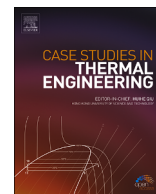




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Heat transfer experimental and numerical study of a three-sided serpentine with the operating fluid directly contacting the PV cell back

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ABSTRACT

The cooling methodologies of photovoltaic/thermal equipment are crucial not only to maintain optimal operating temperatures but also to improve the performance of the photovoltaic systems and prolong their lifespan. Traditional heat exchangers often require physical contact with the material to be cooled, posing challenges for specific projects. Therefore, this study introduces an innovative heat exchanger made of aluminum plate, allowing direct contact of the cooling liquid with the surface to be cooled. The thermal performance of the serpentine, coupled to a steel plate simulating a photovoltaic-thermal panel, was evaluated experimentally. CFD numerical simulations were conducted to analyze the thermal performance of the heat exchanger, providing valuable temperature profiles for single-phase flows. The outcomes showed that the simulated and experimental data agreed well. Particularly, when considering the outlet fluid temperature the mean absolute error between the simulated and experimental results was around 0.5 °C, with a relative error of approximately 1.8 %. To evaluate the influence of the type of material that forms the serpentine, heat exchangers with two different polydimethylsiloxane (PDMS) serpentine were numerically investigated. The PDMS serpentine provided a more heterogeneous steel plate temperature profile compared to the aluminum one; however, such an issue can be corrected with geometry modifications, such as a greater width and cross-sectional area. For all flow rates, the steel plate temperature using aluminum serpentine presented a lower average temperature than that with PDMS serpentine (an average of 6.3 % lower). The wide PDMS serpentine exhibited a better cooling performance than the narrow PDMS serpentine (an average of 92 %) since the heat transfer surface area was enhanced in the former case.

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Nomenclature

A_{FS}	fluid-surface contact area, mm ²
A_{ht1}	aluminum and PDMS serpentine heat transfer surface area, mm ²
A_{ht2}	wide PDMS serpentine heat transfer surface area, mm ²
C_p	specific heat capacity, J/kg.K
D_h	hydraulic diameter, mm
h	heat transfer coefficient, W/m ² .K
H	internal channel height, mm
I	irradiance, W/m ²
k	thermal conductivity, W/m.K
L_{ch}	total channel length, mm
P	total pressure drop, Pa
PP	pumping power, W
Q_v	volumetric flow rate, mL/min
Q_w	input heat power, W
T	temperature, K
T_{in}	inlet temperature of the fluid, K
T_{out}	outlet temperature of the fluid, K
T_w	wall temperature, K
u_{in}	inlet velocity of the fluid, m/s
W	Internal channel width, mm

Greek symbols

ε	emissivity, [–]
μ	kinematic viscosity, kg/m.s
ρ	mass density, kg/m ³

Subscripts

avg	average
f	fluid
in	inlet
out	outlet
max	maximum
min	minimum
w	wall

Abbreviations

PV	Photovoltaic
PV/T	Photovoltaic/Thermal
PCM	Phase Change Material

1. Introduction

In 2021, Portugal's annual electricity consumption was 4630 kWh/inhabitant (INE, DGEG/MAAC, PORDATA, 2023). By comparing this value with the 48 % of solar energy absorbed by the surface, there would be enough energy in 1 h to serve a population of approximately 18 billion people. Such theoretical comparison demonstrates the dimension of this natural energy resource and that scientific or technological research focused on this segment is closely linked to the future of sustainable human development, seeking mainly the reduction of pollutant emissions into the atmosphere.

The solar radiation can be converted into electrical energy by different means including solar chimneys in buildings and photovoltaic (PV) technology using PV cells. Specifically, the latter involves a combination of semiconductor materials, where the most common are monocrystalline (m-Si), polycrystalline (p-Si) and amorphous (a-Si) silicon. The energy assessment and the economics of PV systems depend on the photovoltaic conversion process, which, in turn, depends on the solar modules' working temperature.

Excess solar energy increases the cell's temperature, reducing the system's lifetime and efficiency. Furthermore, anisotropic temperature distributions over the modules can promote damage to the cells [1]. These limitations shorten the lifetime of the photovoltaic panels, increasing the maintenance costs and, in some cases, may lead to the replacement of the modules.

Depending on the type and quality of the materials from which the photovoltaic panels are manufactured, the efficiency is decreased by approximately 0.45 % for each 1 °C increase in the module operating temperature [2]. According to Ref. [3], ensuring that the operating temperature of the photovoltaic module operates within the technical limits established by the manufacturers is essential to obtain a shorter payback period and return on investment of a solar power plant.

In order to overcome such issues, the use of passive and active cooling methods have been studied over the years, such as heat pipes [4], microchannels [5], spraying water [6], liquid immersion [7], phase-change materials [8], serpentine pipes [9] and others [10]. However, some problems are observed in the use of these devices. Heat pipes, for instance, have their performance significantly affected by external environmental conditions [11]. Microchannels have disadvantages such as overheating large walls, critical heat flow at low evaporation points, substantial pressure drops, and poor temperature uniformity [12]. Spray cooling faces challenges in maintaining constant temperatures and requires additional equipment like pumps. It is also influenced by external parameters such as wind speed and equipment characteristics like nozzle shape [13].

Additionally, in liquid immersion cooling, prolonged submersion in deionized water can lead to a decrease in the electrical production of the solar cell module [14]. Although not as efficient in heat exchange as heat pipes, serpentine offer ease of reproduction parameter control and are a widely used type of heat exchanger in industry [15]. However, they require improvements in terms of the contact area.

Cooling systems that use liquids are more common options, where the serpentine tubes are coupled to the rear surface of photovoltaic modules to lower the PVs' working temperature, increasing their power and efficiency [16]. However, conventional serpentine tubes have a low tube-PV panel contact area, making it difficult to achieve good thermal conduction and temperature distribution in the photovoltaic-thermal (PVT) system. Diwania et al. [17] welded the copper serpentine on the rear solar panel surface to conduct experiments with nanofluids as thermal fluids. Menon et al. [18] utilized hinges to maintain the contact between the serpentine and an aluminum plate in a hybrid PV-T system. Krstic et al. [19] simulated the cooling process for a photovoltaic panel using an aluminum heat sink with various geometries. The average difference between the simulated and experimentally obtained values was as low as 1 °C. Moreover, Chen et al. [20] developed a novel type of heat sink with serpentine reentrant microchannels for efficient cooling of PV modules. The researchers feature serpentine flow passages with Ω -shaped cross-sectional configurations, which contributing to promote fluid mixing and disrupt the normal development of thermal boundary layers. By the comparison of a fin heat sink, both numerical and outdoor experimental works were carried to explore the enhancement feasibility of thermal and electrical performance of PV cells. The obtained results confirmed that the serpentine reentrant microchannels reduced the average temperature of the cell and increased the temperature uniformity of the PV cell considerably, given that it exhibited cell temperatures between 25 and 31 °C, much smaller than that between 45 and 63 °C of the compared fin heat sink. The temperature differences of PV cell modules were reduced to be less than 4.4 °C. Besides, the output power increased by as high as 115 %, and the electrical efficiency increased to 15–20 % for the PV module with serpentine reentrant microchannels. Besides, the PV cell module with serpentine reentrant microchannels was also found to provide lower average cell temperatures and better electrical performance than a module with parallel reentrant microchannels. Also, Si et al. [21] investigated numerically the PV-T collector performance considering the effect of using needle fins in a serpentine channel with nanofluids. The authors made the comparison between the hydrothermal characteristics of the PV-T system with finned and plain serpentine channels. The research team found that the use of needle fins improved the thermal efficiency, electrical efficiency, and overall efficiency by between 8.6 % and 10.2 %, 0.1–0.24 %, 5.1–5.7 %, respectively, against the PV-T system employing a plain serpentine channel. Moreover, the thermal exergy efficiency, electrical exergy efficiency, and overall exergy efficiency were enhanced by from 8.6 to 1.2 %, 0.1–0.2 %, and 2.6–2.79 %, respectively, in comparison with those obtained with the plain serpentine channel PV-T system. Park et al. [22] performed a numerical investigation of solar energy heat transfer performances in a water-based serpentine PV-T collector. The researchers found that with increasing pitch size or distance between the tubes from 40 to 120 mm, the thermal efficiencies were reduced from nearly 80.9 % to around 27.6 %, whilst the water outlet temperatures and pressure drop were also decreased from approximately 80.9 to 27.6 % and from around 12,000–4217 Pa, respectively, due to decreasing heat transfer area and surface resistance. Considering the boundary conditions, the optimal pitch size (distance between tubes) was found to be 80 mm while it was shown that the optimal position for the water inlet was on the left side of the unit. In this context, the current work compares, numerically and experimentally, the thermal performance of an aluminum serpentine that allows direct contact of the cooling liquid with the surface to be cooled. In order to evaluate the influence of the serpentine material, a numerical study using ANSYS FLUENT 2022 R2 was carried out, considering a serpentine made from polydimethylsiloxane (PDMS). To the best of our knowledge, we have no information on previous work that made simulations considering PDMS serpentine for solar panel cooling. This is the first instance where a thermal assessment of the use of this material has been conducted. Hence, the novelty of the current study not only comes from the materials used but also from the use of different serpentine with walls on only three sides, with the fourth one being the steel plate surface to be cooled. This enables a cooling effect based on the direct contact of the operating fluid with the solid surface, which can be enhanced with different geometry arrangements like a greater wide of the serpentine.

