Exploration of a plate heat exchanger utilising a phase change material: an experimental and computational study

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Abstract. In this article, the heat transfer mechanism of a phase change material (PCM) chamber during charging and discharging processes was studied both numerically and experimentally. A flat plate thermal storage system with dimensions of 500 x 100 x 12 mm, and a PCM thickness of 12 mm was thus investigated. The thermal storage material selected was commercial paraffin wax with a melting temperature of about 60 °C. A diffusion-advection numerical model was used to study heat transfer in this system, as it was observed that the experimental results were in good agreement with this model. In this way, natural convection was found to be the dominant phenomenon in the model. The effects of heat transfer fluid (HTF) temperature and volume flow rate on the melting and solidification time were also assessed, producing the finding that, by elevating the inlet HTF temperature from $T_h = 343$ K to 348 K and then 353 K, the charging time can be reduced by up to 35%.

1. Introduction

The constant pursuit of new energy sources has led scientists to explore new ground in recent decades, yet while extracting energy from renewable sources is becoming increasingly popular, storing the energy gathered in this way, whether for balancing purposes or to accumulate energy for use in peak hours remains under development [1]. Due to the large amounts of latent heat required for phase change processes, the use of phase change materials (PCMs) such as paraffin wax as a means of storing energy has thus attracted worldwide attention.

Various PCMs have been examined for energy storage purposes, and such materials can be generally divided into three main groups: inorganic materials such as nitrate salts, eutectic mixtures of different components, and organic materials such as paraffin and fatty acids [2]. Several articles have discussed various materials and their applications in thermal storage systems [2–8], and Regin et al. [9] reviewed PCM thermal storage systems from various aspects studying both different materials and encapsulation methods, and presenting a detailed approach for designing a packed bed PCM storage system.

Fan and Khodaddi [10] reviewed previous papers examining different techniques to promote the thermal conductivity of PCMs, while Sharma et al. [2] studied several possible energy storage methods, discussing the properties of PCMs in detail and presenting a classification for them, including reporting on their applications. They also discussed the different numerical methods used to develop PCM models.

Verma et al. [11] similarly studied two types of mathematical models used in PCM energy storage system problems, which were divided based on their use of the first or the second laws of thermodynamics.
Other studies have focused on the heat transfer rates and mechanisms during melting/solidification processes in rectangular containers. Dhaidan and Khodadadi [12] reviewed several such articles focusing on melting in different enclosures. The melting process in a rectangular test configuration was first studied by Hale and Viskanta [13], who showed that natural convection was the dominant mode of heat transfer rather than conduction. Bareiss and Brew [14] experimentally studied rectangular enclosures with constant temperature boundaries. They reported that while only conduction heat transfer exists at the outset of the melting process, as time passes, the effect of natural convection becomes more significant. They also proposed a correlation for predicting the Nusselt number during the charging (melting) process. Bénard et al. [15] studied melting in a similar configuration, finding the local heat transfer rate at a given height in a vertical wall is constant. They also calculated the liquid fraction during this process using an analytical solution. Another study utilised scaling theory to examine natural convection during melting, which also permitted correlations for the Nusselt number and the melting front to be obtained. This proved that, during the melting process, the Nusselt number is proportional to $Ra^{-1/4}$ [16]. Another intensive examination considering a rectangular PCM container used to store energy for peak hours was done by Farid and Husian [17], who proposed an empirical formula to reduce computational errors in predicting thermal conductivity in one-dimensional numerical models, while Shatikian et al. [18, 19] imposed different boundary conditions on a PCM-based heat sink of various dimensions with internal fins, presenting the results using dimensionless parameters. The use of PCM energy storing systems has become a popular topic among researchers due to the evolution of renewable energies, and some papers have thus focused on utilising such materials in heat exchanger configurations. Liu et al. [20] designed an experimental rig to study a thermal storage system using stearic acid as a PCM: they enhanced the overall thermal performance by introducing new types of fins, and the experimental results showed that such fins could greatly enhance the heat transfer rate. Khan and Khan [21] investigated the natural convection of paraffin wax inside a finned shell and tube heat exchanger during the discharge process. Their findings showed that the mean discharge power is enhanced by decreasing the inlet temperature. Likewise, with any increase in volume flow rate, the total discharge time for an equivalent amount of PCM is reduced. Seitz et al. [22] further addressed the use of a thermal storage system within a direct steam generating solar thermal power plant with parabolic troughs from an economic point of view, while Amagour et al. [23] experimentally studied natural convection effects in a compact finned tube heat exchanger. The impact of volume flow rate on the melting and solidification processes was thus considered, showing that elevated flow rates shortened melting time. Youssef et al. [24] experimentally and numerically investigated spiral-wired tubes during the melting and solidification processes, finding evidence that increasing the HTF flow rate from 0.2 to 1 l/min resulted in a reduction in the total time required for melting and solidification processes by factors of 2.5 and 4, respectively.

