Design and modeling of a novel compact solar collector with phase change material

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DESIGN AND MODELING OF A NOVEL COMPACT SOLAR COLLECTOR WITH PHASE CHANGE MATERIAL

By

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A thesis submitted in partial fulfillment of the requirements for the degree of

Master of Science in Mechanical Engineering

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Abstract
The potential of integrating phase change materials (PCM) directly to solar collectors is proposed as a promising innovative modification for solar collectors in recent years. Accordingly, several compact PCM-collector designs are presented and compared to the conventional solar collector systems. In this study, a single vacuum tube is experimentally tested as a building block of the novel evacuated tube collector-storage solar water heater (ETCS). The free space of a single evacuated tube with a U-bend copper pipe is filled with Paraffin wax as a PCM, while water is used as heat transfer fluid (HTF). In addition, a longitudinal aluminum fin is used to enhance the heat transfer between the copper tube and the PCM. The performance of the proposed building blocks is evaluated under several charging and discharging experiments at different operating conditions. The tested tubes are evaluated in terms of cumulative recovered energy, energy storage/recovery efficiency, effectiveness, and charging-discharging cycle time.

The results show that natural convection is the dominating heat transfer mode for 65% of the charging time. For the finned tube, the results show that increasing the PCM initial temperature from 65°C to 80°C, increases the recovered energy and the total energy efficiency by 19% and 24%, respectively. For the un-finned tube, such an increase in PCM initial temperature increases the recovered energy and total energy efficiency by 5% and 9%, respectively. On the other hand, increasing the flow rate of the HTF from 0.3 l/min to 0.5 l/min decreases the effectiveness to 18% and 11% for the finned and un-finned tube, respectively. The heat loss test shows that the heat retention of the finned and un-finned tube after 12 hours test is almost similar at 34% and 32%, respectively. Further, the experimental results are used to validate a numerical model that is used to investigate the effect of tube design on thermal performance. Moreover, the effective heat capacity method is used to model the proposed building block, and the model is validated using both the experimental and numerical results. In addition, a simplified 2-D model is
developed and used for investigating tube performance with different design parameters. Finally, the simplified model is integrated with a design methodology that is developed to be used as a designing tool for novel ETCS systems. The methodology presented in this study is used to design a new generation of solar collectors that work as a stand-alone compact solar water heater for heat demand in the night.
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Nomenclatures

Variables

dp  Copper pipe diameter [mm]
Dg  Glass tube diameter [mm]
E   energy [J]
$H_{\text{L}}$  PCM latent heat of fusion [kJ/kg]
k   Thermal conductivity [W/mK]
L   tube length [mm]
m   mass [kg]
P   power [W]
t   time [min]
T   temperature [°C]
V   volume [m³]
$U_l$  heat loss coefficient
$\Delta T_{\text{max}}$  Maximum temperature [°C]

Subscript

a   ambient
ch  charge
dis discharge
final final temperature
initial initial temperature
max maximum
nom nominal
out outlet temperature
ret retention
In  inlet temperature
st  steady-state
set selected temperature reference

Greek letters

$\dot{V}$  heat transfer fluid volume flow rate [l/min]
$\eta$  efficiency
$\tilde{\varepsilon}$  effectiveness
$\mu$  dynamic viscosity (Pa.s)
$\Delta$  incremental
relative deviation

Velocity (m/s)

angle radian

density (kg/m$^3$)

Small number prevent division by zero

thermal expansion coefficient

**Abbreviations**

ATCT actual total cycle time

CES Chemical energy storage

CF compactness factor

CFD computational fluid dynamics

CNS Chinese national standard

DSC differential scanning calorimeter

EHC effective heat capacity

ESC energy storage capacity

ETC evacuated tube collector

ETC-S evacuated tube collector-storage

FPC flat plate collector

HTF heat transfer fluid

ISO international organization for standardization

KPI key performance indicators

LHS latent heat storage

MCL minimum cycle length

PCM phase change material

RMSE root mean square error

SDWH solar domestic water heating

SHS sensible heat storage

SOC state of charge

SWH solar water heater

TES thermal energy storage
CHAPTER ONE

INTRODUCTION
1- INTRODUCTION

1.1- Introduction
The world energy statistics show a remarkable increase in consumption by 54% in the previous 25 years [1]. Moreover, almost 50% of the energy consumed in 2015 was used in heating applications [2]. Nowadays, the main source of this energy is non-renewable energy resources such as fossil fuels. However, day by day the renewable energy technologies become a promising source to meet the increasing demand for energy. Different renewable energy technologies could be used in heating applications such as solar energy, geothermal, and biomass. One of the most popular and feasible heating technologies for domestic applications is the solar collector. Therefore, the dependence on the solar collector in heating applications significantly increased during the last 10 years, where the total installed capacity was increased worldwide to be 435 GW [1]. This promising technology is facing a huge obstacle that prevents its spread, which is the lack of dispatchability. The solar collector output depends on the solar radiation on the collector surface. This radiation is affected by several factors such as climate, sun location, and shadow from other buildings, which are unpredictable and fluctuating on an hourly basis. Therefore, the domestic thermal solar collectors should include a storage sub-system to overcome the supply-demand problem. On the lights of the previous facts, many researchers directed their attention to innovate and design an economical and technically feasible solution to this problem. One of the most promising solutions is thermal energy storage (TES). The thermal energy could be stored by several means Sensible heat [3], latent by phase change materials [4], and chemical by chemical reactions [5]. Many researchers preferred the latent energy storage by using the phase change materials (PCM) over the other storage technologies. This is because of several advantages of the latent heat thermal energy storage (LHTES)
technologies such as (1) high storage density in minimum volume compared to sensible methods, and (2) the minimum temperature change during melting or solidification processes [6]. Researchers have used PCM in many ways to store thermal energy such as PCM capsules [7] in the water tank, PCM in heat exchanger [8], and integrated PCM-solar collector [9].

In recent years, a notable effort was made to develop a compact evacuated tube solar/collector (ETC-S) heating system. Felinski and Sekret [10], developed and tested a novel collector storage unit using an ETC filled with paraffin wax. The results showed that the heat losses from the proposed system were less than the conventional ETC by 32%, and the useful heat gained from the ETC-S was higher than the energy received from the conventional ETC by 45-79%. Wu et al. [11], experimentally tested two identical ETC systems combined with an oscillating heat pipe. The results showed that the performance of the system with PCM outweighed the conventional system under similar operating conditions. Abokersh et al. [12], [13], integrated commercial-grade paraffin wax to a U-tube ETC system and compared its performance against a conventional ETC system under different charging and discharging scenarios. The results showed that the integration of PCM to the ETC system enhanced the thermal performance of the system in comparison with the typical ETC system by 47% [12], [13].

The previous literature showed that integrating PCM directly into ETC systems was a promising technique to develop an innovative and reliable ETC-S system. The research done on the integrated flat plate systems achieved a pioneering level in redesigning the collector to include the new storage features to the regular design. On the other hand, the research implemented on the ETC-S system based on using ETC was limited and still in the feasibility and validation phase [14]. Thus, most of the research executed in this field was conducted using the standard ETC. Given the fact that the standard ETC was designed and optimized to be used as a solar
collector only, it was clear that a redesigning and optimization effort should be made to achieve the required level of reliability from the ETC-S based on ETC systems.

The novelty of this work that it provides an essential theoretical and experimental methodology as a first step on the progress of redesigning the conventional ETC to combine energy collection and storage features in the same module. This research is the second phase in the ongoing ambitious research project that is aiming to propose a novel integrated ETC to the solar thermal energy industry [12], [13]. Typically, the ETC consists of several glass tubes connected in series or in parallel, which could be easily replaced for maintenance or upgrading. These glass tubes are considered as a building block for the ETC. As such, the first bridge to redesign the ETC-S system that uses ETC is a complete characterization of the thermal performance of a conventional single ETC system’s building block is not conducted yet.

Aligning with this vision, the first part of this study aims to quantify the thermal performance of a single compact PCM-evacuated tube with the purpose of developing a complete characterization methodology to evaluate the thermal performance of this building block. The results of this study will be used as a benchmark in future redesign research to optimize the collector’s design parameters according to the new concept of integrating the thermal storage material directly into the evacuated tube. Generally, the expected outcome of this research project is proposing a design guideline for a novel ETC-S with high reliability, which can assist in increasing the market share of the STES.

In addition, the second part of this research aims to develop a design methodology that uses numerical modeling techniques to redesign a novel compact ETCS and simulate its performance under different operating conditions. This is achieved through using the commercial ANSYS-FLUENT in combination with a developed in-house code to investigate the performance of a
single evacuated tube filled with PCM. The simplified model is validated with experimental results and the results obtained from different CFD techniques. Subsequently, the model is integrated into a novel methodology that proposes a collector design based on the available resources and the final application requirements and standards.

1.2- Contribution of research
This research is dedicated to

- Present a complete thermal characterization for the building block of the targeted novel ETCS.
- Study the effect of using longitudinal fins on the thermal performance of ETCS.
- Model the proposed building block using the commercial computational fluid dynamic (CFD) software ANSYS-FLUENT.
- Compare between different numerical modeling techniques used for simulating PCM.
- Develop a simplified model that simulates the discharging of PCM and use this model for parametric analysis of design parameters.
- Propose a design for a novel ETCS that satisfies the needs of domestic water heating systems.
CHAPTER TWO

LITERATURE REVIEW
2- LITERATURE REVIEW
Solar thermal applications account for 8% of the total renewable energy technologies used in the heating sector [15]. This low percentage of contribution to the heating and cooling sector is mainly due to the uncontrollable energy supply. This drawback prevents the commercial or the domestic sectors from depending on solar energy as a reliable and stable source of heat [16]. Despite this obstacle, the capacity of installed solar thermal collectors in 2015 was near to 38 GWth [2]. All of these facts motivated the researchers to investigate different techniques to store thermal energy as a potential means to overcome the unreliability of solar collectors.

A typical solar water heater collector consists of four main components: Solar collector, storage tank, a circulation pump, and the heat transfer fluid (HTF). Jamar et al.[17] reviewed the recent research done to study the effect of changing these components on the performance of the domestic solar water heater. There are two types of domestic solar collectors either flat plate collector [17] or evacuated tube collector[18]. Zambolin and Col[19] compared the performance of the evacuated tube collector and flat plate collector and concluded that the efficiency of the evacuated tube collector is much higher than the flat plate collector. The second important component of the solar domestic water heater systems (SDWH) is the storage tank. The most common type used to store heat in the SDWH systems is the closed-loop water storage tank which contains an electric heater to heat water on cloudy days. However, the main drawbacks that set barriers to the spreading of SDWH systems are that the storage tank is expensive and needs a large space. Accordingly, intensive research was carried out to investigate the potential of PCM to directly store solar energy in the form of latent heat. The PCM could be used in SDWH system in different ways such as:

1- PCM in the storage tank
2- PCM in flat plate collectors
3- PCM in evacuated tube collectors

2.1- PCM In Storage Tank
The water heat storage tank is a primary part of the SDWH systems, and it is the most expensive part of the system. Accordingly, many researchers attempted to propose PCM as a storage medium instead of /in addition to water to reduce the required capital cost to install a new SDWH system. The integration of PCM as SDWH storage tank could be done through putting PCM capsules in the water tank to combine between sensible and latent heat storage in one tank [27-33] or fill the PCM in a heat exchanger and use it directly as a latent heat storage tank [35-38].

2.1.1- PCM encapsulation
Many researchers studied different shapes of PCM capsules such and as cylindrical [20], [21] as shown in Figure 1, spherical [22],[23] as shown in Figure 2. Kenisarin et al. [24] reviewed the experimental and numerical research that investigated the PCM melting/solidification in spherical capsules. Canbazoğlu et al.[25] investigated the performance of a passive solar water heater while integrating PCM cylindrical capsules in a 190 L water tank. The researchers used sodium thiosulfate as PCM with a melting temperature suitable to the domestic applications 48 °C. The results showed that the integration of PCM capsules maintained the water temperature in the tank at 45°C for 10 hours. On the other hand, sodium sulfite is not a stable material to be used without special encapsulation in a repeated charging-discharging application.
Padma Raju et al. [26] introduced an experimental setup to study the performance of an active solar water heater system while changing the inlet water volume flowrate. The researchers investigated different flowrates 2, 4, and 6 l/min, and they obtained that the charged energy is directly proportional to the inlet flowrate. On the other hand, the charging time was decreased by increasing the inlet flow rate.

For the spherical capsules shape, Khot et al. [27] investigated experimentally the performance of a 10-Liter water storage tank while it was filled with spherical PCM capsules. The researchers encapsulated paraffin wax HS 58 in a 75 mm spherical metal capsules and packed it in the water tank to occupy a volume percentage equal to 26%. The researchers charged the tank and compared the energy remained in the PCM-water tank with the conventional sensible tank. The results showed that the encapsulation of PCM in a storage tank increased its efficiency by 22% compared to the sensible tank. Moreover, after repeating the same experiment with different tank sizes and different PCM volume percentage from 40% to 43%, the results showed that the efficiency of the tank increased by a percentage ranging from 41% to 46%. Moreover, Khodadadi and Zhang [27] conducted an experimental study to investigate the effect of buoyancy forces on the melting time. The authors concluded that the conduction mode is the dominant heat transfer mode at the beginning of the experiment. Also, the top region of the
sphere melted faster than the bottom levels as a result of the natural convection which can be
determined through Rayleigh number.

