An in depth evaluation of matrix, external upstream and downstream recycles on a double pass flat plate solar air heater efficacy

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ABSTRACT

In the present study, the thermal and thermohydraulic characteristics of different types of double pass solar air heaters (DPSAHs) containing three flow channels were analytically investigated. The analyses were conducted at air mass flow rates of 0.01, 0.015, 0.025 kg/s and different reflux ratios of 0.1 to 1. The effects of upstream and downstream recycling patterns were analyzed. Moreover, the impacts of matrix placed between the absorber plate and the second glass cover with various porosities, and variation of solar radiation intensity on the DPSAH performance were examined. Furthermore, the pressure drop due to the existence of matrix was considered to obtain more realistic outcomes. The results demonstrated that as to the downward recycling pattern, using matrix leads to an increase in the DPSAH thermal efficiency presenting the thermal efficiency of 79%; however, it brings about a reduction in its thermohydraulic efficiency at high mass flow rates and reflux ratios and high fan power cost is incurred, subsequently. The most compelling result is: if we consider an upstream recycling pattern, it is true that we have overlooked a certain amount of the DPSAH efficiency compared to a matrix-based downward recycling DPSAH (about 7% less efficiency); however, at high mass flow rates and reflux ratios the thermal efficiency of DPSAH is similar to that of a matrix-based upward recycling DPSAH, and in this way, not only the hot air demand is supplied, also the extra fan power cost and the cost of providing a suitable matrix are eliminated.

1. Introduction

Today, solar air heaters (SAHs) are extensively applied to both industrial and domestic applications. They mostly serve the purposes of providing heat for cold climatic conditions, and drying food materials leading to reaching net zero greenhouse gases emission. Since, enhancing the thermal performance of the SAHS is of great significance in preserving energy, numerous researches have been conducted to advance their performance parameters.

Zheng et al. [1] mathematically investigated the effect of corrugated packing on the thermal and hydraulic efficiencies of an SAH. They also considered the pressure drop within the flow channel and compared the novel proposed SAH performance to that of unglazed and glazed transpired solar collectors. The results showed that in rural cold areas using the proposed novel SAH is systematically and economically more favorable than the other two ones due to its larger heat transfer area. A. Perwez and R. Kumar [2] enhanced the efficiency of a modified SAH up to 35% using a dimpled absorber plate. The highest increase in the outlet fluid temperature was reported as 4.6 °C for 0.009 kg/s flow rate. Moreover, the highest energy efficiency of the SAH was acquired at a 0.025 mass flow rate. Zheng et al. [3] numerically and experimentally surveyed the effect of using perforated corrugated absorber plate on the thermal performance of a glazed transpired collector for space heating applications. The results showed that the proposed transpired collector presents 11.36% and 13.57% more thermal efficiency compared to a transpired collector possessing a silk-like, and a perforated absorber plates, respectively. F. Bayrak et al. [4] conducted energetic and exergetic analyses of an SAH using an aluminum foam as the porous baffles. The experiments were carried out for the mass flow rates of 0.016 kg/s and 0.025 kg/s. The maximum energy and exergy efficiencies of the baffled collector were reported as 77% and 54%, respectively. Z. Wang et al. [5] constructed an integrated collector storage SAH based on lap joint type flat micro heat pipe arrays. The results indicated that the maximum energy efficiency of the solar collector is about 78% regarding 141 m²/hr flow rate. It was also concluded that changing the solar irradiance does not considerably affect the daily efficiency of the collector. P. Charvat et al. [6] used phase change materials for
M. Bazargan [10] utilized the genetic algorithm to optimize the outlet surface. They concluded that the ribs in SAHs enhance the thermal convection conditions are 73.5% and 65%, respectively. M. Ansari and 14% [7,8]. In 2019, M. Salih et al. [9] examined the effects of natural convection in the upper glass cover of an SAH through a specific curvature angle (25°). It was concluded that the packed bed wire mesh with a porosity of 93% can increase the thermal and the thermohydraulic efficiencies of the SAH up to 9% at low air flow rates. A. Kabeel et al. [11] improved the efficiency of a single pass SAH using fins and a guide blade attachment to the heater entrance for well distributing the air flow. They reached the efficiency of 57% for the proposed SAH containing 8 cm height fins. A. Singh and O. Singh [12] enhanced the thermal efficiency of an SAH through a specific curvature angle (25°). It was concluded that the thermal performance of the curved SAH is better than that of a flat one. In addition, Nusselt number for the curved SAH was about two times higher than that of a flat one. However, the heat loss of the flat SAH is lower than that of the curved one. E. Akpinar, and F. Kocyigit [13] investigated the effect of three types of obstacles (triangular, leaf-shaped, rectangular) on the thermal performance of a flat plate SAH. It was deduced that the leaf-shaped obstacles on the absorber plate enhance the efficiency of the SAH more than the other two types. On the other hand, applying matrices causes more turbulence in the air flow leading to the enhancement of heat transfer coefficient.

