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Experimental and Numerical Study of Conical-Shaped Pin Finned Heat Sink with PCM Material

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ABSTRACT

Many researchers have recently become interested in phase change material (PCM) for passive cooling of electronic equipment. The performance of PCM is enhanced by using fins as thermal conductivity enhancers (TCE). For a heat flux of 20,000 W/m^2, three-dimensional numerical (transient) analyses were conducted for pin (circular) and conical-shaped fins with and without holes and dimples on the surfaces to investigate the heat transfer by natural convection. Paraffin wax was used as the PCM material within the fins' spaces by keeping 9% volume of the fins and 91% volume of PCM of the total space above the base plate. The transient heat transfer performance of fins is observed for 20 minutes. Surface areas, temperature curves, liquid fraction curves, and Theta (θ) versus Fourier number curves were compared to analyze the effect of fins of different shapes on the heat transfer rate. Conical-shaped fins with dimples and holes showed the best cooling performance with a 4.6% increment of heat transfer compared with conventional pin (circular shape) fins. With and without PCM, conical fins were subjected to experimental investigation for 20 minutes. Mild steel was the heat sink material, and paraffin wax was the PCM material. Heat flux of 4000 W/m^2 was given to the heat sink's base by a heater. The acrylic glass was used for insulation. The temperature was recorded for nine fins at different heights. In the experiment, the case with PCM gave 5% to 8% better cooling performance than the case with air. The study shows that using conical fins with dimples and holes (a unique fin shape) in PCM-based heat sinks can produce superior thermal performance compared to traditional (circular-shaped) pin-fins.

Keywords:
Heat Sink, Fins, PCM, 3D Computer Model, Transient Simulation

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1. Introduction

Every day, technological improvements in electronic devices are being made. Their sizes are becoming smaller, and they are becoming faster. In 2018, a printed circuit board could contain up to 5×10^{10} transistors. So, thermal management has been a great concern nowadays. Fifty-five percent of failures of electronic devices are found to be due to high temperatures [1]. By reducing 1 ℃, chances of failure were seen to be reduced by 4 %, whereas by increasing 10 ℃-20 ℃ temperature, a 100 % failure tendency occurs. Researchers are carrying out different research on the cooling of electronic devices. There are both active and passive cooling techniques. In the active method, heat is directly ejected from the chip by direct contact through heat sinks. This might be done by forced convection, free convection, liquid cooling, etc. In forced convection, the fan is generally used. But this causes noise and is not suitable for small devices. New developments in the passive thermal management (TM) of electronic devices using phase change materials (PCMs) have recently become popular. Due to high heat storage density, they are at the centre of attention and go under the isothermal process during phase transition. Because of their great thermal storage capacity and energy efficiency, phase change materials (PCMs) are employed to control internal temperatures in buildings and automobiles [2]. When PCM starts to melt at a constant temperature, a small amount of volume (< 10 %) changes during solid to liquid phase transition. Their thermal conductivity is very low. To overcome the low thermal conductivity problem, fins, metal matrices, and nanoparticles are used as thermal conductivity enhancers [3].

PCMs are chemically stable, relatively light, and non-corrosive to many materials. Heat is removed from the device when PCM melts. When the device is not in use, PCM freezes to solid, releasing heat to the surroundings. However, cooling would not be very effective if the operating time of portable electronic devices is longer than the time needed for the phase transformation of PCM [4]. Many studies have focused on the concept of shape modification of fins, inserts, and vortex generators to improve thermal devices' performances [5-6]. A numerical study showed that PCM (n-eicosane) with pin-fins having a 9% volume fraction had better cooling performance with a 3 mm fin thickness than a 2 mm fin thickness. Arshad et al. [7] tested the heat sink at different power levels in that study. The higher temperature rise in the 2 mm thick finned heat sink was due to the larger number of fins than the one with a 3 mm thick fin. Kalbasi *et al.* [8] proved a hybrid heat sink to perform better compared to air-cooled and PCM-based heat sinks. The input parameters were the convective heat transfer coefficient, PCM volume fraction, and heat flux. Temperature variation of the base of the heat sink was the output parameter. Adeel Arshad *et al.* [9] concluded that a 2 mm square pin-fin had the best thermal performance using n-eicosane as the PCM. Using the Taguchi approach, Pakrouh *et al.* [10] numerically optimized the number of fins, the height of fins, the thickness of fins, and the thickness of the base of a heat sink. At low critical temperatures, the number of fins had the biggest influence, followed by fin thickness and height. Base thickness had the least effect on operating temperature. Arshad and his co-authors [11] conducted parametric experimental studies for pin-finned heat sinks with square and circular profiles based on the number of fins, fin thicknesses, and fin configurations filled with six different types of PCMs for constant and intermittent heating. When comparing no fin, 1 mm, 3 mm, and 4 mm fin thicknesses in PCM-filled heat sinks, the authors discovered that 2 mm and 3 mm fin thicknesses with square and circular configurations respectively had the best thermal performance. Ashraf and his co-authors [12] filled circular and square-shaped pin-finned heat sinks having inline and staggered fin distributions with several PCMs. Inline distribution's cooling performance was found to be superior than the staggered distribution's cooling performance for both circular and square-shaped pin-finned heat sinks.

Ali *et al.* [13] investigated the sensitivity of a PCM-filled heat sink to the PCM types in an experimental research. The results showed that n-eicosane outperformed Paraffin wax in each experiment. Saha *et al.* [14] say that an 8 % volume fraction of fins serves the best for PCM-based heat sinks. Increased surface area increases heat transfer rate, and thus, cooling performance of fins improves [15]. Keeping volume fractions constant, different geometries of fins have been tested. They were compared concerning surface area ratios and Fourier number, F_o [16]. Several perforation shapes and alignments were applied on twisted and grooved-shaped pin fins to increase the heat transfer rate of the heat sink, as suggested by Haque *et al. (*2022a) [17]. Haque *et al.* (2022b) [18] added various shapes of perforations and bulges on pin fins to create more surface area and flow pattern. In comparison to in-line arrays, staggered arrays exhibit higher pressure coefficients and heat transfers in another study [19]. After an extensive literature review, it was found that conical fins were not tested with a PCM-based heat sink. No studies on PCM-based heat sinks with pin (circular shape) fins or conical-shaped fins having dimples and holes have been seen in the existing literature. Performance test using mild steel as the heat sink material is also missing. Hence, the present work is an attempt to numerically analyze the cooling performance of pin (circular) fins with dimples and holes and conical fins with and without dimples and holes in a PCM-based heat sink. The results are compared with PCM-based heat sinks' performance with conventional pin (circular shape) fins. A similar volume fraction of fins has been maintained in each case to compare their performances. A heat sink performs better if its base plate's temperature can be kept lower. Thermal performance, therefore, continues to be crucial in this paper. The comparison of the liquid fractions in this study is not detailed since full melting has not occurred. For a fully melted case, the faster the liquid fraction reaches the value 1, the better the heat sink performs. Later, an experiment was conducted on a conical finned heat sink with air and PCM (paraffin wax). It was done to observe PCM's cooling capacity compared to air's. The study's novelty is that it numerically analyzes fins with unique shapes (circular and conical fin surfaces modified with dimples and holes) in a PCM-based heat sink. This study is also unique because it uses mild steel made heat sink with conical fins with PCM for experimentation.