According to Ref. [23], PDMS exhibits strong adherence, enabling its direct assembly within the cooling area. This often eliminates the need for additional elements such as screws, nuts, or adhesives. The material demonstrates excellent wear resistance even when exposed to elevated temperatures and experiences no degradation in typical environmental conditions. The PDMS exhibits high transparency in the visible spectra region, enabling the visualization of the fluid flow during experiments, which is ideal for experimentally studying the behavior of nanoparticles in nanofluids. The mid-infrared regime's transparency and considerable thermal emissivity make the PDMS suitable for outdoor energy conversion applications.

The research gap considering the published works on aluminum and PDMS serpentine-based heat exchangers consists in the use of a heat exchanger based on a rectangular cross-section serpentine with only three sides, making it possible for the operating cooling fluid to directly contact with the surface to be cooled, enhancing the heat transfer capability of the system. Also, it is also innovative to compare in the same work different configurations and materials of the serpentine based on experimental and numerical results.

The main objective of the current work was the development of an innovative serpentine aluminum heat exchanger, enabling the operating fluid to directly contact the heat transfer surface. For this purpose, it was evaluated experimentally the thermal perfor-

mance of the heat exchanger attached to a steel plate, which mimicked the back of a photovoltaic-thermal panel. In conjunction there were conducted CFD numerical simulations were conducted to analyze the thermal performance of the heat exchanger by comparison with one made of PDMS. Section 2 of the current work presents the materials and methodologies to fabricate the aluminum serpentine and the configuration of the system to evaluate the performance of the developed heat exchanger. Section 3 describes the adopted numerical simulation model for thermal performance determination. Section 4 summarizes the main obtained results and corresponding discussion. Finally, Section 5 points out the main concluding remarks retrieved from this study and recommendations for further research studies.

2. Materials and methods

Fig. 1 schematically represents the produced aluminum serpentine. It has a square shape with a total channel length (L) of 1106.00 mm, an internal channel width (W) of 5.00 mm, an internal height (H) of 5.00 mm and a hydraulic diameter (D_h) of 5.00 mm. It has walls on three sides, and the fourth side is the surface to be cooled.

This geometry enables the working fluid to be in direct contact with the surface to be cooled in a maximized area, improving the heat transfer.

2.1. Procedure to construct the aluminum serpentine

A solid plate was used to obtain the aluminum serpentine (Fig. 2), and the flow channel was mechanically made through the milling process. The CNC Milling Machine DECKEL MAHO DMC 63V was used for this process. A thin layer of resin was used to assemble the two solid materials, i.e., the aluminum serpentine and the steel plate (which replicates the back side of a photovoltaic-thermal panel). In order to prevent, after joining both parts, resin leakage into the flow channel, it was necessary to fill the channel with a mixture of liquid caramelized sugar. The liquid caramelized sugar mixture was prepared by the wet method. A mixture of 400 g of white granulated sugar and 250 ml of water was stirred in a saucepan until the sugar was evenly moistened. After that, the

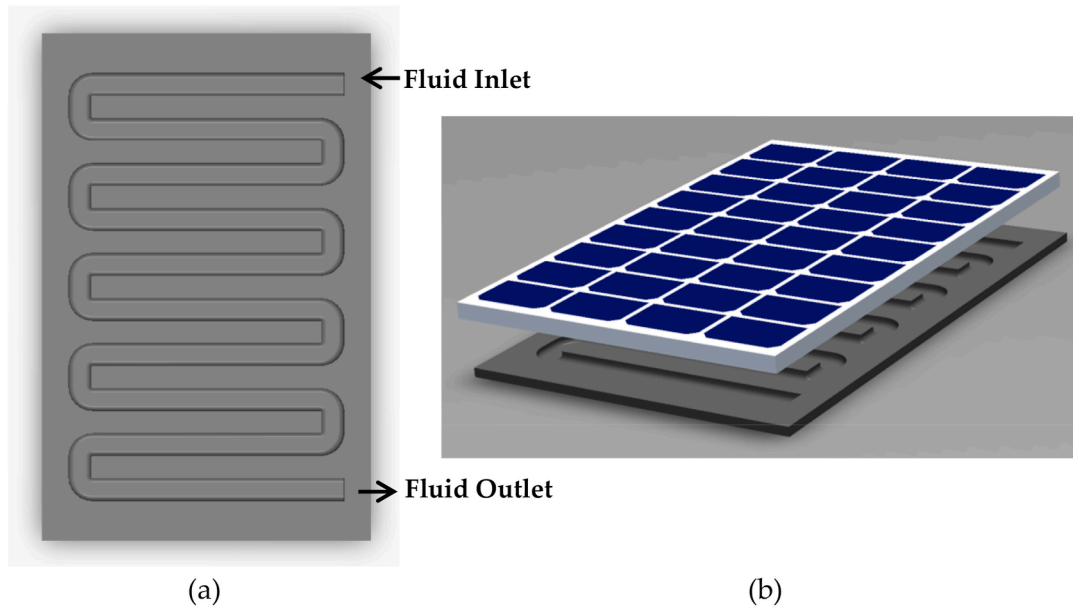


Fig. 1. (a) Aluminum serpentine schematic drawing; (b) Example of the use for PVcooling, showing that the proposed serpentine allows the direct contact of the working fluid with the surface to be cooled.

