

CHARACTERIZATION OF LATENT THERMAL ENERGY IN A STORAGE
UNIT WITH FIBONACCI-SEQUENCE-INSPIRED FINS

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ABSTRACT

CHARACTERIZATION OF LATENT THERMAL ENERGY IN A STORAGE UNIT WITH FIBONACCI-SEQUENCE-INSPIRED FINS

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Latent thermal energy storage (LTES) is a viable method to mitigate one of the main problems with renewable energy sources, such as solar energy, which is their intermittent nature. LTES units take advantage of the large latent heat of fusion in phase change materials (PCMs) to store a substantial amount of energy in a relatively small volume, making them efficient and economical. The major challenge with PCMs is their low thermal conductivity, eventuating long melting (charging) and solidification (discharging) times. Using extended surfaces (fins) is a popular method, with recent developments in novel designs such as tree-like and fractal fins. The complex fin configurations improve the conduction while decreasing the natural convection effects by introducing additional shear stress to the melted PCM. In the current work, a new fin design based on the Fibonacci sequence is proposed in a vertical LTES unit to improve the charging/discharging characteristic. An experimental setup is designed to observe the charging/discharging behavior based on the volume expansion method. A numerical model based on the enthalpy-porosity method is implemented to investigate the physics behind the melting/solidification. The proposed fin performed 20% better in terms of melting time compared to longitudinal fins by introducing more dispersed

natural convection-dominated regions. The average liquid PCM velocity on the melt front was studied as a parameter for evaluating the natural convection behavior. The reduction of 45% in solidification time was established for the novel fin as well.

Keywords: phase change material, nature-inspired, fractal geometry, energy



ÖZ

FİBONACCİ DİZİSİNDEN ESİNLENİLMİŞ KANATÇIKLARA SAHİP BİR DEPOLAMA ÜNİTESİNDE GİZLİ ISIL ENERJİNİN KARAKTERİZASYONU

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Gizli ısı ısı enerjisi depolaması, güneş enerjisi gibi yenilenebilir enerji kaynaklarının temel sorunlarından biri olan kesintili doğalarının etkisi için uygun bir yöntemdir. LTES üniteleri, önemli miktarda enerjiyi nispeten küçük bir hacimde depolamak için faz değiştiren malzemeleri (FDM'ler) verimli ve ekonomik hale getiren büyük miktardaki gizli ısıdan faydalanır. FDM'lerle ilgili en büyük zorluk, uzun erime (şarj) ve katılma (deşarj) sürelerine neden olan düşük termal iletkenlikleridir. Genişletilmiş yüzeylerin (kanatçıkların) kullanımı, ağaç benzeri ve fraktal kanatçıklar gibi yeni tasarımlardaki son gelişmelerle birlikte popüler bir yöntemdir. Karmaşık kanat konfigürasyonları, iletimin etkisini artırırken, erimiş FDM'de ek kesme gerilimleri yaratarak doğal taşıma etkilerini azaltır. Mevcut çalışmada, şarj/deşarj karakteristiğini iyileştirmek için dikey bir gizli ısı ısı enerjisi depolama ünitesinde Fibonacci dizisini baz alan yeni bir kanat tasarımı önerilmektedir. Şarj/deşarj davranışını gözlemlemek için hacim genişletme yöntemine dayalı bir deney düzeneği tasarlanmıştır. Erime/katılmanın ardındaki fiziki araştırmak için entalpi-gözeneklilik yöntemine dayalı sayısal bir

model uygulanmaktadır. Önerilen kanatçık, daha iyi dağılmış doğal taşınım baskın bölgeler ortaya çıkararak, boyuna kanatçıklara kıyasla erime süresi açısından %20 daha iyi performans göstermektedir. Erime bölgesindeki ortalama sıvı FDM hızı, doğal taşınım davranışını değerlendirmek için bir parametre olarak incelenmiştir. Yeni kanatçık için katılma süresinde %45’lik bir azalma tespit edilmiştir.

Anahtar Kelimeler: faz değıştiren malzeme, doğadan ilham alan, fraktal geometri, enerji





To my family

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LIST OF ABBREVIATIONS

2D	2 dimensional
3D	3 dimensional
CTES	chemical thermal energy storage
CSP	concentrated solar power
FVM	finite volume method
HTF	heat transfer fluid
LTES	latent thermal energy storage
PCM	phase change material
PISO	Pressure-Implicit with Splitting of Operators
PRESTO!	PREssure STaggering Option
STES	sensible thermal energy storage
TES	thermal energy storage
TPS	transient plane source

LIST OF SYMBOLS

w	Absolute uncertainty of a quantity, -
R	An independent quantity, -
$\bar{U}$	Average liquid PCM velocity on the melting front, mm/s
$\overline{Nu_D}$	Average Nusselt number, -
A	Area of the expansion tubes balsted with hot air, m^2
c	Cap thickness, mm
h	Convective heat transfer coefficient, W/m^2K
ρ	Density, kg/m^3
μ	Dynamic viscosity, kg/ms
D	Expansion tube outer diameter, mm
$\vec{g}$	Gravitational acceleration, m/s^2
H	Height, mm
$\dot{m}$	HTF mass flow rate, kg/hr
f	Liquid fraction, -
L	Latent heat, kJ/kg
m	Mass, kg
C_{mush}	Mushy zone constant, $kg/(sm^3)$
Pr	Prandtl number, -
P	Pressure, Pa
r	Radius, mm
Re	Reynolds number, -
S	Source term, -
i	Specific enthalpy, kJ/kg

c_p	Specific heat capacity, kJ/kgK
T	Temperature, $^{\circ}C$
t	Time, s
k	Thermal conductivity, W/mK
Q	Thermal energy, kJ
$\mathbf{U}$	Velocity vector, m/s
V	Volume, m^3
β	Volumetric thermal expansion coefficient, $1/K$
x	x-direction, -
y	y-direction, -
z	z-direction, -



CHAPTER 1

INTRODUCTION

1.1 Need for Thermal Energy Storage

A transition to renewable and sustainable energy sources is imminent due to the depletion of fossil fuels and their harmful effects on the environment, such as global warming [1]. It is predicted that the world's total energy consumption will increase by 28% from 2020 to 2050 [2], which is inevitably slowed by an increase in CO_2 emissions if no action is taken for alleviation. The transition to greener energy sources is followed by many challenges that are mainly concerned with the intermittency of renewable energy sources (such as solar energy) [3], the energy lost through conversion (around 31% as predicted by the International Energy Association) [4], lack of reliable energy in peak hours [5], climate volatilities [6], etc. As a solution, energy storage can ensure the balance between energy supply and demand, organize energy distribution, play a significant role in the transition to green energy sources, and help with the reliable supply of electricity [5].

According to European Energy Storage Technology Development Roadmap Towards 2030, thermal energy storage (TES) is a suitable technology for conventional energy generation systems and a possible technology for renewable energy generation systems, transmission, and distribution [7]. Among other energy storage means such as electrical, mechanical, and electrochemical storage methods, TES is proven to be more reliable, practical, easy to integrate, and economically viable in various applications. More specifically, in solar applications such as concentrated solar power (CSP) systems, TES solutions have more capacity and more extended storage periods compared to electrical storage systems [8]. In fact, a study performed in 2017 estimated

that around 50% of the CSP plants utilize TES systems, with an expected higher ratio in the future [9]. In other applications like geothermal energy, fossil fuel, nuclear power plants, and waste heat processes, TES systems have shown promise by taking advantage of the lack of energy management and producing heat [10–13].

TES prevents the waste of valuable energy and stores it in a medium to be used at later times. To do so, various methods with different media have been investigated in the literature. The following section discusses and compares the three main types of TES.

1.2 TES Types

TES systems can be categorized into three main types based on the energy storage method: sensible, latent, and chemical TES. The following subsections provide a descriptive and comparative illustration of the different TES types.

1.2.1 Sensible TES

Sensible TES (STES) uses a temperature rise to store energy. The total amount of stored sensible energy (Q_{stored}) is proportional to the mass of the material (m), specific heat capacity (c_p), and the temperature difference between the initial temperature ($T_{initial}$) and the final temperature (T_{final}) as given in Eqn. 1.1.

$$Q_{stored} = \int_{T_{initial}}^{T_{final}} mc_p dT \quad (1.1)$$

During the charging (increasing the temperature) and the discharging (decreasing the temperature), the material does not undergo any phase change, which explains the “sensible” adjective. The storage medium is usually a liquid or solid material. The common materials for the liquid STES systems are water for low-temperature storage [14], mineral oils for medium-to-high temperature range, and molten salts for high-temperature applications, especially in CSP plants [15]. Solid STES systems

generally use rocks, sand, concrete, etc., with a lower value of specific heat but greater densities than liquid TES materials. Additionally, unlike liquid materials, they do not experience vapor pressure problems; however, with their low thermal conductivity and incapability of circulation, a heat transfer fluid (HTF) is needed for charging/dis-charging processes [16].

Since STES is the simplest way of energy storage, it has been facilitated in many thermal storage plants. Water tank storage systems have proven to be cost-effective and beneficial, especially for low-temperature solar applications [17]. Underground storage systems use rocks or water for seasonal storage applications, with limited outcomes due to high costs [18]. Packed bed STES has been shown to produce a more uniform temperature with high stratification behavior using rocks or pebbles [19].

Although STES systems and materials are mature, stable, and widely available, they suffer from two major problems:

- Their working principle is based on a temperature gradient resulting in the energy discharge at a variable temperature range, which is not desired in many applications such as power generation.
- Since they do not undergo phase change and cannot utilize the latent heat, a large mass is usually required to meet the desired storage capacities, resulting in costly construction and maintenance.

1.2.2 Latent TES

Unlike STES, the energy in a latent TES (LTES) unit is primarily stored in the latent heat, which is the amount of energy provided to a material to undergo a state change from solid to liquid at a constant pressure. LTES units take advantage of the large latent heat in phase change materials (PCMs) to store a substantial amount of energy with a relatively small volume, making them efficient and economical. The total amount of energy stored in an LTES unit can be given by Eqn.1.2

$$Q_{stored} = \int_{T_{initial}}^{T_m} mc_p dT + m f L + \int_{T_m}^{T_{final}} mc_p dT \quad (1.2)$$

where T_m is the melting temperature, f is the liquid fraction, and L is the latent heat. The melting/solidification process typically happens at a constant temperature, which is one of the main advantages of LTES [6]. It is worth noting that an LTES unit is not only comprised of latent heat but also sensible heat contributes to the total amount of stored energy as shown in Eqn. 1.2. However, the amount of stored latent heat is enormous since the latent heat is a significant value; thus, sensible terms do not play a major role in storing energy. For example, considering a 10°C temperature increase in 1 kg of RT42 from Rubitherm [20], 144 kJ of energy is stored in the form of latent heat while this value for the sensible heat is only 20 kJ, about 12% of the total stored energy. Figure 1.1 graphically illustrates the difference between the STES and LTES behaviors.

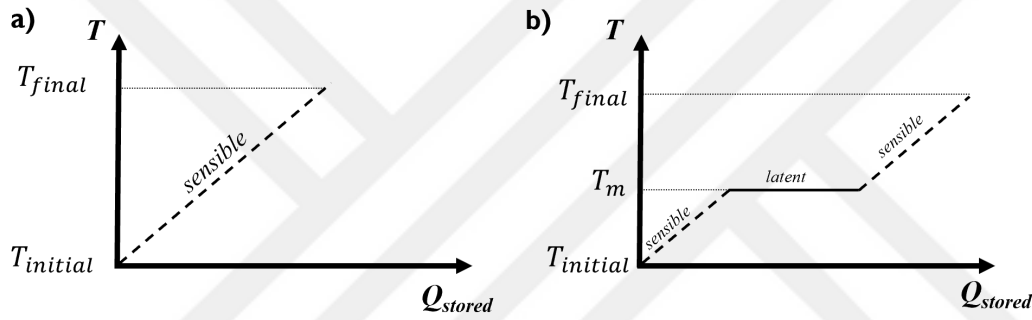


Figure 1.1: Stored thermal energy behavior: **a)** sensible and **b)** latent

PCMs are categorized into two main types: organic and inorganic. Organic PCMs such as paraffin, the most used organic PCMs [21], fatty acids, esters, and glycol are easily found in nature, generally non-toxic and stable though they tend to decompose at high temperatures [22]. Inorganic PCMs, such as salt hydrates, are commonly used in high-temperature applications and have toxic and corrosive properties [23]. LTES systems are popular and effective in solar thermal plants due to their wide range of working temperatures, large energy density, and design and implementation flexibility [7, 24].

Encapsulated PCMs have proven to be influential in controlling the fluctuations in industrial waste heat applications. For example, by augmenting a PCM unit for recovering the waste heat in the steel industry, Dal Magro *et al.* [25] deduced a reduction in the steam generator and turbine size is possible through a thermodynamic analysis.

LTES units are also used in engine waste heat recovery, where the temperatures are relatively low [26].

The main issue with PCMs is their small thermal conductivity value, generally between 0.2 and 0.6 W/mK [3], which results in long charging/discharging times. Various methods have been studied and implemented to mitigate this drawback, the most common of which is using extended surfaces or fins. Further discussion on the enhancement methods will be provided in the following subsections.

1.2.3 Chemical TES

Chemical TES (CTES) uses reversible chemical reactions to store and release heat. They have the highest TES density compared to STES and LTES and are applicable in medium-high temperature ranges up to 400°C . Their current status, including disadvantages such as stability and decomposition problems and slow charging/discharging rates, make them unsuitable for many applications [4]. CTES systems are underdeveloped and need many advances in terms of material science and chemistry. In the current work, CTES materials and methods are out of scope; therefore, the details are left out for curious readers [27, 28].

1.3 Methods of Performance Improvement in LTES

As discussed in Section 1.2.2, the main issue with PCMs is their low thermal conductivity. A low thermal conductivity directly correlates with how fast an LTES unit melts and solidifies. The charging time is an important factor in LTES since the energy source (take solar irradiance for example) is not always available to charge the PCM: in daily storage systems, 6 to 8 hours of melting time is desired to ensure the energy storage. In the solidification process, time is also significant since it determines the power provided by the LTES. The charge/discharge time also evaluates the applicability and cost-effectiveness of the implemented LTES system. There are many methods to improve the charging/discharge performance of the LTES systems, some of which are discussed briefly here.

Additives with high thermal conductivity: One way to increase the thermal conductivity of PCMs is by adding materials with high thermal conductivity values. These materials can help to transfer heat more efficiently within the PCM, leading to improved performance of the PCM in energy storage applications. Carbon-based materials are among the popular additives that have a stable chemical composition and low density [29]. Metal-based materials such as metal foams are also effective in increasing the thermal performance of PCMs by providing a large surface-area-to-volume ratio.

Impressive reductions in melting/solidification times have been reported in the literature using additives in PCMs: Esapour *et al.* [30] reported a decrease in melting time of up to 55% using metal foams, Ramakrishnan *et al.* [31] achieved 49% increase in thermal conductivity using carbon additives which resulted in up to 200% improvement in heat gain of PCM and Sheikholeslami and Mahian [32] were able to decrease the total solidification time by up to 14% using nanoparticles added to PCM. Even though utilization of additives in LTES applications has shown great promise, they alter the properties of PCM and decrease its latent heat, it is difficult to obtain a uniform metal foam, they increase the mass of the LTES unit, and possess low chemical stability [33].

Encapsulation: By enclosing a volume of PCM into a defined shape, the PCM can be encapsulated. Encapsulation increases the heat transfer area and protects the PCM from environmental effects [34]. Additionally, some PCMs, such as several families of non-organic PCMs, should be airtight because there exists a chance of oxidation. Encapsulation is categorized based on its size into two types of micro-and-macro encapsulation. Various geometries for encapsulation have been proposed: spheres [35], cylinders [36], and innovative geometries that can take advantage of volume expansion during phase change in PCMs [37, 38]. The encapsulation techniques are effective ways of improving the performance of LTES units, with the advantage of keeping the PCM in a container that resists external operating conditions. However, they come with a toll of high prices and manufacturing issues that are especially challenging in high-temperature applications.

Extended surfaces (fins): Utilization of fins in LTES units is the most common way of performance improvement. Various conventional fin types such as annular [39],

longitudinal [40] and pin-fins [41] and novel fins like fractal [42] and bio-inspired tree-like fins [43] have established significant enhancement, especially in charging process due to increasing the conduction and natural convection effects by augmenting the heat transfer area. Since the current work is concerned with fins, the following subsections provide more detail on conventional and tree-like fins and discuss their strong and weak points.

Other niche techniques to improve the heat transfer in LTES units are also reported such as using multiple PCMs or cascade arrangements [44], changing the orientation [45], and rotating the LTES tank [46, 47] however, using fins remains the most important and common method. Novel and innovative fin designs have been introduced in recent years to maximize the efficiency and lower melting/solidification times of LTES systems.

1.3.1 Base Design of an LTES Tank

To introduce a base design for the fin augmentation in an LTES, this section briefly discusses the geometrical configuration of a typical LTES unit. A common design for an LTES system involves the use of two concentric cylinders. In this design, a fluid, such as water or oil, is typically flowed through the inner cylinder, while the space between the inner and outer cylinders is filled with a PCM. The HTF tube is used for the HTF to flow in and divide the two media and prevent them from mixing, as depicted in Figure 1.2.

During the charging process, the fluid is heated, causing the PCM to melt and absorb the heat. During the discharge process, the fluid is cooled, causing the PCM to solidify and release the stored heat. The PCM and HTF can change places in a less common configuration [48].

The HTF tube is often made of aluminum or copper thanks to their high thermal conductivity in order to transfer the heat from the HTF to PCM without extra thermal resistance. A bundle of discussed cylinders is commonly put inside a larger tank, resembling a shell-and-tube heat exchanger. Due to its simplicity, the shell-and-tube design is the most common and reliable design for LTES systems [49].

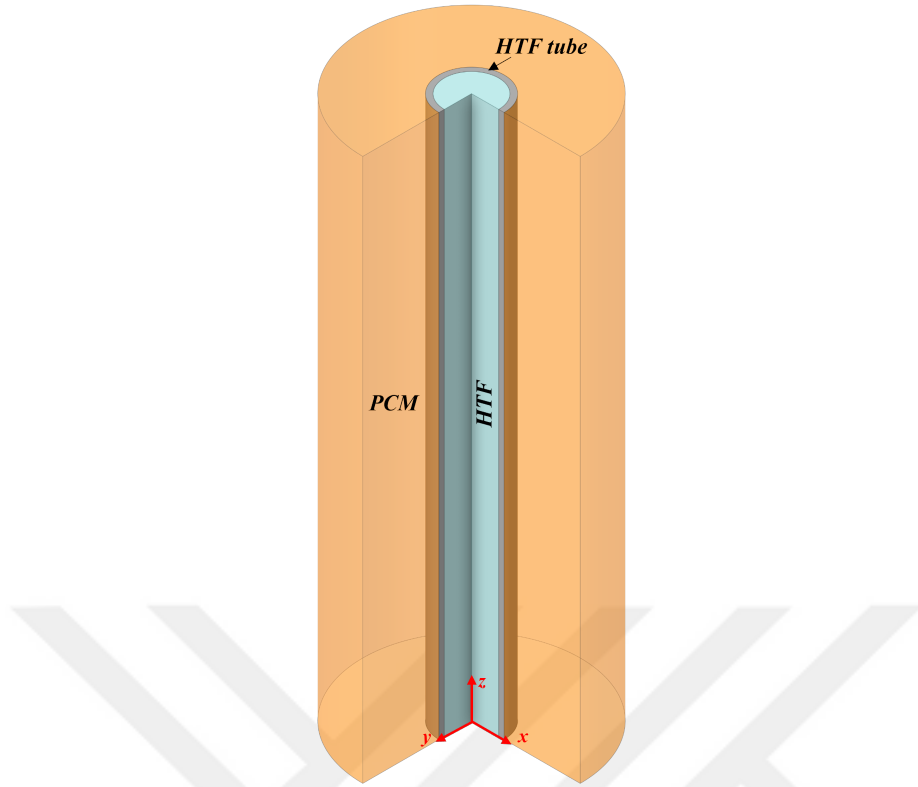


Figure 1.2: The base design of a cylindrical LTES tank

One of the main advantages of the shell and tube design is its simplicity. It is relatively easy to manufacture and maintain, and it can handle a wide range of fluid flow rates and pressure drops. It is also relatively inexpensive compared to other types of heat exchangers. Many parameters regarding the base design have been investigated in the past decades. For the sake of brevity and since the current work is concerned with the attached fins to the HTF tube of the base LTES tank, the studied parameters for simple design are mentioned in Table 1.1 and the discussion is followed by finned configurations.

Table 1.1: Relevant studies to unfinned LTES units

Phase change temp. ($^{\circ}C$)	LTES orientation	Phase change process	Study type	Studied parameters	Main findings	Authors
30	Vertical	Melting/solidification	Numerical	- HTF operating conditions - Length and diameter of the tank	- Negligible effect of HTF mass flow rate on charging/discharging - Stored/released energy is a strong function of HTF inlet temperature	Trp <i>et al.</i> [50]
58-60	Vertical	Melting/solidification	Experimental	HTF mass flow rate	- Weak correlation between HTF mass flow rate and charging/discharging behavior - Melting front progression is from top to bottom and solidification front is from bottom to top	Jesumathy <i>et al.</i> [51]
803	Horizontal	Melting	Numerical	HTF mass flow rate and temperature	- Melting time nearly halves with doubling the HTF mass flow rate - Melting time nearly halves with increase of 40K in HTF inlet temperature	Tao <i>et al.</i> [52]
58	Horizontal	Melting/solidification	Experimental	HTF temperature	- Asymmetric melting behavior due to presence of natural convection during melting - Natural convection, only present in initial stages of solidification - Change in HTF temperature significantly affects the charging/discharging time	Avci and Yazici [53]
35	Vertical	Melting/solidification	Experimental/numerical	HTF injection direction	For charging, a top injection and for the discharging, a bottom injection is recommended	Longeon <i>et al.</i> [54]
50	Horizontal	Melting/solidification	Experimental/numerical	HTF temperature	- Convective and conductive heat transfer modes are established in charging/discharging while in charging/convection is dominant - The rate of heat transfer is directly related to HTF temperature	Hosseini <i>et al.</i> [55]
56	Vertical	Melting/solidification	Experimental	HTF mass flow rate	- The application of PCM in solar water heating systems is proven - With an increase in mass flow rate, total life cycle cost decreases	Mahfuz <i>et al.</i> [56]
59	Vertical	Melting/solidification	Numerical	-	Natural convection dominates the heat transfer process during melting while in solidification, the dominant mode of heat transfer is conduction	Saddegh <i>et al.</i> [57]
220	Vertical/horizontal	Melting	Numerical	- Location of PCM and HTF (inner or outer cylinders) - HTF injection direction (top or bottom) for vertical case	- Natural convection causes a non-uniform interface between solid and liquid phases - The melting time is the shortest for the horizontal configuration with PCM inside the inner cylinder - With the HTF at the top, the natural convection effects do not change in any configuration	Han <i>et al.</i> [58]

The literature states that the melting process is composed of three stages: first, as the melting starts, the heat from the HTF is transferred to the PCM which results in melting with conduction. With the advancement in time, the melted volume of PCM increases, which gives room to the convective flows to take place in the liquid region of PCM. The natural convection flows happen due to the liquid PCM density difference arising from the change in temperature. The natural convection continues until the temperature gradients in the melted PCM diminish and the conduction becomes the dominating heat transfer mode once again. Figure 1.3 shows a typical behavior of melting in an LTES unit consisting of the mentioned three stages. The parameter in the y-axis is the average liquid fraction (f) in the PCM, which indicates how much of the PCM is liquefied.

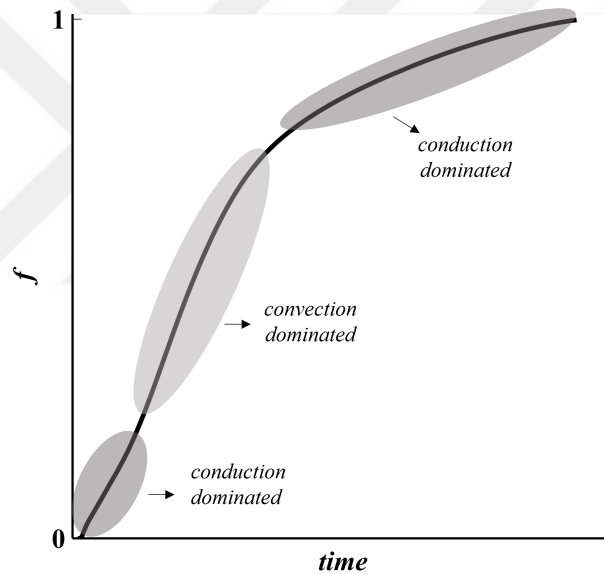


Figure 1.3: Typical behavior of liquid fraction in an LTES unit

1.3.2 Radial Fins

Radial or circular fins are a popular type of extended surface that has a long history of implementation in heat exchangers, heatsinks, condensers, etc. Due to their simplicity, they are widely used in LTES applications as well. Figure 1.4 depicts a typical

vertical LTES unit with radial fins on the tube that carries the HTF. Radial fins extend the heat transfer area and improve the heat transfer from the HTF to the PCM, especially in the radial direction, which results in improvements in conduction and natural convection. Several studies considering the realizations of radial fins in LTES units have been reported in the literature, the summary of which is presented in this subsection.

