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Numerical Simulation and Experimental Study of Thermal Performance in Double-Pass Solar Water Heaters With Paraffin Wax Storage

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ABSTRACT

Solar water heaters are widely used to convert solar energy into thermal energy for residential, commercial, and industrial applications. This study investigates the thermal performance of a double-pass solar water heater (DPSWH) using numerical and experimental approaches under different water mass flow rates. Three configurations were evaluated: a flat absorber plate, a V-corrugated absorber plate, and a V-corrugated absorber plate integrated with paraffin wax-filled aluminum tubes for thermal energy storage (TES). Numerical simulations were conducted using ANSYS Fluent and validated through experimental testing under climatic conditions of Multan, Pakistan at mass flow rates of 0.003, 0.005, and 0.007 kg/s. Results showed that DPSWH with TES achieved the highest thermal performance. A maximum inlet–outlet water temperature difference of 39°C was obtained at 0.003 kg/s while a minimum difference of 35°C occurred at 0.007 kg/s. Solar fraction values ranged from 0.01 to 0.20. Furthermore, incorporation of paraffin wax enhanced thermal efficiency and prolonged heat delivery compared to systems without thermal storage. The findings demonstrate that lower mass flow rates improve utilization of stored thermal energy, resulting in enhanced system performance and extended hot-water availability.

1 | Introduction

The development of eco-friendly and energy-efficient technologies is necessary as global energy demand continues to escalate and environmental concerns intensify [1]. Among them, solar water heaters stand out as a promising solution to meet the demand for clean and cost-effective hot water, capitalizing on inexhaustible solar radiation to provide a sustainable alternative to conventional heating systems [2]. The solar water heater consists of a solar collector often mounted on rooftops, which

absorbs sunlight and transfers it to thermal heat; a storage tank to store heated water; a heat exchanger; and a flowing chamber. In an active kind of system, a pump is used to circulate heated water to the storage tank, while in a passive system, water flows by the natural convection. The consumption of energy can be reduced by using a free source of solar energy for heating water. Solar water heaters are categorized based on two main criteria: the method of fluid circulation and type of solar collector used [3]. The first criterion involves how fluid moves between the

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storage tank and the solar collector. In passive systems, water circulation is driven by natural convection or gravity, with heated water rising from the collector to a storage tank placed above it. In contrast, active systems rely on pumps or fans to actively move the fluid between components. The second criterion focuses on the different types of solar collectors employed in solar water-heating systems. It includes flat-plate collectors, which feature an insulated box with a black absorber plate and evacuated tube collectors. It further consists of glass tubes with metal heat-conductive fins and parabolic trough collectors, which use curved reflective surfaces to concentrate on sunlight [4]. The intermittent nature of solar power and the challenge of maintaining a steady supply of hot water during periods of low sunlight availability have stimulated innovative research efforts [5].

Currently, phase change materials (PCMs) have become widespread and common in industrial, construction, and residential fields. Many appliances are being equipped with PCMs one after the other to utilize the best available energies, such as solar energy [6]. The integration of PCMs is characterized by their ability to store and release thermal energy during phase transition. It has emerged as an innovative strategy to increase the efficiency and performance of solar water-heating systems. Solar water heaters absorb heat energy from the sun and use it to heat water for home use [7]. These systems are made of some large components, such as collectors, temperature changers, storage tanks, PCM material, and valves (e.g., internal and external). It has a coating that acts to capture solar energy and convert it into heat. A PCM is a material with high thermal conductivity and melts and solidifies at a certain temperature, capable of storing and releasing large amounts of energy [8]. Heat is absorbed or released when the material changes from solid to liquid and vice versa. PCMs are thus classified as subtropical heat storage units (HSU). Paraffin wax is safe, reliable, predictable, inexpensive, and noncorrosive [9]. Paraffin waxes have been particularly popular due to their promising properties, such as phase transformation. Paraffin forms a good PCM solution for some applications and selected temperatures. It has a good heat retention capacity, and materials have been proven to freeze without much cooling. The advantages of using paraffin are chemical stability to many heating and cooling cycles, high combustion temperatures, compatibility with all materials, and no longer being applicable to many composite materials [10].

The solar water heater is mainly divided into two sections, such as setting up a water flow channel and constructing a water flow channel. The first phase relates to the configuration of the water-course. The efficiency of the system increases by changing the configuration of the water flow channel [11]. According to the water flow, the solar water heater might be single flow and single pass, double flow and single pass, or double flow and double pass. The efficiency of the system is further improved by changing the structure of the water flow channel by creating a flat plate and V-shaped plates [12]. A V-plate water heater works much better than a flat-plate water heater. This type of water heater consists of a glass cover located on top and a dimple of about V-shape. The whole side of the solar water heater is covered with inserts, such as glass wool and rock wool. The V-plate was designed as a single flow and a double flow. The purpose of the V-shape dimple is to provide a larger surface area and create a barrier to water flow.

When water enters the inlet valve, these dimples produce resistance, which is why the time level automatically rises. This device increases the efficiency of water temperature [13].

Solar thermal collectors are widely used to capture solar energy and convert it into thermal energy, which can be then stored for future usage purposes [14]. Solar thermal collectors are commonly utilized mainly for water heating as well as heating air or specific zones. Solar thermal collectors are generally categorized into two main categories, such as concentrating and non-concentrating systems. The concentrating solar thermal collectors are generally used in large-scale solar power plants, while nonconcentrating kind solar thermal collectors are used for domestic heating purposes. Among all solar thermal collectors, flat-plate solar collectors are widely used part of solar-based heating systems, as they are the most efficient system at a lower temperature range [15]. A flat-plate collector consists of a coated absorber plate and a glazing system that reduces heat loss from the absorber's upper surface. It also includes a heat-transfer fluid and insulation to minimize thermal losses, as well as a protective frame that shields the system from moisture and dust [16].

Luo et al. [17] developed a thermal model of a solar collector integrated with a PCM and demonstrated that the PCM can effectively regulate outlet temperature fluctuations while improving thermal efficiency by storing excess solar energy during periods of peak radiation. The integration of thermal energy storage (TES) has attracted considerable attention as it addresses one of the major limitations of solar water-heating systems: the mismatch between solar energy availability and hot-water demand. Abo-Elfadl et al. [18] investigated a double-pass solar air heater with a modified absorber consisting of adjacent aluminum guide tubes and reported enhanced heat transfer due to the increased contact area between the working fluid and absorber surface [19]. Al-Kayiem et al. [20] examined the performance of PCM nanocomposites and Cu-PCM in solar water-heating applications and showed that latent heat storage can significantly improve temperature retention compared with systems operating without TES. Chen et al. [21] reported that a two-phase thermosiphon solar water heater incorporating enhanced heat-exchange components achieved an 18% improvement in thermal efficiency by reducing heat losses. Hassan et al. [22] demonstrated that corrugated absorber configurations enhance thermal performance by increasing turbulence intensity and heat-transfer area compared with flat absorbers. Mohammed et al. [23] and Elamy et al. [24] highlighted the effectiveness of combining thermal storage materials with advanced absorber designs to improve the solar thermal system productivity and energy utilization. Benli et al. [25] compared various collector configurations under different climatic conditions and emphasized the significant impact of collector geometry on overall system performance. Ho et al. [26] investigated flat-plate solar water heaters with modified flow channels and reported improved heat-extraction characteristics. Chargui et al. [27] investigated the application of solar water heaters integrated with PCM modules for heating football stadiums in Tunisia. Using TRNSYS and EES simulations, the study demonstrated the feasibility of incorporating PCM-based thermal storage to enhance the utilization of solar energy and improve the system performance under varying operating conditions. Bouhal et al. [28] designed a solar water storage

tank integrated with PCM modules for domestic water-heating applications. The results indicated that the complete melting process of the selected PCM required ~ 2.5 h, while increasing the PCM quantity improved thermal storage capacity but reduced the melting rate due to larger energy absorption requirements. Han et al. [29] reviewed the rapid development of the solar water-heating technology in China over recent decades. Their findings indicated that the widespread adoption of solar water heaters has increased significantly, resulting in substantial economic, environmental, and social benefits for the country. Harmim et al. [30] briefly study the design, construction, and testing of an integrated solar water heater with a collector. Chaabane et al. [31] performed a thermal and statistical analysis of an integrated collector-storage solar water heater incorporating PCM. The study confirmed that latent heat storage materials can improve thermal energy retention and contribute to more stable water temperatures during periods of low solar radiation. Fazilati et al. [32] experimentally investigated the effect of incorporating PCM in solar water-heating systems and reported that TES significantly enhances heat retention capability and extends the availability of hot water beyond sunshine hours. Abdullah et al. [33] evaluated the performance of the new double-pass solar heater with and without tabs. Yogesh et al. [34] conducted a comprehensive review on recent developments and innovations of flat-plate solar collectors for enhanced thermal efficiency using reflectors in solar thermal water heaters. The study investigated spiral-shaped flow tubes in a flat-plate solar collector and tested them with three water mass flow rates, such as 0.0058, 0.0083, and 0.0117 kg/s. The optimum thermal efficiency of 70.8% is recorded at a mass flow rate of 0.0117 kg/s, which is 13.7% higher compared to conventional collectors [35]. Based on the literature reviewed, extensive experimental, numerical, and theoretical investigations have been conducted to improve the thermal performance of solar water-heating systems. Previous studies have primarily focused on individual enhancement techniques, such as modified absorber geometries, double-pass flow arrangements, or incorporation of PCMs for TES. However, limited research has addressed the combined effect of V-corrugated absorber

configurations and PCM-based thermal storage within a double-pass solar water heater (DPSWH) under varying operating conditions. Furthermore, comprehensive computational fluid dynamic (CFD) analysis coupled with experimental validation for such integrated systems remains scarce in the literature. Therefore, a significant research gap exists in understanding the thermohydraulic behavior and thermal performance of a DPSWH employing V-shaped flow-enhancement structures and paraffin wax thermal storage capsules. Addressing this gap can provide valuable insights for improving heat transfer, extending thermal energy availability, and enhancing overall efficiency of solar water-heating systems.

The main objective of this work is to design, develop, and evaluate a high-performance DPSWH integrated with latent TES. Three DPSWH configurations, such as a flat absorber plate, a V-corrugated absorber plate, and a V-corrugated plate with paraffin wax-filled aluminum tubes, were investigated using numerical simulations and experiments at mass flow rates of 0.003, 0.005, and 0.007 kg/s, with measurements conducted from 9:00 am to 5:00 pm. The novelty of this study lies in combining V-corrugated flow enhancement with PCM-based thermal storage, validated through comprehensive CFD and experimental testing. The optimal design identified via simulations was fabricated and tested, showing strong agreement between the experimental and numerical results. Incorporating V-shaped baffles and PCM tubes significantly improved the system's thermal performance.