![Figure 2 Spherical PCM capsules](image)

Lee e al.[29] developed a novel numerical model based on the effective thermal conductivity
technique to investigate the effect of using the different ratio for cooling and heating PCM on the
thermal performance of a hybrid storage tank. The results showed that the optimum cooling
PCM filling ratio range between 50% to 90%.

### 2.1.2- PCM in the heat exchanger

The main disadvantage of adding PCM capsules to the conventional water storage tank that the
storage tank dissipates heat to the surrounding environment during the night. Consequently, the
researchers investigated some new techniques to fill a heat exchanger unit by PCM material and
use this exchanger as a storage unit in the SDWH systems [30].

Haillot et al.[30] introduced a heat exchanger filled with PCM as a storage tank and tested this
system under the French climate conditions. The results showed that the solar fraction of the
proposed system was 66% while in the conventional system it was 50%.

There are many designs of heat exchangers that could be filled with PCM such as shell and tube
heat exchangers, double tubes, and triplex heat exchangers.
Akgun and Aydin [31] filled a typical shell and tube heat exchanger with paraffin wax that has a melting temperature of 44°C. The researchers investigated the performance of the heat exchanger during charging and discharging processes by investigating the effect of flow rate and inlet temperature. The results showed that the inlet temperature has a more significant effect on the heat exchanger performance than the mass flowrate.

For the double pipe heat exchangers, Fath [32], developed an analytical model to investigate the performance of a double pipe heat exchanger filled with paraffin wax with a melting temperature 50 °C. The results showed that increasing the inlet water flow rate increased the heat transfer rate between tubes and PCM. Also, the inlet mass flow rate was found to be directly proportional to the amount of energy stored.

In order to enhance the heat transfer rate between the PCM and the heat transfer fluid (HTF), the researchers proposed the triplex heat exchanger as a promising design to benefit from increasing the heat transfer area between PCM and HTF. For Example, Basal and Ahmet [33], numerically investigated a triplex tube filed with RT-52 during a charging process. The researchers studied the heat transfer performance of the system by studying the effect of mass flow rate on a dimensionless number which (Fourier number) represents the melting time. The results showed that increasing the inlet temperature from 50 °C to 70 °C decreased the melting time by 500%.

2.2- Using Computational Fluid Dynamics (CFD) to simulate PCM applications.
Computational Fluid Dynamics (CFD) is a reliable numerical technique to study the performance of latent heat storage applications through numerical model simulation. Several commercial CFD codes could be used to numerically study the melting and the solidification of a PCM in different applications such as Ansys Fluent, Ansys CFX, and COMSOL. Most of these commercial codes
use a finite volume method to numerically solve time-dependent Navier-Stokes equations [34]. Many researchers have investigated the PCM melting and solidification through simulation. For instance, Asses et al.[35] numerically investigated the melting and solidification of PCM in a spherical capsule using commercial CFD code ANSYS-FLUENT. The authors used the validated numerical model to generate a formula that can predict the melting fraction of the capsules at any time. Wang et al.[36] developed a numerical model to investigate the effect of novel stratified on the thermal performance of a hybrid sensible-latent storage tank. The researchers used the validated CFD model to visualize the effect of PCM location on the stratification inside the heat storage tank. Hosseini et al.[37] numerically investigated the design of longitudinal fin on the PCM melting inside a shell and tube TES system. The results showed that the utilization of fins decreased the melting time and increased heat penetration. Tao et al.[38] numerically studied the effect of fins and natural convection on the melting of PCM inside a horizontal tube. The results showed that the utilization of fins decreased the non-uniformity caused by natural convection. Moreover, the researchers used the visualization of PCM temperature distribution and melting front to optimize the design of the used fin. Tiari and Que [39], developed a three-dimension model to simulate a latent heat thermal energy storage unit and studied the effect of different heat transfer enhancement techniques as shown in Figure 3. The results showed that increasing the number of heat pipes within the storage unit significantly improved the heat transfer rate and decreased the required charging time.
In addition, Zhang and Xiao [40], developed a three-dimensional model to simulate the performance of a water storage tank filled with commercial cylindrical PCM capsules coated with nickel foam as shown in Figure 4. The results indicated that coating the PCM capsules with nickel foam enhanced the thermal conductivity of the capsules and improved the total performance of the latent heat storage unit.
2.3- PCM In Solar Collectors

Many research efforts were conducted to decrease the cost of the domestic solar water heater. One of the many suggestions to achieve this goal is by eliminating the storage tank which is the most expensive component from the system. However, the storage tank should be replaced with a storage component to maintain the supply of hot water during the night or the cloudy climate days. A promising solution is to design a tankless solar collector system by integrating the PCM directly into the solar collector to store the solar heat during the day without using any other storage tanks. There are two main design categories of solar collector either flat plate or evacuated tube collectors. Both of these design categories have been studied as a tankless solar collector.

For the flat plate solar collector, Kürklü et al. [41], integrated a PCM storage component directly to a flat plate collector as shown in Figure 5. The storage component was made from galvanized steel and filled with paraffin wax. The researchers investigated the performance of the collector during the fall season in Turkey. The results showed that besides the improvements in the heat transfer rate and the collector absorptivity, the total cost of the system was decreased by 41%.
In addition, many researchers investigated the parameters that affect the performance of compound PCM-solar collectors such as the PCM thickness layer, PCM melting temperature, and HTF charging/discharging flowrate and temperature [42],[43],[44], [45].

Another common type of solar collector is the evacuated tube collector. Although this type of collector is more efficient than the flat plate collector in terms of solar fraction and absorptivity, limited research focused on the integration of PCM directly into the evacuated tubes [13],[46]. Abo Kersh et al. [13],[46], investigated the performance of an evacuated tube collector filled with paraffin wax. The researchers compared the performance of their proposed system and the conventional evacuated solar collector. The results showed that the proposed system efficiency is higher than the conventional system by 35%. Papadimitratos et al. [47] proposed a novel ETCS where they immersed Tritriacontane and Erythritol in the cavity between heat pipes and glass tubes to benefit from the high thermal insulations of the evacuated tubes. Essa et al. [48] used a conventional U-tube direct flow collector as a base for their ETCS system as shown in Figure 6. The results showed that the complete phase change and the full storage benefit is achieved by low flowrates.
The previous literature showed that the integration of PCM directly into solar collectors is a promising technique for TES. Moreover, because the evacuated tube collectors have superior thermal performance compared to the flat plate collectors. There is a need for investigating the integration of PCM to evacuated tube collectors to propose a new collector design that includes the energy collection and storage feature in a single unit.
CHAPTER THREE

METHODOLOGY
3- METHODOLOGY

3.1- Thermal Energy Storage
There are different promising techniques to store thermal energy storage (TES) such as Sensible heat Storage (SHS), Chemical Energy Storage (CES), and Latent heat Storage (LHS). Firstly, the (SHS) technique is the oldest and the most popular storage technique used with solar collectors. In this technique, the heat is stored by increasing the temperature of the storage medium by applying a heat flux to it [49]. The storage medium could be water [50], stone [51], or clinker bricks [52]. Secondly, CES technology became a promising technology in the recent decade because of its ability to store energy in a loss-free status for a long time[5]. This method uses the endothermic and exothermic chemical reactions to store energy [53]. Thirdly, LES techniques which use phase change materials (PCM) to store latent heat during the phase transformation from solid to liquid or vice-versa. Zalba et al.[54] categorized the different types of PCM into organics and inorganics material as shown in Figure 7.

Moreover, Table 1 shows some criteria that should be fulfilled while selecting a material to be used as a PCM [55]. In the last decades, the LES techniques have been highly investigated.
because of the advantage of the PCM as storage material over the conventional sensible storage materials such as i) High storage capacity, ii) Low storage volume, and iii) Isothermal operation during the charging and discharging phases [56], [57].

Table 1 PCM selection criteria [55]

<table>
<thead>
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<th>Properties</th>
<th>Selection criteria</th>
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<tbody>
<tr>
<td>Thermal properties</td>
<td>Suitable operation temperature range</td>
</tr>
<tr>
<td></td>
<td>High latent heat per unit mass</td>
</tr>
<tr>
<td></td>
<td>High specific heat</td>
</tr>
<tr>
<td></td>
<td>High thermal conductivity</td>
</tr>
<tr>
<td>Physical properties</td>
<td>Low-density variation during phase change</td>
</tr>
<tr>
<td></td>
<td>Low or non-super cooling during freezing</td>
</tr>
<tr>
<td>Chemical properties</td>
<td>Chemically stable</td>
</tr>
<tr>
<td></td>
<td>Non-toxic</td>
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<tr>
<td></td>
<td>Non-flammable</td>
</tr>
<tr>
<td>Economic factors</td>
<td>Abundant</td>
</tr>
<tr>
<td></td>
<td>Affordable</td>
</tr>
</tbody>
</table>

3.2- **Commercial CFD Software**

Any computational fluid dynamics code consists of three main software: i) pre-processor (Meshing) software, ii) solver software and iii) post-processing. Each of these programs has a significant role in the development and validation of the CFD models.

1- **Pre-processor**

The main function of the pre-processor program is to aid the user to define is model through three main activities:

i) Define the geometry: most of the CFD pre-processing program have a built-in CAD part or can read the CAD files from the other drawing soft-wares
ii) **Meshing**: this is the main purpose of these programs, the defined domain is divided into tiny cells, either Hexahedral or Tetra cells, where each of these cells represents a control volume

iii) Define the material of the domain and the boundary conditions

**2- Solver**

The solver could be defined as the program that uses one of these numerical methods, Finite difference, Finite element, or Finite volume, to solve the Navier-Stokes equations. Most of the commercial CFD software uses the finite volume method to solve the governing equations through three main steps:

i- **Integration**: of the differential equations with/without respect to time over a control volume

ii- **Discretization**: Convert the differential equations to algebraic equations

iii- **Iterative solution**: Solve the algebraic equation by several iterations until convergence

**3- Postprocessor**

The post-processing software could be defined as the graphics program in the CFD package, the main purpose of this program is to visualize the results and generate graphs through different features and tools such as:

i- Particle tracking

ii- Chart drawer

iii- Planes and cross-sections
3.3- Governing Equations

The conservation concepts are the main physical law that is used to derive the governing equations for fluid flow.

1- Conservation of mass

2- Conservation of momentum

3- Conservation of energy

i- Conservation of mass

Based on the old physics concept that “Mass can neither be created nor destroyed” a continuity equation could be easily derived to its final known form Eq. (1)

\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho U) = 0
\]  

(1)

ii- Conservation of momentum

Based on mechanical law that state the momentum as a conserved quantity the scientists derived the momentum equation

\[
\frac{\partial (\rho U_i)}{\partial t} + \nabla \cdot (\rho U U_i) = - \frac{\partial P}{\partial x_i} + \nabla \cdot (\mu \nabla U_i), i = 1, 2, 3
\]  

(2)

iii- Conservation of energy

Based on the well-known concept that Energy cannot be created or destroyed the energy equation is derived from the following form

\[
\frac{\partial (\rho e)}{\partial t} + \nabla \cdot (\rho eU) = -p \nabla \cdot U + \nabla \cdot (k \nabla T) + \Phi
\]  

(3)

\[
\Phi = \tau_{ij} \frac{\partial U_i}{\partial x_j}
\]  

(4)
In this study, the charging and discharging of the proposed ETCS building block is simulated by using the melting/solidification module of ANSYS Fluent 16.1. The ANSYS’s melting/solidification module uses the enthalpy porosity method that is developed by Voler and Prakash [58]. This numerical technique defines the mushy zone as a porous medium to model the material phase change from solid to liquid as porosity changes from 0 to 1. Moreover, the Boussinesq approximation is used to simulate the effect of natural convection due to buoyancy forces inside the tube. The Governing equations for solving the given problem are illustrated as follows [34], [39].

a- Continuity equation:

\[
\frac{\partial \rho}{\partial t} + \frac{1}{r} \frac{\partial}{\partial \theta} (r \rho v_r) + \frac{1}{r} \frac{\partial}{\partial z} (\rho v_z) = 0
\]  

(7)

Where \( \rho \) is density, \( v_r \) velocity in \( r \), \( \theta \), and \( Z \) directions.

b- Momentum equations

b.1- In \( r \) direction

\[
\frac{\partial v_r}{\partial t} + v_r \frac{\partial v_r}{\partial r} + v_\theta \frac{\partial v_r}{\partial \theta} + \frac{v_\theta^2}{r} + v_z \frac{\partial v_r}{\partial z} = \frac{1}{r} \frac{\partial}{\partial r} (r \tau_{rr}) - \frac{v_r}{r^2} \frac{\partial v_r}{\partial \theta} - \frac{2}{r^2} \frac{\partial v_r}{\partial \theta} + \frac{\partial^2 v_r}{\partial z^2} + S_r
\]  

(8)

b.2- In \( \theta \) direction
\[
\rho \left( \frac{\partial \nu_\theta}{\partial \tau} + \nu_\theta \frac{\partial \nu_\theta}{\partial \tau} + \nu_\theta \frac{\partial \nu_\theta}{\partial \theta} + \frac{\nu_r \nu_\theta}{r} + \nu_\theta \frac{\partial \nu_\theta}{\partial z} \right)
\]
\[
= -\frac{1}{r} \frac{\partial P}{\partial \theta} + \mu \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial \nu_\theta}{\partial r} \right) - \nu_\theta \frac{1}{r^2} \frac{\partial^2 \nu_\theta}{\partial \theta^2} - \frac{2}{r^2} \frac{\partial \nu_r}{\partial \theta} + \frac{\partial^2 \nu_\theta}{\partial z^2} + S_r \tag{9}
\]

b.3- In \textit{z}-direction
\[
\rho \left( \frac{\partial \nu_z}{\partial \tau} + \nu_z \frac{\partial \nu_z}{\partial \tau} + \nu_\theta \frac{\partial \nu_z}{\partial \theta} + \nu_z \frac{\partial \nu_z}{\partial z} \right)
\]
\[
= -\frac{\partial P}{\partial z} + \mu g \beta (T - T_0) + \mu \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial \nu_z}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 \nu_z}{\partial \theta^2} + \frac{\partial^2 \nu_z}{\partial z^2} + S_r \tag{10}
\]

Where \( \mu \) is the dynamic viscosity, \( g \) is the gravity acceleration, \( \beta \) thermal expansion coefficient, and \( S_r \) sink term.