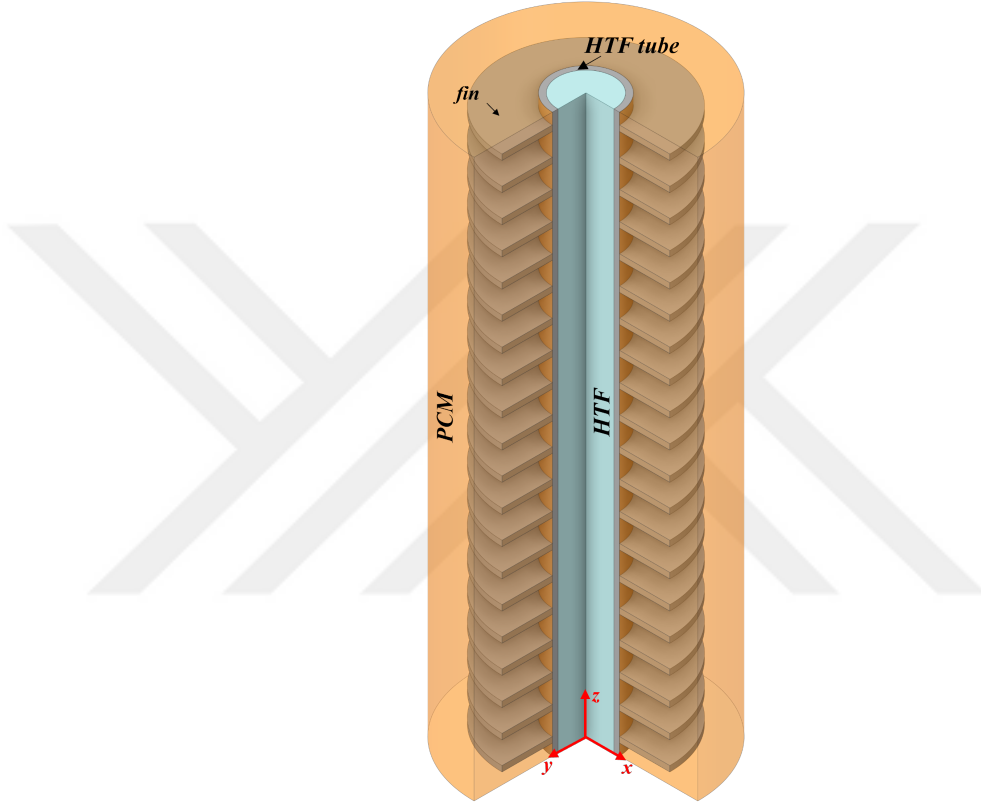


Figure 1.4: A typical design of an LTES unit with radial fins

Agyenim *et al.* [59] experimentally studied the effects of circular fins in a horizontal LTES unit with erythritol ($T_m = 117.7^\circ\text{C}$) as the PCM. They reported a 15% increase in total charged energy and a 46% increase in total discharged energy compared to the case with no fins for the same time interval. Wang *et al.* [60] studied the effects of various geometrical parameters such as fin pitch, radius, thickness, and outer shell diameter and length in a radially finned LTES unit and indicated the dependency of energy efficiency ratio and heat storage rate to them. They also showed that a laminar HTF flow is desired based on the performance of LTES. By studying the discharge

process with no natural convection involvement in LTES with non-uniform radial fin distribution, Kuboth *et al.* [61] established that a higher fin density near the outlet enhances the power output. Yang *et al.* [62] focused on the natural convection effects in vertical LTES units with annular fins by changing the fin number and thickness. They were able to achieve a 65% improvement in charging time with the optimum fin number and thickness. An increase in fin number does not necessarily mean an enhancement in melting since with increasing the fin number, the effects of natural convection tend to demise. The behavior of vertical LTES units equipped partially with radial fins was investigated by Pu *et al.* [63] and it was concluded that a design that is fully finned along the vertical axis is more effective in reducing the charging time.

In an experimental study performed by Tiari and Hockins [64], the effects of radial fins on the melting and solidification of RT55 in a vertical LTES were investigated. It was reported that by just adding four fins, the charging time decreased by 81.9% and the discharging time by 70% whereas the difference between the eight-fin and four-fin designs was not significant. More recently, a comparison between the effects of radial fin and porous material implementation in an LTES unit was investigated, and the results indicated the superiority of radial fins when it comes to charging time, power, and energy density [65].

Radial fins have proven to be effective in the enhancement of LTES performance, though with some shortcomings, such as hindering the natural convective flows during the melting. Additionally, most of the studies have focused on charging characteristics. In the next subsection, another popular fin design in LTES units is discussed with a short literature review of recent studies.

1.3.3 Longitudinal Fins

A common type of fin design used in LTES units is longitudinal configuration. Figure 1.5 depicts a typical LTES unit with longitudinal fins in the z -direction. With the utilization of longitudinal fins, heat transfer improves in the radial and longitudinal directions, resulting in lower charging/discharging times. Similar to radial fins, their implementation is fairly simple in latent storage units and many studies have

investigated their performance. A summary of works related to the application of longitudinal fins in LTES units is presented in this section.

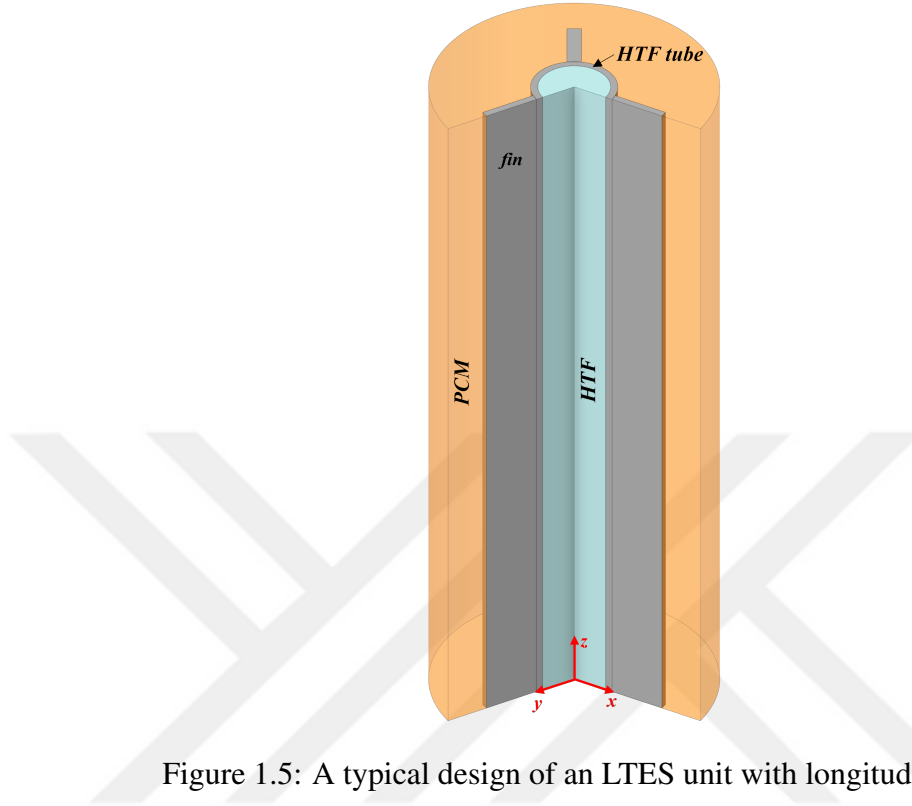


Figure 1.5: A typical design of an LTES unit with longitudinal fins

A set of experimental investigations were performed by Rathod and Banerjee [66] to study the effects of longitudinal fins on shortening the melting/solidification time. They reported that the charging time can be decreased by 24.5% while the discharge time can be reduced up to 43.6%. It was also established that the inlet mass flow rate has a less significant effect compared to the inlet temperature. Mahdi and Nsofor [67] used a 2D model (on the xy -plane) for investigating the effects of fins and nanoparticles in a triplex-tube LTES unit on solidification and concluded that using fins alone resulted in the shortest discharge time. Deng *et al.* [68] used a similar 2D numerical model to the model of Mahdi and Nsofor and studied several parameters including the number, lengths, and configuration of fins, and found that in addition to charging temperature, the mentioned parameters affect the charging performance of an LTES unit with longitudinal fins.

A longitudinal vertical LTES unit was studied experimentally and numerically by Kir-

incic *et al.* [69] and it was shown that the effects of natural convection dominate the melting process while the solidification is mainly dependent on conduction. Li and Tao [70] compared the annular and longitudinal fins in a horizontal LTES unit and established that longitudinal fins restrict the effects of natural convection on melting. They proposed a composite design that utilizes annular fins at the bottom and longitudinal fins at the top of the storage tank. The composite design resulted in better performance when compared to each fin design individually. In a horizontal LTES unit, due to natural convection, the top part of the LTES tank melts faster though solid PCM remains at the bottom of the tank. Utilizing the asymmetrical nature of melting in horizontal LTES units, Patel *et al.* [71] designed an LTES unit with eccentric longitudinal fins and established that using positive eccentricity (downwards) reduced the charging time while the negative eccentricity (upwards) helps with discharge time reduction.

Due to their simplicity, ease of attachment to the HTF tube, and large surface area, longitudinal fins have proven to be a viable solution for the reduction of melting/solidification time in LTES units. However, the basic design is rarely altered to compensate for its disadvantages, such as the hindrance of natural convection flows. Additionally, as discussed, the simple longitudinal fins have worse performance when compared to annular fins and authors have foreseen novel designs to improve the longitudinal fins. The next section discusses different iterations of annular and longitudinal designs.

1.3.4 Nature-inspired Fins

The conventional radial and longitudinal fin designs are generally limited due to their simple design in terms of charging/discharging improvement in LTES units. More specifically, they usually have a non-uniform temperature distribution in the transverse direction of HTF flow (x - or y -directions in Figures 1.3, 1.4 and 1.5). To compensate, several modifications have been introduced such as double and quadruple helical fins [72], Y-shaped fins [73] and spiral fins [74, 75]. More recently, new types of fins inspired by existing patterns in nature have emerged that take advantage of natural inherent properties such as the large surface area to volume ratio.

The nature-inspired fins typically use fractal geometries and fall into three main

shapes: tree-like, snowflake-like, and spiderweb-like geometries as shown in Figures 1.6a to 1.6c respectively. These types of fins often use a profile in the xy -plane and extrude this profile along the HTF direction. These fins have gained attention in recent years due to their ability to promote more efficient heat transfer compared to conventional fin designs, and have been explored for a variety of applications. However, there are also limitations to the use of fractal fins that must be considered in their design and implementation. This section provides a literature survey on fractal fins and discusses their advantages and disadvantages.

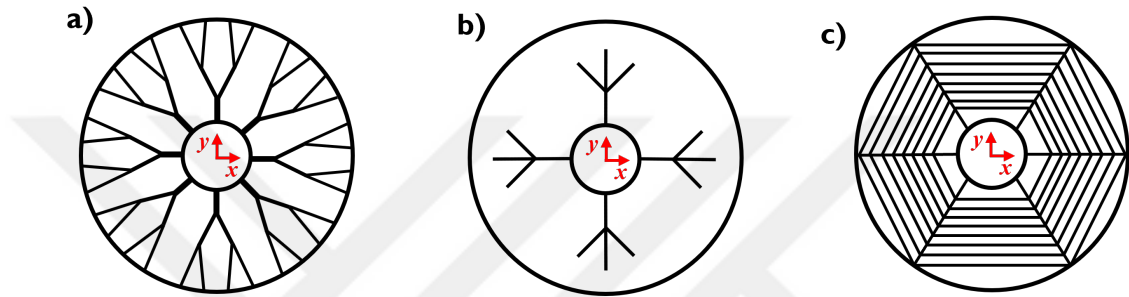


Figure 1.6: Nature-inspired fins: **a)** tree, **b)** snowflake, and **c)** spiderweb

Tree-like fins: Tree-shaped fins are the most studied types of fractal fins. The design is inspired by the distribution of tree branches on the basis that this is the most efficient way of water transport evolved through time and might be effective for heat transfer purposes. The tree-shaped fins are a more advanced iteration of branched fins and several studies, mostly numerical, have been performed in recent years to investigate their behavior in LTES units. Yu *et al.* [76] assumed a numerical domain (similar to the xy -plane in Figure 1.6a) and designed a fractal tree-like fin structure. It was established that the introduced fin suppresses the natural convection effects while increasing the conduction heat transfer during the melting. Compared to longitudinal fins, the tree-like fins performed 26.7% better in terms of charging time reduction and offered more temperature uniformity in the radial direction, resulting in a 45.4% improvement in heat storage rate. Liu *et al.* [77] came up with an uneven tree-like geometry and performed a series of 2D numerical simulations similar to Yu *et al.* They established that with the uneven distribution of fins, the conduction enhancement outperformed the natural convection suppression, and a more uniform radial

temperature distribution can be achieved compared to evenly distributed branches. Deng *et al.* [78] experimentally and numerically showed that fern-fractal fins have shorter melting times compared to conventional fins due to a boost in conduction effects. However, the convection effects are suppressed, especially in the initial melting period. Tian *et al.* [79] introduced a novel design that is similar to leaf structure, which eventuated in significant reductions in melting/solidification times. Huang and Liu [80] stated that the 2D assumption in most of the tree-like fin studies is flawed due to the gradients in the HTF direction, and performed a series of 3D numerical simulations that also consider the effects of flow direction on charging and discharging processes. Several important findings were stated in their work namely, the domination of conduction in the discharge process to the point that natural convection can be neglected, the restriction of natural convection in the tree-shape structures during melting, and an improvement of charging and discharging times up to 34.5% and 49.2%, respectively when compared to longitudinal fins with the same PCM/fin ratio. Shi *et al.* [81] applied the concept of tree-like fins to a triplex heat exchanger that uses waste heat as its energy source and obtained around 27% shorter melting time when compared to traditional longitudinal fins. They also performed an optimization on the tree-like fins, which further improved the melting time. A comprehensive experimental and 3D numerical study by Huang *et al.* [82] was performed with three different fin designs in a horizontal LTES unit; conventional longitudinal, uniformly distributed tree-like and a tree-like design with more fin density at the bottom of the LTES tank to take advantage of non-uniform melting of PCM. The melting time was shortened by 9% while the solidification time was elongated by around 57%.

Snowflake-like fins: Inspired by the snowflake crystals, several fin designs for LTES units have been studied to improve their melting/solidification performance. Sheikholeslami *et al.* [83] proposed a snowflake-like fin configuration to reduce the discharge time and concluded that the fractal fin nearly halves the discharge time compared to longitudinal fins. Hosseinzadeh *et al.* and Alizadeh *et al.* [84,85] investigated various snowflake-like geometries for discharge time reduction alongside the effects of nanoparticle addition in the presence of radiation numerically. They established that by an increase of fractal factor, the complexity of the structure, the snowflake-like fin performed better. Vogel and Johnson [86] studied the effects of natural convection

in a novel fin design based on a combination of snowflake and tree-shaped geometries and concluded that as the fins get more complex and the spacing between them decreases, the convective flows diminish. Additionally, they suggested that due to the promising performance of nature-inspired fins in LTES units, more fractal geometries need to be studied. Based on the inherent characteristics of snowflake design which fills less volume compared to tree-like shape, Ren *et al.* [87] numerically studied melting in a vertical LTES unit that resulted in about 45% reduction in total melting time compared to longitudinal fins.

Spiderweb-like fins: The LTES units with spider-like fin designs are based on their interesting geometrical characteristic, such as effective moisture transport. However, only a few studies have investigated the spiderweb fin configuration in LTES units. To decrease the total solidification time, Wu *et al.* and Ren *et al.* [87,88] numerically investigated the spiderweb-like fins in vertical LTES units and reported up to 47.9% decrease in total discharge time. More studies need to be performed to characterize the fins based on spiderweb geometry, especially when melting is considered.

1.4 Motivation and Problem Statement

The literature reviewed in this study highlights the developments in fin design over the past decade for use in LTES units to address the issue of PCM low thermal conductivity, which is a common challenge in these systems. The nature-inspired fractal fins due to their unique characteristics have proven to be more effective in melting/solidification time reduction in LTES systems. However, the studies taking advantage of fractal fins have some major deficiencies, which are listed below:

- In the melting process, their role in suppressing the natural convective flows is inevitable due to their complex design that introduces extra shear stress in the opposite direction of buoyancy-driven flow during charging. Additionally, their configuration divides the PCM into small cavities with no connection, hindering the free flow of liquid PCM.
- The solidification is often overlooked, even though it is usually the limiting factor when designing an LTES tank. In addition, the investigations that consider

the discharge process, the fractal fins performed worse when compared to conventional fins, which might be attributed to the fact that their design is based on melting characteristics. Additionally, only a handful of studies consider both the melting and solidification performances of nature-inspired fins together.

- Even though the motivation for designing such fins (especially the tree-like fins) is temperature uniformity, except in a few studies, only a 2D numerical simulation is investigated which does not provide any information about the PCM melting/solidification in the direction of HTF flow. This contrasts with the real behavior of both horizontal and vertical LTES units, where the non-melted PCM is left at the bottom of the tanks.
- The design complexity of the available nature-inspired fins makes them questionable in real-life applicability, which explains the scarcity of experimental investigations.

The current work introduces a new type of nature-inspired fins that utilize the Fibonacci sequence, which is one of the most prominent mathematical patterns found in nature. Figures 1.7a to 1.7c show the Fibonacci pattern in seed arrangements of sunflowers and pine and the leaf patterns of the *Aeonium Tabuliforme* plant respectively.



Figure 1.7: Fibonacci series found in nature: **a)** sunflower seeds, **b)** pine seeds, and **c)** *Aeonium Tabuliforme* plant

The Fibonacci spirals have shown some interesting characteristics in fluid flow and heat transfer phenomena. To increase the mixing in low-velocity mixtures, the Fibonacci spiral has shown promising results due to the introduction of secondary flows which accelerate and spread the advective-dominant regions [89,90] that has a potential to accelerate the natural convective flows occurring during the melting process.

Additionally, the Fibonacci-inspired microchannel has shown effective cooling performances due to its unique geometry that promotes conduction [91]. Therefore, the application of Fibonacci-based fin design in LTES units is promising since it could improve conduction and convection and enhance the melting/solidification characteristics of PCM-based storage units. Additionally, the Fibonacci pattern presents the most efficient configurations for the seeds and leaves to be compacted in a confined space, which might be useful when the temperature uniformity in an LTES unit is of concern. Furthermore, their design is less complex when compared to other studied fractal geometries, which makes them more viable for practical applications.

In the current work, a novel fin design is suggested based on the Fibonacci sequence for vertical LTES units. A series of experimental investigations are performed to characterize the melting and solidification processes of the proposed design. The experimental test piece is manufactured using an advanced wire-cutting method. The liquid fraction is used to characterize the phase change processes based on the volume expansion method. This method omits the need of using thermocouples inside the PCM domain, which may disturb the liquid PCM flow. To observe the effects of conduction and convection, a comprehensive set of 3D numerical simulations is performed and validated for each case of experimental investigations. Additionally, the proposed design is compared to an LTES unit that uses traditional longitudinal fins numerically.



CHAPTER 2

METHODOLOGY

2.1 Fin Geometry

Fibonacci sequence dates back to 200 BC, in which each number is the summation of two prior numbers [92]. The Fibonacci sequence has many interesting properties and appears in various areas of mathematics and science, including computer science, biology, and art. In engineering, the Fibonacci sequence has been used to optimize the shapes of structures and systems, such as antennae and solar panels, for improved performance [93, 94]. The Fibonacci spiral can be obtained by drawing squares with dimensions based on the Fibonacci sequence and connecting their corners with an arc, as depicted in Figure 2.1.

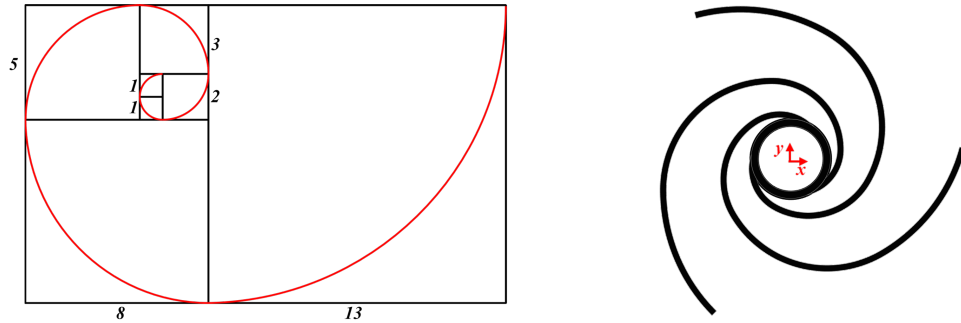


Figure 2.1: Fibonacci sequence and generated fin

Since a spiral based on the Fibonacci sequence is self-repeating, it is categorized as a fractal geometry. To form Fibonacci-inspired fins, a thickness is given to the curves and the original curve is duplicated with equal angles around the center (the z -direction). Then the intersected parts with the HTF tube are deleted.

2.2 Experimental Work

2.2.1 Fin Manufacturing

The test piece unit includes an HTF tube on which the fins described in Section 2.1 are attached. In order to avoid the thermal contact resistances between the HTF tube and fins, a manufacturing method that produces the geometry as a singular unit needs to be employed. Wire cutting is a manufacturing process that involves using a thin wire or wire-like tool to cut through a material. It is a precise and accurate method that can cut a wide range of materials, including metals, plastics, and ceramics.

The tolerances for the wire-cutting technique are around $\pm 0.005 \text{ mm}$ which is suitable for the macro dimensions of the current study. The geometry is cut from a single block of aluminum 6061 series, the photographs of which are given in Figure 2.2. In addition to the wire-cutting method, the use of a water jet to cut the fins out of a solid block of aluminum was attempted, but it proved unsuccessful.

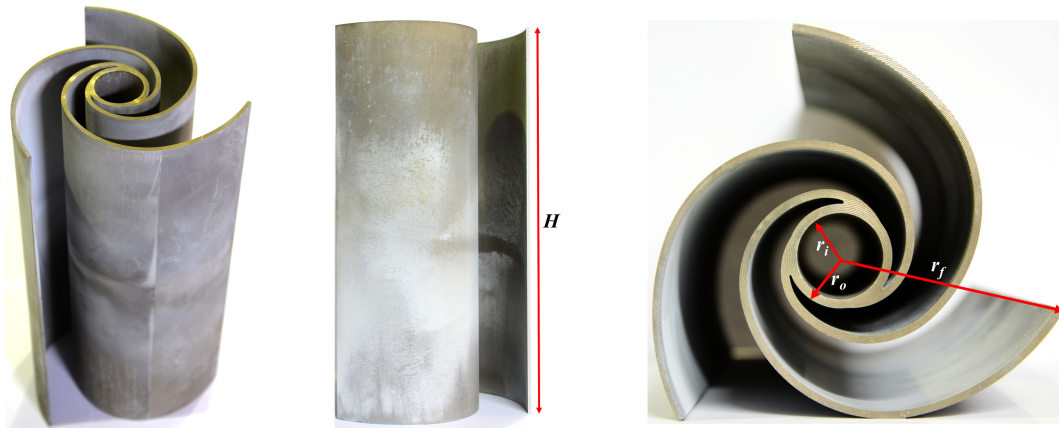


Figure 2.2: The wire-cut fin and its dimension

To ensure a smooth finish, all of the surfaces are polished after the wire-cutting process. A smooth, non-porous surface is important because there is a risk of the PCM sticking to the surfaces during the melting and solidification processes, which is not desirable. The dimensions of the test piece are given in Table 2.1 which are measured with an accuracy of 0.05 mm .

To compare the finned geometry with a conventional unfinned LTES unit, a tube of the same height and inner and outer diameters was obtained with the same thermophysical properties of the finned case. The density, specific heat capacity, and thermal conductivity of aluminum series 6061 are 2700 kg/m^3 , 896 J/kgK , and 167 W/mK , respectively [95].

Table 2.1: The dimensions of the finned geometry

Dimension	Value (mm)
H	200
r_i	10.50
r_o	12.75
r_f	56.30

2.2.2 PCM Selection and Characterization

There are several important factors to consider when selecting a suitable PCM for a study, including its melting temperature, latent heat, and thermal conductivity. However, other factors such as toxicity and availability may also influence the choice of PCM. In this study, the focus is on low to medium temperature ranges, where organic PCMs like paraffin are commonly available.

Due to its availability, RT42 PCM from Rubitherm was chosen for the current work. RT42 is an organic PCM based on paraffin, which has been studied in various research works in the literature. However, there are discrepancies in the reported thermophysical properties of RT42, which can make it difficult to accurately assess its behavior.

In this study, some of the properties that could be measured with the available laboratory equipment were determined, while others were obtained from the manufacturer or other researchers. A discussion of each thermophysical property, including the methods used to measure them in this research, is provided in this section.

Phase change temperature range: PCMs do not undergo phase change at a constant temperature, but rather the phase change takes place within a range of temperatures during which latent heat is released/stored. According to Rubitherm, this temperature

range is $38 - 43^{\circ}\text{C}$ for the melting process, while it is $43 - 37^{\circ}\text{C}$ for the solidification process. The independent measurements of melting temperature with means of a calorimeter in the literature are scarce, and only Siyabi *et al.* [96] has performed such measurements which indicated different values compared to the manufacturer's data. They also mentioned that with different heating rates, the phase change temperature changes, which has also been reported for other PCMs [55].

