

Thermal hysteresis analysis of finned-heat-pipe-assisted latent heat thermal energy storage application for solar water heater system

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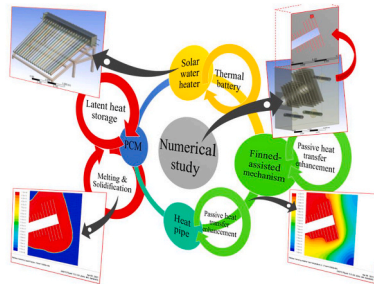
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HIGHLIGHTS

- A novel finned-heat-pipe-assisted solar water heater was numerically investigated.
- The effect of three PCMs and two climate conditions on heat storage was evaluated.
- Temperature distribution and liquid fraction inside the latent heat storage unit were simulated.

GRAPHICAL ABSTRACT



ARTICLE INFO

Keywords:

Solar water heater
Phase change material
Heat pipe

ABSTRACT

This work numerically investigates the performance of an innovative compact-design solar water heater (SWH) using evacuated tube heat pipe solar collectors (ETHPSCs) coupled with a latent heat storage tank (LHS tank). This system is thought to be an unwritten fourth generation of solar

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<https://doi.org/10.1016/j.csite.2022.102490>

Received 18 January 2022; Received in revised form 3 October 2022; Accepted 11 October 2022

Available online 14 October 2022

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Latent heat thermal energy storage
Melting

water heaters known as heat-pipe-assisted thermal batteries. The basic and explicit hypothesis is to store energy than a bulk of hot water, which is the key distinction in compared to the prior classical sorts. Using three phase change materials (PCMs) for high and low solar intensity categories in the tropical region of South East Asia, a computational effort is conducted to model the system and the given temperature headed for the supply output hot water. This research is validated by two interrelated experimental and analytical investigations. According to the results of 12-h charging (melting) process, the maximum mean temperature of the PCMs belongs to PCM C, Rubitherm (RT70HC), which indicates 101 °C during the defined typical high daily solar radiation. This value implies 2.02% error in compared to the reference study. Referring to the solid-liquid interface parameter, the maximum liquefied thickness is also for PCM C with 76 mm thickness. However, reaching the complete melting is required by further studies.

Nomenclature

Abbreviations/acronyms

ETHPSC	Evacuated tube heat pipe solar collector
LHS	Latent heat storage
LHTES	Latent heat thermal energy storage
HP	Heat pipe
HPA	Heat pipe adiabatic unit
HPE	Heat pipe evaporator unit
HPC	Heat pipe condenser unit
HT	Heat transfer
PCMs	Phase change materials
SWH	Solar water heater
TC	Thermal conductivity
TLDSR	Typical low daily solar radiation
THDSR	Typical high daily solar radiation

Symbols

$A(f)$	The porosity function
A_{mush}	The constant of mushy zone
c	The specific heat capacity ($\text{KJ Kg}^{-1} \text{K}^{-1}$)
C_m	A constant which is fundamentally assumed for the melting configuration
C_p	The constant-pressure heat capacity (KJ K^{-1})
f	The volume fraction
h	The sensible enthalpy (KJ Kg^{-1})
h_{sl}	The latent heat by means of the latent heat of the PCM fusion (KJ Kg^{-1})
H	The enthalpy of PCM (KJ)
k	The thermal conductivity of PCM ($\text{W m}^{-1} \text{K}^{-1}$)
L	Heat storage capacity (KJ K^{-1})
S	The momentum sink phrase
T	Temperature ($^{\circ}\text{C}$)
u	The velocity in x direction (m s^{-1})
v	The velocity in y direction (m s^{-1})
X_{pcm}	Liquid-solid location in PCM slab
$X(t)$	The solid-liquid interface

Greeks

ΔH	The latent heat (KJ)
ε	A constant
μ	Viscosity ($\text{Kg m}^{-1} \text{s}^{-1}$)
ρ	Density (Kg m^{-3})
τ	Time (s)

Subscripts

c	Condenser
eff	Effective
hp	Heat pipe
l	Liquid
m	Melting

<i>pcm</i>	Phase change material
<i>ref</i>	Reference
<i>s</i>	Solid
<i>x</i>	X direction
<i>y</i>	Y direction

1. Introduction

Over the last decades, the latent heat storage phenomenon has been enticed by scholars to utilize thermal energy because of its higher potential capacity rather than sensible heat applications. However, it is evident that thermal energy can be stored by two approaches of sensible and latent heat along with also the thermochemical energy storage approach (Y. [1]. If one looks at the unfavorable statistics for fossil fuels, the significant role of thermal energy storage devices becomes clear [2]. Over the last years, and according to the statistics, the fossil fuel resources would encompass nearly 86% or even higher amount of global total energy consumption [3,4]. While the remarkable percentage of 61% of such total energy consumption from fossil fuels are naturally and deleteriously returned back to nature, the greenhouse gas effect has been consequently increased following that of wasted energy [5] (H. [6]. To this end, by recycling such a wasted thermal energy, its usage can be efficiently ameliorated to gain other forms of energy including electricity (J. [7]. However, another most noteworthy aspect of thermal energy storage systems is to employ directly the unbounded solar radiation to convert the heat into any desirable forms of outputs, including providing hot water by the solar water heater units [8].

On the other hand, in the solar water heaters, two technical active and passive methods are typically used for enhancement of heat transfer [9]. In general, active methods are defined as any external power required for the process, and passive methods do not need any additional energy for system improvement considering the thermohydraulic performance. Sometimes, the compound methods of such two approaches are employed for the heat transfer enhancement [10]. For instance, filling materials such as addition of nanomaterials [11,12] and usage of coating materials are sorts of the passive methods [13]. In the reference main mechanism, three passive methods, namely phase change material, fins, and heat pipe have been used for heat transfer improvement.

Thermal energy storage delivers the practical backup technology regarding the energy supply and demand as well as waste heat recovery [14,15]. In addition, phase change materials (PCMs) are taken into account as of the promising energy-saving materials because of their potency for storing and releasing huge amount of latent heat while going through the process of isothermal phase transition, providing applicable heating or cooling energies [16] (Y [17]. (S. [18,19]. In general, such a transition is from either solid or liquid states to the other. To clarify, by melting and solidification at the temperature of phase change, the PCM would be capable of storing and releasing energy (e.g., thermal energy), which is much larger in compared to the sensible heat storage. This while the internal structures of the PCMs change, referring to the latent heat storage capability. Following other advantages of PCM utilizations, they are employed for phase change energy storage in the various renewable and sustainable energy platforms, considering their specific features of superior latent heat as well as their low costs (J [7]. (H. [20]. Their lesser space required for energy storage in compare to the sensible heat storage is another noticeable aspect [21]. Likewise, the stable operational applications of latent heat energy storage phenomenon addressing low, medium, and high temperature ranges are also quite considerable [22,23]. Overall, the substantial potential of latent heat energy storage for energy saving owes to the huge energy storage density [24], fairly stability of temperature through melting and solidification stages (charging and discharging) or the isothermal behavior [25], and all other advantages [26]. Although, it is also noteworthy that the applications of PCMs are interestingly vary from case to case comprising of usages in solar energy, thermal energy management, electronic systems, HVAC systems, buildings, space industries, agricultural mechanisms, and so on, it has received ceaseless attention in solar thermal systems because of all above-mentioned characteristics [27, 28].

Another prevalent passive HT improvement method is the employment of the extended surfaces (fins complex). In fact, in the typical PCM-based fin-assisted mechanism, fins contribute to increase HT surface domain through PCM which enhances the HT rate receiving from the heat source towards the PCM [29]. Such a fin-assisted approach covers up the low thermal conductivity as the main well-known deficiency of the PCMs. For instance, Tiari and Hockins [30] have studied the thermal performance of a latent heat thermal energy storage (LHTES) unit enhanced with fins. Different numbers of annular and radial fins were applied for this target. For example, based on their results, charging and discharging of the system were faster 84.1% and 68.2%, respectively, by ten-annular fins while these amount were 85.8% and 68.6% by the twenty-annular fin. The four-radial and eight-radial fins showed these values by 81.9%, 70.0%, 86.6%, and 80.1%, respectively. In addition, as another aforementioned passive techniques, embedding HP in such PCM-based heat-pipe-assisted systems is widely used owing to its effectiveness for thermal efficiency enhancement. Increasing the number of HPs directly causes to increase HT between the heat transfer fluid and the PCM domain. Citing an example, Ladekar et al. [31] investigated the optimized performance of heat-pipe-embedded LHTES. The heat-pipe-assisted system achieved PCM sensible heating of 15 °C within 75 min while the classical copper-pipe storage tank needed 135 min for the same mass flow rate of 18 liter per minute resulting in 45% enhancement in the charging step.

The entity of using phase change materials (PCMs) in latent heat storage applications cause to accompany thermal hysteresis phenomenon. Thermal hysteresis is delineated as the separation between the actual melting/liquefying point and the crystallization temperature of phase change materials. In other words, substances, solids or liquids, require to be supercooled or superheated before a process of phase changing may happen. Consequently, the actual melting (or charging) point and the crystallization temperature point

of the material is higher or lower, in that order, versus the balance temperature of solid material. Moreover, PCMs, such as organic ones as one general example, would be accepted as desirable applicable choices attributable to their benefits containing dominance of higher latent heat, broader melting temperatures, plus chemical and physical stability, against the deficiencies of low thermal conductivity, as well as leakage during the time of solid-liquid phase change process [32,33]. All in all, numerous experimental, theoretical, analytical, and numerical studies have been done since the 80s for the appraisal of the PCM performance combined with various conditions [34,35].

