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# Numerical Study on the Improvement of Flat-Plate Solar Collector Performance Using a Finned Storage Tank Containing Phase Change Materials

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## Abstract

Various types of solar heaters are designed based on different operational principles. This study focuses on the numerical analysis of the impact of latent heat storage on the performance of a solar water heater over a 24-hour period. A two-dimensional geometry was transiently modeled, and the use of extended surfaces in the air channel and the storage tank containing phase change materials (PCMs) was investigated. Using PCMs in solar latent heat storage systems allows for the storage of energy received from the sun during the day, which can then be utilized for various applications at night. The incorporation of fins enhances the rate of energy storage and release within the system, leading to improved phase change processes. According to the results obtained, increasing the flow velocity enhances the freezing rate, and the use of fins further accelerates this process compared to configurations without fins. By applying extended surfaces placed in the air channel, the heated incoming air is dispersed throughout the channel upon encountering the fins, leading to a more uniform thermal flow. Additionally, after sunset, the heat collected from the absorber is effectively transferred to the air.

***Keywords***: Numerical Simulation; Transient Heat Transfer; Phase Change Materials; Latent Heat Energy Storage; Extended Surfaces.



## Nomenclature

| | | | |
|---|---|---|---|
| $A_c$ | Momentum sink constant (Pa.s/m$^2$) | R | Gas constant (kJ/kgK) |
| $A_{mush}$ | Two-phase region constant | T | Static temperature (K) |
| $T_{liquidus}$ | Minimum temperature of liquid phase (K) | $T_{solidus}$ | Maximum temperature of solid phase (K) |
| $c_p$ | Specific thermal capacity (J/kgK) | $D_h$ | Channel hydraulic diameter (m) |
| h | Specific enthalpy (kJ/kgK) | t | Time (s) |
| k | Turbulent kinetic energy (m$^2$/s$^2$) | u | Longitudinal velocity (m/s) |
| $q''$ | Heat flux (W/m$^2$) | v | Lateral velocity (m/s) |
| $k_f$, $k_s$ | Fluid, solid thermal conductivity (W/mK) | | |

## Greek symbols

| | | | |
|---|---|---|---|
| Nu | Nusselt number | $\alpha$ | Thermal diffusivity (m$^2$/s) |

| | | | |
|---|---|---|---|
| f | Friction factor | $\lambda$ | Thermal conductivity (W/mK) |
| P | Static pressure (Pa) | $\mu$ | Dynamic viscosity (N.s/m$^2$) |
| PCM | Phase change materials | $\varepsilon$ | Radiation emissivity (W/m$^2$) |
| Pr | Prandtl number | $\rho$ | Density (kg/m$^3$) |

## 1. Introduction

The growing energy demand has increasingly shifted focus toward renewable energy sources. The global dependence on fossil fuels has resulted in detrimental environmental impacts and heightened concerns about sustainability. Rising greenhouse gas emissions and the depletion of fossil fuel reserves are significant motivators for humanity to seek and adopt renewable energy alternatives. Among these, solar thermal energy stands out as the most abundant and cleanest form of renewable energy, with the Earth absorbing more energy from the sun in just one hour than the total energy consumed in an entire year [1]. Solar air heaters are one of the key technologies designed to capture and utilize solar thermal energy, employing air as the working fluid [2]. The solar air collector utilizes air as a working fluid to capture and convert solar energy into thermal energy. In this system, air is drawn through a collector that absorbs solar radiation, heating the air before it is circulated for various applications, such as space heating or industrial processes. For example, a solar air heater can be used in residential settings to provide space heating. In this application, air is heated as it passes through a solar collector mounted on the roof. The warmed air can then be circulated into living spaces, reducing reliance on conventional heating methods and lowering energy costs. Extensive research has been conducted on phase change materials (PCMs) and their applications in solar air heating systems. One effective strategy for overcoming the limitations of solar air heaters equipped with PCM storage tanks is the integration of extended heat transfer surfaces, which helps to mitigate the low thermal conductivity of these materials [3,4]. In this system, akin to traditional solar air heaters, the energy storage material and air are heated by solar radiation during the day, while at night, the PCM releases the stored heat back into the air. This process continues until the surrounding temperature reaches the phase change temperature of the energy storage material, ensuring a consistent supply of latent heat.

Al-Hinti et al. [5] conducted an experimental study on traditional water heating systems using paraffin wax. They demonstrated that in systems with energy storage, higher water temperatures could be maintained along with extended operational hours. Additionally, there was a one-hour delay observed between the peak ambient temperature and solar radiation and the peak water temperature. As the sunlight hours drew to a close, thermal losses increased; however, the presence of PCMs allowed for the water temperature to remain stable while extending operational hours. Kumar and Rosen [6] investigated two types of solar water heaters aimed at improving convective heat transfer. In their study, they utilized a collector with a triangular-shaped absorber plate. The results indicated that reducing the distance between the peak and the trough of the absorber plate increased efficiency and enabled the production of hotter water.

Kessentini and Bouden [7] conducted a parametric numerical study. They examined the effects of concentrated reflection, absorber dispersion, and the use of dual coatings. Their analysis revealed that absorbers with low reflectivity reduced heat loss by up to 47% and increased daily efficiency from 18% to 25%. Browne et al. [8] performed an experimental study on photovoltaic systems integrated with PCMs. The results indicated that the system with PCMs exhibited a slower and delayed increase in temperature, demonstrating effective latent heat storage within the system. Papadimitratos et al. [9] utilized two PCMs with melting temperatures of 72°C and 118°C in a solar collector under vacuum conditions. Experiments were conducted under two scenarios: without flow and with circulating water in the system. The results demonstrated a significant increase in efficiency compared to systems without phase change materials.

Wang et al. [10] showed that when the spacing between two consecutive fins exceeds four times the radius of the inner tube in a heat exchanger with a PCM, the height and thickness of the fins have a diminished effect. However, when the fin pitch is small, increasing either the height or width of the fins improves the energy storage system's performance. A detailed examination of important fin parameters in a similar heat exchanger, including quantity, length, and thickness, was conducted by Wang et al. [11]. They observed that increasing both the length and number of fins positively impacted thermal performance, while the thickness of the fins had a minimal effect on the heat transfer rate. A comprehensive study on storage tanks, both horizontal and vertical, was conducted by Seddegh et al. [12]. They examined the thermal behavior of PCMs while considering the combined effects of conductive and convective heat transfer. During the charging process in the horizontal configuration, convective heat transfer significantly influenced the melting of the upper solid portion. Conversely, in the vertical tank, convective heat transfer remained constant throughout the charging process. During the discharging phase, the thermal behavior was similar in both configurations. Overall, the horizontal tank demonstrated higher efficiency compared to the vertical one.

