Research Article

Numerical Study on the Enhanced Thermal Performance of the Porous Media-Assisted Flat-Plate Solar Collector

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In this study, a novel flat-plate solar collector (FPSC) with inserted porous metal foam is designed. A numerical investigation of the thermal performance of FPSC is conducted. The effects of some factors, such as porous block shape (rectangular, trapezoidal, and triangular), number of inserted porous blocks, and permeability parameters, on the enhanced thermal performance of the FPSC channel are analyzed. Numerical results show that, among the three types of porous block shapes, FPSC inserted with rectangular porous blocks has the highest thermal performance. When the number of porous blocks inserted in the FPSC channel increases, the thermal performance is enhanced correspondingly. Furthermore, the permeability of porous blocks has a significant influence, showing that the thermal performance is the best at $Da = 10^{-2}$. Overall, the overall Nusselt number can reach a maximum value of 6.01 under the condition of $Da = 10^{-2}$, $N = 6$ for the rectangular porous blocks. Interestingly, as the nondimensional pressure drop in the channel, the friction factor reaches a maximum value of 8.66 under such a condition of $Da = 10^{-5}$, $N = 4$ for the rectangular porous blocks. Considering the pressure drop due to the porous foam occupying in the FPSC channel, a comprehensive performance evaluation criterion (PEC) is used to assess the thermal performance of the FPSC. The results show that the maximum value of PEC is even up to 1.68 under the condition of $Da = 10^{-2}$, $N = 6$ for inserting the rectangular porous blocks.

1. Introduction

Energy is an irreplaceable driving force for the development of human civilization. Nevertheless, the industrialized development of human society is consuming an enormous amount of fuel. This situation is exacerbating the potential depletion of finite fossil fuel and global warming. To achieve a secure energy future, there is a general worldwide agreement to control the amount of fossil energy needed. Therefore, scientists have made great efforts to carry out extensive research on some suitable alternatives from renewable energy sources (RES) in recent decades [1]. Among these alternative RES, as a promising option, solar energy is preferred on account of its free availability and inexhaustible clean source that mitigates overall warming and protects the environment [2, 3]. Regarded as a simple yet effective technique that converts incident radiant solar energy into thermal energy, the flat plate solar collector (FPSC) has been widely applied in various fields. Generally, FPSC is applied in low-temperature utilizations due to its lower thermal efficiency. Nevertheless, the thermal performance of the collector can be enhanced by various optimizing designs as well as advanced technologies and materials [4–6].

As a passive way of enhancing heat transfer, porous media materials can be availably utilized, which has attracted immense attention for solar thermal applications in the past several decades [7–10]. Due to the presence of voids and energy storage, the porous media matrix plays a vital role in the efficiency improvement of thermal systems. For this random voidage of porous foam, the porous structure presents a large contact surface per unit volume, and the fluid-saturated porous foams normally have superior thermal conductivity compared to ordinary fluids [11]. Consequently, it can promote fluid mixing and accelerate thermal dissipation, which in turn enhances convection heat transfer. In terms of a solar energy collector, when the
porous foam is exposed to solar radiation, the porous substrate can primarily absorb the radiant heat, store thermal energy, and then transfer heat to the working fluid by convection. Based on these advantages, a lot of efforts have been made to further utilize porous foams to elevate thermal efficiency in solar collectors or heat exchangers [12]. Although there have been some important studies on the application of porous media in solar collectors, there are few studies on the use of porous materials to improve the thermal performance of flat plate solar collectors [13–15].

In the past decade, a lot of valuable studies on the use of porous media in solar collectors to enhance the thermal performance of FPSC were carried out. Sopian et al. [16] designed a two-channel solar collector model in which the lower channel is partially filled with a porous medium. The numerical and experimental results showed that compared with the traditional single-channel solar collector, the outlet temperature of the double-channel solar collector with porous medium in the lower channel is higher and the thermal efficiency is higher. Chen and Huang and Chen et al. [17, 18] numerically studied the characteristics of enhanced forced convection in a channel of a solar-water collector by employing multiple metal-foam blocks at the inner wall of the absorber. Both the Darcy-Brinkman-Forchheimer model and the two-equation energy model based on local thermal equilibrium were used within the porous regions. The results showed that the recirculation vortices caused by the metal-foam block in the FPSC channel significantly augment the heat transfer rate on the heated surface, but the enhancement in heat transfer is unfortunately along with an inevitable rise in pressure drop as a penalty. Further, they analyzed in detail the effects of some parameters, such as the Darcy number, Reynolds number, and porosity, to illustrate considerable fundamental and practical results [19]. Rashidi et al. and Bovand et al. [20, 21] numerically studied combined convection–radiation heat transfer inside a porous solar heat exchanger with a sensitivity analysis based on volume-averaged equations within the porous media region. The analysis reveals that the Nusselt number increases with the porous layer thickness at higher values of the Darcy number, and vice versa. Also, the Nusselt number is sensitive to the porous substrate thickness at higher thickness values.

