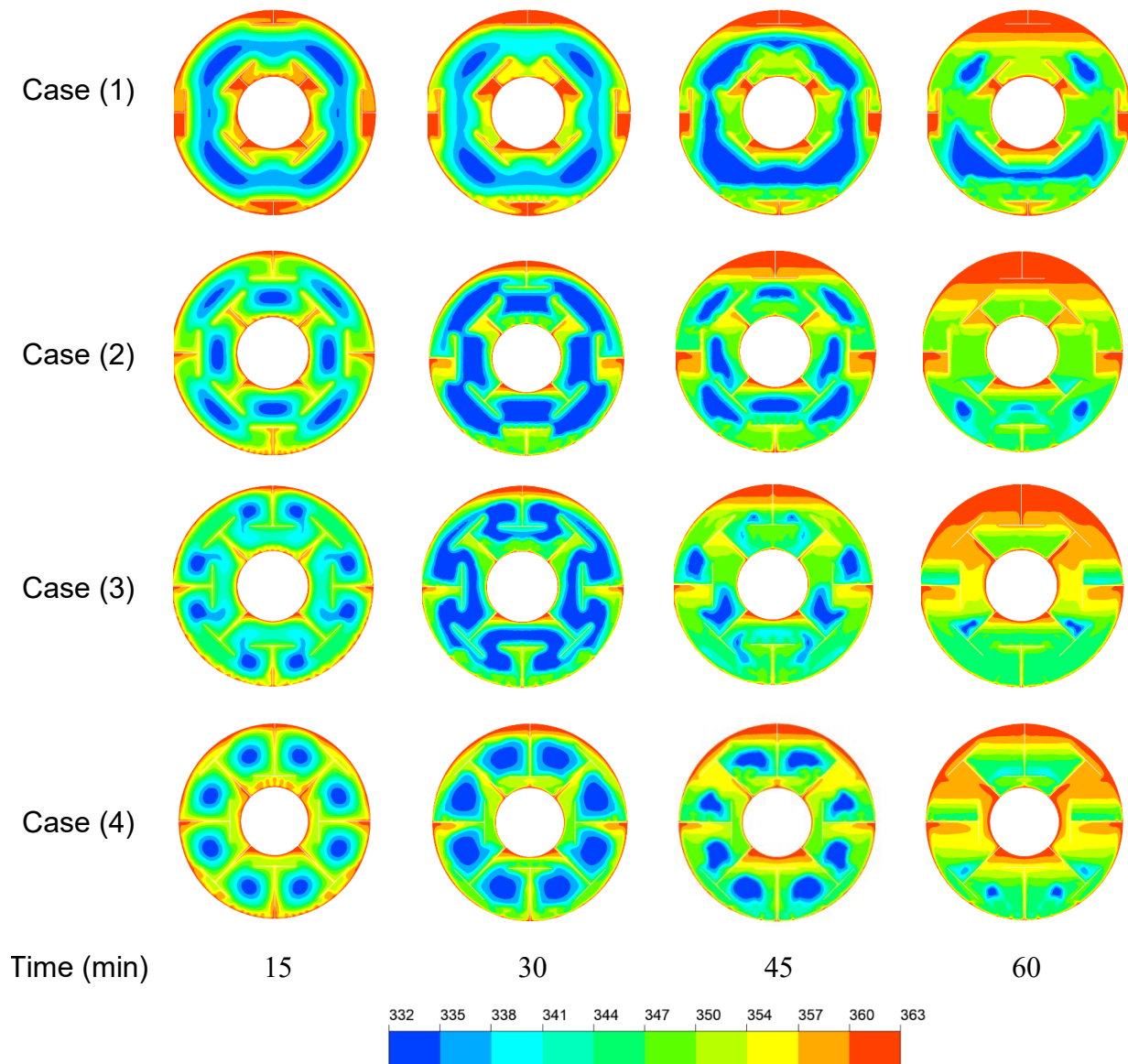


Figure IV. 13 - Average temperature contours with variation of length at a fixed width

Figure IV.15 represents the various temperature of (33, 40, and 45 mm), through the times (15, 30, 45, and 60 min). over time in 15 min was temperature distribution near the surface of tubes on both sides and fins in cases (3) and (5), but in case 6, the temperature distribution in all domains, especially in part highest. At 30 min, the temperature in the same zone in the heat exchanger increased. At 45 min, the temperature was concentrated in the higher region for both case (3) and (5). For case (6), which has a width of 45 mm, the total PCM melted. At 60 minutes, the temperature spread out more in the upper part in case (5) with a width of 40 mm more than in case (3) with a width of 33 mm. The 45mm fin width is important in spreading and distributing the temperature more uniformly, indicating that most of the PCM melting due to surface contact area increases.

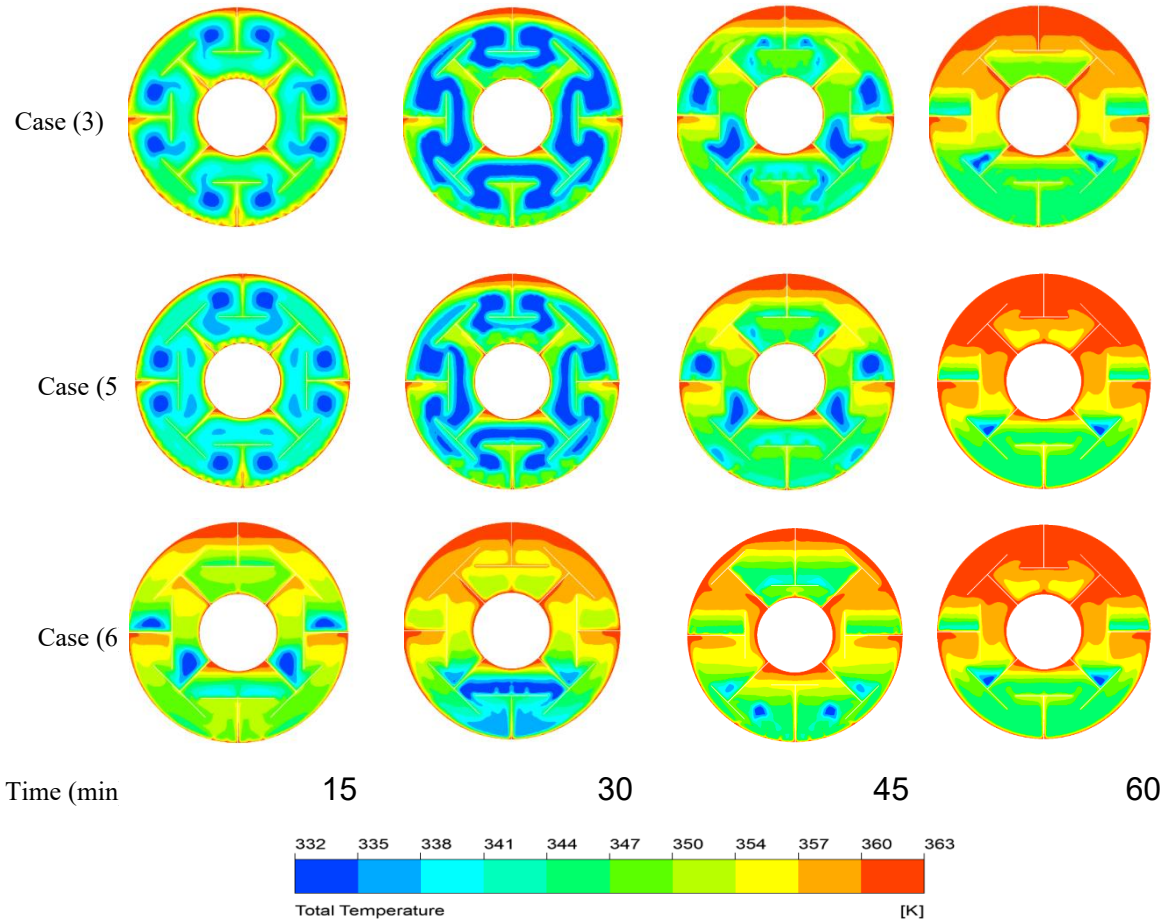


Figure IV. 14 - Average temperature contours with width variation, fixed length

IV.5.2.2 new-shaped fin geometric properties

IV.5.2.2.1 Effect of new-shaped fin length on the melting process

Figure IV.16 presents the influence of fin length on the evolution of the liquid fraction and the average PCM temperature during the charging process. Four fin lengths (30, 35, 40, and 45 mm) were evaluated while maintaining the heat transfer fluid temperature at a constant value of 363 K. The results demonstrate that the melting behavior is highly dependent on the fin geometry, with noticeable differences in both the charging rate and the thermal response of the storage unit. Extending the fin length progressively enhances the melting process by improving the distribution of thermal energy within the PCM. Longer fins provide a larger contact surface between the metallic structure and the storage material, allowing heat to penetrate deeper into the PCM and reducing the distance over which heat must be conducted. Consequently, the temperature field becomes more uniform, thermal resistance decreases, and the phase transition proceeds at a faster rate.