Hamed et al. [13] conducted a numerical study on the dynamic behavior of flat-plate collectors using different PCMs. The results from this two-dimensional model indicated that, over a typical day, systems incorporating PCMs provided lower output temperatures during the charging phase and higher output temperatures during the discharging phase compared to systems without energy storage. Assari et al. [14] introduced an optimized solar collector system by testing a simple system with two types of off-center double-tube heat exchangers. The heated water flows from the top into the storage tank, and cold water is pumped from the bottom of the tank to the collector. The results showed that the melting process in the second heat exchanger, with double the off-center distance, was completed more rapidly. By lowering the heat pipe, a greater amount of PCM became available to buoyancy forces, and increasing the melting rate reduces the melting time. The degree of off-centering of the heat pipe should be optimized to ensure that, given the low thermal conductivity of the latent heat storage materials, heat does not take too long to reach the upper areas, or heat can be effectively transferred to those areas using conventional methods such as finning.

Mahdi and Nsofor [15] demonstrated that a storage solar system utilizing fins could perform better during the melting process compared to systems incorporating nanoparticles or nanoparticles combined with fins. Hosseini et al. [16] investigated the effects of fin length on the thermal performance of the storage system. They found that when the fin length increased from 13 mm to 26 mm, the melting process was completed 41% faster. Sciacovelli et al. [17] assessed energy storage in a cylindrical tank by designing innovative new fins to evaluate the system's thermal performance. The optimization of the fin design was carried out under transient conditions. The use of bifurcated fins resulted in a 24% improvement in discharge efficiency. Additionally, the operational time of the system played a crucial role in the optimal design. For short-term systems, Y-shaped fins with a larger angle between the branches were recommended, while for systems with longer operational times, fins with a smaller angle were suggested.

In the experiments conducted by Palacio and Carmona et al. [18], the use of PCMs significantly reduced fluctuations between the output water temperature and the absorber plate temperature. They observed a temperature difference or fluctuation that was 55% and 50% lower than that of the system without PCMs. Deng et al. [19] investigated several fin arrangements for a finned heat pipe of a heat exchanger with PCM energy storage and identified the optimal number and configuration. In their two-dimensional modeling, various finned and unfinned configurations were analyzed. The results indicated that the lower fin arrangement resulted in the shortest melting time and the highest storage capacity. Conversely, the upper fin arrangement provided the weakest heat transfer enhancement and the longest melting time compared to the unfinned case. The unfinned configuration required more time to complete the phase change and transferred more heat to the PCM. However, the use of fins reduced the mass of the PCM, effectively decreasing the storage capacity. The optimal fin arrangement remains unchanged even with variations in the material of the pipes and fins. Zhou et al. [20] investigated a flat-plate solar collector designed to prevent damage from freezing at night, incorporating PCMs. They found that the energy storage system needs to retain approximately 30% of the total heat received to prevent freezing effectively.

Karami and B. Kamkari [21] investigated the impact of buoyancy forces on the charging and discharging rates by varying the inclination angle. The results showed that increasing the tilt of the rectangular chamber reduced the melting time. Additionally, for a constant Stefan number, reducing the angle of the tank decreased the amount of energy stored. They proposed a relationship involving the dimensionless Stefan, Fourier, and Rayleigh numbers to predict changes in stored energy in angled rectangular tanks. Nematpour and Sheikholeslami [22] numerically studied the melting behavior of PCMs in a triangular chamber with a circular internal tube. The simulation results showed that the melting rate of nanostructured PCMs was higher compared to the non-nanostructured case, attributed to increased thermal conductivity and reduced latent heat. The addition of nanoparticles enhanced the thermal conductivity of the PCMs. The benefits of increased thermal conductivity significantly outweighed the reductions in specific heat and latent heat of the nanostructured phase change materials, effectively reducing the system's entropy production. Moreover, an increase in the Stefan number had a considerable effect on reducing melting time; as the Stefan number increased, the temperature difference between the working fluid and the PCM also increased, leading to a higher melting rate.

Ismail et al. [23] studied the effects of fin material on heat transfer. They demonstrated that fins made from materials with high thermal conductivity enhance the heat transfer between the PCM and the working fluid, thereby reducing the freezing time. Hedau and Singal [24] studied the heat transfer and friction characteristics of a double-pass solar air heater incorporating two specific roughness designs: perforated baffles and semi-cylindrical PCM tubes. The experiments were conducted with varying parameters, including blockage ratios and tube radius ratios, across a range of Reynolds numbers. The enhancements in thermal and hydrodynamical performance compared to a smooth duct were substantial, and the study also provided correlations for predicting the Nusselt number and friction factor with low mean absolute errors. Rehman et al. [25] trained and tested a deep neural network (DNN) model using data collected from a double-pass solar air heater equipped with different fin configurations and energy storage materials. Then, the results were optimized using a genetic algorithm, resulting in a maximum efficiency of 76.328%. This efficiency surpasses the average efficiency of 64.51% achieved without optimization.

Rawat et al. [26] studied the performance of a solar air heater with PCM by varying the thickness-to-length ratio of the PCM container. Both single-pass and double-pass flow configurations were analyzed. The results highlighted the importance of optimizing system parameters, particularly the thickness-to-length ratio, to maximize the energy and exergy performance, as well as the economic viability of the systems. Chaurasiya and Singh [27] explored the potential of wavy PCM units for solar air heater applications. By incorporating porous media and varying the porous region and porosity, they aimed to maximize power and energy density, particularly in low-ambient temperature conditions. Paraffin wax and wire mesh were selected as the PCM and porous media, respectively. Numerical simulations revealed that a porous fraction region of 50% and porosity of 96% yielded optimal results. Parsa et al. [28] introduced a novel solar air heater design incorporating a PCM storage bed and optimized PCM-filled baffles to tackle low heat transfer areas and reduced efficiency during periods of low solar radiation. A 3D transient numerical analysis was conducted to determine the optimal baffle geometries. Results indicated that the height of the baffles significantly impacts the daily effective efficiency. The findings demonstrated the effectiveness of the proposed design in improving the overall performance and sustainability of solar air heating systems.