S. Singh [14] utilized a serpentine matrix with porosity ranging from 85% to 95% in a DPSAH both experimentally and numerically. The results indicated that the packed bed wire mesh with a porosity of 93% can increase the thermal and the thermohydraulic efficiencies of the SAH about 18% and 17%, respectively. Dhiman et al. [15] contributed to enhancement of the efficacy of the absorber plate. They showed that a paraffin-based absorber plate leads to reduction of the peak to peak amplitudes of the outlet fluid temperature to 5 K. Furthermore, many researches have shown that using V-shape corrugated absorber plates leads to reducing the peak to peak amplitudes of the outlet fluid temperature to 5 K. Furthermore, many researches have shown that using V-shape corrugated absorber plates leads to increasing the thermal performance of the DPSAHs up to 114% [7,8]. In 2019, M. Salih et al. [9] examined the effects of natural convection, and forced convection cases on the thermal performance of a DPSAH within January to March. The results demonstrated that the highest energy efficiencies of the DPSAH under natural and forced convection conditions are 73.5% and 65%, respectively. M. Ansari and M. Bazargan [10] utilized the genetic algorithm to optimize the outlet air temperature, and the thermal efficiency of an SAH with a ribbed surface. They concluded that the ribs in SAHs enhance the thermal performance up to 9% at low air flow rates. A. Kabeel et al. [11] improved the efficiency of a single pass SAH using fins and a guide blade attachment to the heater entrance for well distributing the air flow. They reached the efficiency of 57% for the proposed SAH containing 8 cm

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development of using recycles in DPSAHs containing a matrix channel. S. Singh and P. Dhiman [16] investigated the efficiency of two types of packed bed solar air heater with recycles. The first type had three air flow channels; the duct beneath the first glass cover, the duct above the absorber plate through a wire mesh region, and the recycle channel beneath the absorber plate. The second type had only two air flow channels; the duct above the absorber plate through a wire mesh region, and the recycle channel beneath the absorber plate. The results indicated that the thermal and thermohydraulic efficiencies of the first type are about 11% higher than those for the second type. H. Yeh and C. Ho [17] asserted that using internal recycle leads to decreasing the driving force (inlet outlet temperature difference). Hence, application of the external recycle was revealed. Not only does it augment the velocity of the fluid in the flow channel, also it does not reduce the temperature difference between the inlet and the outlet fluids.

In this project, four types (A, B, C, and D) of DPSAHs containing three flow channels and two glass covers have been analyzed analytically through MATLAB software. First, this work was validated by Dhiman’s work [15] to show the reliability of the present results. Then, the investigations were extended for various operational conditions. In Ref. [15], it was concluded that using matrix leads to enhancing the thermal and thermohydraulic efficiencies of a DPSAH. In this research, it is shown that at high mass flow rates and reflux ratios using matrix causes a reduction in the thermohydraulic efficiency of a DPSAH, which imposes extra electricity cost for running the fan. The novelty of this work is to introduce new procedures to approach a matrix-based DPSAH performance without using any matrix; hence, a lot of energy and cost will be saved. In addition, the analyses have been conducted for different air mass flow rates, reflux ratios, solar radiation intensities, and matrix porosities to realize how these parameters affect the mechanical characteristics of DPSAHs under similar environmental conditions.

