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## Heat transfer and energy storage performances of phase change materials encapsulated in honeycomb cells

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1           **Heat Transfer and Energy Storage Performances of Phase Change**  
2           **Materials Encapsulated in Honeycomb Cells**

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15 Thermal energy storage devices are vital for reducing the inconsistency between energy  
16 supply and demand as well as for enhancing the performance of solar thermal systems. The  
17 present study investigates the melting process in metallic honeycombed heat exchangers  
18 filled with n-octadecane as phase change material (PCM). Further, a parametric study is  
19 conducted for four different honeycomb cell sizes, fin thicknesses, and angles of inclinations  
20 of the computational domain. The findings are described in the form of heated surface  
21 average temperature variation, melt fraction of the PCM, and thermal energy stored. Results  
22 show that the honeycomb fin structure considerably improves the heat transfer in the PCM.  
23 Further, it was observed that the honeycomb cell size and fin thickness greatly influence the  
24 variation of the average front surface temperature, melt fraction, and energy storage rate,  
25 whereas the inclination angle doesn't have a significant effect when cell size is less than 0.01  
26 m. From the study, it was also observed that when the computational domain is horizontal  
27 without honeycomb cells, the pure conduction in the PCM results in much higher front  
28 surface temperature, which is greatly reduced in presence of the honeycomb cells.

29 Keywords: Phase change materials; Thermal energy storage; Numerical simulation; Heat  
30 transfer; Melt fraction

31 Nomenclature

32	$C_p$	specific heat [J/(kgK)]
33	$h_{\infty}$	heat transfer coefficient on the front and back surface [W/(m <sup>2</sup> K)]
34	$H$	height of computational domain [m]
35	$k$	thermal conductivity [W/(mK)]
36	$L_c$	cell length [m]
37	$L_f$	latent heat of PCM [J/kg]
38	$P$	pressure [Pa]
39	$Pe$	peclet number [1]
40	$q_0$	incident heat flux on the front surface [W/m <sup>2</sup> ]
41	$t$	time [s]
42	$T$	temperature [K]
43	$T_{\infty}$	ambient temperature [K]
44	$T_{ini}$	initial temperature [K]
45	$u$	velocity field of melted PCM[m/s]
46	$W$	width of the computational domain [m]
47	$\delta_{fin}$	fin thickness [m]
48	$\theta$	tilt angle from horizontal [° ]
49	$\rho$	density [kg/m <sup>3</sup> ]
50	$\rho_l$	density of liquid PCM [kg/m <sup>3</sup> ]
51	$\rho_s$	density of solid PCM [kg/m <sup>3</sup> ]
52	$\beta$	thermal expansion coefficient of PCM [1/K]
53	$\mu$	viscosity of melted PCM [Pa.s]
54	Subscript	
55	$s$	solid
56	$l$	liquid

## 57 1. INTRODUCTION

58 Solar energy is an abundantly available, clean, and sustainable energy source therefore it is  
59 accepted as one of the most favorable alternative energy sources for sustainable development  
60 [1–5]. Due to the intermittency by its nature, unreliability is the major impeding factor for the  
61 massive use of solar energy in agriculture and food processing. The unreliability of solar  
62 energy utilization can be diminished by stowing solar energy when it is excess compared to  
63 the demand and consuming the stowed energy whenever required. The energy storage system  
64 is, consequently, important to any system that is based largely on solar energy. Solar energy  
65 can be stored in the form of thermal energy in thermal energy storage (TES) devices. During  
66 the past few decades, the latent heat TES devices using PCM have gained larger attention due  
67 to their favorable thermophysical properties. However, these materials show poor thermal  
68 conductivity, which restricts the thermal transport in the storage medium and alleviates the  
69 melting/solidification process. To evade this shortcoming and enhance thermal transport  
70 throughout the solid-liquid phase transition, multitudinous approaches have been  
71 recommended in past studies. These approaches are the addition of highly thermal conductive  
72 nanoparticles [6–8], incorporation of metal matrix [9], addition porous matrices [10], and use  
73 of multiple PCMs [11,12]. Amid these approaches, the incorporation of the metal matrix in  
74 PCM is the focus of the present study because it is cheap and easy to integrate with the  
75 system.

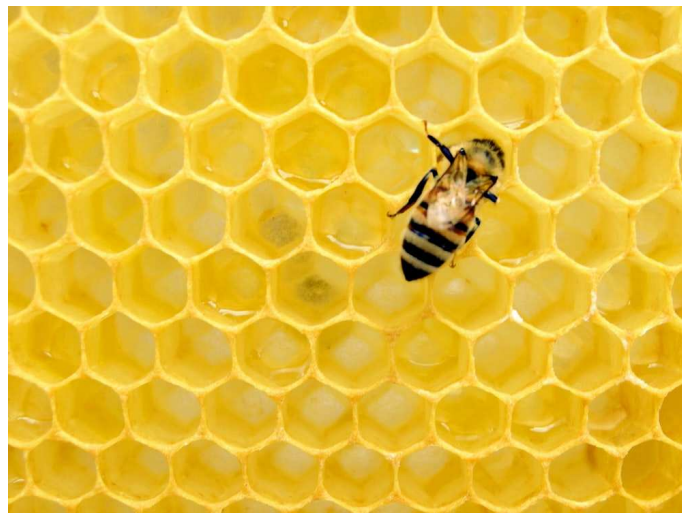
76 In recent years, several studies have been executed to assess the heat transfer behavior of  
77 metal matrices integrated PCMs. In this regard, Biwole et al. [13] conducted a numerical  
78 simulation of RT-25 PCM filled in a rectangular enclosure to assess the effect of fin size and  
79 distribution. The effective fin configuration was investigated and it was perceived that the fin  
80 surface area is an important factor that guides the melting of PCM and the rate of heat  
81 transfer between the PCM and the front hot surface, while the fin structures advance greater  
82 natural convection, leading to an advanced heat flux. Groulx et al. [14] conducted a  
83 numerical simulation to assess the influence of fin placement and angle of inclination on the  
84 melting of RT25 PCM in a rectangular enclosure. Based on the study it was concluded that  
85 the convection heat transfer causes a significant acceleration in the melting process of the  
86 PCM. Also, a negligible effect for the computational domain's inclination angle smaller than  
87 75° from vertical on the heat transfer behavior of all considered configurations was reported.  
88 Darzi et al. [15] studied the melting and solidification behavior of PCM considering radial

89 fins and thermally conductive nanoparticles to enhanced thermal transport. The phase change  
90 process was accelerated by increasing the nanoparticle concentration and the elliptical tube  
91 decreases the melting time as compared to the circular tube. Further, increment the fin  
92 number show system deficient due to the suppression of natural convection. A numerical  
93 study of PCM solidification was conducted by Abidi et al [16] to investigate the best  
94 configuration that minimizes the melting and solidification time of PCM in a triplex tube heat  
95 exchanger with both external and internal fins. In continuation with the thermal transport  
96 enhancement using fin structure, a novel snowflake crystal fin structure was investigated by  
97 Sheikholeslami et al. [17] to enhance the thermal efficiency of the latent heat thermal energy  
98 storage (LHTES) device during the heat discharging process. The obtained results were  
99 indicated that the discharging process of the LHTES is improved by applying Snowflake  
100 shaped fin structure, in comparison with the nanoparticle diffusion, while it does not  
101 considerably reduce the maximum thermal energy storage capacity. Wu et al. [18] performed  
102 numerical analysis and improved the thermal transport behavior of the LHTES device with  
103 spiderweb-like fins. From the obtained results it is observed that the spiderweb-like fins  
104 decrease melting and solidification time by reducing the heat transfer hysteresis region.

105 In the recent past, several investigators have made substantial efforts to improve the thermal  
106 performance of LHTES devices by using different thermally conductive fin structures. The  
107 researchers had paid their most attention toward modified rectangular and annular fin  
108 structures. Conversely, several studies have shown that proposed conventional fins have a  
109 simple structure with a small specific surface area which makes it tough to realize reasonable  
110 thermal transport performance of LHTES devices. Consequently, some fins with mixed  
111 structures and a high specific surface have been found to enhance the thermal transport  
112 effectiveness of LHTES devices. Recently, bionics are advanced in the field of fluid flow and  
113 heat transfer applications including chemical reactor [19], biomedical equipment [20,21], and  
114 electronic cooling [22–26]. The fin structures made by natural assortment encourages the  
115 improvement of the LHTES device. By realizing natural structures, the fins with bionic  
116 structures for instance tree-shaped fins [26,27], Koch fractal fins [28], and snowflake-shaped  
117 fins [17,29] have been recently proposed by researchers to the enhanced thermal performance  
118 of PCM thermal energy storage devices.

119 The bionic fins provide an effective means for thermal transport, therefore increasing the  
120 thermal transport effectiveness of LHTES devices. Though the hysteresis exists during

121 melting and solidification of PCM with various ranges depends on the gaps between fins. The  
122 presence of hysteresis impedes the additional enhancement of the thermal efficiency of such  
123 devices. The extensive existence of honeycomb meshed structure in nature offers an idea to  
124 solve the hysteresis problem. As shown in *Figure 1*, the honeycomb has a compact and  
125 uniformly distributed structure over the entire structure. Based on this quality, the use of  
126 honeycomb-like fin introduces a high interest for improved efficiency of charging and  
127 discharging processes of LHTES devices.



128

*Figure 1 Honey bee honeycomb*

129

130 Through a comprehensive literature assessment, it can be chronicled that not much  
131 investigation has been dedicated to investigating the charging efficiency of LHTES devices  
132 with honeycomb meshed fins. The melting behavior of PCM and interstitial thermal energy  
133 absorption process between honeycomb meshed fin structures and PCM is still imprecise.  
134 Further, the effect of the honeycomb meshed fin structure on the charging efficiency of  
135 LHTES devices has not been exposed so far. The effect of natural convection on the melting  
136 progression has been highlighted by numerous researchers; however, no known study has  
137 been found in the literature on the melting and energy storage behavior of PCM in the  
138 honeycomb structure and limited works have focused on the effect of the container's tilt  
139 angle on the melting behavior of the PCM process. Consequently, the current analysis  
140 provides an advanced insight into how the thorough nature of honeycomb meshed fins  
141 influence the heat absorption behavior of LHTES devices while can be considered as an  
142 important work to enhance the thermal storage and discharge effectiveness.

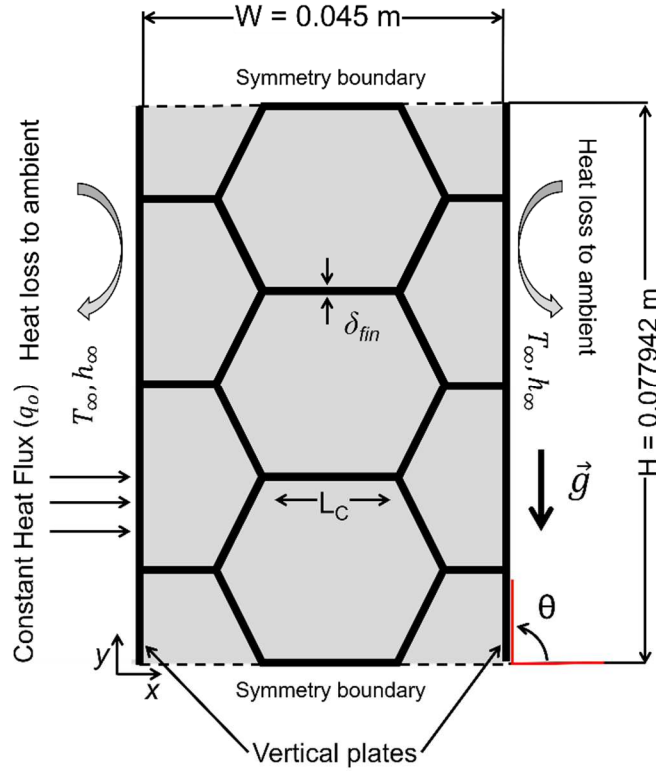
143 The article is structured as follows: In the introduction section, the overall background and  
144 the goal of the current study are presented. After the introduction, the computational model  
145 with all the simulation details including the dimensions and constants used in the study are  
146 described. Next to the computational model, the mathematical formulation, boundary  
147 conditions, and computational technique with model validation are presented. Further,  
148 detailed discussions on the results of the present study are presented, trailed by a summary of  
149 the key conclusions.