2. Methodology

2.1 Computational Model

To preserve thermal equilibrium and avoid overheating, a passive thermal management element called a heat sink is made to absorb and disperse heat away from a hot body [20]. The heat sink is usually connected to the heat source via thermal paste. The bottom boundary of the heat sink is defined with a heat flux value.

The heat sink with conical fins is depicted isometrically in Figure 1(a). The bottom view is displayed in Figure 1 (b). Insulation surrounds the heater, which is located in the middle of the base plate. The acrylic-glass border encircling the heat sink is visible in Figure 1 (c). Heat enters from the base when the fin-tops are open to allow for natural convection. PCM is used to fill in the spaces between fins.

For the numerical analyses, four shapes of fins as shown in Figure 2 were tested. The volume fraction of conical fins (including the remaining three cases) was 8 % to 9 % [14] of the total volume above the heat sink's base. The material used was mild steel. The designs were made in Solid Works. The value of 'H' is 25.43 mm both in Figure 1 and Figure 2. 'H' is the common constant used to dimension all fins. To keep the volume fraction of fins near about 9%, the base diameter of fins and the fin numbers varied. Dimples and holes of similar dimensions and numbers were added to increase the surface area, keeping the overall volume of fins unchanged.

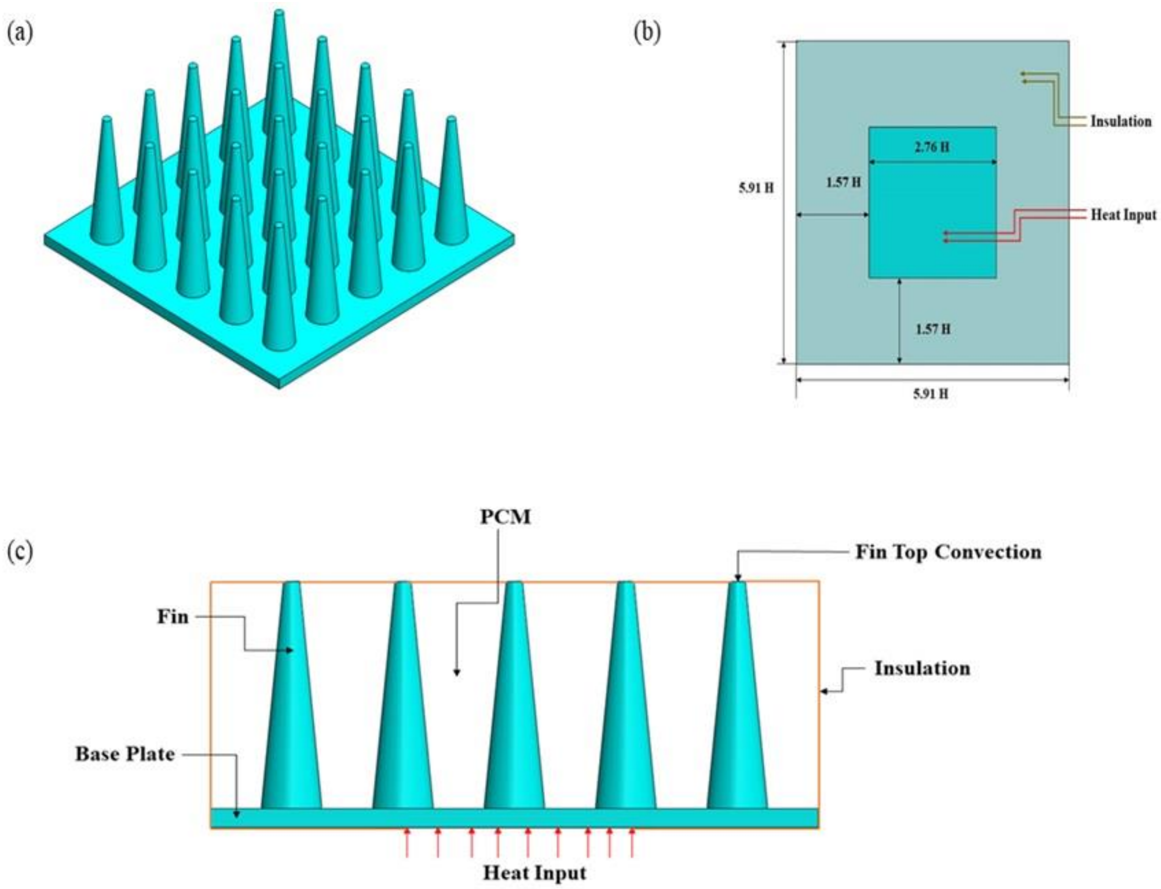


Fig. 1. (a) Conical heat sink, (b) Base of heat sink, (c) Boundary setup of the model

The heat sink's base had dimensions of 150 mm × 150 mm. The height of the heat sink was 65 mm. Twelve fins were selected for the cases, including circular-pin fins (with and without dimples and holes). Twenty-five fins were on the conical-pin finned heat sinks (with and without dimples and holes).

The bar chart in Figure 3 shows the comparisons amongst the surface areas calculated for different geometries of fins. From the chart, it can be seen that conical fins with dimples and holes have the largest surface area. The second largest surface area is for simple conical fins (without dimples and holes). Pin (circular) shaped fins have the least surface area among all these shapes. The experiment was done with a conical (simple) finned heat sink with dimensions similar to those followed for the numerical study.