2. Theoretical study

2.1. Description of the four DPSAHs types

All designs of the DPSAHs are comprised of two transparent glass covers, a steel plate as the absorber plate lying under the second glass cover, a packed bed matrix made of steel positioned in the second channel (the channel between the second glass cover and the absorber plate), a gleaming stainless steel plate as the back plate lying under the absorber plate, and a piece of wood used for insulation. Table 1 indicates

<table>
<thead>
<tr>
<th>Components</th>
<th>Length (m)</th>
<th>Width (m)</th>
<th>Thickness (m)</th>
<th>Depth (m)</th>
<th>Transmissivity (τ)</th>
<th>Absorptivity (α)</th>
<th>Emissivity (ε)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Transparent cover</td>
<td>2.2</td>
<td>0.45</td>
<td>0.003</td>
<td>–</td>
<td>0.95</td>
<td>0.05</td>
<td>0.92</td>
</tr>
<tr>
<td>Absorber plate</td>
<td>2.2</td>
<td>0.45</td>
<td>0.003</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>Matrix</td>
<td>2.2</td>
<td>0.45</td>
<td>–</td>
<td>0.06</td>
<td>–</td>
<td>–</td>
<td>0.95</td>
</tr>
<tr>
<td>Back plate</td>
<td>2.2</td>
<td>0.45</td>
<td>0.003</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>0.94</td>
</tr>
<tr>
<td>Insulation</td>
<td>2.2</td>
<td>0.45</td>
<td>0.05</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>Channels 1, 2, and 3</td>
<td>2.2</td>
<td>0.45</td>
<td>–</td>
<td>0.06</td>
<td>–</td>
<td>–</td>
<td>–</td>
</tr>
</tbody>
</table>

Table 1
Thermophysical properties of the DPSAHs’ components.

<table>
<thead>
<tr>
<th>Type of matrix</th>
<th>d_w (m)</th>
<th>P_t (mm)</th>
<th>n</th>
<th>P</th>
</tr>
</thead>
<tbody>
<tr>
<td>M1</td>
<td>0.75</td>
<td>3</td>
<td>30</td>
<td>0.85</td>
</tr>
<tr>
<td>M2</td>
<td>0.65</td>
<td>2.5</td>
<td>22</td>
<td>0.9</td>
</tr>
<tr>
<td>M3</td>
<td>0.4</td>
<td>1</td>
<td>12</td>
<td>0.95</td>
</tr>
</tbody>
</table>

Table 2
The geometric characteristics of the packed bed matrix sorts.

Fig. 1. Physical models of the investigated DPSAH types.
the physical, and the thermal characteristics of the analyzed DPSAHs components. Moreover, the properties related to the matrix types are shown in Table 2.

The four designed DPSAHs are described as follows:

• Type A: Air flows through channels 1 and 2, and the channel 3 is the recycling channel.
• Type B: Air passes through channels 2 and 3, and the channel 1 is the recycling channel.
• Type C: Air passes through channels 1 and 2, channel 3 is the recycling channel, and there exists a packed bed matrix region in channel 2. This type is the same as type A; the only difference is existence of matrix in channel 2.
• Type D: Air passes through channels 2 and 3, channel 1 is the recycling channel, and there exists a packed bed matrix region in channel 2. This type is the same as type B; the only difference is existence of matrix in channel 2.

There exist three airflow channels for every kind of DPSAH, i.e., two channels for passage of air and one channel for the recycling. Fig. 1 illustrates the physical models of different types of DPSAHs explored in the present work.

2.2. Analytical investigations

The following assumptions have been considered to present mathematical models for the DPSAHs:

• The DPSAH is working under a steady state condition
• The air flow is regarded as one-dimensional. Therefore, the hydraulic and thermal developments have been conducted in the x direction
• The amount of porosity was considered in both x and y directions
• The shadow of walls on the absorber plate is negligible
• The sky is considered as a black body with an equivalent temperature
• The thermophysical characteristics of the air flow are calculated in the mean temperature
• There are not any temperature gradients inside the covers, the absorber plate, and the back plate due to uniform exposure of the collector surface to the sun
• Heat losses from the edges of the DPSAH, and the inlet of recycling channels are negligible

The inlet air temperature is assumed as equivalent to the ambient temperature
The air inside the channels is considered as a fluid, which does not absorb the solar radiation