## 150 2. COMPUTATIONAL MODEL

151 The geometry of the computational domain considered in the present work has been  
152 presented in Figure 2. The computational domain is poised of two vertical metallic plates  
153 connected with hexagonal honeycomb cells; the fin thickness of each hexagonal cell, as well  
154 as the vertical plate, is  $\delta_{fin}$ . The height ( $H$ ) and width ( $W$ ) of the container are fixed, with  
155 values 0.077942 m and 0.045 m respectively and the inclination angle of the computational  
156 domain from horizontal is  $\theta$  ° as shown in Figure 2. Each cell in the container has the same  
157 length ( $L_C$ ) and uniform thickness. Internal fins are also made of the same material as front  
158 and back vertical plates. The cell length  $L_C$ , the inclination angle of the container from  
159 horizontal  $\theta$  and the fin thickness are given in Table 2. The thermo-physical properties of  
160 metallic fins and PCM considered in the present study are shown in Table 1. The  
161 computational domain is primarily set at a temperature  $T_{ini} = 23.2$  °C and the front metallic  
162 plate is subjected to a continuous heat flux of  $q_0 = 1000 W/m^2$  perpendicular to the front  
163 surface for the period of the numerical simulation. As the present study is only focused on the  
164 influence of honeycomb structure on the heat transfer behavior in PCM, the convective heat  
165 transfer coefficient on the front and back surface is fixed. The front and back surface of the  
166 container is opened to convection cooling with an ambient temperature of  $T_\infty = 23.2$  °C as  
167 well as the convection heat transfer coefficient  $h_\infty = 2 W/m^2K$ . The top and bottom of the  
168 computational domain are subjected to symmetry boundaries (adiabatic). The acceleration  
169 due to gravity is always working in the downward direction for the entire tilt angle range. To  
170 conduct numerical simulation the following assumptions has been considered in the present  
171 study: (i) the liquated PCM is incompressible and Newtonian, (ii) the melted PCM's flow is  
172 laminar, radiation and three-dimensional convection are neglected, (iii) volume expansion  
173 due to the melting of PCM is neglected.



174 In the present study, n-octadecane is considered as PCM and its thermo-physical properties  
175 are given in Table 1. Four different angles of inclination and fin thicknesses are studied for  
176 four different honeycomb cell sizes. The study has also been conducted without honeycomb  
177 cells in the rectangular plane of the container considered. The studied combinations of the fin  
178 thickness and inclinational angles are given in Table 2, where  $L_C$  is the honeycomb cell  
179 length,  $\delta_{fin}$  is the fin thickness of the honeycomb cells and  $\theta$  is the angle of inclination of the  
180 computational domain from horizontal as shown in Figure 2. The numerical simulation was  
181 conducted at 4 different angles of inclination  $\theta$  without honeycomb cells as well as at 4  
182 different  $L_C$  (i.e.  $L_C = 0.005$  m,  $L_C = 0.0075$  m,  $L_C = 0.010$  m and  $L_C = 0.015$  m) considering  
183  $\delta_{fin} = 0.0005$  m to investigate the effect of inclination angle as well as honeycomb cell length  
184 on front surface temperature, melt fraction, and energy storage content. Further, to investigate  
185 the effect of the honeycomb fin thickness, the studies are conducted at 4 different honeycomb  
186 fin thickness  $\delta_{fin}$  with 4 honeycomb cell length  $L_C$  while the angle of inclination of the  
187 computational domain is fixed at  $60^\circ$ . Through the change in honeycomb cell size, fin  
188 thickness, tilt angle from horizontal, initial, and boundary conditions to the mathematical  
189 model are described in the next section and thermal energy storage behavior of computational  
190 domain is described arrangements are described.



191

192 *Figure 2 Geometry and boundary conditions of the system considered for the numerical*  
 193 *simulation*

194 *Table 1 Thermophysical properties of PCM and metallic fin*

Properties /Material	PCM	Metallic fin
Melting temperature ( $^{\circ}C$ )	28.2	NA
Specific heat ( $J/kg.K$ )	1934 (s), 2196 (l)	900
Latent heat of fusion ( $kJ/kg$ )	245	NA
Density ( $kg/m^3$ )	814 (s), 775 (l)	2700
Thermal conductivity ( $W/m.K$ )	0.350 (s), 0.149 (l)	238
Thermal expansion coefficient ( $1/^{\circ}C$ )	$9.1 \times 10^{-4}$	NA
Kinematic viscosity ( $m^2/s$ )	$5 \times 10^{-6}$	NA

195

196 *Table 2 Parameters considered in the present study*

	$\theta$ $^{\circ}$ (at $\delta_{fin} = 0.0005$ m)				$\delta_{fin}$ [m] (at $\theta = 60$ $^{\circ}$ )			
Without cells	0	30	60	90	NA	NA	NA	NA
$L_C = 0.005$ m	0	30	60	90	0.00025	0.0005	0.001	0.0015
$L_C = 0.0075$ m	0	30	60	90	0.00025	0.0005	0.001	0.0015

$L_c = 0.010 \text{ m}$	0	30	60	90	0.00025	0.0005	0.001	0.0015
$L_c = 0.015 \text{ m}$	0	30	60	90	0.00025	0.0005	0.001	0.0015

### 197 3. MATHEMATICAL FORMULATIONS

#### 198 3.1. Numerical simulation model description

199 The following equations (1) and (2) with the boundary conditions are applied on the heated  
200 front and back external surface correspondingly:

$$201 \quad -k \left. \frac{\partial T}{\partial x} \right|_{x=0} = q_0 + h_{\infty}(T_{\infty} - T) \quad (1)$$

$$202 \quad -k \left. \frac{\partial T}{\partial x} \right|_{x=\delta_{fin}+W+\delta_{fin}} = h_{\infty}(T - T_{\infty}) \quad (2)$$

203 Further, the equation of heat transfer diffusion realized within the fins and PCM domain is:

$$204 \quad \rho C_p \frac{\partial T}{\partial t} + \nabla(-k\nabla T) + \rho C_p \vec{u} \cdot \nabla T = 0 \quad (3)$$

205 where  $\rho$ ,  $C_p$  and  $k$  represents density, specific heat, and thermal conductivity respectively. For  
206 the PCM, the value of  $\rho$ ,  $k$  and  $C_p$  varies with temperature as given by eq. (7), (8) and (9)  
207 respectively. The fluid velocity  $\vec{u}$  is always zero in the metallic fins as well as on the surface  
208 of metallic fins, and is given in the PCM domain by numerically solving the Navier-Stokes  
209 and continuity equation; the equation of modified momentum conservation for the PCM  
210 domain can be given [14]:

$$211 \quad \rho \frac{\partial \vec{u}}{\partial t} + \rho(\vec{u} \cdot \nabla)\vec{u} = -\nabla P + \rho_l(1 - \beta(T - T_m))\vec{g} + \mu_l \cdot \nabla^2 \vec{u} - \frac{C(1-B(T))^2}{B^3(T)+\varepsilon} \vec{u} \quad (4)$$

212 where  $\rho_l$  and  $\mu_l$  are respectively the density and dynamic viscosity is of melted PCM,  $\beta$  is the  
213 thermal expansion coefficient of PCM and  $\vec{g}$  is the acceleration because of gravity constant.  
214 The value of constant C (Kozeny-Carman permeability) can be determined by the inherent  
215 characteristics of the PCM material. Increasing its value causes a decrease in the melted  
216 PCM's flow in the transition zone, and consequently to a smoother nature of the liquidus  
217 melting front while decreasing its value will have the contrary influence. The value of  
218 constant C in the present study is  $10^5 \text{ kg/ (m}^3 \cdot \text{s)}$ . Here,  $\varepsilon$  is constant with a given value of

219  $10^{-3}$  as in [30] to evade dividing by zero. The melted fraction function  $B(T)$  of PCM is  
 220 defined as:

$$221 \quad B(T) = \begin{cases} 0, & T < \left(T_m - \frac{\Delta T}{2}\right) \\ \frac{T - T_m + \Delta T/2}{\Delta T}, & \left(T_m - \frac{\Delta T}{2}\right) < T < \left(T_m + \frac{\Delta T}{2}\right) \\ 1, & T > \left(T_m + \frac{\Delta T}{2}\right) \end{cases} \quad (5)$$

222 Here  $\Delta T$  is transition temperature and its value is taken  $2^\circ\text{C}$  in the present study. The varying  
 223 thermo-physical properties of PCM are modeled as a function of temperature through Eqs.  
 224 (6)-(10). The viscosity  $\mu$  of PCM is adjusted to ensure a very high value when the PCM is in  
 225 the solid-state [13].

$$226 \quad \mu(T) = \mu_l \left(1 + \frac{C(1-B(T))^2}{C_0(B^3(T)+\varepsilon)}\right) \quad (6)$$

$$227 \quad \rho(T) = \rho_s + (\rho_l - \rho_s) \cdot B(T) \quad (7)$$

$$228 \quad k(T) = k_s + (k_l - k_s) \cdot B(T) \quad (8)$$

229 where the value of  $C_0$  is taken  $1 \text{ kg}/(\text{m}^3 \cdot \text{s})$  to make sure unit consistency,  $\rho_s$  and  $\rho_l$  are the  
 230 density of the solid and liquid phases of PCM, respectively. Similarly, the  $k_s$  and  $k_l$  are the  
 231 PCM's thermal conductivity for solid and liquid phases respectively. The specific heat of  
 232 PCM which is temperature-dependent modeled including an extra term which accounts for  
 233 the latent heat of fusion stored or released in the melting or solidification process,  
 234 consequential in an adjusted heat capacity of PCM can be given by [13]:

$$235 \quad C_p(T) = C_{p_s} + (C_{p_l} - C_{p_s}) \cdot B(T) + L_f D(T) \quad (9)$$

236 where

$$237 \quad D(T) = \frac{e^{-\frac{(T-T_m)^2}{(\Delta T/4)^2}}}{\sqrt{\pi(\Delta T/4)^2}} \quad (10)$$

238 The reason for considering the Gaussian function  $D(T)$  in eq. (9) is conserve the latent heat of  
 239 PCM all over the transition period, because the integral of function  $D(T)$  over the whole  
 240 range of transition temperature  $\Delta T$  is always 1 [31]. The consideration of  $\Delta T$  produces a

241 numerical mushy region. The mushy region thickness grows with the growth in the value of  
242  $\Delta T$ . The value of  $\Delta T$  depends on the nature of the materials and can be fixed empirically if  
243 one has previous information on the melting behavior of PCM. The deformation of geometry  
244 owing to the PCM's thermal expansion is not considered in the present study.