In this numerical study, a pressure-based transient two-dimensional finite volume-based modeling is carried out to analyze the performance of the reference heat-pipe-embedded and fin-assisted energy storage compact unit with the PCM filled into the tank by using ANSYS FLUENT commercial software. Indeed, those three above-mentioned passive approaches for heat transfer enhancement were applied for the reference experimental work, which is elaborated in the next parts of the current study. In addition, three types of paraffin waxes as the PCMs and two typical climate conditions in South East Asia (Kuala Lumpur) were investigated. It is notable to hint that the intermittent variation features of outdoor air temperature as the meteorological and geographical conditions have been

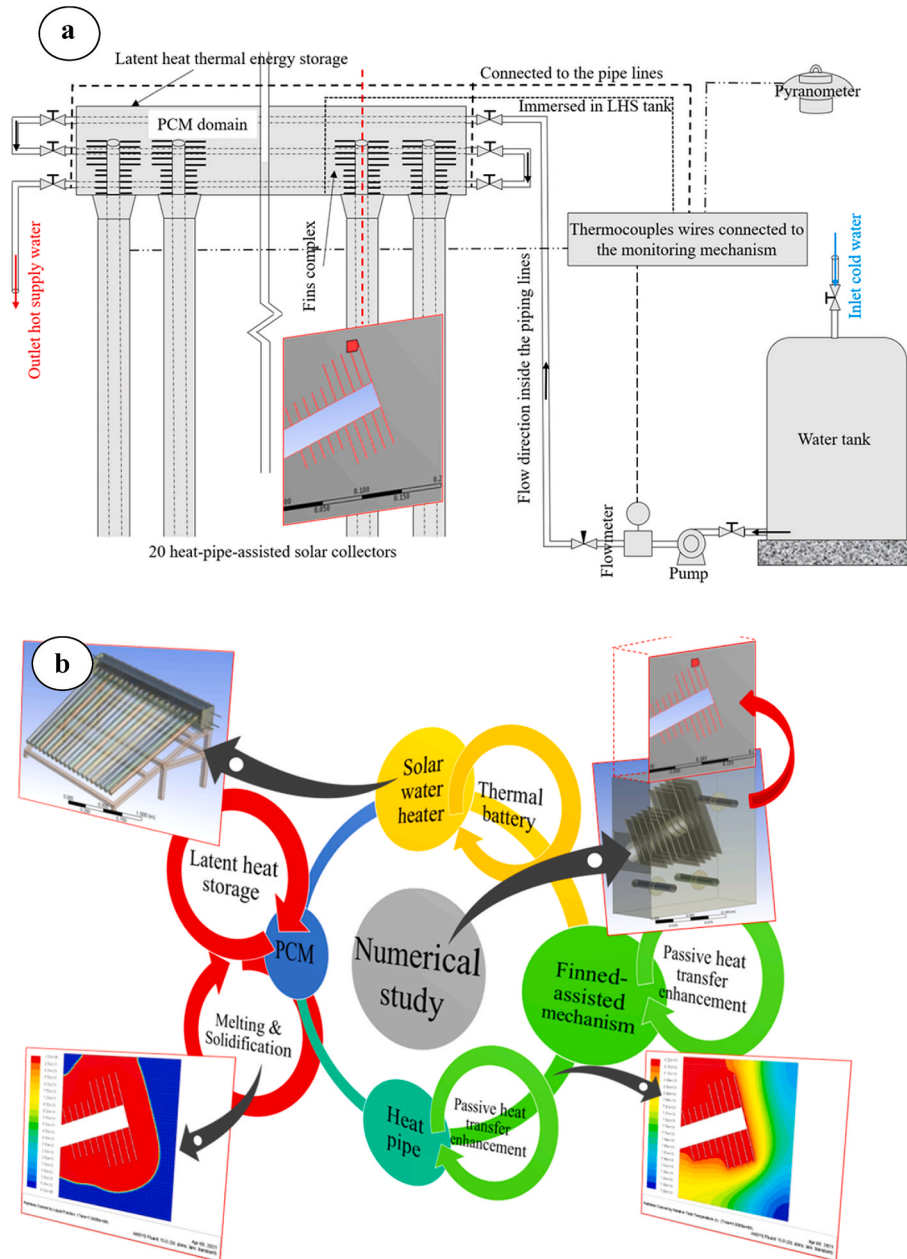


Fig. 1. Schematic design of the experimental mechanism, considering the 2D simulation area (Fig. 1a) [38] addressing the overview of what has been carried out by this numerical work depicted in Fig. 1b.

one aspect of many studies [36]. Although numerous studies on latent heat energy storage have been conducted, according to the best knowledge of the authors following to the interrelated literature review, there is lack of work on the CFD-oriented numerical study of the heat-pipe-based fin-assisted thermal energy storage compact domestic solar water heater unit. This CFD-based numerical work accompanies with simulation of three different PCMs, while the melting points of PCM A and B is far from PCM C, and, moreover, the latent heat storage capacity of PCM A and C in comparison with PCM B are far from each other. This factor causes to possess a good comparison work in between. The objective of this research is to evaluate the charging temperature and the solid-liquid interface parameters. This is a part of typical complexity in the literatures mainly focused on phase equilibrium assessment of PCMs, heat transfer evaluation, and/or storage analysis in terms of the latent heat storage studies (T. [34,37]. The present study can help to improve the understanding of such systems based on the latent heat capacity and melting point of different PCMs alongside with the effect of solar intensity range. The results can be evidently advantageous in terms of the clean energy storage applications.

2. Problem description

As the initial step, it is required to define the physical geometry of the system based on Fig. 1a which has been headed for the Fig. 1b as the overview of this current numerical study. Three different types of paraffin waxes are employed to be analyzed based on their temperature distribution and liquid fraction inside the latent heat energy storage reservoir under two different typical climate condition in this case study in Kuala Lumpur. In such a problem of latent heat storage (LHS)-integrated evacuated tube heat pipe solar collector (ETHPSC), the heat pipe plays a significant role. Its function is technically composed of receiving the heat by the evaporator section (HPE), and rejecting it by the condenser section (HPC). The heat flux, which is applied to the HPE, and, consequently, the temperature of the HPC section, which is directly connected with the fins complex, has been considered as the above-mentioned citing studies. Indeed, the HPC have a complex of fins attached for better spread of heat through the PCMs. In other words, based on the general standpoint of this numerical investigation, using both the HP and fins complex are two approaches of passive heat transfer enhancement to the PCM through charging mode and from the PCM through the solidification mode. In the meantime, the HP and evacuated glass tube have been also assembled with an aluminum fin insertion connecting these two components together. In fact, based on the reference experimental model, the aluminum fin-assisted section was held compactly between the inner side of the evacuated tube and the outer side of the heat pipe. This could be assumed as a significant design-oriented feature with the high-temperature condition over time as the aluminum softness ensures the long-term tight contact enabling the optimized performance. To this end, the incident solar energy is transmitted through the evacuated glass tube in which it is absorbed by such aluminum fins in the HPE section. A big percentage of the heat is absorbed by the aluminum fin is conveyed to the HPC section. Finally, the heat is transferred to the PCM inside the LHS tank.

Regarding the geometry, with the main target of this study at the PCM medium, which lies directly in the melting zone, the entire mechanism generally comprises of twenty number of heat pipes considering the same precise distances among all of the HPs, the fins attached, and the walls of the LHS tank. Through the current 2D simulation, such distances can be observed from side view, which also clearly demonstrates the identical distances of all parts. In other words, the sided 2D view can be a precise representative of the whole system. The tank width is 200 mm while the height is 220 mm based on the actual dimensions, containing the HPC section head, the fins complex, and as the main domain of the PCM storage. The schematic figure of the entire experimental setup is provided according to Fig. 1.

Based on Table 1, three various sorts of paraffin waxes, as the PCMs, are employed in this simulation work of the ETHPSC, referring to our previous works (Naghavi [38] and Bazri et al. [28]. In this two-dimensional (2D) numerical study, only the melting process, which lies in charging mode of the ETHPSC integrated with a LHS tank, is investigated. The study has been performed for different solar radiation incident, namely in the typical high and low solar radiation days in Kuala Lumpur, as a typical tropical city, and, consequently, different inlet temperature through the targeted system. Low and high solar daily radiations imply the typical sunny and cloudy/rainy days monitored carefully, as the average daily radiation of 11 and 20 MJ/m², respectively.

In general and just in parenthesis, the mechanical, thermal, and optical behavior of the materials and the systems are important to be identified for the theoretical and/or numerical assessment of such problems. To this end, and in order to draw a broader outlook, Fig. 2 illustrates the entire properties classifications to determine better the situation of the problem accordingly.

On the other hand, the analytical approaches of melting problems require several substantial assumptions as well as simplifications. In general, and also in the current case, they can be sensibly confined to conduction heat transfer, initial boundary conditions, and thermophysical properties with constant values. To clarify, several assumptions are essential to consider for simplification of the typical solar collector model [40] (D. [41], as follows:

Table 1

The overall specifications of three various paraffin waxes, as the PCMs, using in this numerical research [39].

Property	PCM A	PCM B	PCM C
Manufacturer (Company) name; PCM name	Sigma Aldrich (ASTMD87)	ClimSel (C58)	Rubitherm (RT70HC)
Melting point, T_m (°C)	53-57 (~55)	55-57 (~56)	70-74 (~72)
Heat storage capacity, L (kJ/kg)	163	260	159
Thermal conductivity, k_s/k_l (W/mK)	0.35/0.17	0.57/0.47	0.22/0.20
Density, ρ_s/ρ_l (kg/m ³)	990/916	1470/1330	917/822
Specific heat, c_s/c_l (kJ/kgK)	2.48/2.76	2.26/2.55	2.05/2.34