Three-dimensional analyses were adopted for this study with PCM-based heat sinks having different fins. A common set of governing equations for the PCM and fin domains [21] are given below, where the first one is the continuity equation, the second one is the momentum equation, and the third one is the energy equation:

$$\frac{\partial \alpha_n}{\partial t} + u_i \frac{\partial \alpha_n}{\partial x_i} = 0 \tag{1}$$

Here, α_n represents the volume fraction of the nth fluid phase in the computational cells, t is time, u_i are the components of velocity and x_i are the spatial coordinates.

$$\frac{\partial}{\partial t}\left(\rho_n u_i\right) \frac{\partial(\rho_n u_i u_j)}{\partial x_j} = \mu_n \frac{\partial^2 u_i}{\partial x_j \partial x_j} - \frac{\partial \rho}{\partial x_i} + \rho_n g_i + S_{n,i} \tag{2}$$

Here, ρ_n is the density of the nth fluid phase, μ_n is the dynamic viscosity, g_i represents gravitational acceleration and $S_{n,i}$ denotes any additional source terms.

$$\frac{\partial}{\partial t}(\rho_n h)+\frac{\partial}{\partial x_i}(\rho_n u_i h)=\frac{\partial}{\partial x_i}(k_n\frac{\partial T}{\partial x_i}) \tag{3}$$

Here, h represents the specific enthalpy, k_n is the thermal conductivity of the nth phase, and T is the temperature.

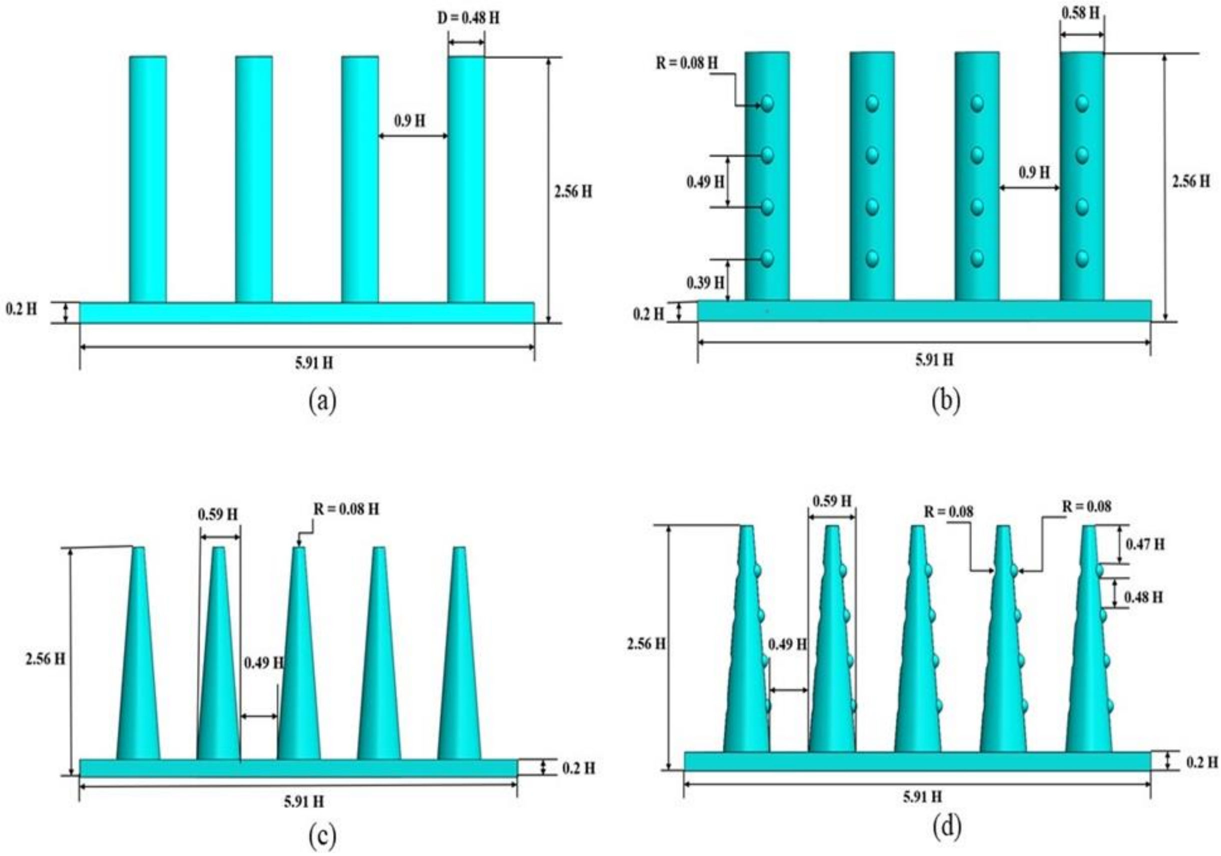


Fig. 2. Side views of heat sinks having different fins with dimensions (a) Pin fins, (b) Pin fins with dimples and holes, (c) Conical fins, (d) Conical fins with dimples and holes

2.2 Numerical Methodology and Experiment

The computational work was done using Ansys Fluent software. The PCM (liquid) was assumed to be incompressible, Newtonian, and went through Boussinesq approximation. The enthalpy porosity method was applied to model the effect of phase transition. The enthalpy-porosity approach is a popular method for creating a numerical model of a PCM. The main benefit of this method is that it eliminates the need for variable transformations and enables a fixed-grid solution of the associated momentum and energy equations. With very low computational requirements, the enthalpy-porosity method has been proven to converge quickly and yield reliable findings for the position and morphology of the melt front at different times [22-23].

The following assumptions were made to model the heat transfer problem of PCM-based heat sinks:

(1) three-dimensional,
(2) laminar flow,
(3) unsteady state,
(4) constant fluid properties,
(5) neglecting heat transfer due to radiation.