### 245 *3.2. Computational method, meshing and validation of the model*

246 The above-mentioned equations are implemented in a fluid and laminar flow module of  
247 COMSOL Multiphysics 5.5. The present study used fixed-grid approaches similar to the one  
248 labeled in this section that consents to evade the necessity to solve the Stefan problem at the  
249 varying melting front. Furthermore, the melting front tracing method necessitates an already  
250 existed liquid layer, which is not the situation for the present study. Through fixed-grid  
251 approaches, the standard method for the liquid flow and energy for instance the finite element  
252 numerical resolution of N-S equations can be applied with partial modifications.

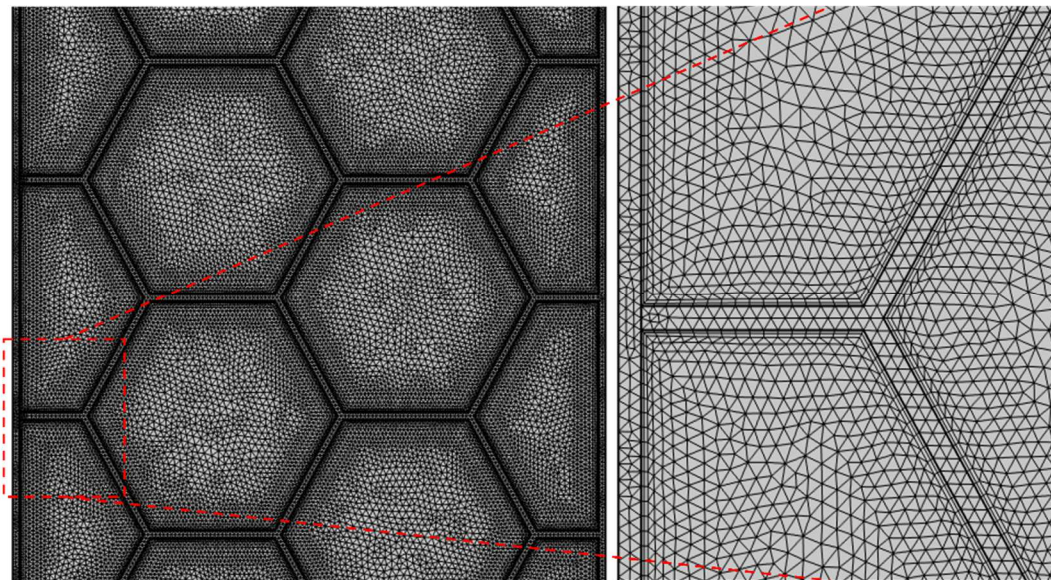
253 To solve the problem, an extremely fine mesh is chosen with a maximum and minimum size  
254 of the element of value  $5.98 \times 10^{-4}$  m and  $6.9 \times 10^{-6}$  m respectively, which results in the total  
255 number of 62716, 72050, 80806, and 95166 linear free triangular mesh elements for  $L_C =$   
256 0.015 m,  $L_C = 0.01$  m,  $L_C = 0.0075$  m and  $L_C = 0.0075$  m respectively. The maximum mesh  
257 size is defined using a condition on the mass mesh Peclet number [31]:

$$258 \quad Pe = \frac{\bar{u}\Delta x}{\vartheta} < 2 \quad (11)$$

259 where  $\Delta x$  is the maximum length of mesh cell in a unit 2D surface. From numerical  
260 simulations, the maximum velocity in the honeycomb cell is taken as  $5 \times 10^{-3}$  m/s. Therefore,  
261 Eq. (11) gives  $\Delta x \approx 2 \times 10^{-3}$  m. In the present study,  $Pe$  is kept lower than 1 to increase  
262 numerical stability. Consequently, the maximum mesh size is  $1 \times 10^{-3}$  m. The mesh is refined  
263 at the borders of the PCM domain (see Figure 3). The first layer thickness of the boundary  
264 layer is  $5.22 \times 10^{-5}$  m. Mesh sensitivity tests were also conducted at three increasing mesh  
265 element sizes and the transient average temperature of the container's front surface was  
266 observed at 38202, 50836, and 62716 mesh elements, as presented in Figure 4 for  $L_C = 0.015$   
267 m. For the other  $L_C$ , the maximum and minimum mesh sizes are the same, therefore the  
268 change in the mesh size will not affect the output results from the model. From this figure, it  
269 is observed that there is a negligible change in the variation of the front surface temperature

270 when increasing the number of mesh elements. Therefore, for the sake of accuracy, maximum  
271 numbers of mesh elements were selected for the present study.

272 The numerical model presented above has been experimentally validated with our previous  
273 experimental study by Biwole *et al.* [31] and a comparison of the temperature variation is  
274 presented in *Figure 5*. For more details about the experimental setup and PCM properties,  
275 etc. please refer to Biwole *et al.* [31]. The model explained above has also been validated  
276 with the experimental results reported by Shokouhmand and Kamkari [32] on the Lauric  
277 acid's melting in another previous study conducted by Biwole *et al.* [13]. The verification of  
278 the model with previous experimental results provides the confidence for the detailed  
279 systematic parametric studies of honeycomb geometry for enhanced thermal transport in  
280 PCM.

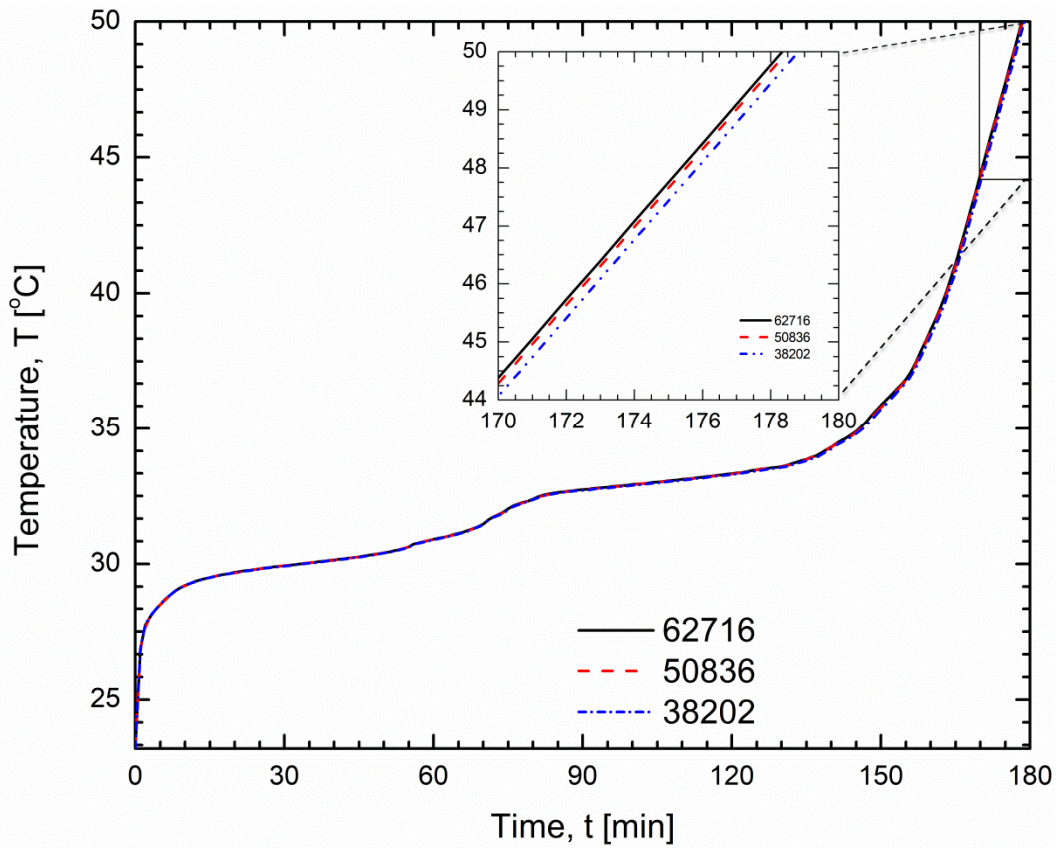


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*Figure 3 Meshing of the computational domain*

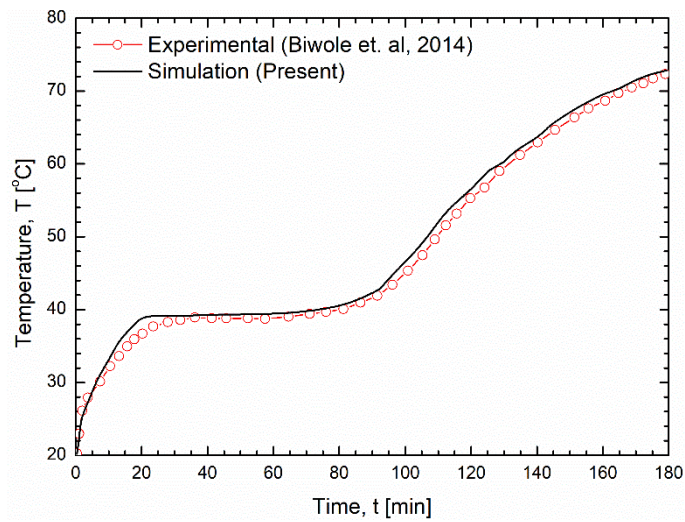
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284

*Figure 4 Mesh sensitivity test of the model*



285

286

*Figure 5 Validation of numerical model with experimental study of Biwole et al [31]*

287

#### 4. RESULTS AND DISCUSSION

288

The findings of this study are described in four different subsections. In the first one, the

289

temperature variation in each cell and the whole computational domain, as well as the

290 observed evolution of the melt fraction are discussed, for a single honeycomb cell size and  
 291 container's angle of inclination. In the next subsections, the effect of honeycomb cell size,  
 292 container tilt angle, and honeycomb fin thickness are discussed. In the present study three  
 293 factors are used to evaluate the efficiency of charging and discharging of PCM: (i) the  
 294 average temperature of the front surface, which must keep on the lowest possible value for  
 295 the longest possible time, (ii) the melt fraction of the PCM with time, (iii) the time-dependent  
 296 energy storage content with a 1,000 W/m<sup>2</sup> constant heat flux on the front surface of the  
 297 computational domain.