2.2.1. The energy balance equations

The scheme of the DPSAH of type A is indicated in Fig. 2. The energy balance equations for the upper cover, the lower cover, the absorber plate, and the back plate of DPSAH of type A are obtained from Eqs (1, 2, 3, and 4), respectively:

\[
\begin{align*}
\dot{m}w_u(T_u - T_a) + \dot{m}_{gl}g(T_{gl} - T_{gl}) + \dot{m}_{p-g}(T_{p-g} - T_{p-g}) + \dot{m}_{p-r}(T_{p-r} - T_{p-r}) + h_{p-b}(T_{p-b} - T_{b}) + h_{p-b}(T_{p-b} - T_{b}) + h_{p-\tau}(T_{p-\tau} - T_{\tau,ave}) &= \dot{I}_{r,gl} \text{(1)} \\
\dot{m}_{gl}g(T_{gl} - T_{gl}) + h_{g-gl}(T_{gl} - T_{gl}) + h_{g-gl}(T_{gl} - T_{gl}) + h_{g-gl}(T_{gl} - T_{gl}) + h_{g-gl}(T_{gl} - T_{gl}) &= \dot{I}_{r,gl} \text{(2)} \\
\dot{m}_{p-g}(T_{p-g} - T_{gl}) + h_{p-g}(T_{p-g} - T_{gl}) + h_{p-g}(T_{p-g} - T_{gl}) + h_{p-g}(T_{p-g} - T_{gl}) &= \dot{I}_{r,gl} \text{(3)} \\
\dot{m}_{p-r}(T_{p-r} - T_{p}) + h_{p-r}(T_{p-r} - T_{p}) + h_{p-r}(T_{p-r} - T_{p}) + h_{p-r}(T_{p-r} - T_{p}) &= \dot{I}_{r,gl} \text{(4)}
\end{align*}
\]

where \(\dot{m}_w\), \(\dot{m}_{gl}\), \(\dot{m}_{p-g}\), \(\dot{m}_{p-r}\), \(h_{p-b}\), \(h_{p-g}\), \(h_{p-r}\), \(h_{g-gl}\), \( \dot{I}_{r,gl}\), and \(U_b(T_b - T_i)\) are the convective heat transfer between the upper cover and the environment, the radiative heat transfer between the lower cover and the upper cover, the convective heat transfer between the upper cover and the air flowing inside channel 1, the absorbed radiation through the upper glass, the absorbed radiation through the absorber plate, and the heat loss from the back plate to the surroundings. The other terms can be interpreted similarly. Furthermore, if we consider a separate control volume within channel 1, the energy balance for the fluid flow in channel 1 is obtained as:

\[
\frac{\dot{m}_c}{w} \frac{dT_{1(i)}}{dx} = h_{g-gl}(T_{p-g} - T_{gl}) + h_{g-gl}(T_{gl} - T_{gl(1)})
\]
\[ \dot{m}_c \rho c_w \frac{dT}{df} f_1(x) \text{dx} \] is the heat transfer rate to the air flow passing through channel 1 within the specific control volume. In the similar approach, considering separate control volumes for the air flowing through channels 2 and 3, leads to acquiring the following equations:

\[ \dot{m}G \rho c_w \frac{dT}{df} f_2(x) \text{dx} = h_{gl} c_u f_1 T_{gu} f_1(x) + h_{gl} c_p f_3 T_{gl} f_2(x) \] (6)

\[ \dot{m}G \rho c_w \frac{dT}{df} f_3(x) \text{dx} = h_{gl} c_p f_2 T_{p} f_3(x) + h_{gl} c_b f_3 T_{gl} f_3(x) \] (7)

Since, the difference between type A and type B is the direction of recycling; the energy balance equations for the aforementioned components of the DPSAH of type B are identical to those for the DPSAH of type A except for the air flowing in channels 1 and 3. The illustration of DPSAH of type B is shown in Fig. 3. The related energy balance equations for these channels are:

\[ \dot{m}G \rho c_w \frac{dT}{df} f_1(x) \text{dx} = h_{gl} c_u f_1 T_{gu} f_1(x) + h_{gl} c_p f_3 T_{gl} f_2(x) \] (8)

\[ \dot{m}_c \rho c_w \frac{dT}{df} f_3(x) \text{dx} = h_{gl} c_p f_2 T_{p} f_3(x) + h_{gl} c_b f_3 T_{gl} f_3(x) \] (9)