2 | Materials and Methods

2.1 | Design Configuration of Collector Plate

The thermal performance of DPSWH with three different design configurations is carried out for various mass flow rates in order to examine thermal characteristics. For this purpose, a numerical simulation approach is used. In addition, an experimental study was also conducted to verify numerical findings. Figure 1 describes the comprehensive research methodology for the current research work.

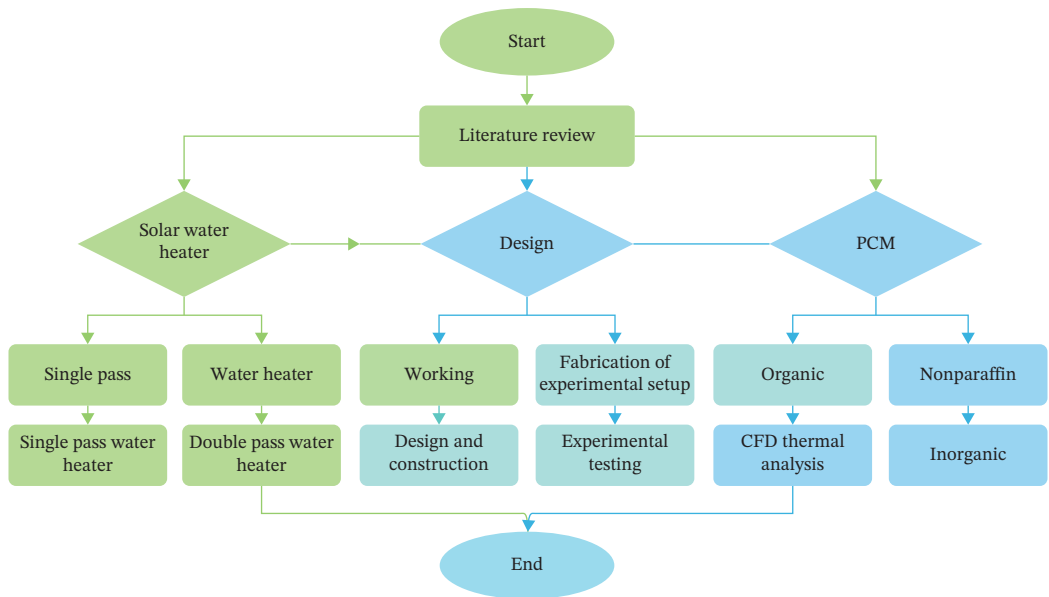


FIGURE 1 | Methodology flow chart.

The thermal performance of DPSWH was investigated using three different configurations: a flat absorber plate, a V-corrugated absorber plate without PCM, and a V-corrugated absorber plate integrated with PCM. Three water mass flow rates, such as 0.003, 0.005, and 0.007 kg/s, were selected to evaluate the performance of each configuration. Experimental tests were repeated on three different days, and average values were used for performance analysis and comparison. Water temperatures at the inlet and outlet sections were measured using calibrated temperature sensors. Figure 2 illustrates the design configurations of the DPSWH, while physical parameters of the collector are summarized in Table 1. In the first configuration, a flat mild-steel absorber plate was employed without any TES medium. Water entered the collector through the inlet valve and exited through the outlet valve, while the temperature rise was monitored at both locations. The second configuration utilized a V-corrugated absorber plate without a PCM to enhance heat transfer by increasing effective heat-transfer surface area and fluid residence time. In the third configuration, paraffin wax encapsulated in

TABLE 1 | Physical parameters of SWHS collector.

Sr. number	Design parameters	Specifications
1	Glass area	0.0036576 m ²
2	Collector glazing	Plain glass with 0.005 m thickness
3	Absorber plate	Width: 0.600 m, length: 1.700 m

aluminum tubes was incorporated as a TES medium to retain excess thermal energy and provide extended heat delivery after sunset. V-corrugated elements with a length of 25.4 mm were integrated into the absorber plate to promote flow disturbance and improve thermal interaction between the absorber surface and flowing water. Figure 2a,b present V-corrugated DPSWH without and with PCM, respectively, whereas Figure 2c shows the flat-plate DPSWH configuration.

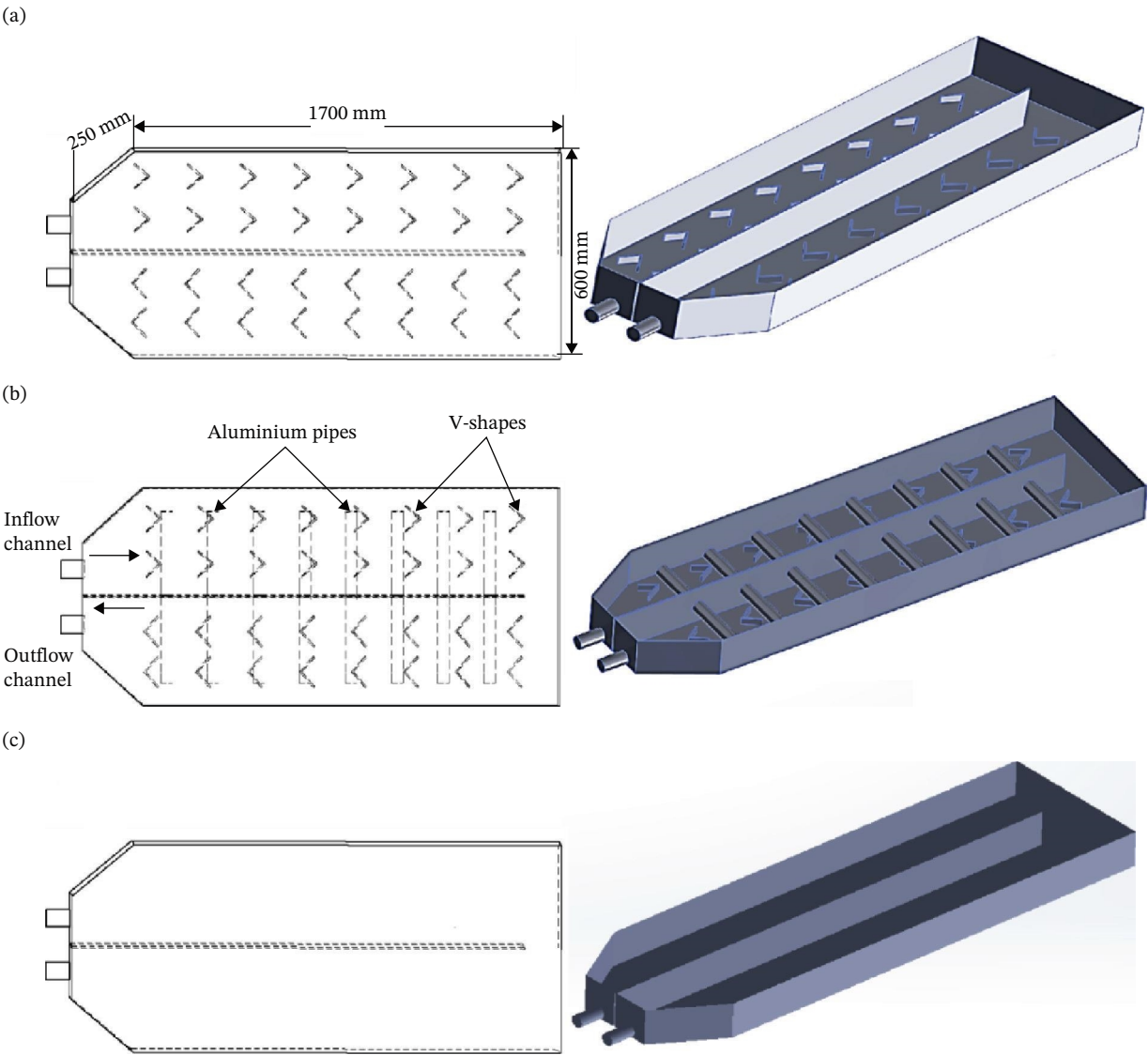


FIGURE 2 | Design configuration of DPSWH: (a) V-plate open channel DPSWH, (b) V-plate with PCM open channel DPSWH, and (c) flat-plate open channel DPSWH.

2.2 | Thermal Performance Parameters

The thermal performance of the DPSWH was evaluated using key heat transfer and energy-based parameters, including useful thermal energy gain, Reynolds number, convective heat-transfer coefficient, instantaneous thermal efficiency, and daily thermal efficiency. These parameters provide a quantitative assessment of heat-transfer characteristics and energy conversion capability of the system under different operating conditions. The governing equations used to determine the thermal performance of the DPSWH are presented in the following section. The useful thermal energy gained by water flowing through the collector (Q_u) is calculated using Equation (1) [36].

$$Q_u = \dot{m} C_p \Delta T_w, \quad (1)$$

where

$\dot{m}$ = Water mass flow rate through the heater ($\frac{kg}{s}$), C_p = Specific heat of the water ($\frac{J}{kg \cdot ^\circ C}$), and ΔT_w = Temperature difference of water across heater ($^\circ C$). The mass flow rate through heater ($\dot{m}$) is given by Equation (2).

$$\dot{m} = \rho_w A_c v_w, \quad (2)$$

where

ρ_w = Water density through the heater

A_c = Area of heater exit pipe (m^2)

v_w = Water velocity at exit of a solar heater pipe ($\frac{m}{s}$).

Besides, Reynolds number of flow (Re) is given in Equation (3).

$$Re = \frac{\dot{m}}{\pi d \mu}, \quad (3)$$

where

d = diameter of heater that exits pipe (m)

μ = Dynamic viscosity of flowing water.

The instantaneous thermal efficiency of the heater (η_{ins}) is given in Equation (4).

$$\eta_{ins} = \frac{Q_u}{IA_p}, \quad (4)$$

where

A_p = Heater projected area (m^2)

I = Solar radiation incident on solar heater ($\frac{W}{m^2}$).

The convective heat-transfer coefficient between water and the absorber plate is given in Equation (5) [37];

$$h = \frac{Q_u}{A_p (T_p - T_w)}, \quad (5)$$

where

Q_u = Heat gain by water (W)

T_p = Average value of absorber plate temperature ($^\circ C$)

T_w = Average water temperature in the heater ($^\circ C$).

The daily efficiency of a collector is defined as the accumulation of accumulated heat in water to combined solar heat that occurs on a heater all day long. Therefore, the efficiency of a collector is obtained by the following formula as given in Equation (6) [38]:

$$\eta_d = \frac{\Sigma Q_u}{\Sigma IA_p}. \quad (6)$$

3 | Numerical Simulation

Numerical simulations were performed using CFDs to investigate the flow and thermal behavior of the DPSWH under different operating conditions. The CFD model was employed to analyze the velocity field, pressure distribution, and temperature distribution within the water channel, as well as to evaluate the influence of absorber plate geometry and mass flow rate on the overall thermal performance of the system. The numerical methodology consisted of computational domain development, mesh generation, mathematical modeling, and specification of appropriate boundary conditions as described in the following sections.