Among the investigated configurations, the 45 mm fin (Case D) exhibits the highest thermal performance. This configuration achieves complete melting in approximately 32.5 min, which is

considerably shorter than the melting times obtained with the shorter fins. Furthermore, it produces the highest average PCM temperature throughout the charging process, indicating a more efficient utilization of the supplied thermal energy. The improvement is attributed to the combined effects of enhanced conductive heat transfer through the extended fin surface and stronger buoyancy-driven convection as the liquid region expands.

Based on these findings, a fin length of 45 mm represents the optimum configuration among the tested cases. It provides the most favorable balance between heat transfer enhancement and charging time, making it the preferred design for the subsequent numerical investigations conducted in this study.

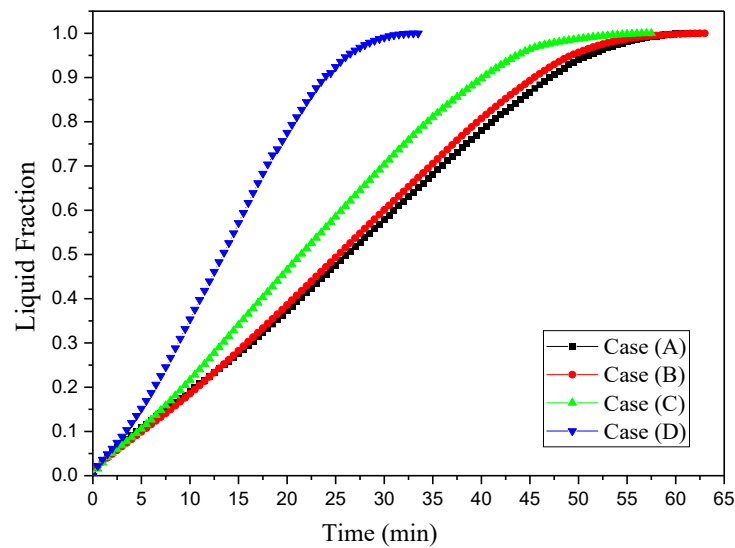


Figure IV. 15 - Effect of the length on the melting PCM

IV.5.2.2.2 The effect of new-shaped fin thickness on the liquid fraction

Figure IV.17 presents the effect of fin thickness on the melting duration of the phase change material (PCM) by considering three different thicknesses (0.5, 1, and 2 mm), while keeping the fin length and the wall temperature constant at 363 K. The results show that increasing the fin thickness leads to a slight decrease in the total melting time. This trend is mainly associated with the improved heat conduction through thicker fins, which promotes a more efficient transfer of thermal energy from the heated wall to the PCM. However, the influence of fin thickness remains relatively weak, as the recorded melting times are only marginally different, reaching approximately 56, 55, and 50 min for fin thicknesses of 0.5, 1, and 2 mm, respectively. These observations indicate that, under the present operating conditions, fin thickness does not play a dominant role in controlling the melting process. Therefore, selecting thinner fins appears to be a more practical design option, since it increases the available volume for PCM inside the heat

exchanger, leading to a higher thermal energy storage capacity and better utilization of the storage material without causing a noticeable reduction in the overall thermal performance.

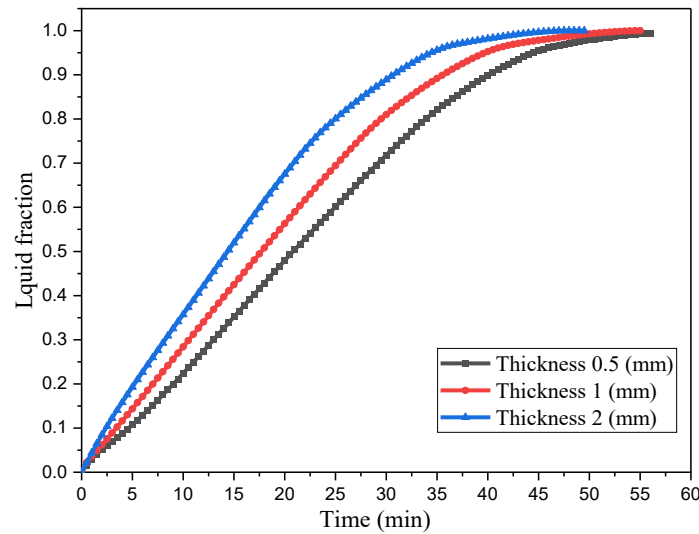


Figure IV. 16 - Effect of fin thickness on the melting phase change material.

IV.5.2.2.3 Contours of length new-shaped fin improvement of heat transfer

Figure IV.18 compares the evolution of the liquid fraction in the triplex-tube latent heat thermal energy storage unit for four fin lengths (30, 35, 40, and 45 mm) under internal–external heating conditions, following the configuration proposed by Sohif et al. [29]. The liquid fraction contours are presented at four charging times (15, 30, 45, and 60 min) to illustrate the influence of fin geometry on the melting process.

At the early stage of charging, melting is initiated at the surfaces directly exposed to the heat transfer fluid, including both the inner and outer tube walls as well as the fin surfaces. As thermal energy continues to penetrate the PCM, the melted region gradually extends toward the core of the storage domain through heat conduction. Once a sufficient volume of liquid PCM is formed, buoyancy forces induce natural convection currents, which become increasingly dominant and promote a faster redistribution of thermal energy within the annular space. This transition from conduction-dominated to convection-assisted heat transfer significantly accelerates the melting process.

The liquid fraction distributions also reveal that fin length strongly affects the progression of the melting front. Increasing the fin length enlarges the heat transfer surface and decreases the conduction distance between the heat source and the PCM, allowing thermal energy to reach previously inaccessible regions more efficiently. Consequently, the PCM melts more uniformly, and the remaining solid regions diminish more rapidly. In contrast, shorter fins leave larger portions of the storage medium dependent solely on heat conduction, resulting in slower melting and greater temperature non-uniformity.

Among the investigated configurations, the 45 mm fin achieves the highest melting rate and the shortest charging time, as summarized in Table IV.5. Its superior performance is attributed to the combined effect of an increased heat transfer area, lower thermal resistance, and stronger natural convection, all of which contribute to a more homogeneous temperature field and a significant enhancement in the thermal efficiency of the latent heat storage unit.

Table IV. 5 - Melting time with different fin lengths

CASE	A	B	C	D
Length (mm)	30	35	40	45
Melting time (min)	62.5	61.5	57.5	32

Figure IV.19 illustrates depicts the temperature field inside the latent heat thermal energy storage unit for different fin lengths (30, 35, 40, and 45 mm) at successive charging times. The thermal contours clearly demonstrate that fin geometry has a pronounced influence on the propagation of heat through the PCM. At the beginning of the charging process, the temperature distribution remains relatively uniform because the thermal energy has not yet penetrated deeply into the storage medium. As heating continues, the fins act as conductive pathways that transfer thermal energy from the heat transfer fluid toward the interior of the PCM, producing warmer regions adjacent to the fin surfaces before the remaining volume is progressively heated.