As indicated in Fig. 4, concerning the DPSAH of type C, the energy equations for the upper cover, the back plate, and the air flows in channels 1 and 3 are identical to the ones for the DPSAH of type A (Eqs. (1), (4), (5), and (7)). Meanwhile, the energy balance equations for the lower glass cover, the packed bed matrix, the absorber plate, and the air flowing in channel 2 are sequentially as follows:

\[ h_{gl} \rho c_p \frac{dT}{df} f_1(x) = h_{gl} \rho c_p f_1(T_{gl} - T_{gl,avg}) + h_{gl} \rho c_p f_2(T_{gl} - T_{gl,avg}) \] (10)
h_{\text{m-gl}}(T_m - T_p) + h_{\text{m-fl}}A_m(T_m - T_{12,avg}) + h_{\text{m-p}}A_p(T_m - T_p) = A_mk_m(T_m - T_p)  
\text{(11)}

h_{p-m}(T_p - T_m) + h_{p-g}(T_p - T_{12}) + h_{p-s}(T_p - T_s) + h_{p-fl}(T_p - T_{12,avg}) = 0  
\text{(12)}

\frac{\dot{m}(1 + G)c_p}{w} \frac{dT_{12(\omega)}}{dx} = h_{\text{gl-gu}} (T_{12} - T_{12(\omega)}) + h_{\text{m-gu}} (T_m - T_{12(\omega)}) + h_{\text{p-gu}} (T_p - T_{12(\omega)})  
\text{(13)}

Also, the energy equations for all components of the DPSAH of type D are identical to those for type C except for the energy balance equations for air flowing in channels 1 and 3 that are the same as those of type B (see Fig. 5).

### 2.2.2. Radiative heat transfer coefficients

Inasmuch as the whole sheets of DPSAHs are flat, parallel, and close to one another, their shape factors with regard to one another are considered as 1. Hence, the radiative heat transfer coefficients are calculated as [18–20]:

\begin{align}
h_{\text{g-u}} &= \sigma_{\text{em}}(T_g^2 + T_u^2)(T_g + T_u)  
\text{(14)}
\end{align}

\begin{align}
h_{\text{g-p}} &= \frac{\sigma(T_g^2 + T_p^2)(T_g + T_p)}{1 + \frac{1}{\epsilon}}  
\text{(15)}
\end{align}

\begin{align}
h_{\text{g-fl}} &= \frac{\sigma(T_g^2 + T_{fl}^2)(T_g + T_{fl})}{1 + \frac{1}{\epsilon} - 1}  
\text{(16)}
\end{align}

where \( \epsilon \) is the emissivity of surfaces (specified in Table 1), \( T \) is estimated in Kelvin, and \( \sigma \) is Boltzmann constant which is regarded as \( 5.67 \times 10^{-8} \text{ W m}^{-2} \text{K}^{-4} \). Moreover, \( T_s \) is the equivalent sky temperature, which is considered as \( 21^\circ \text{C} \):

\begin{align}
T_s = 0.0552T_i^{0.5}  
\text{(19)}
\end{align}

### 2.2.3. Convective heat transfer coefficients

The convective heat transfer coefficient related to the air flowing above the upper glass cover is obtained from [21]:

\begin{align}
h_w = 5.7 + 3.8V  
\text{(20)}
\end{align}

in which \( V \) is the wind velocity. Furthermore, the airflow at inlet of the smooth channels of DPSAHs is assumed fully developed, from a hydrodynamic viewpoint. Hence, the Nusselt number for the airflow in a smooth channel (without matrix) depending on the flow regime is estimated through following equations [22]:

\begin{align}
\frac{Nu}{(1 + 0.00563RePr[\frac{h}{\nu}])^{0.17}}  
\text{for } Re < 2300. \text{ Laminar flow}  
\text{(21)}
\end{align}

Fig. 5. The schematic and the thermal network of the DPSAH of type D.
\[ \text{Nu} = 0.116(Re^{2/3} - 125)Pr^{4/3} \left[ 1 + \left( \frac{D_h}{L} \right)^{2/3} \right] \left( \frac{\mu}{\mu_w} \right)^{0.14} \text{ for } 2300 < Re < 6000, \text{ Transitional flow} \] (22)

\[ \text{Nu} = 0.116Re^{0.4}Pr^{1/4} \text{ for } Re > 6000, \text{ Turbulent flow} \] (23)

where \( D_h \) is the hydraulic diameter defined as:
\[ D_h = \frac{4A_f}{y} = \frac{4(wd)}{2(w+d)} = \frac{2(wd)}{(w+d)} \] (24)