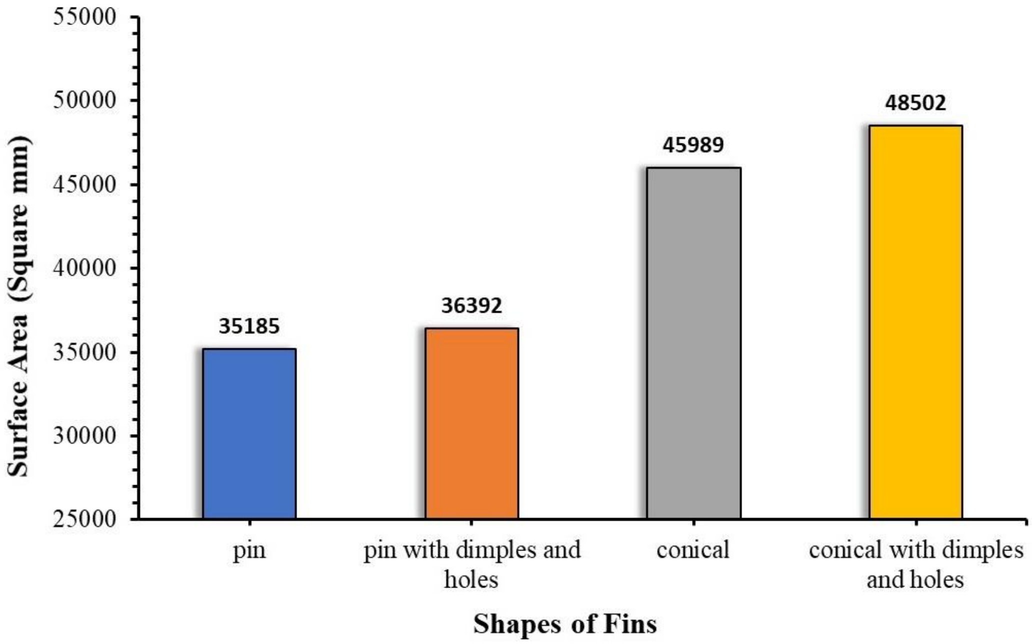


Fig. 3. Surface areas for fins of different shapes

Table 1
Boundary conditions and properties of the test section [24-27]

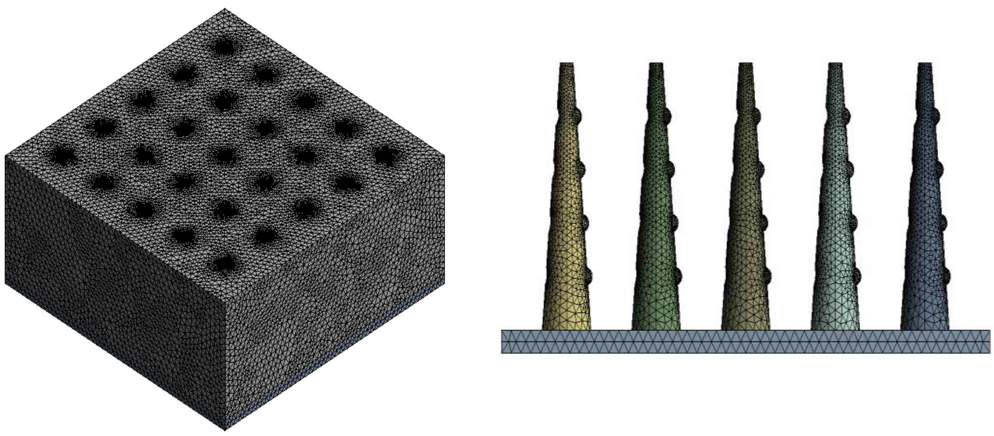
Name	Conditions
Initial Condition	At time t = 0, Temperature all over the body, T= T_i = 300 K
Heat Flux at the Base plate	For (x = 40-70mm, y = 0mm, z = 40-70mm) $\frac{\partial T}{\partial y}$ = - $\frac{q''}{k_s}$
Insulation Boundary	For (x = 0mm, x = 40mm) $\frac{\partial T}{\partial x}$ = 0; for (y = 0mm, y = 60mm) $\frac{\partial T}{\partial y}$ = 0 (not including heater); for (z = 0mm, z = 40mm) $\frac{\partial T}{\partial z}$ = 0.
Parameter	**Properties**
PCM	Density ρ_{PCM} = 790 kg/m^3, specific heat $C_p = 2150$ J/kg − K, thermal conductivity λ = 0.22 W/m-K and dynamic viscosity $\mu_{PCM} = 0.0269$ kg/m-s
Materials	Mild Steel (Thermal Conductivity $K_{ms} = 46$ W/m − K , density $\rho_{ms} = 7850$ kg/m^3)
Materials	Acrylic (Thermal Conductivity $K_{ac} = 0.2$ W/m − K , density $\rho_{ac} = 1180$ kg/m^3)

The boundary conditions mentioned in Table 1 have been used to solve the conservation equations. A Laminar viscous model was used for this CFD (computational fluid dynamics) study. The boundary surfaces were of acrylic material. The fin tops had natural convection with a heat transfer coefficient of 25 W/m^2-K. A heat flux of 20,000 W/m^2 was given at the base of the heat sink. For the pressure-velocity coupling, the SIMPLE scheme was selected. The convergence criteria for residuals was 1×10^{-06}.

Tetrahedron mesh was selected for all cases, as shown for conical fins with dimples and holes in Figure 4. The domain was segmented into multiple sub-domains to generate a grid, each comprising of 'fins solid zone', 'base plate solid zone' and 'PCM zone'. The quality of smoothing was medium. In Table 2, a grid study was conducted to measure the accuracy of the simulations. For meshing, the element number selected was 1329409, as the deviation of base temperature was the lowest for it.

Table 2
(GIT) Meshing Data for 3D Simulation [5]

Size of element (mm)	Element Number	Base Temperature (K)	Percent Deviation
400	127131	337.3876	-
240	400573	336.1578	0.3658 %
150	810259	335.8822	0.0821 %
3	1329409	335.7998	0.0245 %
1.6	2038582	335.6309	0.0503 %



(a) (b)

Fig. 4. Meshing of the computational domain (a) Isometric view, (b) Front view

The current model was validated with PCM based plate finned heat sink (used for electronic cooling) having similar dimensions as in the study of S.K. Saha *et al.* [28]. The initial temperature of the PCM was 27 ℃. The length of the sink was 20 mm. Half fin thickness was 0.2 mm. The total height was 10 mm. The validation graph is shown in Figure 5. The values of liquid fraction were validated. Applied heat flux, q was 11337 W/m^2. At 9 s, PCM started melting, and at 184 s, the melting was completed. The validation had a 7 % error in it.