To simulate the PCM melting and solidification processes Voler and Prakash developed the Enthalpy porosity technique \cite{58}. In this technique, the porosity is added to the momentum equations (9)-(10) and (11)-(10) as a sink term \( S \) as shown in Eq. (11)-(14) and the phase of each cell is determined based on the liquid fraction in this cell as shown in Eq. (14)-(14)

\[
S_r \frac{(1-f)^2}{f_1+f_\text{e}} A_{\text{mush}} \nu_r \tag{11}
\]

The liquid fraction \( f \) is calculated through equations

\[
f = 0 \quad T < T \text{ solidification} \tag{12}
\]
\[
f = 1 \quad T > T \text{ melting} \tag{13}
\]
\[
f = \frac{T-T_{\text{solidification}}}{T_{\text{melting}}-T_{\text{solidification}}} \quad T \text{ solidification} < T < T \text{ melting} \tag{14}
\]

c. Energy equation
\[
\frac{\partial (\rho H)}{\partial \tau} + \frac{1}{r} \frac{\partial}{\partial r} \left( r \nu_r H \right) + \frac{1}{r} \frac{\partial}{\partial \theta} \left( r \nu_\theta H \right) + \frac{\partial}{\partial z} (\nu H)
\]
\[
= \frac{1}{r} \frac{\partial}{\partial \theta} \left( k \frac{\partial T}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 T}{\partial \theta^2} + \frac{1}{r} \frac{\partial}{\partial \theta} \left( k \frac{\partial T}{\partial \theta} \right) + \frac{\partial^2 T}{\partial z^2} \tag{15}
\]
Where $H$ is the heat energy of PCM, and $K$ is the thermal conductivity of PCM. The enthalpy is calculated by Eq. (16) [59].

$$H = h + \Delta H$$  \hfill (16)$$

Where

$$h = h_{\text{ref}} + \int_{T_{\text{ref}}}^{T} C_p dT$$  \hfill (17)$$

$$\Delta H = f_l h_{\text{sl}}$$  \hfill (18)$$

$h_{\text{ref}}$ and $T_{\text{ref}}$ are the reference enthalpy and temperature. $C_p$ is the PCM specific heat. The latent heat $\Delta H$ is calculated based on the liquid fraction ($f_l$) at each time step.

### 3.4 - Performance Parameters

#### 3.4.1. Nominal Temperature ($T_{\text{nom}}$)

The nominal temperature of the TES module is defined as the required HTF outlet temperature, which varies according to the application requirements [60]. The proposed integrated tube is designed to be used as a building block for an ETC-S system. So, the $T_{\text{nom}}$ of this application is determined according to solar water heaters standards such as the Chinese National Standard (CNS) 7277-12558 [61] and the International Organization for Standardization (ISO) standard 97.100.99 [62]. For instance, in the domestic water heating applications, the required water temperature is between 45 and 55°C [63], [64]. Meanwhile, in the commercial and industrial applications, the nominal temperature could reach more than 200 °C [65], [66].

#### 3.4.2. Nominal power ($P_{\text{nom}}$)

Theoretically, the TES module has a designed constant charging/discharging nominal power. This theoretical power is determined according to the nominal temperature of the application ($T_{\text{nom}}$) and the normal ambient temperature ($T_a = 25 ^\circ C$) by Eq. (19) [60].
Where \( \dot{m}_{\text{HTF}} \) is the minimum flowrate available to be used in the application, and \( C_{\text{HTF}} \) is the specific heat of the circulating fluid. However, in real applications, the charging or discharging power is not constant. The power changes during the charging/discharging cycle due to: (i) decrease in temperature difference between HTF and PCM, (ii) the increasing PCM thermal resistance with time, and (iii) the heat losses from the LHS unit to surrounding environment [31–33].

### 3.4.3. Energy storage capacity (ESC)

The energy storage capacity (ESC) is defined as the maximum energy storage capacity of a TES device if its temperature raised from ambient temperature to a specific higher temperature. It is calculated as follows:

\[
\text{ESC} = \sum_{i=1}^{n} M_i C_i \Delta T + M_p H_L
\]  

(20)

Where \( n \) is the number of components in the TES device. \( M \) and \( C \) are the mass and specific heat of each component, respectively. \( M_p \) is the mass of PCM material. \( H_L \) is the latent heat of the PCM material.

### 3.4.4. Actual energy delivered/discharged

The actual energy charging/recovered represents the accumulative thermal energy transferred between the HTF and the PCM during the charging/discharging cycles. The actual energy delivered or recovered is calculated by Eq. (21) and Eq. (22), respectively [19, 23].
\[
E_{ch} = \rho_{HTF} \dot{V} C_{HTF} \int_{0}^{t_{ch}} [T_{in} - T_{out}] \, dt \\
E_{dis} = \rho_{HTF} \dot{V} C_{HTF} \int_{0}^{t_{dis}} [T_{out} - T_{in}] \, dt
\]

Where \( \rho_{HTF} \) is HTF density, \( \dot{V} \) is the HTF volume flow rate, \( C_{HTF} \) is the specific heat of the circulated HTF, \( T_{in} \) and \( T_{out} \) are HTF inlet and outlet temperature, respectively. Moreover, \( t_{ch} \) and \( t_{dis} \) represent the time intervals required to complete the charging and discharging processes, respectively.

3.4.5. Charging/Discharging Energy Efficiency (\( \eta_{ch}, \eta_{dis}, \eta_{T} \))

The charging and discharging experiments are conducted to determine the efficiency of the tested thermal storage unit. Firstly, the energy accumulated energy delivered is compared with the ESC to determine the charging efficiency by Eq. (23) [70]. Then, the recovered energy is compared with the ESC to determine the recovery efficiency through using Eq. (24) [25, 26]. Finally, the total energy charging/discharging efficiency is calculated, which is a non-dimensional parameter that represents the ratio between the energy delivered to charge PCM and the energy gained by HTF during recovery experiments by using Eq. (25) [34–36].

\[
\eta_{ch} = \frac{E_{ch}}{E_{ESC}} \\
\eta_{dis} = \frac{E_{dis}}{E_{ESC}} \\
\eta_{T} = \frac{E_{dis}}{E_{ch}}
\]

3.4.6. Effectiveness

The storage unit’s effectiveness is defined as the ratio between the actual heat energy transferred and the maximum heat energy available to transfer [37–39]. In the LHS modules, the major part
of the energy is stored in the latent heat form. Therefore, the module’s effectiveness during the sensible energy storage is negligible, and the maximum effectiveness is obtained when the HTF outlet temperature equals the PCM melting temperature as shown in Eq. (26) [32, 40].

\[ \varepsilon = \frac{T_{in} - T_{out}}{T_{in} - T_m} \]  

(26)

Where \( T_{in} \) is the HTF inlet temperature, \( T_{out} \) is HTF outlet temperature, and \( T_m \) is the PCM melting temperature.

The module effectiveness decreases with time as the PCM thermal resistance to heat transfer increases during the melting/solidification process [41, 42]. Thus, the module’s effectiveness represents the average of instantaneous effectiveness through the melting/solidification process, as shown in Eq. (27) and Eq. (28) [32, 42].

\[ \bar{\varepsilon}_{ch} = \frac{1}{t_{ch}} \sum_{0}^{t_{ch}} \varepsilon_{ch} \]  

(27)

\[ \bar{\varepsilon}_{dis} = \frac{1}{t_{dis}} \sum_{0}^{t_{dis}} \varepsilon_{dis} \]  

(28)

### 3.4.7. Heat loss coefficient

The heat loss is defined as “the rate at which heat is lost from the storage device per degree temperature difference between the average storage medium temperature and the ambient temperature” [79]. For any TES unit, the storage efficiency and unit’s reliability are directly related to the heat losses from this TES unit to the surrounding environment.

In order to estimate the passive heat loss coefficient of an LHS unit, the tested unit is charged to the maximum temperature, then the inlet fluid is halted, and the storage unit is disconnected from the charging system. The unit is allowed to discharge passively for an extended period (12 Hours
or 24 Hours). After completing this discharge period, the unit is recharged again to its maximum temperature, and the passive heat losses are estimated by calculating the energy required to recharge the unit to the uniform maximum temperature. The heat loss coefficient is estimated by Eq. (29) and Eq. (30) [46, 47].

\[
\eta_{\text{ret}} = \frac{T_{\text{final}} - T_a}{T_{\text{initial}} - T_a} \tag{29}
\]

\[
U_L = \frac{\text{ESC}}{\Delta t} \ln \left[ \frac{T_{\text{initial}} - T_a}{T_{\text{final}} - T_a} \right] \tag{30}
\]

The \( \eta_{\text{ret}} \) is the retention efficiency where \( T_{\text{initial}} \) is the initial PCM temperature at the start of the heat loss experiment, \( T_{\text{final}} \) is the final PCM temperature after the extended retention time, \( T_a \) is the average ambient temperature during the test. Moreover, the overall heat loss coefficient \( U_L \) (W/K) is calculated by Eq. (30) where ESC is the energy storage capacity of the unit when it is charged to the maximum PCM temperature, and \( \Delta t \) is the retention time before the recharging processes start.

### 3.4.8. Minimum cycle length

The minimum cycle length is defined as the time required to complete a sequential charging and discharging cycle under nominal conditions. This parameter is used to evaluate and compare the performance of the TES modules from a time perspective with tangible values. This comparison is conducted regardless of the variation of cycle time under different operation conditions. The minimum time required for the charging or discharging cycle (MCL) is calculated by the (ESC) and nominal power (\( P_{\text{nom}} \)) of this cycle, as shown in Eq. (31) [43, 44]. The ATCT is the sum of the charging and discharging cycle time, as shown in Eq. (32) [83].
However, in the real applications, the time required to complete the charging/discharging processes highly depends on the heat transfer rate between the PCM integrated to the tubes and the HTF circulated inside the copper pipes. Therefore, the actual total cycle time \((ATCT)\) is affected by the PCM temperature \(T_{\text{init}}\) and \(T_{\text{fin}}\), and HTF volume flowrate \(\dot{V}\).

### 3.4.9. Compactness factor

The compactness factor \((CF)\) is defined as the ratio of the volume of PCM to the volume of the tank [84]. This parameter is essential to compare the geometry of different LHS units. The main benefit of this parameter is to evaluate the losses in energy storage due to the utilization of heat transfer enhancement methods like fins [12], [85], multiple tubes [50, 51], and metal foam [88], [89]. The \(CF\) is calculated through Eq. (33) [48, 54].

\[
CF = \frac{V_{\text{tank}} - V_{\text{PCM}}}{V_{\text{tank}}} \quad \text{(33)}
\]

### 3.5. Uncertainty Calculations

An uncertainty analysis is essential to determine the accuracy of the obtained experimental results. The uncertainty of the calculated results depends on the instrumental error of the independently measured variables \((x_1, x_2, \ldots, x_n)\) such as PCM temperature, water temperature, and water mass flow rate. The relation between independent variables \((x_i)\) and the dependent results \((y_i)\) such as (accumulated energy, storage capacity, energy efficiency, etc.) is described by [35, 55]:
\( y = f(x_1, x_2, \ldots, x_n) \) \hspace{1cm} (34)

In addition, the general form for calculating the uncertainty of the dependent variables \((y_i)\) is calculated by Eq. (35) [35, 55):

\[
u(y) = \left[ \sum_{i=1}^{n} \left( \frac{\partial f}{\partial x_i} u(x_i) \right)^2 \right]^{1/2} \hspace{1cm} (35)
\]
CHAPTER FOUR

EXPERIMENTAL WORK
4- EXPERIMENTAL WORK

4.1. Experimental Setup
The main objective of this study is to present a detailed characterization for the thermal performance of a building block of a compact novel ETC-S system. This building block consists of an evacuated tube with 1450 mm long, 45 mm and 34 mm outer and inner diameter is filled with Paraffin wax type (Alexandria 600) [12], with thermal and physical properties shown in Table 2. This building block can include different heat enhancement configurations, such as circular and longitudinal fins. In this research, the thermal performance of this building block with longitudinal aluminum fin will be tested and compared to the performance of a building block with a plain tube. The experimental setup shown in Figure 8 is mainly consisting of two circuits for charging and discharging cycles where water was used as HTF in each circuit.