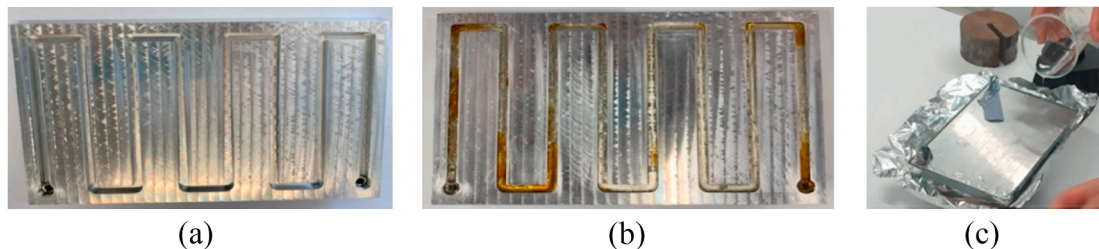


Fig. 2. Procedure of aluminum serpentine: (a) milled aluminum serpentine, (b) caramelized sugar mixture to fill the channel, (c) aluminum serpentine being attached to the steel plate using a thin liquid layer of resin.

mixture in the saucepan was placed over a medium-to-high heat and constantly stirred until the sugar began to dissolve. When the sugar had completely dissolved and the mixture began to boil, the stirring was stopped to prevent the crystallization of the mixture. Then, the heat was reduced to low, and the mixture simmered for up to around 20 min. The stirring of the mixture was avoided since the water evaporated and the sugar began to caramelize, and the stirring effect will incorporate air into the mixture and reduce the temperature, preventing the sugar from browning adequately. When the entire contents reached a uniform brown color and was slightly thickened, the caramelizing process was complete. After that, the mixture was immediately removed from the heat to avoid its potentially burning of the mixture. In this stage it was added 50 ml of water to adjust the viscosity of the mixture. Finally, the serpentine was filled with the caramelized sugar. After the solidification of the caramelized sugar mixture, the parts were joined. On average, it took 48 h to cure completely at room temperature. In order to remove the solidified caramelized sugar from the channel, the assembly was immersed in water. The water was heated up to a temperature high enough to melt the sugar, around 40 °C, and its complete removal was carried out by pumping fluid through the channel.

The cost-effectiveness of the developed heat exchanger or cooling system is practically guaranteed by the comparatively affordable price of aluminum. Besides this fact, the required amount of aluminum is somewhat reduced as the serpentine has only three sides. The relatively high cost of PDMS is compensated by the small quantity needed to glue the serpentine to the steel plate. Hence, the cost-effectiveness is a reality when the use of only relatively affordable materials of aluminum, PDMS, and caramelize sugar solution are included in the manufacturing procedure of the serpentine. Also, the scalability inherent to large-scale production can be achieved with the unavoidable adaptations and refinements of the aluminum serpentine machining and attachment of the serpentine in the back of the photovoltaic panels. A rigorous economic analysis of the manufacturing cost and industrial scale production and required adjustments is out of the scope of the current work, given that it will make it too complex and long, and outside of the main goals of this work.

2.2. Experimental configuration for assessing the efficiency of heat exchange

The experimental system depicted in Fig. 3 consists of a pump (1) connected by hoses to pump the liquid (2); the test section, provided by the aluminum serpentine and steel plate (3); a heat source (4); temperature sensors (5); a system for acquiring data (6); a laptop (7) and containers for maintaining and regulating the temperature of the inlet fluid (8 and 9).

Prior to initiating the experimental tests, the steel plate containing the aluminum heat exchanger was positioned at a predetermined distance of 250 mm from the artificial light source (heat source) perpendicular to the plate. The heat source provides 400 W. Six T-type temperature sensors were installed at the back side of the steel plate and applied to measure wall temperature as a function of time and fluid flow, as shown in Fig. 4. The liquid inlet and outlet temperatures were measured using two additional sensors. A system for acquiring data was used to acquire all temperature measurements.

Then, the serpentine channel was filled with Deionized Water (DI water) (used for all tests) through the peristaltic pump from the reservoir to control the system's initial temperature. Subsequently, the heat source was turned on. The peristaltic pump was activated after the average temperature, provided by the six sensors at the back side of the steel plate, reached 45 ± 0.3 °C.

Volumetric flow rates between 40 and 200 ± 0.035 ml/min were utilized, and the inlet temperature of the liquid was set at 22 ± 0.3 °C for all tests. The test interval depended on the selected flow rates, so intervals between 15 and 25 min were used to reach

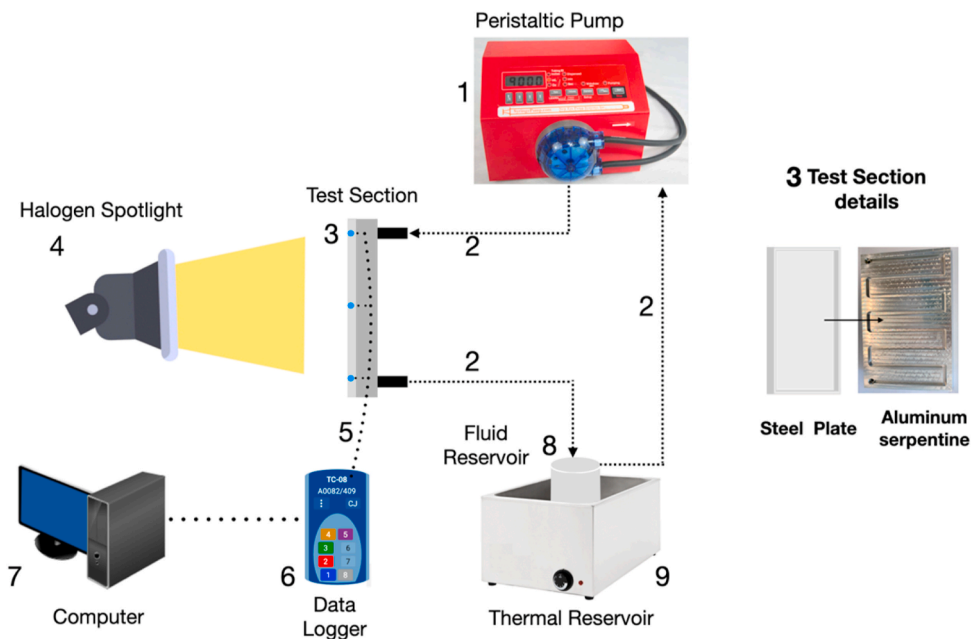


Fig. 3. Diagrammatic representation of the experimental setup for assessing heat exchange efficiency.

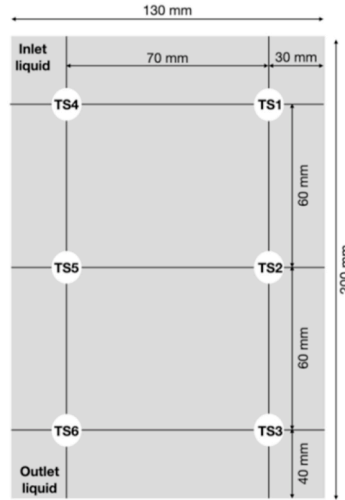


Fig. 4. Six thermal sensors located on the back side of the steel plate.

the steady state. The tests ended when the wall temperature provided by the average of the six thermocouples and inlet and outlet liquid temperatures stabilized (temperature variations remained below the uncertainties). Fig. 5 shows the temperature variation measured during an experimental test with a volumetric flow rate of 200 ml/min. In this case, after a period of 15 min, it was observed that the system reached equilibrium, characteristic of the steady state. Hence, the average of the last 200 experimental data points was used for the analysis of the results.

Table 1 shows the uncertainty values for the current work; the method described by previous work [24] was used for the sensor's uncertainty. Regarding the device, the manufacturers' uncertainties were taken into account.