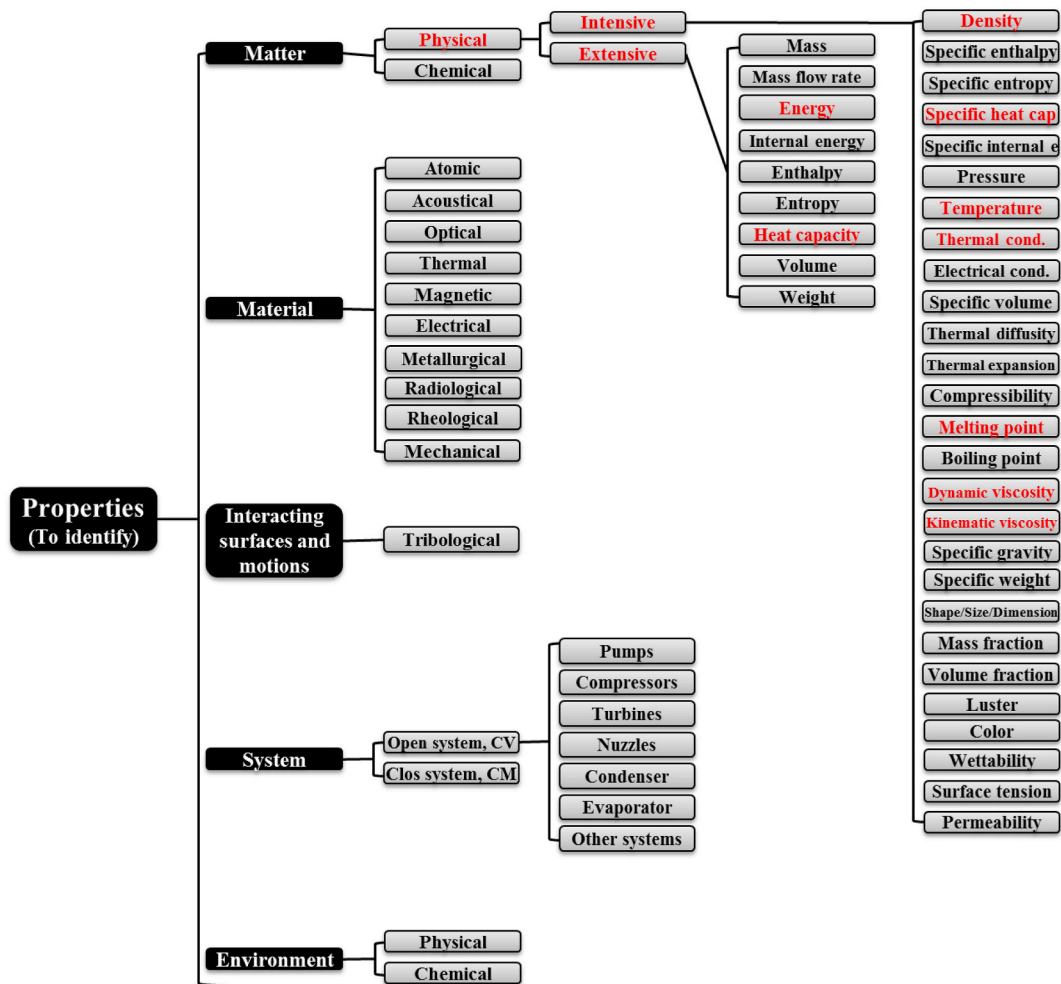


Fig. 2. Where do we fall in among all defined properties to identify the required properties of the PCMs in the present defined simulation problem?.

- i. Another necessary assumption, directly affecting the simulation is that the heat transfer from the solar collector to the heat pipes and vice versa is assumed as steady state.
- ii. The entire sides and edges of the tank, and solar collector have been well insulated; consequently, the heat loss measurements in such sections could be neglected.
- iii. As a general assumption, the thermal resistance of vapor flow would be negligible.
- iv. The heat loss in the heat pipe adiabatic unit (HPA) is considered to be ignored.
- v. The temperature gradient does not exist in the axial direction of the heat pipe.
- vi. The contact thermal resistance among all walls of the cylindrical absorbers and the heat pipes are assumed to be neglected as well.
- vii. In terms of materials, the thermophysical properties of all solid material, phase change material, and water in this system, affecting the simulation, are considered as constant.
- viii. The thickness of fins and walls are negligible.
- ix. The natural convection of the process, specifically of the PCM, is neglected because the natural convection effect in the solicitation process can be assumed insignificant.
- x. For modelling simplification, the finned heat-pipe-assisted latent heat storage tank of the solar water heater and all contact surfaces are set as symmetry plane, as the computational domain.

3. Governing equations

The thermal performance of a finned evacuated heat pipe-assisted solar water heater integrated with a latent heat thermal energy storage unit is modeled. A transient two-dimensional (2D) finite volume-based modeling is applied for the analysis of the characteristics of the storage unit with paraffin wax as the phase change material. According to the problem, the governing equations are as follows:

3.1. Phase change material (PCM)

Continuity equation:

$$\frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} = 0 \quad (1)$$

where ρ indicates as the density of PCM, and the velocity in x and y directions lie in u and v , respectively.

x-momentum equation:

$$\frac{\partial(\rho u)}{\partial \tau} + \frac{\partial(\rho uu)}{\partial x} + \frac{\partial(\rho uv)}{\partial y} = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left(\mu \frac{\partial u}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu \frac{\partial u}{\partial y} \right) + S_x \quad (2)$$

where μ shows the viscosity of PCM.

y-momentum equation:

$$\frac{\partial(\rho v)}{\partial \tau} + \frac{\partial(\rho uv)}{\partial x} + \frac{\partial(\rho vv)}{\partial y} = -\frac{\partial p}{\partial y} + \frac{\partial}{\partial x} \left(\mu \frac{\partial v}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu \frac{\partial v}{\partial y} \right) + \rho g \beta (T - T_m) + S_y \quad (3)$$

Moreover, the porosity (porous medium is followed by mushy zone based on the enthalpy-porosity approach) in every element would be considered as the equivalent to that of element liquid fraction. The liquid fraction (f) is associated with a primary modification based on the enthalpy formulation. It is clear that the approach can be employed while the liquid phase is allowed to flow. The source term of momentum equation is as below:

$$S = -A(f) \cdot V \quad (4)$$

where $A(f)$ signifies the porosity function, using for gradual reduction of the velocities from liquid with a certain magnitude to the solid with zero value. $A(f)$ definition makes the equation of momentum mimic the relation of Carman-Kozeny for flow through the porous media as follows:

$$A(f) = \frac{C_m (1-f)^2}{(f^3 + \varepsilon)} \quad (5)$$

where ε symbolizes a tiny constant, using for prevention of zero in denominator, and C_m shows another constant which is fundamentally assumed for the melting front configuration. The parameters/features of dimensions, and shapes of the mushy zone relates to such constant values. Therefore, such parameters affect the rates of melting and solidification modes. Following the source term of momentum equation, the momentum sink phrases, S_x and S_y , are as below, by considering the reduction of porosity medium in the mushy zone:

$$S_x = -\frac{(1-f_l)^2}{(f_l^3 + \varepsilon)} A_{mush} (u - u_p) \quad (6)$$

$$S_y = -\frac{(1-f_l)^2}{(f_l^3 + \varepsilon)} A_{mush} (v - v_p) \quad (7)$$

where f_l indicates the volume fraction of liquid, and its definition according to the PCM temperature is:

$$f_l = 0 \text{ if } T < T_{solidus} \quad (8)$$

$$f_l = 1 \text{ if } T > T_{liquidus}$$

$$f_l = \frac{T - T_{solidus}}{T_{liquidus} - T_{solidus}} \text{ if } T_{liquidus} < T < T_{solidus}$$

ε shows a tiny value to avoid denominator of zero, and also A_{mush} signifies the constant of mushy zone, which calculates the damping amplitude. The meaning is the higher the damping amplitude is, the more downward will be the transition of material velocity towards zero while solidifies.

Energy equation:

$$\frac{\partial(\rho H)}{\partial \tau} + \frac{\partial(\rho u H)}{\partial x} + \frac{\partial(\rho v H)}{\partial y} = \frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(k \frac{\partial T}{\partial y} \right) \quad (9)$$

where, k and H represent the thermal conductivity and the enthalpy of PCM, respectively. Enthalpy is measured as:

$$H = h + \Delta H \quad (10)$$

ΔH indicates the latent heat, and h is the sensible enthalpy defining as:

$$h = h_{ref} + \int_{T_{ref}}^T C_p dT \quad (11)$$

h_{ref} , T_{ref} , and C_p implies the reference enthalpy, reference temperature, and the constant-pressure specific heat. In addition, the latent heat by means of the latent heat of the PCM fusion, h_{sl} , is as below:

$$\Delta H = f_l h_{sl} \quad (12)$$

It means that the latent heat range is from zero (solid PCM) to h_{sl} (liquid PCM).

3.1.1. Latent heat of PCM

Over charging stage, the boundary condition of the interface location ($X(t)$), is added. Consequently, the equation of transient heating (13) process is solved by the below equations and boundary conditions:

$$\rho_s c_s \frac{\partial T}{\partial t} = k_s \frac{\partial^2 T}{\partial x^2} \quad (13)$$

where ρ_s , c_s , and k_s specify the density, specific heat, and thermal conductivity at solid phase,

Now, at boundary conditions for the range of $0 < x < X(t)$:

$$T_{pcm}(0, t) = T_{hp}(t) \quad (14)$$

where T_{pcm} , and T_{hp} show the temperatures of the PCM and heat pipe,

$$T_{pcm}(X_{pcm}(t), t) = T_m \quad (15)$$

where T_m indicate the temperatures of the melting point,

$$\rho_l L X_{pcm}(t) = -k_l \frac{\partial T_{pcm}}{\partial t} \text{ where } X_{pcm}(t) \quad (16)$$

where L implies the latent heat of PCM.

Solid-liquid location in the PCM slab, X_{pcm} : (While also the slab of PCM is $0 \leq x \leq L$)

$$X_{pcm}(0) = 0 \quad (17)$$

In equations of 14 to 17, the properties values of the PCM are assumed in the liquid state (L).