For the experimental purpose, a conical shape was given to the fins with the help of a lathe machine. Fine machining was done as much as possible. Acrylic glasses were used as the boundary materials. Acrylic glass was selected for its lower thermal conductivity and easy visualization. An acrylic cover was drilled for the ease of fin top convection and thermocouple insertion. Drillings were done on specific fins at 10 mm and 30 mm heights to take the temperature readings. K-type thermocouples were used to measure the temperatures. Thermocouples were encoded with Arduino Mega. Thermocouple readings were shown on the display I2C module. Wood was used under the heat sink base to protect the lower acrylic glass from excessive heat. The heater was made with dimensions of 7 cm × 7 cm. The voltage regulator and thermocouples were attached accordingly. A voltage regulator (AC) was used to give power to the heater. A voltage of 130 V was given for 20 minutes. The current supply was 155 mA.

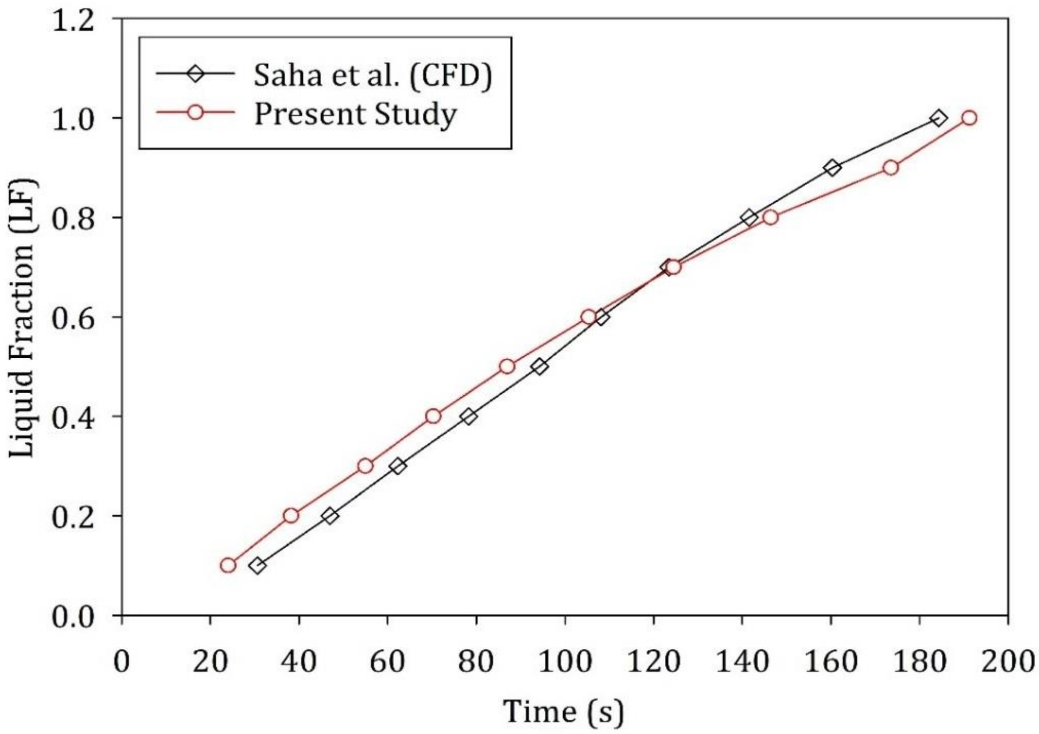


Fig. 5. Validation with numerical data from the study by Saha *et al.* [28]

3. Results

The influence of fin shapes on the cooling performance of PCM-based heat sinks was studied here. Temperature and Liquid Fraction contours and curves were analyzed. The heat was supplied at the centre of the base plate. So, the temperature at the centre of the base plate was very high. The central fins were more heated than the ones at the periphery. Data for a heat supply of 20 minutes or 1200 seconds was collected for this study.

The contours of the combined temperature of the base plate, fins, and PCM after 20 minutes of heat supply are shown in Figure 6. The Temperature legend's range was selected based on the similarity of the maximum and lowest temperatures for each case after 20 minutes. From the contours of Figure 6, it is visible that the temperature rise in the heat sink was comparatively less for the case of conical-shaped fins with and without dimples and holes.

Contours for the liquid fraction of PCM for each of the cases under investigation after a 20-minute heat supply are shown in Figure 7. In ANSYS Fluent, the liquid fraction denotes the portion or percentage of a certain volume that is made up of liquid in a multiphase flow [29]. Its value falls between 0 and 1, where 0 denotes total solidification and 1 denotes a fully melted state which is represented in the 'Liquid Fraction' legend of Figure 7. From the contours of Figure 7, it can be seen that the liquid fractions for all the cases are almost similar. For complete melting, more heat flux or more time duration must be applied. The average liquid fraction values of PCM for every case over a 20-minute duration are displayed in Figure 8(b). As shown in the Figure, after 20 minutes, the average liquid fraction of PCM for the conical fin with dimples-holes was 0.1885, the highest of all. PCM melted the most in that case by absorbing heat faster from the base. Thus, its cooling performance was better than the rest three cases. As PCM has a huge latent heat of fusion (180,000 J/kg for Paraffin wax), it easily absorbs the heat supplied at the base. As time rises, PCM becomes liquid, and convection increases.