298 The melt fraction of the PCM in the computational domain is defined as:

$$299 \quad \phi = \frac{\text{Volume of the melted PCM } (T > (T_m + \Delta T/2))}{\text{Total volume of PCM}} \quad (11)$$

300 The energy stored in the PCM is calculated based on the temperature of the PCM.

$$301 \quad \text{for } T < T_m: E = mC_p(T - T_{ini}) \quad (12)$$

$$302 \quad \text{for } T = T_m: E = m\{C_p(T_m - T_{ini}) + L_f\} \quad (13)$$

$$303 \quad \text{for } T > T_m: E = m\{C_p(T_m - T_{ini}) + L_f + C_p(T - T_m)\} \quad (14)$$

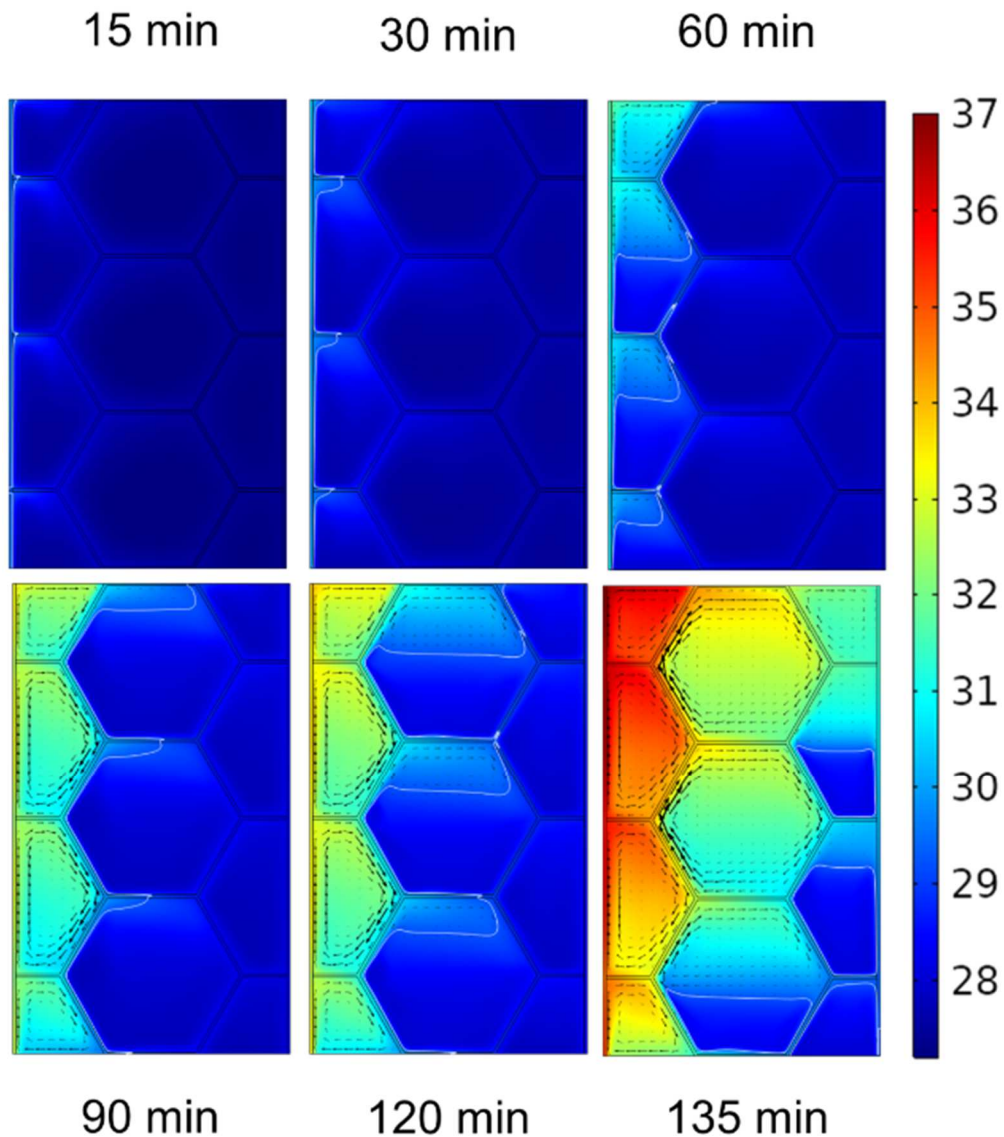
304 Here  $E$  is energy storage content and  $m$  is the mass of the PCM filled in the computational  
 305 domain. The amount of energy stored in the melted PCM is computed as the energy stowed  
 306 in entire numerical elements plus the computational domain.

#### 307 *4.1. Melting of PCM in honeycomb cells*

308 The PCM's melting in honeycomb cells is presented in Figure 6, at different time intervals  
 309 with 0.015 m cell length and 0.0005 m honeycomb fin thickness. The color contours  
 310 represent the variation of temperature, arrows show the velocity field, and the white line  
 311 denotes the melting front of the PCM. The heat flux applied to the front (left) surface of the  
 312 computational domain heats the PCM filled in the cells near the front surface. The PCM starts  
 313 melting and melted PCM moves upward due to the buoyancy force and collects at the top of  
 314 each cell. It can also be noted that the thermal behavior on each cell is related to that of its  
 315 vertical neighbors. Despite the fact that they receive the same normal heat flux from the left,



316 the cells situated at the top of the container heat up faster than the cells situated at the bottom.  
 317 The warmer liquid PCM collected at the top of each cell, thus warming the cell directly  
 318 above, also explains this. For all these reasons, a double temperature gradient is observed  
 319 from top to bottom both in each cell and in the PCM container. As expected, a horizontal  
 320 temperature gradient in the container is also noted as the heat propagates from the heated  
 321 plate.



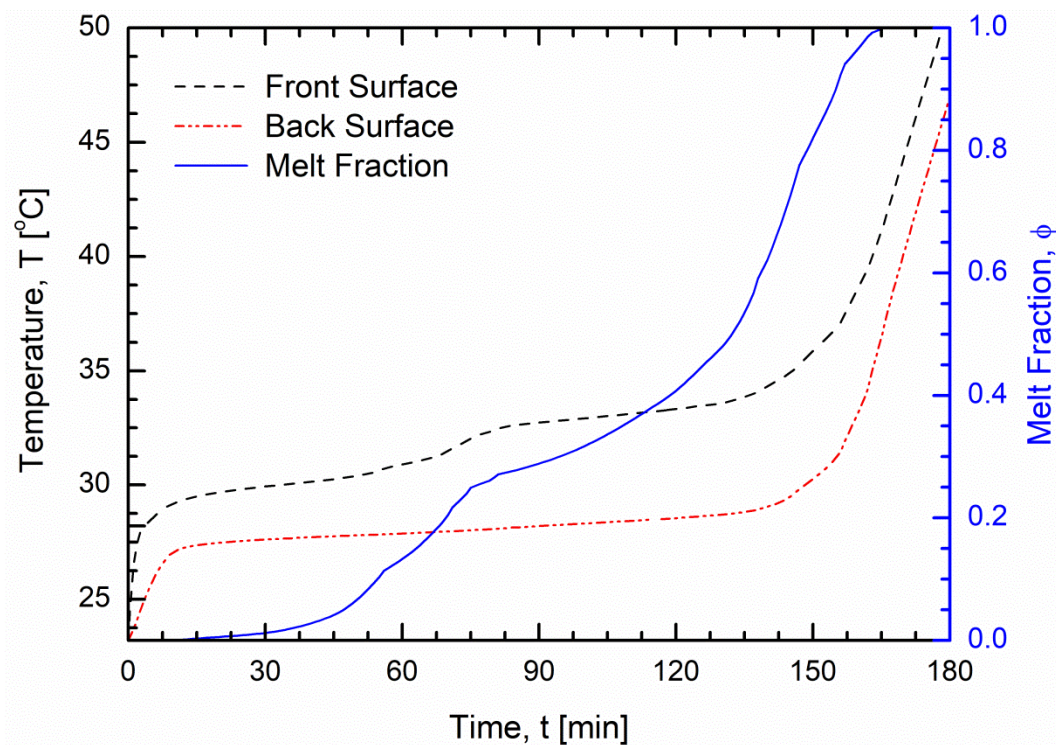
322

323 *Figure 6 Temperature field, melting front (white contours) and moving direction (black*  
 324 *arrows) of n-octadecane in honeycomb cells at different times*

325 The average temperature variation of the front and back surfaces, as well as the variation of  
 326 melt fraction ( $\phi$ ) with time, are presented in Figure 7. The two inflexion points noted on the

327 curves of each surface average temperature are explained by the start and end of the PCM  
 328 melting in the container, as obvious when comparing with the melt fraction curve. At first,  
 329 the average front surface temperature increases sharply up to the melting temperature of the  
 330 PCM. Then, it increases more gradually until the melt fraction of the PCM reaches its  
 331 maximum value (i.e. 1). It resumes increasing sharply again as the heat storage is done in the  
 332 form of sensible heat.

333 From  $t = 60$  min to  $t = 90$  min, a sudden increase in the front surface temperature of the  
 334 computational domain is observed because of the complete melting of the cell near the front  
 335 surface, which affects the melting of the PCM in the other cells of the container. At the same  
 336 time, it was observed that there is a gradual increment in the melt fraction of the PCM  
 337 compared to the initial stage of melting. The average back surface temperature gradually  
 338 increases from the start of melting until the complete melting of PCM. Except at the start and  
 339 end of PCM melting, there is no sudden increment or decrement in its temperature variation.

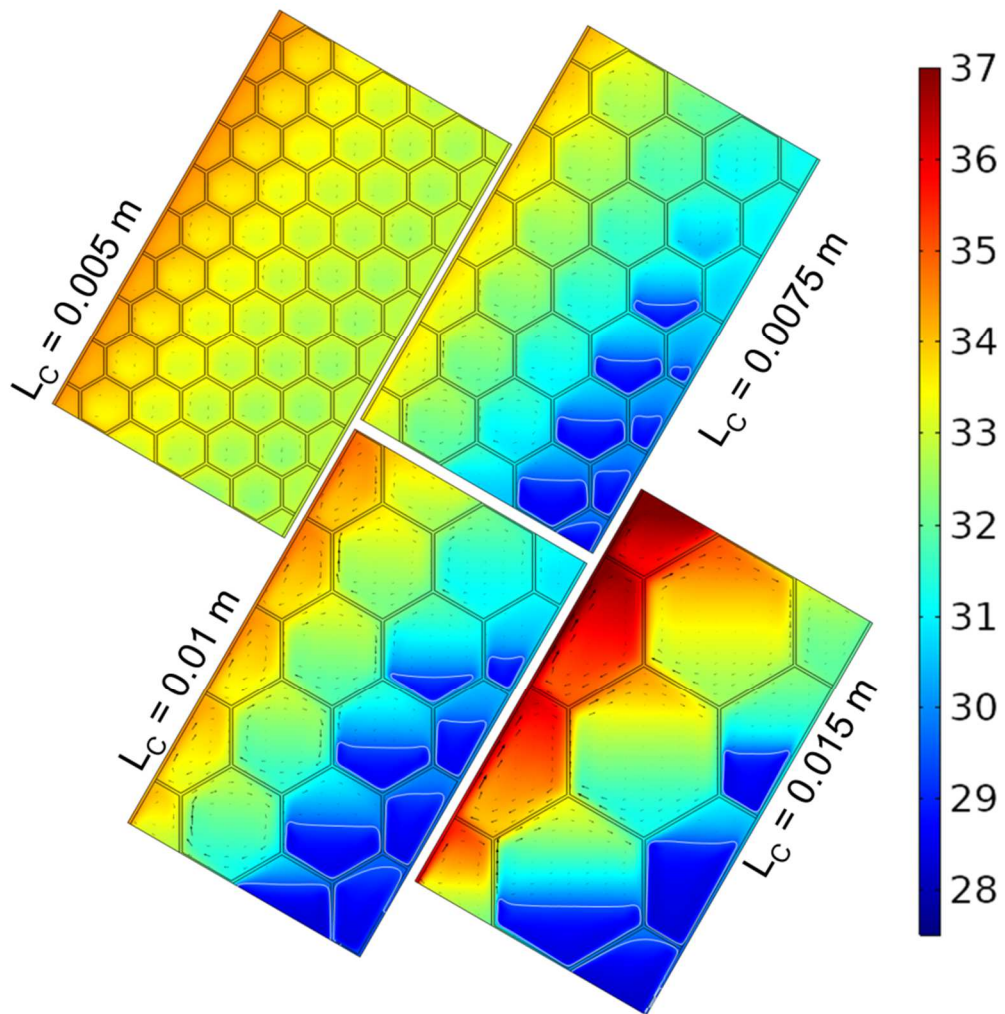


340

341 *Figure 7 Variation of melt fraction ( $\phi$ ) and average front and back surface temperature of*  
 342 *the container*

343 4.2. Effect of honeycomb cell size

344 Figure 8 represents the temperature field, the melting front and shows the moving direction of  
345 the molten PCM in the computational domain for four different honeycomb cell sizes, at  $t =$   
346 150 min for  $\delta_{fin} = 0.0005$  m and  $\theta = 60^\circ$ . For the smallest honeycomb cell size ( $L_c = 0.005$   
347 m), the PCM is completely melted, whereas for  $L_c = 0.0075$  m, 0.01 m and 0.015 m, some  
348 un-melted PCM remains. For the smaller cell sizes, the temperature gradient in the  
349 computational domain is reduced as compared to the larger cell sizes.