Moreover, the convective heat transfer coefficients for the airflow in channels 1, 2, and 3 without matrix (types A and B) are, respectively,
\[ h_{c,1}\text{avg} = \frac{Nuk_1}{D_{h1}} \] (25)
\[ h_{c,2}\text{avg} = \frac{Nuk_2}{D_{h2}} \] (26)
\[ h_{c,3}\text{avg} = \frac{Nuk_3}{D_{h3}} \] (27)

where \( k_1 \) is the conductivity of air. On the other hand, the convective heat transfer coefficient between the matrix and the airflow in channel 2 is acquired as [23,24]:
\[ h_{c,m2} = S_{st}G_0C_p \] (28)

in which \( G_0 \) and \( S_t \) are respectively defined as:
\[ G_0 = \frac{\dot{m}}{A_{ch}P} \] (29)
\[ S_{st} = J_{st}Pr^{-2/3} \] (30)

where \( J_{st} \) is obtained from [24]:
\[ J_{st} = 0.647 \left[ \frac{1}{\eta P} \left( \frac{\rho_0}{\rho} \right) \right]^{1.104} Re_m^{-0.55} \] (31)
in which \( P \) is acquired from [24]:
\[ P = \frac{pp_1 d_1 - \left( d_1 \right)^n}{pp_1 d_1} \] (32)

The convective heat transfer coefficient between the airflow in channel 2 with matrix (types B and D) and the lower glass cover is acquired from:
\[ h_{c,2gl} = \frac{Nu_{m2}k_1}{PD_{h2}} \] (33)
in which \( Nu_{m2} \) is the Nusselt number for the porous media calculated from [24]:
\[ Nu_{m2} = 0.2Re_m^{0.4}Pr^{1/3} \] (34)
in which \( Re_m \) is defined as [24]:
\[ Re_m = \frac{4\rho_0 G_0}{\mu} \] (35)

where \( \rho_0 \) depends on the porosity of matrix defined as:
\[ \rho_0 = \frac{Pd_s}{4(1-P)} \] (36)

Meanwhile, the heat transfer coefficient between the airflow in channel 2 containing matrix and the absorber plate is equal to \( h_{c,2gl} \).

### 2.2.4. Solution process and boundary conditions
In this section, the acquired linear differential equations for the airflow in channels have been solved analytically. The boundary conditions for solving the equations related to the DPSAHs of types A and C (downward recycling) are:
\[ T_{1(1)x=0} = T_a \] (37)
\[ T_{2(1)x=0} = \frac{GT_{1(1)x=L} + T_s}{1 + G} \] (38)
\[ T_{3(1)x=0} = T_{2(1)x=L} \] (39)

It is implied from Eq. (37) that the inlet fluid temperature is considered equal to the ambient temperature in the coding process. In addition, the boundary conditions for solving the equations pertaining to the DPSAHs of types B and D (upward recycling) are:
\[ T_{1(2)x=0} = T_{2(2)x=L} \] (40)
\[ T_{2(2)x=0} = \frac{GT_{1(2)x=L} + T_s}{1 + G} \] (41)
\[ T_{3(2)x=0} = T_s \] (42)

Concerning the aforementioned boundary conditions, the mean temperatures in channels 1, 2, and 3 are determined based on the mean value theorem for definite integrals as follows:
\[ T_{\text{avg}} = \frac{1}{L} \int_{0}^{L} T_{i(x)} \, dx \] (43)

by which the amounts of mean air temperatures in channels 1, 2, and 3 in the DPSAH of type A are, respectively:
\[ T_{1,\text{avg}} = \left( 1 - e^{-K_{L1}/K_{0,L}} \right) T_a + \left( 1 - e^{-K_{L1}/K_{0,L}} \right) K_{L1} T_{st} + \left( 1 - e^{-K_{L1}/K_{0,L}} \right) K_{0,L} T_{gl} \] (44)
\[ T_{2,\text{avg}} = Z_sS_sT_{gl} + (Z_sS_s + Z_sR_s)T_p + Z_sR_sT_{st} + Z_sT_s \] (45)
\[ T_{3,\text{avg}} = Z_sS_sT_{gl} + (Z_sS_s + Z_sR_s)T_p + Z_sR_sT_{st} + Z_sT_s \] (46)