3.1 | Computational Domain and Mesh Generation

The computational domain represents the physical region in which governing equations of mass, momentum, and energy conservation are solved. To obtain accurate numerical solutions, the DPSWH geometry was discretized using an unstructured tetrahedral mesh. Tetrahedral elements were selected due to their ability to accurately capture the complex geometry of the V-corrugated absorber plate, flow passages, and inlet-outlet regions while providing sufficient flexibility for local mesh refinement. A finer mesh density was applied near the absorber surface and channel walls to resolve thermal and hydrodynamic boundary layers, whereas a relatively coarser mesh was used in regions with lower gradients to reduce computational cost. The final mesh consisted of ~512,436 nodes and 298,754 tetrahedral elements. The mesh quality was evaluated using standard CFD quality metrics, yielding an average skewness of 0.24 and an orthogonal quality of 0.87, indicating good mesh quality and numerical stability. The selected mesh provided an appropriate balance between solution accuracy and computational efficiency for the present simulations. The computational domain and corresponding tetrahedral mesh of the DPSWH model are illustrated in Figure 3a,b.

3.2 | Mathematical Modeling

The flowing fluid field can be described by using the Reynolds number. The Reynolds number can be expressed as in Equation (7).

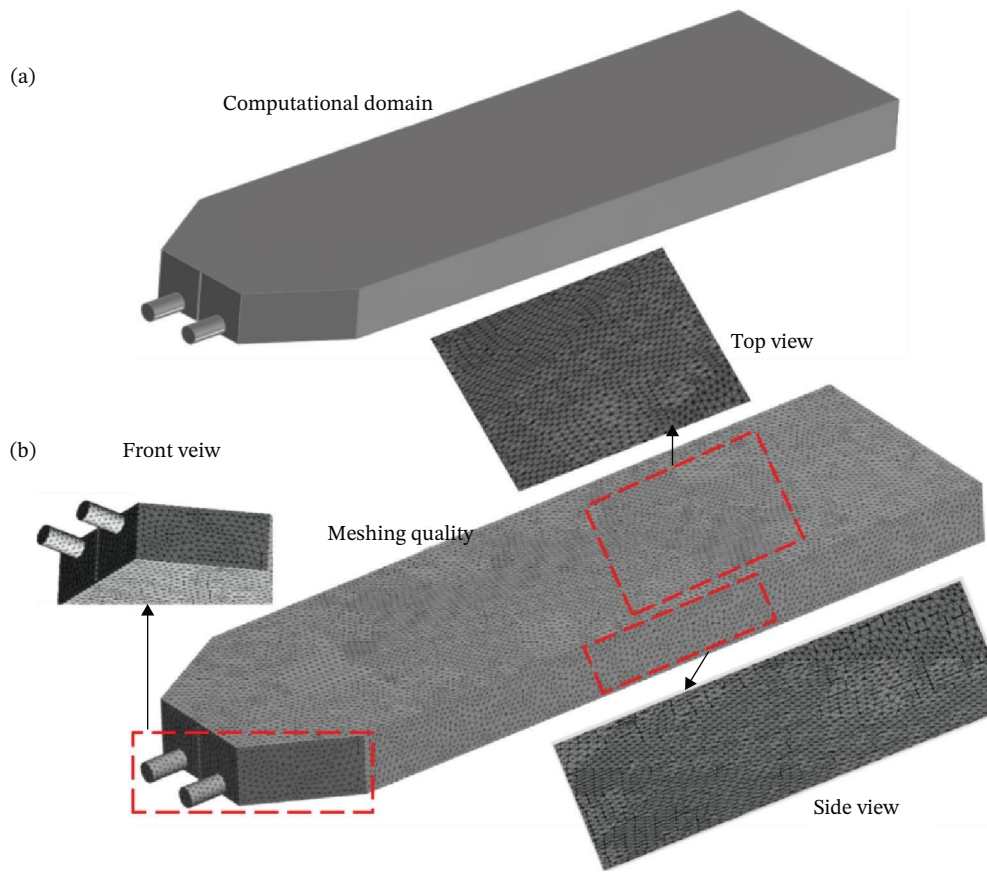


FIGURE 3 | (a) Computational domain of DPSWH and (b) meshing of DPSWH.

$$Re = \frac{\rho U_{\sigma} D}{\mu}, \quad (7)$$

where Re is Reynolds number, ρ is the fluid density, and U_{σ} is the free stream velocity. Likewise, D is the diameter and μ is the dynamic viscosity. The numerical simulations were performed for a Reynolds number $\sim 14.33 \times 10^6$. The formula of conservation of mass is given in Equation (8) [39].

$$\nabla(\vec{u}) = 0. \quad (8)$$

Similarly, the formula for the conservation of momentum is given in Equation (9).

$$\rho \left\langle \frac{\partial(\vec{u})}{\partial t} + \mu \cdot \nabla \vec{\mu} \right\rangle = -\nabla p + \nabla \cdot \tau + S_M, \quad (9)$$

where u is velocity vector, ρ is density, t is time, p is the pressure, τ is stress tensor, and S_M is momentum source. The turbulence kinetic energy is determined as given in Equation (10) [40].

$$\frac{\partial k}{\partial t} + U_j \frac{\partial k}{\partial x_j} = \tau_{ij} \frac{\partial U_i}{\partial x_j} - \beta k \omega + \frac{\partial}{\partial x} \left[(\nu + \sigma_k \nu_r) \frac{\partial k}{\partial x_j} \right]. \quad (10)$$

The formula for the specific dissipation rate is given in Equation (11).

$$\begin{aligned} \frac{\partial \varpi}{\partial t} + U_j \frac{\partial \varpi}{\partial x_j} &= \alpha S^2 - \beta \varpi^2 + \frac{\partial}{\partial x_j} \left[(\nu + \sigma_{\nu} \nu_r) \frac{\partial \varpi}{\partial x_j} \right] \\ &+ 2(1 - F_1) \sigma_{\nu} \frac{1}{\varpi} \frac{\partial k}{\partial x_j} \frac{\partial \varpi}{\partial x_j}, \end{aligned} \quad (11)$$

where k is turbulence kinetic energy, ϖ is specific dissipation rate, and α is closure coefficient. Besides, S is mean rate-of-strain tensor and F_1 is a blending function in simulated system. Similarly, the irradiative transfer formula for energy transfer in form of electromagnetic radiation is given in Equation (12) [41].

$$\begin{aligned} \frac{dI(\vec{r}^{\rightarrow}, \vec{s}^{\rightarrow})}{ds} + (\alpha + \sigma_s) I(\vec{r}^{\rightarrow}, \vec{s}^{\rightarrow}) &= \alpha n^2 \frac{\gamma T^4}{\pi} \\ &+ \frac{\sigma_s}{4\pi} \int_0^{4\pi} I(\vec{r}^{\rightarrow}, \vec{s}^{\rightarrow}) \Phi(\vec{s}^{\rightarrow} \cdot \vec{s}^{\rightarrow}) d\Omega', \end{aligned} \quad (12)$$

where $\vec{r}^{\rightarrow}$ = position vector

$\vec{s}^{\rightarrow}$ = direction vector

$\vec{s}'^{\rightarrow} \rightarrow \vec{s}^{\rightarrow}$ = scattering direction vector

σ = path length

γ = absorption coefficient

n = refractive index

σ_s = scattering coefficient

σ = Stefan–Boltzmann constant ($5.670374419 \times 10^{-8} \text{ W}/(\text{m}^2 \times \text{K}^4)$)

I = radiation intensity which depends on position ($r \rightarrow$ and direction $s \rightarrow$)

T = local temperature

Φ = phase function

Ω' = solid angle.

3.3 | Boundary Conditions and Numerical Scheme

Appropriate boundary conditions are essential for obtaining accurate and stable numerical solutions in CFD simulations. In the present study, water entered DPSWH through the inlet channel and exited through the outlet section. The flow was considered incompressible and turbulent, as the Reynolds number was $\sim 3.45242 \times 10^5$, which is significantly higher than the critical Reynolds number of 2320. Therefore, the SST $k-\omega$ turbulence model was employed to accurately predict flow and heat-transfer characteristics within the collector. The thermophysical properties of water were assumed to be constant, with a density of $998.2 \text{ kg}/\text{m}^3$ and a dynamic viscosity of $0.00103 \text{ kg}/\text{m}\cdot\text{s}$. The inlet fluid temperature was specified as 25°C , while three mass flow rates of 0.003, 0.005, and $0.007 \text{ kg}/\text{s}$ were considered to evaluate the thermal performance of DPSWH under different operating conditions. A pressure–velocity coupled scheme was adopted to solve the governing equations and ensure effective coupling between pressure and velocity fields. The numerical solution was considered converged when the residuals satisfied the prescribed convergence criteria, which was typically achieved within 500 iterations.

3.4 | Model Validation

The accuracy and reliability of the developed CFD model were assessed by comparing the numerical predictions with the experimental measurements obtained from the fabricated DPSWH under identical operating conditions. For validation purposes, the thermal efficiency of the V-corrugated DPSWH integrated with the PCM was selected as the primary performance parameter. As it directly reflects the combined effects of fluid flow, heat transfer, and TES within the system. Table 2 presents the comparison between the numerical and experimental thermal efficiencies at three different mass flow rates. At flow rates of 0.003, 0.005, and $0.007 \text{ kg}/\text{s}$, the CFD model predicted thermal efficiencies of 80%, 73%, and 54%, respectively, while the corresponding experimental values were 78%, 70%, and 52%. The calculated relative errors were 2.56%, 4.29%, and 3.85%, respectively. The

TABLE 2 | Comparison of numerical and experimental thermal efficiencies for model validation.

Mass flow rate (kg/s)	CFD efficiency (%)	Experimental efficiency (%)	Relative error (%)
0.003	80	78	2.56
0.005	73	70	4.29
0.007	54	52	3.85

maximum deviation between the numerical and experimental results was less than 5%, indicating good agreement and confirming the capability of the CFD model to accurately predict the thermal behavior of the DPSWH. The slight inconsistencies between the numerical and experimental results can be attributed to inevitable heat losses to the surroundings and fluctuations in solar radiation during the experiments. Additional factors include uncertainties in thermophysical properties and measurement errors of the experimental instruments. Furthermore, the numerical model was developed under certain assumptions, including uniform material properties and idealized boundary conditions, which may differ slightly from actual operating conditions. Despite these limitations, the close agreement between the numerical predictions and experimental measurements demonstrates that the adopted numerical methodology, turbulence model, and boundary conditions provide a reliable representation of the physical system. Therefore, the validated CFD model can be confidently used to investigate the flow and thermal characteristics of DPSWH configurations under different operating conditions.