![Experimental setup](image)

Figure 8 Experimental setup (1) circulating pump, (2) connecting pipes, (3) cold-water tank, (4) hot water tank, (5) rotameter, (6) finned tube (A), (7) un-finned tube (B), (8) thermocouples, (9) data logger and (10) radiation shield.
Table 2: Thermophysical properties of the used paraffin wax (Alex PW600) [12]

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Melting temperature</td>
<td>61.8°C</td>
</tr>
<tr>
<td>Solidification temperature</td>
<td>58.3°C</td>
</tr>
<tr>
<td>Latent heat of solidification</td>
<td>201.8 kJ/kg</td>
</tr>
<tr>
<td>Latent heat of melting</td>
<td>189.8 kJ/kg</td>
</tr>
<tr>
<td>Density (solid)</td>
<td>902 kg/m³ at 22.2°C</td>
</tr>
<tr>
<td>Specific heat capacity @25°C</td>
<td>1.8 kJ/kg K</td>
</tr>
</tbody>
</table>

A schematic drawing for the experimental setup is shown where the main components are as follows: vertical evacuated pipes filled with a commercial grade of Paraffin wax (Alex-600), U-bend copper pipes, hot water tank, cold water tank, circulating pump, rotameters, thermocouples and data acquisition device.

A U-bend copper tube with 8 mm outer diameter and length 1430 mm is placed vertically in the middle of the evacuated tube to deliver and retract energy during the unit’s charging and discharging tests, as shown in Figure 10, which represents the un-finned ETC tube. Besides, as most of the PCM materials are known for their low thermal conductivity that ranges between 0.2 and 0.3 W/m.K [92], this PCM disadvantage limits the TES units’ thermal efficiency and requires the integration of different enhancement techniques to overcome this obstacle. As such, in this study, a longitudinal aluminum fin is as attached to the copper tube of
an un-finned ETC tube to extend the heat transfer surface and enhance the heat transfer rate between the HTF and PCM, as shown in Figure 10-b(i). Four thermocouples type K with uncertainty (±0.5 °C), (A1: A4) and (B1: B4), are located centrally in each evacuated tube at longitudinal distances (80 mm, 520 mm, 960 mm, 1400 mm) to measure the PCM temperature as shown in Figure 10-a. Besides, the water inlet and outlet temperatures of each unit are measured by thermocouples type J with uncertainty (±0.5 °C). The thermocouples data is stored by a data logger (Campbell Scientific-CR3000) with scanning intervals 1 second. The piping system is well insulated using 45 mm of glass wool material to minimize the heat losses to the surrounding environment. Moreover, the whole apparatus setup is covered with a reflective aluminum sheet to reduce the radiation effect to negligible levels, as shown in Figure 8. In the following sections, the finned tube will be mentioned as a finned tube (A) while the un-finned tube will be mentioned as an un-finned tube (B).

In the charging circuit, an electric heater with power (2000W) is connected to a thermostat (Autonics-TC4M) to maintain the water temperature inside the hot tank at the selected level within ±1 °C differential. A centrifugal pump (Calpeda-NGM4E) with a capacity of 1 HP and 4l/min at a head of 41 m is used to supply the hot water to the storage units at a constant flow rate until the end of the charging cycle. During the charging experiments, outlet water from the tubes was returned to the hot water tank to minimize the energy required to maintain the water temperature at the required level. On the other hand, the discharging loop used a continuous feed of water drawn from the municipal supply with a temperature around 25 ±3 °C to maintain a constant level in the cold tank during the heat recovery cycle. Meanwhile, the same centrifugal pump is used to circulate the cold water inside the tubes.
Figure 10 cross-section for the tested tubes A and B, where (a) a longitudinal cross-section view shows the location of thermocouples (A1: A4) and (B1: B4) inside tubes A and B. (b) the top view of the tested tubes shows the used aluminum fin where (i) evacuated tube (A); and (ii) evacuated tube (B).
4.2. Experimental Procedure
Before starting the charging/discharging experiment, two conditions should be achieved. Firstly, the uniformity of the PCM temperature inside the module must be verified where the temperature difference between the PCM and HTF should be (±1 °C). Secondly, the variation in the inlet mass flow rate must be less than 1% to be considered as a steady-state flow [33, 56].

The standard test procedure consists of four sequential periods, as shown in Figure 11-a. Firstly, verify the achievement of the previously described initial steady-state conditions for the time $t_{st}$. Secondly, the charging experiment during the charging time $t_{c}$, where the HTF inlet temperature and mass flow rate are adjusted to the selected values, then the experiment continues until the PCM temperature reaches the uniform condition, and the difference between the HTF inlet and outlet temperature reaches the steady-state condition. Thirdly, the tested module is maintained at the steady-state condition for at least one hour, time $t_{st}$ to ensure the homogeneity of PCM temperature inside the module. Finally, the discharging experiment starts, discharging time $t_{d}$, after obtaining the steady-state conditions where the HTF inlet temperature is decreased to a value lower than the PCM temperature (e.g., ambient temperature), and the test is performed until the PCM temperature and the HTF inlet/outlet difference reach the steady-state condition as shown in Figure 11-a and Figure 11-b. During this experiment, the data is collected and stored for the essential measurements such as HTF inlet temperature, HTF inlet mass flow rate, HTF outlet temperature, PCM temperature, ambient temperature, and time. The necessary data is recorded and used to calculate the energy recovered and the discharging time.
Figure 11: The standard experimental procedure where (a) PCM temperature during the complete testing procedure for latent storage modules, (b) Water temperature during the complete charging and discharging cycle.

Figure 12: Shows the set of experiments that are selected to study the performance of the proposed building block. Both finned tubes A and un-finned tube B are experimented using different PCM initial/final temperature. Moreover, the effect of changing the HTF volume flowrate $\dot{V}$ on the thermal performance of the ETC tubes are studied using three different flowrates. Firstly, for the PCM initial/initial temperature, considering that the maximum working temperature for the used wax (Alex 600) is 85 °C [94], the selected maximum initial/initial temperature was 80°C. Moreover, three additional levels of PCM temperatures with difference 5 °C were studied to evaluate the
effect of PCM temperature on the thermal performance of the tested tubes. Secondly, for the HTF volume flow rate $\dot{V}$, the previous study showed that the maximum allowable flowrate ($\dot{V}_{\text{max}}$) to avoid glass thermal shock in the solar collector is 4 l/min [12]. As such, the tested $\dot{V}_{\text{max}}$ for one tube was 0.5 l/min, and the effect of $\dot{V}$ on thermal performance was studied using two additional lower levels of 0.3 and 0.4 l/min.

\begin{figure}[h]
    \centering
    \includegraphics[width=\textwidth]{figure12}
    \caption{Set of experiments in this study}
\end{figure}

4.3. Experimental Results

The results obtained from the conducted experimental work are presented and discussed in the following section. The results are subdivided into four main sections. These sections comprise (i) analyze for the temperature profiles of PCM and circulated HTF during different charging/discharging experiments, (ii) the thermal performance of the tested tubes A and B, (iii) the heat loss coefficients from tubes A and B, and finally (iv) analyzing the effect of using fin on the performance of the tested tube.

4.3.1- Repeatability

The repeatability verification was done by repeating charging/discharging experiments with operating conditions $T_{\text{initial/final}} = 80^\circ$C and $\dot{V} = 0.5$ l/min as shown in Figure 12. The selected experiment was repeated for three times, and the results
repeatability is shown in Figure 13-a and Figure 13-b. The maximum root means square error (RMSE) of charging experiments is 1.24°C, while the maximum RMSE for discharging experiments is 2.31°C which confirm the repeatability of the experimental results.

Figure 13 Repeatability experiments: (a) Charging experiment at $\dot{V}=0.5$ l/min and $T_{\text{initial}}=80^\circ\text{C}$, (b) Discharging experiment at $\dot{V}=0.5$ l/min and $T_{\text{initial}}=20^\circ\text{C}$
4.3.2- Temperature profile
   a- PCM temperature profiles
Aiming at describing the thermal behavior of the PCM during the conducted charging/discharging experiments, the transient change in PCM temperature is recorded and analyzed in the following section. The analysis of PCM temperature is essential to characterize the performance of the tested LHS module from several aspects such as identify the dominant heat transfer mode, determine the actual phase transition temperatures, and study the temperature distribution inside the module due to natural buoyancy forces. As such, the PCM temperature is registered during the charging/discharging experiments using four thermocouples positioned at different elevations, as described in section 4.3. Figure 14 presents the PCM average temperature profile for both tubes A and B during a complete charging/discharging experiment where the tubes A and B were charged/discharged using operation conditions $T_{\text{initial/finat}} = 80^\circ\text{C}$ and $\dot{V} = 0.5$ l/min. Figure 14 presents the time required to complete the melting process in the experiment, which is approximately 40 and 80 minutes for the tubes A and B, respectively. Moreover, during the discharging processes, the presented average PCM temperature profile for both tubes at the same experiment illustrates that the time required to complete the discharging processes is higher than the charging time. Explicitly, the complete energy recovery processes time was approximately 60 and 100 minutes for the tubes A and B, respectively. This is mainly because, during the discharging processes, the solidified PCM creates a layer around the copper pipes that decrease the heat transfer rate between the PCM and HTF due to the low thermal conductivity of the utilized paraffin wax.
Figure 14 PCM average temperature profile for tubes A and B during the experiment (1.1) with $\dot{V}=0.5$ l/min and $T_{\text{init}}=80^\circ\text{C}$.

Figure 15 illustrates the change in PCM temperature at different elevations during the charging processes. Firstly, during the charging process, the temperature profiles show that the PCM temperature increases gradually with time independently of the thermocouple elevation. This proves that conduction is the main heat transfer mode during this period. Afterward, the melting phase starts around 70°C, and the measured PCM temperatures start to diverge based on the elevation of the thermocouple. This separation occurs due to the natural convection inside the tube that drives the melted PCM upwards, this increases the melting rate in the top elevations, and the convection heat transfer becomes the dominant heat transfer mechanism since it appears at more than 65% of the charging experiment time. On the other hand, Figure 15-a and Figure 15-c illustrate that the used longitudinal aluminum fin mitigates the effect of natural convection on the PCM temperature inside the tube. The divergence between the measured temperatures at different elevations in the tube A, as shown in Figure 15-a is less than the difference obtained in tube B, as shown in Figure 15-c. Moreover, Figure 15-c presents that the PCM temperature is
almost flat at 70 °C for an extended period until the end of the melting phase. However, in tube A, Figure 15-a shows that the utilization of longitudinal fin homogenizes the PCM temperature at different elevations and minimizes this phase change time where the PCM temperature reaches the final required temperature without wasting extended time in the melting phase.

Secondly, during the discharging processes, Figure 15-b and Figure 15-d show that the PCM temperature rapidly decreases in the first phase of the discharging processes until the PCM starts to solidify after 8 and 9 minutes for tubes A and B, respectively. Afterward, the phase change stage begins when $T_{PCM}$ was around 63°C, and the PCM temperature remains constant for a short period of time. Specifically, in tube B, the observed solidification time $t_{s}$, approximately 5 minutes, is less than the recorded melting time, which lasted for more than 13 minutes. This is mainly because, during the charging cycle, a respectful amount of energy must transfer from the HTF to the PCM at the melting front to melt the PCM surrounding the thermocouple located in the center of the tube. While the melting process proceeds, the temperature difference between the PCM and the HTF decrease, which slow the heat transfer rate between the two mediums and increase the melting time. In contrast, during the discharging processes, the PCM surrounds the thermocouple probe solidify faster than the other areas in the tube. As such, the recorded PCM temperature showed a much lower time for solidification compared to the melting time.

Finally, a similar pattern was obtained in the other charging/discharging experiments. As such, to avoid redundancy, only the results of the experiment with $\dot{V}=0.5$ l/min and $T_{initial}=80^\circ$C are presented in this publication.
Figure 15 PCM temperature profile at different elevations for $V=0.5$ l/min and $T_{\text{initial}}=80^\circ$C which includes (a) charging experiment for finned tube A, (b) discharging experiment for finned tube A, (c) charging experiment for un-finned tube B, (d) discharging experiment for un-finned tube B

**b- Water temperature profile**

The main application for the proposed ETC-S system is domestic water heating. Hence, the outlet water temperature is an essential parameter that should be studied to determine the feasibility of any new ETC-S design. Because the proposed ETC-S will consist of several tubes connected in series and/or parallel, the required nominal temperature described in section 3.4 will not be achieved by a single tube, the HTF outlet temperature was recorded throughout all conducted experiments to determine the level of temperature increase caused by a single integrated tube.
Figure 16 shows the water temperature difference recorded during the charging and discharging experiment with $V=0.5$ l/min and $T_{\text{initial}}=80^\circ\text{C}$ for the finned and un-finned tubes. Firstly, for the charging experiment, it was observed that the difference between inlet/outlet water temperature decreased to less than 5$^\circ$C after approximately 7 and 13 minutes from starting the experiment for the finned and un-finned tubes, respectively. Moreover, it is clear that the maximum temperature difference ($T_{\text{max}}$) in the finned tube was lower than the $T_{\text{max}}$ obtained in the un-finned one, where the maximum difference between inlet and the outlet water temperature was 43$^\circ$C and 38$^\circ$C, respectively. Similar observations were obtained during the discharging experiment.
where the temperature difference obtained from the finned and un-finned tubes decreased below 5℃ after 8 and 12 minutes, respectively. However, in the discharging experiments, the maximum temperature difference obtained in the finned tube (A) was 33℃, which is slightly higher than the difference obtained in the un-finned tube (B), which is equal to 32℃.