3. Numerical procedure

For the numerical simulation, the ANSYS Fluent software was utilized, and the Shear-Stress Transport $k-\omega$ turbulence model was chosen to compute the flow velocity profiles in the serpentine and air surrounding the model. The $k-\omega$ performs superior in boundary layer flows delimited by walls; thus, it was chosen to model near-wall interactions more accurately since it provides more accurate predictions of turbulence characteristics in these regions. The study was conducted with the same inlet flow rate and temperature considered in the experimental tests. Moreover, the experiments followed the following protocol: the plates were heated to

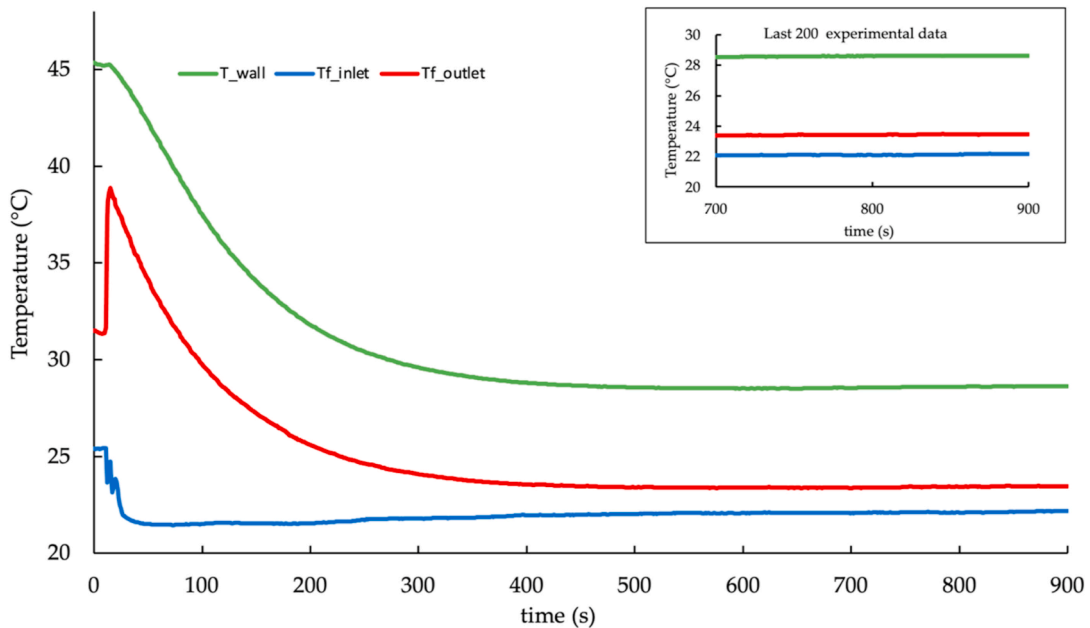


Fig. 5. Temperatures variation measured during an experimental test for a volumetric flow rate of 200 ml/min.

Table 1
The current study's experimental uncertainties.

Variable	Uncertainty range
Serpentine parameters, L , h , w , D_h	± 0.05 mm
Temperature, T	± 0.3 °C
Volumetric flow rate, Q	± 0.035 ml/min
Irradiance, I	± 9.61 W/m ²

45 $\pm$ 0.3 °C by our heat source and then the pump was turned on to initiate the cooling process. The measured temperatures were monitored in real time and after a period varying between 15 and 25 min, it was observed that the system reached equilibrium, which are characteristics of steady state. Simulating this entire cooling process using the transient method should yield the same results, but it would require an immense amount of computational time to reach the equilibrium condition, making this possibility practically unfeasible. Apart from this, a steady-state simulation will offer a quicker solution convergence, making it easier to validate against the experimental results since the time-averaged results can be compared. Finally, by simulating the steady-state flow regime through the developed heat exchanger, it can be optimized its design to maximize the heat transfer capability of the system, whilst minimizing the pressure drop.

Considering the complexity of the system, the problem under study is numerically solved by properly addressing the conservation equations of flow, heat transfer, and radiation. The continuity equation (mass conservation) is valid for compressible and incompressible fluids and is described by Equation (1).

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0 \quad (1)$$

where $\mathbf{u}$ is the velocity vector, and ρ is the mass density of the fluid [25].

Equation (2) is the momentum conservation equation that describes the motion of fluids during flow, being a non-linear partial differential equation.

$$\frac{\partial (\rho \mathbf{u})}{\partial t} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) = -\frac{\partial p}{\partial \beta} + \nabla \cdot (\mu \nabla \mathbf{u}) + S_M \quad (2)$$

where p is the static pressure, μ is dynamic viscosity, β represents the components from directions x , y and z , and S_M represents momentum source [25]. The terms on the left present the variation in density over time and the convective term while the last one is the source of momentum per unit volume per unit time linked with body forces like, for instance, gravity being in this case S_M given by $-\rho \mathbf{g}$. In incompressible flows this term could be related to the buoyancy effect.

The energy equation to be solved takes the following form (Equation (3)).

$$\frac{\partial (\rho i)}{\partial t} + \nabla \cdot (\rho i \mathbf{u}) = -p \nabla \cdot \mathbf{u} + \nabla \cdot (k \nabla T) + \Phi + S_i \quad (3)$$

Where ρ is the density of the fluid, i is the internal energy, $\mathbf{u}$ is the velocity, p is the pressure, k is the thermal conductivity, T is the temperature, Φ is the dissipation function, and S_i is the additional source term [25]. The last one is the additional source term per unit volume per unit time and it is associated to different effects that impact on the fluid flow and behavior including, for instance, chemical reactions in reactive flows and applied external forces. The term S_i is equal to $S_E - \mathbf{u} \cdot S_M$, where S_E is the source of energy per unit volume per unit time, $\mathbf{u}$ is the fluid velocity, and S_M is the already mentioned momentum source term.

Since the study includes radiation, Equation (4) describes this phenomenon for absorption, emission, and scattering.

$$\frac{dI(\vec{r}, \vec{s})}{ds} + (a + \sigma_s) I(\vec{r}, \vec{s}) = an^2 \frac{\sigma T^4}{\pi} + \frac{\sigma_s}{4\pi} \int_0^{4\pi} I(\vec{r}, \vec{s}') \Phi(\vec{s} \cdot \vec{s}') d\Omega' \quad (4)$$

where $\vec{r}$ is a position vector, $\vec{s}$ is direction vector, $\vec{s}'$ is scattering direction vector, s is path length, a is absorption coefficient, n is the refractive index, σ_s is the scattering coefficient, σ is the Stefan-Boltzmann constant, I is the radiation intensity, Φ is the phase function, Ω' is solid angle and $(a + \sigma_s)$ is the optical thickness or opacity of the medium.

The problem was solved using the Surface-to-Surface (S2S) Radiation Model, which considers the energy exchange between the radiating and absorbing surfaces. This model considers the size, distance between the surfaces, and orientation.

In ANSYS Fluent, this model assumes that the interacting surfaces are gray and diffuse, meaning that emissivity and absorptivity are independent of the radiated wavelength. Since the surfaces are diffuse, reflectivity is independent of incident or outgoing radiation directions. Another important consideration is that the model is not affected by the medium between the surfaces, in this case, the air.

In the radiation model, the energy flux leaving a specific surface is obtained by summing the directly emitted energy with the reflected energy. The energy flux reflected in the medium depends on the incident flux and is treated as a function of the outgoing energy flux, which can be given by Equation (5):

$$q_{out,k} = \varepsilon_k \sigma T_k^4 + \rho_k q_{in,k} \quad (5)$$

where ε_k is the emissivity and $q_{in,k}$ is the incident energy flux on the surface from the surroundings.