As the phase transition advances, buoyancy-driven flow becomes increasingly important in the liquid PCM. The density variation associated with temperature gradients induces natural convection currents that transport heat from the upper heated regions to the colder zones, thereby accelerating the melting process. The effectiveness of this mechanism depends strongly on the fin length. Increasing the fin length enlarges the contact area between the metal fins and the PCM while simultaneously shortening the heat conduction path. Consequently, thermal energy is distributed more uniformly throughout the storage domain, reducing localized hot and cold regions and promoting a more homogeneous melting front.

In contrast, shorter fins provide limited thermal penetration into the PCM, restricting efficient heat transfer to the vicinity of the heat exchanger walls. As a result, significant temperature gradients persist in the central regions of the storage unit, increasing thermal resistance and delaying the progression of the melting front. Among the investigated configurations, the 45 mm fin exhibits the best thermal performance, as it achieves the most uniform temperature field, minimizes thermal stratification, and shortens the overall charging time. These observations confirm that fin length is

a key design parameter for improving the heat transfer characteristics and charging efficiency of

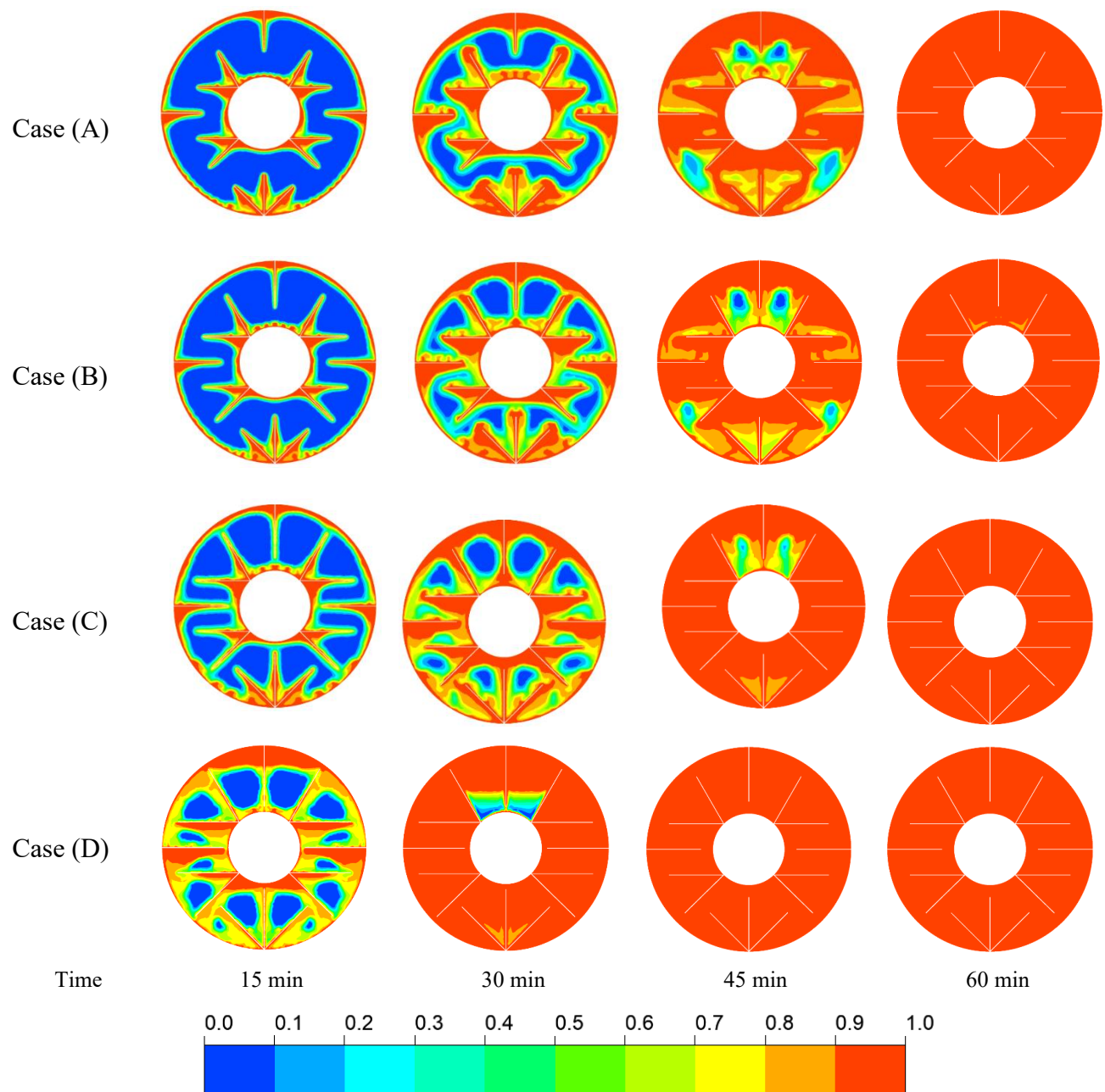


Figure IV. 17 - Liquid fraction contours with variation of length over time

latent heat thermal energy storage systems.