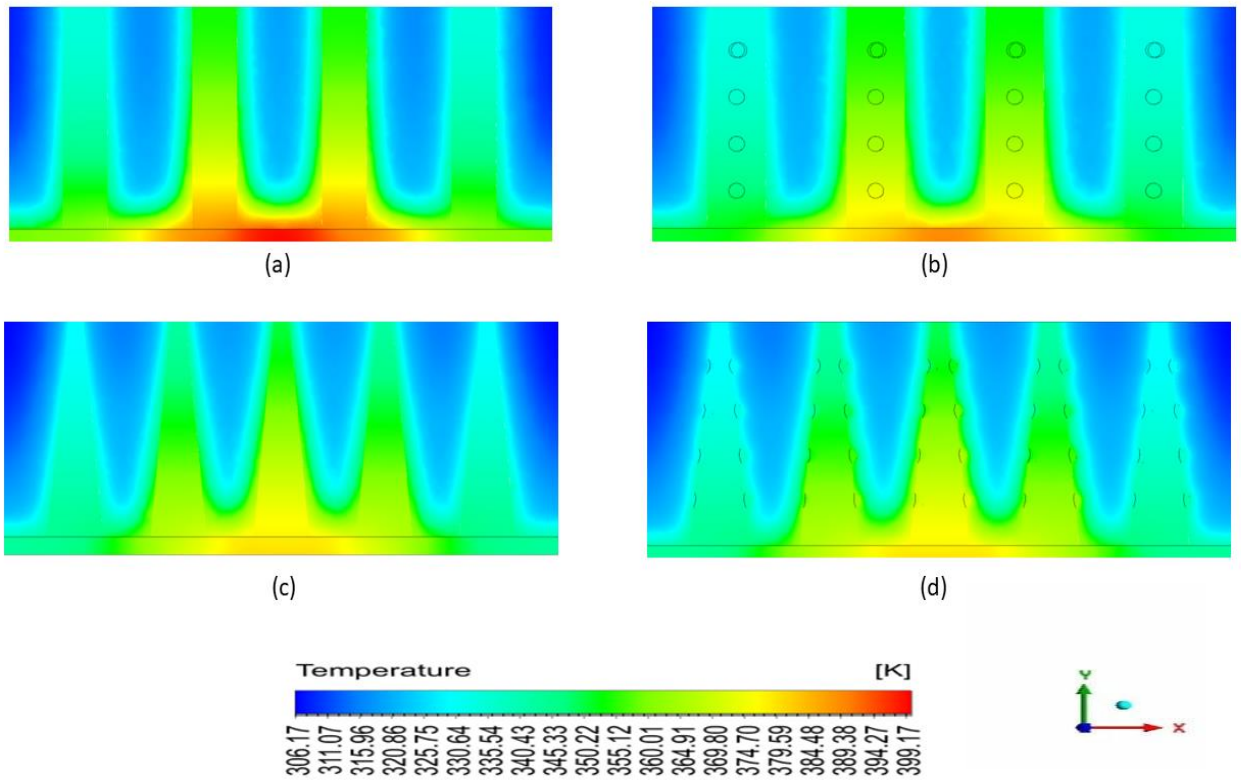


Fig. 6. Temperature contours of heat sinks after 20 minutes having (a) Pin fins, (b) Pin fins with dimples and holes, (c) Conical fins, (d) Conical fins with dimples and holes

The lower the curve of the base-plate temperature, the better the cooling capacity. A cooler base plate results from faster heat absorption from the base by the fins and the PCM above it. The base-pate temperatures of the heat sink after 20 minutes are 366.92 K, 358.75 K, 350.26 K, and 350.04 K for the fins with pin (circular) shape, pin shape with dimples and holes, conical shape and conical shape with dimples and holes respectively. The fins, which have conical shapes with dimples and holes, show the best result among the four shapes. The term 'Temperature Difference' in the y-axis of Figure 8 (a) refers to the difference of the base plate's temperature for each case at a given time from its initial temperature (300 K). The graphs in Figure 8 also demonstrate that the heat sink's base plate with conical fins (having holes and dimples) has the lowest temperature rise after 20 minutes of heat supply. The heat sink with conical fins devoid of holes and dimples is seen to be the second-best case. 'Dh' stands for Dimples and Holes in Figure 8.

Increased surface area increases heat transfer rate [15]. Surface area ratio ($\in_A$) used to compare the surface areas for different shapes [16]. The surface area ratio is the proportion of a bigger surface area ($A_{dominat\ shape}$) to a smaller one ($A_{weaker\ shape}$), shown in equation 4. Heat Sink having conical-shaped fins with dimples and holes has a temperature decrement of 0.22 K compared to the conical fins, 8.71 K compared to the pin (circular) shaped fins with dimples and holes and that of 16.88 K compared to the pin (circular) shaped fins. Firstly, taking the surface area ratio of conical-shaped fins with dimples and holes to the simple conical fins, it is obtained that $\in_A$ is 1.06. It depicts that the total surface area of conical-shaped configuration with dimples and holes is more than the surface area of (simple) conical configuration.

$$\in_A = \frac{A_{dominat\ shape}}{A_{weaker\ shape}} \tag{4}$$

Taking the surface area ratio of conical-shaped fins with dimples-holes to the pin (circular) shaped fins with dimples and holes, it is found that $\in_A$ is 1.33. Then $\in_A$ of 1.38 was found by taking surface area ratio of conical-shaped fins with dimples-holes to the pin (circular) shaped fins. This state, when the surface area of fins is smaller, it adversely affects the heat transfer rate. When $\in_A$ is more than 1, the cooling performance improves. The phase-changing heat transfer process is normalized using dimensionless analysis. Dimensionless average heat sinks temperature (θ) is used to express the temperature distribution when melting inside the heat sink [7]. Equation 5 provides its formula, whereby 'k' represents the PCM's thermal conductivity, 'T' denotes the base plate's temperature at that time,'T_{ini}'is the initial temperature,'h_f'denotes the fin-height and 'q' denotes the heat flux.