3.2. Heat pipe (HP)

The special characteristic of heat pipes is their capability to transfer heat at considerably higher rate in comparison with solid conductors (e.g., copper) through taking advantages of the phenomenon of liquid and vapor transition. Nevertheless, there are rare studies for simulation of the mechanism inside heat pipes due to the difficulties of the whole physical processes. Most of the studies accomplished so far assumed the HP as a tool/device with constant and high thermal conductivity (TC). In this numerical study, HP has been modeled as a device with high constant TC. The equation of energy for the HP is:

$$(\rho C_p)_{eff} \frac{\partial T}{\partial \tau} = k_{eff} \left[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right] \quad (18)$$

3.3. Fins complex

The equation of energy for the fins complex is:

$$(\rho C_p)_f \frac{\partial T}{\partial \tau} = k_f \left[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right] \quad (19)$$

4. Solution method and description of the numerical method

A numerical analysis has been employed for this work to investigate the heat transfer through an evacuated tube heat pipe solar collector (ETHPSC) integrated with a latent heat storage (LHS) tank. The commercial ANSYS FLUENT software has been utilized to solve the defined problem. The physical model has been created by ANSYS design modeler. As previous clarification, the complete setup comprises of twenty number of heat pipes considering the same precise distances from each other, the fins attached among all of the HPs, and the walls of the LHS tank. As it was mentioned earlier, the thickness of all walls, heat pipe, and the fin complex were neglected, and the PCM domain has been in direct contact with the finned heat-pipe-assisted complex in the computational model. Through the 2D simulation, such distances are observed from side view, which also clearly demonstrates the identical distances of all parts. In other words, such 2D view (from side view) can be a precise model of the whole system. The structured uniform grid has been applied for discretization of the computational domain. The interior part of the model comprising of the PCM apart from the fins complex attached to heat pipe was considered as the interior body boundary condition. In the meantime, heat flux of all walls were

assigned by zero value (W/m^2). Moreover, the analyses were carried out by hourly consecutive simulation. The pressure based transient technique within FLUENT has been employed to solve the momentum equations of all PCMs by considering SIMPLE scheme for the velocity-pressure coupling. Least Square Method was used for spatial discretization. Also, Second Order Upwind differencing technique has been employed to discretize the convective terms in the equations of momentum and energy. To develop the stability of convergence mode, the under-relaxation factors of the pressure, momentum, liquid fraction, and energy, were chosen as 0.3, 0.7, 0.9, and 1.0, respectively.

4.1. Monitoring the PCM temperature inside the latent heat storage tank

Considering the actual experimental model and its results, and by observing the heat pipe performance, their effects upon the overall throughput and performance of the solar collector water heater play a huge role. In order to obtain the wall temperature of heat pipe, five divided sectors (as the presumption sectors) of the heat pipes, in which assigned to record the temperature of heat pipe wall, were assumed by employing 41 T-type-copper-constantan thermocouples according to Fig. 3 and as follows: heat pipe condenser unit (HPC) (from TC22 to TC26), heat pipe adiabatic unit (HPA) (from TC27 to TC31), and heat pipe evaporator unit (HPE) (from TC32 to TC36 and from TC37 to TC41) sectors.

Following this simulation model, and referring to the data and calculations of the experimental study, and also by assuming the above-mentioned explanation based on Fig. 3, the PCM (paraffin) temperatures have been recorded as the mean temperature in the LHS tank, $\overline{T_{PCM}}$, the temperature in middle of the tank, $T_{PCM,m}$, and the temperature in the bottom of the tank, $T_{PCM,b}$. The mean temperature of PCM has been calculated as the average PCM temperature by the time.

Hence, according to the results of the analytical study by Bazri et al. [39] and the experimental study by Naghavi [38], the temperatures of three types of paraffin, as the PCMs, at 1 cm distance from the fin wall as well as the mean temperature inside the LHS tank are supposed to be obtained by the simulation study and to be compared with those two studies as the validation part.

4.2. Justification for simplification of the HPC simulation

Based on Fig. 4, and as simplification of the simulation model, the HPC part temperature, which is directly connected with the fins complex, is considered to display the temperature inlet in the melting or charging process of the current two-dimensional (2D) model. This can be as the justification of such simplification beside the fact that through time passing of melting and solidification stages, whatever the PCM is absorbing and then storing the thermal energy would follow a predictive trend because of the low thermal conductivity of the PCMs. In other words, owing to the fact that the temperature variations between the HPC and the PCM layer in direct contact with it has been recorded very low, the HPE and the HPC temperatures, and subsequently the inlet temperature into the PCM are almost equal.

The temperatures of the PCMs at the prechosen distance of 1 cm, perpendicular distance from the last fin plate next to the end of heat pipe direction (based on Fig. 4 which is precisely measured in the CAD/CAM step of the simulation), and the PCM mean temperatures are investigated. The most important point here is that because of the present 2D computational model, the simulated symmetric face of the PCM domain represent the overall temperature throughout the entire tank, which is completely logical. The reason is that the whole actual cubic latent heat tank, comprises of the fins complex and the PCMs can be considered identical according to the designated experimental model.

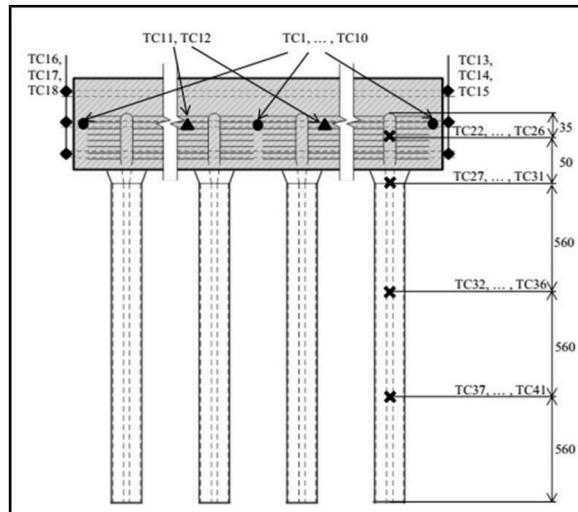


Fig. 3. The locations of the mounted thermocouples (TCs) [38] (Reprinted from Ref. [42] and with permission from Elsevier).

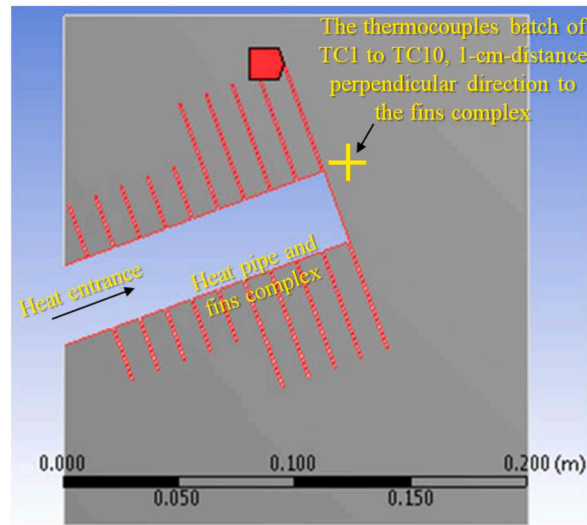


Fig. 4. The 2D ANSYS-based schematic view.

5. Model validation (Reliability tests of the numerical simulation)

In order to validate the numerical model, the results obtained from the simulation were compared with the experimental and analytical results of two studies by Naghavi [38] and Bazri et al. [39]. Validation tests were performed for different solar radiation incident already indicated. Therefore, such variations evidently caused varied melting and solidification, considering the charging and discharging modes for three types of phase change materials. For the purpose of the model validation, the test results are illustrated that display the comparison between two previous reference studies and the current numerical model.

The existence of very noticeable agreement among the results of previous reference works and the present computational study illustrates the acceptable outputs. In addition, the small difference of this comparison can be caused by numerical parameters, 2D simulation, the two phases of PCMs, the differences in PCM actual and simulated properties, and so on.

5.1. Grid and time-step independence tests

For such a finned-heat pipe-assisted solar collector, the quadrilateral dominant grid was applied for the meshing method. The grid around the finned-heat pipe complex was denser to advance the accuracy of computations, regarding the whole heat transfer problem from the finned-heat pipe complex to the PCM domain. The basic grid distribution or meshing sketch of the computational model is illustrated in Fig. 5. The independence of the grid size (meshing size) and time step were tested. All of the numerical models were checked for grid independence aimed at examination of the mesh number sensitivity to the simulation results. Three mesh sizes of fine to coarse, defining from higher to lower number of grids were checked. Consequently, the finalized meshing with assured outcome was selected, considering the grid and time-step independence checking (Table 2). For the finalized optimum meshing, the body sizing definition was set as the element size of 1 mm while the minimum and maximum sizing of the grids were selected as 0.1482 and 29.639 mm, respectively. 40657 and 39394 were the number of nodes and elements as the statistics of the basic meshing model.

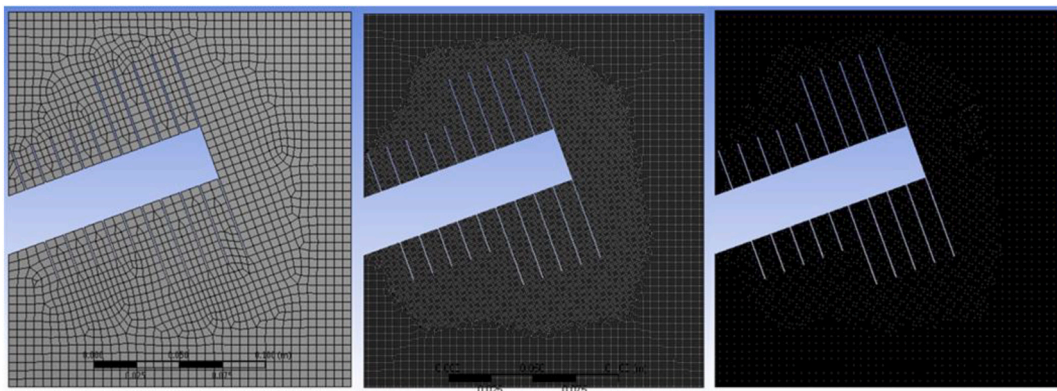


Fig. 5. Meshing or the grid distribution (as the sketch of mesh) of the finned heat-pipe-assisted solar water heater based on the grid independence test.

Table 2

The checking of the grid and time-step independence outputs.