In the current work, the values provided by the manufacturer were used for the thermophysical properties of the PCM, as there is only one other source of information besides the manufacturer's datasheet.

Density: It is evident that the density of a PCM changes with the phase change process because the molecular structure of the material is rearranged as it transitions from one phase to another. During a phase change, the distance between the molecules in the material changes, which can alter the overall density of the material.

The provided density data by the manufacturer ($\rho_{solid} = 880 \text{ kg/m}^3 @ 15^{\circ}\text{C}$ and $\rho_{liquid} = 760 \text{ kg/m}^3 @ 80^{\circ}\text{C}$) did not result in the same volume change during the phase change process observed in the experiments. This indicates that there may be discrepancies in the manufacturer's data, or that other factors may be affecting the density and volume of the PCM during the phase change process.

Therefore, the density of the liquid and solid phases of the PCM were measured in order to ensure that reliable values were used in the calculations. The PCM liquid density was measured using a TAG hydrometer (depicted in Figure 2.3). The melted PCM was poured into a glass vial and then heated up to 72°C using an industrial-grade oven. The temperature of PCM was controlled using two T-type thermocouples. Once the PCM was out of the oven, the measurements began as the temperature of the PCM dropped due to heat dissipation to the environment. Figure 2.3 shows the configuration used for measuring the liquid PCM density as a function of temperature and Figure 2.4 depicts the change of PCM liquid density as a function of temperature which presents a nearly linear behavior.

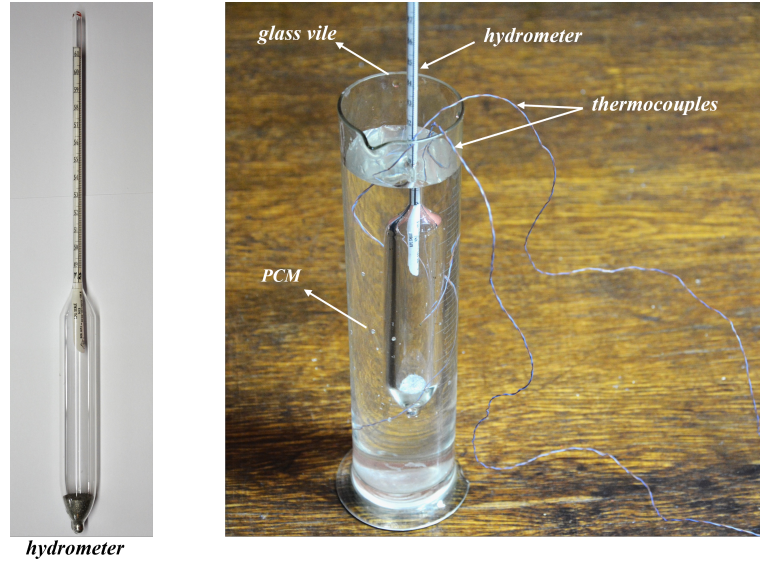


Figure 2.3: Liquid PCM density measurements at different temperatures

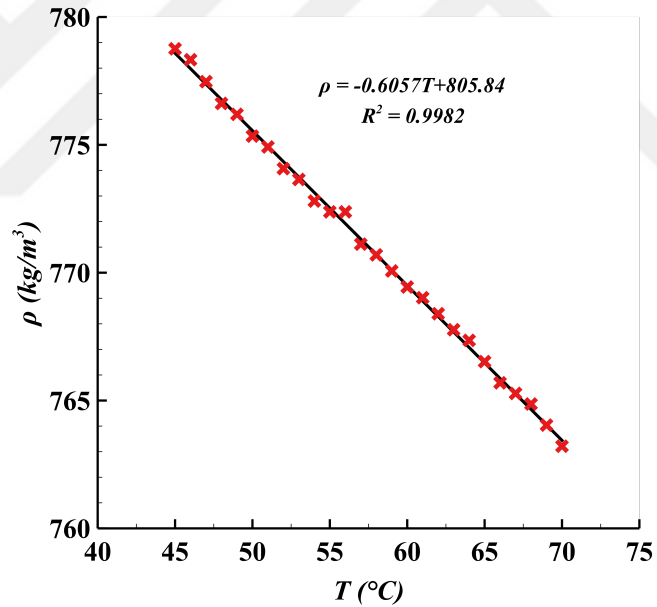


Figure 2.4: Measured liquid PCM density as a function of temperature

It is important to note that the current measurements are not conducted in a fully controlled environment, and the density could be a function of the heating/cooling rate. This is because the rate of phase change is directly related to the temperature difference within the PCM. However, the relationship between heating rate and den-

sity is not necessarily linear and can depend on the specific properties of the PCM. Some PCMs may have a higher density at lower heating rates, while others may have a lower density at higher heating rates [97].

Volumetric thermal expansion coefficient: The volumetric thermal expansion coefficient is defined as the fractional change in volume per unit temperature change. It is a measure of the tendency of a material to expand or contract as its temperature changes and is defined as in Eqn. 2.1:

$$\beta = -\frac{1}{\rho} \left(\frac{\partial \rho}{\partial T} \right)_p \quad (2.1)$$

According to Figure 2.4, the value of $7.85 \times 10^{-4} K^{-1}$ is obtained based on the density measurements for the volumetric thermal expansion coefficient. However, various values have been reported in the literature ranging between $1 \times 10^{-4} K^{-1}$ and $8 \times 10^{-4} K^{-1}$ [98–101]. In the current study, an average value of $4 \times 10^{-4} K^{-1}$ is used.

Thermal conductivity: Rubitherm states a constant thermal conductivity of $0.2 W/mK$ for both phases [20] though the literature provides different thermal conductivity values for liquid and solid phases. The thermal conductivity of a material is generally related to the movement of heat-carrying particles, such as atoms or molecules. In the solid phase, these particles are typically more densely packed, which can lead to a higher thermal conductivity compared to the liquid phase. Motahar and Jahangiri [102] reported a value of $0.254 W/mK$ at $20^\circ C$ and $0.158 W/mK$ at $60^\circ C$. In the current study, the solid PCM thermal conductivity was measured at $25^\circ C$ using HOT DISK 2500S which uses the transient plane source (TPS) technique for measuring the thermal conductivity of a material. In the TPS method, a temperature-sensor-embedded heater is sandwiched between two samples of PCM, and a sufficient voltage is passed through the sensor to increase its temperature. Since the increase in temperature results in voltage changes within the sensor, the heat flow between the PCM and the sensor can be evaluated based on the voltage variations. With the time-dependent heat flow, the thermal conductivity of the PCM can be evaluated [103,104]. To test the accuracy of the measurements, the thermal conductivities of acrylic and

NBR rubber at $25^{\circ}C$ were also measured and compared to the literature, the values of which are presented in Table 2.2.

Table 2.2: Thermal conductivity of acrylic and NBR rubber

Material	Data in literature (W/mK)	Measured value (W/mK)
Acrylic	0.187-0.209 [105]	$0.19 \pm 5\%$
NBR rubber	0.24 [106]	$0.24 \pm 5\%$

The measured values are similar to those reported in the literature. Therefore, it can be established that for solid materials with a similar thermal conductivity to RT42, the thermal conductivity measurements are reliable. In the current investigation, solid PCM thermal conductivity was measured as $0.26 W/mK$ at $25^{\circ}C$ and for the liquid phase, the value provided in the literature is used. Other properties of RT42, namely specific heat capacity [20], dynamic viscosity [102] and latent heat [20], are consistent throughout the literature. Table 2.3 summarizes the thermophysical properties of RT42, including the values from the literature and measured values.

Table 2.3: Thermophysical properties of RT42

Property	Data in literature	Measured value
Melting range ($^{\circ}C$)	38 – 43	-
Solidification range ($^{\circ}C$)	43 – 37	-
Density, ρ (kg/m^3)	880 @ $15^{\circ}C$ and 760 @ $80^{\circ}C$	$-6057 T + 805.84$ for $45^{\circ}C < T < 70^{\circ}C$ and 845 @ $25^{\circ}C$
Volumetric thermal expansion coefficient, β ($1/K$)	$1 \times 10^{-4} - 8 \times 10^{-4}$	7.85×10^{-4}
Thermal conductivity, k (W/mK)	0.254 @ $20^{\circ}C$ and 0.158 @ $60^{\circ}C$	0.26 @ $25^{\circ}C$
Dynamic viscosity, μ (kg/ms)	0.02351	-
Specific heat capacity, c_p (kJ/kgK)	2	-
Latent heat, L (kJ/kg)	144	-

2.2.3 Conceptual Design and Experimental Setup

The experimental setup needs to be designed such that the melting/solidification processes can reliably be performed and repeated. There are several important factors to consider when designing an experimental setup for a phase change process in a vertical shell-and-tube-like geometry:

- The HTF temperature and mass flow rate need to be controllable.
- The direction of HTF must be designed.
- The test piece must have no leakage issues.
- The state of phase change must be monitored.

Based on the mentioned factors, a conceptual design of the experimental setup is considered with the main experimental loop comprising of the test piece, a micropump, and a water heater/chiller as depicted in Figure 2.5.

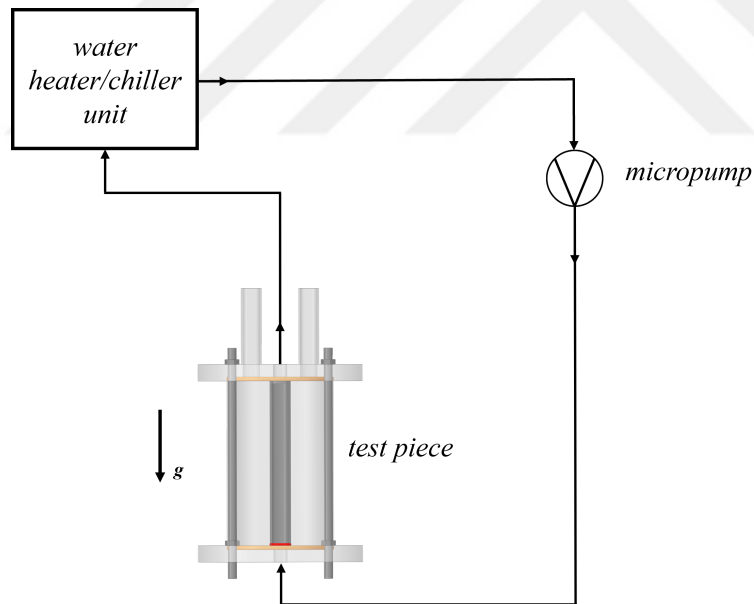


Figure 2.5: Schematics of the test setup and water flow direction

The direction of HTF (water) is from bottom to top in all the cases. Such direction is chosen so that during the solidification process, due to density difference, the possible void spaces are filled with PCM [54] which prevents the entrapment of air bubbles

in the solid PCM. Several attempts were made to flow the HTF in a top-to-bottom direction, but this resulted in the entrapment of small air bubbles that disrupted the melting process. As a result, the bottom-to-top HTF direction was chosen for all melting/solidification cases, with or without fins.

Figure 2.6a depicts the test piece, while Figure 2.6b represents its exploded view. The test piece is comprised of an acrylic shell for the LTES tank with the inner radius of $r_s = 60 \text{ mm}$, the thickness of $\delta = 5 \text{ mm}$, and the height of $H = 200 \text{ mm}$. Figure 2.6c shows the primary dimension of the LTES tank.

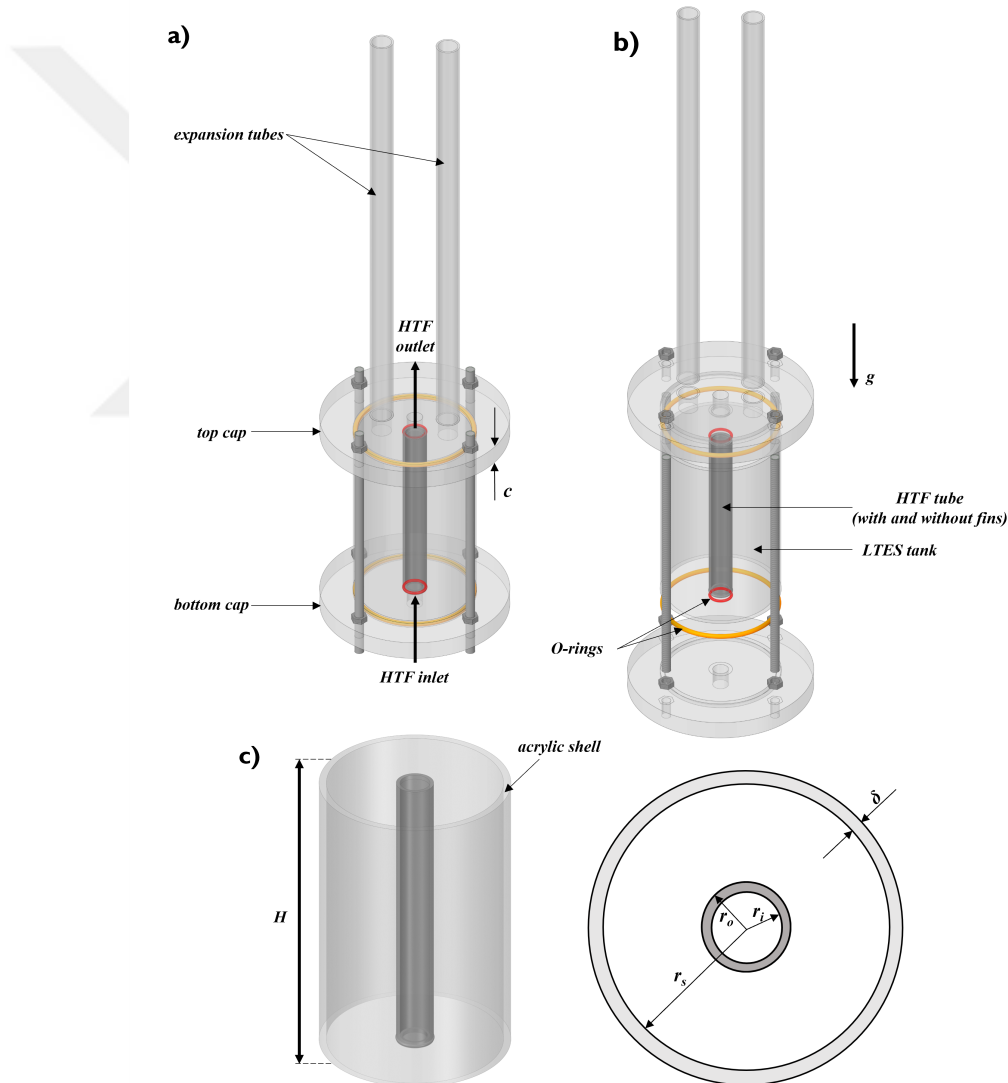


Figure 2.6: LTES tank: **a)** unfinned test piece, **b)** exploded view, and **c)** dimensions

The acrylic shell was chosen due to its transparency so that the phase change process can be viewed and tracked from outside. On the other hand, the thermal conductivity of acrylic is low, which can prevent heat dissipation to the environment to an extent. The acrylic shell is sandwiched between two thick acrylic caps, $c = 20\text{ mm}$, which are tightened with four equally-spaced bolts and knots. The knots were tightened carefully to make sure that the distance between the top and bottom caps is constant. In terms of leakage, the more important connection was the location where the expansion tubes meet the top cap. To avert the risk of leakage, the expansion tubes were merged into the upper cap using chloroform and acrylic dust. After the merging process, a heat-resistive adhesive was utilized to fill the gaps.

In order to maintain the water tightness of the LTES tank, temperature-resistant O-rings were placed between the HTF tube, acrylic shell, and caps. These O-rings are made of materials that can withstand high temperatures, silicone for the large O-rings, and viton for the small O-rings, and provide a flexible seal to prevent leaks. Additionally, the bottom cap is attached to the acrylic shell with a temperature-resistant silicone sealant, which acts as a sealing agent to further ensure the tank's water tightness. The water tightness of the finned and unfinned test pieces was examined through flowing hot water (around 90°C) inside the HTF tube for a day that simulated the maximum thermal expansion for the fittings.

Most of the studies related to PCMs use thermocouples embedded in the PCM to monitor the state of charging/discharging. However, it has been reported in the literature that the use of thermocouples in the PCM can alter the natural behavior of PCM during the melting/solidification [107]. Additionally, there exists a possibility that the thermocouples interfere with the natural convective flows in the liquid PCM. Therefore, to avert this issue, two expansion tubes with an inner diameter of 21 mm and a height of 500 mm are placed on the top cap open to the PCM where liquid PCM can freely flow. With these expansion tubes, the need for embedding thermocouples within the PCM is eliminated. The expansion tubes allow a direct derivation of liquid fraction based on the expanded liquid PCM height, which is an indication of the state of melting/solidification. The dimensions of the expansion tubes are selected such that they can accommodate the expanded volume of the PCM during phase change. This method has been effectively used in several studies to calculate the evolution

of liquid fraction during the charging processes [65, 108–111], however, the current work would be the first investigation using the expansion tubes for monitoring the liquid fraction for melting and solidification processes in a shell-and-tube type LTES.

Based on the mentioned conceptual design of the experiments using the volume expansion method, an experimental setup is constructed. The experimental setup includes a heater/chiller device (Thermo NesLab RTE740), a 4.8 W micropump (UniChrome Zauss) which is controlled by a DC power supply (SIGLENT SPD3303C), a data logger (Agilent 34972A), the test piece and a series of tubes and fittings to direct the flow of HTF. Figure 2.7 depicts the experimental setup and its components.

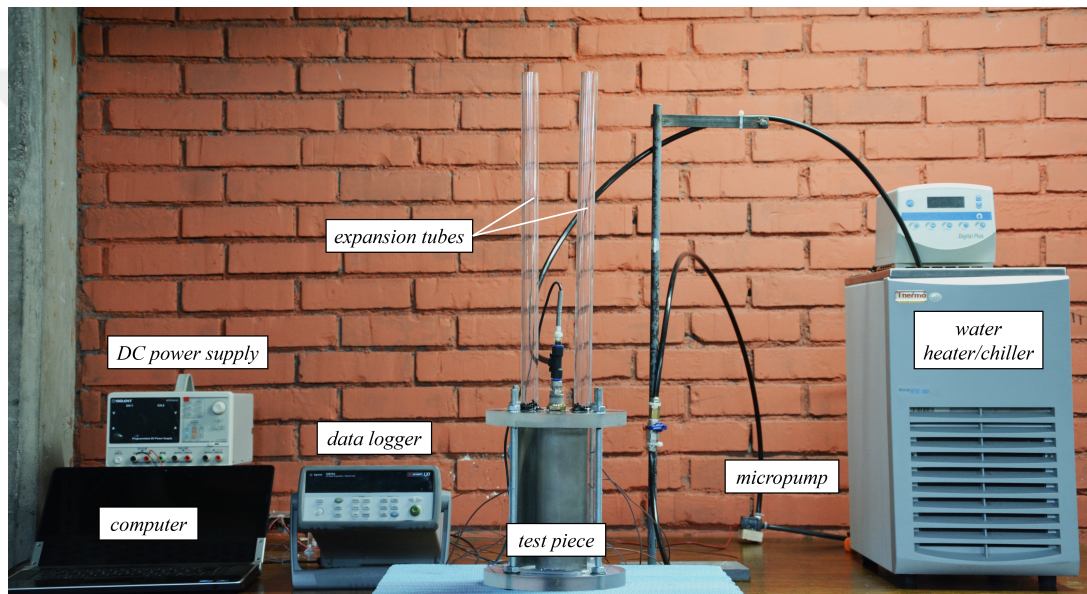


Figure 2.7: Experimental setup and its components

The temperature data from the thermocouples are recorded using the data logger every 10 seconds. A J-type thermocouple is used for monitoring the inlet/outlet water temperatures, while six T-type thermocouples are used to measure the temperature on the outer surface of the acrylic shell, as depicted in Figure 2.8. The reason behind using the J-type thermocouples for measuring the inlet and outlet temperatures is that the T-type thermocouples when in contact with high-temperature water, resulted in lower temperature measurements when compared to the mercury-in-glass thermometer since they did not have any coating on their junction point (unlike the J-type thermocouples) which might have resulted in oxidization over time.

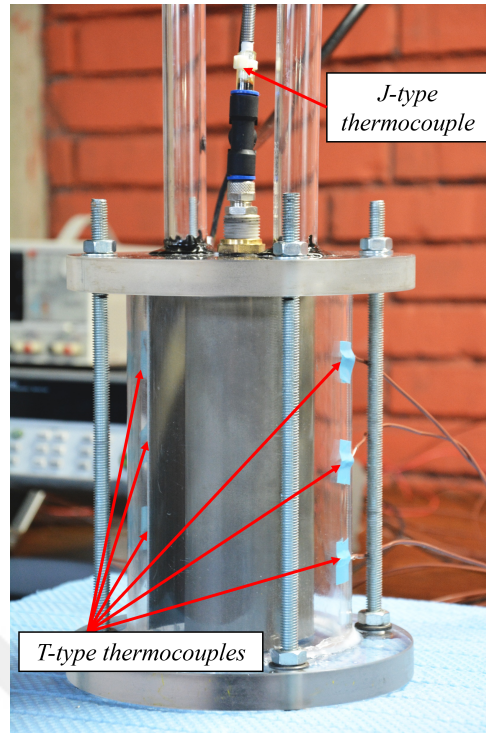


Figure 2.8: Thermocouple locations on the test piece

2.2.4 Experimental Procedure

To ensure reliable results, the experimental work was carefully planned and carried out. Before the experiments, the thermocouples were calibrated in a water bath of constant temperature using a mercury-in-glass thermometer, and the micropump was tested to ensure it was functioning properly. Then, the liquid PCM was filled inside the tank by weight calculated with its respective density value. During the filling process, many air bubbles would trap inside the PCM and to resolve this issue, the melted PCM was left inside the tank for a day while being constantly heated by the HTF, which provided time for the air bubbles to escape.

The melting and solidification processes were performed in a cyclic manner such that the PCM was melted fully, and then the solidification process started by flowing the cold water through the HTF tube.

As the solidification starts, the PCM shrinks because of its higher density in the solid state. This shrinkage is characterized by the reduction of liquid PCM level inside

the expansion tubes, as depicted in Figure 2.9. Next to the expansion tubes, a ruler with the precision of 1 mm was placed and the liquid PCM level inside the tubes was recorded every five minutes by a DSLR camera for calculating the liquid fraction as a function of time.

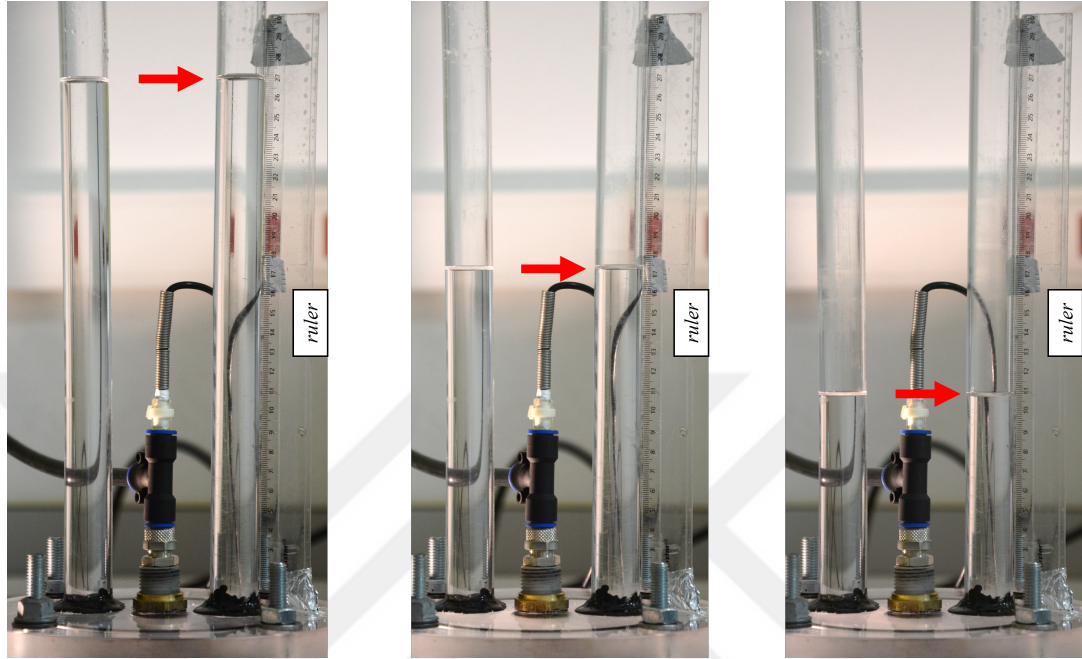


Figure 2.9: Liquid PCM level during solidification (arrows show liquid PCM level)

It is worth noting that because of heat dissipation to the environment and the cooling of the PCM, the PCM inside the expansion tubes begins to solidify over time, making it difficult to accurately measure the PCM level as depicted in Figure 2.10. This issue was more apparent in the unfinned case due to the longer solidification time. To address this issue, the expansion tubes were heated with hot air from a heat gun in short bursts every few minutes. This helped to keep the PCM in a liquid state and allowed for more accurate measurements of the PCM level.