The amount of incident energy on a surface from another one directly depends on the surface-to-surface view factor, as expressed in Equation (6) for the incident radiation.

$$A_k q_{in,k} = \sum_{j=1}^N A_j q_{out,j} F_{jk} \quad (6)$$

where A_k is the area of surface k and F_{jk} is the view factor between surface k and surface j .

So,

$$q_{in,k} = \sum_{j=1}^N F_{kj} q_{out,j} \quad (7)$$

From where,

$$q_{out,k} = \varepsilon_k \sigma T_k^4 + \rho_k \sum_{j=1}^N F_{kj} q_{out,j} \quad (8)$$

By rewriting Equation (8), we obtain

$$J_k = E_k + \rho_k \sum_{j=1}^N F_{kj} J_j \quad (9)$$

where $J_{k,j}$ is the energy emitted by radiation from surface k and j , and E_k is the emissive power of surface k . The surface-to-surface view factor is the proportion of the radiative heat flux that leaves surface A and strikes surface B. This parameter measures how well one surface can “see” another surface. View factors are purely geometrical parameters independent of the physical surface properties and temperature. Nonetheless, they are important in determining the radiative heat flux between two surfaces. The view factor between two finite surfaces, i and j , can be expressed by Equation (10).

$$F_{ij} = \frac{1}{A_i} \int_{A_i} \int_{A_j} \frac{\cos \theta_i \cos \theta_j}{\pi r^2} \delta_{ij} dA_i dA_j \quad (10)$$

where δ_{ij} is obtained by the visibility of dA_j for dA_i .

The selected turbulence model is the Shear-Stress Transport (SST) k - ω Model [26], as it incorporates refinements over the Baseline (BSL) model [27] and considers the transport of shear stress in turbulence when defining turbulent viscosity. This makes the SST k - ω Model more accurate and reliable than the standard and BSL models. Therefore, the turbulence kinetic energy and specific dissipation rate are obtained through Equations (11) and (12) [26].

$$\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_i} (\rho k u_i) = \frac{\partial}{\partial x_j} \left(\Gamma_k \frac{\partial k}{\partial x_j} \right) + G_k - y_k + G_b + S_k \quad (11)$$

and

$$\frac{\partial}{\partial t} (\rho \omega) + \frac{\partial}{\partial x_i} (\rho \omega u_i) = \frac{\partial}{\partial x_j} \left(\Gamma_\omega \frac{\partial \omega}{\partial x_j} \right) + G_\omega - y_\omega + S_\omega + G_{\omega b} \quad (12)$$

where G_k is the generation of turbulence kinetic energy due to mean velocity gradients and G_ω is the generation of ω . Γ_k and Γ_ω are the effective diffusivity of k and ω , respectively. y_k and y_ω are the dissipation of k and ω due to turbulence. G_b and $G_{\omega b}$ consider fluctuation terms. S_k and S_ω are user-defined source terms.

The governing equations presented earlier were numerically solved using the finite volume method with the coupled algorithm, where the equations are solved simultaneously, and the dependent variables are calculated at the cell centers.

Thermal power transferred to the fluid was also calculated according to the fundamental equation of sensible heat calorimetry expressed in Equation (13).

$$\dot{Q} = \dot{m} C_p \Delta T \quad (13)$$

where $\dot{Q}$ is the power transferred to the fluid, $\dot{m}$ is the mass flow rate, C_p is the specific heat capacity, and ΔT is the temperature variation.

3.1. Computational model, mesh and boundary conditions

The computational domain is shown in Fig. 6. A radiating surface replaces the lighting source used in the experimental study, and an enclosure is created around the system to account for air movement's effects.

In order to fully define the situation under study, the boundary conditions must be defined. In the current study, all surfaces have a non-slip boundary condition applied, with the sides being treated as non-adiabatic and the outlets having zero-gauge pressure. The air inlet and outlet openings are assumed to occupy the entire sides of the box. The air flow is assumed to be turbulent flow. An inlet velocity of 0.0005 m/s was determined using an anemometer (Hot Wire 8903). More information about the surface boundary conditions can be found in Table 2.

The inlet fluid and ambient temperatures are based on the experimental conditions. The radiation source was modeled as a radiating surface with a temperature of 513 ± 0.3 K, experimentally determined. The front surface of the PV module (corresponding to the steel plate surface facing the radiation source) was defined as the radiation receiver. The standard software settings were used for the serpentine materials except for resin, which was used to connect the steel plate and aluminum exchanger; the settings shown in Table 3 were used.

Fig. 7 depicts the mesh constructed to discretize the computational domain under study. The mesh geometry was meticulously crafted to ensure uniform division throughout the domain, prioritizing seamless connectivity between cells while maintaining orthogonality. The generated vertices were grouped into equal sets and selected together to guarantee consistent subdivision. In order to capture the flow-field details, the mesh near the solar panel and fluid flow was refined. As a result, this specific region exhibits a higher density of elements, gradually decreasing as one moves away from it. Additionally, the face meshing command was applied to all faces to guarantee the overall high quality of the mesh.

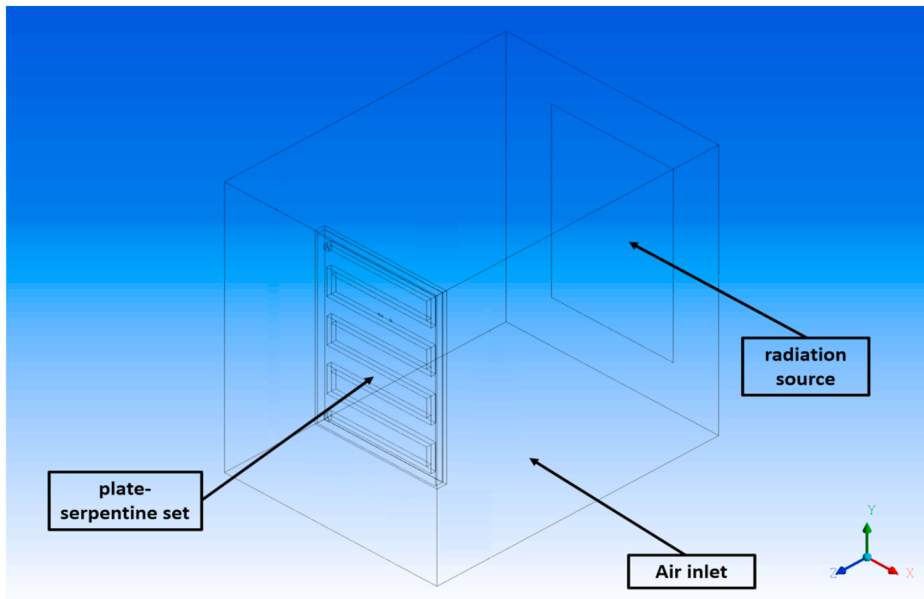


Fig. 6. Computational domain used.

Table 2
Surface boundary conditions.

Surface	Boundary Conditions
Fluid–Solid Interface	No-slip Interface and Convection
Interfaces between solid materials	Thermal Coupled
Air inlet	Velocity Inlet (0.0005 m/s)
Fluid inlet	Velocity Inlet (from experimental data)
Air and Fluid outlet	Zero-gauge pressure

Table 3
Thermophysical properties of the materials at 25 °C used in the simulation.