Time step (s)	Number of time steps	Element size (m)	Min size (m)	Max size (m)	Number of elements
1	3600	5e-004	0.741e-004	1.1319e-002	157701
2	1800	1e-003	1.482e-004	2.2639e-002	39394
5	720	5e-003	7.41e-004	14.8195e-002	1842
1	3600	5e-004	0.741e-004	1.1319e-002	157701
2	1800	1e-003	1.482e-004	2.2639e-002	39394
5	720	5e-003	7.41e-004	14.8195e-002	1842
1	3600	5e-004	0.741e-004	1.1319e-002	157701
2	1800	1e-003	1.482e-004	2.2639e-002	39394
5	720	5e-003	7.41e-004	14.8195e-002	1842

6. Results and discussion

The consequent outputs of the present numerical model are illustrated in this part to clearly show the temperature distribution and melting fraction in the finned-heat pipe-assisted ETHPSC-LHS unit, based on the 2D model. In general, during melting step, which is called charging mode, the heat which is transfers from glass absorber, heat pipe evaporator, heat pipe condenser, fins complex, and in the end into the paraffin wax as the phase change materials can be stored in it in both sensible and latent heat energy. The most significant part of the mechanism in this step is that the supply water is not passed through the mounted water pipes inside the latent heat storage tank in the charging only step. Whereas the supply water passes through the pipes in the discharging mode to receive the stored heat energy from the liquefied PCM. The current strategy is drawn to utilize the supply water while there is weak solar energy such as at night or during rainy hours.

Over the typical melting and solidification processes of PCMs, the configuration and orientation of the fins complex, as the passive heat transfer approach, inside the latent heat storage unit can evidently affect its enhancement. Heat is flows through the solid state of PCM as the conduction, as the very beginning of charging process. It is evident that one of the main characteristics of PCMs is their low thermal conductivity. However, since the liquified PCM shows a lower thermal conductivity in comparison with the solid state, the conduction heat transfer can be assumed trivial. Indeed, the subsequent additional melting phenomenon is by natural convection as a result of the density gradient inside the liquid state. Accordingly, the PCM nearby the fins complex does not affect the enhancement of the conduction rate. Moreover, it is quite important that as long as the temperature of PCM has not been touched the melting point, the heat absorption is sensible, once the melting temperature is reached, the latent heat phenomenon is begun, and the role of latent heat storage shows its capacity. According to the three various PCMs, different runs (Table 3) have been conducted to simulate both aforementioned studies, considering the inlet HPC temperature to transfer into the PCM domain as it was already justified in previous sections.

Fig. 6 depicts the temperature gradient of PCMs, A, B, and C, over the time based on the actual scale of the heat transfer domain of the PCMs in the symmetric view of the present two-dimensional simulation, which is entirely identical throughout the real physical model. The justification is that the actual rectangular prism tank filled with the PCM, comprising of the fins complex has been experimentally and theoretically designated identical. In other words, all twenty heat pipe and fins complex inside the PCM tank have arrayed and mounted at the same distances and positions. To clarify, the 2D simulated plane is defined at the center of the actual experimental finned HPC, which has been mounted as 20 number throughout the tank with identical distances from each other and also because of the technical symmetric state of the input heat flux from each finned HPC to the PCM, the justification for doing the present research has been made. However, this evaluation is in the charging only mode and for discharging only stage, the analysis will be different.

By considering typical low daily solar radiation (TLDSR) and high daily solar radiation (THDSR) in Kuala Lumpur, Malaysia, and following the aforesaid experimental and analytical studies, the temperatures of the three PCMs at the chosen distance of 1 cm,

Table 3

HPC inlet temperature for all three PCMs during the typical high (h) and low (l) daily solar radiation within the considered hours of charging mode from 7am to 7pm.

Time (hours)	HPC temperature (°C)					
	PCM A, h	PCM A, l	PCM B, h	PCM B, l	PCM C, h	PCM C, l
7:00	58	58	59	59	77	75
8:00	60	60	59.5	59.5	78	77
9:00	61	61	60	60	79	79
10:00	68	66	64	62	90	88
11:00	79	73	70	66	105	97
12:00	89	79	75	70	119	106
13:00	96	84	80	73.5	130	113
14:00	99	87	82.5	76	136	118
15:00	100	88	84	77	138	120
16:00	99	86	83.5	76	137	119
17:00	98	85	83	75	136	118
18:00	96	84	82.5	74.5	135	117
19:00	95	83	82	74	134	116

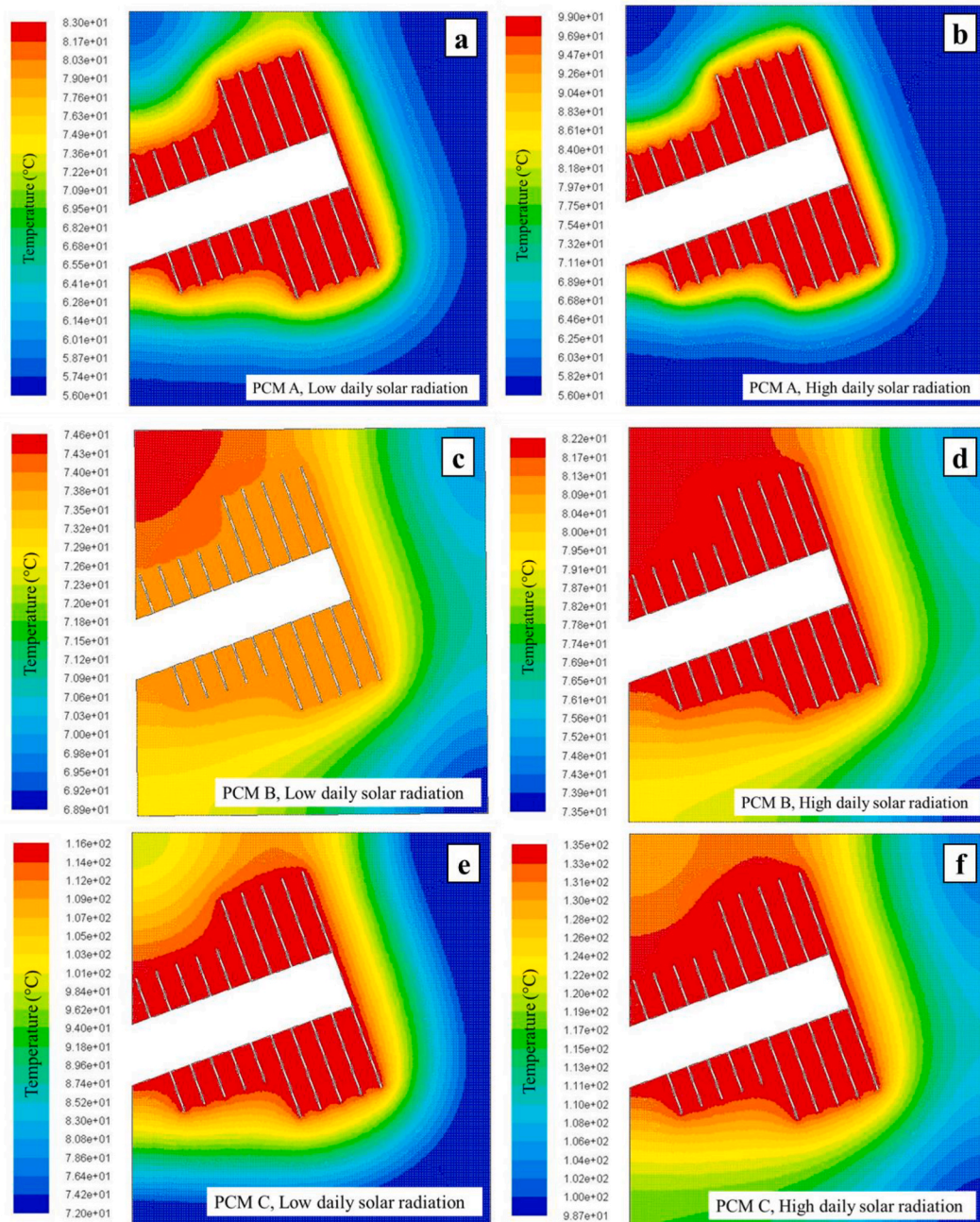


Fig. 6. The temperature distribution of PCM A, B, and C at 7pm after the entire charging or melting mode, at both low and high daily solar radiation typical days, as the average daily radiation of 11 and 20 MJ/m², respectively.

perpendicular distance from the last fin plate next to the end of heat pipe direction, and the PCM mean temperatures have been investigated. To clarify, the mean temperature of the PCMs has been assumed as the average temperature of each PCM section accumulated around each finned-heat pipe set in 3D state by the time. Therefore, the average temperature of the PCM in the 2D plane view of this section, cutting out through the middle of finned heat pipe complex somehow as a cross-section, would be the PCM mean temperature.

Based on Fig. 6 and then 7, at 1-cm distance chosen point, for PCM A during the THDSR, the PCM simulated temperature reaches 86 °C while the mean temperature touches the maximum of 65 °C versus 90 °C and 68.5 °C in the reference analytical study. These values are 76 °C and 61 °C for the TLDSR in compared to the analytical values of 79.5 °C and 63 °C that imply engineering errors from the range of 0.8% towards less than 5.88% for PCM A. Besides, the highest temperature of heat pipe condenser has been considered around 100 °C and 88 °C for such high and low typical days based on the pivot study. The simulation study represents the

improvements of 13.16% and 6.56% for the 1-cm-distance and the mean temperatures from the TLDSR to THDSR, respectively.

For PCM B, the simulated values for the 1-cm distance and the mean temperatures at the typical high and low daily regional solar radiations reach 81 °C, 66 °C, 73 °C, and 62 °C, respectively. In the analytical study, these temperatures were reported as 78 °C, 64 °C, 70.5 °C, and 60 °C, respectively, in which these values indicate errors from 3.8% to around 5%. While the highest temperature of HPC is less than 85 °C for THDSR and not more than 77 °C for TLDST, the general trend and the plotted scattered lines are somehow similar to PCM A but with different temperature levels. It demonstrates the amounts of 10.96% and 6.45% enhancement for the two defined pivot temperatures from the TLDSR to THDSR through the simulation work.

Nonetheless, for PCM C and during a THDSR or even in a TLDSR, the scenario has differences. In the THDSR, the temperature of PCM C at the 1-cm distance is around 129.5 °C while the mean temperature is almost 101 °C, considering the highest temperature of HPC up to 138 °C as the inlet temperature. These temperatures are 128 °C and 99 °C in the analytical study. These values over the TLDSR are obtained as 108 and 88 °C versus 110.5 and 89.5 °C in the reference study. To clarify, the present simulation study indicates the values of 19.91% and 14.77% of improvements for the 1-cm-distance and the mean temperatures from the TLDSR to THDSR, respectively, that is the highest amounts among the two other PCMs. From the apparent side, it seems that PCM C showing higher temperatures can be the best choice, but, with consideration of other parameters, namely the latent heat energy storage capacity and the coefficient of thermal conductivity, the output temperature of the supply hot water delineates the most practical phase change material. However, according to the analytical study, PCM C showed the highest output temperature.