Hedau and Singal [29] investigated the potential of incorporating PCM-filled baffles into solar air heaters to enhance their efficiency and performance. By varying the geometric parameters of the baffles, the study aimed to optimize thermal efficiency, heat storage rate, and pumping power. Numerical simulations revealed that specific configurations of tube amplitude ratios and relative blockage height ratios yielded significant improvements in both thermal performance and heat storage capacity. However, these enhancements came at the cost of increased pumping power. Overall, the findings demonstrate the potential of PCM-filled baffles to enhance the efficiency and versatility of solar air heating systems. A study evaluated the performance of a baffled solar air heater with and without an organic PCM storage unit by Dinesh et al. [30]. Experiments were conducted under identical conditions during two solar days in February 2021. In the solar heater, PCM incorporated at the beneath of the absorber and baffled plates, demonstrated significantly improved energy efficiency compared to the solar heater without PCM. The addition of PCM increased efficiency by 11.25%.

This study presents a novel approach to enhancing the performance of flat-plate solar collectors by integrating a finned storage tank containing PCMs. This innovative design eliminates the need for separate storage components and simplifies its construction. By incorporating extended surfaces within the storage tank, the study aims to significantly improve heat transfer rates and overall system efficiency. To gain a deeper understanding of the critical parameters influencing the thermal performance of PCM-based systems, it investigates the effects of inlet flow velocity on the system's behavior. This research will provide valuable insights for optimizing the design and operation of solar collectors equipped with PCM storage tanks.

## 2. Physical Description of the Geometry and Fluid Properties

The geometry of a two-dimensional rectangular solar heater is considered in this study. Fig. 1 provides a simplified schematic representation of the problem's geometry. The geometric parameters of the two-dimensional channel are detailed in Table 1.

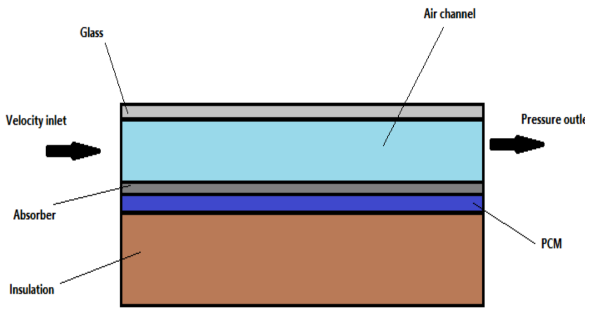



**Fig. 1. Schematic of the problem geometry**

**Table 1. The geometrical parameters of the case studied**

| Parameter | Value (m) |
|---|---|
| Length of the channel | 1 |
| Width of the air channel | 0.05 |
| Insulation thickness | 0.06 |
| Glass thickness | 0.005 |
| Absorber plate thickness | 0.003 |

In the present study, air is considered as the working fluid. The PCM used in this research is a substance called Paraffin 01, borrowed from Ref. [31]. The materials for the absorber plate, lower insulation, and fins are copper, wood, and aluminum, respectively. The thermophysical properties of air and the PCM are assumed to be constant, except for density. They are presented in Table 2. The density of the PCM varies linearly with temperature in a piecewise-linear trend using the data provided in Table 3.

**Table 2. Thermo-physical properties of air and PCM**

| Property | Air | PCM |
|---|---|---|
| $c_p$ (J/kg.K) | 1007 | 1800 |
| k (W/m.K) | 0.0263 | 0.45 |
| μ (kg/m.s) | 0.00001846 | 0.00023 |
| $h_f$ (kj/kg) | - | 224 |
| $T_{melting}$ (K) | - | 345 |
| $T_{freezing}$ (K) | - | 345 |

**Table 3. Density of PCM as a function of temperature**

| T (K) | ρ (kg/m³) |
|---|---|
| 290 | 850 |
| 345 | 810 |
| 390 | 780 |

## 3. Governing Equations, Assumptions, Boundary Conditions, and Key Variables

To obtain the flow field and temperature distribution, the governing equations, including continuity, momentum, and energy equations for the fluid, are solved numerically based on Eqs. (1) to (4) [32,33]. Furthermore, the energy equation for the PCM material is provided by Eq. (5).

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{1}$$

$$\rho\left(\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=\mu\left(\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)-\frac{\partial P}{\partial x}+A_c u \tag{2}$$

$$\rho\left(\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right)=\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial z^2}\right)-\frac{\partial P}{\partial y}+A_c v \tag{3}$$

$$\rho c\frac{\partial T}{\partial t}+\rho c\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=\frac{\partial}{\partial x}\left(k_f\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k_f\frac{\partial T}{\partial y}\right)-\rho\Delta H\frac{\partial f}{\partial t} \tag{4}$$

$$\frac{\partial H}{\partial t}=\nabla[\alpha(\nabla h)] \tag{5}$$

where $u$ and $v$ indicate the velocity in the direction of x and y, respectively, while $\mu$, $k_f$ and $\alpha$ shows respectively the dynamic viscosity, fluid thermal conductivity, and thermal diffusivity. $h$ addresses enthalpy and $P$ and $T$ illustrate the static pressure and temperature, respectively.

The enthalpy method helps to have a two-phase region to prevent discontinuities arising from phase change, which can lead to numerical instability and divergence. The two-phase region constant, $A_{mush}$ varies between $10^3$ and $10^8$ kg/m³s. This constant significantly affects the melting rate of the PCM and generally controls the thermo-hydraulic characteristics of the problem. Therefore, accurately determining this constant is essential for correctly modeling the thermal behavior of the phase change material. In this study, a mushy constant of $10^5$ is considered. The $A_c$ in the momentum equation is calculated using

$$A_c = -\frac{A_{mush}(1-f)^2}{f^3+\varepsilon} \quad (6)$$

where $f$ represents the melting fraction of the PCM and indicates the liquid portion present in each cell according to the enthalpy method. Consequently, a melting fraction of zero signifies that the phase of the material within each cell is solid.

$$\begin{cases} 0 & T > T_{solidus} \\ \dfrac{T - T_{solidus}}{T_{liquidus} - T_{solidus}} & T_{solidus} > T > T_{liquidus} \\ 1 & T_{liquidus} > T \end{cases} \quad (7)$$