To the best of the authors’ knowledge, flat plate heat exchanger thermal storage systems have not yet been sufficiently studied; in particular, further research is needed to increase the thermal efficiency of such systems while reducing or maintaining costs. In this study, these issues are addressed using both experimental and numerical approaches, with paraffin wax as the PCM, drawing attention to the influence of HTF flow rate and inlet temperature on this problem.

2. Experimental configuration and procedure

2.1 Heat storage system

The system consisted of a modified flat plate heat exchanger constructed specifically for energy storage purposes. The storage unit, as displayed in Figure 1, was built in the engineering workshops of the Department of Mechanical Engineering at Kerbala University, and consisted of two PCM chambers, each of which was 500 mm high and 12 mm thick, made of 2.0 mm 304 stainless steel. The HTF chamber was 600 mm high and 10 mm wide. Each PCM chamber was equipped with drainage for disassembling purposes, and the thermal energy storage unit was connected to a heating/cooling system, the TD36a model from Didacta Italia, shown in Figure 2 alongside the storage unit and measurement instruments.
The maximum flow rate of this system is 45 l/min. During the experiments, the flow rate and temperature of the water were measured using a rotameter and two thermometers (one at the top and the other at the bottom of the storing unit), respectively. The temperature of the PCM chamber was measured at 12 positions, as depicted in Figure 3.

Figure 1 Experimental storage unit

Figure 2 Integrated storage system: 1. TFA-1048 thermometer for measuring the inlet HTF temperature; 2. HTF inlet port; 3. Flat plate storage unit; 4. Data logger; 5. Laptop; 6. Heating/Cooling system thermostat; 7. PCM chamber drainage; 8. HTF outlet; 9. Hot and cold tanks; 10. High temperature and low temperature liquid flow meter; 11. Contactor; 12. Power outlet; 13. Pressure gauge; 14. Circulating pumps, each of which is of 0.75 H.P.
2.2 Phase change material

Paraffin wax was used as the PCM in this study. It has a melting point of 60 °C, and the other PCM and the HTF properties are provided in table 1.

| Table 1 Thermophysical properties of PCM and HTF. |
|-----------------------------------------------|
| PCM Descriptions                  | Unit      | Value     |
| Solid state density               | kg. m\(^{-3}\) | 850       |
| Liquid state density              | kg. m\(^{-3}\) | 825       |
| Latent heat                       | J. kg\(^{-1}\) | 141340    |
| Melting range                     | K         | 328-333   |
| Specific heat                     | J.kg\(^{-1}\). K\(^{-1}\) | 5625.88   |
| Thermal conductivity in solidus   | W.m\(^{1}\)K\(^{-1}\) | 0.43      |
| Volumetric expansion              | 1/K       | 0.0005    |
| Viscosity                         | kg.m\(^{1}\). s\(^{-1}\) | 0.00733   |
| HTF Descriptions                  | Unit      | Value     |
| Inlet temperature                 | K         | 343, 348, 353 |
| Density                           | kg. m\(^{-3}\) | 998.2     |
| Flow rate                         | Lit. min\(^{-1}\) | 15, 30, 45 |
| Specific heat                     | J.kg\(^{-1}\). K\(^{-1}\) | 4180      |
| Thermal conductivity              | W.m\(^{1}\)K\(^{-1}\) | 0.58      |

2.3 Thermal instantaneous and cumulative energy

The thermal power \((q)\) and energy given or gained by water \((Q_{\text{ch & dis}})\) during the charging and discharging processes can be calculated as follows:

\[
q_{\text{ch}} = \dot{m}C_p(T_{\text{in}} - T_{\text{out}}) \tag{1}
\]

\[
q_{\text{dis}} = \dot{m}C_p(T_{\text{out}} - T_{\text{in}}) \tag{2}
\]

\[
Q_{\text{ch & dis}} = \sum q_{\text{ch & dis}}\Delta t \tag{3}
\]
where $m$ denotes the mass flow rate, $C_p$ is the specific heat capacity, and $T_{in}$ and $T_{out}$ are the inlet and outlet temperatures of the HTF, respectively.