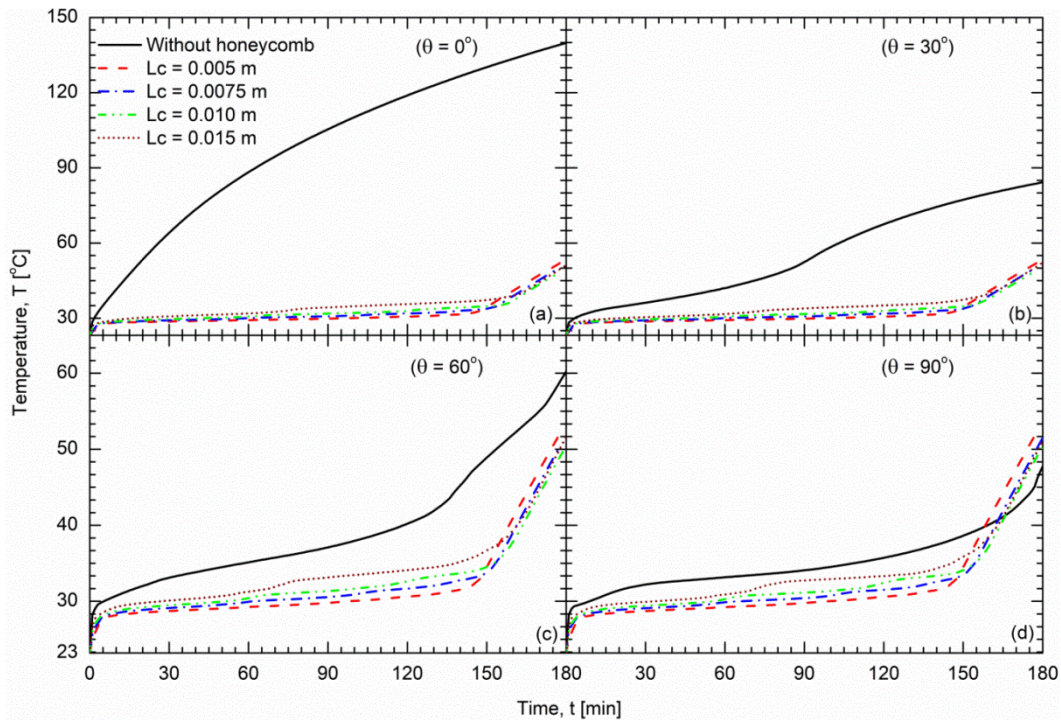


350

351 *Figure 8 Temperature field, melting front (white contours) and moving direction (black*  
352 *arrows) of n-octadecane in honeycomb container for different cell size at 150 min ( $\delta_{fin} =$*   
353 *0.0005 m and  $\theta = 60^\circ$ )*

354 4.2.1. Effect of honeycomb cell size on the average front surface temperature

355 The average temperature variation of the front surface with time for different cell sizes and  
356 angles of container inclination is presented in *Figure 9*. The comparison includes a case  
357 without hexagonal cells in the computational domain. When the computational domain is  
358 horizontal (*Figure 9(a)*,  $\theta = 0^\circ$ ), the average temperature of the front surface without  
359 hexagonal cells increases sharply and reaches around  $135^\circ\text{C}$  at 180 min because of pure  
360 conduction in the PCM. The effect of PCM is hardly noticeable. However with the  
361 honeycomb heat exchanger at  $L_c = 0.005\text{ m}$ ,  $0.0075\text{ m}$ ,  $0.01\text{ m}$  and  $0.015\text{ m}$  the average front  
362 surface temperature increases sharply only up to  $30^\circ\text{C}$  at  $t = 5\text{ min}$ , and then increases much  
363 more gradually with time until  $t = 150\text{ min}$ . This behavior is due to the presence of  
364 hexagonal cells in the container that facilitates a smooth heat transfer in the PCM. The fin  
365 structure transports heat all over the computational domain efficiently, thanks to its higher  
366 thermal conductivity. An increment in the tilt angle results in an enhancement of the  
367 convective heat transport in the PCM, which in turn further reduces the average front surface  
368 temperature. It is also observed that the angle of inclinations have a minor effect when the  
369 cell size is small ( $L_c = 0.005\text{ m}$ ,  $L_c = 0.01\text{ m}$  and  $L_c = 0.015\text{ m}$ ). From *Figure 9(a)-(d)* it is  
370 noted that the inclination angle only affects the average front surface temperature when the  
371 cell size is large. A major influence of the inclination angle on average front surface  
372 temperature is only observed when there are no hexagonal cells in the computational domain.  
373 Due to the addition of honeycomb cells, it is possible to keep the front surface temperatures  
374 at  $32^\circ\text{C}$ , which is around  $90^\circ\text{C}$  below as compared to the horizontal case (i.e.  $\theta = 0^\circ$ , pure  
375 conduction),  $50^\circ\text{C}$  lower as compared to  $\theta = 30^\circ$  case,  $15^\circ\text{C}$  lower as compared to  $\theta = 60^\circ$   
376 case and  $8^\circ\text{C}$  lower as compared to  $\theta = 90^\circ$  case. It is also probable to keep an approximately  
377 constant temperature of the front surface for up to 140 min.



378

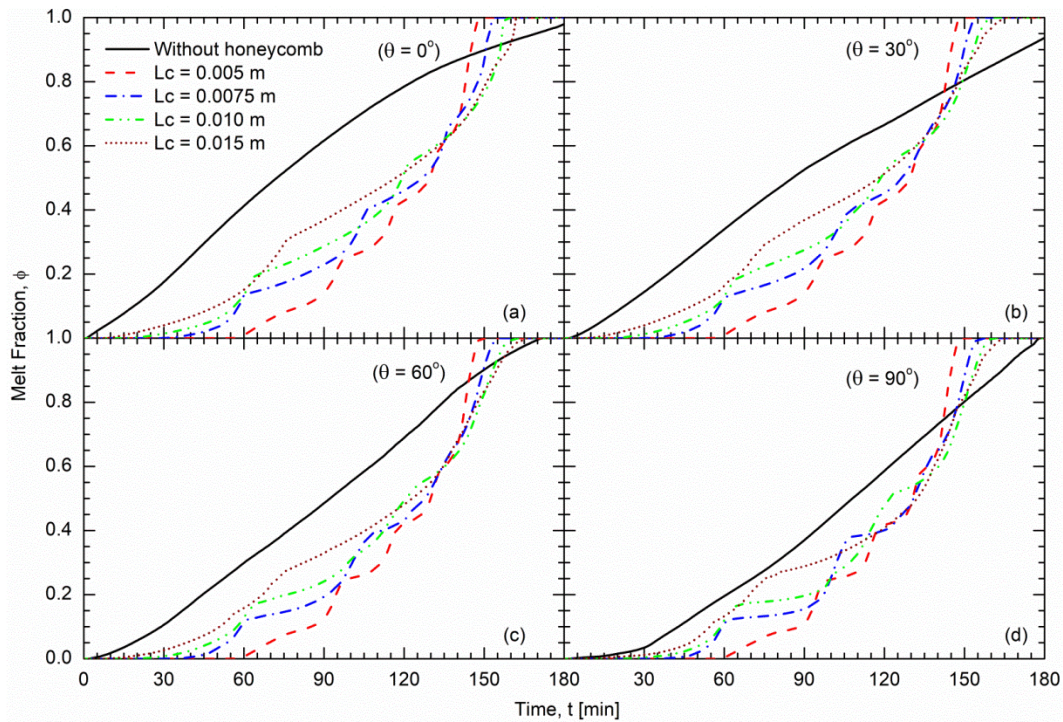
379 *Figure 9 Average front surface temperature variation with time for different honeycomb cell*  
 380 *sizes: (a) at  $\theta = 0^\circ$ , (b)  $\theta = 30^\circ$  (c)  $\theta = 60^\circ$  and (d)  $\theta = 90^\circ$*

381 *4.2.2. Effect of honeycomb cell size on the melt fraction*

382 The variation of melt fraction in the computational domain for different cell sizes and various  
 383 inclination angles is shown in Figure 10. Four disparate inclination angles (i.e.  $0^\circ$ ,  $30^\circ$ ,  $60^\circ$ ,  
 384 and  $90^\circ$ ) are considered in the present study. The melt fraction without honeycomb cells is  
 385 advanced as compared to that with honeycomb cells up to 140 min, regardless of the cell  
 386 sizes. This behavior is observed because of two reasons. Firstly, the presence of honeycomb  
 387 cells transfers heat to the entire computational domain, not only to the PCM available near  
 388 the front heating surface. Due to the absorption of heat as sensible heat by the computational  
 389 domain, the melt fraction of the PCM without cells is higher for a certain time and melting  
 390 starts later for the smallest cell size as compared to larger cell size. Secondly, in the  
 391 beginning, the convection heat transfer is accentuated in the configuration without cells, since  
 392 the melt PCM has more room for movement. After a certain amount of time (140 min here),  
 393 the latter convection process fails to achieve fast melting of the solid PCM situated as the  
 394 bottom left of the container and becomes less efficient than that of celled configurations  
 395 where heat is transported everywhere, as shown in Figure 8.

396 The latter reasoning holds when comparing meshed configurations. One can observe that for  
397 cases with higher cell sizes, melting starts earlier but ends later than for smaller cell sizes. As  
398 the heat which is transferred from the heated surface to the entire computational domain is  
399 reduced with larger cells, their initially higher melting rate decreases, and they finally take a  
400 longer time to achieve complete melting. For all the inclination angles it is observed that the  
401 melting starts at around 60 min, 35 min, 20 min, and 10 min for  $L_c = 0.005$  m,  $L_c = 0.0075$  m,  
402  $L_c = 0.01$  m, and  $L_c = 0.015$  m respectively, meaning that the finer the mesh, the later the  
403 melting start.

404 There are several inflection points observed in the melt fraction variation for all the cell sizes  
405 and inclination angles. The first inflection point for  $L_c = 0.005$  m,  $L_c = 0.0075$  m,  $L_c = 0.01$   
406 m, and  $L_c = 0.015$  m is observed when melt fraction reaches around 0.1, 0.15, 0.2 and 0.3  
407 respectively. The second inflection point in the melt fraction is not observed when  $L_c =$   
408  $0.015$ , however for  $L_c = 0.005$  m,  $L_c = 0.0075$  m,  $L_c = 0.01$  m the inflection point is observed  
409 when the melt fraction reaches 0.225, 0.45, and 0.55 respectively. The number of inflection  
410 points in the melt fraction variation depends on the number of cells in the direction of the  
411 heat flux application. The inflections in the melt fraction are because of the uneven heat  
412 transfer that happened simultaneously by the additional contact resistance at the surface of  
413 each cell and to convection in the melted PCM within the cells. Regarding the latter cause,  
414 the heat transfer to the honeycomb cell is faster when the PCM filled in the previous cell is  
415 completely melted. If the PCM filled in the previous cell is not completely melted, the heat is  
416 absorbed by the solid PCM and a reduced amount of energy is transferred to the next cell  
417 which reduces the melting rate in that cell, resulting in a reduced melt fraction.