$T_{liquidus}$ is the minimum temperature of the PCM in the liquid phase, while $T_{solidus}$ is the maximum temperature in the solid phase. For the melting of a pure substance, the phase change occurs at a specific melting temperature, $T_m$ making $T_{liquidus}$ and $T_{solidus}$ equal. When the temperature of the cell is below $T_{solidus}$, $f$ equals zero, and when it exceeds $T_{liquidus}$, $f$ equals unity. Otherwise, $f$ is calculated from the second condition using Eq. (7). The enthalpy function $H$ is temperature-dependent and, for constant thermo-physical properties, is defined as the sum of the sensible enthalpy and the latent enthalpy of the phase change material, defined by

$$H(T) = h(T) + \rho\, f(T)\Delta H \quad (8)$$

$$h(T) = \int_{T_m}^{T} \rho c_P \, dT \quad (9)$$

In heat transfer, the Nusselt number, presented below, indicates the rate of convective heat transfer relative to conductive heat transfer. Larger Nusselt numbers indicate a greater share of convective mechanism to heat transfer.

$$Nu = \frac{D_H \,.\, q''}{k_f\,(T_s - T_f)} \quad (10)$$

where $q''$ represents the heat flux, $D_H$ is the hydraulic diameter, $T_s$ is the surface temperature, and $T_f$ is the fluid temperature. In the present study, air is considered as an incompressible ideal gas. The equation of state for ideal gases is given by Eq. (11). Additionally, the Reynolds number is calculated using Eq. (12).

$$\rho = \frac{P_{op}}{RT} \quad (11)$$

whereby $P$ represents the system's operating pressure, and $R$ is the universal gas constant for ideal gases. The assumptions of the current work are as follows.

- The PCM behaves ideally, with no negative variables such as supercooling during freezing and no changes in thermo-physical properties.
- The PCM is considered homogeneous.
- The dependence of temperature and the density is included in the solution.
- The PCM is free of impurities, and the boundaries of the components in the problem are fixed and stationary.
- The working fluid is Newtonian and incompressible.
- The thermal conductivity and specific heat capacity of the PCM are assumed to be constant.
- The flow of the working fluid in the air channel of the collector is considered laminar and single-phase.
- The working fluid enters the air channel of the collector vertically and has a temperature equal to that of the ambient environment.
- The movement of the PCM in the reservoir beneath the absorber plate is laminar.
- The PCM is in complete contact with the absorber plate.

This study investigates a solar water heater in the city of Mosul, Iraq, on June 21 for 24 hours, using real weather conditions from the study by Ahmad et al. [34]. The lateral and upper boundaries of the solar water heater are subject to convective and conductive heat transfer. The fluid velocity at the inlet is considered fully uniform, set at 2.0 or 6.0 m/s. The inlet fluid temperature is assumed to be 298.15 K, and the pressure at the outlet is set at one atmosphere. The heat flux is equivalent to the flux from study Ref. [31], illustrated in Fig. 2.

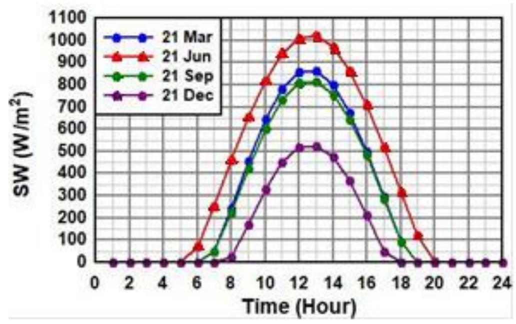
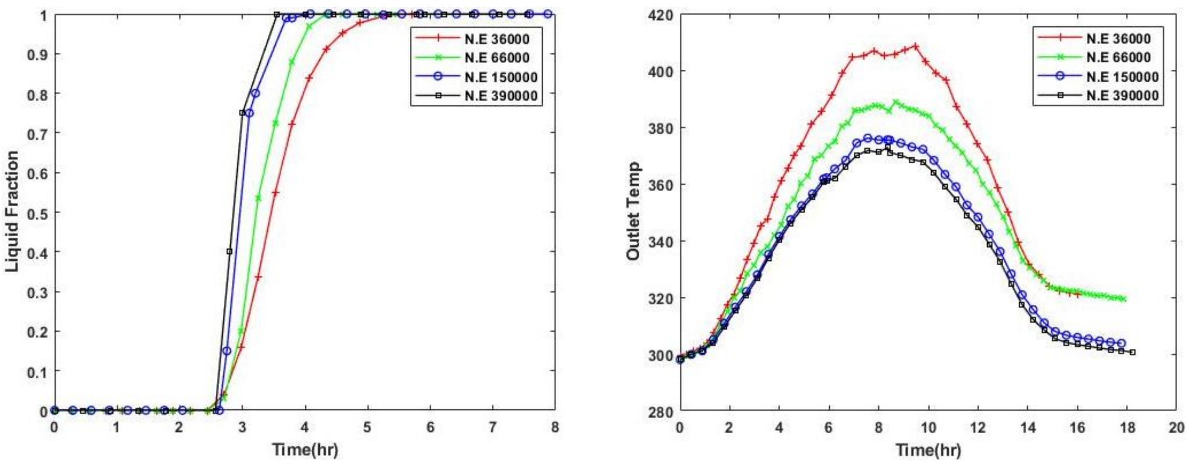