2.4 Uncertainty analysis

To examine the reliability of the experimental results, a consideration of uncertainty ranges was required. To calculate this range, correlation (4) was used, where $i$ is the number of independent variables, and $R$ is a linear function. In this approach, it is assumed that each variable is distributed with a standard deviation $\sigma_i$; thus, the standard deviation for $R$ can be estimated as [26]

$$\sigma_R = \left[ \left( \frac{\partial R}{\partial \sigma_1} \right)^2 \sigma_1^2 + \left( \frac{\partial R}{\partial \sigma_2} \right)^2 \sigma_2^2 + \cdots + \left( \frac{\partial R}{\partial \sigma_i} \right)^2 \sigma_i^2 \right]^{0.5}$$ (4)

The uncertainty related to each instrument is presented in Table 2; in this study, the maximum calculated uncertainty was 5% for all cases.

### Table 2 Instruments’ accuracy and maximum uncertainty

| Equipment                    | Measurement section | Accuracy         | Maximum uncertainty |
|------------------------------|---------------------|------------------|---------------------|
| TFA1048 thermometer          | Temperature of HTF  | ±0.15 to ±0.25 °C| 0.25 °C             |
| K-type thermocouple          | Temperature of PCM  | ±0.5 °C          | 0.15 °C             |
| BTM-4208SD data logger       | Temperature recorder| ± (0.4%+0.5 °C)  | 0.2 °C              |

2.5 Experimental procedure

In all cases, before the initiation of the melting process, the PCM was at equilibrium with its surroundings, which were maintained at 292 K. During the process, the water received heat from three heaters (3 kW each), and its inlet temperature was maintained at the pre-set levels (343, 348, or 353 K) with the help of a manual thermostat. In addition, the HTF temperature at the inlet and outlet of the storing unit was recorded using two TFA thermometers. The HTF was circulated in the heating loop by means of a centrifugal pump with 0.75 HP, and its flow rate was set (15, 30, or 45 l/min) by a gate valve. When the paraffin reached thermal equilibrium with the HTF, the heating loop was switched off, ending the data acquisition process.

In the discharge process, the water was cooled to the pre-set temperature (293, 298, or 303 K) by application of the low temperature circuit, which consisted of a cold-water tank, a refrigeration unit, a circulating pump, a rotameter, and a cold temperature cut out. When all the paraffin reached the inlet temperature of the HTF, the experiment was considered complete, and the low temperature circuit was switched off. The data logger used in this experiment recorded the temperature every minute, and each experiment was repeated in triplicate. All experimental data presented in this research is thus a mean value of the relevant three tests.

3. Numerical model

A numerical model was used to model the thermal energy storage system utilising the PCM. The problem was assumed to be 2D based on neglecting the impacts of the third dimension (Z-direction). Moreover, as the symmetry in the storage system allowed for further simplifying assumptions, only half of the complete system was modelled. The final numerical domain and its boundary conditions are depicted in Figure 4, which shows two main zones, one for the HTF flow and the other for the PCM chamber. The upper and lower boundaries of the PCM chamber were assumed to be adiabatic, as those ends were carefully insulated.
3.1 Governing equations

The coupled physical problem required solutions to the mass conservation and Navier-Stokes equations, as well as the energy equation, as modified by appropriate treatment to allow phase changing models to predict heat transfer, fluid flow, and phase change characteristics in the flat plate heat storage system. The Boussinesq approximation was employed to take into account temperature-induced density variations, which affected the buoyancy term in the Y momentum equation, while to simulate the phase alterations during the processes of melting and solidification, the enthalpy-porosity technique, presented by Voller and Prakash [27, 28], was used. This technique permits the user to simulate two-phase problems using a single set of conservation equations, which requires defining a new parameter called the fluid stage fraction. A list of assumptions made for this model is offered below:

a- The depth of the plate heat exchanger is assumed to approach infinity in the Z-direction,
b- The PCM in its liquid form is incompressible and Newtonian,
c- The flow regime in the liquidus PCM is laminar,
d- Viscous dissipation terms and radiation are negligible,
e- No heat losses to the environment, and
f- The HTF flow is turbulent, incompressible, and two dimensional.