$$\theta = \frac{k(T-T_{ini})}{qh_f} \tag{5}$$

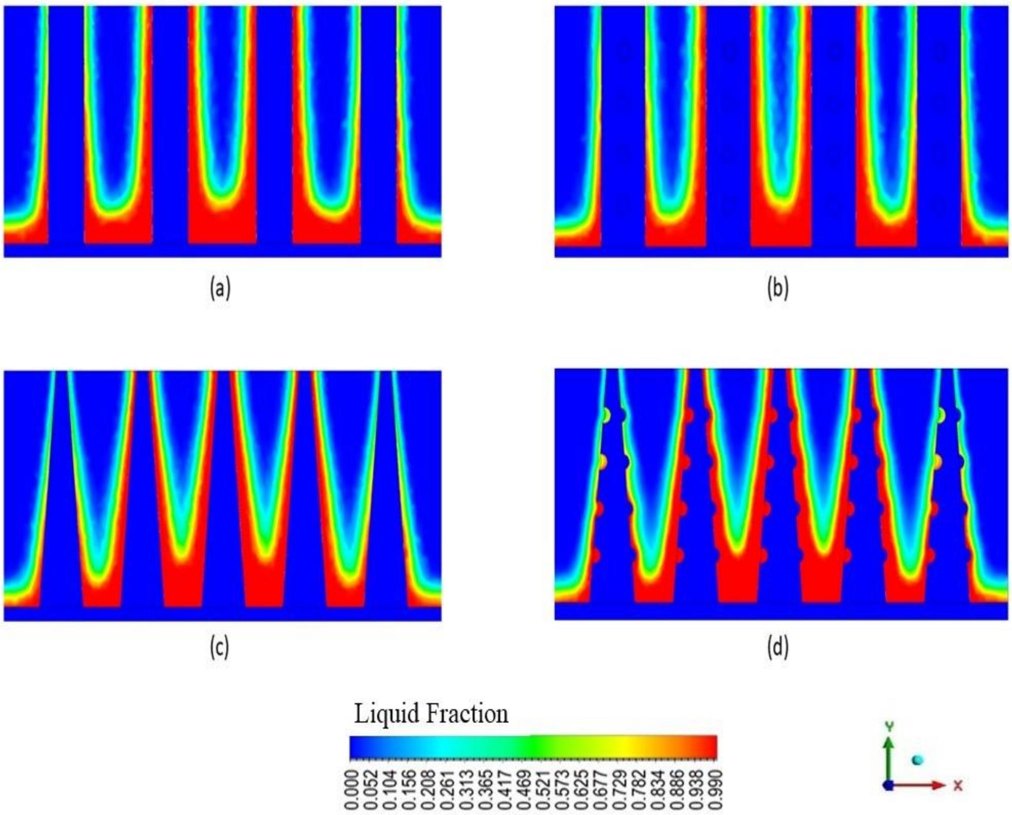


Fig. 7. Liquid-fraction contours of heat sinks after 20 minutes having (a) Pin fins, (b) Pin fins with dimples and holes, (c) Conical fins, (d) Conical fins with dimples and holes

The Fourier number (*Fo*), which is referred to as the ratio between diffusive transport and energy storage, is the non-dimensional number used in transient heat conduction problems. The time for heat diffusion over the PCM thickness is shown by *Fo* [7]. Its formula is provided in equation 6, where 'α' is the PCM's thermal diffusivity, 't' is the time (seconds), and 'h_f' is the fin height.

$$Fo = \frac{\alpha t}{{h_f}^2} \tag{6}$$

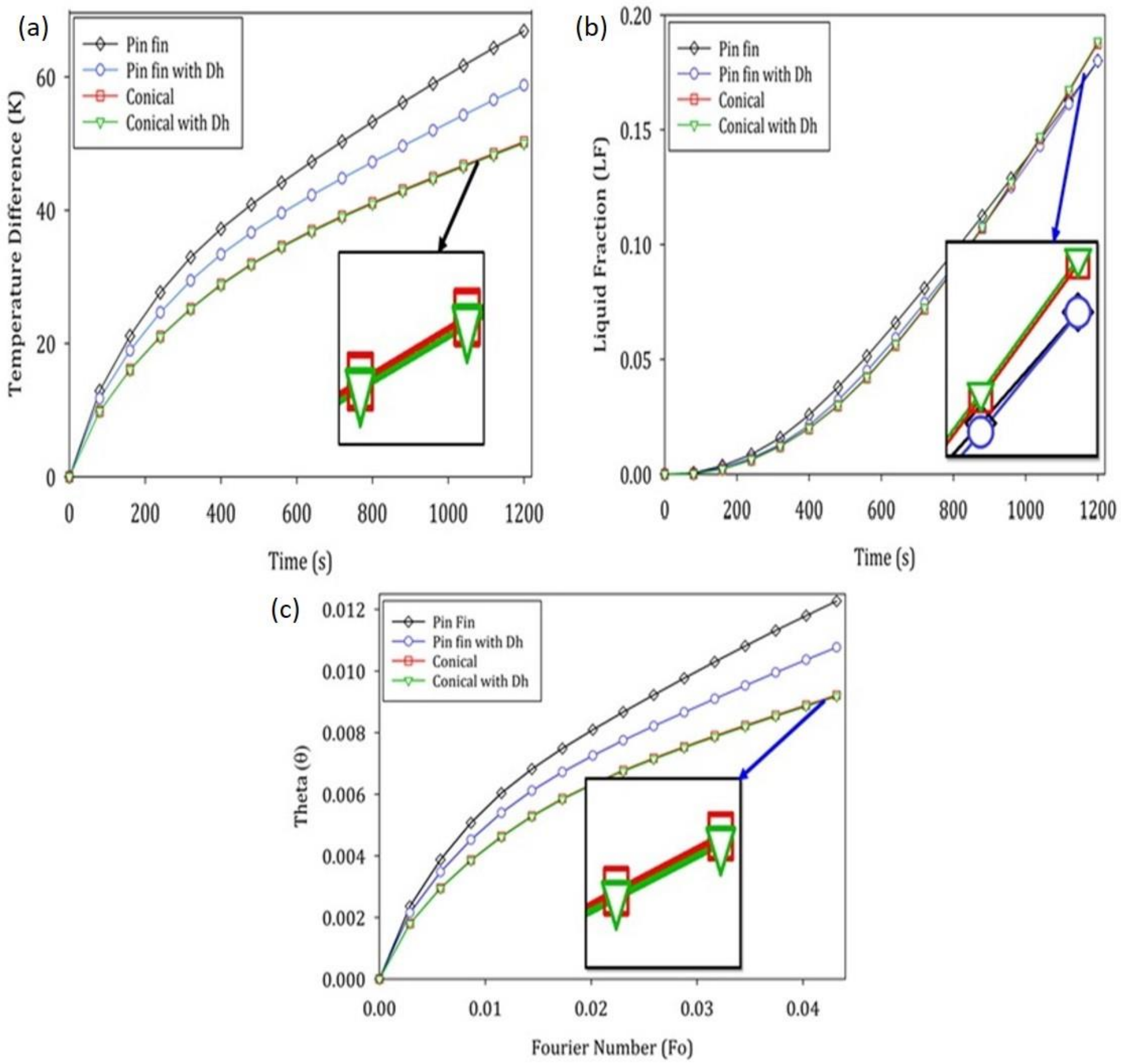
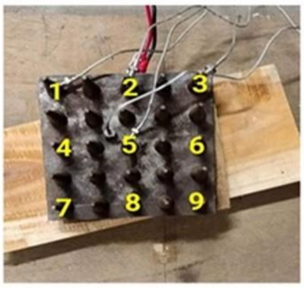


Fig. 8. The graphs of (a) Temperature difference vs time, (b) Liquid fraction vs time and (c) Variation of θ vs *Fo* for different fin shapes