The supplied heat to the expansion tubes affects the liquid fraction evolution which is compensated for by adjusting the liquid fraction during the solidification of unfinned geometry the details of which are provided in Section 2.2.5.

The end of the solidification process was marked when the liquid PCM levels stop moving. Once the solidification was finished, the experiments were stopped by halt-

ing the flow of water. Then, the warm water was flown into the tubes, which marked the start of the melting process. Similar to the solidification process, during the melting, the liquid PCM level was recorded as it rose inside the expansion tubes due to lower density.

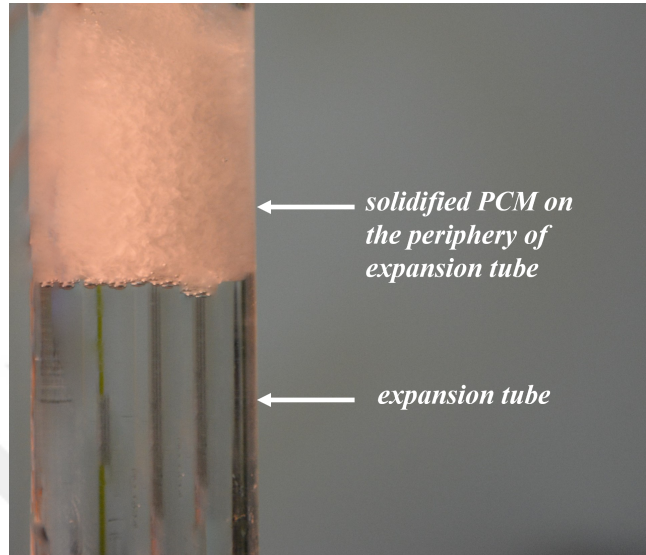


Figure 2.10: The solidified PCM in the expansion tubes

Aside from the liquid PCM level and recorded temperatures by thermocouples, the water mass flow rate was measured using a scale for each process. The experimental matrix includes one solidification and one melting process for each of the finned and unfinned geometries. The experiments were performed three times to ensure repeatability, and the results of the last experiment were recorded.

2.2.5 Data Reduction

Several parameters are extracted from the experiments throughout the melting and solidification processes. This subsection discusses each which are the temperature, HTF mass flow rate, and expansion height values of each experiment.

Liquid fraction: The liquid fraction, f , reveals the state of melting/solidification in the LTES tank. This value is calculated in accordance with Eqn. 2.2 [65]:

$$f = \frac{m_{liquid}}{m_{total}} \quad (2.2)$$

where m_{liquid} is the mass of the melted PCM and m_{total} is the total mass of the PCM. A value of one for liquid fraction indicates a fully melted LTES tank, while a value of zero reflects a fully solid LTES tank. The liquid fraction can be calculated as:

$$V_{tank} + V_e = \frac{m_{liquid}}{\rho_{liquid}} + \frac{m_{total} - m_{liquid}}{\rho_{solid}} \quad (2.3)$$

In Eqn. 2.3, V_{tank} , V_e , ρ_{liquid} and ρ_{solid} are the volume of the TES tank, volume in the expansion tubes and liquid, and solid PCM densities, respectively. Eqn. 2.3 can be rearranged for obtaining the liquid fraction value as:

$$f = \frac{m_{liquid}}{m_{total}} = \frac{\rho_{solid}\rho_{liquid}(V_{tank} + V_e) - \rho_{liquid}m_{total}}{(\rho_{solid} - \rho_{liquid})m_{total}} \quad (2.4)$$

Since the diameter of the tubes is constant, the expanded volume is only dependent on the height of the PCM in the expansion tubes, which is monitored during the experiments. In this study, the evolution of the liquid fraction over time is used as a means of validating the numerical models. This is an important step in ensuring the reliability and effectiveness of the numerical model for simulating the melting and solidification processes.

Supplied heat to the expansion tubes: As mentioned in the previous subsection, during the solidification of the unfinned geometry, to avoid solidification of liquid PCM inside the expansion tubes, a heat gun was used to supply heat to the solidified sections of PCM. An adjustment is applied to the solidification liquid fraction values to account for the supplied heat.

The Churchill and Bernstein [112] relation for the average Nusselt number of a cylinder in cross flow is utilized for calculating the convective heat transfer coefficient between the hot air and the expansion tubes:

$$\overline{Nu}_D = \frac{\bar{h}D}{k_{air}} = 0.3 + \frac{0.62Re_D^{1/2}Pr^{1/3}}{[1 + (0.4/Pr)^{2/3}]^{1/4}} \left[1 + \left(\frac{Re_D}{28200} \right)^{5/8} \right]^{4/5} \quad (2.5)$$

where $\bar{h}$ is the average convective heat transfer coefficient, Pr is the Prandtl number of air and Re_D is the Reynolds number based on the outer diameter of the expansion tubes, D :

$$Re_D = \frac{\rho_{air} U_{hg} D}{\mu_{air}} \quad (2.6)$$

in which U_{hg} is the velocity of air blown from the heat gun that was measured to be 3.5 m/s . With the convective heat transfer coefficient extracted from Eqn. 2.5, the supplied heat to the system at each burst of heating with the heat gun can be evaluated as:

$$Q_{supplied} = \bar{h}A (T_{air} - T_{exp.tube}) \Delta t \quad (2.7)$$

where A , $T_{exp.tube}$, T_{air} and Δt are the area of the expansion tubes where the hot air was blown, the surface temperature of the expansion tubes, the hot air temperature from the heat gun, and the period of hot air blast (≈ 30 seconds). The supplied heat is then extracted from the latent heat at each interval of measurement:

$$Q_{adjusted}^n = Q_{latent}^n - Q_{supplied} = (m_{liquid}^n - m_{liquid}^{n-1})L - Q_{supplied} \quad (2.8)$$

where the superscript n denotes the n th measured interval of liquid fraction.

Then, the adjusted liquid fraction can then be calculated in accordance with the adjusted heat:

$$f_{adjusted}^n = \frac{m_{liquid, adjusted}^n}{m_{total}} = \frac{1}{m_{total}} (m_{liquid, adjusted}^{n-1} - \frac{Q_{adjusted}^n}{L}) \quad (2.9)$$

It is worth mentioning that this adjustment is based on the assumption of the supplied heat being uniformly distributed to all the steps at which the liquid fraction was measured.

HTF inlet temperature and mass flow rate: To maintain consistent and controlled conditions during the melting and solidification processes, the water mass flow rate, $\dot{m}_{HTF}$, was kept constant using the DC power supply. The values for the charging and discharging processes were set to 99 kg/hr and 96 kg/hr, respectively. The slight difference between these values is due to the difference in water temperature at the melting and solidification processes. By keeping the mass flow rate constant, the effects of variations in water flow on the melting and solidification processes are minimized.

The water inlet temperature was set to $T_{inlet, m} = 61.5^\circ C$ and $T_{inlet, s} = 21.5^\circ C$ for charging and discharging, respectively. These values are selected based on the average temperatures at which the maximum latent heat is released ($42^\circ C$) and stored ($41^\circ C$) which was provided by the manufacturer [20]. Therefore, a $20^\circ C$ of superheating during the charging and subcooling during the discharge is set for the PCM.

Initial temperatures for the numerical models: As a result of not using thermocouples inside the PCM domain in the experiments, it is necessary to devise a method for accurately determining the initial temperatures at the start of the melting and solidification processes. The measured values can then be used in numerical models to accurately simulate the processes. Without a reliable method for establishing these initial temperatures, the accuracy of the numerical models may be compromised.

The method is based on conduction assumption inside the PCM domain after each melting/solidification process. To guess the initial temperature of the domain, six T-type thermocouples are placed on the outer surfaces of the acrylic shell with a 180° angle between each three as depicted in Figure 2.11.

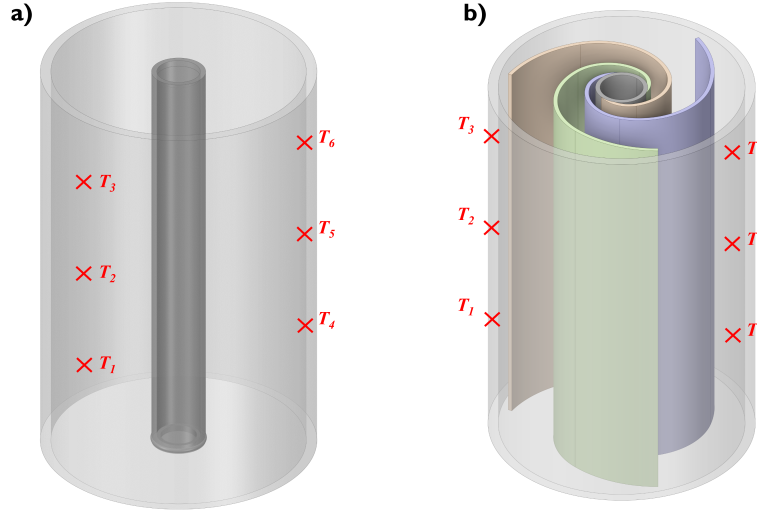


Figure 2.11: Thermocouple locations for initial temperature guess in **a)** unfinned and **b)** finned geometries

The temperatures T_1 through T_6 for each geometry are averaged at the beginning of solidification and melting as:

$$T_{outer,m} = \left(\frac{1}{6} \sum_{i=1}^{i=6} T_i \right)_{@t=t_{initial,m}} \quad (2.10)$$

$$T_{outer,s} = \left(\frac{1}{6} \sum_{i=1}^{i=6} T_i \right)_{@t=t_{initial,s}} \quad (2.11)$$

At the beginning of solidification, since all the PCM is melted, there exists a small temperature gradient inside the PCM. It is important to note that even though the temperature gradient is small, there still exist natural convective flows inside the liquid PCM, and they are neglected for guessing the initial temperature of the PCM domain. On the other hand, at the beginning of melting since the PCM domain is fully solidified, there exists no convective motion of PCM and therefore the conduction-only behavior is valid without any assumptions. With Eqns. 2.10 and 2.11, the outer shell temperature of each finned and unfinned case, are known just before the melting/so-

lidification processes. Since the thermal conductivity of the aluminum is high, the thickness of the HTF tube is small (2 mm) and the temperature difference between the HTF inlet and outlet is small at the end of each phase change process, it is safe to assume that the HTF temperature is very close to the PCM temperature in the vicinity of the inner pipe. Therefore, by averaging the inlet and outlet temperatures, at the beginning of the melting/solidification processes, the inner temperature of the PCM can be obtained:

$$T_{inner,m} = \left(\frac{T_{inlet} + T_{outlet}}{2} \right)_{@t=t_{initial,m}} \quad (2.12)$$

$$T_{inner,s} = \left(\frac{T_{inlet} + T_{outlet}}{2} \right)_{@t=t_{initial,s}} \quad (2.13)$$

Based on the assumptions mentioned, the unfinned and finned cases at the beginning of the melting and solidification processes can be treated as steady-state problems. The domains for guessing the initial temperatures and their boundary conditions are provided in Figures 2.12a and 2.12b for the unfinned and finned cases.

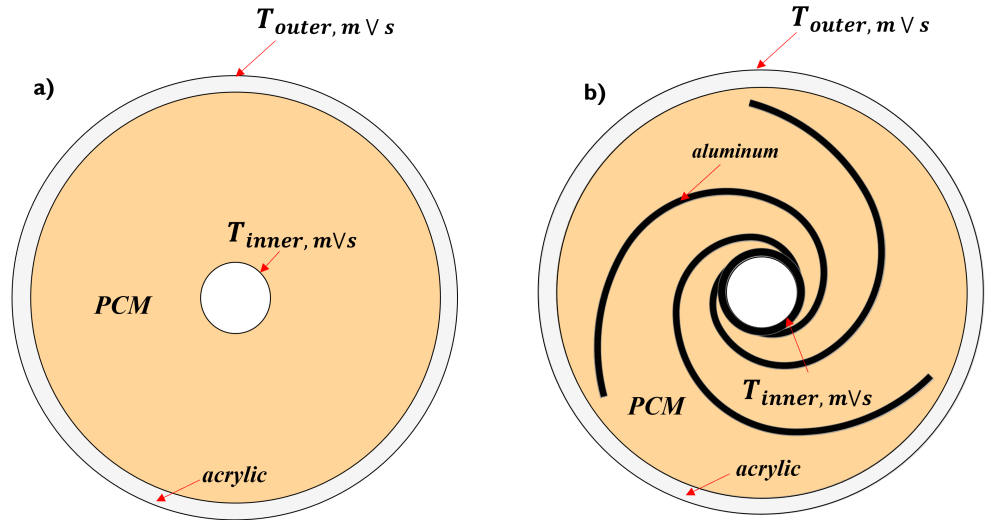


Figure 2.12: Domains of **a)** unfinned and **b)** finned geometries for initial temperature guessing

The solution of this problem for the unfinned case can be resolved analytically by solving the steady-state conduction equation and averaging the temperatures in the domain. On the other hand, for the finned case, a 2D steady-state numerical model is devised based on the mentioned boundary conditions. The surface average temperature in the 2D model then evaluates the initial temperature.

Table 2.4 gives the initial temperature values for the finned and unfinned cases and their respective melting and solidification processes based on the methodology developed in this chapter. These initial temperatures will be used in the numerical modeling of the phase change process described in Section 2.3.

Table 2.4: Initial temperature of the unfinned and finned geometries

Geometry	Phase change process	Initial temperature ($^{\circ}C$)
Unfinned	Melting	26
	Solidification	48
Finned	Melting	28.2
	Solidification	46

Heat dissipation to the environment: To observe the melting/solidification of PCM, in this study, the LTES tanks are not insulated which makes the heat loss to the environment inevitable. For evaluating the heat losses to the environment which will be needed in numerical models, four thermocouples on each side of the acrylic shell of the LTES tank and two thermocouples on each side of the top and bottom acrylic caps are placed as depicted in Figure 2.13.

The experiments related to finding the heat loss values were performed separately, meaning that during the main experiments for determining the liquid fraction, no thermocouples were placed inside the PCM. For calculating the heat losses to the environment, at each instant, one-dimensional conduction is assumed in each of the acrylic parts.

Therefore, it can be stated that the rate of conduction through an acrylic thickness is equal to the rate of heat dissipation through convection to the environment at each instant.

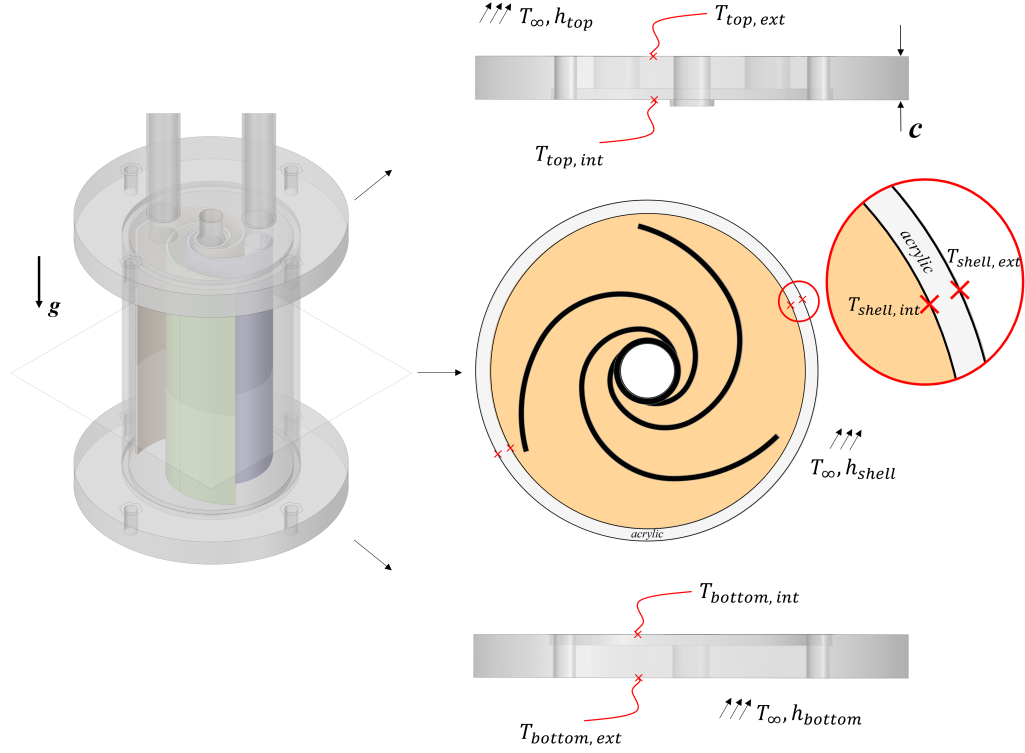


Figure 2.13: Thermocouple for estimating the heat dissipation to the environment

For each surface, the convective heat transfer coefficients can be calculated as:

$$k_{\text{acrylic}} \frac{T_{\text{top}, \text{int}} - T_{\text{top}, \text{ext}}}{c} = h_{\text{top}} (T_{\text{top}, \text{ext}} - T_{\infty}) \quad (2.14)$$

$$k_{\text{acrylic}} \frac{T_{\text{shell}, \text{int}} - T_{\text{shell}, \text{ext}}}{\ln \left(\frac{r_s + \delta}{r_s} \right)} = h_{\text{shell}} (r_s + \delta) (T_{\text{shell}, \text{ext}} - T_{\infty}) \quad (2.15)$$

$$k_{\text{acrylic}} \frac{T_{\text{bottom}, \text{int}} - T_{\text{bottom}, \text{ext}}}{c} = h_{\text{bottom}} (T_{\text{bottom}, \text{ext}} - T_{\infty}) \quad (2.16)$$

In Eqns. 2.14 to 2.16, T_{∞} is the ambient temperature of the room, which is kept constant at 25°C by using a heater and fan configuration controlled by a thermostat. Figure 2.14 depicts the behavior of h_{shell} for the unfinned and finned geometries respectively for melting and solidification processes.

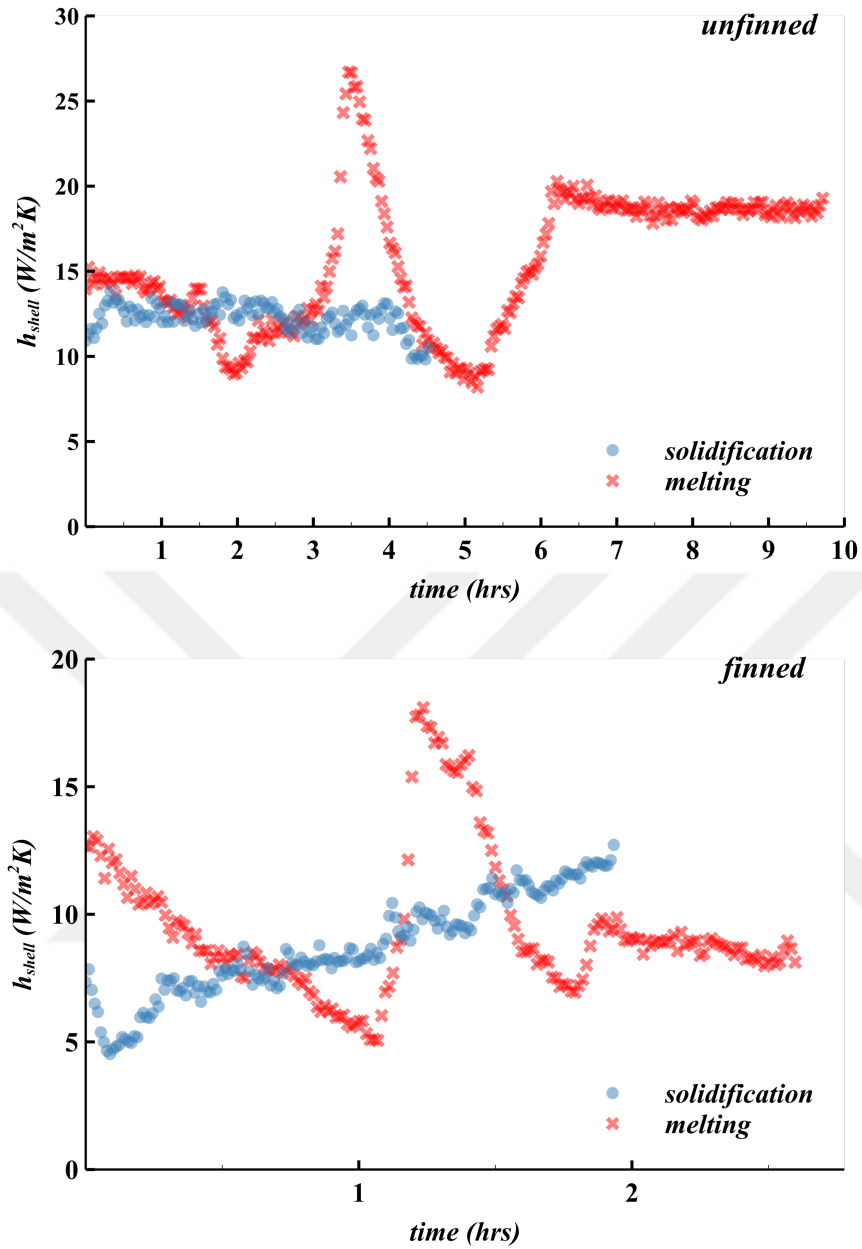


Figure 2.14: Variation of convective heat transfer coefficient on the outer surface of the shell

For the melting process, a distinct jump is observed nearly halfway through the phase change process, which is due to the PCM liquid-solid phase movement near the shell. The temperature difference between the liquid and solid phases of PCM during the experiments passes 10°C which results in a sudden jump in convective heat transfer to the environment with an increase in $T_{shell,ext}$. It is noteworthy that the values

of the convective heat transfer coefficient to the environment are larger than what is typically expected for losses due to natural convection, which generally hovers around $5 \text{ W/m}^2 \text{ K}$ for a vertical cylinder [54]. The humidity of the room was controlled using a fan, resulting in air circulation and measured air velocities of up to 0.1 m/s by an anemometer near the test piece. This supports the values of the calculated convective heat transfer coefficient. Table 2.5 presents the time-averaged values of the convective heat transfer coefficient for the unfinned and finned geometries at the top and bottom parts.

Table 2.5: Time-averaged values of convective heat transfer coefficient for the top and bottom caps

Convective heat transfer coefficient	Geometry	Process	Value ($\text{W/m}^2 \text{ K}$)
h_{top}	Unfinned	Solidification	2
		Melting	4
	Finned	Solidification	1
		Melting	5
h_{bottom}	Unfinned	Solidification	11
		Melting	12
	Finned	Solidification	7
		Melting	8

2.2.6 Uncertainty Analysis

Due to the accuracy of measurement equipment, there are inevitable uncertainties in the experiments. The uncertainty of measured and calculated quantities throughout the experiments is estimated using the Gauss error propagation law, which is a mathematical formula used to estimate the uncertainty in the result of a mathematical operation involving two or more variables [113]. This law states that if a quantity, R , is a function of several independent quantities such that:

$$R = \psi(x_1, x_2, x_3 \dots x_n) \quad (2.17)$$

then, the absolute uncertainty of R can be estimated as:

$$w_R = \left[\left(\frac{\partial R}{\partial x_1} w_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} w_2 \right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n} w_n \right)^2 \right]^{0.5} \quad (2.18)$$

where w_n is the uncertainty of the n th independent quantity. Table 2.6 provides the relative uncertainty values for the measured and reduced parameters. The relative uncertainty values are calculated based on the measured value and the calculated uncertainty values.

It is worth mentioning that the time-dependent uncertainty of liquid fraction, f , is provided on the relative graphs in error bar form in the results section.

Table 2.6: Relative uncertainty of the measured and reduced parameters

Parameter	Relative uncertainty (%)
PCM liquid density, (kg/m^3)	0.056
PCM solid density, (kg/m^3)	0.71
Total PCM mass, (kg)	0.06
Expanded volume, (m^3)	1.45
LTES tank volume, (m^3)	0.18
HTF mass flow rate, (kg/hr)	0.5
Temperatures, ($^{\circ}C$)	0.75

2.3 Numerical Study

2.3.1 Numerical Domains

To understand the behavior of the melting and solidification processes and the contributions of convection and conduction, a series of numerical simulations are constructed based on the experimental setups. These simulations are used to model the behavior of the actual experimental test pieces. Before the introduction of the model-

ing methodology, the geometries are simplified to make them suitable for numerical simulations in terms of complexity and computational costs. By comparing the results of the numerical simulations to the actual experimental data, the effectiveness of the model in predicting the melting and solidification behavior can be evaluated.

For the unfinned geometry, an axisymmetrical 2D domain is assumed based on the geometrical symmetry around the z -axis as depicted in Figure 2.15. Therefore, the 3D domain is reduced to a 2D domain on the xz -plane, which reduces the computational time significantly.