Consequently, the governing equations for the PCM can be expressed as

Continuity equation [28]:

\[ \nabla \vec{V} = 0 \]  \hspace{1cm} (5)

Momentum equation for the PCM chamber [28]:

\[
\frac{\partial \vec{V}}{\partial t} + \vec{V} \cdot \nabla \vec{V} = -\nabla P + \frac{1}{\rho} \left( -\nabla \beta \right) + \rho \frac{g}{\rho} \beta (T - T_{ref}) + \vec{S}
\]  \hspace{1cm} (6)

where \( \vec{S} \) is the Darcy's law damping term (as the source term) in the fluid region; this term is included in the equation to simulate phase change effects in the convective heat transfer, and is defined as [27]
\[ S = \frac{(1-\lambda)^2}{\lambda^3} A_{mush} \bar{V} \]  

(7)

In this relationship, \( A_{mush} \) is the mushy zone constant that lies between \( 10^4 \) and \( 10^7 \). In this study, it is assumed to be \( 10^5 \) [27].

Energy equation [28]:

\[
\frac{\partial h}{\partial t} + \nabla \cdot (Vh) = \nabla \cdot \left( \frac{k}{\rho C_p} \nabla h \right) \tag{8}
\]

The total enthalpy for the material is calculated as the sum of the sensible enthalpy, \( h \), and the latent heat, \( \Delta H \) [29]:

\[ H = h + \Delta H \tag{9} \]

where

\[ h = h_{ref} + \int_{T_{ref}}^{T} C_p \, dT \tag{10} \]

and

\[ \Delta H = \lambda L \tag{11} \]

where \( \lambda \) is a liquid fraction. \( \Delta H \) varies from zero for solid state to \( L \) for liquid. Thus, \( \lambda \) is defined as [28]

\[
\lambda = \begin{cases} 
\frac{\Delta H}{L} = 0 & \text{if } T < T_s \\
\frac{\Delta H}{L} = 1 & \text{if } T > T_{liq} \\
\frac{\Delta H}{L} = \frac{T-T_s}{T_{liq}-T_s} & \text{if } T_s < T < T_{liq}
\end{cases} \tag{12}
\]

The under-relaxation factors for PCM liquid fraction, pressure, momentum, and energy were thus found to be 0.9, 0.3, 0.7, and 1, respectively; as the HTF is assumed to be incompressible, turbulent, and unsteady, the (k-\( \varepsilon \)) model was employed to solve the water domain.

### 3.2 Numerical procedure and Validation

The SIMPLE algorithm [29] was used to solve the governing equations, with a second-order upwind method employed to calculate fluxes through cell faces for the momentum and energy equations; the PRESTO scheme was used for the pressure correction equation. The liquid mass fraction was updated utilising Eq. (11) at each time step, and three different meshes, with 85,000, 169,680, and 340,000 cells were used alongside three different time steps (0.05, 0.1, and 0.2 seconds). The cell size and time step independencies were examined carefully for the full melting process and the same method was also used to determine the best mesh and time step for the solidification process. Tables 3 and 4 display mesh and time step independence for the charging and processes. The best choices of mesh and time step were found to be 169,680 and 0.1, respectively. Convergence was tested at each time step, with the convergence criterion of \( 10^{-7} \) for all variables.

### Table 3. Cell number verification for accurate results for thermal storage during the charging process.

| PCM thickness | Cell numbers |
|---------------|--------------|
| 12mm          | 85000        |
|               | 169680       |
|               | 340000       |

### Table 4. Time step verification for accurate results for thermal storage during the charging process.

| PCM thickness | Time step |
|---------------|-----------|
| 12mm          | 0.05      |
|               | 0.1       |
|               | 0.2       |
Figure 5 (a-b) and Figure 6 (a-b) show the temperature profile in the PCM versus time for the numerical results and the experimental temperatures for the melting process in different locations. Relatively good agreement can be observed between the predicted and experimental outcomes.

Figure 5 Comparison of experimental and numerical temperature variation versus time at position No. 5 (a) charging process and (b) discharging process.

(a) Charging process.  
(b) Discharging process.