It is clear that the effect of using longitudinal fin with the proposed conventional design is controversial. On one hand, the used fin design increased the max temperature difference between the HTF inlet and outlet temperatures compared to the observed temperature difference in the un-finned tube (B), as shown in Figure 16-a and Figure 16-b. On the other hand, the used fin decreased the effective time available before the difference decreased to lower than the useful limit 5℃, as shown in Figure 16-a. This is mainly because the used fin design creates a short circuit between the inlet and outlet tubes, as shown in Figure 10-b. Thus, the fin design is pinpointed as one of the essential parameters that must be studied and optimized in the new ETC-S system.

4.3.3- Nominal temperature \( T_{nom} \) and Nominal Power \( P_{nom} \)
As explained in section 3.4 the nominal temperature \( T_{nom} \) of any TES module is determined according to the prospected application requirements and standards. Therefore, as the proposed integrated tube is designed to be used as a building block for an ETC-S system, the \( T_{max} \) of this application is selected to equal 45℃ according to the selected standards [24, 25] and previous research [26, 27]. Moreover, the nominal power is calculated by Eq. (19) using the minimum mass flowrate defined by the Chinese...
standard [61] which is equal to \( \dot{m}_{HTF} = 0.02 \text{ kg/s.m}^2 \). Thus, the \( P_{\text{nom}} \) for the proposed integrated tube is equal to 239 W.

4.3.4 - Tube Performance in charging and discharging experiments

a- Energy storage capacity

The theoretical storage capacity of the tested, integrated tubes is calculated by Eq. (20) and presented in Figure 17. The storage capacity increases proportionally with the PCM final charging temperature. The theoretical capacity of both finned and unfinned tubes increased from 274 ± 1.05 kJ to 414 ± 1.15 kJ and from 290 ± 1.05 kJ to 437 ± 1.15 kJ by increasing the final charging temperature from 60°C to 80°C, respectively.

![Figure 17 Theoretical storage capacity for the finned tube (A) and un-finned tube (B) at different \( T_{\text{initial/final}} \) PCM temperatures.](image)

Figure 17 shows the actual energy required to charge or available to be retrieved from the tested tubes, which are calculated by Eq. (21) and Eq. (22). The results showed that the effect of the PCM temperature on the actual energy transferred/retrieved is more dominant than the effect of charging/discharging flowrate.
The actual energy retrieved from the finned tube (A) decreased by 19% from 379 ± 32 kJ to 307 ± 33 kJ as a result of changing the initial temperature from 80°C to 65°C while the same discharging flow rate 0.5 l/min was maintained as shown in Figure 18. Similarly, the same observations were obtained from the discharging experiments of the un-finned tube (B). Changing the initial PCM temperature from 80°C to 65°C decreased the energy recovered from tube (B) using HTF with (\( \dot{V} = 0.5 \) l/min, and \( T_{\text{inlet}} = 25 \pm 3°C \)) by 24% from 400 ± 34 kJ to 303 ±24 kJ.

On the other hand, as shown in Figure 18, the volume flowrate has a limited effect on the energy-charged or retrieved from the tested tubes. For example, for experiments conducted with PCM initial or final temperature \( T_{\text{initial/final}} = 80°C \), it is observed that changing the HTF flowrate from 0.5 l/min to 0.3 l/min barely affected the required energy to reach the fully charged status by 4% and 8% while the recovered energy was affected by 3.7% and 4% for the finned tube (A) and the un-finned tube(B), respectively.
Figure 18 Actual energy charged and recovered from finned tube (A) and un-finned tube (B) under different operation conditions where a) actual charging energy of finned tube (A), b) actual recovered energy of finned tube (A), c) actual charging energy of un-finned tube (B), d) actual recovered energy of un-finned tube (B)

b- . Energy efficiency

Figure 19 shows the change in charging energy efficiency $\eta_{ch}$ of both tubes A and B during different operating conditions. It is obtained that increase the $T_{fina}$ proportionally enhance the charging efficiency $\eta_{ch}$. For example, in experiments 1.1 to 1.4, increasing the $T_{fina}$ temperature from 65℃ to 80℃ enhanced the charging efficiency from 94% to 99% in the finned tube (A) and from 89% to 98% in the un-finned tube (B), as shown in Figure 19-a and Figure 19-d. Moreover, the recovery efficiency $\eta_{rec}$ showed a similar trend where changing the PCM temperature $T_{init}$ from 65℃ to 80℃ enhanced the $\eta_{dls}$ by 5% and 8% for the finned and un-finned
tubes, respectively. As the rest of the experiments showed a similar trend, the details of the results shown in Figure 19 will not be discussed.
Figure 19 Charging, discharging, and Total efficiency for tubes A and B
Also, the effect of changing the HTF flow rate and temperature on the total efficiency $\eta_T$ is shown in Figure 20. The results present that both HTF flowrate and PCM temperature have a proportional relation with the total efficiency. Moreover, it is observed that the effect of operation condition is more significant on the efficiency of the un-finned tube (B) compared to their effect on the performance of the finned tube (A). For example, the increase of HTF flow rate to 0.5 l/min and the PCM temperature to 80°C enhanced the $\eta_T$ of the finned tube (A) by 7% compared to more than 14% in the un-finned tube (B), as illustrated in Figure 20. This is a result of increasing the HTF volume flowrate, which increases the heat transfer rate and decreases the required time to complete the charging and discharging processes. Meanwhile, the reduction in the required time to complete the cycle decreases the un-avoidable heat losses to the environment during the charging and discharging processes and increases the total efficiency of the tube.
Figure 20 effect of HTF flow rate on the total efficiency of tubes A and B

c- Charging/ discharging cycle time

Figure 21(a) and Figure 21(b) show the actual time consumed to complete the charging test for the finned and un-finned tubes. The results showed that the effect of PCM temperature is more dominant than the impact of the HTF flowrate on the ATCT required to complete the charging/discharging cycle. For instance, the results present that, for PCM temperature $T_{initial/ final} = 80^\circ C$, decreasing the HTF charging/discharging flowrate $\dot{V}$ from 0.5 l/min to 0.3 l/min only increased the ATCT by
17% and 20% for the finned tube (A) and the un-finned tube (B), respectively. On the other hand, decreasing the PCM temperature $T_{\text{initial/finale}}$ from 80°C to 65°C minimized the temperature difference between HTF and PCM which hinders the heat transfer rate between the two mediums. Accordingly, the total time required to complete the charging and discharging cycle done with HTF flowrate $\dot{V} = 0.5 \text{ l/min}$ was increased by 26% and 28% for the finned tube (A) and un-finned tube (B), respectively. Besides, to quantify the effect of different $T_{\text{initial/finale}}$ on the cycle time with a normalized value, the MCL concept, as described in section 3.4, is calculated and compared to the actual time as displayed in Figure 21. Moreover, the utilization of aluminum longitudinal fin enhanced the heat transfer rate between the PCM and HTF and consequently decreased the ATCT. As shown in Figure 21-a and Figure 21-b the ATCT required to complete a sequential charging and discharging cycle is decreased by almost 50% by using the proposed fin design. Also, the ATCT in the experiments conducted on the finned tube (A) is closer to the MCL than the ATCT obtained from the experiments conducted on the un-finned tube (B), as presented in Figure 21-a and Figure 21-b.
**d- Charging and discharging effectiveness**

Figure 22 shows the average charging effectiveness ($\bar{\varepsilon}_c$) of both tested tubes A and B during the charging experiment 1.1 with the operating conditions described in Figure 12. The graph clearly shows that the $\bar{\varepsilon}_c$ of both tubes is inversely proportional with the circulated HTF flowrate. Firstly, in charging experiments, the $\bar{\varepsilon}_c$ of the finned tube (A) decreased by 35% as a result of increasing the HTF flowrate from 0.3 l/min to 0.5 l/min. The same conclusion was obtained for the un-finned tube (B), the $\bar{\varepsilon}_c$ decreased from 22% with HTF flowrate equal to 0.3 l/min to 12% when the HTF flow rate increased to 0.5 l/min.

Also, it is clear that the used fin design has a positive effect on the process effectiveness were the $\bar{\varepsilon}_{ch}$ of the finned tube (A) was higher than the $\bar{\varepsilon}_{ch}$ of the un-finned tube (B) in all cases, as shown in Figure 22. This enhancement is obtained because the difference between HTF inlet and outlet temperature during the phase change period is higher in the finned tube (A) compared to the difference obtained in the un-finned tube (B), as previously illustrated in Figure 10.
Figure 22 Average effectiveness during experiment 1.1 for Finned tube (A) and Un-finned tube (B) where (a) charging experiment and (b) discharging experiment

Further, the average discharging effectiveness ($\overline{\varepsilon}_{d\text{is}}$) of both tested tubes under different operation conditions is determined by Eq. (28) and presented in Figure 22b. Similar to the charging effectiveness, it is clear that in both tubes A and B the $\overline{\varepsilon}_{d\text{is}}$ is inversely proportional with the HTF flow rate, as shown in Figure 22b. In details, for tests with $T_{\text{initial}} = 80^\circ\text{C}$, rising the HTF flowrate from 0.3 l/min to 0.5 l/min decreased the $\overline{\varepsilon}_{d}$ of the experiment from 26 % and 21 % to 18% and 11 %for the finned tube (A) and un-finned tube (B), respectively.

4.3.5 Heat loss calculation
Figure 23(a) shows the PCM average temperature profile during the 12h heat loss test. Both tubes A and B were charged to the maximum temperature of 80°C and allowed to naturally cool down for 12 hours and 24 hours. For experiments were done with an ambient temperature $T_a$ equal 21°C, the average temperature of PCM after 12 hours test declined to almost 48°C for both tubes and declined to the ambient temperature after almost 20 hours in the 24 hours test.
Figure 23(b) shows the difference between the initial ESC charged to both tubes and the required energy to recharge the tubes to the initial uniform temperature after the 12 hours heat loss test. Again, both tubes have almost the same performance in the heat loss test. Firstly, for the finned tube (A), the required recharge energy was equal to 270 kJ, which is 66% of the initial ESC. Also, the required energy to recharge the un-finned tube (B) was equal to 293 kJ, which is 68% of the initial ESC of the tube.

4.3.6 Effect of fin integration

The compactness factor of the used aluminum fin is calculated by Eq. (33), and it equals approximately 5%. The integration of the longitudinal fin that demonstrated in Figure 10 decreased the ESC of the finned tube (A) by a maximum of 6%. On the other hand, the used fin design enhanced the heat transfer between the circulated HTF and the PCM during the charging and discharging cycles. The overall efficiency $\eta_T$ of the finned tube (A) was higher than the efficiency of the un-finned tube (B). The maximum $\eta_T$ difference is obtained in experiment 1.4 by approximately 8%. The fin’s enhancement effect decreases in the high-temperature experiments because the natural convection inside the un-finned tube (B) enhances the heat transfer between the PCM and
the circulated HTF. Moreover, the utilization of fin remarkably decreased the cycle time by almost 50% in all experiments compared to the cycle time of the un-finned tube (B). Also, the used fin enhanced the effectiveness by at least 3% and 5% for the charging and discharging cycles. Figure 24 shows a summary of the effect of fin integration on the thermal performance of the tested building block.

Figure 24 The fin effect on the thermal characteristic indicators
CHAPTER FIVE

NUMERICAL WORK
5- NUMERICAL MODEL

5.1. Detailed Numerical Model

5.1.1- Physical model

The ultimate goal of the LHS units designing process is the maximization of heat transfer rate between the HTF and the PCM within the available unit’s volume. The highest heat transfer density is achieved by minimizing thermal flow resistance and increase the heat transfer contact area. The conventional ETC systems use a U-tube pipe to circulate the HTF in counterflow within the system as shown in Figure 25. In this research, aiming to redesign the conventional ETC system, the effect of circulating the HTF as a parallel flow on the thermal performance of the tested building block is investigated using the developed numerical model. Thus, a new design using double parallel pipes is proposed and compared with the thermal performance of the counter flow pipe as shown in Figure 25. The parallel flow design offers the ability to increase the discharged flowrate from a single tube which is essential to minimize the required storage volume. Moreover, the proposed parallel flow pipe is tested using a half flow rate to compare the thermal performance of parallel and counter flow pipe under the same discharging load.
5.1.2 - Governing equations

In this study, the charging and discharging of the proposed ETCS building block is simulated by using the melting/solidification module of ANSYS Fluent 16.1. The ANSYS’s melting/solidification module uses the enthalpy porosity method that is developed by Voler and Prakash [58]. This numerical technique defines the mushy zone as a porous medium to model the material phase change from solid to liquid as porosity changes from 0 to 1. Moreover, the Boussinesq approximation is used to simulate the effect of natural convection due to buoyancy forces inside the tube.