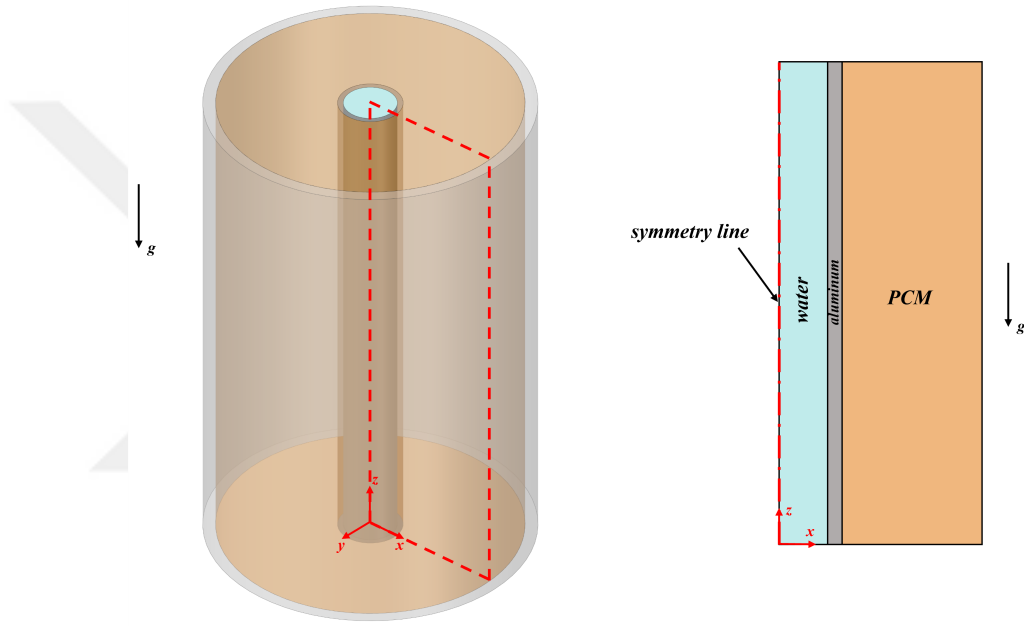


Figure 2.15: Simplified numerical domain of the unfinned geometry

The finned geometry is also simplified into a smaller domain that contains one-third of the elements in the full 3D domain. The finned geometry can be divided into three equal parts as depicted in Figure 2.16 with three division curves which are 120° apart. The division is based on the fact that the fin geometry is inherently periodic along the z -direction which states that any property is periodic with the phase angle of 120° . Figure 2.16a depicts the full 3D domain, while Figure 2.16b depicts the 2D representation of the simplified numerical domain and division curves on the xz -plane. The result of the division is depicted in Figure 2.16c. With this simplification,

it is important to carefully consider the boundary conditions in order to ensure the accuracy of the results which will be discussed in further detail in the next subsection.

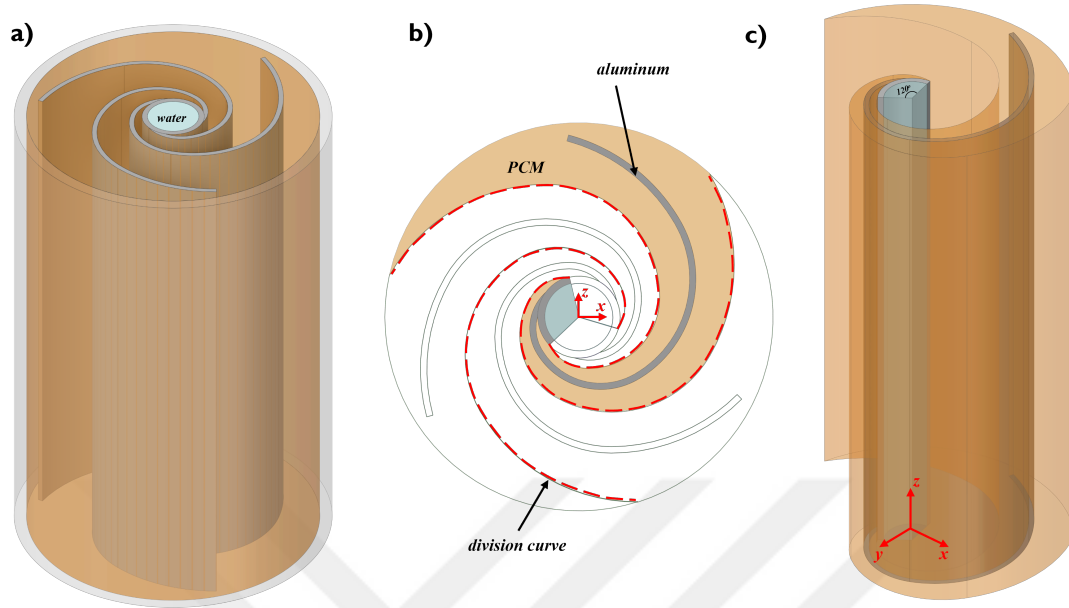


Figure 2.16: Finned geometry: **a)** 3D domain, **b)** division curves, and **c)** simplified domain

2.3.2 Governing Equations, Initial, and Boundary Conditions

The numerical model is based on several assumptions for modeling the melting/solidification processes:

- The flow of water and liquid PCM is unsteady, laminar, and incompressible
- Boussinesq approximation is assumed for modeling the buoyancy-driven flows in the melted PCM
- Viscous dissipation and radiation are negligible in all the domains

The Boussinesq approximation is not always valid in the modeling of flows with the phase change process. If the heat source (here, the surface of the fins and HTF tube) is perpendicular to the melting PCM, then a small layer of liquid PCM is continuously being squeezed by the solid PCM where the effects of density difference between the solid and liquid PCM becomes deterministic in the melting characteristics [96].

However, in the current configuration, the fins are parallel to the melting PCM, and the mentioned phenomenon (commonly known as close-contact melting) does not occur.

The governing equations are the continuity, momentum, and energy equations. Eqns. 2.19 and 2.20 describe the continuity and momentum equations for an unsteady, incompressible fluid flow.

$$\nabla \cdot \mathbf{U} = 0 \quad (2.19)$$

$$\rho \frac{\partial \mathbf{U}}{\partial t} + \rho(\mathbf{U} \cdot \nabla) \mathbf{U} = \mu \nabla^2 \mathbf{U} - \nabla P + \rho \vec{g} \beta (T - T_{ref}) + S_{\mathbf{U}} \quad (2.20)$$

where $\mathbf{U}$, ρ , μ , P , and $\vec{g}$ are the velocity vector, average PCM density (solid and liquid states), dynamic viscosity, pressure, and gravitational acceleration vector, respectively. The first term on the left-hand side of Eqn. 2.20 represents the temporal acceleration of fluid, while the second term is the convective term.

On the right-hand side of Eqn. 2.20, the first and second terms represent the viscous and pressure forces, respectively. The term $\rho \beta g(T - T_{ref})$ depicts the Boussinesq approximation in which β is the volumetric thermal expansion coefficient, T is the temperature and T_{ref} is the solidus temperature of PCM. The Boussinesq approximation neglects the effects of density change on the convective terms of momentum since the density change is a strong function of temperature.

To model the melting/solidification, the enthalpy-porosity model, first introduced by Voller *et al.* [114] is utilized. In this method, the PCM is assumed as a semi-porous material, where the porosity is replaced by the PCM liquid fraction. $S_{\mathbf{U}}$ in the right-hand side of Eqn. 2.20 models the melting and its definition, which is based on a form of Darcy equation for porous media, is:

$$S_{\mathbf{U}} = C_{mush} \frac{(1 - f)^2}{f^3 + \varepsilon} \mathbf{U} \quad (2.21)$$

In Eqn.2.21, f is the PCM liquid fraction and C_{mush} is a mushy zone constant. Additionally, ϵ is a small number (0.001) to avoid division by zero when the liquid fraction is zero. The mushy zone constant determines the intensity of liquid PCM velocity damping near the melt front. This value is ranged between 10^4 and 10^8 [115–117] and is dependent on the morphology of PCM and needs to be determined by comparing the experimental and numerical results. In this study, an investigation considering different mushy zone constant values is performed for both geometries and proper values are selected based on the experimental results. The enthalpy-porosity method considers the melting PCM as a mushy zone where if the liquid fraction is one, the PCM is fully melted and when its value is zero, the PCM is fully solid. The liquid fraction in the experimentations was defined as in Eqn. 2.2. However, in the numerical models, in between the liquid fraction values of one and zero, a linear relationship based on the liquidus, and solidus temperatures are assumed [114]:

$$f = \frac{T - T_{solidus}}{T_{liquidus} - T_{solidus}}, \quad T_{solidus} < T < T_{liquidus} \quad (2.22)$$

The liquidus and solidus temperatures are the temperature at which PCM is completely liquid and solid respectively. The relationship between the temperature and liquid fraction is assumed to be linear even though it might be a non-linear function. However, until now most of the studies that used the enthalpy porosity method successfully used this linear relationship and were able to validate the models based on the experiments [54,55].

For solving the heat transfer phenomenon, the energy equation is given based on the enthalpy formulation in Eqn. 2.23:

$$\rho \frac{\partial i}{\partial t} + \rho \nabla \cdot (\mathbf{U}i) = \nabla \cdot (k \nabla T) \quad (2.23)$$

where k is the thermal conductivity and the enthalpy, i , accounts for the sensible and latent components:

$$i = i_{sensible} + i_{latent} = \int_{T_1}^{T_2} c_p dT + fL \quad (2.24)$$

In Eqn. 2.24, c_p and L are the specific heat capacity and latent heat.

Eqn. 2.23 includes an unsteady term (first term on the left-hand-side), a convective term (second term on the left-hand-side), and a diffusive (conductive) term (the term on the right-hand-side). For the solid domains (HTF tube and fins) only the unsteady and diffusion terms in Eqn. 2.23 are solved.

The initial temperatures are obtained in accordance with the methodology developed in Section 2.2.5 with the values stated in Table 2.4. Both fluid domains are initially at zero gauge pressure.

At the inlet, a constant velocity based on the constant mass flow rate obtained in the experimentation is applied. In addition, the HTF is set to a constant temperature on the basis of melting/solidification experimental analyses:

$$U_{z=0} = \frac{\dot{m}_{HTF}}{\rho \pi r_i^2} \quad (2.25)$$

$$T_{z=0} = T_{inlet, mVs} \quad (2.26)$$

At the water domain outlet, the pressure is set to zero gauge:

$$P_{z=H} = 0 \quad (2.27)$$

Where the HTF meets the HTF tube and the fins contact the PCM domain, the no-slip condition is applied while the interface condition for the conjugate heat transfer modeling is utilized by setting the temperature and heat transfer rate at the interface of each numerical domain (water, aluminum, and PCM) the same. The periodic

boundary conditions are applied for the finned geometry such that at the division curves, all the independent parameters and their spatial derivatives at each instant are equal [118]. Figure 2.17a and 2.17b present the boundary conditions applied to the unfinned and finned numerical domains, respectively.

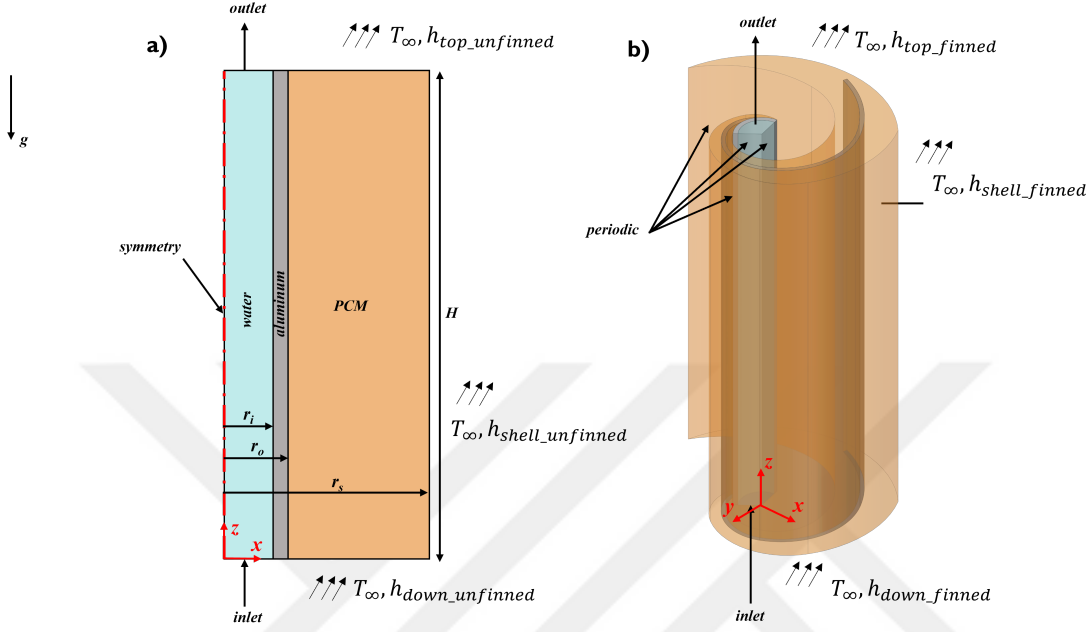


Figure 2.17: Applied boundary conditions to the **a)** unfinned and **b)** finned geometries

The governing equations with the mentioned boundary and initial conditions are discretized with the finite volume method (FVM) using a second-order upwind scheme for momentum and energy equations, while the pressure field is discretized using PREssure STaggering Option (PRESTO!) method. The pressure velocity coupling is performed using the Pressure-Implicit with Splitting of Operators (PISO) scheme, and the time marching is done with the first-order implicit formulation.

The convergence criteria for the residuals of all variables except energy are chosen to be 10^{-5} and for energy, the criterion of 10^{-7} is set. A time step of $0.5s$ is selected for the unfinned cases, while a time step of $0.25s$ is selected for the finned cases. The time steps are picked such that the convergence criteria can be reached after a reasonable number of iterations (less than 40 iterations) at each time step. The geometry creation, meshing, and solving of the governing equations are performed using ANSYS SpaceClaim, ANSYS Meshing, and ANSYS FLUENT 2021 R2.

2.4 Mesh Independence Study

The PCM flow in the melting process is more complex due to natural convection within the liquid PCM, therefore the mesh independence study is performed only for melting for the finned and unfinned geometries.

2.4.1 Unfinned Geometry

The rectangular shape of the unfinned geometry makes it suitable for using a structural grid. To study the effect of element number on the melting behavior of the unfinned geometry, three different structured meshing configurations are investigated. Figure 2.18 shows the time-dependent behavior of the melting for three differently-sized meshing configurations.

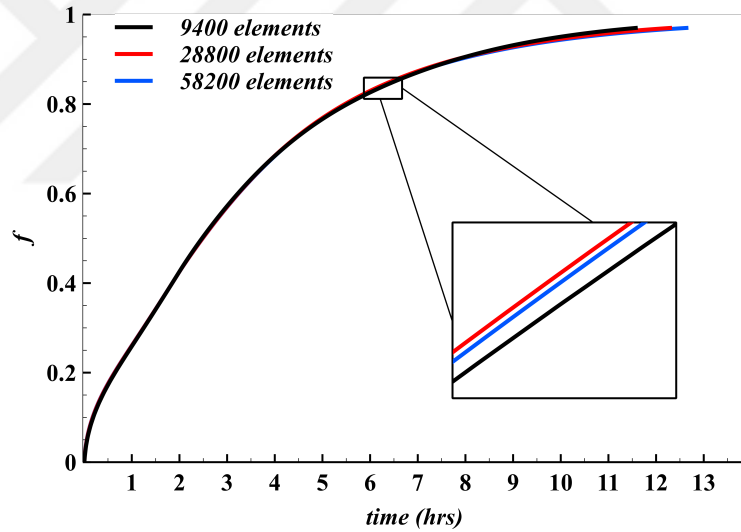


Figure 2.18: Liquid fraction for different element numbers of the unfinned geometry

Even though the behavior over time for all the meshes is similar, the total melting time for the mesh with element numbers 9400 and 28800 is 8.4% and 2.7% shorter when compared to the finest mesh.

The reason behind the total melting time difference lies in the fact that different mesh numbers in the cylindrical domain, lead to a slight change in the total PCM mass in

numerical analyses. Additionally, with larger elements, the model struggles to capture small vortices in the liquid PCM domain which results in different total melting times. Therefore, to save computational time, the mesh with a total element number of 28800 is chosen for further investigations.

Figure 2.19 depicts the selected mesh for the unfinned case. The PCM domain is meshed without a growth ratio because the melting front changes over time, while the mesh in the water domain was made finer in the vicinity of the no-slip boundary condition to capture the large temperature and velocity gradients.

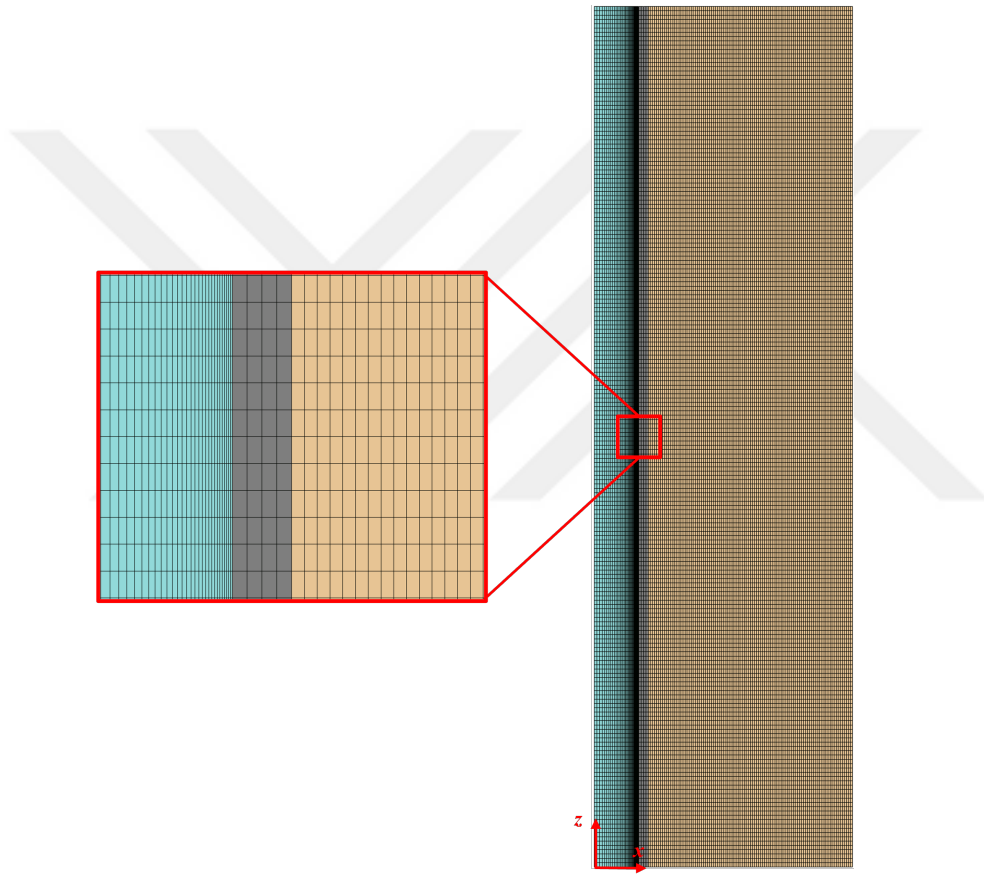


Figure 2.19: Chosen mesh for the unfinned geometry

2.4.2 Finned Geometry

The meshing configuration for the finned domain is of more importance because of several reasons:

- As the domain being studied is 3D, it is important to be mindful of the number of elements used in the numerical simulations.
- It is expected that the flow of liquid PCM in the domain will be more complex than in the domain with an unfinned geometry since fins can create local variations in the temperature field, which can affect the solid-liquid phase change process, potentially affecting the flow patterns.
- To accurately model the behavior of the phase change, it is desirable to use a conformal mesh at the periodic boundary conditions. A conformal mesh is a type of mesh that matches the shape of the boundaries without requiring any manipulations.

The meshing method is based on sweeping a 2D mesh on a surface in the z -direction, which ensures a conformal mesh between the two periodic boundary conditions. Figure 2.20 compares the liquid fraction behavior for three meshes used for the mesh independence study.

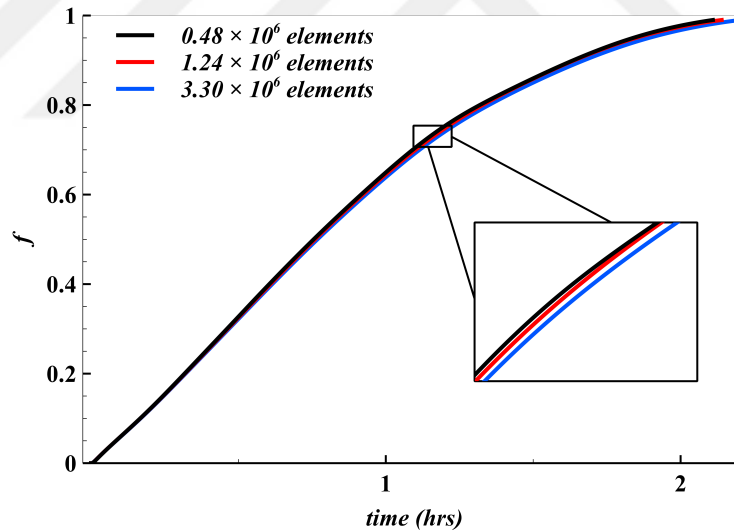


Figure 2.20: Liquid fraction for different element numbers of the finned geometry

With an increase in the element number, the liquid fraction evolution slightly slows, though the behavior between the different mesh elements is similar. The total melting time for mesh with element numbers of 0.48×10^6 and 1.24×10^6 is 4.17% and 2.8% faster when compared to the finest mesh. Therefore, to save computational time, the

mesh with an element number of 1.24×10^6 was chosen for further investigation. The chosen mesh is depicted in Figure 2.21.

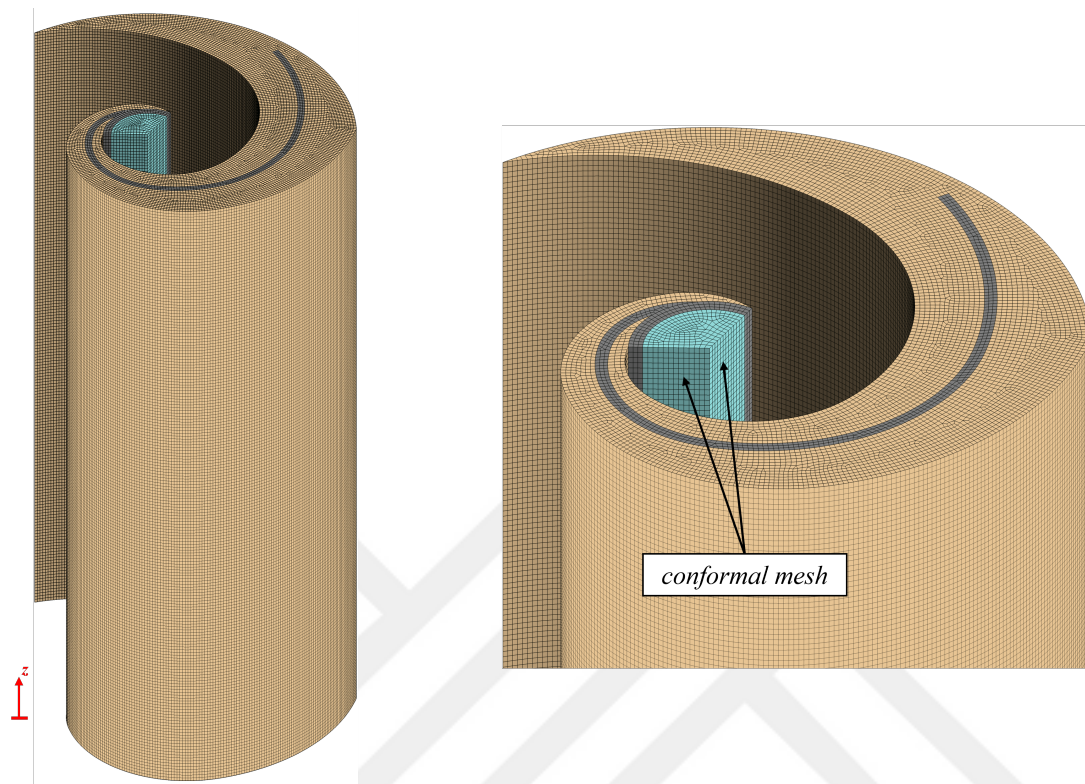


Figure 2.21: Chosen mesh for the finned geometry

CHAPTER 3

RESULTS AND DISCUSSION

3.1 Validation of the Numerical Models

The governing equations and relative boundary conditions using the methodology in the previous chapter are solved for the melting and solidification processes. In this section, the numerical results are compared to the experimental outcomes which are divided into two main subsections that separately discuss the unfinned and finned geometries. In each subsection, first the melting and then the solidification phase change processes are investigated.

3.1.1 Unfinned Geometry

Figure 3.1a characterizes the liquid fraction behavior for the unfinned geometry. It also compares the numerical results with different C_{mush} values to the experimental values. Even though the behavior is similar, the relative error between the numerical and experimental results is large. Upon investigating the experimental melting behavior, it was apparent that the melting volume at the top of the LTES tank was progressing slower than the numerical results, which indicates that the numerical model overestimates the effects of natural convection in the liquid region of PCM. To address this issue, the value of the volumetric thermal expansion coefficient (β), which controls the magnitude of natural convection within the liquid PCM, is lowered for the unfinned geometry to a value that is within the reported values in the literature ($10^{-4} K^{-1}$).

Figure 3.1b presents the comparison between the experimental and numerical results

of the unfinned geometry with the adjusted β value of 10^{-4} K^{-1} . For C_{mush} values of 10^5 , 10^6 , and 10^7 , the time-averaged relative error between the numerical and experimental results are 15.14%, 6.07%, and 2.95%, respectively. It is evident that by adjusting the volumetric thermal expansion coefficient, the liquid fraction behavior of the numerical study matches the experimental work.