Moreover, for PCM A, the maximum obtained temperature of the melted PCM nearby the fined HPC is less than 99 °C in a THDSR, whereas this value is about 83 °C for a TLDSR in Kuala Lumpur. These simulated results are approximately 90 °C and 80 °C in the referred previous studies, as the pivot work. These simulated temperatures reach up to 82 °C and 75 °C for PCM B in compared to the values of 78 °C and 70 °C in the reference analytical study for PCM B, whereas these parameters go up to 135 °C and 116 °C in simulation for PCM C versus 125 °C and 110 °C, respectively. The differences are quite close that can be in an acceptable agreement as the validation which has been elaborated in more details further.

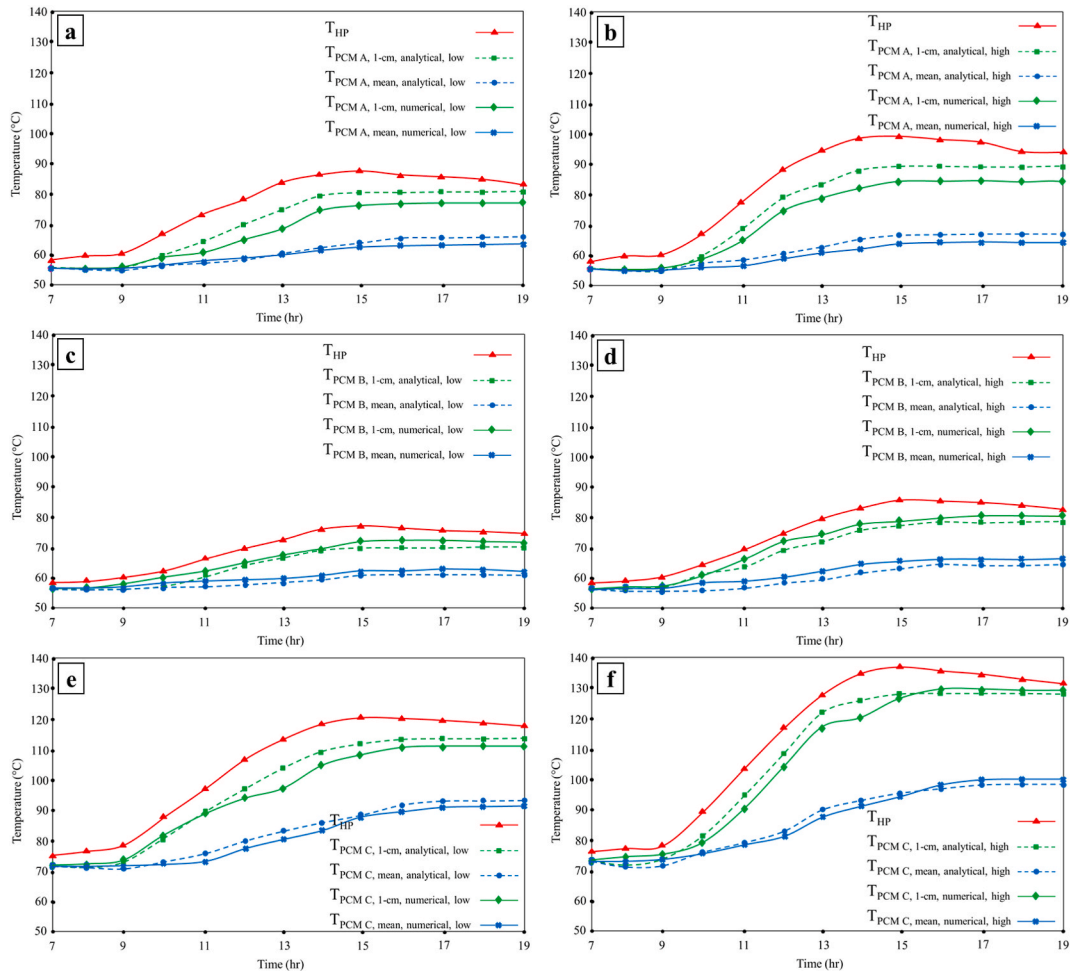


Fig. 7. The overall comparison among the temperatures of heat pipe compressor section, 1-cm-distance, and mean value for the referenced analytical versus the present numerical study in both low and high daily solar radiation conditions, as the average daily radiation of 11 and 20 MJ/m², respectively.

As the discussed Fig. 7, during afternoon, while the solar radiation would be lower, the temperature of PCMs remains almost steady due to the low differences of temperature between the finned heat pipe and the adjacent layer of the PCM. Indeed, with the time passing, the higher temperature of the PCM front layers (the closer layers of the PCM bulk to the extended surfaces of fins) leads the internal heat flux towards the rear layers, and, consequently, the smaller gradient of temperature inside the PCM is come up that can be visually observed based on Fig. 6 as well. In other words, it is evident that initially the PCM layers closer to the fins complex are getting hot and the heat is gradually transferred to the rear layers farther from the fins. In the meantime, due to the low conductivity of the PCMs, with time passing the assistance of the fins helps for heat transfer through the PCM material for smaller gradient of the interior

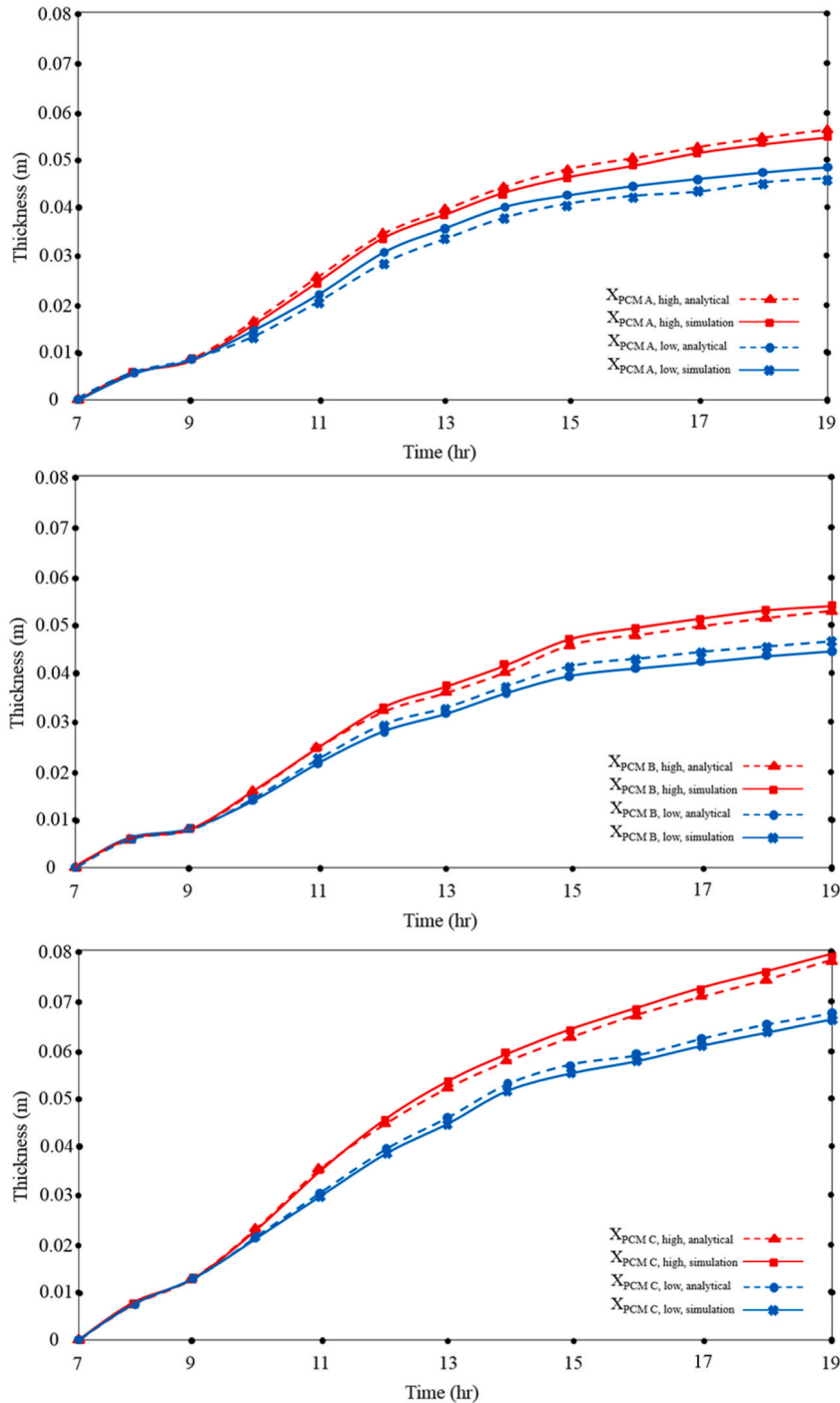


Fig. 8. Simulated solid-liquid interface (X) of PCM A, B, and C through the entire 12-h of charging or melting mode, at high and low daily solar radiation typical days, in compared to the analytical study.

temperature. On the other hand, because the heat transmission of heat pipe flows technically only in one direction from the evaporator side into the condenser side, similar to a type of check valve, the stored energy is not wasted out of the PCM domain inside the LHS tank.

In order to investigate the solid-liquid interface, referring to the schematic view of the geometry, the PCM medium domain, which entirely occupies the internal space of LHS tank and directly lies in the solidification and liquefying zone, is evidently considered. The entire mechanism comprises of twenty heat pipes considering the same precise distances among all HPs, the fins attached, and the walls of the LHS tank. Through the current 2D simulation study, such distances can be observed from side view, which also clearly demonstrates the identical distances of all elements. In other words, the sided 2D view can be a precise representative of the whole system. For the further future numerical research, the 3D study would be taken into account to be simulated. To this end, the assumption would be obviously different, and, the PCM inside the PCM reservoir can be divided into 20 equalized numbers of slabs.