**Fig. 2. The solar heat flux (W/m$^2$) [31,35]used in the current work**

## 4. Numerical Solution Method and verification

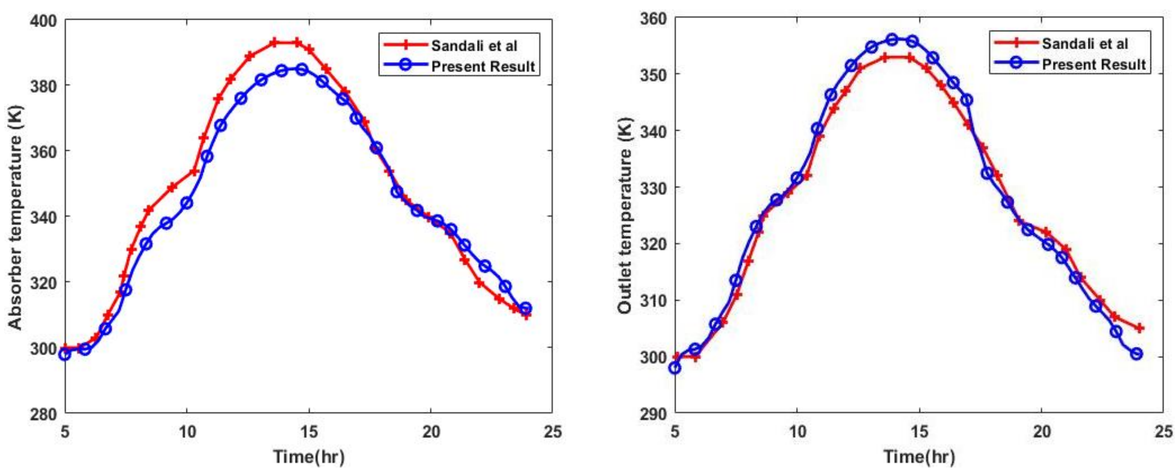
The simulation of heat transfer and airflow, along with PCM, was conducted using ANSYS Fluent, employing the finite volume method and a coupled algorithm. The pressure–velocity coupling was discretized using the SIMPLE method, while the momentum and energy equations were discretized with second-order upstream scheme. The grid independence test was examined with varying cell numbers by four different meshes. The criterion for grid independence was determined by stabilizing the melting fraction and outlet temperature parameters. As shown in Fig. 3, a mesh with 150,000 cells was selected for a relatively accurate and economical solution. To determine an appropriate time step, four different time steps of 5, 7, 10, and 20 seconds were evaluated. Considering the time required to complete the modeling for each case, a time step of 10 seconds was chosen.

To verify the current study, research conducted by Sandali et al. [31], which considered multiple PCM and various thicknesses, is utilized. This study presents graphs of outlet temperature, absorber plate temperature, and melting fraction. For verification in the current study, the obtained absorber plate temperature and outlet temperature are compared with the results from Ref. [31]. According to the results shown in Fig. 4, the mean relative error for the absorber plate temperature and outlet temperature is respectively 4% and 3%. Therefore, the present numerical simulation can accurately predict heat transfer by PCM and air in the solar air heater. The reason for comparing the absorber plate temperature is that the absorber plate in a solar heater plays a crucial role in converting solar energy into thermal energy. Typically made of a material with high thermal conductivity, such as metal, the absorber plate absorbs sunlight and converts it into heat. This heat is then transferred to a heat transfer fluid that flows through or over the plate. The effectiveness of the absorber plate is enhanced by its design, such as the use of coatings that improve solar absorption and reduce heat loss. By maximizing the collection of solar energy, the absorber plate significantly contributes to the overall efficiency of solar heating systems, making them effective for residential and commercial applications in providing hot water or space heating.




(a) (b)

**Fig. 3. Grid independence test using (a) melting fraction and (b) outlet temperature**




(a) (b)

**Fig. 4. Verification with numerical results of Ref. [31] for (a) absorber plate temperature and (b) outlet temperature**

## 5. Results and Discussion

In this section, the effects of flow velocity and the use of fins in the solar air heater and PCM reservoir are studied. The results are presented in the form of appropriate graphs for analysis. The dependent parameters studied include the melting fraction of the PCM, outlet temperature, absorber plate temperature, and Nusselt number. These parameters are evaluated in three scenarios: without fins, with fins in the air channel, and with fins in the both air channel and PCM reservoir. Table 4 presents the various variables in the cases under study.

According to Fig. 5, with the increase in airflow velocity in all cases, there is little change in the duration of the melting process throughout the daytime before sunset. However, as solar heat flux decreases and the gradual release of energy by the PCM begins, the impact of flow velocity on the phase change process, particularly freezing, becomes evident. As shown in Fig. 5, in the no-fin condition, the melting fraction of the phase change material at the end of the time studied reaches a maximum range of 0.3 to 0.4. In the case with fins in the air channel, it ranges from 0 to 0.3, while in the scenario with fins in the PCM reservoir, complete freezing occurs. Furthermore, as flow velocity increases, the freezing process accelerates. For instance, comparing case 3 with a flow velocity of 2.0 m/s and case 4 with a flow velocity of 6.0 m/s in Fig. 5 (b), it is observed that the increased velocity leads to complete freezing in case 4, while the melting fraction of case 3 is 0.3 at the midnight.

**Table 4. Various variables in the cases under study**

| Case number | Inlet velocity (m/s) | Conditions |
|---|---|---|
| 1 | 0.2 | Without fins |
| 2 | 0.6 | Without fins |
| 3 | 0.2 | Fins in the air channel |
| 4 | 0.6 | Fins in the air channel |
| 5 | 0.2 | Fins in the both air channel and PCM reservoir |
| 6 | 0.6 | Fins in the both air channel and PCM reservoir |