4 | Experimental Analysis

4.1 | Materials and Specifications

In this study, a flat plate of mild steel with a black coating is used as a uniform absorber plate. The length of a plate is 1700 mm, as shown in Figure 4a. The new V-shaped objects are used for better thermal contact with the flowing water. The length and angle of object V are 3 mm and 60° , respectively, as shown in Figure 4b. An aluminum tube is used to fill hot storage items. The round fins have been added to increase the rate of heat transfer, as shown in Figure 4c. The physical parameters used in SWHs for the storage capsule are given in Table 3. Thermal storage material is used to store thermal energy during experiments, as shown in Figure 4d. Paraffin wax was selected as the PCM for TES due to its high latent heat capacity, chemical stability, noncorrosive nature, low cost, and reliable thermal cycling performance. The PCM used in this study was commercial-grade paraffin wax without the addition of nanoparticles or other thermal conductivity enhancement additives. Paraffin wax was primarily composed of hydrocarbon compounds with 20–40 carbon atoms, remains solid under ambient conditions, and undergoes a phase transition at $\sim 323 \text{ K}$. This enables it to absorb and store excess thermal energy during periods of high solar radiation and release it when solar intensity decreases, thereby extending hot-water availability beyond peak sunshine hours. The thermophysical properties used in this study were adopted from the literature [10] and are presented in Table 4. The digital thermometer is used as a temperature sensor. The internal temperature display measures temperature within a specified minimum and maximum range and includes a mounting clip for attachment to any flat surface, as shown in Figure 4f. The physical specifications of the temperature sensors are provided in Table 5. A solar meter (MASTECH MS 6612, USA) was used to measure solar radiation in W/m^2 , as shown in Figure 4e. The parametric operation is assessed thoroughly according to the variation of the charge/discharge rate of the hybrid thermoelectric systems [42]. Thermocol, technically known as polystyrene, is a lightweight and nonbiodegradable hydrocarbon polymer derived from styrene monomers. It is

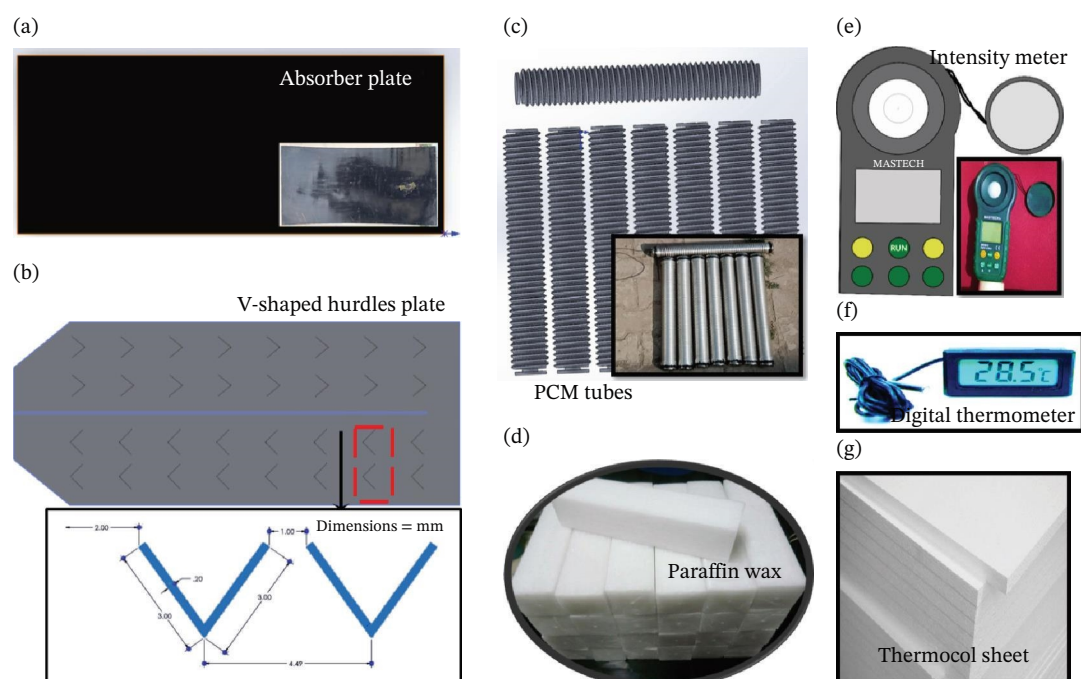


FIGURE 4 | Experimental components: (a) Absorber plate, (b) V-shaped plate, (c) PCM tubes, (d) paraffin wax, (e) sunlight intensity meter, (f) digital thermometer, and (g) thermocol sheets.

TABLE 3 | Physical parameters for storage capsule in SWHS.

Sr. number	Design material/parameter	Specification
1	Pipe, inner diameter	0.040 m
2	Pipe, outer diameter	0.050 m
3	Pipe length	0.500 m
4	Total number of pipes	8
5	Weight paraffin per capsule	0.475 kg

TABLE 4 | Characteristics of paraffin wax [10].

Sr. number	Properties	Values
1	Specific heat capacity	$2.14 \text{ J g}^{-1} \text{ K}^{-1}$
2	Heat of fusion	220 J g^{-1}
3	Melting temperature	323 K

widely used as a thermal insulation material in various solar thermal applications. In this study, thermocol sheets were employed to insulate the sidewalls and bottom of the DPSWH collector to minimize heat losses and improve thermal energy retention within the water channel, as shown in Figure 4g. The insulation was wrapped around the sealed collector chamber to reduce conductive heat transfer and enhance the overall thermal performance. The thermophysical properties of the thermocol used include a density of $15\text{--}30 \text{ kg/m}^3$, thermal conductivity of $0.032\text{--}0.038 \text{ W/m}\cdot\text{K}$, a specific heat capacity of $1200\text{--}1400 \text{ J/kg}\cdot\text{K}$ and an operating temperature range of -50 to 75°C . Its low thermal conductivity and lightweight nature make thermocol a cost-effective and suitable insulation material for solar water-

TABLE 5 | Physical parameters for temperature sensors in SWHS.

Sr. number	Design material/parameter	Specification
1	Temperature range	$-50\text{--}+70^\circ$
2	Usage temperature	$-10\text{--}+50^\circ$
3	Usage humidity	$5\%\text{--}80\%$
4	Accuracy	$\pm 0.1^\circ$
5	Operation range	$4\text{--}30$ volts

heating systems. Table 6 represents the thermophysical properties of the thermocol sheet.

4.2 | Experimental Setup

The thermal performance of a DPSWH incorporating V-shaped longitudinal baffles on both sides of the absorber plate was investigated through numerical and experimental approaches. For the experimental study, the DPSWH was fabricated using a mild-steel structure with overall dimensions of 1700 mm in length and 600 mm in width. A 1 mm -thick aluminum absorber plate coated with black paint was employed to maximize solar energy

TABLE 6 | Thermophysical properties of thermocol sheet.

Sr. number	Properties	Values
1	Density	22 kg/m^3
2	Thermal conductivity	$0.035 \text{ W/m}\cdot\text{K}$
3	Specific heat capacity	$1300 \text{ J/kg}\cdot\text{K}$
4	Porosity	96%

absorption. To reduce heat losses to the surroundings, the bottom and side walls of the collector were insulated with glass wool of 1 ft 0.5-inch thickness. In addition, the top surface of the collector was covered with a 5 mm-thick glass sheet to minimize convective and radiative heat losses. Circular inlet and outlet ports were provided to ensure continuous water circulation through the collector. A 25 L water storage tank supplied water to the inlet section through a connecting pipe equipped with a control valve. The mass flow rate of water was determined using the theoretical approach given in Equation (13).

$$\dot{m} = \rho \cdot Q, \quad (13)$$

where $\dot{m}$ is the mass flow rate, ρ is the fluid density, and Q is the volumetric flow rate. The mass flow rate was determined by an uncertainty of ± 0.0005 kg/s based on multiple experimental trials. A corresponding relative uncertainty of $\sim 2.5\%$ was observed in the mass flow rate measurements. Flow pulsations within the water channel were identified as the primary source of measurement uncertainty and may be further reduced in future studies using a regulated flow meter. To enhance the thermal performance of the DPSWH, a latent TES system was integrated into the collector. Commercial-grade paraffin wax was encapsulated in eight aluminum tubes, each having a diameter of 50 mm and a length of 500 mm, to store excess thermal energy during periods of high solar radiation. A black-coated absorber plate was employed to maximize solar energy absorption and improve heat transfer to working fluid. Furthermore, three calibrated temperature sensors were installed at selected locations to monitor the water temperature at the inlet and outlet sections of the system. Figure 5a,b illustrate the schematic layout and fabricated experimental setup of the DPSWH, respectively.

5 | Results and Discussion

Thermal analysis of three different DPSWH designs (e.g., flat plate, V-shaped, and V-shaped with PCM) was numerically

examined at different mass flow rates to determine the thermal performance of DPSWH. In addition, an experimental study is carried out to validate the numerically calculated results for the best design. For this purpose, the third simulated configurations (DPSWH) were selected to further correlate and ascertain the results. After a series of experiments, mass flow rates of 0.003, 0.005, and 0.007 kg/s are selected for numerical and experimental study. Basically, the primary parameters of the third configurations were recorded and observed, which fed the temperature measurements into the simulation study. Other parameters, which are as important as the primary parameters are derived using mathematical relationships, such as input, output, and energy storage. A series of simulation experiments were performed, and then the third configurations were selected for a comparison study. Experimental data were recorded from 9:00 am to 5:00 pm. The DPSWH was at the local latitude of Multan conditions (30.20°N or South) and facing south to receive the maximum amount of solar radiation. Various types of line graphs were used to present results from experimental data. Graphical representation can be divided into two types. The former describes a single parameter under the experimental study, while the latter indicates multiple parameters under the experimental study. This analysis provides an insight into the various parameters associated with the experimental setups being studied.