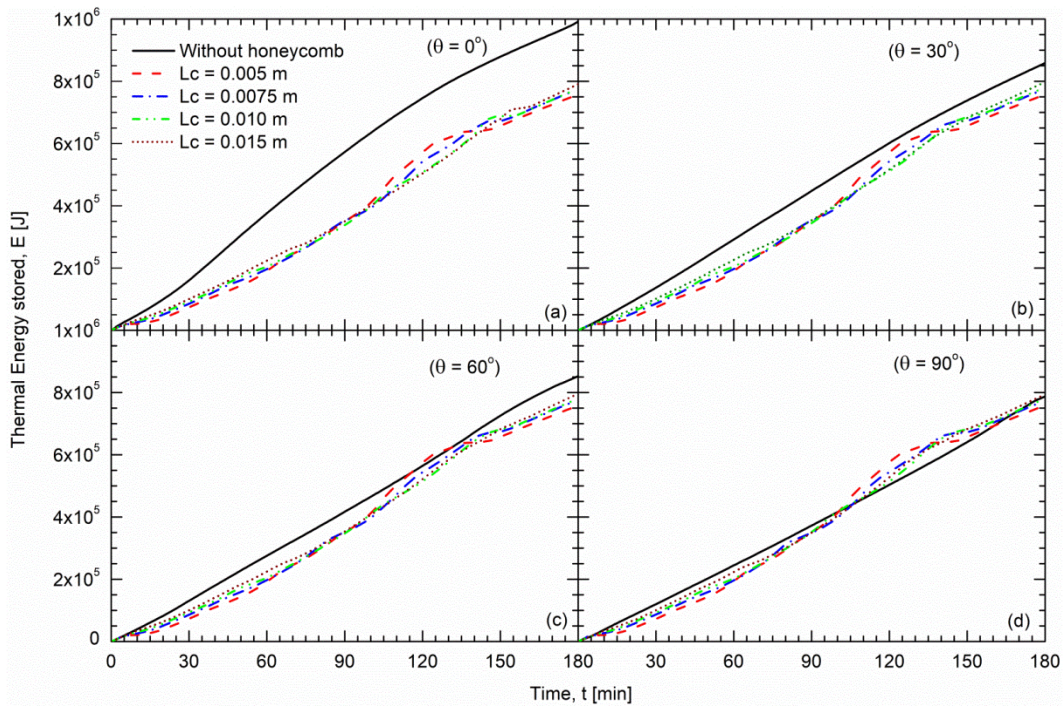


418

419 *Figure 10 Melt fraction variation with time for different honeycomb cell sizes: (a) at  $\theta = 0^\circ$ ,*  
 420 *(b)  $\theta = 30^\circ$  (c)  $\theta = 60^\circ$  and (d)  $\theta = 90^\circ$*

421 *4.2.3. Effect of honeycomb cell size on the energy stored*

422 The energy stored is calculated for different cell sizes and tilt angles, as presented in Figs.  
 423 Figure 11(a)-(d). The maximum energy is stored for the computational domains that have no  
 424 hexagonal cells because, without honeycomb cells, the temperature of the computational  
 425 domain is higher as compared to other configurations, which results in additional energy  
 426 stored in as sensible heat. The honeycomb cells facilitate the heat transfer to the backplate  
 427 and the environment beyond, thus reducing the thermal energy remaining in the container.  
 428 The decrease in the cell size causes an increased number of cells in the computational  
 429 domain, which reduces the actual volume of the PCM in the container. Therefore, finer mesh  
 430 configurations exhibit a lower heat storage capacity. For meshed configurations, it is also  
 431 observed that the inclination angle does not significantly affect the energy content in the  
 432 PCM.



433

434 *Figure 11 Energy stored in PCM with time for different honeycomb cell sizes: (a) at  $\theta = 0^\circ$ ,*  
 435 *(b)  $\theta = 30^\circ$  (c)  $\theta = 60^\circ$  and (d)  $\theta = 90^\circ$*

### 436 4.3. Effect of the inclination angle

437 The melting of PCM in a rectangular container also depends on the inclination angle due to  
 438 the convection effect. The effect of the tilt angle of the computational domain from horizontal  
 439 on the average temperature of the front surface, melt fraction, energy stored in the container  
 440 is presented for two different cell sizes,  $L_C = 0.015\text{ m}$  and  $L_C = 0.0075\text{ m}$ .

#### 441 4.3.1. Influence of inclination angle on the average front surface temperature

442 Figure 12(a) and (d) show the average front surface temperature variation at different angles  
 443 of inclination for two different cell sizes  $L_C = 0.015\text{ m}$  and  $L_C = 0.0075\text{ m}$ , respectively.  
 444 For both cell sizes, the minimum average front surface temperature is obtained with  $\theta = 90^\circ$   
 445 because of higher convective heat transfer in the computational domain. The impact of the  
 446 angle of inclination on the average front surface temperature is negligible with smaller cell  
 447 sizes because of the enhanced thermal transport due to the existence of the fin structure. The  
 448 presence of the fin structure reduces the velocity field; however, it enhances thermal transport  
 449 via the metallic structure.



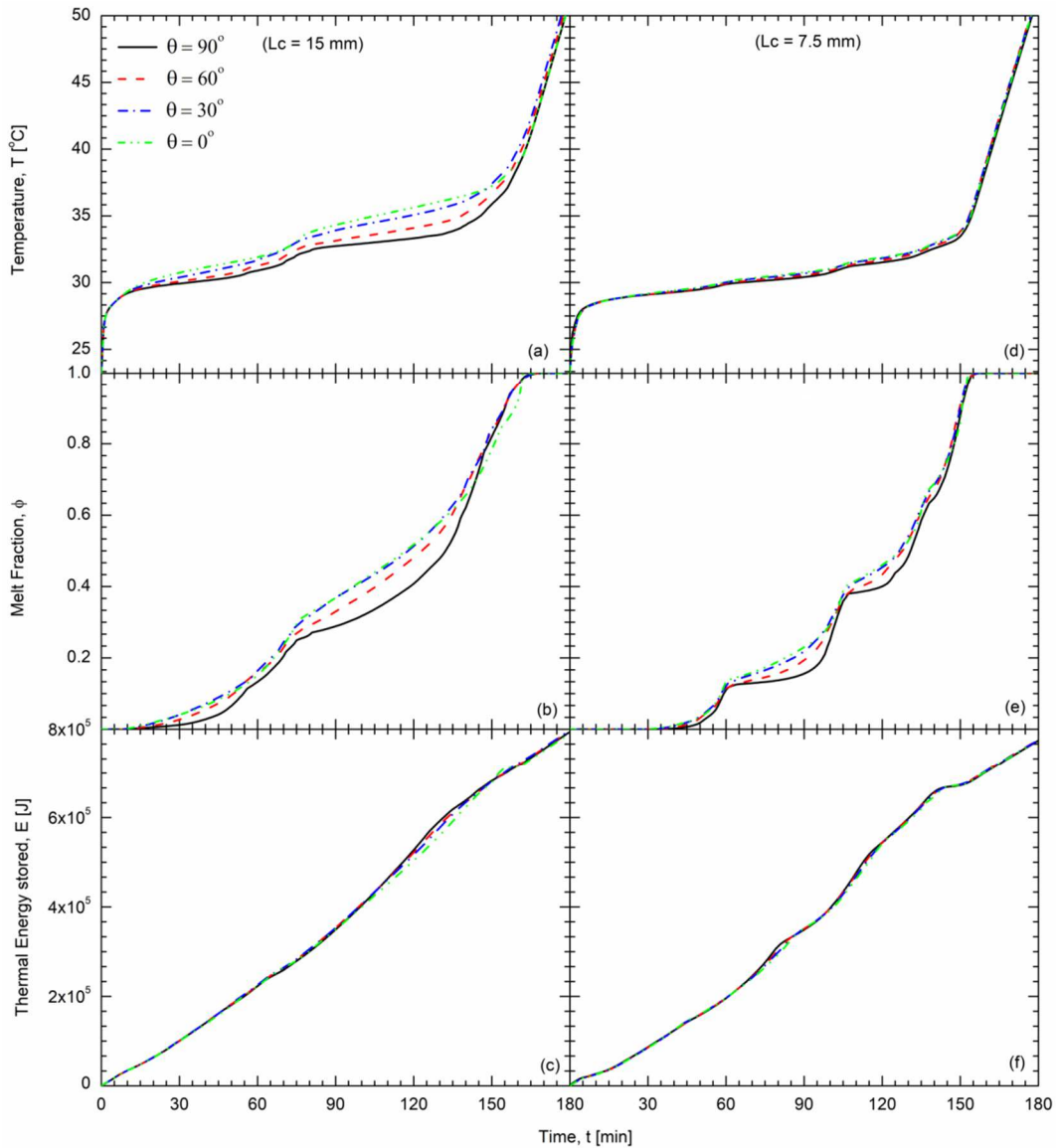
450 *4.3.2. Influence of inclination angle on the melt fraction*

451 The Influence of the tilt angle of the computational domain on the melt fraction for two  
452 different cell sizes is presented in Figures 12(b) and (e). For a certain time duration, the melt  
453 fraction is advanced when the computational domain is completely horizontal ( $\theta = 0^\circ$ ) for  
454 both cell sizes. Further, there are inflection points in the melting curve of both cell sizes. As  
455 previously mentioned, this is due to the uneven heat transfer that happened because of the  
456 additional contact resistance at the surface of each cell and to convection in the melted PCM  
457 within the cells. The number of inflection points in the melt fraction depends on the number  
458 of hexagonal cells present in the computational domain.

459 *4.3.3. Influence of inclination angle on the energy stored*

460 The energy storage content in the PCM with time at different inclination angles is presented  
461 in Figure 12(c) and (f) for  $L_C = 0.015$  m and  $L_C = 0.0075$  m, respectively. There is a small  
462 variation are observed for the different angle of inclination during energy storage with  $L_C =$   
463  $0.015$  m, however, there is no variation with  $L_C = 0.0075$  m. for given cell size and fin  
464 thickness the overall rate of energy storage is the same, regardless of the angle of inclination.  
465 This is due to the hypothesis of normal heat flux on the front surface in all configurations.  
466 The results might be different in real applications where the amount of solar radiation  
467 received depends on the container's tilt angle.