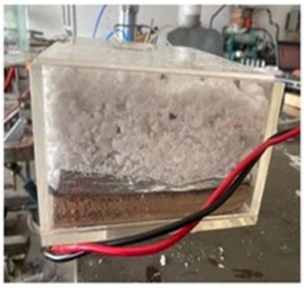
Figure 8 (c) shows the distributions of the dimensionless average temperature (θ) of the heat sinks as a function of the Fourier number (Fo) for all the fin shapes. Variations of θ in finned heat sinks demonstrate that the heat sink's interior temperature is distributed uniformly when Fo increases. Temperature differential ($\Delta T = T - T_{ini}$) and *Fo* affect the variation of θ. For a constant Fo, the higher the value of θ, the higher is the temperature rise. At a constant *Fo*, the highest θ is achieved for the pin-shaped fin case. At a constant *Fo*, the temperature rise is maximum inside the heat sink with pin-shaped fins compared to the other three shapes. At a constant *Fo*, θ is the lowest for conical fins with dimples and holes. Among all the cases, heat sinks having conical fins with dimples and holes gave the least rise to temperature, as seen in Figure 8 (c) graphs. The geometries of all these fins and the conduction-convection of PCM with them also influenced the cooling performance. Here Stefan number, Ste [7], is constant as PCM and heat flux are constant. So, Stefan number is not calculated in this study. Its formula is given in equation 7, where c_p'is the PCM's specific heat, *'q'* is the heat flux,'h_f'is the fin-height, 'k' is the thermal conductivity, and 'L' is the latent heat of fusion.

$$\text{Ste} = \frac{c_p q h_f}{kL} \tag{7}$$

(a)

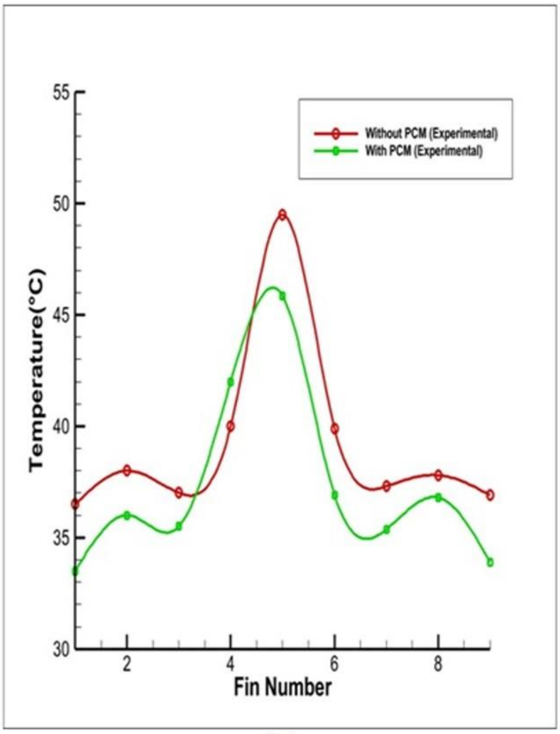


(b)



(c)

(i)



(ii)

Fig. 9. (i) Experimental setup (a) Conical finned heat sink, (b) Numbering of fins to take a temperature reading, (c) Heat sink with PCM; (ii) Fin temperature (at 30 mm distance from the base) vs Fin number

The experimental setup of the conical finned heat sink is depicted in Figure 9 (i). The layout of the nine fins selected for taking temperature readings is depicted in Figure 9 (i) (b). The x-axis in Figure 9 (ii) is labeled 'Fin Number' to denote those specific fins. PCM inside the heat sink can be seen in Figure 9 (i) (c). Figure 9 (ii) graphs show the experimental results for the heat sink having conical fins with PCM (paraffin wax) and air. The heat flux was near about 4000 W/m^2. The graphs show a temperature difference of 3°C to 2 °C between the two cases. The PCM-based heat sink proves to work better. The graph rises at the centre. It represents that the fins near the centre of the heat sink became hotter as heat was supplied. In Figure 9 (ii), unlike the other numbers, for fin number 4, the temperature is higher for the case with PCM than for those without PCM. While PCM was given, it seems that there were gaps near the fin number 4. So, the thermal efficacy of using PCM was not found for that fin. As a result, its temperature was higher than the case without PCM. Moreover, as wood was used below the heat sink for height adjustment, heat propagation seems to be disturbed in the heat sink's base.

4. Conclusions

Three-dimensional transient numerical analyses were done to observe the heat transfer phenomena of four shapes of fins in a PCM-based heat sink which are respectively, pin fins, truncated conical fins, pin fins having dimples and holes, and truncated conical fins having dimples and holes. Natural convection was applied there. The simulations were conducted for 20 minutes using paraffin wax as the PCM, maintaining a 9 % volume fraction of fins. Truncated conical fins with dimples and holes had the highest surface area. Dimensionless average temperature, θ versus Fourier number, *Fo* graphs were used to observe the cooling performance of all the cases. The key findings are stated below:

i. The best result was truncated conical fins with dimples and holes. For that shape the value of θ was 0.0092 at the *Fo* of 0.04.
ii. Pin (circular shape) fins with dimples and holes performed far better than simple pin fins, with a temperature variation of 8.17 K between the two.
iii. The cooling performances of conical fins with and without dimples and holes were closer to each other as their surface areas did not differ much. A Case with conical fins with dimples and holes performed better than the one with simple conical fins with a temperature variation of only 0.22 K.
iv. Experimental analyses were carried out for the truncated conical fins with air and PCM. Better cooling performance was found in the case with PCM having a temperature difference of 3°C to 2°C.

From this study, it can be said that conical fins, specifically with dimples and holes on the surface, give better cooling performance than the conventional pin (circular shape) fins. Fins of this shape can be used in PCM-based Heat Sinks for a variety of applications that require efficient thermal control. Potential applications include computers, laptops, batteries for electric cars (EVs) and hybrid electric vehicles (HEVs), solar thermal systems, HVAC systems, furnaces, ovens, reactors, and residential and commercial structures. Further studies can be carried out on these using different fin materials and different PCM materials. Other researchers can use our numerical study to fully melt and solidify the PCM for additional thermal performance analyses. Using our findings, they can conduct additional research into surface modification of fin-shapes for PCM-based heat sinks. Other researchers may work with our experimental study to generate more refined data and new findings.

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