5.1 | Performance Analysis of Flat Plate DPSWH

The thermal performance of the flat-plate DPSWH without the PCM was numerically investigated at three water mass flow rates of 0.003, 0.005, and 0.007 kg/s. Figure 6 illustrates the variations in outlet water temperature, temperature difference, solar radiation intensity, and thermal heat energy during the operating period from 9:00 a.m. to 5:00 p.m. As shown in Figure 6c, the available solar radiation increased from ~ 760 W/m² in the morning to a peak value of 1112 W/m² at 1:00 p.m. before gradually decreasing to about 770 W/m² in the evening. This variation directly influenced the thermal response of the collector and the outlet water temperature. Figure 6a shows that the outlet water temperature

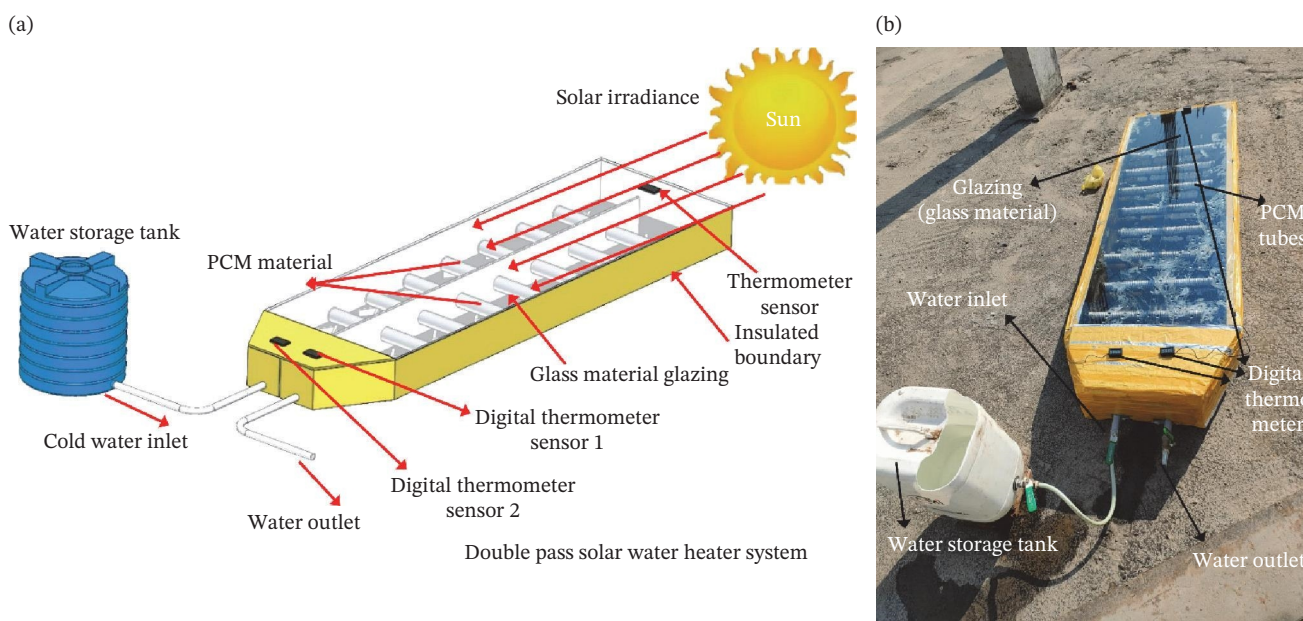


FIGURE 5 | Diagram of DPSWH: (a) Schematic of double-pass solar water heater and (b) experimental setup of double-pass solar water heater.

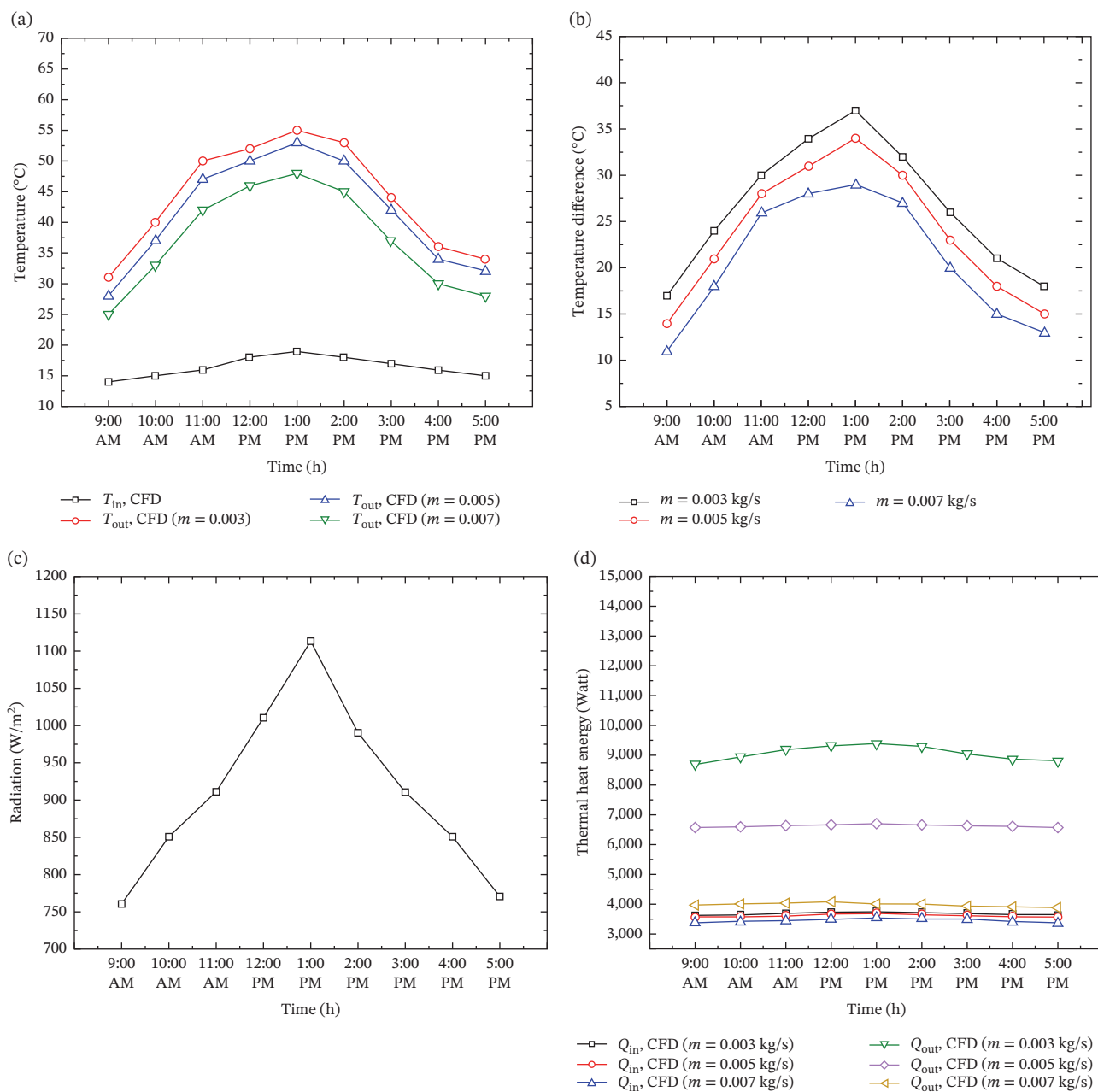


FIGURE 6 | (a) Variation of outlet temperature with time at different mass flow rates of water, (b) variation of temperature difference with time at different mass flow rates of water, (c) variation of radiation intensity with time, and (d) variation of thermal heat energy of input and output with time at different mass flow rates of water.

increased with increasing solar intensity and reached its maximum value at noon for all operating conditions. At a mass flow rate of 0.003 kg/s, the outlet temperature increased from ~31 at 9:00 a.m. to 55°C at 1:00 p.m., corresponding to a temperature rise of 24°C. In comparison, the maximum outlet temperatures at 0.005 and 0.007 kg/s were ~52°C and 48°C, respectively. The lower flow rate produced a higher outlet temperature because the water remained inside the collector for a longer residence time, allowing greater heat absorption from the absorber plate.

The temperature difference between the inlet and outlet water, as shown in Figure 6b, followed a similar trend. The maximum temperature difference of ~36°C was achieved at 0.003 kg/s, whereas peak values of about 33 and 29°C were observed at

0.005 kg/s and 0.007 kg/s, respectively. Compared with the highest flow rate, the lowest flow rate enhanced the peak temperature difference by nearly 24%, demonstrating the strong influence of residence time on heat-transfer effectiveness. Figure 6d presents the variation of thermal energy transferred to water. The useful thermal energy increased with the solar radiation intensity and reached its maximum value around noon. The collector operating at 0.003 kg/s consistently delivered a higher thermal energy gain than the other cases due to the larger temperature rise achieved within the collector. Overall, the results indicate that the flat-plate DPSWH performs more effectively at lower mass flow rates, where the increased fluid residence time enhances heat absorption and improves the thermal utilization of the available solar energy.

Figure 7 presents the variation of thermal efficiency and temperature distribution contours for the flat-plate DPSWH at different water mass flow rates. As shown in Figure 7a, the thermal efficiency increased from morning to noon due to the increase in solar radiation intensity and subsequently decreased during the afternoon. The maximum thermal efficiency was observed at ~1:00 p.m. for all cases. Among the investigated flow rates, 0.003 kg/s achieved the highest thermal efficiency of about 73%, whereas maximum efficiencies of ~56% and 43% were recorded at 0.005 kg/s and 0.007 kg/s, respectively. Compared with the highest flow rate, the lowest flow rate improved the thermal efficiency by nearly 70%, indicating more effective utilization of the absorbed solar energy. This improvement is mainly attributed to the longer residence time of water within the collector, which enhances the heat transfer between the absorber plate and the flowing water. The temperature distribution contours shown in Figure 7b exhibit a similar trend. The highest temperature levels are observed at a mass flow rate of 0.003 kg/s,

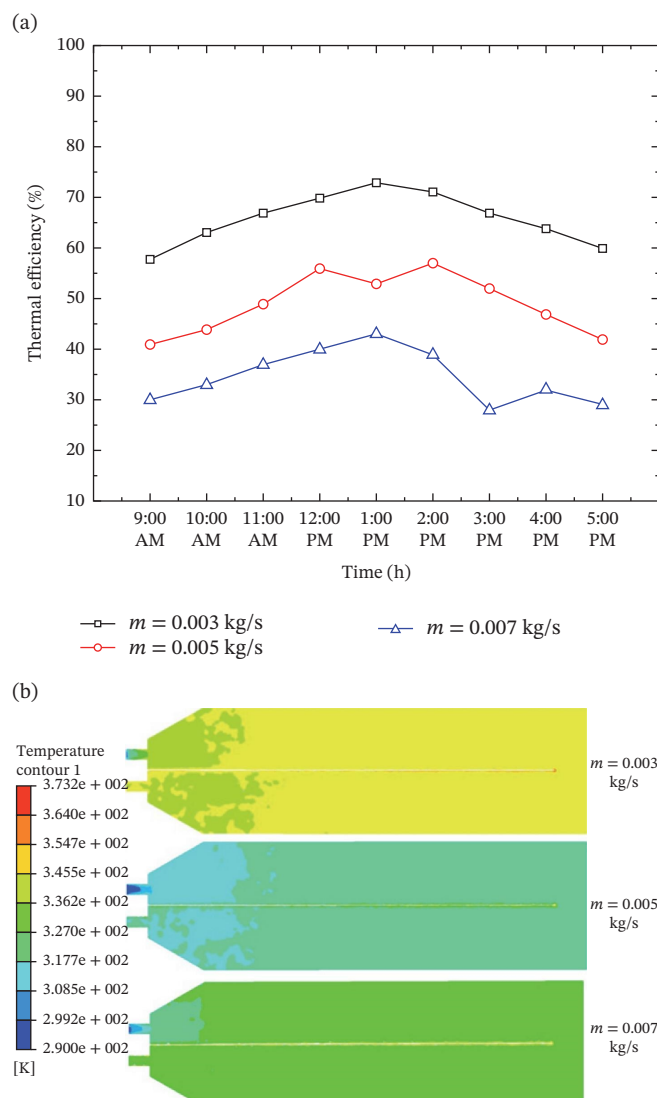


FIGURE 7 | Performance analysis of flat-plate DPSWH: (a) Variation of thermal efficiency with time at different mass flow rates of water and (b) temperature distribution along the open channel DPSWH at different mass flow rates of water.

whereas lower temperatures are obtained at 0.005 and 0.007 kg/s due to the reduced thermal interaction time. The contour plots also indicate a relatively uniform temperature distribution along the collector surface, demonstrating effective heat transfer within the flow channel. Furthermore, the solar fraction was found to vary between 0.40 and 0.55. The highest solar fraction was achieved at 0.003 kg/s, indicating a greater contribution of solar energy to water heating and reduced dependence on auxiliary heating sources. Overall, the results confirm that lower mass flow rates provide superior thermal performance and solar energy utilization in the flat-plate DPSWH.