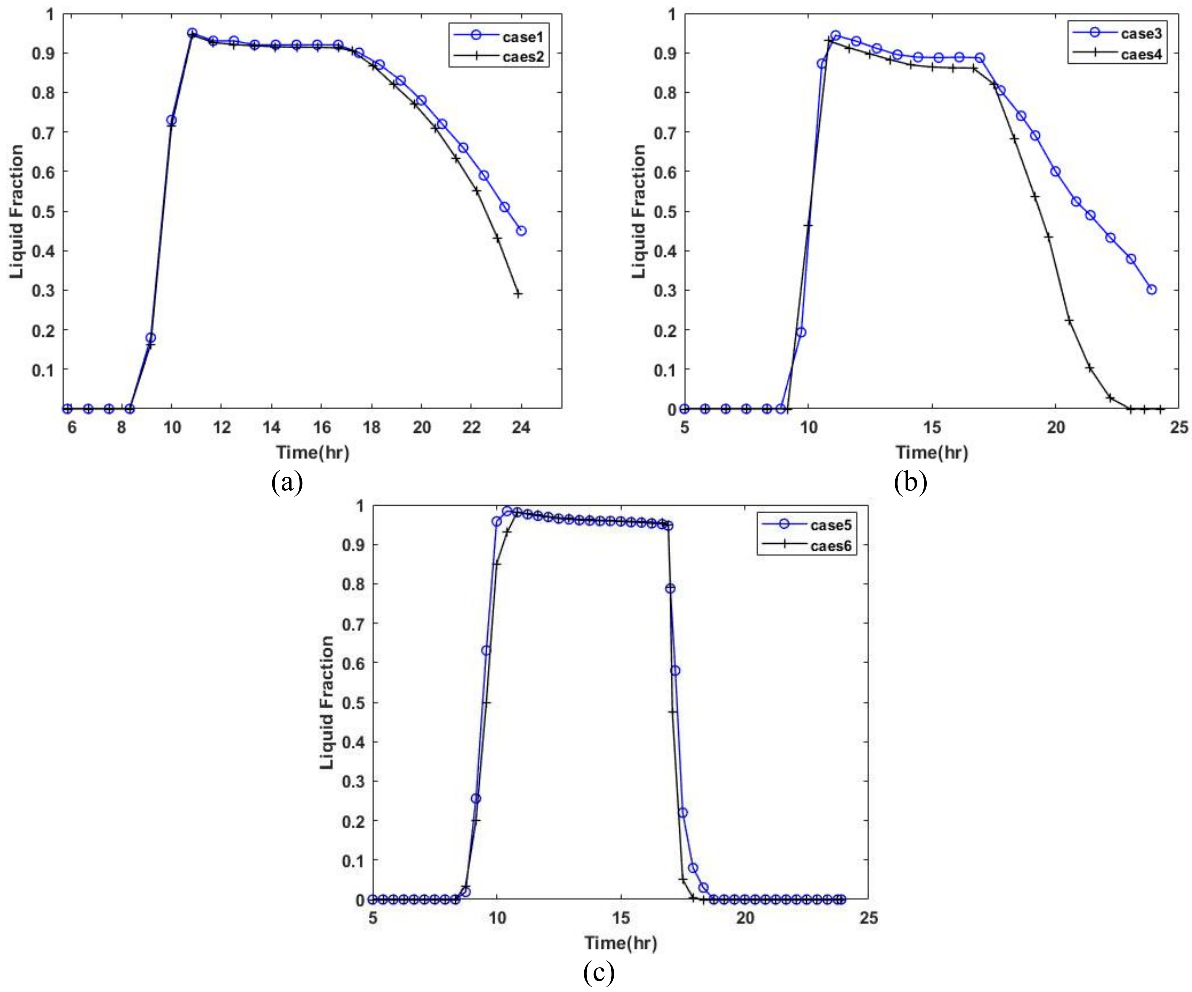



**Fig. 5. Comparison of the Effect of flow velocity on the phase change process for case of (a) without fins, (b) with fins in the air channel, and (c) with fins in the air channel and reservoir**

In the no-fin cases, as shown in Fig. 6 (a), the absorber plate temperature remains nearly constant at different flow velocities. In the finned cases, however, Fig. 6 (b) clearly shows a reduction in the absorber plate temperature. The maximum value of the absorber temperature occurs at hour 15. The maximum temperature decreases by using the fins either in the channel or channel and reservoir (Fig. 6(c)). The difference between the maximum temperatures for cases with different inlet velocities become larger by applying the fins; it is such that the difference can touch 40K by increasing the inlet velocity. The figure shows that rising the inlet velocity and applying fins can reduce the temperatures, meaning a better cooling.

Fig. 7 arranges a similar figure to Fig. 6, except for the outlet temperature. Comparing Fig. 7(a) and (b), it can be observed that the outlet temperature in the finned case is, on average, 12 degrees Celsius higher at midday than in the no-fin condition. Additionally, as seen in the figure, the outlet temperature during the night for the case with fins in the air channel and PCM reservoir is higher than in the other two scenarios. With an increase in airflow velocity, there is less time for heat exchange between the air and the absorber plate, leading to a smaller increase in its temperature. Conversely, the absorber plate will experience a greater temperature drop due to enhanced heat dissipation to the passing air. Consequently, with increasing flow velocity, both the outlet temperature and the absorber plate temperature will decrease.

Fig. 8 shows the variations in Nusselt number across different flow velocities for the cases with or without fins. It is evident that the applying fins improves the Nusselt number; it is such that the Nusselt number increases by a factor of about 2.5 and 1.5, on average, using fins in the air channel, respectively for the lower and higher inlet velocity. The difference between case 3 and 4 touches 80%, showing the effect of inlet velocity increment. The counterpart increase for using fins in both air channel and PCM reservoir is about 30%. Additionally, the figure shows that using the fins only in the air channel improves more in the Nusselt number compared to that with fins in both air channel and PCM reservoir. Besides heat transfer enhancement, the benefits of applying fins is keeping the Nusselt number with the lowest value of variations during the most day time and even after sunset.