468



469

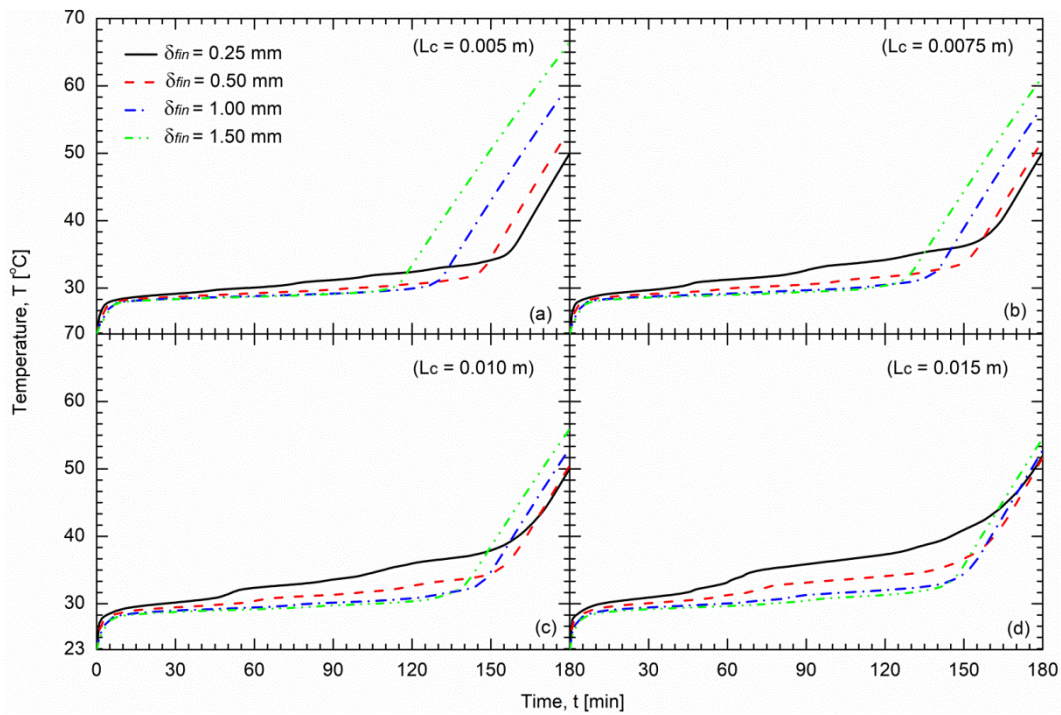
470 *Figure 12 Effect of tilt angle on average front surface temperature, melt fraction and energy*  
 471 *stored in the PCM: (a)-(c)  $L_C = 0.015$  m, (d)-(f)  $L_C = 0.0075$  m*

#### 472 4.4. Effect of the honeycomb cell fin thickness

##### 473 4.4.1. Effect of the honeycomb cell fin thickness on the average front surface temperature

474 *Figure 13* represents the average front surface temperature variation for four different  
 475 honeycomb cell fin thicknesses (i.e.  $\delta_{fin} = 0.25$  mm, 0.5 mm, 1 mm and 1.5 mm). Four  
 476 different cell sizes are also considered. *Figure 13* (a) represents the average temperature  
 477 variation for  $L_C = 0.005$  m. At first, the minimum temperature is obtained with a thicker fin  
 478 because of higher heat transmission to the backplate. But due to enhanced heat transfer to the  
 479 PCM and reduced volume of PCM in the container, the thicker fin configuration is also the

480 one with a shorter melting time and its average front surface temperature becomes highest  
 481 after the complete melting of PCM. The same behavior is observed for all fin thicknesses,  
 482 with the inversion depending on the fin thickness, except before the end of melting where  
 483 there is a very small temperature difference for  $\delta_{fin} = 0.5 \text{ mm}, 1 \text{ mm}$  and  $1.5 \text{ mm}$ . As  
 484 shown in *Figure 13* (b)-(d) the cell size does not significantly change the above-described  
 485 pattern. However larger cell sizes reduce thermal transport to the PCM and therefore, the  
 486 average front surface temperature is higher for the higher cell size.



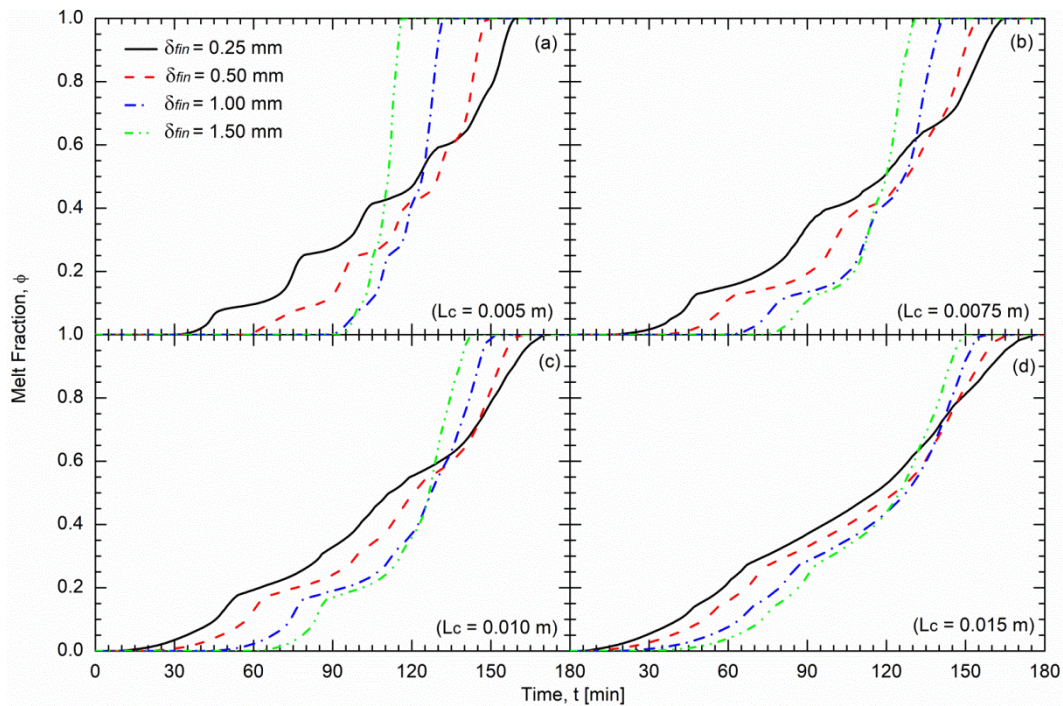
487

488 *Figure 13* Average front surface temperature variation with time at different honeycomb fin  
 489 thickness: (a) at  $L_C = 0.005 \text{ m}$ , (b)  $L_C = 0.0075 \text{ m}$  (c)  $L_C = 0.010 \text{ m}$  and (d)  $L_C = 0.015 \text{ m}$

#### 490 4.4.2. Effect of the honeycomb cell fin thickness on the melt fraction

491 The effect of honeycomb cell fin thickness on melt fraction for different cell sizes is shown in  
 492 *Figure 14*(a)-(d). *Figure 14*(a) represents the melt fraction at  $L_C = 0.005 \text{ m}$  for different fin  
 493 thicknesses. The melting of PCM starts later for higher fin thicknesses as compared to lower  
 494 ones, because of the higher fin thickness transport to the backplate and higher storage of  
 495 sensible heat in the fins. Due to sensible heating below the melting temperature, the melting  
 496 of PCM starts later which can be visible in *Figure 14*(a). Similar to the previous subsection,  
 497 after fin sensible heating, the heat transfer is enhanced with higher fin thickness, with less  
 498 volume of PCM, which explains the end of melting, earlier than that of lower fin thicknesses.

499 A similar pattern is obtained with all other cell sizes; however, the complete melting time is  
 500 higher for higher cell sizes.

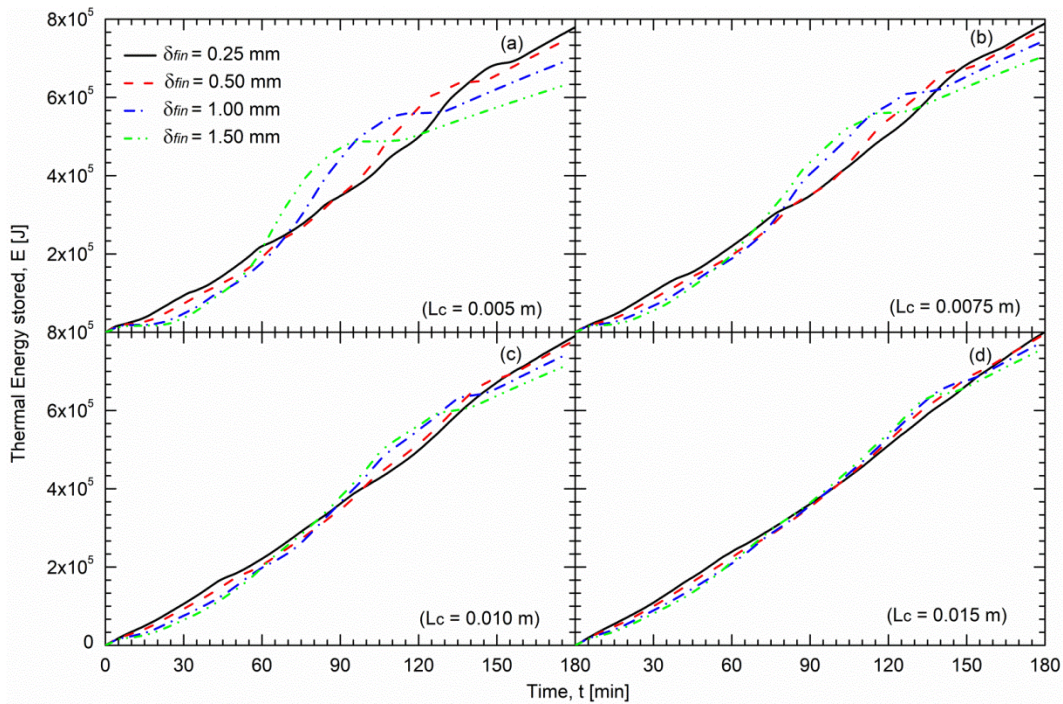


501

502 *Figure 14 Melt fraction variation with time at different honeycomb fin thickness: (a) at  $L_c =$*   
 503 *0.005 m, (b)  $L_c = 0.0075$  m (c)  $L_c = 0.010$  m and (d)  $L_c = 0.015$  m*

504 *4.4.2. Effect of the honeycomb cell fin thickness on the energy stored*

505 The variation of energy storage content in the computational domain with time for different  
 506 fin thicknesses for four different cell sizes is presented in Figure 15(a)-(d). Initially, the  
 507 energy storage rate in the PCM is higher for smaller fin thicknesses, as the energy is quickly  
 508 transported to the PCM (reduced sensible heat accumulation in the fins). After the melting of  
 509 PCM starts, it becomes lower for lower fin thicknesses, as previously explained. The overall  
 510 energy storage content after complete melting is higher for lower fin thickness and minimum  
 511 for highest fin thickness because of the reduction in the actual volume of PCM filled in the  
 512 computational domain. Increasing the cell size tends to reduce the effect of fin thickness, as  
 513 the rate of energy storage becomes similar for all cases (Figure 15 (c)-(d)).



514

515 *Figure 15 Energy stored in PCM with time at different honeycomb fin thickness: (a) at  $L_c =$*   
 516  *$0.005$  m, (b)  $L_c = 0.0075$  m (c)  $L_c = 0.010$  m and (d)  $L_c = 0.015$  m*

517 **5. Conclusions**

518 In the current study, the influence of honeycomb meshed fins on the heat transport  
 519 performance of a rectangular PCM container inclined at different angles from horizontal has  
 520 been numerically studied. A 2D numerical model of the rectangular container with  
 521 honeycomb structure considering conduction and convection modes of heat transfer in the  
 522 melted PCM is developed. The heat transfer and energy storage behavior without honeycomb  
 523 cells was looked up to that of four other configurations where the PCM is filled in  
 524 honeycomb cells of four different lengths, thicknesses, and tilted at four different inclination  
 525 angles. The evaluation of the charging and discharging efficiency of the PCM-filled in  
 526 honeycomb fins structures were based on the melting time, the regulation of the front plate  
 527 temperature to retain it as low as possible for the longest possible duration, as well as the  
 528 energy storage rate.