Jouybari et al. [22] experimentally investigated the thermal properties and pressure losses of FPSC with a fully filled porous channel. The results manifested that the insertion of porous foam improves the optimal efficiency and reduces the heat losses at low Reynolds number, and a Nusselt number augment of 82% is obtained by inserting a porous substrate. However, higher pumping power is demanded to achieve this increment due to the increased pressure drop in the channel. Furthermore, they also conducted an experimental study on the thermal performance of porous FPSC using the working fluid of SiO$_2$/deionized water nanofluids [23]. They reported that the porous channel containing nanofluids provided higher heat transfer but also resulted in a significant pressure drop. In general, the performance of porous channel with nanofluids was illustrated to be lower than that of empty channels. Besides, Jouybari et al. [24] theoretically analyzed the heat transfer in thin FPSC with a fully saturated porous channel considering the influence of solar radiation. But, the property of the optically thick material for the porous media was not considered in their analytical solutions. In Saedodin’s report [25], the local thermal equilibrium (LTE) assumption in the porous foam region was preferred for numerically calculating the thermal performance of the solar collector. The results show that the fully filled channel has a higher thermal performance than that of the empty channel, in which the inserting of porous media in the channel enhances the maximum thermal efficiency and Nusselt number up to 18.5% and 82%, respectively. However, the nondimensional pressure drop, i.e., friction factor, was observed to increase due to the use of porous insertion. Furthermore, a parameter was defined as performance evaluation criteria (PEC) to assess the FPSC performance, in which the collector performance consequently decreases with the increase in flow rate in the channel.

More specifically, some advances in the nanofluids used in the solar thermal system were reviewed by Said et al. [26], in which the emphasis was primarily focused on the aspects of energy, exergy, economic analysis, and environmental impact. Xiong et al. [27] numerically analyzed the combined results of porous media and mixed nanofluids in FPSC by using the two-phase model. Thermal radiation was used to study the thermo-hydrodynamic characteristics and heat transfer of a horizontal flat-plate solar collector. The results showed that the thermal properties of the porous solar collector have a significant effect on the thermal fluid dynamics. Anirudh and Dhinakaran [28] numerically studied the performance enhancement of FPSC in which the collector channel was filled with fully saturated porous metal foam. Both the extended Darcy-Brinkman-Forchheimer flow model and the radiation model of the Rosseland approximation were suggested to model the porous region. The computational results demonstrated that the flow and thermal fields varied when modeling buoyancy and radiation influences were comprehensively considered, and the porous metal foam in the FPSC channel could enhance the thermal performance because of better thermal mixing, along with buoyancy parameter and volumetric radiation parameter. Further, they performed numerical simulations to study the thermal performance of an FPSC by inserting intermittent porous blocks on the inner absorber wall taking into account different inserted points [29]. Kansara et al. [30] numerically and experimentally investigated the thermal performance of a novel flat-plate collector filled with porous media and air as a working fluid. The predictions showed that the effects of the number of fins, porous foam materials, and foam porosity on the performance of the collector were prominent, in which the air temperature in the collector with porous media rises by 16.17% compared with that of the empty channel structure. Moreover, the collector configuration was optimally proposed concerning the pressure drop in the channel. Recently, we reported a novel configuration design of an FPSC with an upper porous substrate on the inner of the absorber. The numerical analysis manifested that the porous substrate with chamfered cavities is conducive to the thermal performance enhancement of FPSC [31].
Based on the aforementioned review of the state-of-the-art, it is obvious that the insertion of porous media in the FPSC channel can promote the enhancement of heat transfer between the solar absorber and the working fluid, whereas the pressure drop of the FPSC channel occurs simultaneously as a penalty of higher pump power consumption. Therefore, there are a lot of design types for the FPSC configuration in which the porous material is arranged within the channel. In the current study, to aim at developing some novel passive design strategies for thermal performance enhancement in an FPSC channel, some factors, such as the structural shapes, permeability, and number of inserting intermittent porous blocks in the FPSC channel are mainly considered. The primary objectives are to numerically predict the optimal configuration of FPSC concerning both high heat transfer performance and the low-pressure drop caused by the insertion of porous blocks. The effects of shapes (rectangle, trapezoid, and triangle), number, and permeability of porous blocks on the thermal performance of FPSC are investigated in the present numerical endeavor. The effects of various factors on the Nusselt number ($N_u$) of heat transfer characteristics and the nondimensional pressure drop, i.e., friction factor ($f_m$) of flow characteristics in the FPSC channel are analyzed. Afterward, the performance evaluation criterion (PEC) is analyzed by simultaneously considering the balance between heat transfer enhancement and pressure loss in the FPSC channel, which is regarded as a vital value to designing the optimal FPSC system configuration with inserted porous blocks. The research results can be used as a practical basis for the design of porous media-assisted FPSC in future commercial applications.