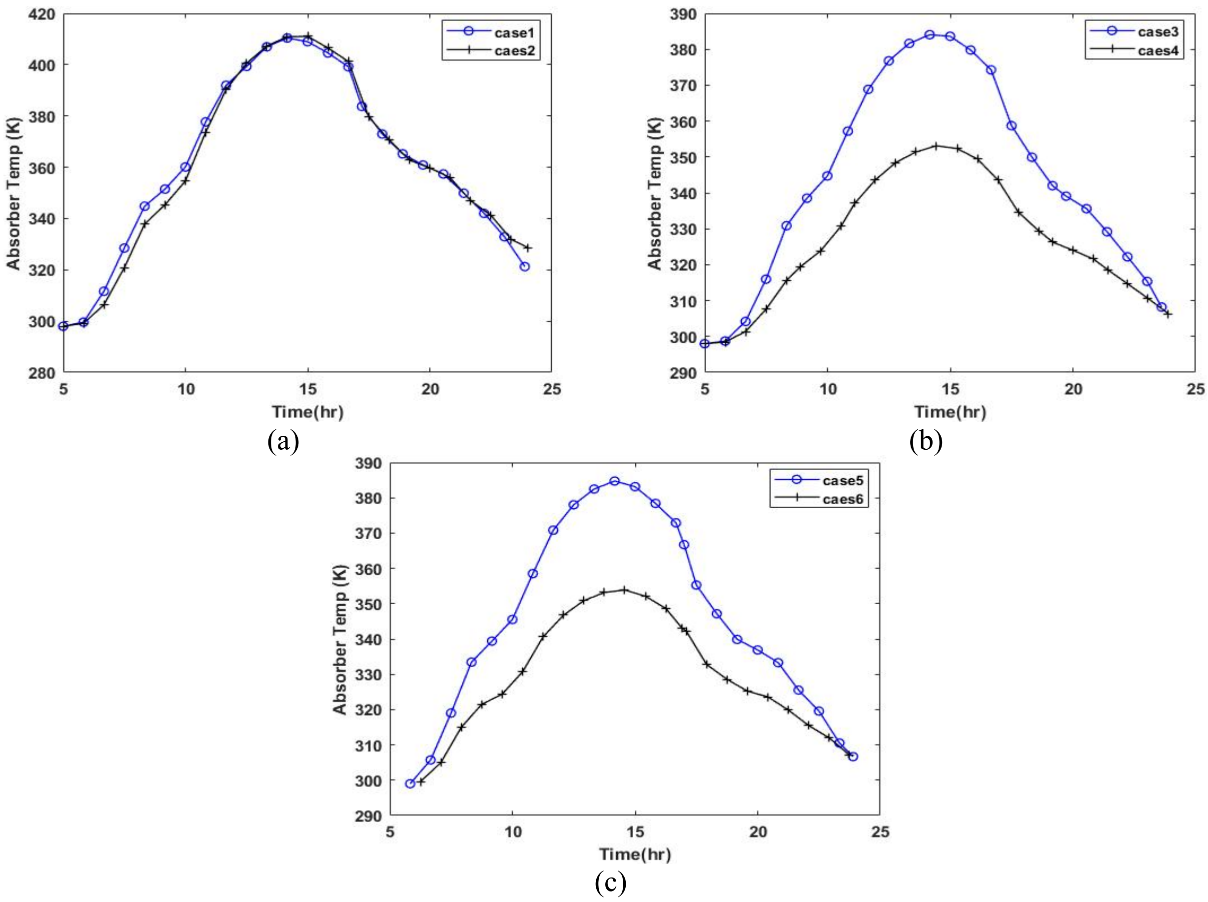
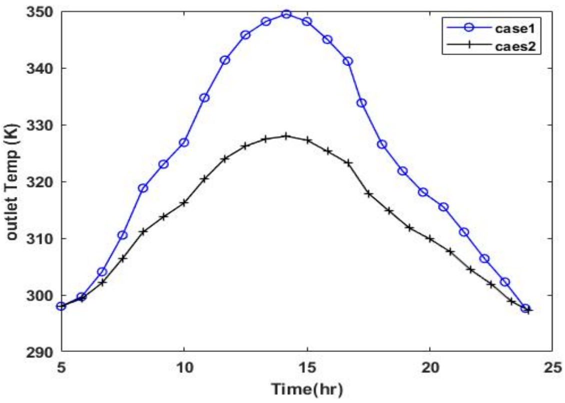
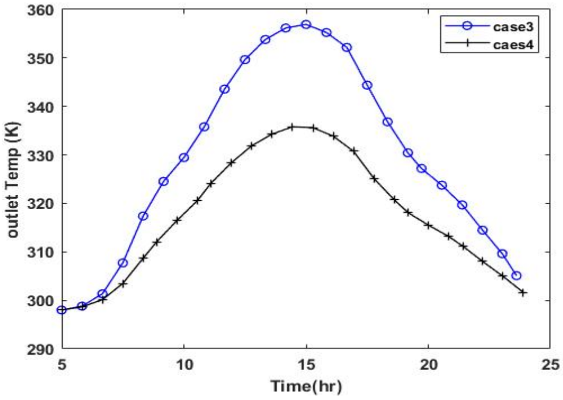


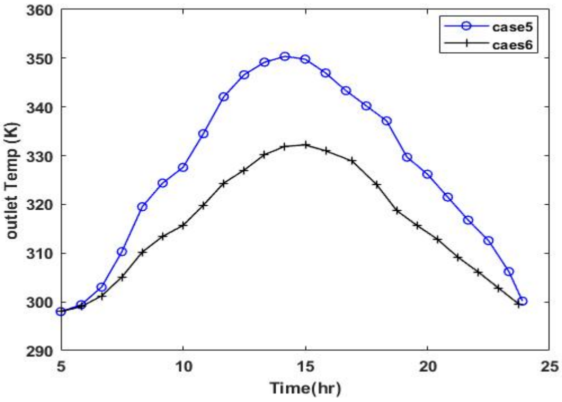

**Fig. 6. A comparison of the effect of flow velocity on absorber plate temperature among cases of (a) without fins, (b) with fins in the air channel, and (c) with fins in the air channel and reservoir**




(a)

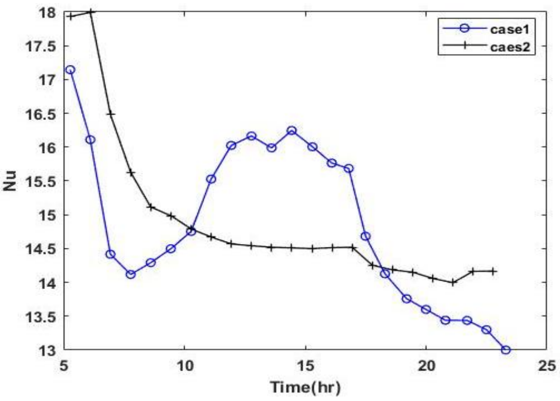



(b)

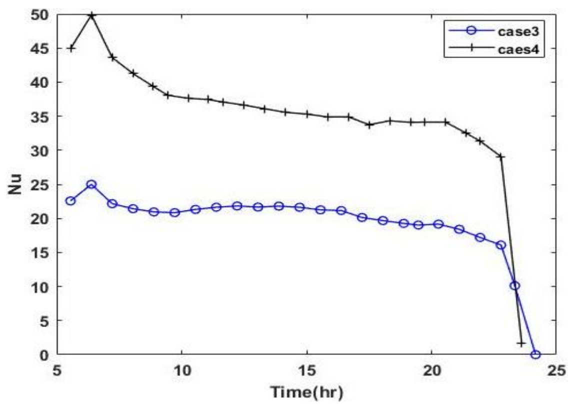



(c)

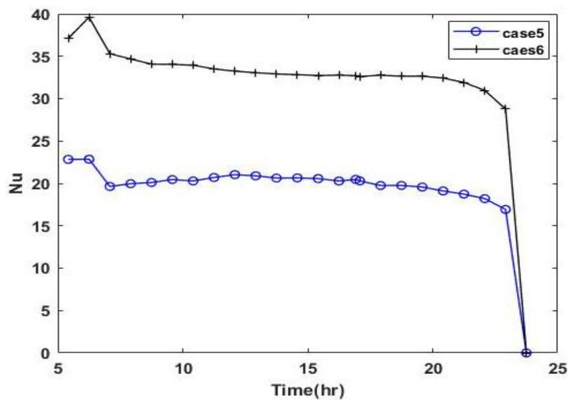
**Fig. 7. A comparison of the effect of flow velocity on the outlet temperature among cases of (a) without fins, (b) with fins in the air channel, and (c) with fins in the air channel and reservoir**