529 Results have shown that natural convection plays an important role in maintaining the  
 530 average temperature of the front surface to a constant value and for continuous thermal  
 531 transport in PCM. Further, optimized honeycomb cell length and fin thickness choices would  
 532 be vital for a rapid heat transfer to the rear end of the computational domain. It was also

533 found that a tilt angle of the honeycomb cells has a minor influence on the heat transfer  
534 behavior of all studied patterns when the cell size is small. The total melting time is minimum  
535 for lower cell sizes and higher fin thickness regardless of the orientation of the computational  
536 domain. With such a configuration, it is possible to keep the front surface temperatures at 32  
537 °C, which is around 90°C lower compared to the horizontal case (pure conduction,  $\theta = 0^\circ$ ).  
538 The energy storage behavior of the PCM with varying time is also investigated for the range  
539 of parameters selected in the present study and it is found that the total energy content is  
540 higher without honeycomb mesh because of the higher amount of PCM, while the tilt angle  
541 has a marginal effect on the energy storage rate in cases where the heat flux remains normal  
542 to the heated surface. Finally, it is concluded that this type of fin structure leads to enhanced  
543 thermal transport in the PCM to maintain the front surface temperature constant. Besides, the  
544 charging and discharging time of PCM is reduced as compared to without meshed  
545 configuration. Overall, such studies for PCM encapsulated in honeycomb cells may be  
546 utilized for appropriate thermal management and to optimize heat transfer and energy  
547 storage.

548

549 Declaration of competing interest

550 The authors have no conflict of interest regarding this work.

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554

555   References

- 556   [1]   X. Qiao, X. Kong, H. Li, L. Wang, H. Long, Performance and optimization of a novel  
557       active solar heating wall coupled with phase change material, *Journal of Cleaner*  
558       *Production*. (2020). <https://doi.org/10.1016/j.jclepro.2019.119470>.
- 559   [2]   X. Kong, L. Wang, H. Li, G. Yuan, C. Yao, Experimental study on a novel hybrid  
560       system of active composite PCM wall and solar thermal system for clean heating  
561       supply in winter, *Solar Energy*. (2020). <https://doi.org/10.1016/j.solener.2019.11.081>.
- 562   [3]   S. Gorjian, F. Calise, K. Kant, M.S. Ahamed, B. Copertaro, G. Najafi, X. Zhang, M.  
563       Aghaei, R.R. Shamshiri, A review on opportunities for implementation of solar energy  
564       technologies in agricultural greenhouses, *Journal of Cleaner Production*. 285 (2021)  
565       124807. <https://doi.org/10.1016/j.jclepro.2020.124807>.
- 566   [4]   K. Kant, A. Shukla, A. Sharma, A. Kumar, A. Jain, Thermal energy storage based  
567       solar drying systems: A review, *Innovative Food Science & Emerging Technologies*.  
568       34 (2016) 86–99. <https://doi.org/10.1016/j.ifset.2016.01.007>.
- 569   [5]   A. Shukla, A. Sharma, K. Kant, Solar greenhouse with thermal energy storage: A  
570       review, *Current Sustainable/Renewable Energy Reports*. 3 (2016) 58–66.  
571       <https://doi.org/10.1007/s40518-016-0056-y>.
- 572   [6]   Y. Dutil, D.R. Rousse, N. Ben Salah, S. Lassue, L. Zalewski, A review on phase-  
573       change materials: Mathematical modeling and simulations, *Renewable and Sustainable*  
574       *Energy Reviews*. 15 (2011) 112–130. <https://doi.org/10.1016/j.rser.2010.06.011>.
- 575   [7]   K. Kant, A. Shukla, A. Sharma, P. Henry Biwole, Heat transfer study of phase change  
576       materials with graphene nano particle for thermal energy storage, *Solar Energy*. 146  
577       (2017) 453–463. <https://doi.org/10.1016/j.solener.2017.03.013>.
- 578   [8]   K. Kant, A. Anand, A. Shukla, A. Sharma, Heat transfer study of building integrated  
579       photovoltaic (BIPV) with nano-enhanced phase change materials, *Journal of Energy*  
580       *Storage*. 30 (2020) 101563. <https://doi.org/10.1016/j.est.2020.101563>.
- 581   [9]   X. Tong, J.A. Khan, M. Ruhul Amin, Enhancement of heat transfer by inserting a  
582       metal matrix into a phase change material, *Numerical Heat Transfer, Part A:*  
583       *Applications*. 30 (1996) 125–141. <https://doi.org/10.1080/10407789608913832>.
- 584   [10]  J.M. Marín, B. Zalba, L.F. Cabeza, H. Mehling, Improvement of a thermal energy  
585       storage using plates with paraffin-graphite composite, *International Journal of Heat*



- 586 and Mass Transfer. (2005). <https://doi.org/10.1016/j.ijheatmasstransfer.2004.11.027>.
- 587 [11] H.A. Adine, H. El Qarnia, Numerical analysis of the thermal behaviour of a shell-and-  
588 tube heat storage unit using phase change materials, *Applied Mathematical Modelling*.  
589 33 (2009) 2132–2144. <https://doi.org/10.1016/j.apm.2008.05.016>.
- 590 [12] Y. Tian, C.Y. Zhao, Thermal and exergetic analysis of Metal Foam-enhanced  
591 Cascaded Thermal Energy Storage (MF-CTES), *International Journal of Heat and*  
592 *Mass Transfer*. (2013). <https://doi.org/10.1016/j.ijheatmasstransfer.2012.11.034>.
- 593 [13] P.H. Biwole, D. Groulx, F. Souayfane, T. Chiu, Influence of fin size and distribution  
594 on solid-liquid phase change in a rectangular enclosure, *International Journal of*  
595 *Thermal Sciences*. (2018). <https://doi.org/10.1016/j.ijthermalsci.2017.10.038>.
- 596 [14] D. Groulx, P.H. Biwole, M. Bhourri, Phase change heat transfer in a rectangular  
597 enclosure as a function of inclination and fin placement, *International Journal of*  
598 *Thermal Sciences*. (2020). <https://doi.org/10.1016/j.ijthermalsci.2020.106260>.
- 599 [15] A.A. Rabienataj Darzi, M. Jourabian, M. Farhadi, Melting and solidification of PCM  
600 enhanced by radial conductive fins and nanoparticles in cylindrical annulus, *Energy*  
601 *Conversion and Management*. 118 (2016) 253–263.  
602 <https://doi.org/10.1016/j.enconman.2016.04.016>.
- 603 [16] S. Mat, A.A. Al-Abidi, K. Sopian, M.Y. Sulaiman, A.T. Mohammad, Enhance heat  
604 transfer for PCM melting in triplex tube with internal-external fins, *Energy Conversion*  
605 *and Management*. (2013). <https://doi.org/10.1016/j.enconman.2013.05.003>.
- 606 [17] M. Sheikholeslami, S. Lohrasbi, D.D. Ganji, Numerical analysis of discharging  
607 process acceleration in LHTESS by immersing innovative fin configuration using  
608 finite element method, *Applied Thermal Engineering*. (2016).  
609 <https://doi.org/10.1016/j.applthermaleng.2016.06.158>.
- 610 [18] L. Wu, X. Zhang, X. Liu, Numerical analysis and improvement of the thermal  
611 performance in a latent heat thermal energy storage device with spiderweb-like fins,  
612 *Journal of Energy Storage*. 32 (2020) 101768.  
613 <https://doi.org/10.1016/j.est.2020.101768>.
- 614 [19] Y. Chen, Z. Deng, Hydrodynamics of a droplet passing through a microfluidic T-  
615 junction, *Journal of Fluid Mechanics*. (2017). <https://doi.org/10.1017/jfm.2017.181>.
- 616 [20] C. Zhang, F. Yu, X. Li, Y. Chen, Gravity–capillary evaporation regimes in

- 617 microgrooves, *AIChE Journal*. (2019). <https://doi.org/10.1002/aic.16484>.
- 618 [21] M. Liu, L. Su, J. Li, S. Chen, Y. Liu, J. Li, B. Li, Y. Chen, Z. Zhang, Investigation of  
619 spherical and concentric mechanism of compound droplets, *Matter and Radiation at*  
620 *Extremes*. (2016). <https://doi.org/10.1016/j.mre.2016.07.002>.
- 621 [22] Y. Chen, C. Zhang, M. Shi, Y. Yang, Thermal and hydrodynamic characteristics of  
622 constructal tree-shaped minichannel heat sink, *AIChE Journal*. (2010).  
623 <https://doi.org/10.1002/aic.12135>.
- 624 [23] C. Zhang, Y. Chen, R. Wu, M. Shi, Flow boiling in constructal tree-shaped  
625 minichannel network, *International Journal of Heat and Mass Transfer*. (2011).  
626 <https://doi.org/10.1016/j.ijheatmasstransfer.2010.09.051>.
- 627 [24] X. Daguene-Frick, J. Bonjour, R. Revellin, Constructal microchannel network for  
628 flow boiling in a disc-shaped body, *IEEE Transactions on Components and Packaging*  
629 *Technologies*. (2010). <https://doi.org/10.1109/TCAPT.2009.2027427>.
- 630 [25] R. Wu, Y.P. Chen, J.F. Wu, M.H. Shi, Flow boiling characteristics in constructal tree-  
631 shaped minichannel, *Kung Cheng Je Wu Li Hsueh Pao/Journal of Engineering*  
632 *Thermophysics*. (2010).
- 633 [26] A. Sciacovelli, F. Gagliardi, V. Verda, Maximization of performance of a PCM latent  
634 heat storage system with innovative fins, *Applied Energy*. (2015).  
635 <https://doi.org/10.1016/j.apenergy.2014.07.015>.
- 636 [27] C. Zhang, J. Li, Y. Chen, Improving the energy discharging performance of a latent  
637 heat storage (LHS) unit using fractal-tree-shaped fins, *Applied Energy*. (2020).  
638 <https://doi.org/10.1016/j.apenergy.2019.114102>.
- 639 [28] J. LI, Y. Huang, C. Zhang, X. Liu, Numerical study on the solidification performance  
640 of a latent heat storage unit with Koch-fractal fin, *Fractals*. 27 (2019) 1950108.  
641 <https://doi.org/10.1142/S0218348X19501081>.
- 642 [29] M. Sheikholeslami, S. Lohrasbi, D.D. Ganji, Response surface method optimization of  
643 innovative fin structure for expediting discharging process in latent heat thermal  
644 energy storage system containing nano-enhanced phase change material, *Journal of the*  
645 *Taiwan Institute of Chemical Engineers*. (2016).  
646 <https://doi.org/10.1016/j.jtice.2016.08.019>.
- 647 [30] K. Kant, A. Shukla, A. Sharma, P.H. Biwole, Heat transfer studies of photovoltaic

648 panel coupled with phase change material, *Solar Energy*. 140 (2016) 151–161.  
649 <https://doi.org/10.1016/j.solener.2016.11.006>.

650 [31] P.H. Biwole, P. Eclache, F. Kuznik, Phase-change materials to improve solar panel's  
651 performance, *Energy and Buildings*. 62 (2013) 59–67.  
652 <https://doi.org/10.1016/j.enbuild.2013.02.059>.

653 [32] H. Shokouhmand, B. Kamkari, Experimental investigation on melting heat transfer  
654 characteristics of lauric acid in a rectangular thermal storage unit, *Experimental  
655 Thermal and Fluid Science*. (2013).  
656 <https://doi.org/10.1016/j.expthermflusci.2013.06.010>.

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