Properties	Aluminum Resin Reference
Mass density (kg/m^3), ρ	2700 0.97 [28]
Specific heat (J/kg.K), C_p	900 1460 [28]
Thermal conductivity (W/m.K), k	237 0.17 [29]

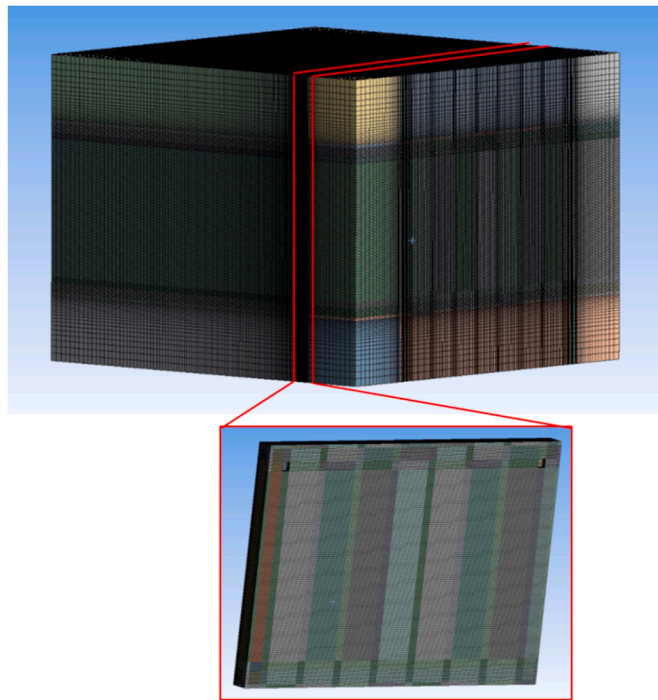


Fig. 7. Mesh used in the numerical simulation focusing on the panel and serpentine zone.

The mesh employed is orthogonal, non-uniform, and structured. A study of the non-interference of the mesh on the results was carried out, and the results are presented in Table 4. The mesh's aspect ratio, skewness, and orthogonality were used to validate its quality. The selected mesh was the number 2 (Table 5), comprising 7 225 344 hexahedral elements [30].

The mesh was initially generated based on the results of the lowest tested volumetric flow rate. Changes were then applied proportionally throughout the mesh geometry. The mesh was progressively refined until the numerical results matched the experimentally obtained results, ensuring the most coherent outcomes with the minimum number of elements and, consequently, the lowest possible computational time.

It was assumed a normalized wall distance y^+ equal to one, given that the intention is to capture the boundary layer heat transfer using a near wall model approach with mesh refinement.

For the grid independence test, three distinct grid configurations were used for the computational model adopted in the present work. There were used the computational representative outcomes of a volumetric flow rate of 191.48 ± 0.035 ml/min and an inlet velocity of 0.005 m/s, and the average outlet temperature as benchmark criterion. Fig. 8 illustrates the evolution of the average outlet

Table 4
Properties of the meshes that are employed in convergence.

	Mesh 1	Mesh 2	Mesh 3
Nodes	1234625	8311681	7340242
Elements	1198352	8185536	7225344
Minimum size (mm)	2.27e-2	1.91e-2	1.80e-2
Maximum size (mm)	1.91	1.13	1.59
Aspect ratio	4.49	3.05	2.963
Skewness	6.08e-7	6.22e-7	6.37e-7
Orthogonal quality	1	1	1

A comparison of the outlet fluid temperature results was performed, in which mesh number 2 presented the best results.

Table 5
Validation mesh comparison with experimental results.

	Experimental Temperature, °C	Numerical Results					
		Mesh 1		Mesh 2		Mesh 3	
		°C	Error (%)	°C	Error (%)	°C	Error (%)
Outlet Fluid	29.28	27.06	7.58	30.18	3.07	27.11	7.41

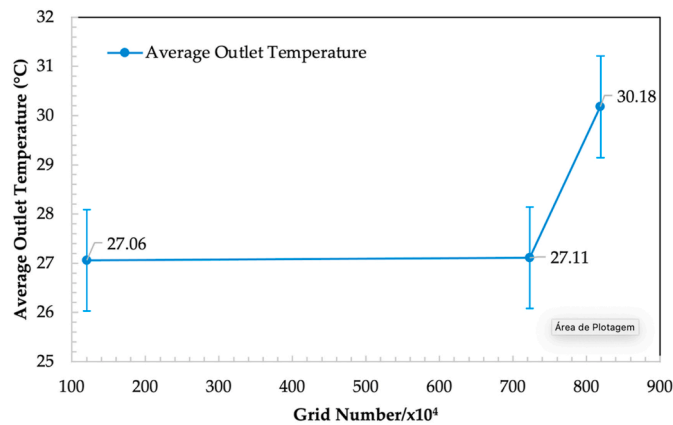


Fig. 8. Grid independence test using the outlet temperature.

temperature with grid number of elements. The analysis of Fig. 8 confirms that the modification rate in outlet temperature values stabilized and then sharply increased for the finest grid, being the obtained value, as already been referred, the closest to the experimentally measured one. Also, a subsequent analysis determined that the relative discrepancies between the average outlet temperature acquired from Mesh number 2 and finest meshes with more elements were negligible. Considering these facts, Mesh number 2 was the selected grid configuration comprising more than 8,185,000 elements.

4. Results and discussion

4.1. Validation of numerical results

The first step to be carried out is the validation of the numerical simulations with those obtained experimentally. As shown in Fig. 9, there was excellent agreement between the numerical and experimental results for the outlet fluid temperature. The mean absolute error was 0.46 °C, with a relative error of 1.78 %.

The results for flow rates below 60 ± 0.035 ml/min do not have a mean as close to the experimental values as those in the subsequent sequence; however, they are within the expected error range and can therefore be considered virtually correct. For the highest flow rate, the simulation result, although not within the considered error margin, has a relative value below 10 %, which is deemed satisfactory in the vast majority of works in this field. Factors not accounted for in the simulation, such as wall roughness, the velocity profile of peristaltic pumps, and potential dimensional deviations in the steel plate and the serpentine, may have introduced some additional noise into the results. Apart from this, the use of a sub-routine in the numeric simulation concerning the case of the highest volumetric flow rate would dim the discrepant values.

4.2. Influence of the serpentine material

In order to evaluate the serpentine material's influence, a numerical study was conducted considering another serpentine made with polydimethylsiloxane (PDMS). The PDMS is a prominent elastomer in biomedical engineering and microfluidics because of its special characteristics [31]. Recently, PDMS has been employed as a gas separation membrane as well [32], as a cell to enhance the measurements of nanofluids' thermal conductivity [29] and as an elastomer-based heat exchanger [33].

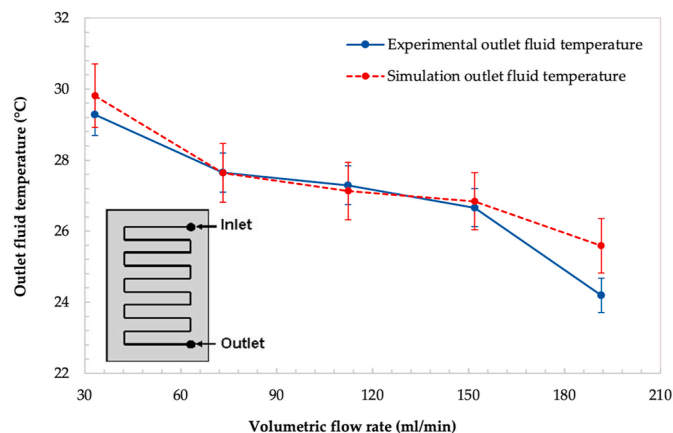


Fig. 9. Comparison of the aluminum serpentine's outlet fluid temperature measured experimentally and numerically.