5.1.3 - Boundary and initial conditions details

A 3-D geometry is drawn using CAD software (Solidworks-2018) and used to generate a hexagonal structured mesh using ANSYS-ICEM software as shown in Figure 26.
Figure 26 a) front view of ICEM structured mesh, (b) 3-D view for the grid

The pressure-based solver available in the commercial software ANSYS-Fluent 16.1 is used for solving the previously explained governing equations. Further, the SIMPLE scheme is used for pressure-velocity coupling, and the second-order upwind differencing scheme is used to solve the convective terms in momentum and energy equations. Moreover, the PRESTO scheme is selected for the pressure correction equation. The outer glass is defined as adiabatic walls, the inlet boundary is defined as mass flow inlet while the outlet boundary is defined as a pressure outlet. Table 3 shows the boundary conditions and the operation condition of the validated CFD case.

Table 3 Boundary conditions and case setup details of the validated CFD case.

<table>
<thead>
<tr>
<th></th>
<th>380,510, 270,453 and 525,643 nodes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Grid size</td>
<td></td>
</tr>
<tr>
<td>Inlet flowrate</td>
<td>0.5 l/min</td>
</tr>
<tr>
<td>Average Inlet water temperature</td>
<td>25°C</td>
</tr>
<tr>
<td>Initial PCM temperature</td>
<td>80°C</td>
</tr>
<tr>
<td>Total simulated time</td>
<td>3600 second</td>
</tr>
<tr>
<td>Time step</td>
<td>0.1 second</td>
</tr>
</tbody>
</table>
5.1.4. Model validation

The relative deviation $\varepsilon$ is calculated to validate the developed numerical models and calculate the accuracy of each model in comparison with the results obtained from the conducted experiments. The relative deviation is calculated through Eq. (36) [95].

$$\varepsilon = \frac{|T_{\text{CFD}} - T_{\text{exp}}|}{T_{\text{exp}}} \times 100$$  (36)

5.2. PCM Modeling Using Effective Heat Capacity Methods

The modeling of LHS units using the enthalpy porosity technique is preferred by many commercial CFD software due to its high reliability and stability for problems with small phase change temperature range. However, for the model used in this study the utilization of the available melting and solidification module in ANSYS-Fluent required a small-time step, between 0.1 and 0.2 seconds, to achieve the convergence criteria and obtain results with acceptable accuracy. This small-time step increased the total computational time and cost and limited the ability to use the CFD model for optimization problems and/or general design studies. The studies reviewed in the literature showed that the simulation of LHS units can be simplified in many cases by using the effective heat capacity method (EHC) [96], [97]. The heat capacity method is developed and used in many studies as a simple technique to model the latent heat as a sensible heat using a temperature-dependent specific heat and convert the problem to a simple heat conduction model [98]–[100]. Accordingly, the proposed model in this study is simplified by using the EHC method to compare the results and the computational cost with the commonly
used enthalpy-porosity technique. In this study, the paraffin wax is defined as a sensible medium with specific heat is introduced as a temperature-dependent property using Eq. (37). The specific heat-temperature profile is smoothed to avoid the instability caused by the dramatic change in specific heat due to phase change as shown in Figure 27 (b).

\[
C = \begin{cases} 
C_s + \frac{C_l}{2} 
& \text{for } T < T_f - \Delta T \\
\frac{L}{4\Delta T} 
& \text{for } T_f - 2\Delta T \leq T \leq T_f + 2\Delta T \\
C_l 
& \text{for } T > T_f + \Delta T 
\end{cases}
\] (37)

Where \(C_s\) and \(C_l\) are the PCM specific heat in solid and liquid phases, respectively. \(L\) is the PCM latent heat. \(T_f\) the phase change temperature and \(\Delta T\) is the phase change temperature interval as shown in Figure 27 (a) and (b).

Figure 27 a) PCM specific heat profile b) modified PCM specific heat profile

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5.2.1 Material properties

The performance of the tested building block depends on the properties of the selected PCM. The melting temperature, specific heat, and latent heat are the most important properties that affect the performance of the LHS unit during charging and/or discharging processes. Accordingly, the developed numerical model is used to test the performance of the proposed building block while using four different PCM. The properties of the tested PCMs are shown in Table 4.

<table>
<thead>
<tr>
<th>Property</th>
<th>Melting temperature</th>
<th>Cp solid (kJ/kg K)</th>
<th>Cp liquid (kJ/kg K)</th>
<th>Latent heat (kJ/kg)</th>
<th>Density (kg/m³)</th>
<th>Thermal Conductivity (W/m. K)</th>
<th>Ref</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alex 600</td>
<td>58-62</td>
<td>1.8</td>
<td>2.2</td>
<td>189</td>
<td>0.920</td>
<td>0.21</td>
<td>[101]</td>
</tr>
<tr>
<td>SP58</td>
<td>56-59</td>
<td>2</td>
<td>2</td>
<td>250</td>
<td>1.3</td>
<td>0.6</td>
<td>[102]</td>
</tr>
<tr>
<td>Paraffin 54</td>
<td>52-54</td>
<td>1.67</td>
<td>2.065</td>
<td>251</td>
<td>0.918</td>
<td>0.18</td>
<td>[103]</td>
</tr>
<tr>
<td>C22</td>
<td>58-60</td>
<td>2.14</td>
<td>2.9</td>
<td>185</td>
<td>0.795</td>
<td>0.21</td>
<td>[104]</td>
</tr>
</tbody>
</table>

5.3. Simplified Numerical Model

Several researchers [105]–[108] showed the possibility to simplify the LHS units and propose methodologies to develop numerical models that simulate the charging/discharging cycles based on a simplified energy equation. In this research, the proposed building block is simplified to decrease the computational resources and time required to simulate a complete discharging cycle. Further, this simplified model is essential to develop a general design tool that will be used to design the targeted novel ETCS system.
5.3.1- Geometry and mesh

Based on the results obtained from the CFD simulations using the enthalpy and EHC methods the proposed building block is simplified to a two-dimensional axisymmetric domain as shown in Figure 28. Moreover, the counterflow configuration is simplified through converting the U tube pipe into an elongated straight pipe with double length. The pipe thickness is divided into several small circular cells as shown in Figure 28 (C). Moreover, based on the temperature contours obtained from the CFD simulations, the PCM is divided into several circular segments as shown in Figure 28 (a). A different number of nodes are tested to verify the model grid independency and the selected number of nodes were: 5 in the copper radial direction, 8 in the PCM, 1 for the HTF. Meanwhile, in the y-direction, the domain is discretized into 200 nodes.

Figure 28 Sketch of the modeled system.
5.3.2- Governing equations

The heat transfer of the discharging process is governed by the energy equation as shown in Eq (38–38) [105].

\[
\frac{\partial H}{\partial t} + \vartheta_y \frac{\partial H}{\partial y} = k \left( \frac{\partial^2 T}{\partial X^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (38)
\]

Where \( X \) is cells in radial directions and \( y \) is cells in the \( y \)-direction. Moreover, \( H \) is the total volumetric enthalpy, \( \vartheta_y \) is HTF velocity in the \( y \)-direction, and \( k \) is thermal conductivity. The fully implicit finite volume method is used to solve the governing energy equation through the following initial and boundary conditions [109]:

(i) \( T = 0 \quad T = T_i \)

(ii) \( X = X_0 \quad \frac{\partial T}{\partial x} = 0 \)

(iii) \( X = X_1 \quad T_{HTF} = T_{copper} \)

(iv) \( X = X_n \quad -k_{copper} \frac{\partial T}{\partial x} = h_1(T_n - T_{pcm}) \)

(v) \( Y = 0 \quad T = T_{inlet} \)

(vi) \( Y = L \frac{\partial T}{\partial y} = 0 \)

Further, the governing energy equation is discretized as shown in the figure and described in Eq (39–39) [105].

\[
a \cdot (T_p^{t+1} - T_p) = b \cdot (T_p^{t+1} - T_W^{t+1}) + c \cdot (T_p^{t+1} - T_x^{t+1}) + d \cdot (T_p^{t+1} - T_p^{t+1}) + e \cdot (T_k^{t+1} - T_p^{t+1}) + m \cdot C_p(T_p^{t+1} - T_W^{t+1}) \quad (39)
\]
Where $a$, $b$, $c$, $d$, and $e$ are the heat accumulation coefficients and defined as follows:

$$a = \frac{\rho \cdot C_p \cdot V_p}{\Delta t}$$

Meanwhile, the constants $b$, $c$, $d$, and $e$ are defined based on the heat transfer mode as shown in Table 5:

<table>
<thead>
<tr>
<th>Conduction</th>
<th>$b = k_y S_y / \Delta y$</th>
<th>$c = k_z S_z / \Delta x$</th>
<th>$d = k_n S_n / \Delta x$</th>
<th>$e = k_e S_e / \Delta y$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Convection</td>
<td>$b = h S_w$</td>
<td>$c = h S_x$</td>
<td>$d = h S_n$</td>
<td>$e = h S_e$</td>
</tr>
</tbody>
</table>

### 5.3.3- Assumptions and boundary conditions

Furthermore, several assumptions are considered to minimize the complexity of the problem and maintain an acceptable level of accuracy. These assumptions are backed by information gathered from the literature as follows:

1. **Neglect the water temperature gradient inside the copper pipe and use a single node for water.**
ii- The thermal losses through the outer glass walls are negligible.

iii- The thermophysical properties of the PCM, copper pipe, and the HTF are independent of the temperature.

iv- The PCM is considered homogeneous and isotropic with no supercooling and the buoyancy effects are neglected.

v- The PCM latent heat is modeled as a dynamic specific heat depends on temperature.

The convective heat transfer coefficient (h) is calculated as a function of empirical correlations of Nusselt number which is calculated using the well-known Dittus–Boelter correlation for straight pipes as shown by equations (40):

\[ Nu = 0.023Re^{0.8}Pr^{0.3} \]  \hspace{1cm} (40)

In this research, the results obtained from this simplified model are compared with the results of different CFD modeling techniques to validate the proposed 2-D model. Further, several design parameters, such as copper pipe diameter, glass tube diameter, tube length, and PCM type, are investigated in a parametric analysis using the validated 2-D model.

The parametric analysis is essential to study the effect of design parameters on the thermal performance of the proposed building block. Moreover, the results obtained from this analysis in addition to the validated model will be used to build reliable tools and for designing the targeted novel tankless collector. Therefore, a parametric study is conducted to explore the effect of design parameters on the thermal performance of the building block. These design parameters include copper pipe inner diameter, glass tube...
diameter, and tube length. Several design options are selected based on the available standard parts as shown in Table 6 with all dimensions in mm.

Table 6 Dimensions of design parameters examined in parametric analysis

<table>
<thead>
<tr>
<th>Copper pipe diameter</th>
<th>d1=8</th>
<th>d2=10</th>
<th>d3=12</th>
<th>d4=14</th>
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</thead>
<tbody>
<tr>
<td>Glass diameter</td>
<td>D1=35</td>
<td>D2=47</td>
<td>D3=58</td>
<td>D4=70</td>
</tr>
<tr>
<td>Tube length</td>
<td>L1=700</td>
<td>L2=1200</td>
<td>L3=1500</td>
<td>L4=1800</td>
</tr>
</tbody>
</table>

5.4 Collector Design Methodology

The validated simplified model granted the opportunity to obtain the thermal performance of the building block in less time compared to other modeling techniques. This computational time and cost reduction are essential to design the novel tankless solar collector through modeling different configurations of the novel ETCS system’s building block. In this methodology, the simplified model is used as the core of a general design tool that is developed to propose a general design of the ETCS system based on the available resources, cost, and user needs. The procedure of this general design tool is explained as follows as shown in Figure 30:

Step 1: Define the building block properties

The first step in the design methodology defines the design parameters of the building block, including copper pipe diameter, glass tube diameter, PCM thermophysical properties, and copper pipe configuration (CON), according to the available standard parts, economics, and user preferences.

Step 2: Define the collector parameters

The conventional solar collector consists of a set of series and parallel tubes. Firstly, several tubes connected in series to increase the temperature of the circulated HTF to the required temperature. Secondly, a few parallel sets to increase to HTF flow rate to the required outflow. In this step, the collector design constraints are defined as the
maximum number of series tubes per set (ST), and the maximum number of parallel blocks (PB).

**Step 3: Define the application parameters**

The ETCS system KPIs are defined based on the application requirements. These KPIs include the discharge temperature after a certain time ($T_{set}$), the required flow rate $\dot{V}_{set}$, and the discharging time. Subsequently, the user-defined parameters are used as inputs for the simplified model and the obtained results are compared with these system requirements such as $T_{set}$ and $\dot{V}_{set}$.

**Step 4: Run the optimization algorithm**

The previously defined parameters are used as inputs for the initial case in the optimization algorithm. The algorithm is developed to propose the optimum building block parameters that satisfy the user needs or at least the closest to the ideal design. The optimization algorithm includes the following sub-steps:

**Sub-step 1: Run the simplified model**

In this step, the validated simplified model is used to obtain the water outlet temperature profile.

**Sub-step 2: Optimize parameters**

At the end of the simulation, if the outlet temperature is less than the predefined $T_{set}$ the building block parameters are changed and the results of the new design are compared with $T_{set}$. The optimization algorithm continued until the user needs are achieved or the closest to the ideal design is obtained.