2. Mathematical Model

2.1. Problem Description and Assumptions. In this study, the schematic of the two-dimensional numerical model of FPSC is shown in Figure 1. The design of FPSC geometries is inspired based on the experimental configuration [25] with a channel size of 13 mm(H) $\times$ 800 mm(L). The thickness and optical properties of both the glass cover and absorber plate of the collector are neglected. To determine the steady heat transfer of FPSC, an equivalent amount of solar irradiation intensity is assumed to be in uniform with the constant heat source on the absorber ($q_w = 800$ W/m$^2$) in this numerical calculation, which can represent one real quantity of geographical incident radiation [32]. For the geometrical shapes of a porous block inserted on the inner wall of the FPSC absorber, there are four types to be designed as rectangular (REC), trapezoidal (TRA$_1$, TRA$_2$), and triangular (TRI), respectively. For each case, the number of inserted porous blocks is also shown in Figure 1, while the total basic width of all porous blocks has remained the same. Besides, the porous metal-foam matrix is homogeneous and isotropic aluminum (Al$_2$O$_3$) material, and its porosity and permeability of porous media are uniformly distributed. The thermophysical properties of porous blocks are assumed to be constant, as shown in Table 1. The working fluid in the collector channel is water ($Pr = 7$), and the inlet flow rate is 1.5 L/min ($Re = 351$). The fluid flows into the collector at a uniform velocity ($u_{in}$) and a constant temperature ($T_{in} = 20^\circ$C) at the FPSC inlet. The flow field is assumed to be steady, incompressible, and laminar in the run process, while buoyancy and radiation effects are neglected in this numerical study.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
<th>Unit</th>
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<tr>
<td>Total length</td>
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<td>m</td>
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<tr>
<td>Height</td>
<td>0.0078</td>
<td>m</td>
</tr>
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<td>Density</td>
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<tr>
<td>Thermal conductivity</td>
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<tr>
<td>Specific heat capacity</td>
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<td>J/(kg·K)</td>
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<tr>
<td>Darcy number</td>
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<tr>
<td>Material type</td>
<td>Al$_2$O$_3$</td>
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</tbody>
</table>
2. Governing Equations. Based on the above assumptions, the fluid viscous and inertial effects are taken into account in the porous media-assisted flat plate solar collector. Therefore, the extended Darcy-Brinkman-Forchheimer model \([33–36]\) is applied in the porous region. Meanwhile, the assumption of the local thermal equilibrium (LTE) model is made between the two phases in the area near the porous media. The 2D mathematical governing equations of flow and heat transfer in porous media are expressed as follows:

**Continuity equation:**

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0, 
\tag{1}
\]

where \(u\) and \(v\) are the velocity components of \(x\) and \(y\) directions.

**Momentum equation:**

\[
\frac{\rho}{\varepsilon^2} \left( u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = -\frac{\partial p}{\partial x} + \frac{\mu_{\text{eff}}}{\varepsilon} \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) - \frac{\mu}{K} u - \frac{\rho F}{\sqrt{K}} |V| u, 
\]

\[
\frac{\rho}{\varepsilon^2} \left( u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \right) = -\frac{\partial p}{\partial y} + \frac{\mu_{\text{eff}}}{\varepsilon} \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) - \frac{\mu}{K} v - \frac{\rho F}{\sqrt{K}} |V| v, 
\]

\(\bar{V} = \sqrt{u^2 + v^2}\) is the combined velocity of transverse and longitudinal, \(\varepsilon\) is the porosity, and \(K\) is the permeability.
of porous media. When there is no porous media material arrangement in the collector, it can be regarded as $K \to \infty$, $\varepsilon = 1$ in the channel. $\rho$ is the fluid density, $p$ is the fluid pressure, $\mu$ is the fluid viscosity, $\mu_{\text{eff}}$ is the effective viscosity, and $F = 1.75/\sqrt{150} \cdot 1/e^{1.5}$ is the inertial factor [35].

In addition, based on the hydraulic radius theory of Carman-Kozeny, the relationship of the permeability takes the form as follows [37]:

$$K = \frac{d_p^2 \varepsilon}{180(1 - \varepsilon)^2}. \quad (3)$$

Here, $d_p$ is the effective average pore diameter.

Energy equation:

$$\frac{1}{\varepsilon} \left( u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) = \alpha \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right), \quad (4)$$

where $T$ is the temperature and $\alpha$ refers to the thermal diffusivity.

2.3. Boundary Conditions. The boundary conditions calculated by numerical simulation in this study are as follows:

At the inlet of the FPSC channel, uniform velocity and temperature distribution are considered.

$$u = u_{\text{in}}, \quad v = 0, \quad T = T_{\text{in}}. \quad (5)$$

At the outlet of the FPSC channel, the fully developed condition is specified for all variables.

$$\frac{\partial v}{\partial x} = 0, \quad \frac{\partial u}{\partial x} = 0, \quad \frac{\partial T}{\partial x} = 0. \quad (6)$$

The bottom of the collector channel is assumed as an adiabatic wall with no-slip boundary conditions.

$$u = v = 0, \quad \frac{\partial T}{\partial y} = 0. \quad (7)$$

The top of the collector channel is exposed to solar radiation, and the radiation intensity is evenly distributed on the absorber plate. The cover plate also has no-slip boundary conditions.

$$u = v = 0, \quad \frac{\partial T}{\partial y} = -\frac{q_m}{k_{\text{eff}}} (\text{porous blocks}), \quad (8)$$

where $k_{\text{eff}}$ is the effective thermal conductivity of the fluid-saturated porous media.