The two serpentine analyzed in the current work, Aluminum Serpentine and PDMS Serpentine, have different thermal properties; one is made of an insulating elastomer, PDMS, and the other is made of a conductive material, aluminum. In order to evaluate the surface cooling effect of different materials, *i.e.*, the influence that the serpentine material has on the steel plate surface (which simulates the back rear of a photovoltaic solar panel), numerical simulation was carried out to obtain the thermal profiles when the steel plate is exposed to a heat source, simulating the solar radiation in the experiment. Additionally, a Wide PDMS Serpentine with 3x width, maintaining the same hydraulic diameter of 5 ± 0.05 mm, was simulated to investigate the effect of channel size.

Fig. 10 compares the steel plate average temperature values determined through numerical simulation, considering the same input data, for different volumetric flow rates and serpentine mentioned above. It is worth mentioning that the liquid flow direction is from top to bottom, as shown in Fig. 4.

One may observe that the steel plate temperature is lower when the Aluminum Serpentine is used compared to the PDMS Serpentine case (an average of 6.3 % lower) for all flow rates. It is also observed that with an increase in the fluid-surface contact area, by using a wide PDMS serpentine, the disadvantages of PDMS low material's thermal properties can be overcome.

In order to analyze the influence of the serpentine material on the heat transfer to the working fluid, Fig. 11 shows the outlet fluid temperature as a function of volumetric flow rates. The Aluminum Serpentine has more capability of transferring heat to the fluid for the lower flow rates. As the mass flow rate increased, this advantage for the Aluminum Serpentine became negligible since the fluid-surface contact area for the Wide PDMS Serpentine positively affected the overall thermal performance.

The sensible heat gain by fluid was also evaluated according to the flow rate for each serpentine. Fig. 12 corroborates what was shown regarding fluid temperature; the Aluminum Serpentine achieved better thermal performance results for lower flow rates ($< 160 \pm 0.035$ ml/min), while the Wide PDMS Serpentine performed better as the flow rate increased. Moreover, the wide PDMS serpentine exhibited a better cooling performance than the narrow PDMS serpentine (an average of 92 %) since the heat transfer surface area was enhanced in the former case.

In addition, Fig. 13 shows the steel plate temperature distribution for Aluminum and PDMS serpentine and a fixed volumetric flow rate (191.48 ml/min). Such a figure illustrates the influence of serpentine material on surface temperature behavior. Blue and red hues show the minimum and highest temperatures for each surface temperature contour. This contour shows that the liquid flow

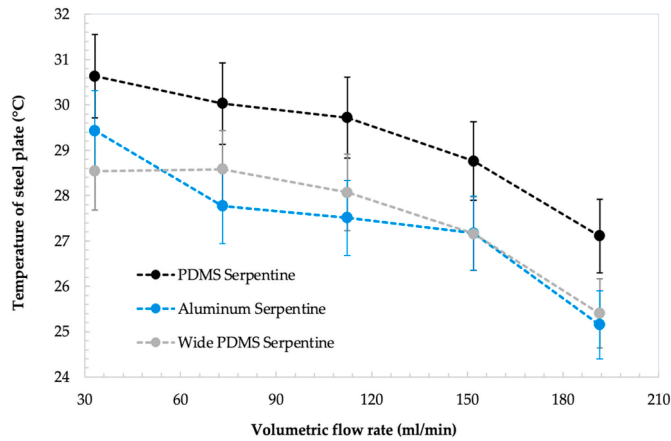


Fig. 10. Numerical results for the steel plate temperature for different volumetric flow rates and serpentine materials.

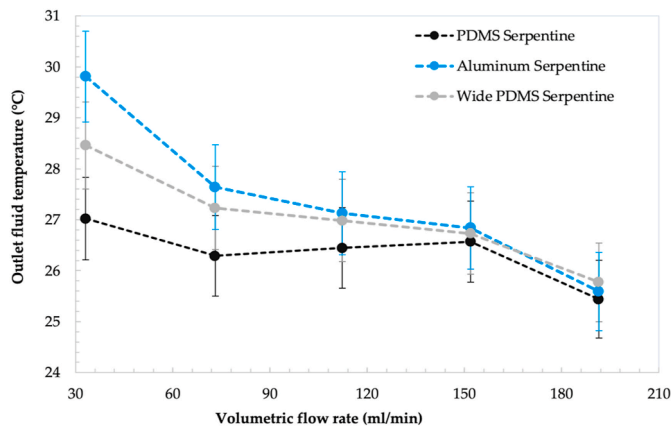


Fig. 11. Outlet fluid temperature from numerical simulation for different volumetric flow rates.

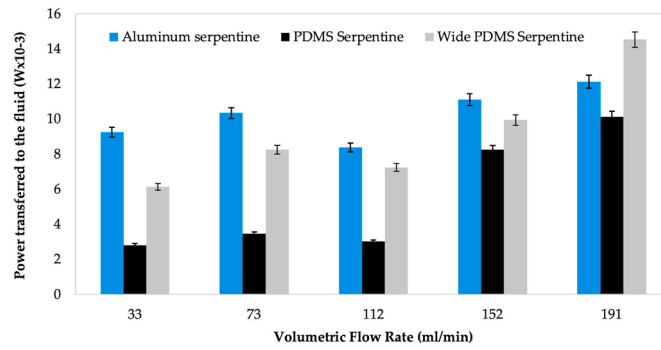


Fig. 12. Numerical results for the power transferred to the fluid for different volumetric flow rates.

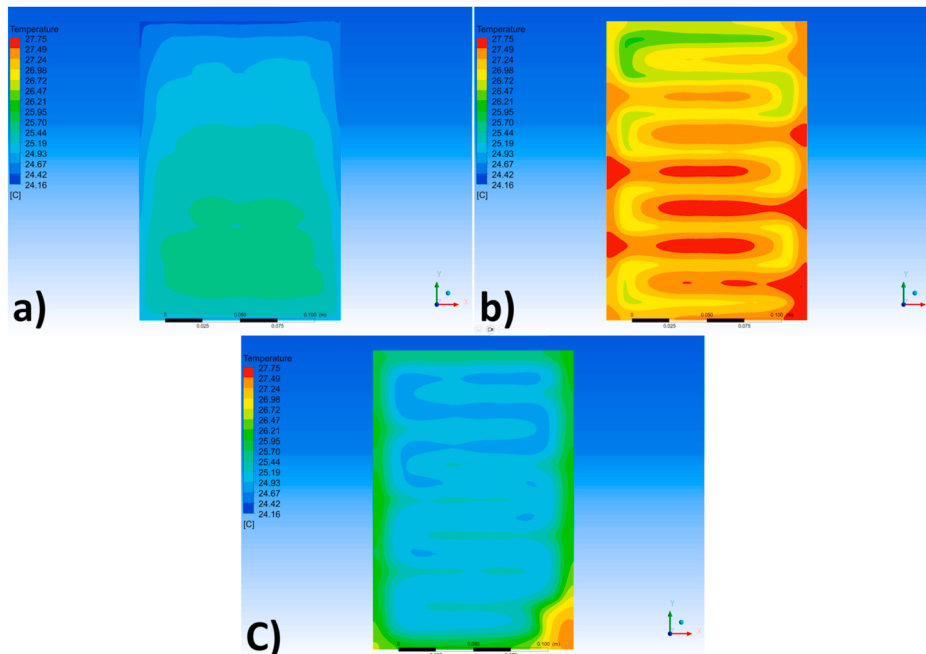


Fig. 13. Temperature distribution on the steel plate surface for a volumetric flow rate of 191.48 ± 0.035 ml/min a) aluminium serpentine, b) PDMS serpentine, and c) wide PDMS serpentine.

increases thermal activity as it passes through the heat exchanger. For Aluminum Serpentine (Fig. 13a), the hue of the surface near the outflow port becomes almost green, indicating a higher heat transfer rate than PDMS Serpentine. The results for the Wide PDMS Serpentine indicated a greater temperature homogeneity compared to the other serpentes, only showing a slight increase in the temperature at the edges of the plate where there is no contact of the fluid with the steel plate.