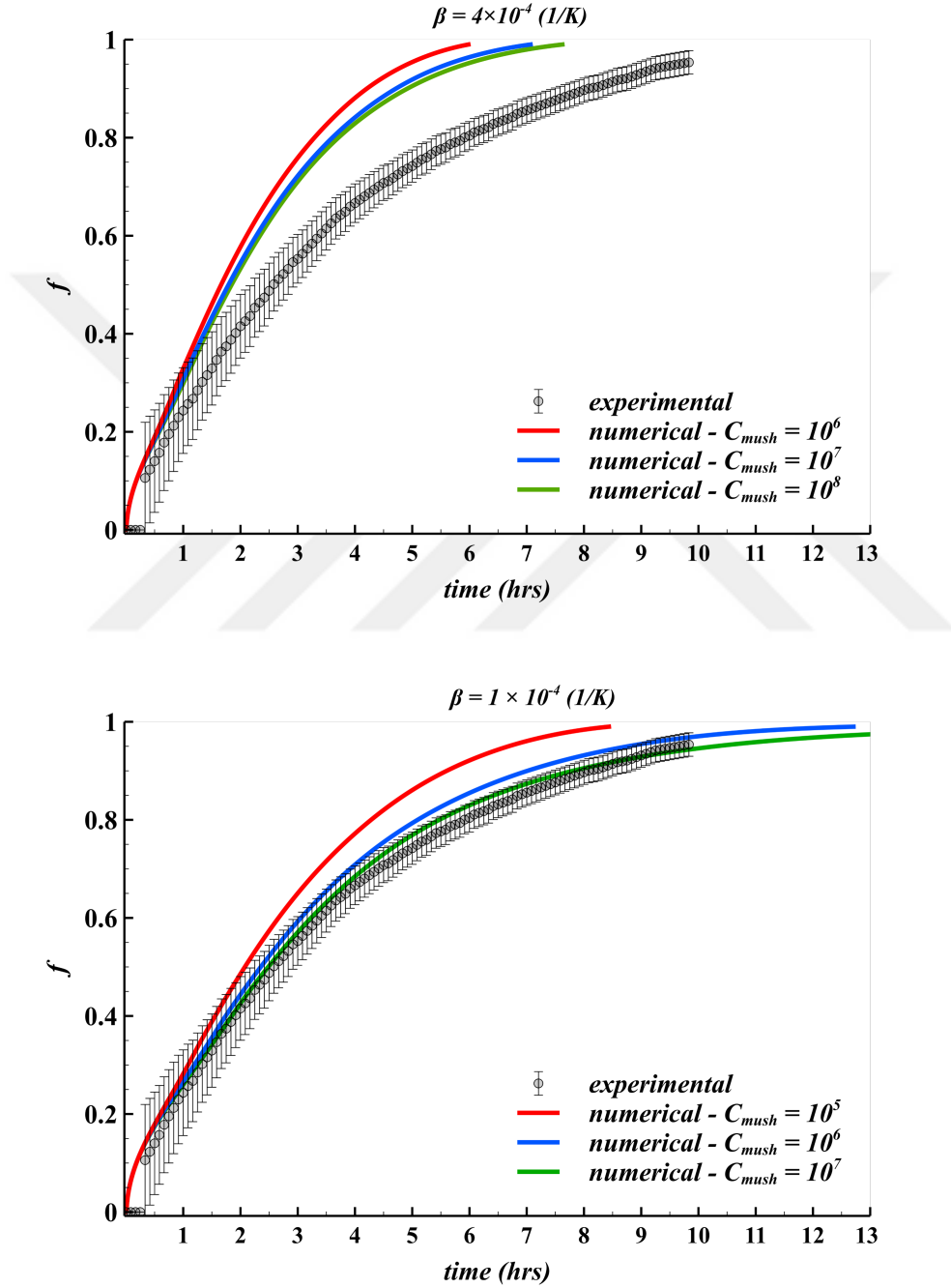


Figure 3.1: Liquid fraction in melting for the unfinned geometry

One possible reason for this difference is that the unfinned case takes a long time to melt, so, at any given moment, there is less amount of liquid PCM in the LTES tank compared to a design that melts more quickly, such as the finned design. As a result, the total thermal expansion coefficient (solid + fluid) is smaller for the unfinned case. When a PCM is melting, it will expand as it absorbs latent heat. This expansion may be pronounced depending on the heating rate, with higher heating rates tending to result in more rapid expansion, and researchers have reported that the thermal expansion coefficient of paraffin, varies depending on the heating rate [119].

During the experiments, the interface between the melted and solid PCM on the inner surface of the acrylic shell was apparent. To compare the melting morphology of the numerical model and the experimentation, the height of the solid column in the LTES tank is measured at 270 and 430 mins after the melting and depicted in Figure 3.2.

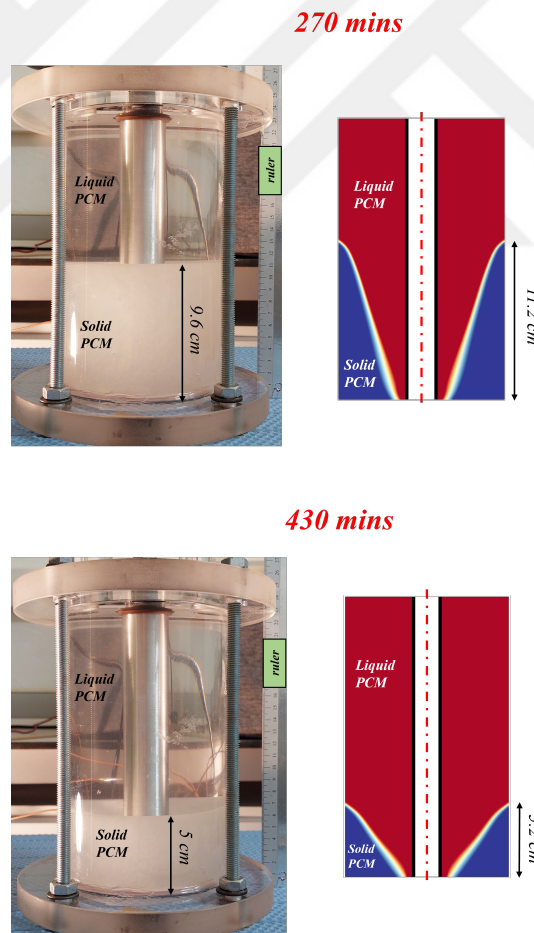


Figure 3.2: Exp. and num. PCM melting morphology of unfinned geometry

Based on Figure 3.2, it can be established that the 2D axisymmetry assumption is valid for the unfinned geometry and the melting morphology behavior can be captured accurately with the presented numerical model. This suggests that the numerical model with its simplified geometry is able to accurately predict the melting behavior of the unfinned geometry.

The small difference between the solid columns in the experimental and numerical results is possibly due to the enthalpy porosity methodology and the assumptions based on it that effects the morphology of PCM, possible experimental artifacts, and uncertainties in the thermophysical properties of PCM.

Figure 3.3 compares the experimental and numerical results for the solidification of the unfinned geometry. C_{mush} values from 10^4 up to 10^7 are modeled, and their results indicate that during the solidification of unfinned geometry, C_{mush} does not play a significant role. The time-averaged error for all the numerical simulations is about 25% with the numerical model solidifying faster than the experimental results.

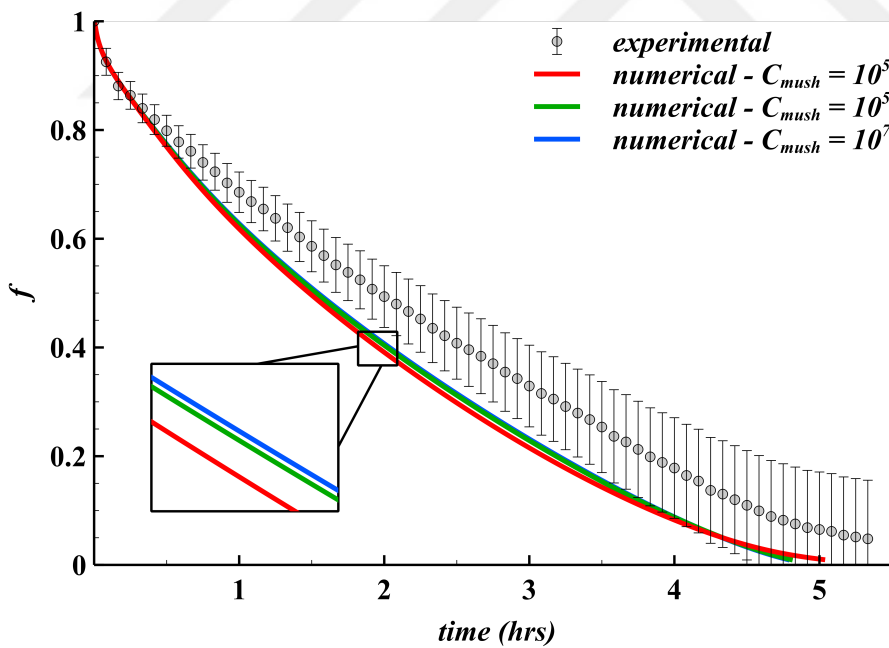


Figure 3.3: liquid fraction in solidification for the unfinned geometry

One likely explanation for this behavior is that during the experiments, the PCM in-

side the expansion tubes was blasted with hot air to prevent it from solidifying, which inevitably added thermal energy to the system, resulting in a longer solidification time.

The adjustment motioned in Section 2.2.5 is applied to account for the supplied heat during solidification. Figure 3.4 compares the adjusted liquid fraction evolution with the numerical results ($C_{mush} = 10^7$). The numerical results are closer to the adjusted results and fall within the error bars with an average error of around 7%.

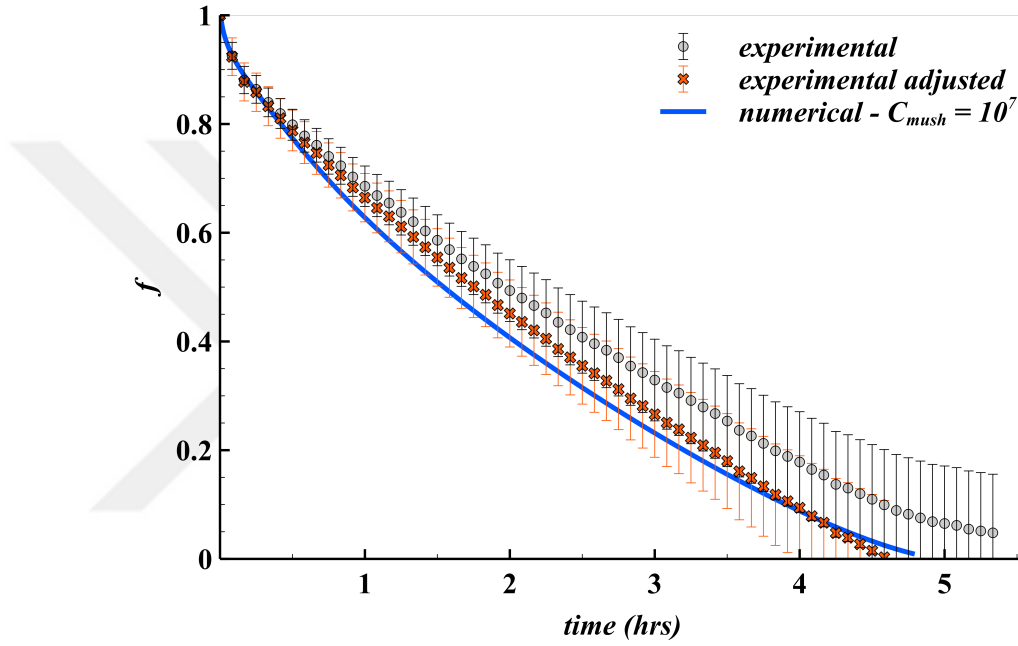


Figure 3.4: Adjusted exp. liquid fraction in solidification for unfinned geometry

3.1.2 Finned Geometry

Figure 3.5a compares the liquid fraction progression for the melting phase change between the numerical and experimental results for different C_{mush} values. For the C_{mush} value of 10^4 the time-averaged relative error is 3.92% while this value is 5.93% and 11% for C_{mush} values of 10^5 and 10^6 respectively.

The C_{mush} value of 10^4 captures the liquid fraction behavior for the melting until the latest stages of melting when it diverges. With the value of 10^5 , this divergence is

resolved, but until the latest stage of melting, it underestimates the liquid fraction. However, the errors with the C_{mush} value of 10^4 are smaller throughout the melting process.

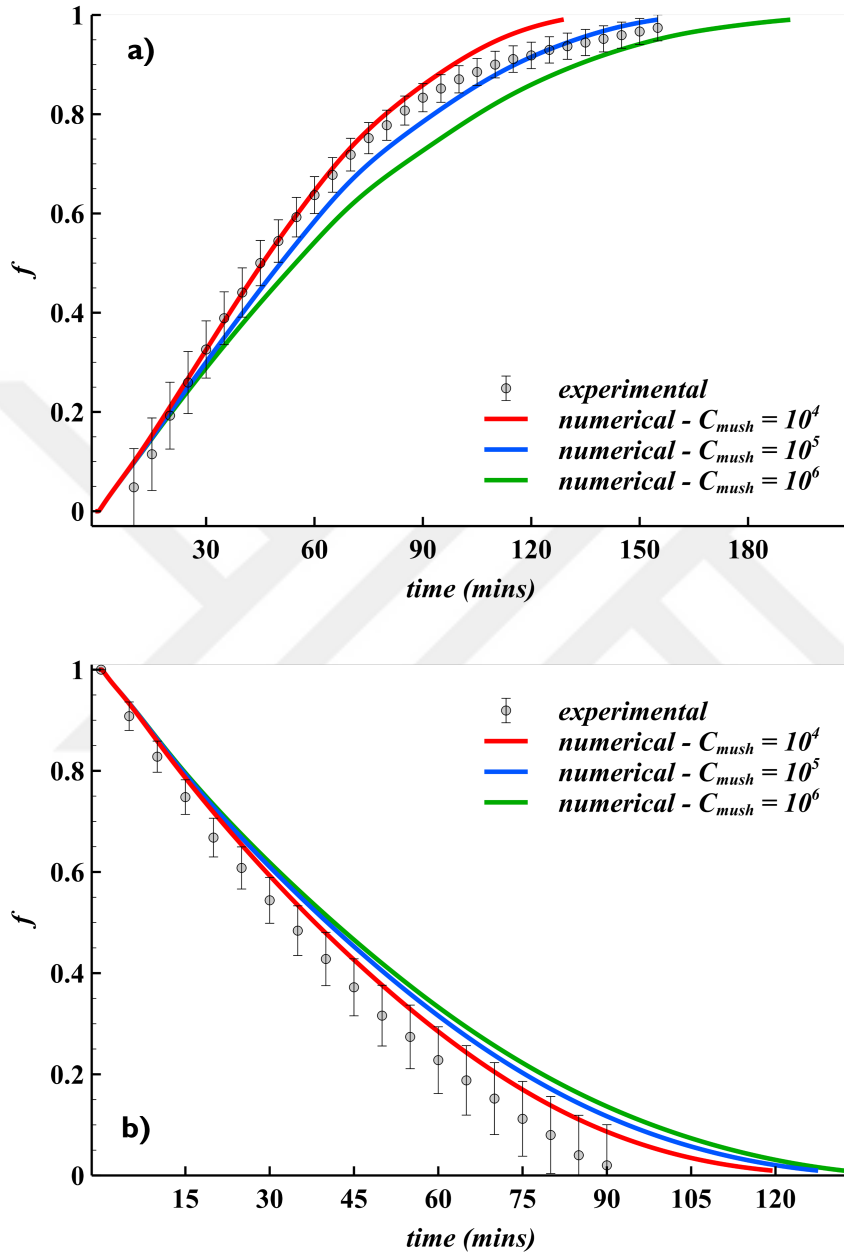


Figure 3.5: Liquid fraction for finned geometry: **a)** melting and **b)** solidification

As the mushy zone is increased, the melting fraction progression gets faster since the smaller C_{mush} allows larger velocity values on the melting front, thus increasing the

natural convection effects [120]. Figure 3.6 shows the liquid fraction on the xy -plane at $z/H = 0.5$ for different C_{mush} values after one hour of melting. It is evident that by increasing the C_{mush} value, the melting slows down and the progression of liquid fraction into the solid PCM domain gets smaller. This behavior is more prominent in the regions between the fins.

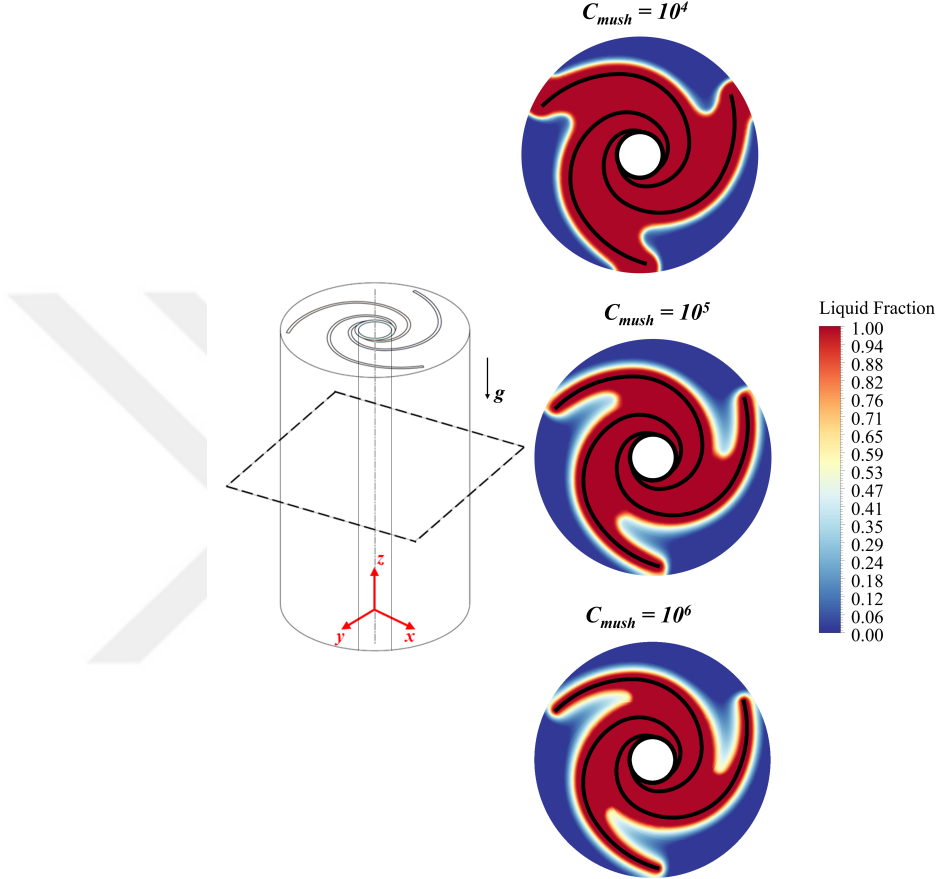


Figure 3.6: Liquid fraction in melting on $z/H = 0.5$ plane for different C_{mush} values, time = 1 hour

Figure 3.5b compares the liquid fraction progression for the solidification phase change between the numerical and experimental results for different C_{mush} values. The difference between the curves with different C_{mush} values is small and gets more pronounced towards the end of the solidification. For the C_{mush} value of 10^4 the time-averaged relative error is 15.2% while this value is 21.23% and 24.36% for C_{mush} values of 10^5 and 10^6 respectively. It is important to note that as the liquid fraction decreases, the uncertainty increases, which is the reason behind the relatively large

errors. In fact, at almost all the experimental data points, the numerical model with C_{mush} value of 10^4 falls within the uncertainty limits of the experimental data.

With an increase in the C_{mush} , the solidification time increases, which has been reported in the literature [121] as well. Figure 3.7 presents the liquid fraction on the xy -plane at $z/H = 0.5$ for different C_{mush} values after one hour of solidification.

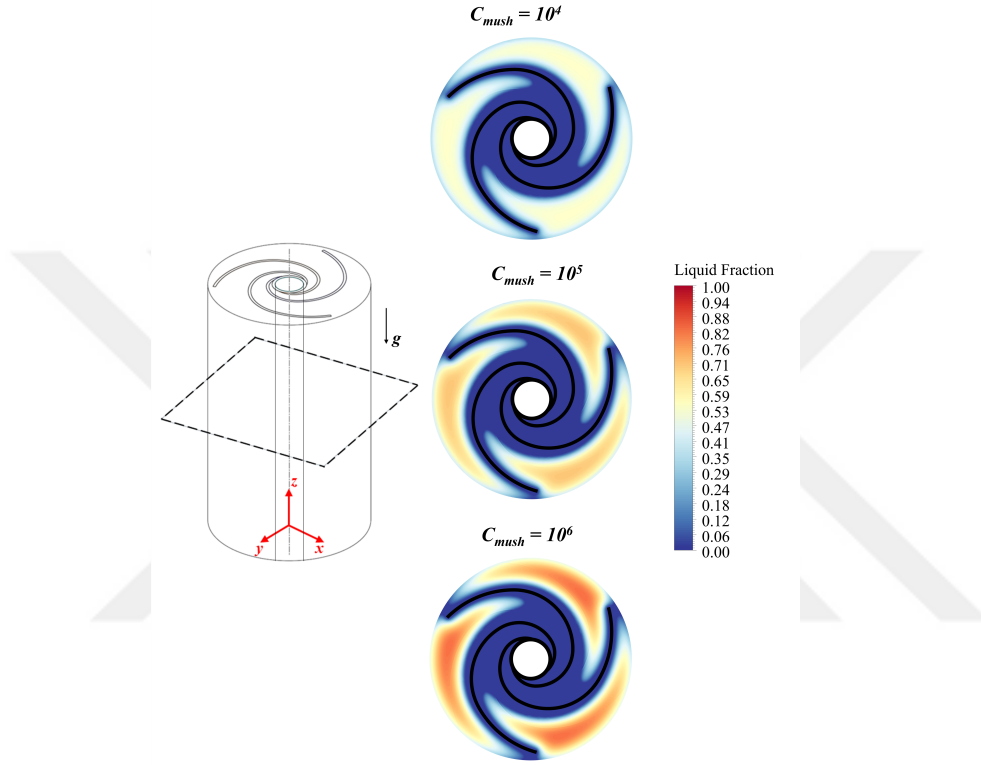


Figure 3.7: Liquid fraction in solidification on $z/H = 0.5$ plane for different C_{mush} values, time = 1 hour

By increasing the value of C_{mush} , the rate of solidification slows down. This is reflected in the observation of higher liquid fractions in the mushy zone for higher values of C_{mush} . However, the location of the solidified PCM does not change with changes in C_{mush} . This suggests that conduction plays a dominant role in the solidification process, as the liquid fraction changes in response to changes in C_{mush} , rather than the location, of the solidified PCM. This can have important implications for the design and optimization of systems involving PCMs, as the rate of solidification can significantly impact the overall performance of the system. It is therefore critical to

carefully consider the effects of variables such as C_{mush} on the solidification process in order to optimize the performance of PCM-based systems.

The numerical results not only can predict the liquid fraction evolution for melting and solidification but also can predict the phase change front accurately during the phase change process. The phase change front prediction is important since if the liquid fraction is the only comparison factor between the experimental and numerical results, the numerical results might predict the liquid fraction accurately while the PCM morphology is different. This issue is one of the main problems faced in the numerical results presented in the literature that use the enthalpy-porosity method. Since the caps are made of transparent acrylic, the phase change front can easily be observed.

Figure 3.8 compares the melting front between the experimental and numerical results after 20 minutes of melting, and Figure 3.9 depicts the solidification front after 25 minutes of solidification at $z/H = 1$ plane. The results of the comparison show that the numerical models accurately reproduce the phase change fronts, indicating that they are reliable representations of the experimental investigations and can be used with confidence for further study.