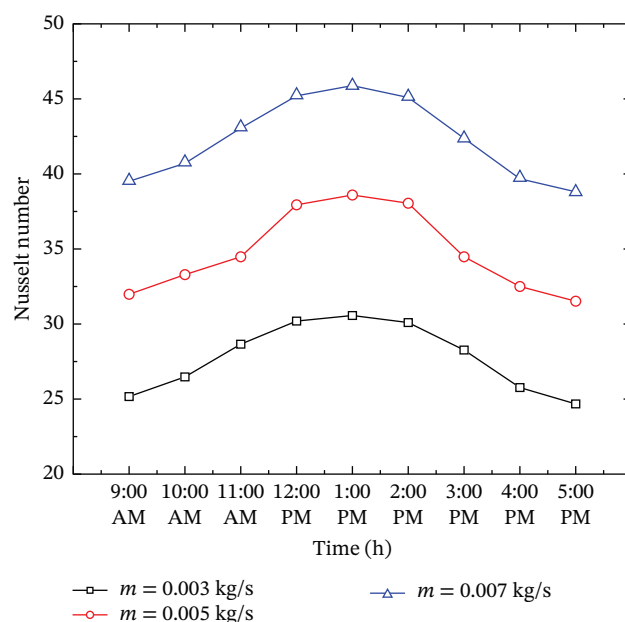


FIGURE 8 | Variation of Nusselt number with time at different mass flow rates for design 1.

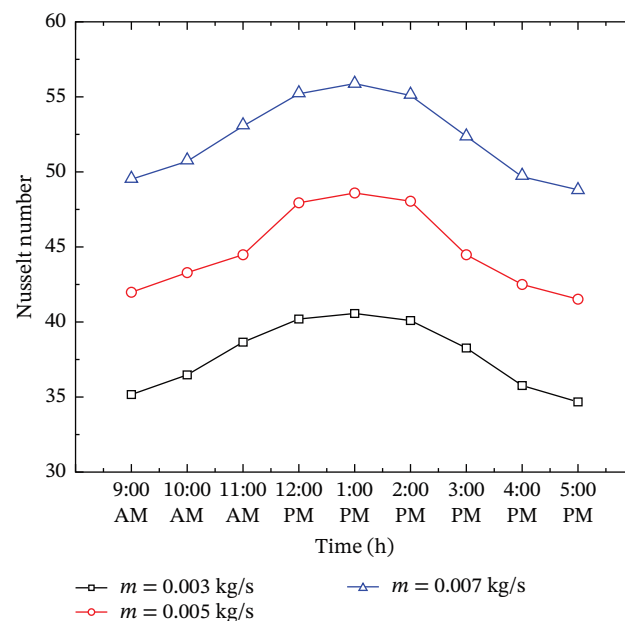


FIGURE 9 | Variation of Nusselt number with time at different mass flow rates.

Furthermore, the Nusselt number is an important parameter for evaluating the convective heat-transfer performance in solar water heaters. Figure 8 illustrates the variation of the Nusselt number with time at three different mass flow rates of 0.003, 0.005, and 0.007 kg/s. For all operating conditions, the Nusselt number increased from morning to noon and subsequently decreased during the afternoon, following the variation in solar radiation intensity and absorber plate temperature. At a mass flow rate of 0.003 kg/s, the Nusselt number increased from ~25.3 at 9:00 a.m. to 30.3 at 1:00 p.m. Similarly, peak values of about 38.6 and 46.1 were obtained at 0.005 kg/s and 0.007 kg/s, respectively. Compared with the lowest flow rate, the maximum Nusselt number increased by ~27% and 52% for mass flow rates

of 0.005 kg/s and 0.007 kg/s, respectively. The higher Nusselt numbers obtained at larger mass flow rates indicate enhanced convective heat transfer due to increased fluid velocity and greater flow turbulence within the water channel. Although higher flow rates improve the convective heat-transfer coefficient, the corresponding outlet water temperature and thermal efficiency decrease due to reduced residence time of water inside the collector. Therefore, a tradeoff exists between convective heat-transfer enhancement and thermal energy absorption. Overall, the results demonstrate that increasing the mass flow rate promotes stronger convective heat transfer, while lower mass flow rates remain more effective for achieving higher water temperatures and improved thermal performance of the DPSWH.

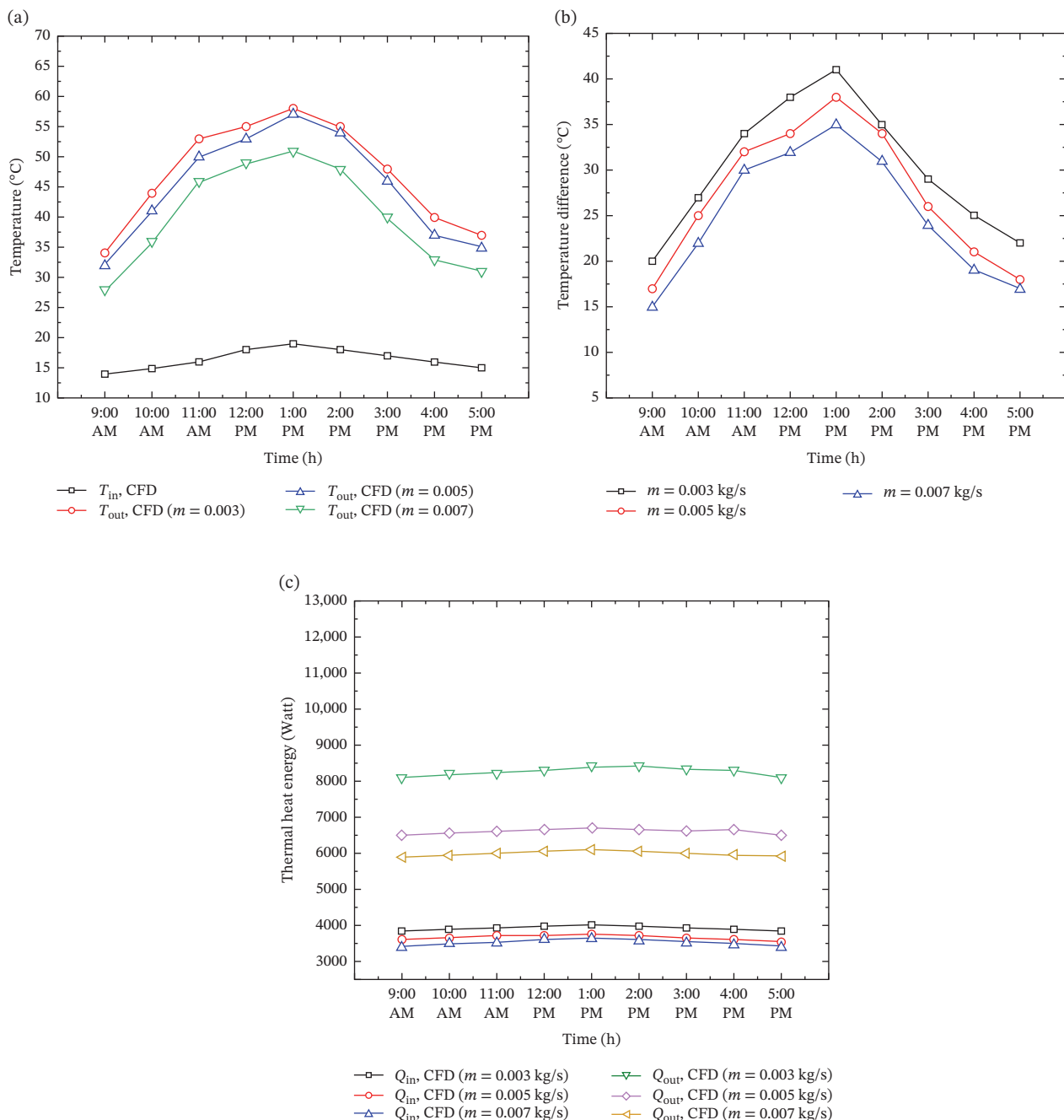


FIGURE 10 | (a) Variation of outlet temperature with time at different mass flow rates of water, (b) variation of temperature difference with time at different mass flow rates of water, and (c) variation of thermal heat energy with time at different mass flow rates of water.