2.4. Thermo-Hydrodynamic Parameters of FPSC. The overall friction factor in the collector is defined as

$$f_m = \frac{2H \Delta p}{\rho u^2 L}, \quad (9)$$

where $f_m$ is the overall friction factor, $H$ is the height of the collector channel, $L$ is the length of the collector channel, and $\Delta p$ is the pressure drop in the collector channel.

The local Nusselt number is calculated as

$$Nu = \frac{hH}{k} = \frac{q_m H/k}{T_w - T_m}, \quad (10)$$

where $Nu$ is the local Nusselt number, $h$ is the convection coefficient, $k$ is the thermal conductivity of the fluid, $T_w$ is the wall temperature, and $T_m$ is the average temperature of the fluid, which is calculated by
The formula for calculating the mean Nusselt number of single porous blocks is defined by

$$ T_m = \frac{\int_{y=0}^{H} uTdy}{\int_{y=0}^{H}udy} $$ \hspace{1cm} (11) $$

$$ \text{Nu}_{mi} = \frac{1}{I} \int_{X_i}^{X_{i+1}} \text{NudX}, $$ \hspace{1cm} (12)

The formula for calculating the mean Nusselt number of single porous blocks is defined by

where $I$ is the width of a single porous block, $X_i$ is the spatial position of the $i$th block from the channel entrance.

Figure 5: Streamline contours in the FPSC channel for various numbers of inserted porous blocks at $Da = 10^{-4}$: (a) $N = 6$, (b) $N = 4$, (c) $N = 2$, (d) $N = 0$. 
The overall Nusselt number ($N_u$) in the collector is calculated by the following formula:

$$N_u = \frac{\sum_{i=1}^{N} N_{u_{mi}}}{N}, \quad (13)$$

where $N$ is the number of inserted porous blocks.

3. Numerical Methodology

3.1. Numerical Details. In the present study, all simulations are performed by using ANSYS Fluent 19.1 software, which is based on the finite volume method. To obtain a proper mesh quality, the meshing near porous regions is refined, as shown in Figure 2(a). This meshing scheme is made to capture the sharp gradients that are present close to porous/fluid interfaces and boundaries, as well as ensure
appropriate grid refinement at the interface regions. The double precision option is chosen for the numerical solution. The calculation of pressure-velocity coupling is performed by using the SIMPLE algorithm. The second-order upwind scheme is applied for the momentum and energy equations. The laminar model is adopted in handling steady-state flow problems due to the low velocity in the collector channel. Finally, the convergence criteria for all solver variables are set to be $10^{-8}$ for one case of $N = 4$, $Da = 10^{-3}$ with inserted REC porous blocks. It indicates that the numerical scheme is feasible in the present study.

3.2. Validation of Grid Independence. To validate the grid independence under the conditions of $q_{in} = 800$ W/m$^2$, $T_i =$ 20°C, and an inlet flow rate of 1.5 L/min, a test of grid quality is performed to determine the accuracy of the numerical solution. Taking four REC porous blocks at $Da = 10^{-4}$ in the collector channel as a case of validation, three sets of grid points (30000, 120,000, and 250,000, respectively) are tested. Figure 3 displays the profiles of both pressure drop and temperature near the nonporous bottom surface of the collector channel. As shown in Figure 3, it can be seen that the variations of the pressure and temperature variables along the channel are almost coincident with each other for the two cases of 120000 and 250000 grids. Therefore, it can be confirmed that 120000 grids are sufficient for the numerical accuracy of this work.

Furthermore, the current solver is also verified with the numerical results of Vijaybabu et al. [35] at various Darcy number values, and satisfactory confidence is obtained, which is detailed in Figure 4. The nondimensional x-velocity ($U$) distribution is in good agreement with their numerical results. Sufficient confidence is achieved by using the present numerical methodology to continue with the next numerical experiments.

4. Results and Discussion

4.1. Hydrodynamic and Thermal Characteristics in Collector

4.1.1. Effect of the Number of Inserted Porous Blocks. To reveal the effect of the number of inserted porous blocks on the flow characteristics in the collector channel, Figure 5 shows the streamline contours of a various number of REC porous blocks at $Da = 10^{-4}$ in the FPSC channel. It can be found that the streamlines are considerably distorted in the collector due to the insertion of the porous block arrangement. And clearly, some small flow recirculation regions appear behind each porous block in addition to the most of streamlines flowing around the porous blocks. As the number of inserted porous blocks reduces, the number of recirculation regions decreases correspondingly, so that no recirculation region eventually exists in the empty channel, as shown in Figure 5(d). Based on the hydrodynamic principle, when the porous blocks are mounted on the upper wall of the FPSC channel, the flow pressure drop near the porous blocks increases undoubtedly. Under the condition of a constant inlet velocity, since the porous blocks incompletely occupy the entire cross channel, the fluid tends to flow along such a path line in a low-resistance space without porous media. Therefore, only a small portion of fluid flows through the porous blocks with higher frictional resistance. As a result, there is a certain difference in fluid flow momentum near the back of each porous block that results in the formation of recirculation vortexes.