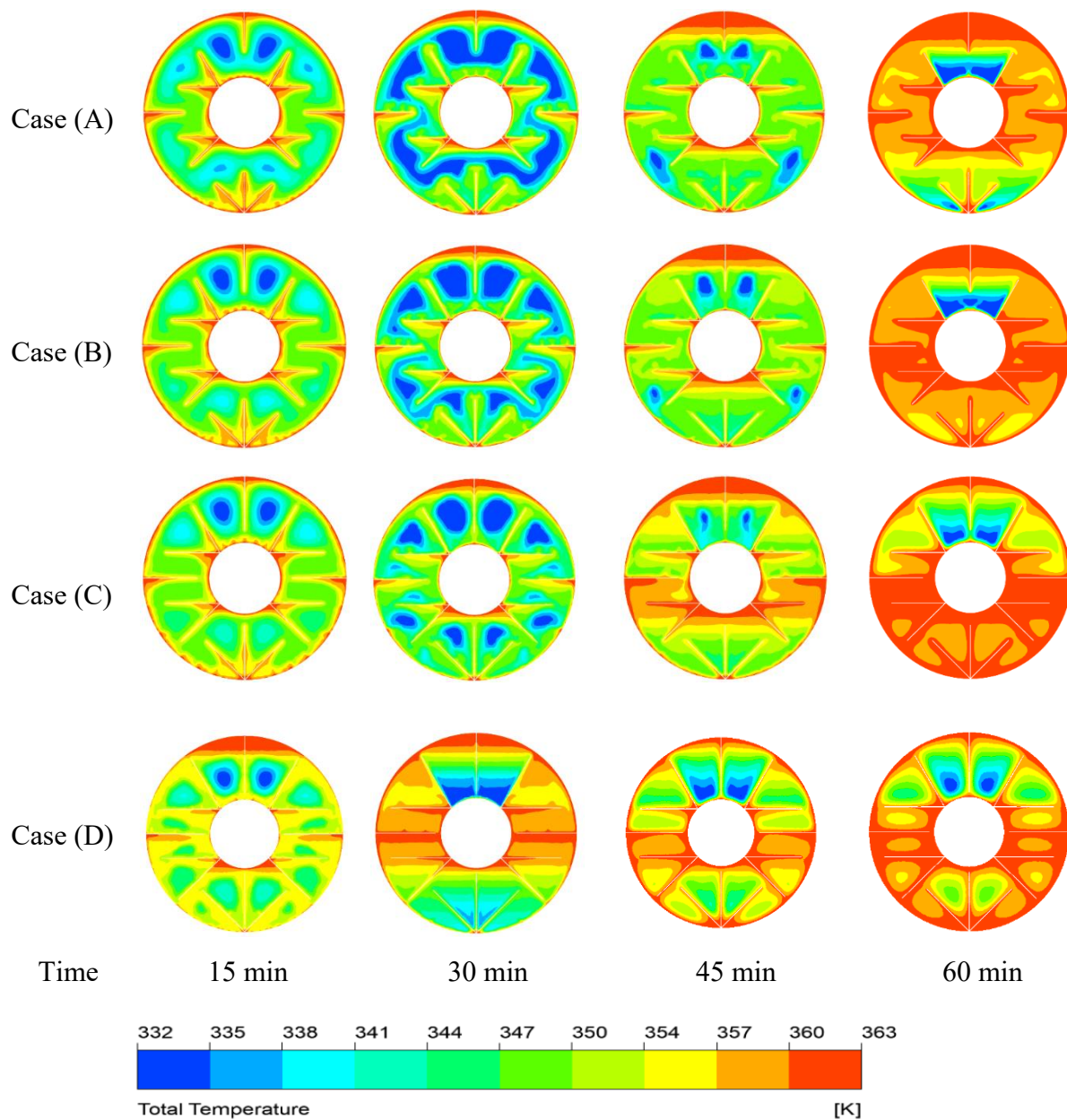


Figure IV. 18 - Average temperature contours with variation of length over time

IV.5.3 CFD simulation with PCM incorporating nanoparticles

Incorporating nanoparticles into (PCM) influences the coalescence behavior during melting, one of the most effective approaches for improving the thermal performance of phase change materials is the incorporation of highly conductive nanoparticles into the PCM matrix. This modification aims to overcome the inherently low thermal conductivity of conventional PCMs, which often limits the rate of heat absorption and release during the charging and discharging processes. By providing additional conductive pathways within the material, nanoparticles facilitate

more rapid heat diffusion, resulting in a more uniform temperature distribution and a faster phase transition.

In the present study, the influence of three widely used nanoparticles aluminum oxide (Al_2O_3), copper (Cu), and titanium dioxide (TiO_2) is systematically examined. To ensure a fair comparison, each nanoparticle is dispersed in the base PCM at a constant volume fraction of 3%, while all other operating conditions are kept unchanged. This approach makes it possible to evaluate the individual contribution of each nanoparticle to the melting behavior without the influence of concentration-related variations.

The comparative analysis focuses on the evolution of the melting process, the liquid fraction, and the overall charging performance of the thermal energy storage unit. Particular attention is given to identifying the nanoparticle that provides the greatest enhancement in heat transfer efficiency, thereby reducing the melting time and improving the thermal response of the PCM. The findings are expected to provide useful guidance for selecting suitable nano-enhanced PCMs for high-performance latent heat thermal energy storage applications.

IV.5.3.1 Thermophysical Property Evaluation of Paraffin RT82 Enhanced with Nanoparticles

The effective thermophysical properties of the paraffin RT82 mixed with nanoparticles (Al_2O_3 , CuO, TiO_2) were calculated using a weighted average model for density and specific heat capacity. Thermal conductivity was estimated using the Maxwell model, which accounts for the influence of nanoparticle concentration on heat transfer. The results in the table show that density and thermal conductivity increase with higher nanoparticle concentrations, while the effective specific heat decreases due to the lower specific heat of nanoparticles compared to paraffin. This enhancement improves the thermal storage performance by facilitating better heat transfer.

Table IV. 6 - Table thermophysical properties of the paraffin RT82 with nanoparticles (Al_2O_3 , CuO, TiO_2)

Material	Al_2O_3 [29]	CuO [68]	TiO_2
Density ($\text{kg}\cdot\text{m}^{-3}$)	3970	6500	4250
Specific Heat ($\text{J}/\text{kg}\cdot\text{K}$)	765	550	700
Thermal Conductivity ($\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$)	40	33	11

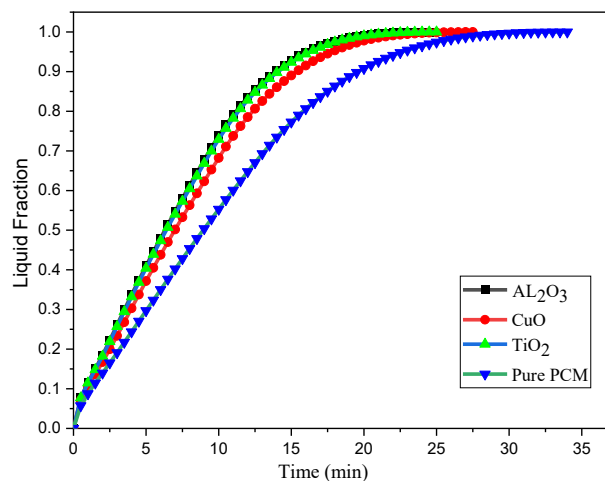
Effective Thermophysical Properties of RT82 with Nanoparticles (Based on Maxwell Model) and Effect of the concentration fraction volumes (1%, 2%, 3% and 4%)

Table IV. 7 - Concentration fraction volumes of nanoparticles

Nanoparticle	Volume Fraction (%)	Effective Density ($\text{kg}\cdot\text{m}^{-3}$)	Effective Specific Heat ($\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$)	Effective Thermal Conductivity ($\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$)
Al_2O_3	1%	987.2	1971.8	0.2047
	2%	1024.4	1944.3	0.2095
	3%	1061.6	1917.3	0.2143
	4%	1098.8	1890.8	0.2191
CuO	1%	1006.5	1960.7	0.2044
	2%	1063	1921.2	0.2089
	3%	1119.5	1883.4	0.2134
	4%	1176	1847	0.2179
TiO_2	1%	988	1970.8	0.2024
	2%	1026	1941.7	0.2049
	3%	1064	1913	0.2073
	4%	1102	1884.6	0.2097

IV.5.3.2 The T-shaped fin effect of nonadditive incorporation

The impact of nanoparticles on the fusion (charging) process was investigated by selecting three distinct nanoparticle materials: aluminum oxide (Al_2O_3), copper oxide (CuO), and titanium dioxide (TiO_2). Each material was analyzed at a fixed concentration of 3% in PCM. The curve depicted in Figure IV.20 highlights that aluminum oxide demonstrates the most pronounced enhancement in thermal transfer among the tested materials. This improvement is reflected in a significant increase in thermal conductivity, emphasizing its superior effectiveness in optimizing heat transfer during the fusion process.

**Figure IV. 19** - Liquid fraction of nano-MCPs as a function of time.

IV.5.6.3 The new new-shaped fins effect of nonadditive incorporation

The influence of nanoparticle incorporation on the melting characteristics of RT82 is illustrated in Figure IV.20 by comparing the liquid fraction evolution of pure PCM with nano-enhanced PCMs containing 3 vol.% of Al_2O_3 , Cu, and TiO_2 . A clear improvement in the charging process is achieved after adding nanoparticles, as evidenced by the shorter time required to reach complete melting. The base PCM exhibits the slowest thermal response, requiring nearly 3600 s to melt completely. In contrast, the addition of Al_2O_3 , TiO_2 , and Cu reduces the melting duration to approximately 2550 s, 2580 s, and 2790 s, respectively. The enhanced performance is attributed to the higher effective thermal conductivity of the nano-enhanced composites, which facilitates faster heat propagation within the PCM and reduces thermal resistance during the phase transition. Among the investigated nanoparticles, Al_2O_3 produces the greatest enhancement, providing the most rapid melting behavior. This superior performance indicates that Al_2O_3 is particularly effective in promoting heat transfer inside the storage medium, thereby improving the charging efficiency of the latent heat thermal energy storage system.