Besides, the Wide PDMS serpentine has a higher aspect ratio (ratio between the hydraulic diameter and channel width) and a larger cross-sectional area. Hence, for a given inlet volumetric flow rate in the wide serpentine with a higher aspect ratio, the fluid flow velocity decreases, the vortex region at the outlet shrinks, and the stagnant region at the bottom of the structure expands. The widening of the serpentine is expected to increase the heat transfer surface area, enhancing the heat dissipation capability. Hence, the cooling effect will be improved by increasing the aspect ratio under the same inlet flow rate. Among the temperature distribution profiles of the PDMS serpentes in Fig. 13, it can be seen that the profile of the Wide PDMS Serpentine presents a lower surface temperature. Even though a larger serpentine geometry implies a heavier structure, this negative impact will be offset by the better heat transfer performance of the Wide PDMS Serpentine.

For cooling applications in solar PV panels, aluminum can efficiently dissipate heat from a PV panel due to its high thermal conductivity; however, a slight increase in the PDMS serpentine width can result in more efficient energy conversion efficiency and a longer lifespan of the PV system.

Regarding heat transfer capability, the thermal conductivity of PDMS is 0.17 W/m.k, which is very low compared with the thermal conductivity of, for instance, thermal pastes, which is usually between 3 and 5 W/m.K. However, directly comparing the two materials in the specific PV application, thermal pastes are a worse option, as they are unsuitable for outdoor applications and easily dry up, degrading performance. Hence, a periodic replacement would be needed, which does not occur with PDMS. Moreover, the ther-

mal pastes require a time-consuming application that may cause an uneven heat distribution when not properly done. Also, a too-large amount of paste will make it act as a heat insulator. Besides, they are very difficult to apply in the form of thin film with uniform thickness, which is comparatively simple in the case of the PDMS material. Apart from this, if transparency is not critical, nanoparticles and other nanostructures can be incorporated into the PDMS to enhance its thermal conductivity [34], benefiting heat transfer. Furthermore, using PDMS allows the creation of flow channels of different geometries for the heat exchanger, which is highly complicated compared to conventional ones [35].

5. Conclusions

The current work has analyzed two serpentine with different thermal properties. The serpentine made of an insulating material (PDMS) and another made of a conductive material (aluminum) were analyzed under the same conditions and cooled by water at the same volumetric flow rates. The following conclusions can be drawn.

- The simulation carried out in CFD proved to be valid and in agreement with the experimental results, thus allowing a reliable analysis of the parameters that can be difficult to measure experimentally, in addition to allowing extrapolations for different materials and boundary conditions.
- The outlet temperature of the operating fluid showed a significant difference only at the lower flow rates, demonstrating little influence of the serpentine material from a certain volumetric flow rate onwards.
- For all flow rates, the steel plate temperature using aluminum serpentine presented a lower average temperature than that with PDMS serpentine (an average of 6.3 % lower);
- Considering the aluminum serpentine, the results obtained in the experiments and numerical simulations agreed well. Particularly, when considering the outlet fluid temperature the mean absolute error between the simulated and experimental results was around 0.5 °C, with a relative error of approximately 1.8 %.
- The outlet fluid temperature in the case of aluminum serpentine is higher than that of PDMS serpentine, showing that aluminum serpentine has more capability of transferring heat to the fluid, especially at lower flow rates.
- Although aluminum has thermal advantages over PDMS, which can lead to more efficient cooling of the solar cells, potentially increasing the energy conversion efficiency, PDMS is a lightweight material, flexible, and generally less expensive compared to aluminum. Such characteristics make PDMS suitable for applications where good thermal contact and the overall weight of the cooling system are critical. Additionally, the thermal conductivity of PDMS can be increased by adding materials with superior thermal properties, which might be a cost-effective option for cooling systems.
- Increasing the serpentine's contact area effectively reduces the surface temperature and provides a more uniform temperature distribution at high flow rates.
- The wide PDMS serpentine showed a better cooling performance than the narrow PDMS serpentine with an average increase of 92 % in the power transferred to the fluid, given that the heat transfer surface area was enhanced in the former case.
- The numerical results indicated that the power transferred to the fluid increased with increasing volumetric flow rates for the Wide PDMS serpentine and PDMS serpentine. Increasing the volumetric flow rate from 33 ± 0.035 ml/min to 191 ± 0.035 ml/min induced an increase of 150 % and 340 % in the power transferred to the fluid for the Wide PDMS serpentine and PDMS serpentine, respectively.

In conclusion, choosing a PDMS serpentine or an aluminum serpentine for cooling solar PV panels involves considering a combination of factors, including the specific design and requirements of the PV panel, the desired cooling efficiency, the manufacturing process, the environmental conditions at the installation site, and the cost considerations.

As main future recommendations, more experimental and numerical works should be conducted to achieve a better understanding of the effect derived from the direct contact of the working fluid with the back of the PV panels. It strongly suggested experiments under real operating conditions testing an actual PV panel with a back made from polymeric materials like EVA or Tedlar, instead of using the steel plate. Also, more studies upon the feasibility and structural factors of the PDMS three-sided serpentine are most welcome. It should be tested more constitutive materials of the serpentine presenting distinct thermophysical properties like, for instance, thermal conductivity. Also, the heat exchanger design should be optimized in further research experimental and numerical works evaluating directly the main effects incoming from potential modifications of the serpentine configuration like altering the number and shape of the turns, internal angles, length, and cross-section dimensions. The obtained results will bring some more useful insights about the ideal configuration and width of the serpentine as well as their most appropriate composing material for an enhanced heat transfer performance.

CRediT authorship contribution statement

F. Barbosa: Writing – original draft, Validation, Software, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **G. Nobrega:** Validation, Software, Methodology, Formal analysis, Data curation. **R.R. Souza:** Writing – review & editing, Writing – original draft, Validation, Software, Methodology, Investigation, Funding acquisition, Formal analysis, Data curation, Conceptualization. **R.P. Ralha:** Writing – review & editing, Methodology, Investigation, Formal analysis. **E.M. Cardoso:** Writing – review & editing, Supervision, Software, Methodology, Investigation, Formal analysis, Data curation. **J.E. Pereira:** Writing – review & editing, Validation, Methodology, Investigation, Formal analysis, Data curation. **I.S. Afonso:** Validation, Software, Formal analysis, Data curation. **F. Marques:** Methodology, Investigation, Data curation. **J.C.F. Teixeira:** Validation, Supervision, Software, Methodology. **R. Lima:** Writing – review & editing, Supervision, Project administration, Methodology, Investigation, Funding acquisition.

sition. **A.S. Moita:** Writing – review & editing, Validation, Supervision, Project administration, Methodology, Investigation, Funding acquisition.

Declaration of competing interest

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

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Data availability

Data will be made available on request.

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