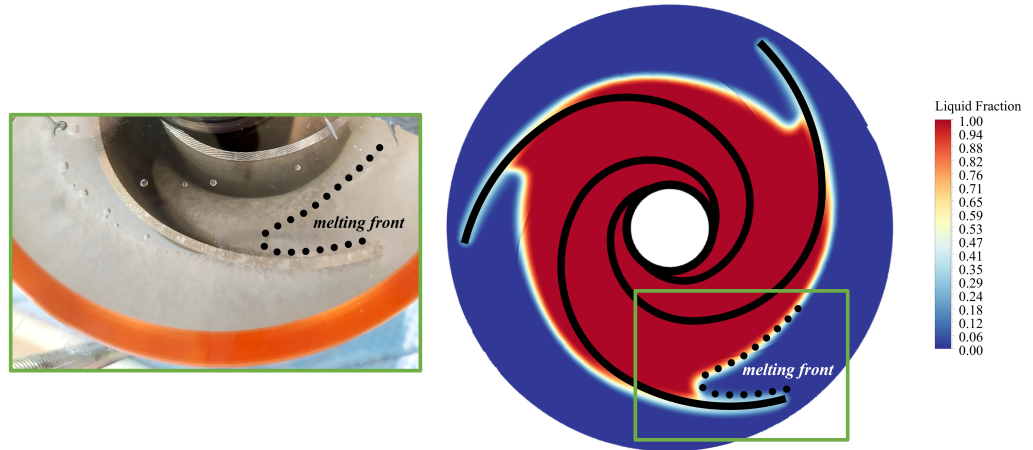


Figure 3.8: Melting front for experimental and numerical analyses on $z/H = 1$ plane, time = 20 mins

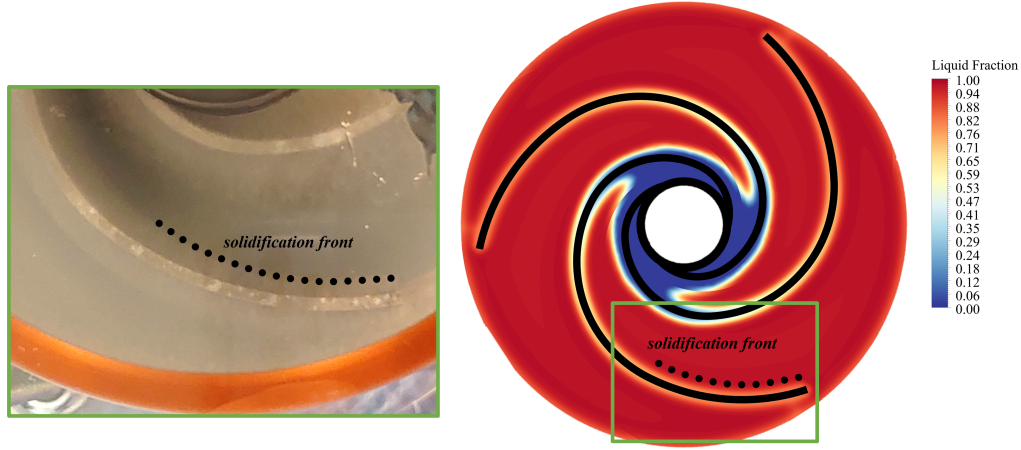


Figure 3.9: Solidification front for experimental and numerical analyses $z/H = 1$ plane, time = 25 mins

It is worth noting that the adjustment process for the solidification phase change of the finned geometry is not performed here, since the solidification process was short and the heat gun was used for very brief times and periods.

3.2 Discussion on Phase Change Characterization

Some modifications are made to the validated numerical models to facilitate their comparison as listed below, and, other than the mentioned items, all the other boundary conditions are kept the same.

- All the outer surfaces of the numerical domains are insulated in order to eliminate the effects of heat dissipation to the environment.
- For all the cases, the temperature initial conditions are the same: for melting, 28.2°C and for solidification, 46°C .
- The HTF mass flow rate is kept constant for all the cases.
- The C_{mush} values of 10^4 and 10^7 are used for the finned and unfinned geometries, respectively.

Additionally, an LTES tank with longitudinal fins (as depicted in Figure 1.5) is numerically modeled with the same number of fins for comparing the conventional design to the proposed Fibonacci-inspired fins. The total mass of the PCM is kept the same as the case with Fibonacci-inspired fins by increasing the thickness of the fins.

The following sections are divided into melting and solidification categories, with the physics of phase change in the unfinned geometry being discussed first. This is followed by a comparison between conventional longitudinal fins and the proposed Fibonacci-inspired fins. The comparison is based on the liquid fraction, temperature, and melt front velocities. The goal of this comparison is to evaluate the performance of the different fin designs and understand how they affect the phase change process.

3.2.1 Melting

Figure 3.10 depicts the velocity vector-embedded contours, temperature, and liquid fraction contours for the unfinned geometry as the melting progresses. After 0.5 hours of melting, a small layer of melted PCM is present in the vicinity of the HTF tube which is mostly uniform in the z -direction except for the uppermost section of the PCM domain where the natural convective flows heat up the solid PCM and create a melted region with a thickness larger than that of other parts of the domain. Within the melted thickness, a circulation of liquid PCM is present with velocities of around 0.25 mm/s . The liquid PCM in the vicinity of the HTF tube is warmer than the liquid PCM in the vicinity of the solid PCM, which is the reason behind this circulation.

As the melting continues, The melted thickness gets larger in the x -direction, which is due to conduction heat transfer. At time = 1 hour, the region carved by natural convective flows grows, and its temperature increases as the velocities in the vicinity of the melt front increase, which is an indication of the onset of natural-convective-dominant melting. When the melting reaches the 2 hours mark, the velocity in the vicinity of the melting front is the highest, which results in complete domination of natural convective-driven melting.

The high-velocity flows on the melting front continue to grow until the 6th hour of melting when the velocities on the melting front start to decrease due to lower tem-

perature gradients within the PCM. As the melting gets to its ending process, the velocities on the melting front start to diminish, and once again, conduction becomes the dominant mode of heat transfer.

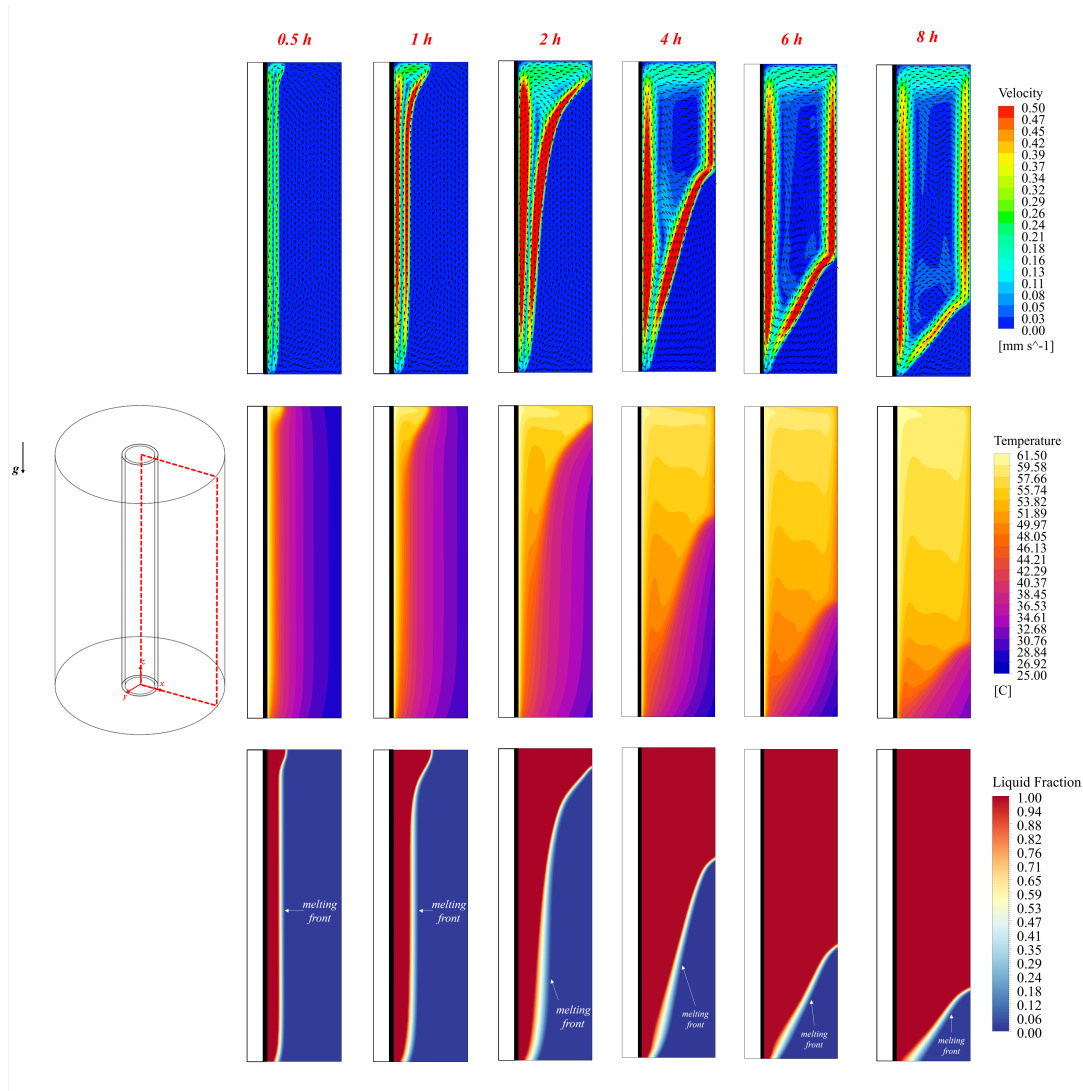


Figure 3.10: Velocity, temperature, and liquid fraction behaviors of the unfinned LTES during melting

Figure 3.11 shows the evolution of liquid fraction and the ratio of the average velocity of liquid PCM at the melting front to the maximum velocity of liquid PCM on the melting front (taken as an isoline where $f = 0.95$) for the unfinned geometry. The melting process is divided into three main stages:

- The strong conduction dominance stage at the beginning and ending of melting where the velocities on the melting front are small.
- The weak convection dominance stage when the velocities of the melting front start to grow.
- The strong convection dominance stage when the velocities on the melting front are the largest.

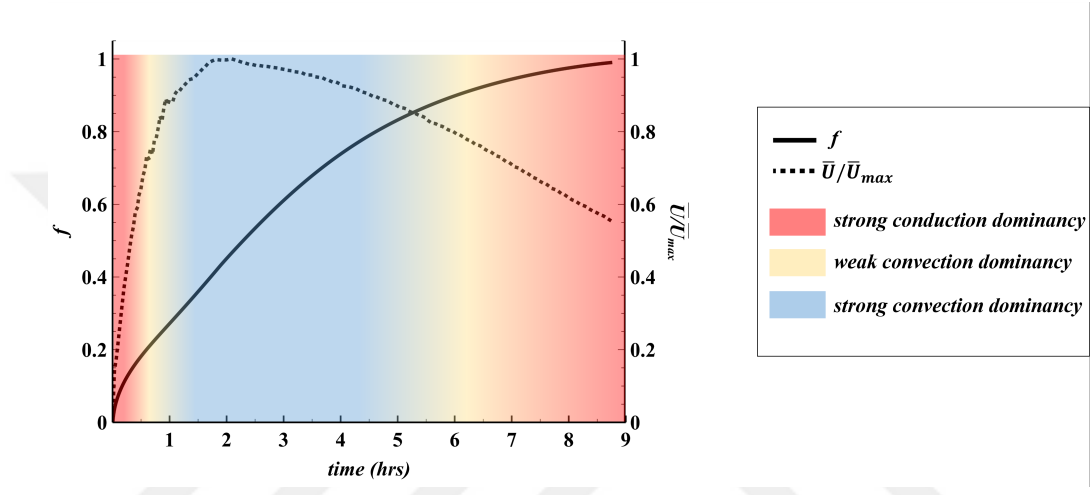


Figure 3.11: The evolution of liquid fraction and average melting front velocity for the unfinned LTES

It is important to note that even though there are circulations within the liquid PCM when the melting is dominant by conduction, these circulations happen due to temperature gradients within the liquid PCM, especially between the HTF tube and the insulated walls of the domain. The total melting time in the unfinned geometry is around 9 hours, with about 38% of it within the strong convection dominant region. The heat transfer towards the x -direction is inefficient because of the low thermal conductivity of PCM and the small role of natural convection.

Figure 3.12 presents the liquid fraction contours for the Fibonacci-inspired and longitudinal fins on the xy -plane at different z/H values of 0.1, 0.5, and 0.9 which respectively represent the bottom, middle, and top sections of the LTES tank. The evolution of liquid fraction is depicted for 20, 40, and 80 minutes after the melting.

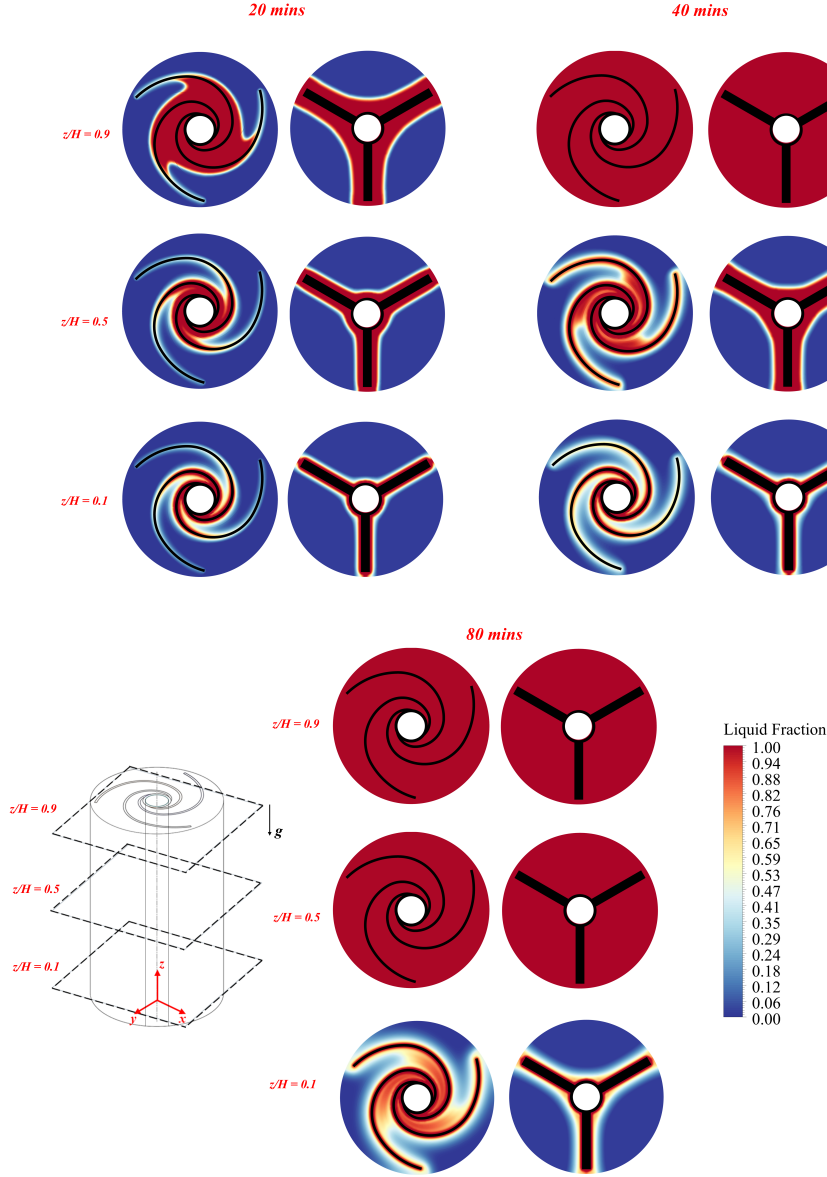


Figure 3.12: The evolution of liquid fraction in fin-integrated LTESs

The top of the LTES tank, $z/H = 0.9$, melts the fastest in both geometries due to the effects of natural convection while its bottom, $z/H = 0.1$, melts the slowest since that region is mostly conduction-dominant melting. The results show a large gradient of liquid fraction in the HTF direction, indicating that the melting process is actually three-dimensional. This is in contrast to many studies on nature-inspired fins, which often consider only a 2D domain that may not accurately capture the behavior of PCM during melting.

The key difference between the proposed fin and the conventional design arises from the fact that the proposed design covers more area of the solid PCM with its curved structure, the effect of which is eventuated at the bottom of the LTES tank where the longitudinal fin struggles to melt the solid PCM. Towards the end of the melting, on the plane where $z/H = 0.1$, the surface-averaged liquid fraction value for the longitudinal fins is 0.16 while the Fibonacci-inspired fins have a value of 0.42 which indicates the effectiveness of the design in melting the regions where the conventional design experiences major problems. Figure 3.13 presents the 3D temperature field and velocity contours in the vicinity of the fins for both geometries.

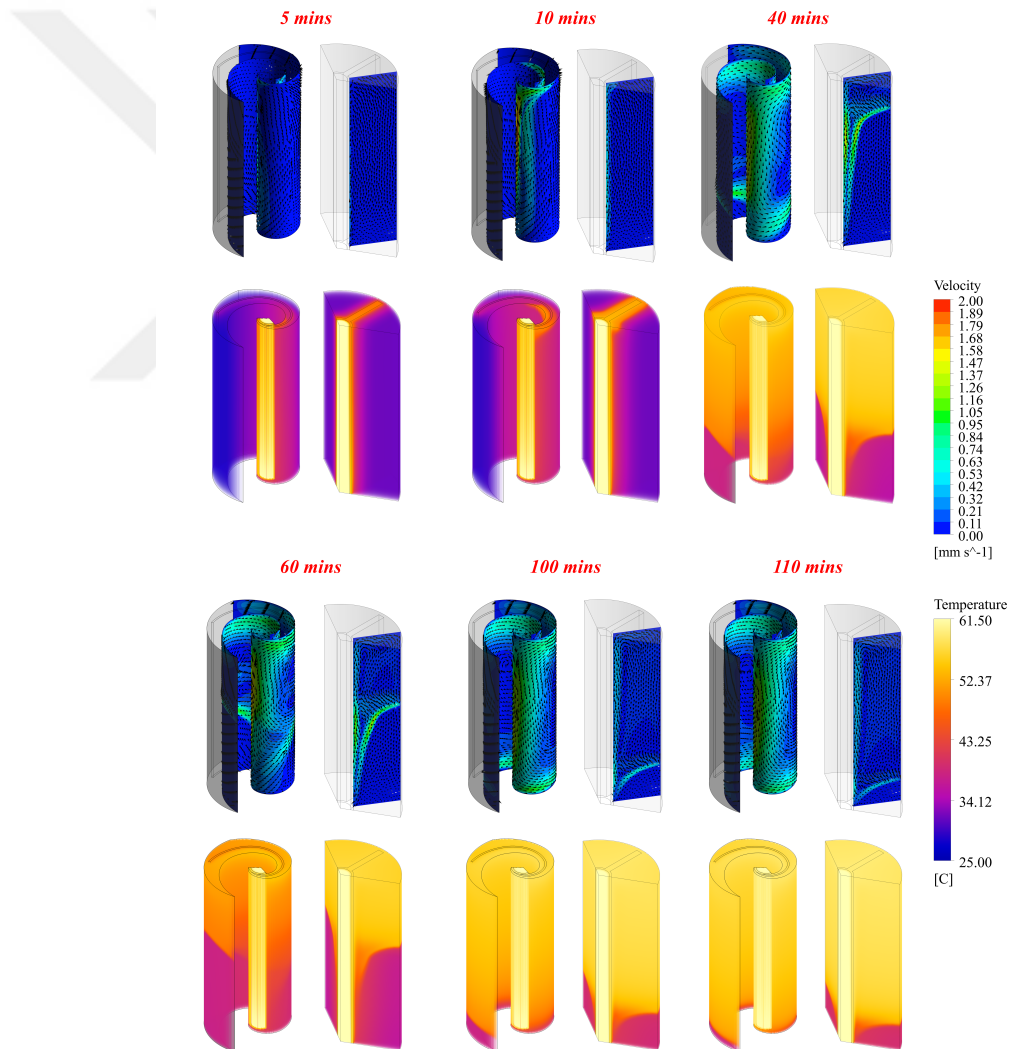


Figure 3.13: The evolution of velocity and temperature for fin-integrated LTESs

The temperature uniformity within the PCM domain is improved compared to the longitudinal fins, resulting from flow mixing due to curved structures in the Fibonacci-inspired design. The liquid PCM flow is accelerated towards the z -direction in both geometries due to bouncy effects though, in the proposed design, the flow velocity is larger near the HTF tube with a concentration of high-velocity flows on the upper half of the LTES tank. The results establish that the convective melting process is more evenly distributed throughout the PCM domain in the Fibonacci-inspired fins, due to the circulation of the liquid PCM in directions other than the main flow direction (upwards towards the top). In contrast, in the longitudinal fin design, the main liquid PCM circulation is observed from the HTF tube wall to the side of the LTES tank. At time = 40 minutes, the temperature uniformity is particularly noteworthy, with the longitudinal fin showing a large temperature gradient in the z -direction while the Fibonacci-inspired design exhibits a smaller temperature gradient in all directions. This suggests that the Fibonacci-inspired design is more effective at promoting temperature uniformity within the PCM domain compared to the longitudinal fin design.

To observe the effect of flow mixing on the melting front more closely, Figure 3.14 compares the velocity contours on the melting front (an isosurface where the liquid fraction is 0.95). Normalized velocity vectors are also augmented on the velocity contours to show the direction of liquid PCM flow.

During the melting process, the velocities on the melting front are more dispersed in the domain containing the PCM for the Fibonacci-inspired fin design, while the melting front is concentrated near the straight fin. This suggests that the Fibonacci-inspired design is more effective at promoting more evenly distributed flow and heat transfer within the PCM, leading to more efficient melting. In contrast, the straight fin design leads to concentrated and localized heat transfer, which results in less efficient melting.

The distribution of high-velocity liquid PCM is particularly important for improving the melting where the distance between the heating source and the solid PCM is the farthest. In these locations, the conventional fin struggles to facilitate heat transfer while the proposed design has the advantage of dispersing the convective heat transfer effects albeit with lower velocities.

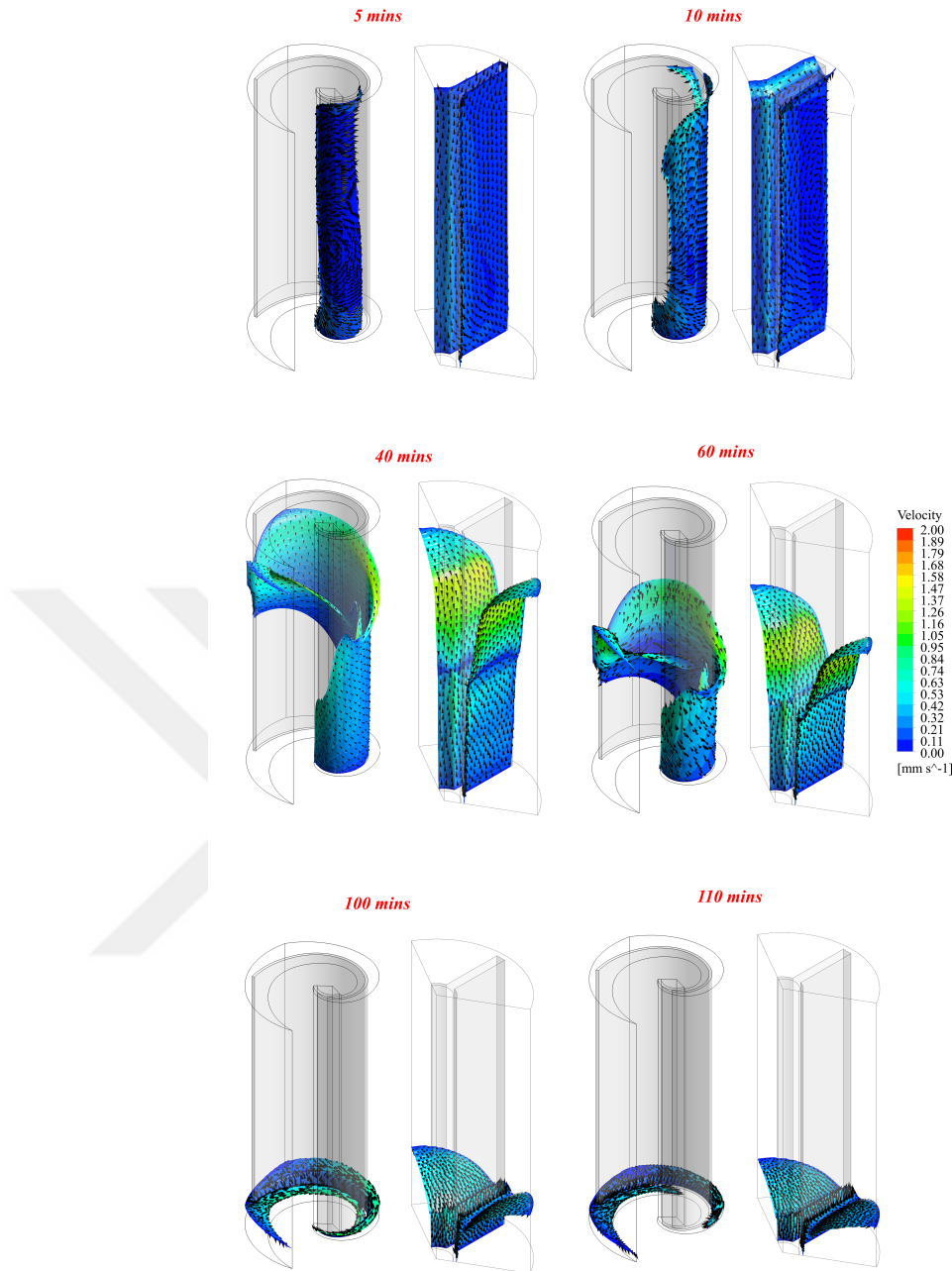


Figure 3.14: Velocity contours on the melting front for the fin-integrated LTESs

Secondary flows refer to the flows of fluid that are not in the same direction as the main flow. In some cases, secondary flows can be beneficial, such as when they help to mix or transport materials within a fluid. These flows can be caused by various factors, such as the shape of the container through which the fluid is flowing. There exist secondary flows in the Fibonacci-inspired fins which are perpendicular to the

z -direction, while the main liquid PCM flows in the z -direction in the longitudinal fins. The presence of secondary flows in the Fibonacci-inspired design enhances the mixing and heat transfer within the PCM, leading to more efficient melting. In contrast, the unidirectional flow in the longitudinal fin design leads to less mixing and less efficient heat transfer. Figure 3.15a and 3.15b present the liquid fraction and the ratio of average velocity on the melt front to the maximum velocity on the melt front of the Fibonacci-inspired and longitudinal fins.