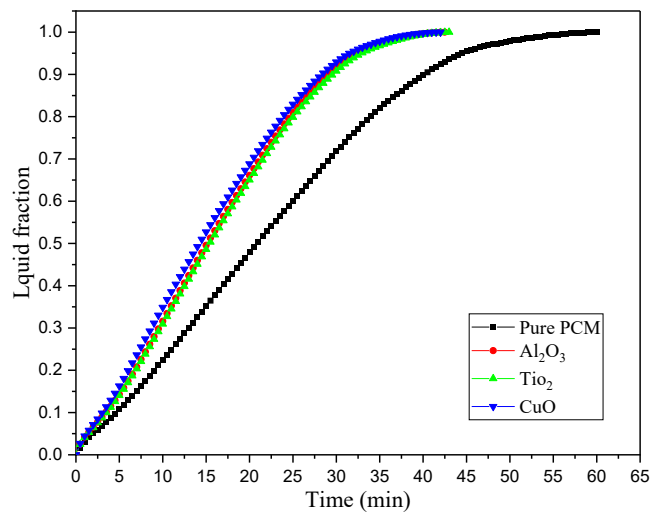


Figure IV. 20 - Variation of the liquid melting fraction according to the nanoparticles type

IV.5.4 The effect of (HTF) temperature on the pcm liquid fraction

Figure IV.22 illustrates the effect of the heat transfer fluid temperature (HTF) on the PCM melting time for three different temperatures: 363 K, 368 K, and 373 K, while keeping the optimal values of fin length, width, and thickness constant. The results indicate that the melting time decreases significantly with increasing HTF temperature. Specifically, the melting time is 33.5 min at 363 K, 24.5 min at 368 K, and 20.5 min at 373 K.

An inverse relationship between the PCM melting time and the HTF temperature can clearly be observed. This behavior is attributed to the enhancement of heat transfer caused by the increase in the temperature difference between the HTF and the PCM, which accelerates thermal energy transfer and promotes faster melting of the PCM. These results confirm that the HTF temperature has a significant influence on the PCM melting process and overall thermal performance of the storage system.

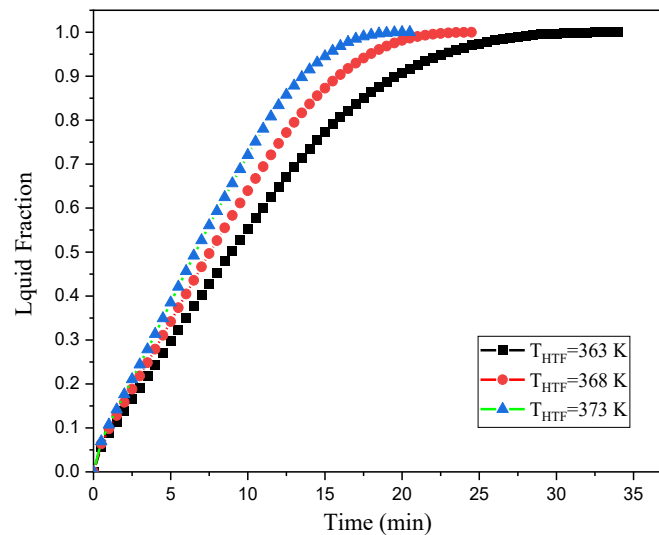


Figure IV. 21 - Variation of heat transfer fluid temperature on liquid melting of PCM

IV.5.5 The effect of PCM type on melting time

Figure IV.23 compares the melting behavior of three paraffin-based phase change materials, namely RT50, RT60, and RT82, under identical operating conditions with the heat transfer fluid maintained at 363 K. A noticeable difference in the melting duration is observed among the three materials. RT82 requires the longest time to complete the melting process (32.5 min), whereas RT60 and RT50 melt more rapidly, with melting times of 19.5 min and 16.5 min, respectively. This behavior is primarily associated with the melting temperature of each PCM relative to the temperature of the heat transfer fluid. Materials with lower melting temperatures experience a larger temperature difference, which enhances heat transfer and accelerates the phase change process. In contrast, RT82, whose melting point is higher, undergoes a slower melting process because the available thermal driving force is reduced. Although RT50 and RT60 offer shorter charging times, RT82 is capable of storing thermal energy at a higher temperature level, making it more suitable for applications where greater thermal storage capacity is required. Therefore, the choice of PCM should be based on the specific operating temperature and the desired balance between charging rate and energy storage performance.

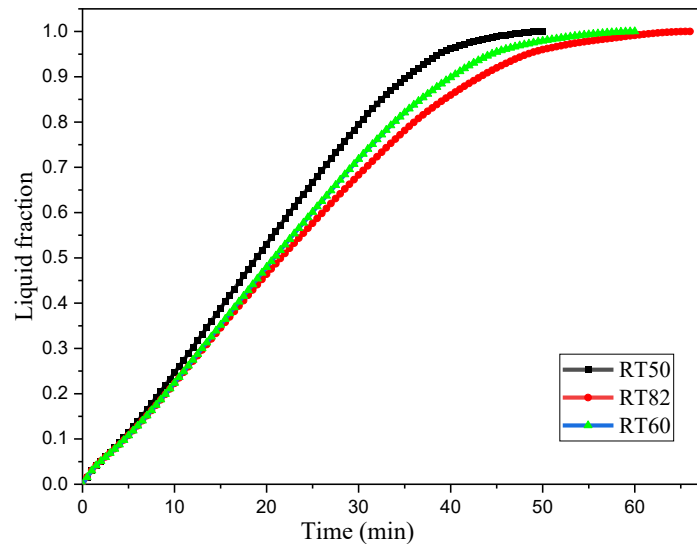


Figure IV. 22 - Variation of the phase change material type on liquid fraction

IV.5.6 Configuration shaped fin effect on the melting PCM

Figure IV.24 illustrates the evolution of the liquid fraction of the phase change material (PCM) over time for three different fin configurations: straight fins, T-shaped fins, and the newly proposed fins. During the initial stage, the liquid fraction values are nearly identical for all cases, as heat transfer is mainly governed by thermal conduction. As time progresses, differences between the curves become apparent due to variations in the heat distribution efficiency within the PCM.

The results show that PCM melting is completed after approximately 70 minutes when straight fins are used, whereas the melting time decreases to 33.5 minutes with T-shaped fins and to 32 minutes with the new fins. This improvement is attributed to the ability of the new fins to transfer heat to the central regions and areas far from the tube surface, thereby reducing the presence of cold zones and enhancing the effect of natural convection within the phase change material.

In general, the curves indicate that improving the fin geometry significantly increases the thermal charging rate and reduces the melting time. The new fins exhibit the best thermal performance due to their ability to improve heat distribution within the PCM and enhance both conductive and natural convective heat transfer mechanisms. Consequently, fin geometry has a significant influence on the thermal performance and efficiency of the thermal energy storage system.

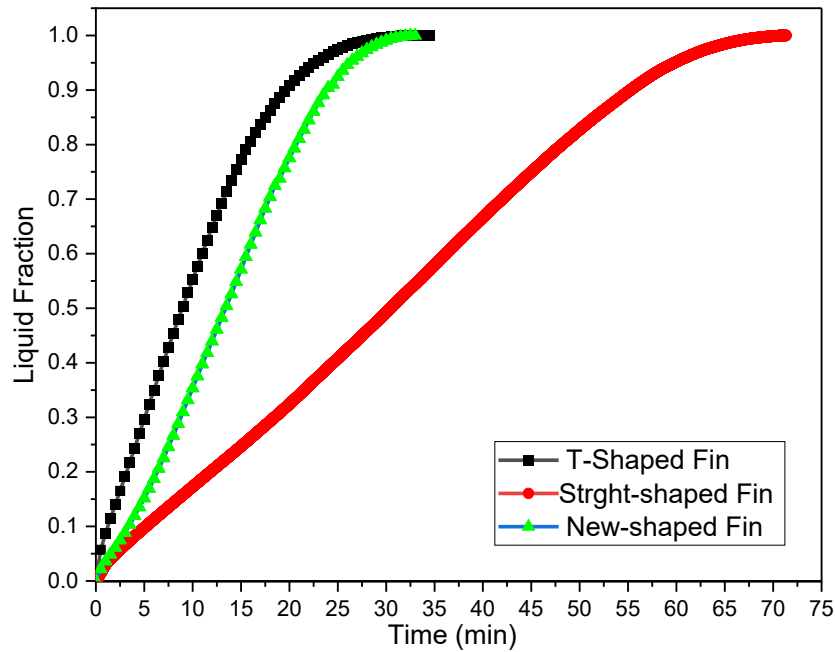


Figure IV. 23 - Effect of fin geometry on the liquid fraction.