4. Results and discussion

The thermal response (average temperature) of the PCM during charging at the inlet HTF temperatures of $T_h = 343$ K, 348 K, and 353 K and during the discharging process at the inlet temperatures of $T_c = 293$ K, 298 K, and 303 K was assessed at a constant flow rate of 45 l/min (Fig. 7). The overall average temperature in the system was assessed using 12 thermocouples (Figure 3). The temperature profiles in Fig. 7a clarify the various states of heating, including solid, sensible heating; phase change; and sensible liquid heating. The final melting temperatures and upward trend of the average temperature demonstrate that when HTF temperature increases, the heat transfer rate also increases proportionally. The temperature profiles in Fig. 7b clarify the various states of cooling, including liquid sensible cooling; phase change; and solid, sensible cooling. The reason for this proportionality is the clear physical principle that temperature difference is the impetus for every heat transfer process.
Hence, elevated inlet HTF temperature curtails the PCM charging time, while diminished inlet HTF temperature leads to shorter discharge times. Figure 8 shows the times of complete charging and discharging of the PCM in terms of inlet HTF temperature, indicating that the total charging time and the inlet water temperature are negatively correlated such that the total charging time is shortened both as the inlet water temperature is raised from 343 K to 348 K and from 348 K to 353 K. The opposite applies to the solidification process. At the beginning of the discharge process, the expected sharp fall in PCM temperature occurs based on the fact that the driving potential (temperature difference) between the PCM and the cooling water is larger. The heat transfer rate is then decreased based on the less intense temperature gradients that follow this sharp temperature drop.

![Figure 7. Comparison of average temperature profile in the phase change material at various inlet heat transfer fluid temperatures](image)

![Figure 8. The effect of inlet heat transfer fluid temperature on complete charging and discharging time](image)

Figure 9 shows the liquid fraction contours for the paraffin wax at various times with different inlet temperatures; examining this figure leads to the conclusions below:

In the liquid fraction contours, paraffin melting is parallel to both heat flow and the plate walls in the vertical direction (Figure 9a). As the paraffin wax grows hotter, natural convection appears in the charging phase (Figure 9b). Finally, all the paraffin wax is melted (Figure 9d). The charging process thus begins from the top and extends to the bottom part of the PCM chamber, causing the liquid fraction profile to be inclined. In the bottom part of the thermal storage system, the solid phase changes to liquid.
phase very slowly. The times required for the charging process were about 55.56 minutes, 42.98 minutes, and 35.986 minutes for HTF inlet temperatures of 343 K, 348 K, and 353 K, respectively.

![Figure 9](image)

**Figure 9.** The phase interface position at various moments for various inlet heat transfer fluid temperatures (charging process): (a) 3 min (b) 10 min (c) 30 min, (d) 55.56 min (343 K), 42.98 min (348 K), and 35.986 min (353 K)

Figure 10 shows the variation of liquid fraction contours for paraffin over time for different HTF inlet temperatures during the discharge process. The paraffin solidifies at the lower part of the thermal storage system much more rapidly than in the upper part most likely be due to the fact that as the paraffin adjacent to the HTF plate begins to solidify, it forms a solid layer of paraffin near the wall, increasing the thermal resistance between the liquid paraffin and the cold HTF. As the thickness of the solid paraffin layer increases, the heat transfer rate decreases. Due to the lack of liquid paraffin in the solid layer, conduction dominates the heat transfer from HTF to the liquid paraffin. Thus, buoyancy forces cause the hot liquid paraffin to flow upward.

![Figure 10](image)
Fig. 10. The phase interface position at different times for various inlet heat transfer fluid temperatures (discharging process): (a) 3 min (b) 10 min (c) 20 min and (d) 22.95 min (293 K), 25.25 min (298 K), 28.26 (303 K)

As observed in Figure 11, at the beginning of the charging and discharging processes, the rate of heat transfer is at its highest when the driving temperature gradient between the water and PCM is largest. The thermal power gradually decreases as the PCM temperature approximates the water temperature (Figure 11a).

Figure 11b shows the energy of the plate thermal energy storage for paraffin with a flow rate of 45 l/min for HTF temperatures of 343, 348, and 353 K during the charging process and 293, 298, and 303 K during the discharging process. Initially, the thermal energy is stored in the form of sensible thermal energy only, while once the paraffin reaches the phase change temperature, the thermal energy becomes latent thermal energy.
5. Conclusion

According to the results of this experimental and computational study of charging and discharging inside a plate heat exchanger,

1. Initially, conduction causes heat transfer from hot water to the paraffin wax; as melting starts, natural convection plays a role in the transfer of this heat from the melting zone to the solid zone.

2. Melting starts from the upper part of the enclosure, whereas solidification begins in the lower part of the plate heat exchanger for downward HTF flow.

3. The rate of heat transfer and total charge and discharge time are mainly associated with the HTF inlet temperature.

4. This work showed that elevated inlet HTF temperature reduces charging time by up to 35% when the inlet temperature of HTF increases from 343 K to 353 K. However, the discharging time decreased by up to 19% when the HTF inlet temperature was reduced from 303 K to 293 K.

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