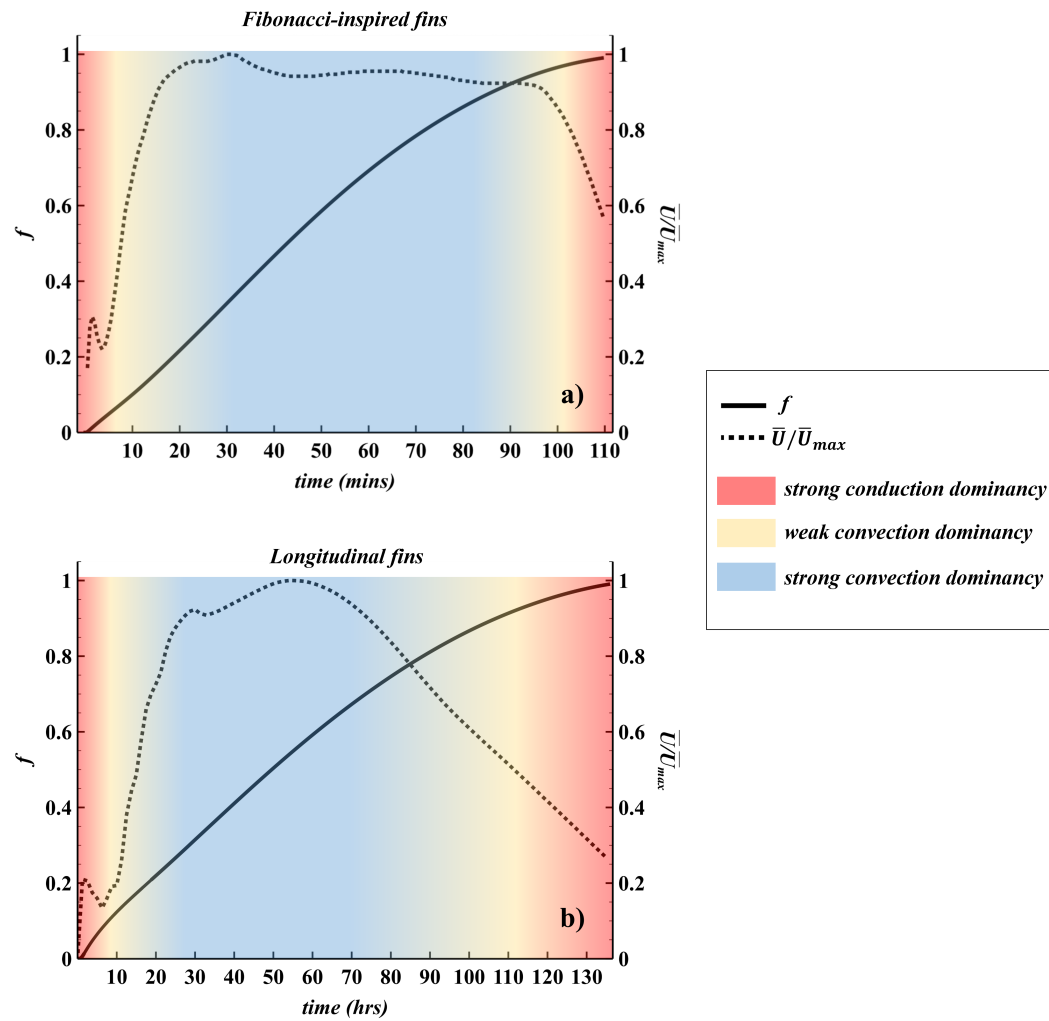


Figure 3.15: The evolution of liquid fraction and average melting front velocity for the **a)** Fibonacci-inspired and **b)** longitudinal fins

The melting process ends 20% faster in the Fibonacci-inspired fins when compared to the longitudinal fins. Even though the velocities in the longitudinal fin LTES are larger

in magnitude, its storing convection dominant region is shorter: 63% versus 50%. The velocity ratio stays above 80% for a longer time in the proposed design in contrast to the conventional design where a sharp decrease in the velocities is observed after 90 minutes of melting which results in to weak convection dominance and eventually longer conduction dominancy. This dominance is due to the fact that the proposed geometry, redirects the flow in the x and y -direction as depicted in Figure 3.14 which results in a more solid PCM area to be in contact with the convective flows through the intensity of these flows are higher in the longitudinal fin LTES due to less wall shear stress resulting from lower heat transfer area. This provides more space for the bouncy-driven flows to expand resulting in higher velocities.

3.2.2 Solidification

Figure 3.16 represents the velocity, temperature, and liquid fraction contours of the unfinned case after half, one, two, and six hours of solidification. The solidification front at the beginning of the phase change process is thicker at the bottom of the domain since it is closer to the cold inlet. At the first half hour of melting, low magnitude velocities exist in the PCM domain which quickly diminish since as the PCM solidifies, it loses its mobility. The solidification fronts develop in a near one-dimensional manner towards the x -direction, which indicates that the solidification process is mostly dominated by conduction heat transfer.

The combination of a lack of convective heat transfer, a low thermal conductivity value of the PCM, and an increasing distance between the solidification front and the cold surface (HTF tube) result in progressively larger thermal resistances between the solidification front and the cold surface as time passes. This is due to the fact that these factors work together to reduce the overall rate of heat transfer from the HTF tube to the PCM, leading to a slower solidification process.

The solidification process in the unfinned geometry is mostly one-dimensional while in the melting, due to the presence of natural convection, the phase change process is two-dimensional. However, since the thermal conductivity of RT42 in the solid state is 60% larger than in the liquid state, the melting happens nearly at the same speed as solidification. This behavior results in a symmetrical melting/solidification phase

change for the unfinned case which is not desired since a symmetrical phase change process indicates large temperature gradients and low level of liquid PCM mixing.

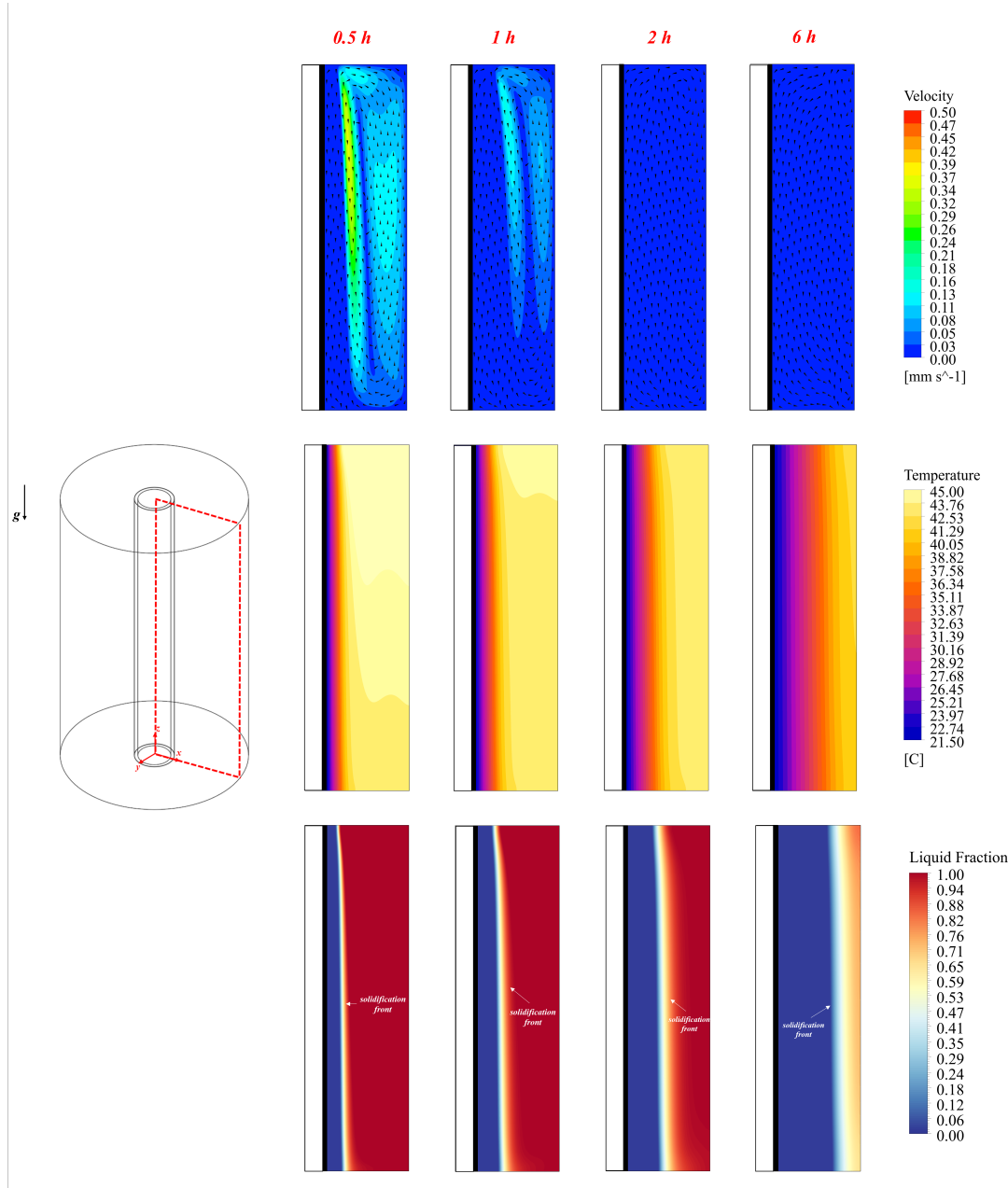


Figure 3.16: Velocity, temperature, and liquid fraction behaviors of the unfinned geometry during solidification

Figure 3.17 compares the liquid fraction evolution on the xy -planes at different z/H values for the proposed Fibonacci-inspired and conventional longitudinal fins, while Figure 3.18 compares their temperature and velocity behaviors.

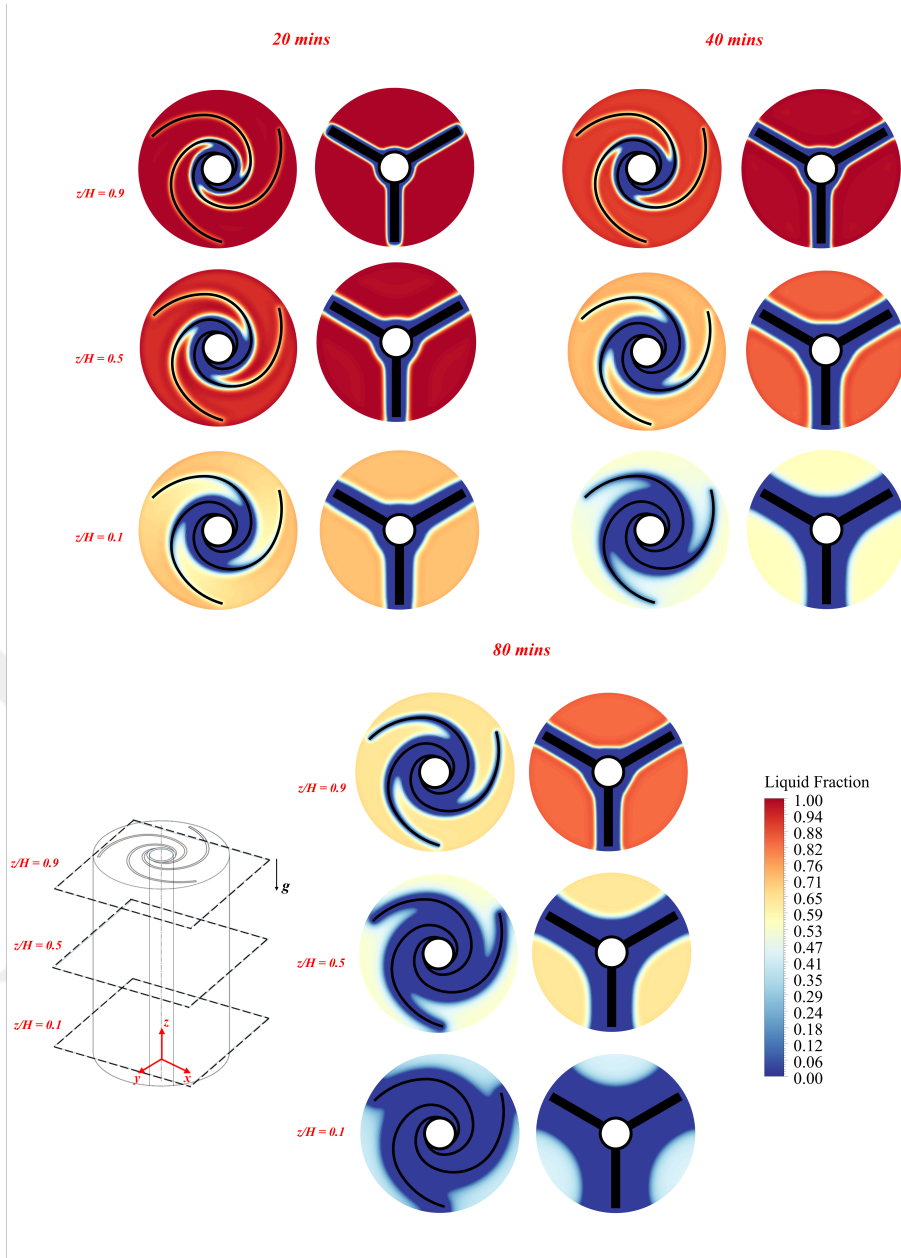


Figure 3.17: Liquid fraction behaviors in solidification for fin-integrated LTESs

The clear advantage of the proposed fins in the solidification process is apparent due to several observations as listed below:

- The velocity magnitudes for the proposed design are larger compared to the conventional design which indicates that in the proposed design, the effects of natural convection are more significant, especially in the beginning stages of

the solidification process. This effect can be clearly observed by comparing the liquid fraction contours on the $z/H = 0.9$ plane at time = 20 minutes.

- The fins in the proposed design extend further into areas where the PCM solidification process is more challenging, improving the overall heat transfer performance of the LTES tank. This is because the fins can help to promote heat transfer from the HTF to the PCM, reducing the thermal resistance between these two regions and speeding up the solidification process. The liquid fraction at time = 80 minutes on the $z/H = 0.1$ plane emphasizes this difference.
- The temperature uniformity in the proposed design is superior, which is more pronounced towards the end of the discharge process.

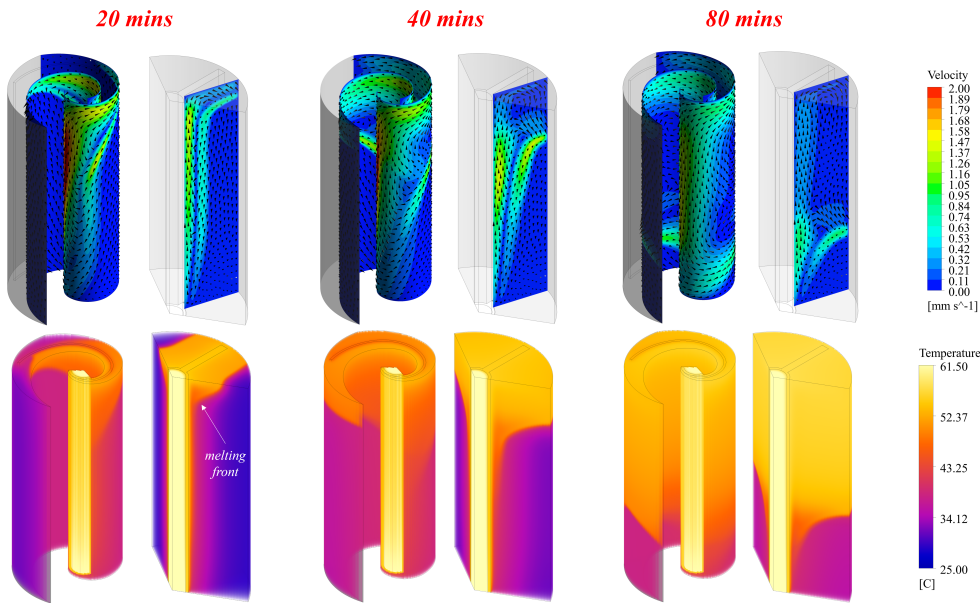


Figure 3.18: Velocity and temp. behavior in solidification of the unfinned LTESs

Compared to the unfinned geometry, the temperature gradient in the z -direction is larger, which is attributed to the larger heat transfer area in the finned case resulting in faster heat dissipation at the bottom of the LTES tank. Figure 3.19 compares the liquid fraction of the unfinned, Fibonacci-inspired, and longitudinal geometries. The proposed geometry solidifies 65% faster compared to the unfinned geometry and 42% faster compared to the longitudinal design. The longitudinal fins struggle at the last

two hours of its solidification because they do not reach the farthest locations from the HTF tube which create a large distance between the cold surface and liquid PCM resulting in large conductive thermal resistances.

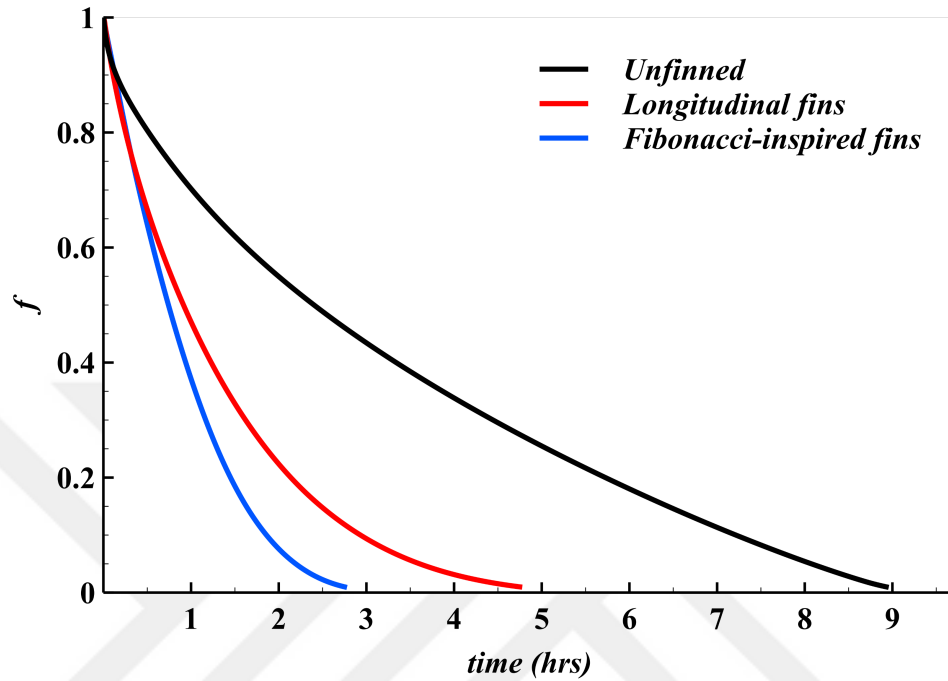


Figure 3.19: Liquid fraction evolution of the finned and unfinned geometries during solidification



CHAPTER 4

CONCLUSION AND FUTURE WORK

LTES is considered the future of thermal energy storage due to its reliability and high storage density. However, due to its low thermal conductivity, the charging/discharging time can exceed the requirements of energy storage using renewable energy sources such as solar energy. To resolve this issue, researchers have proposed solutions such as using extended surfaces (fins), adding particles with high thermal conductivity to the PCM, and using porous material. Among the viable means of decreasing the charge/discharge time, adding fins is the most applicable solution since it is easy to implement and does not alter the properties of PCM.

Various conventional fin designs have been studied fins such as radial and longitudinal fins. However, they usually deliver a large temperature gradient inside the PCM domain that leaves unmelted/melted regions of PCM during charging/discharging, prolonging the phase change time. Novel fins inspired by nature that use fractal structures, such as trees, snowflakes, and spiderwebs, have shown promise for decreasing the melting and solidification times. Even so, the proposed designs have several major issues: they usually suppress the effects of natural convection during the melting process, the solidification is often overlooked, there is a lack of experimental evaluation for such designs and the literature is mainly investigating their performance using numerical tools which by themselves have critical flaws since most of the studies consider a 2D domain that cannot accurately model the phase change phenomenon in the LTES tank. Additionally, the nature-inspired fins are restricted mostly in terms of shape to tree-like fins with limited variety, which indicates an opportunity for exploring other novel geometries.

This study introduces a novel fractal fin geometry based on the Fibonacci sequence,

which is an integral part of many species, mainly flowers, and ferns. Fibonacci-inspired geometries have proven to have significant mixing performance, especially in low-velocity fluid flows, which could be useful in liquid PCM buoyancy-drive motion mixing that inherently possesses low velocities in the order of millimeters per second. Additionally, since the proposed fins do not contain the PCM in small volumes, unlike available fractal fins, it has the potential to maintain the natural convection flows without decreasing their effects on melting. To investigate the proposed design, an experimental setup was designed including an LTES unit with Fibonacci-inspired fins manufactured using a wire-cutting method out of aluminum.

Additionally, a base design with no fins was also manufactured to compare the difference to the suggested fins. The PCM of choice was RT42 and its density (in liquid and solid phases), thermal expansion coefficient, and thermal conductivity in the solid phase were measured in-house. The non-intrusive volume expansion method was utilized to directly estimate the liquid fraction evolution for both melting and solidification phase change processes, enabling the elimination of thermocouples inside the PCM that may cause interference with the melting/solidification. It is worth mentioning that this study is the first of its kind to evaluate the liquid fraction behavior for the solidification process in shell-and-tube-like geometries.

To study the effects of the proposed fin design more clearly, a 3D numerical model for each configuration was constructed using the experimental data obtained. The numerical model was based on the enthalpy-porosity method combined with the Boussinesq approximation to model the natural convective flows. The numerical model was validated successfully by comparing the liquid fraction graphs for melting and solidification processes. The effects of the mushy zone constant, which is a critical parameter for characterizing the phase change process, were studied extensively for both geometries.

To compare the Fibonacci-inspired fins to the conventional longitudinal fins, a numerical model by keeping the total mass of PCM constant was developed for the longitudinal fins. The main findings of the experimental and numerical works are listed below:

- In melting, natural convection plays a significant role. However, during solidifi-

cation, conduction is the dominant mode of heat transfer in all the geometries.

- The convection heat transfer was characterized by the average velocity of liquid PCM on the melting front. In the unfinned geometry, around 38% of melting was dominated by strong convective flows while for the proposed and longitudinal fins, this value was about 63% and 50% respectively.
- Even though the velocities on the melting front were lower in magnitude for the proposed fin design, they covered more surface area of the melt front compared to the longitudinal fins, thus having a longer effect in the melting process.
- Secondary flows due to the curvature of the Fibonacci-inspired fins were observed by investigating the velocity contours within the liquid PCM domain. These flows helped with heat transfer and resulted in a more uniform temperature in the PCM domain.
- Compared to the unfinned geometry, the melting duration was shorter by 80% while this value was 20% when compared to the conventional fins.
- Due to the fin structure, the melted pockets (especially at the bottom of the tank) during the solidification process are minimized compared to the conventional fins.
- The proposed geometry solidifies 65% faster compared to the unfinned geometry and 42% compared to the longitudinal fins.
- The decrease of liquid fraction for the proposed design is sharper and mostly linear, which is an indication of its fast response to the phase change.

Therefore, with the mentioned results, the proposed design presents a significant improvement in terms of thermal response both in melting and solidification processes which is backed by exhaustive experimental and numerical results. On the other hand, the main issue of the Fibonacci-inspired fins is their manufacturing challenges, where an advanced and costly method was used to produce such a geometry.

The future work, based on the experimental and numerical experiences, can be divided into three categories:

PCM thermophysical properties: The experimental work revealed that there are various discrepancies within the literature in terms of different properties of PCM. Some

of the properties were able to be measured though some, due to measurement limitations, could not be measured. An extensive study on the property measurement is needed especially, the values of dynamic viscosity, liquid PCM thermal conductivity, and specific heat capacity. A more accurate set of thermophysical properties could resolve the issues with the numerical simulations.

Experimental work: Several challenges were faced throughout the experiments, including keeping the PCM in the expansion tubes in the liquid phase during the solidification process and a large amount of heat dissipation to the environment. Additionally, the leakage problem of the liquid PCM was a major issue. In future work, a more comprehensively-designed experimental setup could be manufactured that includes a totally sealed LTES tank. The heat dissipation could be eliminated by insulating the LTES shell, though it does not allow visualizing the phase change process. Additionally, advanced image-capturing techniques such as using LIDAR could make the melt front visualization within the PCM a feasible deed.

Numerical analysis: The experimentally validated numerical models could be used to optimize the shape of the Fibonacci-inspired fins. The optimization parameters are suggested to be the number of fins, fin thickness, and the distance of the fin to the acrylic shell while keeping the total volume of the fin constant with the objective functions of shortest melting/solidification times. Since the experimental work needs a new fin design for each experiment, the experimentally-validated numerical analyses instead, can be a viable tool for the optimization process, and then the optimized geometry can be tested experimentally.

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