5.2 | Performance Analysis of V-Shaped DPSWH

The thermal performance of the V-shaped DPSWH without the PCM was evaluated to investigate the influence of absorber geometry on heat transfer and flow characteristics. In this configuration, V-shaped obstacles were incorporated into the absorber plate to increase the effective heat-transfer area and promote flow disturbance within the water channel. Three mass flow rates of 0.003, 0.005, and 0.007 kg/s were considered for the numerical analysis. Figure 9 illustrates the variation of the Nusselt number with time for different mass flow rates. For all cases, the Nusselt number increased from morning to noon and subsequently decreased during the afternoon, following the variation in solar radiation intensity and absorber plate temperature. The highest Nusselt number was observed at a mass flow rate of 0.007 kg/s, reaching ~53.5 at 1:00 p.m., whereas peak values of about 46.0 and 38.2 were obtained at 0.005 kg/s and 0.003 kg/s, respectively. Compared with the lowest flow rate, the maximum Nusselt number increased by ~40% at 0.007 kg/s, indicating a significant enhancement in convective heat transfer due to the higher fluid velocity and stronger flow mixing generated by the V-shaped obstacles. Although the higher flow rate improved the convective heat-transfer coefficient, the reduced residence time limited the outlet water temperature rise. The variations in outlet temperature, temperature difference, and thermal heat energy are presented in Figure 10. As shown in Figure 10a, the outlet water temperature increased with solar radiation intensity and reached its maximum value near noon. At a mass flow rate of 0.003 kg/s, the outlet temperature increased from ~34 in the morning to 58°C at noon before decreasing to about 37°C in the evening. Compared with the flat-plate configuration, the V-shaped absorber enhanced the outlet temperature by ~5% due to the increased heat-transfer surface area and improved thermal interaction between the absorber plate and flowing water. Figure 10b shows that the maximum temperature difference of ~39°C was achieved at 0.003 kg/s, while lower peak values were obtained at higher flow rates. Similarly, Figure 10c indicates that the useful thermal energy gain reached its maximum value around noon because of increased solar energy absorption. Overall, the V-shaped DPSWH demonstrated superior thermal performance compared with the flat-plate collector, particularly at the lower mass flow rate of 0.003 kg/s, where the combined effects of enhanced heat transfer and a longer residence time resulted in the highest temperature rise and thermal energy gain.

Furthermore, the thermal efficiency and temperature distribution in the water channel at different mass flow rates are presented in Figure 11. As shown in Figure 11a, the thermal efficiency increased from morning to noon and then gradually decreased during the afternoon due to the variation in solar radiation intensity. The maximum thermal efficiency was achieved at a mass flow rate of 0.003 kg/s, reaching ~78% at 1:00 p.m., whereas peak efficiencies of about 70% and 50% were obtained at 0.005 kg/s and 0.007 kg/s, respectively. Compared with the highest flow rate, the lowest flow rate improved the thermal efficiency by ~56%, indicating more effective utilization of the absorbed solar energy. The higher efficiency at lower flow rates is mainly attributed to the increased residence time of water within the collector, resulting in greater heat absorption and temperature rise. Similarly, the temperature contours shown in Figure 11b indicate that the highest temperature distribution

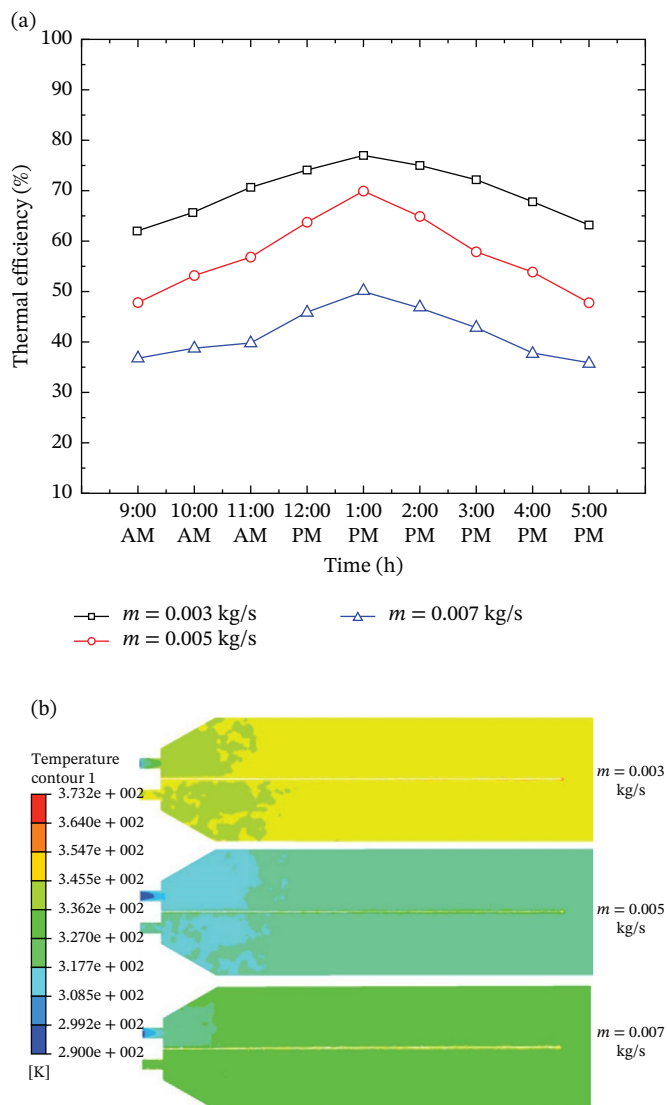


FIGURE 11 | Performance analysis of V-shaped DPSWH without PCM: (a) Variation of thermal efficiency with time at different mass flow rates of water and (b) temperature distribution along the open channel DPSWH at different mass flow rates of water.

occurs at a mass flow rate of 0.003 kg/s, while lower temperatures are observed at higher flow rates. Compared with the flat-plate collector, the V-shaped absorber enhanced thermal efficiency by ~5% due to the increased heat-transfer area and improved flow disturbance generated by the obstacles. In addition, the solar fraction was found to vary between 0.30 and 0.35. The higher solar fraction demonstrates a greater contribution of solar energy to water heating and a reduced dependence on auxiliary heating sources. Overall, the V-shaped DPSWH exhibited improved thermal performance and is therefore a suitable option for residential solar water-heating applications.

5.3 | Performance Analysis of V-Shaped DPSWH With PCM

In this section, the thermal performance of the V-shaped DPSWH integrated with PCM tubes is evaluated using both numerical and experimental approaches. The primary purpose of incorporating PCM tubes was to store excess thermal energy during periods of

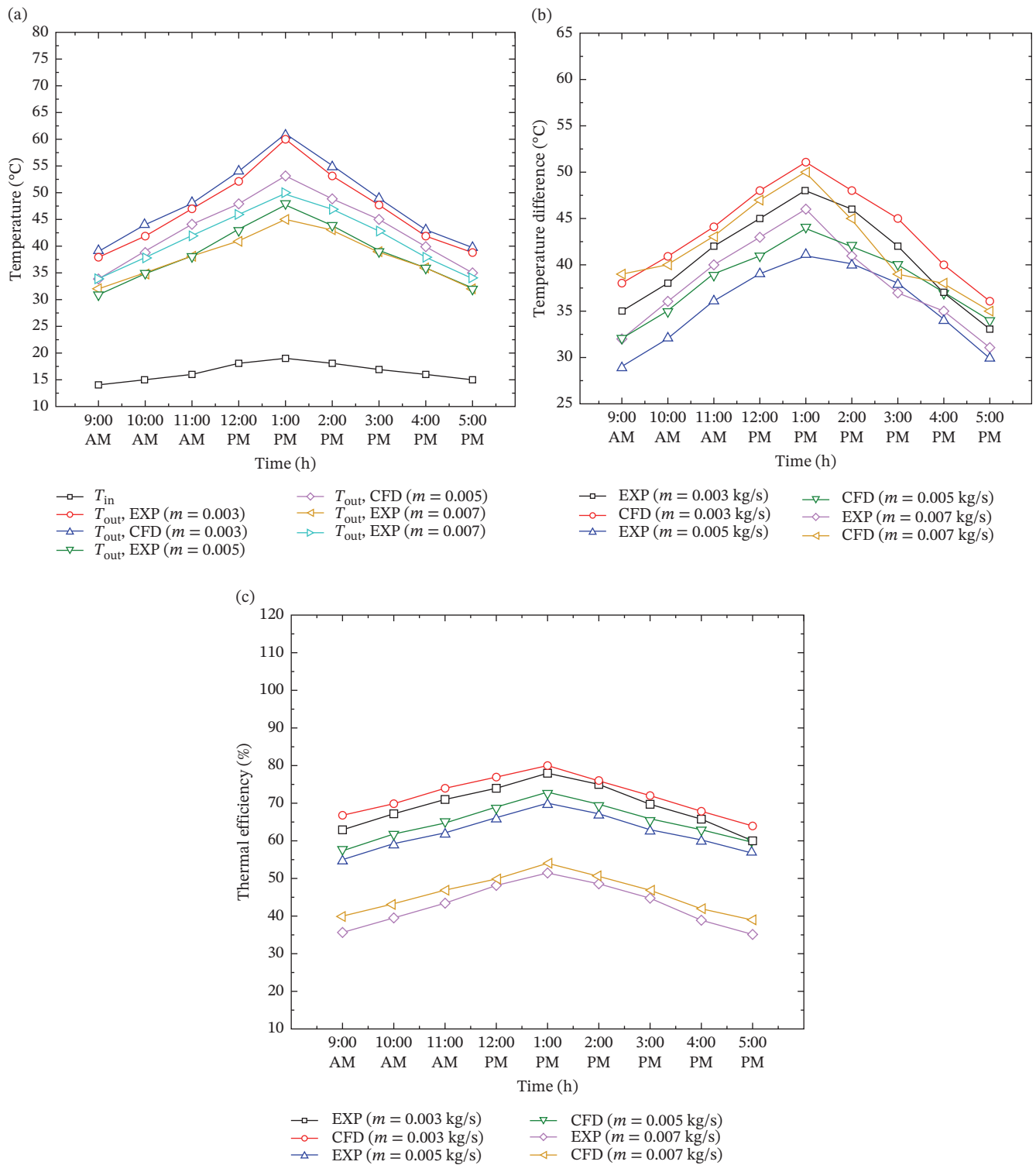


FIGURE 12 | (a) Variation of outlet temperature with time at different mass flow rates of water. (b) Variation of temperature difference with time at different mass flow rates of water. (c) Variation of thermal efficiency of CFD and experimental results over time at different mass flow rates.

high solar radiation and release the stored energy during periods of low or no solar availability. The combined effect of V-shaped baffles and PCM storage enhanced heat transfer, improved thermal retention, and extended the hot-water supply duration beyond the peak solar period. Figures 12 and 13 present the thermal characteristics of the system, including outlet temperature, temperature difference, thermal efficiency, heat energy gain, and temperature distribution, at water mass flow rates of 0.003, 0.005, and 0.007 kg/s.