**Step 5: Increase the number of series tubes**

If the optimum single tube design is obtained, the required $T_{set}$ is achieved through increasing the number of series tubes to increase the heat transfer contact area between PCM and circulated HTF which increases the outlet temperature. However, adding units to the set of series tubes increase the HTF pressure drop in the collector which in turn requires a larger pump and higher initial cost to overcome this problem. Therefore, in
some cases, the required $T_{\text{set}}$ should be compromised with the available pump capacity and initial cost to achieve the optimum design.

**Step 6: Increase the number of parallel blocks**

In this step, the set of series tubes is considered as a block unit and the number of PB is determined based on the required flow rate and the maximum allowable collector volume $\text{PB}_{\text{max}}$.

The aforementioned explained methodology of general collector sizing will provide a guideline for engineers and researchers to design novel ETCS systems that are mainly based on evacuated tubes and PCM. This methodology combines between the available on the shelf components and application requirements to develop a general primary design of novel ETCS systems.
In this research, the proposed methodology will be used to design an ETCS system that will serve in domestic water heating applications. As such the required temperature and flow rate are selected based on the literature to be $T_{set}$ equal 45 °C and the $\dot{V}_{set}$ is 4 l/min [110], [111].

5.5- Results and Discussion

5.5.1- Validation and grid independence study

The developed numerical model is compared with the experimental results obtained from a discharging experiment conducted using the previously demonstrated ETCS building block with an initial temperature 80°C and the HTF flow rate 0.5 l/min. Figure 31(a) and (b) show the validation of results obtained from the CFD model with the experimental results. In detail, Figure 31(a) shows an acceptable match between the average PCM temperature obtained from experimental and CFD model results. Moreover, Figure 31(b) shows the validation of the water outlet temperature CFD results with the results obtained from the conducted experiment where the maximum error deviation $\varepsilon$ is 6%.
Figure 31 comparison of CFD results with experimental results a) Average PCM temperature b) water outlet temperature
The model grid independency is verified by repeating the same CFD case with a different grid that has a higher and lower number of nodes, namely, 270,453 and 525,643 nodes. Further, the time step independence is verified by repeating the same case using two-time steps 0.15 second and 0.05 second. The grid independence study showed that there is no significant difference between the results of both grids and both time steps where the maximum difference was less than 5%. Accordingly, the medium grid with 380,510 nodes and the time step 0.1 second is selected as the basic case set-up for the following research simulations.

Furthermore, the figure 32 shows the results of the outlet water temperature obtained from different numerical models developed in this research. The CFD model that based on the EHC method and the simplified 2-D model is validated with the results obtained from the CFD model that used the enthalpy method where the maximum deviation error is equal to 5% and 12% for the EHC method and the simplified model, respectively. Moreover, the computational time is significantly decrease from 2350 min to 6 min as shown in figure 32 (b).
5.5.2- Results of the enthalpy method

a- Temperature and liquid fraction contours

Figure 33 presents the liquid fraction contours of the simulated base case with initial temperature 80 °C and HTF flow rate 0.5 l/min at locations A1: A4. The contours show the propagation of the solid-liquid interface at different time steps starting from initial status to 45 minutes with 15 minutes step. The complete solidification of the tested building block with a U-tube copper pipe is achieved after 36 minutes, and the solidification rate at the lower plane A4 is higher than the upper one A1 as shown in Figure 33. This is mainly because the natural convection forces draw the liquid PCM to the upper section and the solidified PCM sink downward. Moreover, it is noticed that the solid PCM layer around the copper pipes propagates as an almost similar circular segment surrounding both sides of the copper U-tube.

Figure 33 PCM liquid fractions contours at locations A1: A4
Figure 34 presents the temperature contours for the discharging of the counter flow and parallel flow tubes with initial temperature 80 °C and the HTF flow rate 0.5 l/min. The contours show the change of PCM temperature at the top and bottom levels A1 and A4 after 30 minutes interval from 15 to 45 minutes. The comparison of the PCM temperature contours illustrates that the temperature of the PCM center of the counterflow design decreased from 80 ℃ to 64 ℃ after 15 minutes. Meanwhile, the parallel flow design increased the heat discharging rate which decreased the PCM temperature to 58 ℃ after 15 minutes. The observation is noticed from the comparison between the PCM temperature contours after running the discharging simulation for 45 minutes. Also, the temperature contour of the parallel flow design showed that the proposed design inhibited the effect of natural convection where the temperature difference between the PCM centers at the top and bottom levels decreased from 7 ℃ to 3 ℃ in the counter flow and parallel flow design, respectively.

Figure 34 Change in temperature contours of counter and parallel flow designs at locations A1 and A4 and different time steps.

b- Effect of copper pipe configuration
The water outlet temperature and discharging power are the main performance indicators for evaluating and optimize the design of TES units. The main aim of the optimum design is to maintain the water outlet temperature and the discharge power as high as possible at the end of the discharging cycle. Figure 35 shows the effect of copper pipe configuration on the thermal performance parameters of the proposed building block. The parallel flow design allows circulating more HTF in the unit which in turn increases the energy recovery rate and decreases the required time to reach the complete solidification status. Generally, the water outlet temperature of the parallel flow configuration is lower than the counterflow design along the whole energy recovery cycle. Specifically, the time required to reach 25 °C is 10.5 minutes for the parallel flow while the water outlet temperature obtained from the counter flow reaches the same target temperature after only 6.5 minutes. Moreover, the parallel flow is tested using a lower flow rate to maintain the same heat discharging rate and investigate the effect of flow rate on the performance of the parallel flow design. The water temperature profile is shown in Figure 35-(a) shows that splitting the flowrate into two tubes increased the water outlet temperature along the whole discharging cycle compared to the conventional counter flow design with the same flowrate. Further, the outlet temperature reaches the target temperature at 18 and 6.5 minutes for the half flow rate and full flowrate options, respectively.

Furthermore, Figure 35-(b) shows the discharging power calculated from the obtained outlet temperature. The results show that the increase in the circulated HTF attained through using parallel flow configuration enhanced the discharging power by 41%. Compared to the counterflow design. Also, the effect of increasing the water outlet
temperature through using the parallel flow configuration with a half flow rate increased the maximum discharging power by 12 % compared to the maximum power obtained from a conventional counterflow.

![Temperature and Power Graphs](image)

**Figure 35** effect of using different copper pipe configurations a) water outlet temperature profile b) discharging Power of different configurations and flow rates.

### c- Parametric study

**Figure 36** (a) shows the effect of PCM initial temperature on the solidification time of a counter flow pipe simulated using the validated CFD model with discharging flow rate 0.5 l/min and water inlet temperature 22 ℃. Increasing the initial temperature from 70 ℃ to 80 ℃ increased the solidification time by 40% from 26 minutes to 37 minutes, respectively. Additionally, the effect of the circulated flow rate on the
solidification time is investigated using three different flowrates 0.5, 0.4, and 0.3 l/min as illustrated in Figure 36-(b). Subsequently, the proposed tube is modeled using three different flowrates while maintaining the same PCM initial temperature of 80 °C and the HTF inlet temperature 22 °C. The results showed that solidification time decrease by 33 % through increasing the flow rate from 0.3 l/min to 0.5 l/min. Further, the results showed that the solidification time is dominated by the PCM initial temperature more than the HTF flow rate.

Figure 36 effect of PCM initial temperature and HTF flow rate on the solidification time.
5.5.3- Results of effective heat capacity method

The EHC method is used to minimize the computational time required to simulate the solidification of the proposed building block. The simulation of the proposed LHS unit using the EHC method eliminated the need for small-time steps that is necessary to track the propagation of the liquid-solid interface. As such, the proposed domain is simulated as a simple heat conduction model and the time step increased from 0.1 seconds to 5 seconds. The total time required to complete the energy recovery simulation decreased from 2350 minutes to only 72 minutes as shown in Figure 32.

a- Effect of PCM type

Figure 37 shows the thermal performance of the simulated tube using four different PCM as described in Table 4. The ESC calculations showed that the Stearic acid has the highest storage capacity with 438 kJ compared to the other PCM that has 311 kJ, 375 kJ, and 293 kJ for PCMs Alex 600, paraffin 54, and C22, respectively. Moreover, the highest outlet water temperature at the end of the discharging simulation, 23 °C, is obtained using the Stearic acid as shown in Figure 37(a). However, the state of charge profile showed that the low thermal conductivity of the stearic acid inhibited the heat transfer between PCM and HTF which decreased the energy recovery percentage at the end of the simulation. The highest energy recovery percentage is 97.5% for the Alex 600 wax, and the lowest obtained percentage is 72% for the Stearic acid as shown in Figure 37(c).
5.5.4 - Results of the simplified model

The validated model is used to investigate the effect of different design parameters on the thermal performance of the proposed building block as explained in subsection 5.3. To conduct this parametric analysis the simplified model is used to simulate a discharging process with PCM initial temperature 80°C and $V = 0.5 l/min$ results of this parametric analysis will be shown in the following sections.
a- Effect of the copper pipe diameter
The Investigation of the effect of copper pipes diameter on the thermal behavior of the proposed ETCS building block is necessary to give valuable insights for design optimization. The study conducted through modeling different levels of pipe diameters, as shown in Table 6, while maintaining the same glass diameter, 47 mm, and tube length, 1.5 m, and use paraffin wax Alex 600 as PCM in all cases. The operation condition of these cases remained constant with PCM initial temperature 80 °C and HTF inlet temperature and flow rate 22°C and 0.5 l/min, respectively. Generally, as the pipe diameter increase, the rate of heat transfer and the HTF outlet temperature increase because of the large heat transfer area. Figure 38-(a) and Figure 38-(b) show that increasing the pipe diameter proportionally increases the heat transfer rate between PCM and HTF, where at the end of the discharging simulation the highest HTF outlet temperature and discharging power are obtained by using pipe diameter 14 mm. Although the large pipe diameter reduces the PCM volume and minimizes the available ESC, the enhancement achieved in the heat transfer rate overcome these issues, and the highest rate of energy recovery is achieved by using large pipe diameter. Figure 38-(c) shows that the ESC of different pipe design decreased from 663 kJ to 581 kJ by increasing the pipe diameter from 8 to 47, respectively. However, the actual recovered energy using the 47 mm copper pipe is 25% higher than the energy recovered from the tube with the smallest copper pipe, 8mm, as shown in Figure 38-(c). Moreover, Figure 38-(d) shows the benefit of using a large pipe diameter on the SOC where 95% of the energy store in the modeled tube is recovered by the end of the discharging simulation compared to 63% recovered by using the smallest pipe. The same observations are obtained from simulating different copper pipe diameter with the
different levels of the glass tube diameter. As such, to avoid redundancy only the results of one set of simulations are presented in this section.

Figure 38 Effect of copper pipe on thermal performance
a) HTF outlet temperature profile for different glass tube diameters, b) change in discharging power for different glass tube diameters, c) Theoretical and actual energy recovered from different designs, d) SOC change during the discharging process of different glass diameters

b- Effect of Glass diameter
Similarly, the effect of changing the glass tube diameter on the thermal performance of the proposed building block is investigated through modeling four tubes with diameters 35 mm, 47 mm, 58 mm, 70 mm. The simplified numerical model is used to obtain the effect of glass tube diameter on a 1-hour discharging process with initial temperature 80°C and HTF flow rate 0.5 l/min. All these simulations are done using the same copper
pipe diameter, 12 mm, and Alex 600 as PCM. Figure 39 shows the ESC results for different tube sizes, where the maximum ESC, 1782 kJ, is obtained by increasing the available volume for PCM through using the maximum available glass diameter and minimum copper pipe diameter.

However, the results obtained from the numerical models showed that the actual energy recovered from the tubes highly depends on the ratio between the glass tube and copper pipe diameters. Figure 40 shows that the HTF outlet temperature of the tubes with diameters 47 mm, 58 mm, and 70 mm was almost the same. This is mainly because maintaining the same copper pipe diameter minimized the benefit of increasing the storage capacity. Moreover, Figure 40 (a) and Figure 40 (b) show that the tube with 35 mm diameter was the only design that reached complete energy recovered status by the end of the discharge simulation. Moreover, it is clear that increasing the glass diameter increased the storage capacity but at the same time did not benefit the HTF outlet temperature and the SOC during discharging processes.
Figure 39 Theoretical energy storage capacity of different tube sizes

Figure 40 effect of glass tube diameter on thermal performance

(a) HTF outlet temperature profile for different glass tube diameters, (b) SOC profile during the discharge process for different diameters

c- Effect of tube length
A similar procedure is used to investigate the effect of tube length on the HTF outlet temperature. Four levels of tube length 700 mm, 1200 mm, 1500 mm, and 1800 mm are studied in a model that used Alex wax 600 as PCM while the other design parameters
glass tube and copper pipe diameters maintained constant at 70 mm and 14 mm, respectively. The results showed that increasing the tube length enlarges the contact area between HTF and PCM which enhances the heat transfer between both mediums. At the end of the discharging simulation, the HTF outlet temperature increased from 22°C to 25°C as a result of increasing the tube length from 700 mm to 1800 mm as shown in Figure 41.