IV.5.7 Case study : Tissemsilt City

In this case study, the solar radiation corresponding to the selected days is first calculated to evaluate its effect on the outlet temperature of the flat-plate solar collector in Tissemsilt City. Appropriate mathematical models are employed to accurately estimate the thermal energy gained by the solar collector and to assess the influence of local climatic conditions on the performance of the solar thermal system. Subsequently, an energy balance analysis of the solar thermal collection system is performed, including calculations of the reflected and absorbed solar radiation, in order to determine the amount of energy transferred through the solar collector.

Finally, numerical simulations of the melting process of RT-82 paraffin are carried out to evaluate the effectiveness of latent thermal energy storage using (PCMs) for the storage and utilization of solar energy under the climatic conditions of Tissemsilt City.

IV.5.7.1 Solar radiation analysis in the Tissemsilt region

Solar radiation was calculated for four representative days: July 17, December 15, April 15, and September 15, selected to represent different seasons in Tissemsilt City, Algeria. The Perrin Brichambaut model was used to estimate the solar radiation values for these selected days. The radiative analysis, presented in Figure IV.25, provides the necessary input for evaluating the performance of the flat-plate solar collector and predicting the outlet temperature of the heat transfer fluid. The results indicate a significant seasonal variation in solar radiation intensity. The highest

solar radiation values are observed on July 17, corresponding to the summer season, due to the high solar altitude and longer daylight duration, while the lowest values are recorded on December 15 during the winter season. This variation helps identify the optimal periods for solar energy utilization in the region.

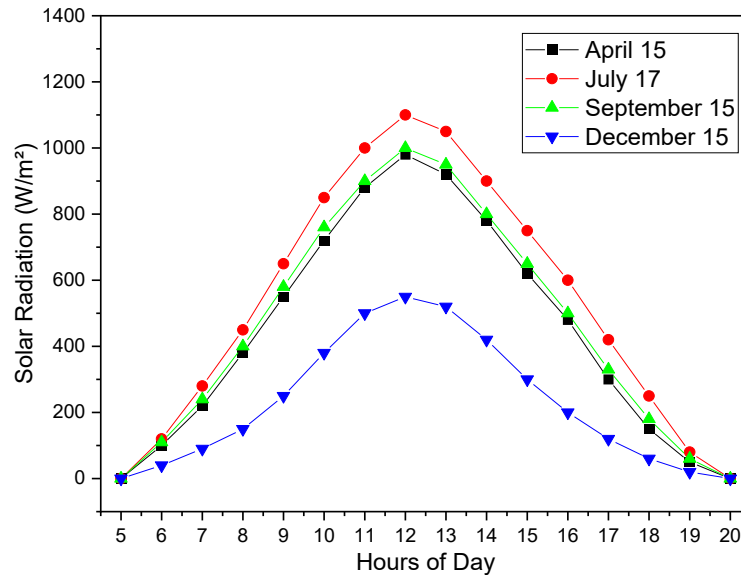


Figure IV. 24 - Variation of solar radiation during the typical days.

IV.5.7.2 Collector outlet temperature

Figure IV.26 illustrates the variation of the solar collector outlet temperature as a function of solar time throughout the day for the representative days considered in this study. The selected days are July 17, December 15, April 15, and September 15, which were chosen to represent the different climatic seasons of Tissemsilt City, Algeria. The results show that the outlet temperature closely follows the temporal evolution of solar irradiance for all cases, with a typical diurnal profile characterized by an increase in the morning, a peak around solar noon, and a subsequent decrease in the evening. Furthermore, the incorporation of nanoparticles enhances the thermal performance of the solar collector, leading to higher outlet temperatures compared to the base fluid. This improvement is particularly pronounced around solar noon, when solar radiation reaches its maximum intensity.

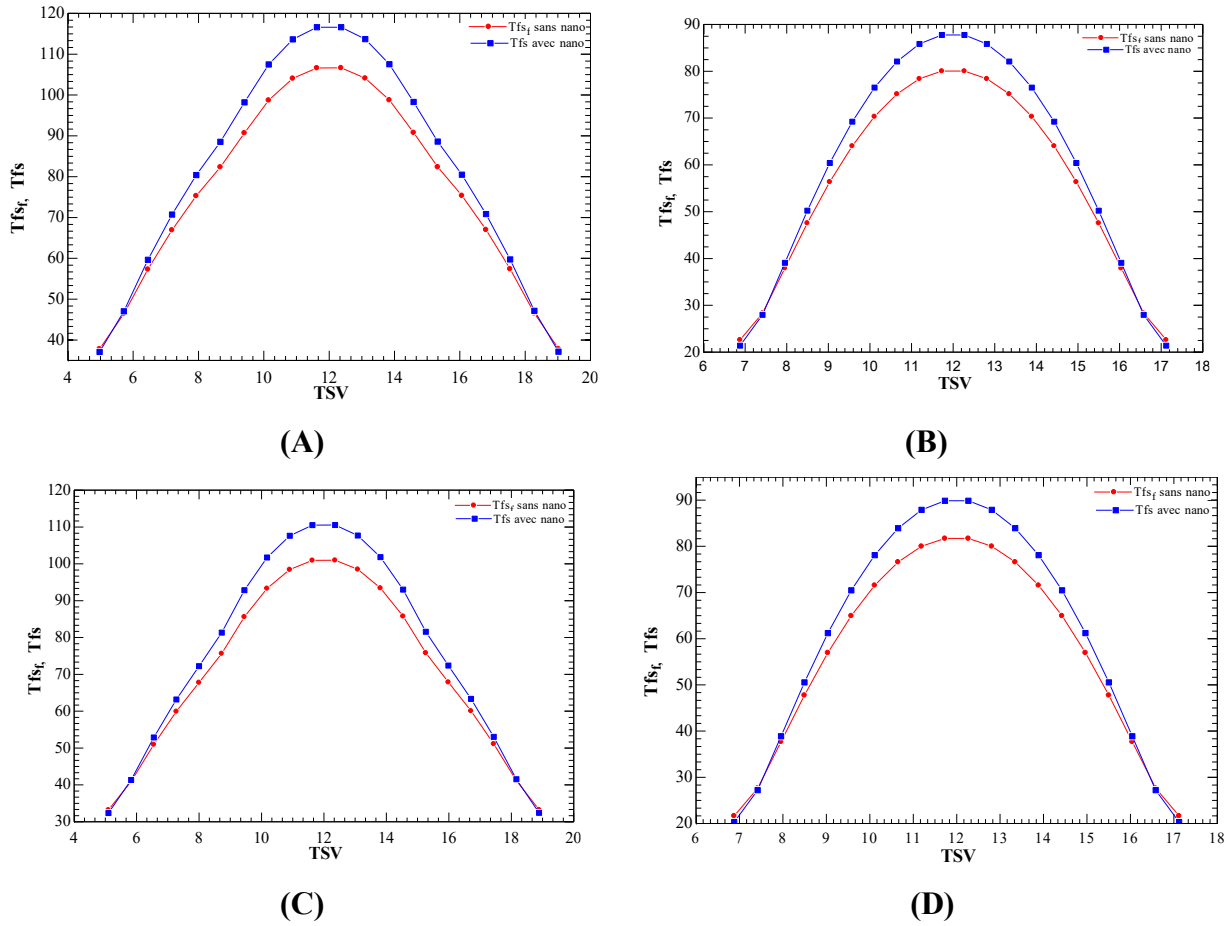


Figure IV. 25 - Collector outlet temperature with and without nanoparticles on the days of (A) July 17, (B) December 15, (C) April 15 and (D) September 15, 2024