Corresponding to the above characteristics of flow fields shown in Figure 5, Figure 6 shows the temperature field characteristics in the channel. Clearly, the recirculation regions occur near the back of the porous block to promote flow mixing. The interaction between the recirculation vortex inside the intermittent space of the porous blocks and the external core flow plays a predominant role in affecting the temperature field. As a consequence, a thinner thermal boundary layer exists and becomes significantly distorted in the region near the inner wall of the FPSC absorber, wherein the convective heat transfer is generally enhanced in the channel. Furthermore, there is a constant total basic width ($L = 36$ cm) of all porous blocks, and the increasing number of porous blocks will lead to the emergence of more recirculation regions in the intermittent space of porous blocks, and thereby, it intensifies more frequently the flow mixing to enhance the heat transfer.

To further analyze the heat transfer in the FPSC channel, the temperature curves at the exit are displayed in Figure 7. When the number of REC porous blocks inserted is $N = 6$, the local outlet temperature of the FPSC channel is always higher than in other cases. Therefore, it can be further confirmed that the thermal performance of the FPSC channel with the number of inserted porous blocks ($N = 6$) is intensified to the best. Interestingly, for the numbers of inserted porous blocks with $N = 2$ and $N = 4$, the local outlet temperatures are not always higher than those of the empty channel. In the nonporous space of the bottom exit of FPSC, the local outlet temperature of the empty channel is lower.
than that of FPSC with inserted porous blocks of $N = 2$ or $N = 4$. However, at the upper exit with the porous block obstacle, the local temperature curve of the empty channel exceeds those of the $N = 2$ and $N = 4$ porous blocks. Nevertheless, by a comparison of the averaged outlet temperatures for the various numbers of inserted porous blocks, it can be found that the averaged outlet temperature rises with the increase in the number of inserted porous blocks. Considering the complexity of the flow in the FPSC channel with the inserted porous blocks, the overall thermal performance will be quantitatively analyzed later.

4.1.2. Effect of Various Shapes of Porous Blocks. Figure 8 shows the flow characteristics in the channel with the number of porous blocks $N = 4$ and $Da = 10^{-4}$ for different structural shapes of porous blocks which present four various shapes of REC, TRA$_1$, TRA$_2$, and TRI, respectively. As can be seen from Figure 8, compared with the cases of TRI and
TRA porous blocks, the recirculation regions are more likely to appear behind the back of REC porous blocks. As the porous block shape changes to the TRA1, the size of recirculation regions behind the back of porous blocks obviously reduces until the recirculation vortexes disappear behind the TRA2 and TRI porous blocks. When the number and permeability of porous blocks remain constant, the volume of REC porous blocks is the largest among the four shapes, and thus it will cause the greatest flow viscous and inertial resistances within the porous blocks. The flow fields are more easily disrupted near the back of the porous block due to the larger pressure gradient, and thus the recirculation vortexes behind the back of the REC porous blocks are more likely to emerge. Nevertheless, the recirculation regions behind the TRI porous blocks are less likely to occur since they have the smallest resistance loss.

Figure 9 shows the temperature field characteristics in the channel under the conditions in Figure 8. As can be seen
from Figure 9, compared with the TRA and TRI porous blocks, the REC porous blocks are more likely attached with recirculation regions near the backs of the porous block. And thus, a thinner boundary layer is formed to enhance heat transfer in FPSC. Moreover, the REC porous blocks have the greatest volume due to more fluid flowing through the REC porous media zone, which promotes stronger capacities for heat exchange and energy storage.

4.1.3. Effect of the Darcy Number (Da). Figure 10 shows the effect of the Darcy number of the REC porous blocks on the flow characteristics in the collector for a case of $N = 6$. It is clear that the variation in porous block permeability has a significant impact on the flow characteristics in the FPSC channel. In Figures 10(a) and 10(b) at $Da = 10^{-2}$ and $Da = 10^{-3}$, no recirculation region occurs behind the backs of the porous blocks. As the Darcy number of the porous

Figure 10: Streamline contours around REC porous blocks at various Darcy numbers (Da) for $N = 6$: (a) $Da = 10^{-2}$, (b) $Da = 10^{-3}$, (c) $Da = 10^{-4}$, (d) $Da = 10^{-5}$. 
blocks decreases to $Da = 10^{-4}$ in Figure 10(c), the streamlines are observably distorted in the collector and more amount of fluid flows around the porous blocks with higher flow resistance. As a consequence, under the condition of a large pressure gradient between the porous and nonporous regions, recirculation vortexes occur behind each porous block. When the Darcy number of the porous blocks further decreases to $Da = 10^{-5}$ in Figure 10(d), the flow resistance in porous regions further rises to generate a greater driving potential, and thus the disturbance resulting in recirculation

![Isotherm contours in the FPSC channel with REC porous blocks at various Darcy numbers for $N = 6$: (a) $Da = 10^{-2}$, (b) $Da = 10^{-3}$, (c) $Da = 10^{-4}$, (d) $Da = 10^{-5}$.](image)

Figure 11: Isotherm contours in the FPSC channel with REC porous blocks at various Darcy numbers for $N = 6$: (a) $Da = 10^{-2}$, (b) $Da = 10^{-3}$, (c) $Da = 10^{-4}$, (d) $Da = 10^{-5}$. 
regions becomes sharper, which boosts the recirculation regions behind the porous blocks, and the recirculation regions slightly move spatially from the absorber wall.