5.5.5- General Collector Design

In this case study, the authors selected the SDWH system as the reference system to determine the design requirements and constraints. The literature showed that a reliable SDWH supply HTF at temperature 45°C for 6 hot water tapping with 22 liters each and in total 132 l/day. Thus, the main objective of the desired ETCS system is to deliver a continuous feed of hot water at temperature 45 °C and flow rate 2 l/min for a total time of
66 minutes. The simplified model is used in the proposed design methodology, explained in section 2.4, to generate a new collector design. Hence, the building block parameters used for this collector design case are selected based on the insights gained from the previous parametric analysis. It was concluded from the parametric study results that the highest water outlet temperature is achieved through using a 14 mm pipe diameter and the 1800 tube length. Although the Dg has a significant effect on the ESC, the minimum effect on the outlet temperature is recorded. As such, to decrease the collector volume the glass tube with Dg =58 mm is selected for the building block of this case study. Moreover, the Rubitherm-SP58 is selected as the filling PCM to obtain the highest ESC and HTF outlet temperature. The design parameters of the building block used in this case study are shown in Table 7.

Table 7 Summary of initial building block design parameters

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Pipe diameter (mm)</th>
<th>Glass diameter (mm)</th>
<th>Length (mm)</th>
<th>PCM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Value</td>
<td>14</td>
<td>47</td>
<td>1800</td>
<td>SP58</td>
</tr>
</tbody>
</table>

Further, several collector constraints are determined based on information collected from the industry experts as shown in Table 8. Firstly, the maximum number of tubes in the collector is 30 tubes arranged in a set of series and parallel tubes as shown in Figure 42. These tubes are arranged in a set of series tubes with the maximum number of tubes in series order (STmax) is 6 tubes. Meanwhile, these sets of series tubes considered as a block that is connected in parallel with similar blocks with the maximum number of parallel blocks (PBmax) is 4. Finally, the maximum operating temperature of
the selected PCM (Tmax) is 85 °C, and the maximum flow rate per tube is 1 l/min to avoid tube breakage due to thermal shocks.

Table 8 Summary of collector requirements and constraints

<table>
<thead>
<tr>
<th>Constrains</th>
<th>STmax (Tube)</th>
<th>PBmax (Block)</th>
<th>Tmax (°C)</th>
<th>Maximum flow per tube (l/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Value</td>
<td>6</td>
<td>5</td>
<td>85 °C</td>
<td>1</td>
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</tbody>
</table>

Figure 42 General schematic of a solar collector showing the series and parallel connections.

The design algorithm is activated using the above-mentioned inputs and constraints. The results showed that the desired ETCS collector that satisfies the needs of a typical DWH system consists of a set of 5 tubes in series and 4 parallel blocks. The outlet temperature of each tube is shown in Figure 43. The collector consists of four parallel blocks to ensure that the flow rate per tube is below the allowable maximum flow rate and, at the same time, maintain the outlet flow rate at the desired level for the end-user.
Table 9 collector design parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pipe diameter</td>
<td>14 mm</td>
</tr>
<tr>
<td>Glass diameter</td>
<td>58 mm</td>
</tr>
<tr>
<td>Length</td>
<td>1800 mm</td>
</tr>
<tr>
<td>PCM</td>
<td>Rubitherm-SP58</td>
</tr>
<tr>
<td>Number of Tubes</td>
<td>20</td>
</tr>
<tr>
<td>ST</td>
<td>5 tubes</td>
</tr>
<tr>
<td>PB</td>
<td>4 blocks</td>
</tr>
</tbody>
</table>

The results showed that the proposed collector design successfully satisfied the DWH requirements. The collector supplied hot water at a temperature higher than 45°C with flow rate 4 l/min for more than 35 minutes as shown in Figure 43. Also, it is important to note that the water outlet temperature remained higher than the required temperature through all the discharging simulations. As such, there is a remarkable
potential to use the smart control tools to adapt the HTF flow rate and maintain the outlet temperature at 45 °C which extend the serving time.
CHAPTER SIX

CONCLUSION AND FUTURE WORK
6- CONCLUSION AND FUTURE WORK
In the first phase of this ETC-S research project, a conventional ETC system was compared with a novel ETC-S system [22, 23]. Based on the promising results of this previous research, a single evacuated tube integrated with PCM material is considered as the building block of the new ETC-S system. Thus, it was necessary to characterize the thermal performance of this building block as a prologue for the redesigning and optimization of the conventional evacuated tubes collectors to be used as an ETC-S system in the future studies. In this study, a set of experiments is conducted to generate a full thermal characterization for the proposed building block. This characterization includes the temperature profiles of PCM and circulated water, a detailed calculation, and discussion of the thermal performance parameters, results of passive heat loss test, and analysis for the effect of the used longitudinal fin on the thermal performance of the tube. Also, The proposed building block is modeled through different numerical techniques that address different levels of details. The developed numerical models are validated with experimental results and compared against each other in terms of accuracy and computational time. Moreover, the developed models are used to investigate the effect of design parameters and operating conditions on the thermal performance of the proposed building block. These investigations include pipe configurations, PCM material, and tube dimensions. Also, the study includes analysis for the effect of initial temperature and HTF flow rate on the PCM solidification. A summary of the key findings of the thermal characterization of the ETC-S system is the following:

- For the temperature profiles, the results showed that the discharging time is higher than the charging time in all conducted experiments. For example, in the experiment with $\dot{V} = 0.5 \text{ l/min and } T_{\text{Initial}} = 80^\circ\text{C}$, the discharging time was higher by 50% and 25% for the
finned and un-finned tubes, respectively. Moreover, it was obtained that the natural convection was the dominant heat transfer mode during 65% of the charging process time.

- For the thermal performance parameters, both tubes are compared in terms of $ESC$, energy efficiency, cycle time, and effectiveness during charging and discharging cycles. Moreover, the effect of introducing longitudinal fin to enhance the heat transfer rate is studied on all previously mentioned thermal aspects, and they are compared to the results of the plain tube. To conclude, the $ESC$ of the proposed building block increases proportionally with the PCM initial/final temperature. The maximum $ESC$ was achieved at charging temperature 80 °C where the maximum $ESC$ was 414 kJ and 437 kJ for both finned and un-finned tubes, respectively. Moreover, the charging, discharging, and total energy efficiency is enhanced by increasing both the PCM final/initial temperature and the HTF flow rate. The $\eta_T$ of the finned tube (A) increased by 5% while the $\eta_T$ of the un-finned tube (B) increased by 9% as a result of increasing the PCM temperature by 25°C.

- The effects of the HTF flow rate on the effectiveness of the tube during the charging and discharging process is studied for both tubes. The results showed that the effectiveness of the tube during the charging/discharging cycles is inversely proportional to the HTF flow rate. It was obtained that increasing the HTF flow rate from 0.3 l/min to 0.5 l/min caused a remarkable drop in the $\bar{\varepsilon}_c$ and $\bar{\varepsilon}_d$ for both finned and un-finned tubes by 35%, 45%, and 45%, 51%, respectively.

- The analysis of the conducted heat loss tests showed that almost the same heat retention performance is obtained from both tubes where the finned and un-finned tubes lost 66% and 68% of their initial stored energy after 12 hours test, respectively.

- The analysis of the effect of using the longitudinal aluminum fin showed that the used fin homogenized the PCM temperature profile during the charging tests and decreased the
required time for completing the charging/discharging cycles by 50%. Moreover, it was obtained that the used fin design enhanced the thermal characteristics of the tube in all aspects, excluding the ESC due to the volume occupied by the fin.

- The results obtained from the CFD model using the enthalpy method showed a perfect match with the results of the experimental work.
- The EHC method could be used to model the melting/solidification of PCM and minimize the computational time and cost. The results of the EHC method is validated with the experimental results and the computational time is reduced by more than 300%.
- The EHC model is used to compare the thermal performance of the unit using different types of PCM. The results showed that Rubitherm-SP58 is the best option among the tested PCMs.
- The high thermal conductivity of SP58 enhanced the heat transfer rate and a 100% energy recovery is achieved. Moreover, the ESC of the SP59 is 505 kJ, the highest among the tested PCMs.
- A simplified 2-D model based on the EHC method is developed and validated with both the CFD and experimental results. The validated simplified model is essential for optimization and general design stages.
- The parametric analysis conducted using the simplified model showed that the highest outlet temperature is obtained by increasing the copper pipe diameter and tube length.
- The collector design methodology is used to propose a collector design that stratifies the needs of a typical DWH system.
- The optimum building block design is d=14 mm, Dg=47 mm, L=1800 mm, and use SP58 as PCM.
- The proposed collector consists of four series tubes per set and four parallel blocks.
In conclusion, this study showed that the proposed technique of integrating the PCM directly to the vacuum tubes is a promising technique to design a compact ETC-S collector. The heat retention test of the tested building block showed that the proposed collector could be used as a collector and storage because it was able to maintain the PCM temperature at a high level for more than 12 hours. However, the charging and discharging experiments showed that the level of HTF temperature increase within a single tube is limited and stability of hot water supply is questionable. Accordingly, a redesign effort should be conducted to achieve the required levels of temperature increase and stability from this compact collector.

Generally, the methodology of this study could be used in future research to provide a complete thermal characterization of thermal storage modules and the building blocks of TES systems. Subsequently, it would provide useful guidelines for the researchers during the testing phase of the novel TES units and enhance the deployment of these units in renewable energy systems. Moreover, this study showed that numerical modeling techniques could be used to design and test novel ETCS systems. The modeling technique is selected based on the required level of details. Further, the developed simplified model in this study showed a remarkable potential to be used in future research that tackles the design optimization problems and the integration of smart control tools.
References


[40] P. Zhang, F. Ma, and X. Xiao, “Thermal energy storage and retrieval characteristics of a molten-salt latent heat thermal energy storage system,” Appl.


Appendix 1 Calibration
Calibration of Finned tube thermocouples

A1 - PCM Thermocouple calibration

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<tr>
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<tr>
<td>Cold</td>
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<td>25.96</td>
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<tr>
<td>Hot</td>
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<td>25.68</td>
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\[
y = 0.947x + 0.337 \\
R^2 = 0.9998
\]

A2 - PCM Thermocouple calibration

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<td>25.96</td>
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<td>Hot</td>
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A3- PCM Thermocouple calibration

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R^2 = 0.9998
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A4- PCM Thermocouple calibration

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\[
y = 0.9496x + 0.264 \\
R^2 = 0.9998
\]
### B1 - PCM Thermocouple calibration

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\[ y = 0.9522x + 0.2572 \]
\[ R^2 = 0.9998 \]

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<td>23.98614</td>
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<tr>
<td>Cold</td>
<td>0.5</td>
<td>25.96</td>
<td>0.196307</td>
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<tr>
<td>Hot</td>
<td>104</td>
<td>25.68</td>
<td>98.72459</td>
</tr>
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</table>

\[ y = 0.9438x + 0.776 \]
\[ R^2 = 0.9999 \]
**B3- PCM Thermocouple calibration**

<table>
<thead>
<tr>
<th>Ref</th>
<th>Tamb</th>
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<tbody>
<tr>
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</tr>
<tr>
<td>Hot</td>
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<td>25.68</td>
</tr>
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</table>

**B4- PCM Thermocouple calibration**

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<tr>
<td>Cold</td>
<td>0.5</td>
<td>25.96</td>
</tr>
<tr>
<td>Hot</td>
<td>104</td>
<td>25.68</td>
</tr>
</tbody>
</table>
$y = 0.9473x + 0.2753$

$R^2 = 0.9998$
Appendix 2 - Calculation sample

Energy storage capacity (ESC):

the energy storage capacity is calculated by

\[ \text{ESC} = \sum_{i=1}^{n} M_i C_i \Delta T + M_p H_L \]

For example:

\[ \text{ESC} = \sum (M_{pcm} C_{pcm} \Delta T + M_{copper} C_{copper} \Delta T + M_{water} C_{water} \Delta T) + M_p H_L \]

- Copper = 34.9 kJ
- Water = 127 kJ
- PCM = 275 kJ
- Total = 437 kJ

Actual energy delivered/discharged

\[ E_{ch} = \rho_{HTF} \dot{V} C_{HTF} \int_{0}^{t_{ch}} [T_{in} - T_{out}] dt \]

- \( \rho_{HTF} = 995 \text{ kg/m}^3 \)
- \( \dot{V} = 0.5 \text{ l/min} \)
- \( C_{HTF} = 4.18 \text{ kJ/kg K} \)

Effectiveness

\[ \varepsilon = \frac{T_{in} - T_{out}}{T_{in} - T_m} \]

- \( T_m = 62 ^\circ \text{C} \)
- Tin = inlet temperature during melting period
- Tout = outlet temperature during melting period

<table>
<thead>
<tr>
<th></th>
<th>80</th>
<th>75</th>
<th>70</th>
<th>65</th>
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</thead>
<tbody>
<tr>
<td>0.5</td>
<td>10%</td>
<td>10%</td>
<td>10%</td>
<td>10%</td>
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<tr>
<td>0.4</td>
<td>12%</td>
<td>13%</td>
<td>18%</td>
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<td>21%</td>
<td>21%</td>
<td>26%</td>
<td>26%</td>
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</tbody>
</table>
**Minimum cycle length**

The minum cycle time is calculated by

\[ MCL = \frac{ESC}{P_{\text{nom.ch}}} + \frac{ESC}{P_{\text{nom.dis}}} \]

\[ MCL = \frac{437}{0.239} + \frac{437}{0.239} = 60.1 \text{ minutes} \]