Figure 12a shows the variation of inlet and outlet temperatures obtained from both CFD and experimental analyses. The highest outlet temperature was achieved at a mass flow rate of 0.003 kg/s, reaching $\sim 60^{\circ}\text{C}$ at noon, whereas peak temperatures of about 55 and 50°C were observed at 0.005 kg/s and 0.007 kg/s, respectively. Similarly, Figure 12b indicates that the maximum temperature difference of $\sim 51^{\circ}\text{C}$ was obtained at 0.003 kg/s, while lower values were recorded at higher flow rates due to the reduced residence time within the collector. The comparison between

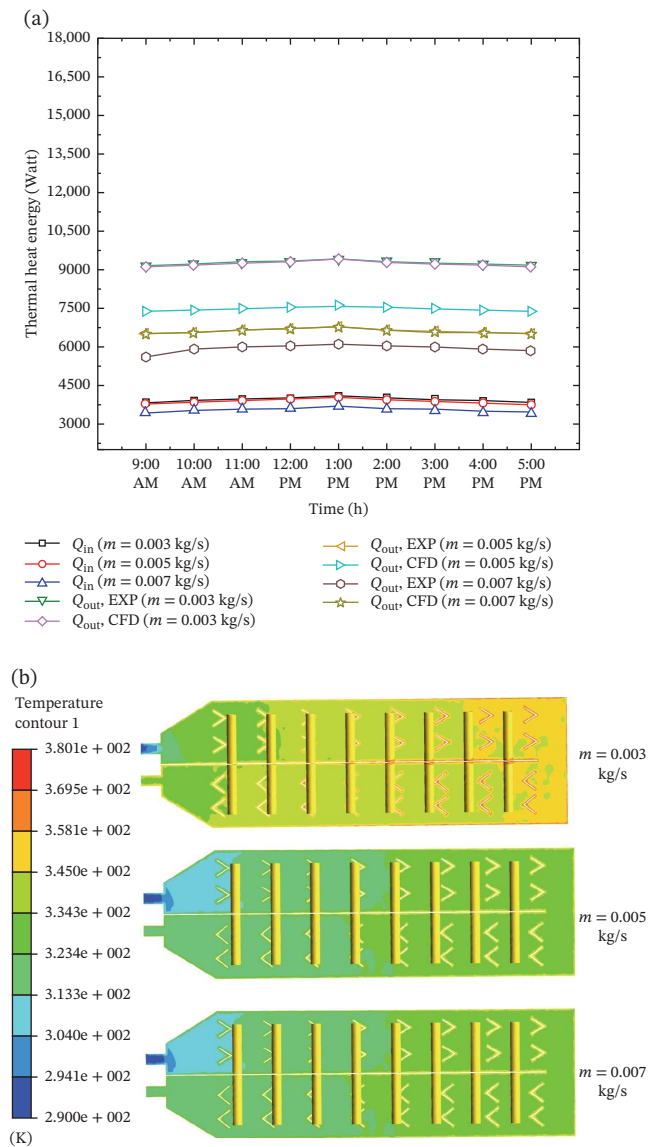


FIGURE 13 | (a) Variation of thermal efficiency with time at different mass flow rates of water and (b) temperature distribution along the open channel DPSWH with PCM at different mass flow rates of water.

CFD and experimental results demonstrates good agreement, with only minor deviations attributed to heat losses, measurement uncertainties, and fluctuations in solar radiation intensity. The variation of the thermal efficiency with time is shown in Figure 12c. The maximum experimental thermal efficiency was ~80% at 0.003 kg/s, whereas efficiencies of about 70% and 54% were obtained at 0.005 kg/s and 0.007 kg/s, respectively. Figure 13a further shows that the useful thermal energy gain was the highest at 0.003 kg/s because the lower flow rate enabled greater heat absorption and more effective utilization of the stored thermal energy. The PCM tubes continued releasing heat after the peak solar period, thereby reducing the cooling rate and maintaining higher outlet temperatures during the late afternoon. The temperature contours shown in Figure 13b confirm that the highest temperature distribution occurred at 0.003 kg/s, while lower temperature levels were observed at higher flow rates. Furthermore, the solar fraction varied between 0.01 and 0.20, indicating the contribution of stored solar energy

to the overall heating demand. Overall, the V-shaped DPSWH integrated with the PCM exhibited the best thermal performance and demonstrated the effectiveness of latent TES for extending heat delivery and improving system efficiency.

Figure 14 shows the variation of the Nusselt number over time at mass flow rates of 0.003, 0.005, and 0.007 kg/s, based on both CFD and experimental results. For all conditions, the Nusselt number increased from morning to noon and decreased in the afternoon, following changes in solar radiation and absorber plate temperature. The highest Nusselt number occurred at 0.007 kg/s, reaching ~61 (CFD) and 59 (experimental) at 1:00 p.m., while peak values of 53 and 45 were observed for 0.005 kg/s and 0.003 kg/s, respectively. Compared to the lowest flow rate, the maximum Nusselt number increased by ~36% at 0.007 kg/s, indicating enhanced convective heat transfer. The higher Nusselt numbers at larger flow rates result from increased fluid velocity and stronger mixing induced by the V-shaped baffles. The close agreement between CFD and experimental results validates the numerical model. While higher flow rates improve the convective heat-transfer coefficient, the reduced residence time limits the outlet temperature rise. Thus, higher flow rates enhance convective performance, whereas lower flow rates favor greater thermal efficiency and energy gain.

Additionally, a comprehensive comparison study was conducted among all three designs to evaluate the variations in thermal performance. The purpose of this analysis was to determine the influence of different design configurations on the efficiency and effectiveness of the system. Table 7 presents all computed results obtained from the simulation and experimental work, allowing for a detailed examination of performance differences. The findings from this comparative study indicate noticeable variations in the thermal efficiency of DPSWH across three designs. Among the evaluated configurations, design 3 demonstrated the highest

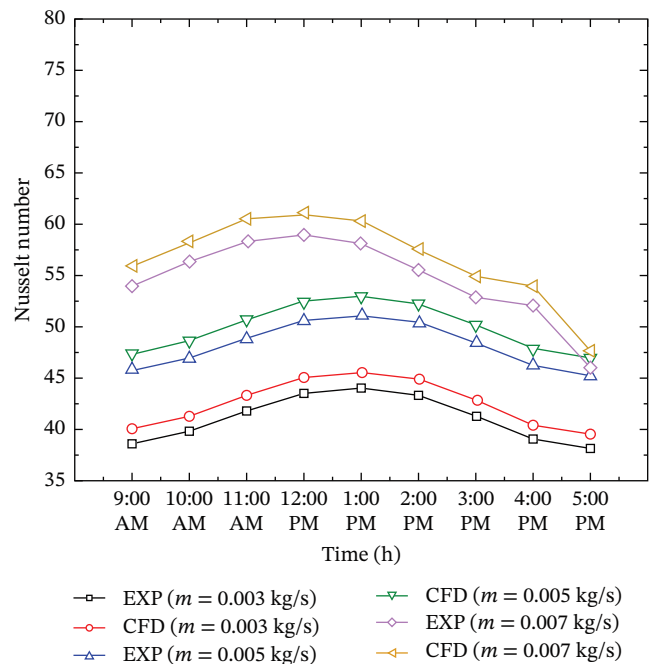


FIGURE 14 | Variation of Nusselt number with time at different mass flow rates for V-shaped DPSWH with PCM.

TABLE 7 | Comparative study of thermal performance between all designs.

Design	Thermal efficiency (%) (0.003 kg/s at 1 pm)	Thermal efficiency (%) (0.005 kg/s at 1 pm)	Thermal efficiency (%) (0.007 kg/s at 1 pm)
1	73	53	43
2	77	70	50
3 (CFD)	80	73	54
3 (EXP)	78	70	52

efficiency, reaching an impressive 80% thermal efficiency at a mass flow rate of 0.003 kg/s. This superior performance can be attributed to its improved heat absorption capacity and reduced thermal losses. The integration of a thermal storage medium utilized heat more efficiently, which was effectively retained. Design 2 followed closely behind, achieving a thermal efficiency of 77%, and benefited from moderate heat capture and controlled energy losses. Conversely, design 1 exhibited the lowest thermal efficiency at 73%, likely due to its higher heat losses and relatively lower heat retention capability. The results highlight that modifications in the collector design and incorporation of a thermal storage medium play a crucial role in improving thermal performance.

6 | Conclusions

Performance analysis of the DPSWH integrated with a PCM was carried out using numerical simulation and experimental methods. Experiments were conducted from 9:00 a.m. to 5:00 p.m. at mass flow rates of 0.003, 0.005, and 0.007 kg/s under the climatic conditions of Multan, Pakistan (30.1864°N, 71.4886°E). Based on the obtained results, the following conclusions are drawn:

- The thermal performance of three DPSWH configurations was analyzed experimentally and numerically at different mass flow rates. Simulations for designs 1 and 2 were conducted from 9:00 a.m. to 5:00 p.m., whereas design 3 was experimentally evaluated until 8:00 p.m. due to additional TES by the PCM.
- Designs 1 and 2 did not include a thermal storage medium. The performance decreased significantly after 5:30 p.m. because of the reduced solar radiation intensity.
- In design 3, paraffin wax PCM was filled in circular tubes with aluminum fins to enhance thermal storage. The V-shaped absorber plate accelerated heat absorption and reduced the PCM melting time to ~25 min.
- Design 3 exhibited significantly higher temperature and thermal efficiency than designs 1 and 2. The improved performance was attributed to the combined effect of V-shaped barriers and PCM tubes, which enhanced heat transfer and increased water–surface contact time.
- The use of the PCM extended the heating duration up to 8:00 p.m., whereas the other designs provided heating only up to 5:00 p.m.

- At a mass flow rate of 0.003 kg/s, the thermal efficiencies of designs 1, 2, and 3 were 73%, 77%, and 80%, respectively.
- The results demonstrate that integrating the PCM and V-shaped barriers improves the thermal performance and energy storage capability of the DPSWH system. Future work should focus on detailed economic analysis to evaluate the commercial feasibility of the proposed system. The findings of this study can support researchers and engineers in the design and development of efficient solar water-heating systems.

Nomenclature

DPSWH:	Double-pass solar water heater
SWH:	Solar water heater
TSM:	Thermal storage material
GDP:	Gross domestic product
LHS:	Latent heat storage
PCM:	Phase change material
Q_u :	Thermal heat energy (W)
M :	Mass flow rate (kg/s)
C_p :	Specific heat of the water (J/kg, °C)
P :	Density of water (kg/m ³)
h :	Heat transfer coefficient
T_a :	Average temperature of water (°C)
Re :	Reynolds number
μ :	Viscosity of water (m ² /s)
T_p :	Average temperature of absorber plate (°C)
ΔT :	Temperature difference (T _{out} –T _{in}) (°C)
I :	Solar radiation (W/m ²).

Author Contributions

Lan Zhang: writing – original draft, methodology, investigation. **Muhammad Waqas:** formal analysis, conceptualization, writing – review and editing. **Lun Zhao:** resources, funding acquisition, project administration, supervision. **Zeshan Abbas:** resources, writing – original draft, supervision. **Jiajin Zhang:** validation, writing – review and editing. **Muhammad Imran:** validation, software, investigation. **Hafiz Muhammad Shahzad:** visualization, writing – review and editing.

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Ethics Statement

The authors have nothing to report.

Consent

The authors have nothing to report.

Conflicts of Interest

The authors declare no conflicts of interest.

Data Availability Statement

The data that support the findings of this study are available upon request from the corresponding author. The data are not publicly available due to privacy or ethical restrictions.

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