(a)




(b)




(c)

**Fig. 8. A comparison of the effect of flow velocity on the Nusselt number among cases of (a) without fins, (b) with fins in the air channel, and (c) with fins in the air channel and reservoir**

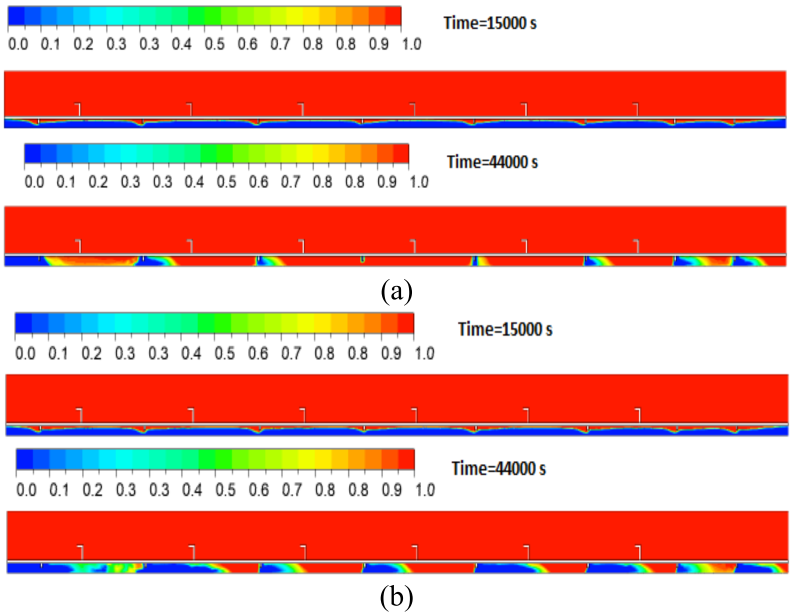



**Fig. 9. Melting fraction contours in various times for (a) case 5, (b) case 6.**

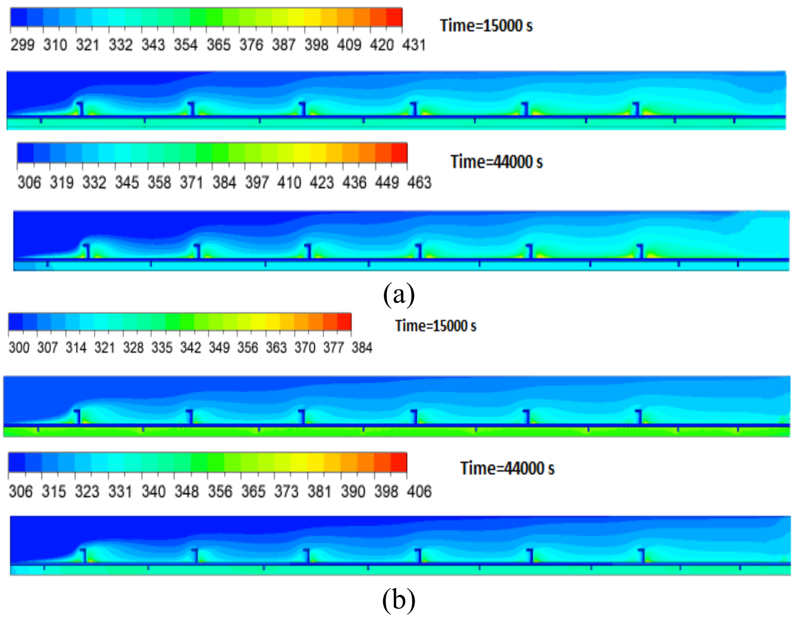



**Fig. 10. Temperature contours in various times for (a) case 5, (b) case 6.**

To examine the melting pattern of the PCM and the impact of using fins on the thermal behavior of the system, contour plots of the melting fraction and temperature along the air channel are presented in a two-dimensional view at two different times of the day, respectively in Fig. 9 and 10. The time of presentation in these figures is 9AM and 17PM. With the implementation of fins in the reservoir, phase change initiates around the fin areas. As shown, the use of fins enhances heat transfer within the system, significantly reducing the melting and freezing process times. Increasing the inlet velocity speeds up the phase change process. As flow velocity increases, both the outlet temperature and the absorber plate temperature decrease. The heated air, upon contacting the fins, moves away from the absorber plate and disperses in the channel. The incorporation of fins results in an increase in outlet temperature while substantially lowering the temperature of the absorber plate. The same effect is found by increasing the inlet flow velocity.

## 6. Conclusions

In the present study, a numerical simulation of a latent heat storage system integrated into a solar air heater with a phase change material (PCM) reservoir was conducted. The study examined the impact of key parameters in latent heat storage systems, such as flow velocity and the effect of fins in the air channel and PCM reservoir. The significant findings of this research are summarized below.

- **Output Temperature Analysis**: Considering two coefficients of 1 and 1.5 for the baseline heat flux, the outlet temperature and absorber plate temperature were analyzed under different conditions. The results indicated that increased heat flux rose both the outlet temperature and the absorber plate temperature. In the finned condition, the absorber plate temperature rose less than in the no-fin condition with increasing heat flux.
- **Nusselt Number Variations**: The Nusselt number increased with higher heat flux. In the finned condition, the Nusselt number significantly rose compared to the no-fin condition with increased flow velocity. For similar flow velocities, the Nusselt number was higher in the finned samples than in the no-fin ones.
- **Flow Velocity Influence on Freezing**: Increasing the airflow velocity in the examined samples did not significantly alter the melting process. However, as the heat flux decreased and the PCM gradually released energy, the effect of flow velocity on the freezing process becomes apparent. An increase in flow velocity enhanced the freezing rate, and the incorporation of fins accelerates the freezing process compared to the no-fin scenario.
- **Heat Exchange Dynamics**: With higher flow velocities, the incoming air has less time for heat exchange with the absorber plate, leading to a decrease in outlet temperature. Conversely, the absorber plate experienced a greater temperature drop due to enhanced heat dissipation to the passing air. Consequently, both the outlet temperature and absorber plate temperature decreased with increasing flow velocity, with a notable reduction in the plate temperature in the finned condition. In the finned scenario, the outlet temperature was 12°C higher than in the no-fin condition.

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