IV.5.7.3 Thermal Energy Storage under Tissemsilt Climatic Conditions

The thermal system was comprehensively analyzed, including the solar collector, where the outlet temperature was obtained after enhancing the thermophysical properties of the heat transfer fluid under the climatic conditions of Tissemsilt, Algeria. The numerical simulation of the thermal storage unit was performed using the previously defined parameters and obtained results. Paraffin RT-82 was selected as the phase change material and incorporated into a triplex-tube heat exchanger equipped with newly designed fins to enhance enhance the heat transfer surface area. A representative day and the solar hour corresponding to maximum solar irradiance were selected, and nanoparticles were added to further improve the thermal behavior of the system. The performance of the thermal storage unit was evaluated in terms of average temperature evolution and melting rate, as illustrated in the corresponding results. The findings demonstrate a significant reduction in melting time and an increase in stored thermal energy compared to the conventional configuration, confirming the effectiveness of the proposed enhancement techniques.

IV.5.7.4 System study on december 15th

Based on the previous simulation results of the thermal energy storage system employing a triple-tube heat exchanger with newly designed fins and copper oxide (CuO) nanoparticles, a detailed analysis of the overall thermal behavior was carried out. The heat output from the solar collector was also included in the simulation model for the winter period (15 December), under optimal operating conditions and a suitable nanoparticle concentration for this case. The outlet temperature was found to be 348 K. The results indicated that the melting process was completed in 43 minutes only, as shown in Figure IV.27. This enhancement in the melting performance can be attributed to the increase in the effective thermal conductivity of the storage medium due to the addition of CuO nanoparticles, as well as the improved heat transfer achieved by the presence of fins inside the triple-tube heat exchanger. As a result, thermal energy transfer was intensified and the thermal resistance within the storage material was reduced, leading to a faster melting process and higher efficiency in utilizing solar energy during its availability period.

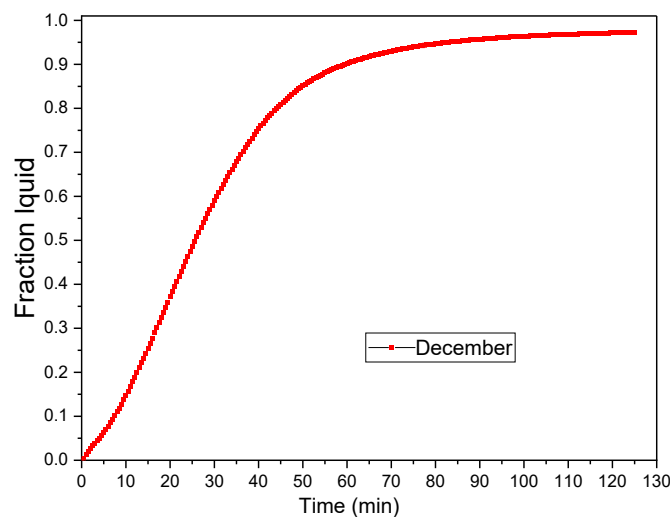


Figure IV. 26 - Melting process studied on december

IV.5.7.5 System study on July 17th

Based on previous simulation results of the thermal energy storage system using a triple-pipe heat exchanger with novel fins and copper oxide (CuO) nanoparticles, the overall thermal behavior of the system was investigated in this stage. The heat released from the solar collector was also incorporated into the simulation model during the summer period (July 17), using the optimal operating conditions and appropriate nanoparticles for this case. The outlet temperature reached 391 K, and the melting process started at 11:00 AM. The results showed that the melting process was completed within only 17 minutes, as illustrated in Figure IV.28. This reflects a significant

improvement in the melting rate and the effectiveness of solar thermal energy utilization during its availability period.

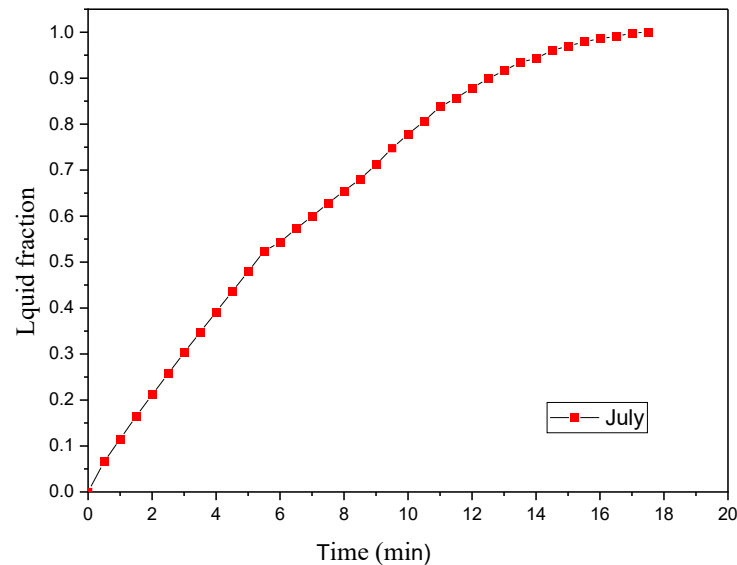


Figure IV. 27 - Melting process studied on July

IV.5.7.5 Performance of a PCM Storage System under Tissemsilt Climatic Conditions

Figure IV.29 shows the effect of fin geometry and nanoparticle integration on the melting performance of the thermal energy storage system. The straight-fin configuration exhibited the longest melting time (60 min), while the T-shaped and new-shaped fins reduced the melting time to 33.5 min and 32.5 min, corresponding to reductions of 44.17% and 45.83%, respectively as show inTable IV.8. The best performance was achieved with the new-shaped fin combined with nanofluids in the solar collector and nanoparticles in the storage unit under the climatic conditions of Tissemsilt, where the melting time decreased to 17 min, representing a reduction of 71.67% compared with the conventional configuration. These results demonstrate that the combined effect of fin geometry optimization and nanoparticle integration significantly enhances heat transfer and accelerates the melting process within the PCM.

Table IV. 8 - Performance of the thermal energy storage system

Configuration	Straight Fin	T-Shaped Fin	New-Shaped Fin	New-Shaped Fin + Nanofluid Collector + Nano-Enhanced PCM
Surface Area (mm ²)	336	312	315	315
Volume (mm ³)	168000	156000	157500	157500
Melting Time (min)	60	33.5	32.5	17