Based on the conditions in Figure 10, Figure 11 shows the corresponding temperature field characteristics in the channel. It can be found that under the conditions of the high permeability of REC porous blocks at $Da = 10^{-2}$ and $Da = 10^{-3}$ in Figures 11(a) and 11(b), the temperature field distribution is relatively uniform, and the averaged outlet temperatures are higher than those of low permeability of REC porous blocks at $Da = 10^{-4}$ and $Da = 10^{-5}$. For the cases of the high permeability of porous blocks, more amount of fluid flows through the porous blocks without the emergence of recirculation regions. As a result, liquid-solid convective heat transfer performs better inside porous blocks. Additionally, the more fluid contacts more sufficiently with the upper heat source of the absorber in a thermal boundary layer so that the convective heat transfer is further enhanced, whereas for the conditions of low permeability in Figures 11(c) and 11(d), the temperature distributions emerge in a more non-uniform channel due to the larger resistance of porous blocks in the FPSC channel. Thereby, less amount of fluid flow contacts within both the porous blocks and the upper wall of the thermal boundary, and a decrement in heat transfer rate must occur. Undoubtedly, it can be illustrated that the FPSC channel with high-permeability porous blocks displays better heat transfer performance.

4.2. Variation of the Overall Nusselt Number ($Nu_g$). To quantitatively analyze the heat transfer characteristics of the porous media-assisted FPSC channel, Figure 12 presents the profiles of the overall Nusselt number ($Nu_g$) in the FPSC channel with the effects of different porous block shapes, the number of inserted porous blocks, and the Darcy number under the condition of an inlet flow rate of 1.5 L/min. From Figure 12 and Table 2, it can be seen that the overall Nusselt
number monotonically rises with the change of porous block shapes from the TRI to the REC, indicating that with the same number of inserted porous blocks, the heat transfer performance in the FPSC channel with inserted REC porous blocks is the best. On the one hand, as the total basic width of the porous blocks mounted on the inner wall of the absorber remains constant, the total contact area of heat transfer within the REC porous blocks is larger than that of the TRI and TRA porous blocks, so the heat transfer inside the REC porous blocks is more sufficient. On the other hand, based on the qualitative results in Figure 8, some potential vortexes readily occur behind the back of REC porous blocks, which can intensify the thermal mixing of the fluid and the temperature gradient close to the inner wall of the absorber. Thus, the heat convection behind the back of porous blocks is enhanced.

Furthermore, in Figures 12(a)–12(c), as the number of inserted porous blocks increases at $N = 2$, $N = 4$, and $N = 6$, respectively, the flow resistance disturbance will become more dramatic, and thus the thermal flow mixing will be further intensified. It indicates that the increase in the number of porous blocks is conducive to enhancing the heat transfer performance in the FPSC channel. Besides, the permeability of porous blocks can also influence heat transfer in the porous regions. In Figure 12, it can be found that the overall Nusselt number augments with the increase of the Darcy number. Further, by a comprehensive comparative analysis in Figure 12(c), the results show that the heat transfer performance in the FPSC channel is the best with inserted REC porous blocks at $N = 6, Da = 10^{-2}$, and the overall Nusselt number can reach a maximum value of 6.01, which is twice as much as that in the empty channel.

4.3. Variation of Friction Factor ($f_m$). The insertion of porous blocks in the FPSC channel can significantly enhance the heat transfer performance, but the resistance dissipation effect in the channel is potentially increased due to the larger pressure loss caused by porous blocks. Therefore, it is necessary to conduct a detailed analysis of the pressure drop characteristics in the FPSC channel with inserted various porous blocks. In conjunction with the quantitative analysis in Table 2 again, Figure 13 shows the variations of friction factor ($f_m$) in the channel under various shapes, inserting numbers, and Darcy numbers of porous blocks at the inlet flow rate of 1.5 L/min. It can be notably seen that the variations of friction factor present a significant increase when the shape of porous blocks changes from the TRI to the REC. Among the porous blocks, the inserted REC porous blocks cause the maximum pressure drop in the collector due to its largest obstacle volume. As $Da$ decreases, inertial and viscous resistances offered to the fluid flow increase, and thus $f_m$ almost linearly increases with the change of the porous block shape from the TRI to the REC, as shown in Figure 13. By comparisons of all cases, it is interestingly found that $f_m$ has a maximum value of 8.66 for the case of the REC porous blocks $N = 4, Da = 10^{-5}$, whereas $f_m$ comparatively decreases to 8.26 for the case of $N = 6$ and $Da = 10^{-5}$. And it indicates that the cases of $N = 4$ are the most serious to result in the largest pressure drop in the collector. It is reasonable to infer that the flow disruption is the largest in the case of $N = 4$ as the working fluid flows through the porous regions. When the number of porous blocks increases to $N = 6$, it discourages the formation of a continuous hydrodynamic layer, which reduces the viscous resistance to some extent, and hence, the lower $f_m$ values are

Table 2: Comparisons of the overall Nusselt number ($Nu_\theta$) and friction factor ($f_m$) in the FPSC channel by inserting porous blocks with various shapes, inserting number, and Darcy number.