As previously-mentioned parts, within the charging (melting or liquefying) mode, the heat transferred from the heat pipe is stored in the selected paraffin wax, as the PCM. In this case, from each of HPs, the equal heat is transferred, and the thermocline as well as the interface location in each section can be logically the same. It is scientifically evident that as soon as the heat pipe condenser temperature reaches greater than the PCM temperature at the melting point level, the melting or liquefying process, which is technically called charging is also commenced.

The simulated liquid fraction of the PCMs is compared with the experimental and analytical data by Naghavi and Bazri et al., which were computed by an approximation approach using for the estimation of the liquid fraction of the PCMs in the LHS tank and Matlab programming, respectively. Referring the experimental and analytical data, which is the basis of this numerical research, the approximation technique was employed by utilization of the recorded temperature data of the PCM inside the tank, considering both in the middle and the bottom locations.

As a result, the present 2D study's solid-liquid interface measurements may be generalized to be simulated alongside the temperature distribution of each variety of paraffin wax. Fig. 8 depicts the movement of the solid-liquid interface on typical Kuala Lumpur sunny and rainy days, taking into account all three types of PCMs and both low and high radiations. For example, the PCM simulated molten thickness on each side of the finned HPC for PCM A is roughly 55 and 46 mm at the end of the melting stage at 7pm, compared to the analytical research findings of 57 and 48 mm for THDSR and TLDSR, respectively. In the row for THDSR and TLDSR, the liquefied thickness spans about 54 and 47 mm for PCM B at the end of the charging mode at 7pm, compared to 53 and 45 mm for the reference research. Finally, for PCM C, these similar values are about 76 and 67 mm in the simulation, and 75 and 68 mm in the analytical research, in that order.

Furthermore, the influence of the solid-liquid interface and temperature distribution may be investigated also by taking into account both radial heat transfer, which refers to the HP wall, and angular heat transfer, which is caused by the fins complex that surrounds the HP. In other words, apart from all interpretation of the figures, the solid layers accumulating around both heat pipe and the fins complex shaping the radial and angular thickness of PCM materials can be studied by radial and angular heat transfer, which is also associated with the design of the system. This part can be considered as a further mathematical or numerical research. In terms of other future researches, the characteristics such as the number, size, and shape of the fins, as well as the varied melting temperatures of the PCMs, will be investigated using an optimization study to create the optimum feasible configuration.

As for the finalized interpretation, Table 4 has been provided to demonstrate the overall summary of the entire numerical results in comparison with the reference analytical study, which is also according to the reference experimental study, following both the PCM temperatures and solid-liquid interface, which depicts the liquified thickness of the PCMs in the charging mode. As it can be seen, PCM C, Rubitherm (RT70HC), has reached the highest temperature during the 12-h melting process. Moreover, its error has been obtained by the simulation somehow as the minimum amount among the other PCMs. Likewise, the maximum error of 5.11% can be observed for PCM A at the mean temperature, which arises also from a high error of 4.44% at 1-cm distance temperature for this PCM in THDSR. The main reason can be lie on the lowest amount of melting point for PCM A among the two others, considering the solidus and liquidus simulated temperatures, which can cause a more critical simulation in comparison with the other PCMs. However, by virtue of the fact that the errors are quite close to each other, and this maximum amount of error still can be interpreted somehow as an acceptable value. On the other hand, by considering the temperature distribution, the current simulation study signifies the improvements of 13.16% and 6.56% for PCM A for 1-cm-distance and mean temperature values moving from the TLDSR to THDSR, respectively. This goes for the amounts of 10.96% and 6.45% enhancement for these two situations in case of PCM B, and, most interestingly, the simulation specifies the values of 19.91% and 14.77% of progresses from the TLDSR to THDSR at the two pre-chosen defined temperatures, respectively, that is the maximum amounts among all three PCMs. Building upon the above-mentioned indication, it would be evident that the solid-liquid interface has a superior growth for PCM C. Nevertheless, since the latent heat storage capacity of PCM C is less than PCM B, the finalized analysis can be drawn from the magnitude and volume of the output temperature of the supply hot water. This aspect of the research is going to be carried as the discharging process. Discharging stage has its specific technical challenges, namely the rapid solidification of PCM around the finned-heat-pipe complex resultant in big temperature drop. The discharging mode investigation is going to be reported through our upcoming papers.

7. Conclusion

A transient 2D actual-scale model was used to simulate the charging phase of a revolutionary compact-design solar water heater comprised of evacuated tube heat pipe solar collectors coupled with a latent heat storage tank.

- Two further analytical and experimental tests of previously published articles verified the numerical model.

Table 4

A summary of full numerical results in compared to the reference analytical study.

Comparison results							
PCM conditions		THDSR			TLDSR		
		Present simulation	Reference analytical	Engineering error (%)	Present simulation	Reference analytical	Engineering error (%)
PCM A temperature (°C) at:	1-cm distance	86	90	4.44	76	79.5	4.40
	Mean	65	68.5	5.11	61	63	3.17
PCM A, liquefied thickness (mm)		55	57	3.5	46	48	4.16
PCM B Temperature (°C) at:	1-cm distance	81	78	3.85	73	70.5	3.55
	Mean	66	64	3.12	62	60	3.33
PCM B, liquefied thickness (mm)		54	53	1.85	47	45	4.44
PCM C temperature (°C) at:	1-cm distance	129.5	128	1.17	108	110.5	2.26
	Mean	101	99	2.02	88	89.5	1.67
PCM C, liquefied thickness (mm)		76	75	1.33	67	68	1.47

- The performance of the system was investigated by different parameters such as different PCMs, fins geometry, and two various typical climate conditions in the tropical region of South East Asia, Kuala Lumpur, that can be a good sample for other similar climate conditions.
- The present study was implemented to simulate the stored latent thermal heat energy and the melted-PCM temperature aimed at the supply output hot water using three different phase change materials (PCMs) for the typical high and low solar intensity days.
- Melting points of PCM A and B versus PCM C as well as the latent heat storage capacity of PCM A and C versus PCM B are far from each other resulting in a good comparison work.
- According to the results, PCM C, Rubitherm (RT70HC), reached the highest mean temperature of 101 °C with 2.02% engineering error (compared to the analytical study) within the 12-h melting/charging process over the typical high daily solar radiation. Moreover, its engineering error was somehow the minimum amount among the other PCMs.
- In this regard, it would be technically evident that the solid-liquid interface has the superior growth for the PCM C whereas the latent heat storage capacity of PCM C is less than PCM B that plays a significant role to choose the appropriate phase change material in the end by the user. To this end, the maximum thickness of the liquified PCM, which also depicts the most progress of the melted PCM, is for PCM C with 76 mm thickness, which is quite close to the walls.

Author statement

Shahab Bazri: Software, Data Curation, Writing - Original Draft, Writing - Review & Editing, Visualization, Project administration.

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Mohammad Sajad Naghavi: Conceptualization, Investigation, Resources.

Abdul Rahman Mallah: Validation, Formal analysis.

Ali H. Abdelrazek: Methodology, Formal analysis.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

The authors do not have permission to share data.

Acknowledgments

The authors extend their appreciation to the Deanship of Scientific Research at King Khalid University for funding this work through the Large Groups Project under grant number RGP. 2/101/43.

References

- [1] Y. Wang, K. Yu, X. Ling, Experimental and modeling study on thermal performance of hydrated salt latent heat thermal energy storage system, *Energy Convers. Manag.* 198 (2019), 111796, <https://doi.org/10.1016/j.enconman.2019.111796>, April.