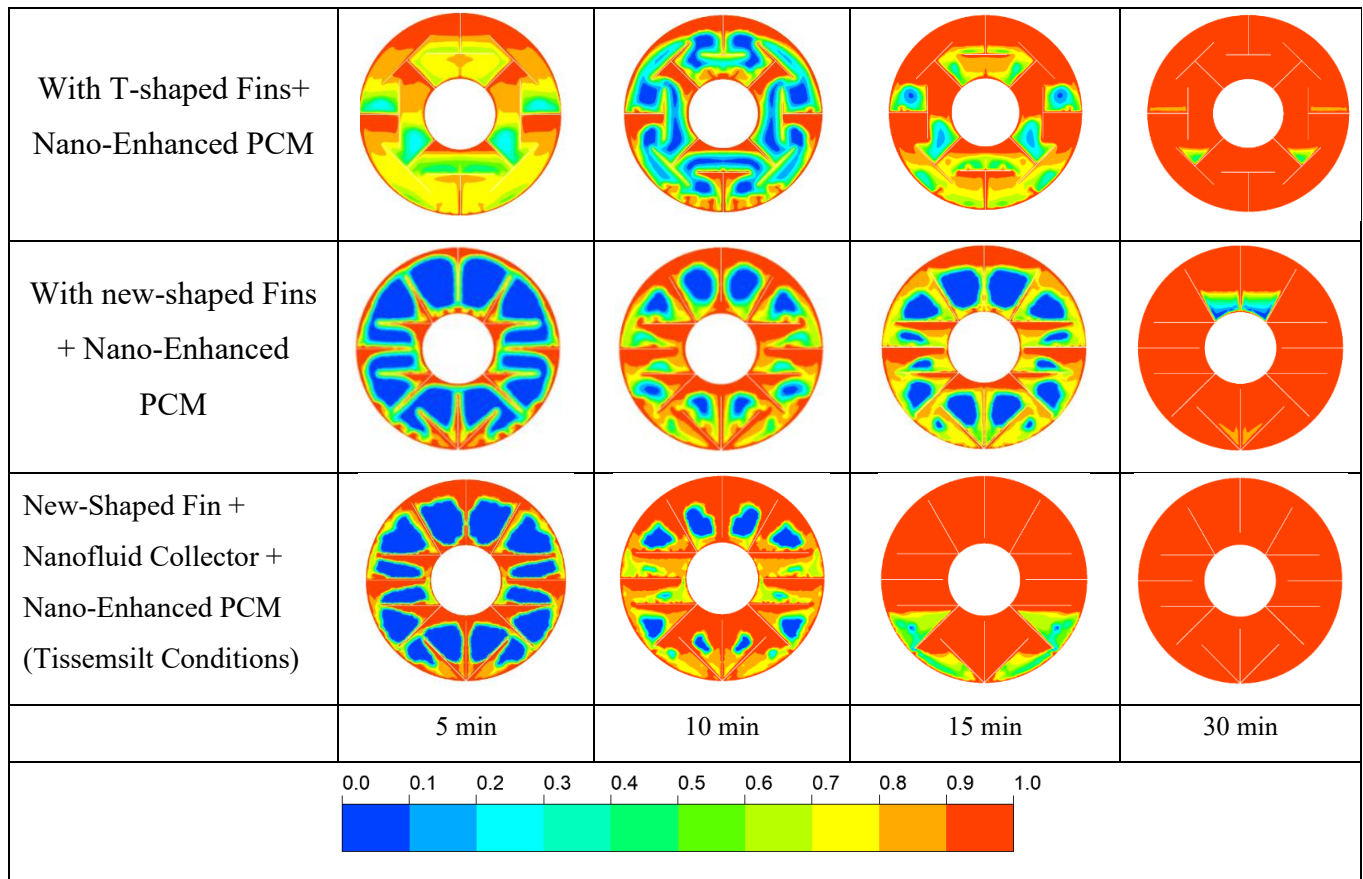


Figure IV. 28 - Liquid fraction contours with nanoparticles over time

IV.6 Conclusion

This chapter presented and analyzed the results obtained through numerical modeling and simulations, focusing on improving the performance of the studied systems. The findings revealed the significant influence of key variables, including solar radiation, the performance of solar collectors, and heat storage processes utilizing (PCMs).

The results demonstrated that incorporating nano-particles improved thermal distribution and significantly reduced melting time, thereby enhancing the thermal efficiency of the heat storage system. The study also highlighted the critical role of fin designs, particularly T-shaped fins, in optimizing heat transfer within three-tube heat exchangers, leading to superior thermal performance.

Validation of the employed numerical models was achieved by comparing the obtained results with reference data, ensuring the accuracy and reliability of the adopted approaches. The study further confirmed its practical applicability by adapting the analysis to the specific climatic conditions of the Tissemsilt region, considering local environmental factors and their impact on solar energy systems. overall, this research underscores the importance of numerical modeling and simulation in developing innovative renewable energy solutions and efficient TESS

General Conclusions and Perspectives

General conclusions

This thesis presents a comprehensive study of a solar thermal energy collection and storage system under the climatic conditions of Tissemsilt, Algeria. The research contributes to enhancing heat transfer performance in both the solar collector and the thermal energy storage system. For the solar collector, a mathematical model was developed to estimate solar irradiance on four representative days (July 17, December 15, April 15, and September 15), selected to represent the four seasons of the year in the region.

The performance of both the solar collector and the thermal storage unit was validated through comparison with previously published studies, showing strong agreement and confirming the reliability of the adopted numerical models. In addition, the effect of incorporating nanoparticles into the heat transfer fluid was investigated, and the collector performance was analyzed under the environmental conditions and geographical characteristics of the installation site.

Regarding the thermal storage unit, its thermal performance was enhanced using (PCMs) and by improving their thermal conductivity through the development of an innovative storage unit design. Mathematical modeling and two-dimensional numerical simulations were employed to analyze heat transfer in a triple-tube storage system equipped with innovative T-shaped fins, while nanoparticles were integrated to further improve the thermal performance.

The results demonstrated that improving thermal efficiency requires enhancing the thermal conductivity of both the heat transfer fluid and the phase change material through effective integration between the solar collector and the thermal storage unit. The main findings can be summarized as follows:

- **Thermal storage unit:** The integration of innovative fin geometries in the proposed designs significantly improved the performance of the thermal storage systems and accelerated the melting process of the RT-82 material.
- **Practical effectiveness:** The developed fin design was characterized by simplicity, low cost, and ease of manufacturing and implementation.
- **Comparative performance:** The proposed designs outperformed conventional configurations by reducing the melting time by 33.5% while ensuring a more uniform thermal distribution within the storage unit.
- **Optimal fin dimensions:** The optimal fin dimensions were determined to be 30 mm in length, 0.5 mm in thickness, and 45 mm in width, achieving the highest thermal performance while maintaining a high storage capacity.
- **Effect of nanoparticles:** The incorporation of nanoparticles (Al_2O_3 , Cu, and TiO_2) at a concentration of 3% enhanced the heat transfer rate. Among them, aluminum oxide (Al_2O_3)

General Conclusions and Perspectives

nanoparticles exhibited the best performance, reducing the melting time from 3600 s to 2550 s.

- **Effect of nanoparticles in system with Tissemsilt city:** The best performance was achieved using the new-shaped fin combined with nanofluids in both the solar collector and the thermal energy storage unit under the climatic conditions of Tissemsilt. In this configuration, the melting time decreased significantly to only 17 min, compared with 60 min for the conventional straight-fin configuration. This corresponds to a reduction of 71.67% in melting time, demonstrating a substantial enhancement in heat transfer and thermal charging performance.

This remarkable improvement can be attributed to the synergistic effect of fin geometry optimization and nanoparticle integration. The nanofluid enhanced solar energy absorption and improved heat transfer within the solar collector, while the nanoparticles dispersed in the phase change material (PCM) increased its thermal conductivity and reduced internal thermal resistance. Consequently, heat propagation within the storage unit was accelerated, leading to a faster melting process and improved overall thermal performance.

These findings demonstrate that the combination of an optimized fin design and nano-enhanced heat transfer media is an effective strategy for improving the efficiency of solar thermal energy storage systems. Under the climatic conditions of Tissemsilt, this approach significantly reduces the thermal charging time and enhances the overall system performance.

Perspectives

These results provide a solid foundation for improving heat transfer in both the solar collection system and the thermal storage system, in addition to enhancing the understanding of the behavior of (PCMs) and nanoparticles. Furthermore, these findings can be exploited for further studies and developments in the field of heat transfer and the advancement of thermal energy storage units.

Moreover, future research in this field may be extended through the adoption of innovative concepts and advanced technologies, such as the implementation of a solar tracking system to improve the efficiency of solar energy harvesting, as well as the use of hybrid nanoparticles characterized by superior thermal conductivity enhancement capabilities. Concerning the thermal energy storage unit, new designs based on rotational storage systems or the incorporation of hybrid nanoparticles may be developed with the aim of improving the thermal energy storage capacity of (PCMs) and enhancing the overall thermal performance of the system thermal performance of the system

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