<table>
<thead>
<tr>
<th>Shape</th>
<th>$Da$</th>
<th>$Nu_\theta$</th>
<th>$f_m$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$N = 2$</td>
<td>$N = 4$</td>
<td>$N = 6$</td>
</tr>
<tr>
<td>REC</td>
<td>$10^{-3}$</td>
<td>4.07</td>
<td>4.40</td>
</tr>
<tr>
<td>REC</td>
<td>$10^{-4}$</td>
<td>4.18</td>
<td>4.70</td>
</tr>
<tr>
<td>REC</td>
<td>$10^{-5}$</td>
<td>4.26</td>
<td>4.78</td>
</tr>
<tr>
<td>REC</td>
<td>$10^{-2}$</td>
<td>4.93</td>
<td>5.91</td>
</tr>
<tr>
<td>REC</td>
<td>$10^{-3}$</td>
<td>3.76</td>
<td>3.86</td>
</tr>
<tr>
<td>REC</td>
<td>$10^{-4}$</td>
<td>3.79</td>
<td>3.94</td>
</tr>
<tr>
<td>REC</td>
<td>$10^{-5}$</td>
<td>3.93</td>
<td>4.26</td>
</tr>
<tr>
<td>TRI</td>
<td>$10^{-2}$</td>
<td>4.40</td>
<td>5.17</td>
</tr>
<tr>
<td>TRI</td>
<td>$10^{-3}$</td>
<td>3.36</td>
<td>3.69</td>
</tr>
<tr>
<td>TRI</td>
<td>$10^{-5}$</td>
<td>3.52</td>
<td>3.91</td>
</tr>
<tr>
<td>TRI</td>
<td>$10^{-2}$</td>
<td>3.82</td>
<td>4.48</td>
</tr>
<tr>
<td>TRI</td>
<td>$10^{-3}$</td>
<td>3.11</td>
<td>3.40</td>
</tr>
<tr>
<td>TRI</td>
<td>$10^{-4}$</td>
<td>3.13</td>
<td>3.46</td>
</tr>
<tr>
<td>TRI</td>
<td>$10^{-5}$</td>
<td>3.29</td>
<td>3.72</td>
</tr>
<tr>
<td>TRI</td>
<td>$10^{-2}$</td>
<td>3.61</td>
<td>4.25</td>
</tr>
<tr>
<td>Nonporous channel</td>
<td></td>
<td>2.98</td>
<td></td>
</tr>
</tbody>
</table>

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presented for the case of \( N = 6 \) in comparison to the case of \( N = 4 \).

4.4. Performance Evaluation Criteria (PEC). Compared with the thermal performance of conventional FPSC empty channel, the insertion of porous blocks in the FPSC channel can enhance heat transfer. However, because of the increasing pressure losses in the FPSC channel, the cost of the system pumping power also rises. Therefore, the balance between heat transfer enhancement and pressure losses in the FPSC channel should be taken into consideration. As a comprehensive evaluation parameter, the performance evaluation criterion (PEC) is used to analyze the thermal enhancement performance of porous media-assisted FPSC channel. The PEC expression is as follows \([2, 12]\):

\[
\text{PEC} = \frac{\left( Nu_{N=6}/Nu_{N=0} \right)}{\left( f_{N=6}/f_{N=0} \right)^{1/3}}.
\]  

(14)

Details of the PEC variation are shown in Figure 14 for various cases included in the study. It is clear that the PEC value increases with the change of the porous block shapes from the TRI to the REC at \( Da = 10^{-2} \) and \( Da = 10^{-3} \), in which the PEC reaches a maximum value of 1.68 at \( N = 6 \), \( Da = 10^{-2} \) for the cases of inserted REC porous blocks. It is demonstrated that FPSC shows better thermal performance with inserted REC porous blocks at higher Darcy number values \( (Da = 10^{-2}, 10^{-3}) \) under the number of \( N = 2 \sim 6 \) due to the lower pressure losses. Besides, the PEC value increases with the change of porous block shapes from the TRI to the REC. However, as shown in Figures 14(b) and 14(c), under the conditions of a low Darcy number \( (Da = 10^{-4}, 10^{-5}) \), it can be found that the change of the porous block shape has no significant influence on the PEC values. Especially, for the condition of \( N = 6 \) at a low Darcy number, the PEC profiles show a trend of almost horizontal development in spite of the change of porous block shape. Furthermore, it

Figure 13: Variations of friction factor with various shapes, inserting numbers, and the Darcy number of porous blocks: (a) \( N = 2 \), (b) \( N = 4 \), (c) \( N = 6 \).
is evident that the PEC values under the conditions of high Darcy number \((Da = 10^{-2}, 10^{-3})\) exceed 1.0 for \(N = 4\) and 6 in Figures 14(b) and 14(c), which means that the heat transfer enhancement outbalances the penalty of pressure losses in the FPSC channel. In conclusion, if the pressure drop is not a major concern, the inserted REC porous blocks can provide maximum heat transfer augmentation in the channel. And for such evaluations of low Darcy number cases, regardless of porous block shapes, the PEC values are always less than 1.0, which implies that the effect of pressure loss is greater than that of heat transfer enhancement in the FPSC channel.