- [2] A. Shafieian, M. Khadani, A. Nosrati, A review of latest developments, progress, and applications of heat pipe solar collectors, *Renew. Sustain. Energy Rev.* 95 (2018) 273–304, <https://doi.org/10.1016/j.rser.2018.07.014>. June.
- [3] T.H. Hsu, C.H. Chung, F.J. Chung, C.C. Chang, M.C. Lu, Y.L. Chueh, Thermal hysteresis in phase-change materials: encapsulated metal alloy core-shell microparticles, *Nano Energy* 51 (2018) 563–570, <https://doi.org/10.1016/j.nanoen.2018.06.021>. March.
- [4] C. Suresh, R.P. Saini, Thermal performance of sensible and latent heat thermal energy storage systems, *Int. J. Energy Res.* 44 (6) (2020) 4743–4758, <https://doi.org/10.1002/er.5255>.
- [5] V. Bhalla, H. Tyagi, Parameters influencing the performance of nanoparticles-laden fluid-based solar thermal collectors: a review on optical properties, *Renew. Sustain. Energy Rev.* 84 (2018) 12–42, <https://doi.org/10.1016/j.rser.2017.12.007>. November 2017.
- [6] H. Wang, X. Wang, H. Zhang, W. Ma, L. Wang, X. Zong, Organic–inorganic hybrid perovskites: game-changing candidates for solar fuel production, *Nano Energy* 71 (2020), 104647, <https://doi.org/10.1016/j.nanoen.2020.104647>.
- [7] J. Wang, Y. Li, D. Zheng, H. Mikulčić, M. Vujanović, B. Sundén, Preparation and thermophysical property analysis of nanocomposite phase change materials for energy storage, *Renew. Sustain. Energy Rev.* 151 (2021), <https://doi.org/10.1016/j.rser.2021.111541>. July.
- [8] I. Sarbu, A. Dorca, Review on heat transfer analysis in thermal energy storage using latent heat storage systems and phase change materials, *Int. J. Energy Res.* 43 (1) (2019) 29–64, <https://doi.org/10.1002/er.4196>.
- [9] R. Dai, J. Mostaghimi, N. Li, T. Deng, Q. Wang, M. Zeng, Charging time and energy storage rate analysis of fin effect inside the horizontal tube for thermal energy storage, *J. Clean. Prod.* 273 (2020), 123030, <https://doi.org/10.1016/j.jclepro.2020.123030>.
- [10] H. Soltani, M. Soltani, H. Karimi, J. Nathwani, Optimization of shell and tube thermal energy storage unit based on the effects of adding fins, nanoparticles and rotational mechanism, *J. Clean. Prod.* 331 (2022), 129922, <https://doi.org/10.1016/j.jclepro.2021.129922>. April 2021.
- [11] M.E.H. Amagour, M. Bennajah, A. Racheq, Numerical investigation and experimental validation of the thermal performance enhancement of a compact finned-tube heat exchanger for efficient latent heat thermal energy storage, *J. Clean. Prod.* 280 (2021), 124238, <https://doi.org/10.1016/j.jclepro.2020.124238>.
- [12] Z. Khan, Z.A. Khan, Role of extended fins and graphene nano-platelets in coupled thermal enhancement of latent heat storage system, *Energy Convers. Manag.* 224 (2020), 113349, <https://doi.org/10.1016/j.enconman.2020.113349>. June.
- [13] T.K. Abdelkader, Y. Zhang, E.S. Gaballah, S. Wang, Q. Wan, Q. Fan, Energy and exergy analysis of a flat-plate solar air heater coated with carbon nanotubes and cupric oxide nanoparticles embedded in black paint, *J. Clean. Prod.* 250 (2020), 119501, <https://doi.org/10.1016/j.jclepro.2019.119501>.
- [14] P.T. Sardari, R. Babaei-Mahani, D. Giddings, S. Yasserli, M.A. Moghimi, H. Bahai, Energy recovery from domestic radiators using a compact composite metal Foam/PCM latent heat storage, *J. Clean. Prod.* 257 (2020), 120504, <https://doi.org/10.1016/j.jclepro.2020.120504>.
- [15] M. Sheikholeslami, O. Mahian, Enhancement of PCM solidification using inorganic nanoparticles and an external magnetic field with application in energy storage systems, *J. Clean. Prod.* 215 (2019) 963–977, <https://doi.org/10.1016/j.jclepro.2019.01.122>.
- [16] J. Shi, M. Qin, W. Aftab, R. Zou, Flexible phase change materials for thermal energy storage, *Energy Storage Mater.* 41 (June) (2021) 321–342, <https://doi.org/10.1016/j.ensm.2021.05.048>.
- [17] Y. Zhang, X. Li, J. Li, C. Ma, L. Guo, X. Meng, Solar-driven phase change microencapsulation with efficient Ti407 nanoconverter for latent heat storage, *Nano Energy* 53 (2018) 579–586, <https://doi.org/10.1016/j.nanoen.2018.09.018>.
- [18] S. Zhang, D. Feng, L. Shi, L. Wang, Y. Jin, L. Tian, Z. Li, G. Wang, L. Zhao, Y. Yan, A review of phase change heat transfer in shape-stabilized phase change materials (ss-PCMs) based on porous supports for thermal energy storage, *Renew. Sustain. Energy Rev.* 135 (2021), 110127, <https://doi.org/10.1016/j.rser.2020.110127>. January 2020.
- [19] A. Buonomano, F. Calise, A. Palombo, Solar heating and cooling systems by absorption and adsorption chillers driven by stationary and concentrating photovoltaic/thermal solar collectors: modelling and simulation, *Renew. Sustain. Energy Rev.* 82 (2018) 1874–1908, <https://doi.org/10.1016/j.rser.2017.10.059>. November 2017.
- [20] H. Xu, W.Y. Lin, F. Dal Magro, T. Li, X. Py, A. Romagnoli, Towards higher energy efficiency in future waste-to-energy plants with novel latent heat storage-based thermal buffer system, *Renew. Sustain. Energy Rev.* 112 (May) (2019) 324–337, <https://doi.org/10.1016/j.rser.2019.05.009>.
- [21] A. Kumar, S.K. Saha, Experimental and numerical study of latent heat thermal energy storage with high porosity metal matrix under intermittent heat loads, *Appl. Energy* 263 (2020), 114649, <https://doi.org/10.1016/j.apenergy.2020.114649>. February.
- [22] M. Koşan, M. Aktaş, Experimental investigation of a novel thermal energy storage unit in the heat pump system, *J. Clean. Prod.* 311 (2021), <https://doi.org/10.1016/j.jclepro.2021.127607>. November 2020.
- [23] C. Ji, Z. Qin, S. Dubey, F.H. Choo, F. Duan, Three-dimensional transient numerical study on latent heat thermal storage for waste heat recovery from a low temperature gas flow, *Appl. Energy* 205 (2017) 1–12, <https://doi.org/10.1016/j.apenergy.2017.07.101>. August.
- [24] X. Lin, X. Zhang, L. Liu, J. Liang, W. Liu, Polymer/expanded graphite-based flexible phase change material with high thermal conductivity for battery thermal management, *J. Clean. Prod.* 331 (2022), 130014, <https://doi.org/10.1016/j.jclepro.2021.130014>. August 2021.
- [25] M.J. Zarei, H. Bazai, M. Sharifpur, O. Mahian, B. Shabani, The effects of fin parameters on the solidification of PCMs in a fin-enhanced thermal energy storage system, *Energies* 13 (1) (2020), <https://doi.org/10.3390/en13010198>.
- [26] Y.H. Diao, L. Liang, Y.H. Zhao, Z.Y. Wang, F.W. Bai, Numerical investigation of the thermal performance enhancement of latent heat thermal energy storage using longitudinal rectangular fins and flat micro-heat pipe arrays, *Appl. Energy* 233–234 (100) (2019) 894–905, <https://doi.org/10.1016/j.apenergy.2018.10.024>.
- [27] C. Guo, W. Zhang, Numerical simulation and parametric study on new type of high temperature latent heat thermal energy storage system, *Energy Convers. Manag.* 49 (5) (2008) 919–927, <https://doi.org/10.1016/j.enconman.2007.10.025>.
- [28] S. Bazri, I.A. Badruddin, M.S. Naghavi, O.K. Seng, S. Wongwises, An analytical and comparative study of the charging and discharging processes in a latent heat thermal storage tank for solar water heater system, *Sol. Energy* 185 (2019) 424–438, <https://doi.org/10.1016/j.solener.2019.04.046>. January.
- [29] A. Jaber Kosroshahi, S. Hossainpour, A numerical investigation on the finned storage rotation effect on the phase change material melting process of latent heat thermal energy storage system, *J. Energy Storage* 55 (PA) (2022), 105461, <https://doi.org/10.1016/j.est.2022.105461>.
- [30] S. Tiari, A. Hockins, An experimental study on the effect of annular and radial fins on thermal performance of a latent heat thermal energy storage unit, *J. Energy Storage* 44 (PB) (2021), 103541, <https://doi.org/10.1016/j.est.2021.103541>.
- [31] C.L. Ladekar, S.K. Chaudhary, S.S. Khandare, Experimental investigation for optimization of heat pipe performance in latent heat thermal energy storage, *Mater. Today Proc.* 4 (8) (2017) 8149–8157, <https://doi.org/10.1016/j.matpr.2017.07.156>.
- [32] B. Eanest Jebasingh, A. Valan Arasu, A comprehensive review on latent heat and thermal conductivity of nanoparticle dispersed phase change material for low-temperature applications, *Energy Storage Mater.* 24 (2020) 52–74, <https://doi.org/10.1016/j.ensm.2019.07.031>. May 2019.
- [33] R. Raud, S. Bell, T.C. Ong, G. Will, T.A. Steinberg, Optimized salt selection for solar thermal latent heat energy storage, *Advanced Sustainable Systems* 2 (11) (2018), <https://doi.org/10.1002/adsu.201800074>.
- [34] S. Tiari, S. Qiu, M. Mahdavi, Numerical study of finned heat pipe-assisted thermal energy storage system with high temperature phase change material, *Energy Convers. Manag.* 89 (2015) 833–842, <https://doi.org/10.1016/j.enconman.2014.10.053>.
- [35] M. Zeneli, I. Malgarinos, A. Nikolopoulos, N. Nikolopoulos, P. Grammelis, S. Karellas, E. Kakaras, Numerical simulation of a silicon-based latent heat thermal energy storage system operating at ultra-high temperatures, *Appl. Energy* 242 (2019) 837–853, <https://doi.org/10.1016/j.apenergy.2019.03.147>. March 2018.
- [36] T. Zhou, Y. Xiao, H. Huang, J. Lin, Numerical study on the cooling performance of a novel passive system: cylindrical phase change material-assisted earth-air heat exchanger, *J. Clean. Prod.* 245 (2020), 118907, <https://doi.org/10.1016/j.jclepro.2019.118907>.
- [37] T. Xu, E.N. Humire, J.N.W. Chiu, S. Sawalha, Numerical thermal performance investigation of a latent heat storage prototype toward effective use in residential heating systems, *Appl. Energy* 278 (2020), 115631, <https://doi.org/10.1016/j.apenergy.2020.115631>. July.
- [38] M.S. Naghavi, *Theoretical and Experimental Investigation on Heat Pipe Solar Collector Integrated with Latent Heat Thermal Energy Storage* Mohammad Sajad Naghavi Sanjani Thesis Submitted in Fulfilment of the Requirements for the Degree of Doctor of Philosophy Faculty of, PhD Thesis, (University of Malaya), 2016.

- [39] S. Bazri, I.A. Badruddin, M.S. Naghavi, O.K. Seng, S. Wongwises, An analytical and comparative study of the charging and discharging processes in a latent heat thermal storage tank for solar water heater system, *Sol. Energy* 185 (2019) 424–438, <https://doi.org/10.1016/j.solener.2019.04.046>. January.
- [40] S. Bazri, I.A. Badruddin, M.S. Naghavi, M. Bahiraei, A review of numerical studies on solar collectors integrated with latent heat storage systems employing fins or nanoparticles, *Renew. Energy* 118 (2018) 761–778, <https://doi.org/10.1016/j.renene.2017.11.030>.
- [41] D. Zhang, H. Tao, M. Wang, Z. Sun, C. Jiang, Numerical simulation investigation on thermal performance of heat pipe flat-plate solar collector, *Appl. Therm. Eng.* 118 (2017) 113–126, <https://doi.org/10.1016/j.applthermaleng.2017.02.089>.
- [42] M.S. Naghavi, K.S. Ong, I.A. Badruddin, M. Mehrli, H.S.C. Metselaar, Thermal performance of a compact design heat pipe solar collector with latent heat storage in charging/discharging modes, *Energy* 127 (2017) 101–115, <https://doi.org/10.1016/j.energy.2017.03.097>.