5. Conclusions

In this paper, numerical simulations are carried out on the thermal enhancement performance of porous media-assisted flat-plate solar collectors. The effects of the shape, number, and permeability of inserted porous blocks on the flow and heat transfer performance in the channel of a flat plate solar collector are studied, and the following conclusions are summarized:

1. Considering the effects of different porous block shapes, the recirculation regions are more likely to emerge behind the back of the REC porous block under such conditions of low Darcy number \((Da = 10^{-5}, 10^{-4})\), which is beneficial to enhance convective heat transfer in the FPSC channel, whereas no recirculation regions emerge in the channel at high Darcy number \((Da = 10^{-3}, 10^{-2})\)

2. For the parametric analysis of the effect of porous blocks on the thermal performance of FPSC, the convective heat transfer is the strongest under the condition of \(N = 6\) and \(Da = 10^{-2}\), and especially, the overall Nusselt number can reach a maximum value of 6.01 for the case of inserted REC porous blocks

Figure 14: Variations of performance evaluation criteria (PEC) with the change of porous block shapes at various Darcy numbers for the FPSC channel: (a) \(N = 2\), (b) \(N = 4\), (c) \(N = 6\).
(3) As the pressure drop cannot be neglected due to the porous block occupying in the FPSC channel, a comprehensive analysis of performance evaluation criteria (PEC) is necessary. The best performance evaluation criteria are under the condition of high permeability \((Da = 10^{-3})\) with \(N = 6\) for REC porous blocks, and the PEC value reaches 1.68, which is improved by about 68% compared to the empty channel.

**Nomenclature**

- \(c_p\): Specific heat at constant pressure (J/kg·K)
- \(f_m\): Friction factor
- \(F\): Inertial factor
- \(D_h\): Hydraulic diameter (m)
- \(d_p\): Effective average pore diameter (m)
- \(g\): Gravitational acceleration (m/s²)
- \(K\): Permeability of porous media (m²)
- \(Da\): Darcy number
- \(Q\): Volume flow rate (lit/min)
- \(Rd\): Volumetric radiation parameter (\(4\alpha T_{w}\)³/\(k\beta_r\))
- \(Ri\): Richardson number (\(g\beta_r(T_{w} - T_{m})/D_h\))
- \(Nu\): Nusselt number
- \(Nu_{mi}\): The average Nusselt number at each block
- \(Nu_{m}\): The overall Nusselt number
- \(N\): Number of blocks
- \(Pr\): Prandtl number (\(\nu/\alpha\))
- \(H\): Collector channel height (m)
- \(L\): Collector channel length (m)
- \(h\): Porous block height (m)
- \(s\): Porous block spacing (m)
- \(I\): Porous block width (m)
- \(w\): Porous block width (
- \(v\): \(x, y\)-component of velocity at the inlet (m/s)
- \(q_w\): Radiative heat flux (W/m²)
- \(x, y\): Horizontal, vertical distance (m)
- \(k\): Thermal conductivity of the material (W/m·K)
- \(k_{eff}\): The effective thermal conductivity of the material (W/m·K)
- \(p\): Pressure (Pa)
- \(T_w\): The wall temperature (K)
- \(T_m\): The average temperature of the fluid (K)
- \(F\): The inertial factor.

**Greek Symbols**

- \(\alpha\): Thermal diffusivity (m²/s)
- \(\mu\): Viscosity (kg/m·s)
- \(\mu_{eff}\): Effective viscosity (kg/m·s)
- \(\nu\): Kinematic viscosity (m²/s)
- \(\varepsilon\): Porosity
- \(\sigma\): Stefan-Boltzmann constant
- \(\rho\): Density (kg/m³)
- \(\beta\): Channel inclination angle (°)
- \(\beta_{r}\): Extinction coefficient.

**Abbreviation**

- FPSC: Flat-plate solar collector
- TRI: The shape of the porous blocks, triangle
- TRA1: The shape of the porous blocks, trapezoid with slope, 6 h/l
- TRA2: The shape of the porous blocks, trapezoid with slope, 3 h/l
- REC: The shape of the porous blocks, rectangle
- PEC: Performance evaluation criterion
- LTE: Local thermal equilibrium
- 2D: Two-dimensional.

**Data Availability**

The original/processed data required to reproduce these findings cannot be shared at this time as the data also forms part of an ongoing study.

**Conflicts of 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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