Accordingly, the mean air temperatures in channels 1, 2, and 3 in the DPSAH of type B are, respectively:
\[ T_{1,\text{avg}} = \left( 1 - e^{-K_{L1}/K_{0,L}} \right) C_1 + \frac{h_{c,2gl}}{\left( \frac{\alpha_{ch}}{\rho c_p} \right) E_0} T_{gs} + \frac{h_{c,2gl}}{\left( \frac{\alpha_{ch}}{\rho c_p} \right) E_0} T_{gl} \] (47)
\[ T_{2,\text{avg}} = \left( 1 - e^{-K_{L1}/K_{0,L}} \right) C_2 + \frac{h_{c,2gl}}{\left( \frac{\alpha_{ch}}{\rho c_p} \right) E_0} T_{gs} + \frac{h_{c,2gl}}{\left( \frac{\alpha_{ch}}{\rho c_p} \right) E_0} T_{gl} \] (48)
\[ T_{3,\text{avg}} = \left( 1 - e^{-K_{L1}/K_{0,L}} \right) C_3 + \frac{h_{c,2gl}}{\left( \frac{\alpha_{ch}}{\rho c_p} \right) E_0} T_{gs} + \frac{h_{c,2gl}}{\left( \frac{\alpha_{ch}}{\rho c_p} \right) E_0} T_{gl} \] (49)

Moreover, the acquired mean air temperatures in channels 1, 2, and 3 in the DPSAH of type C are, respectively:
\[ T_{1,\text{avg}} = \left( 1 - e^{-K_{L1}/K_{0,L}} \right) C_1 + \frac{h_{c,2gl}}{\left( \frac{\alpha_{ch}}{\rho c_p} \right) K_0} T_{gs} + \frac{h_{c,2gl}}{\left( \frac{\alpha_{ch}}{\rho c_p} \right) K_0} T_{gl} \] (50)
The proposed DPSAHs are determined based on Eqs. (58) and (59), respectively.

In addition, this parameter for the DPSAHs of types B and D is estimated by [19]:

$$\eta = \frac{Q_f}{A_{\text{fan}}}$$  \hspace{1cm} (58)

$$\eta_{\text{low}} = \frac{Q_f - P_{\text{fan}}}{A_{\text{fan}}}$$  \hspace{1cm} (59)

Subsequently, the thermal and the thermohydraulic efficiencies of proposed DPSAHs are determined based on Eqs. (58) and (59), respectively [19]:

$$\eta = \frac{Q_f}{A_{\text{fan}}}$$  \hspace{1cm} (58)

$$\eta_{\text{low}} = \frac{Q_f - P_{\text{fan}}}{A_{\text{fan}}}$$  \hspace{1cm} (59)

In Eq. (59), $P_f$ is the fan power defined as [19]:

$$P_{\text{fan}} = \frac{\rho \dot{V}^2}{2} \frac{L}{n}$$  \hspace{1cm} (60)

in which $\dot{V}$ and $\eta_{\text{low}}$ are considered to be 0.7 and 0.9, respectively [25]. Furthermore, $P_{\text{low}}$ is the air pumping power calculated for the DPSAHs of types A and B as [19]:

$$P_{\text{low}} = \frac{\dot{m}(1 + G)(\Delta P_{\text{d}}) + \dot{m}G(\Delta P_{\text{d}}) + \dot{m}(\Delta P_{\text{d}})}{\rho}$$  \hspace{1cm} (61)

and as to the DPSAHs of types C and D [19]:

$$P_{\text{low}} = \frac{\dot{m}(1 + G)(\Delta P_{\text{d}}) + \dot{m}G(\Delta P_{\text{d}}) + \dot{m}(\Delta P_{\text{d}})}{\rho}$$  \hspace{1cm} (62)

in which $\Delta P_{\text{d}}$ and $\Delta P_{\text{d}}$ are the pressure drops in the packed duct, and the smooth duct, respectively obtained from [26]:

$$\Delta P_{\text{d}} = \frac{f_m \rho \dot{V}^2}{2} \frac{L}{n}$$  \hspace{1cm} (63)

$$\Delta P_{\text{d}} = \frac{2f \rho \dot{V}^2}{D_b} (j = 1, 2, 3)$$  \hspace{1cm} (64)

where $f_m$ and $f$ are the friction factors of the packed bed channel, and the smooth channel, respectively. Moreover, the subscript $j$ in $\dot{Q}_j$ refers to the number of channels, through which the air is passing. The parameters $f_m$ and $f$ are respectively defined as [26]:

$$f_m = 2.484 \left( \frac{1}{D_b} \right)^{0.8} Re_m^{0.44}$$  \hspace{1cm} (65)

$$f = 0.059Re^{0.2}$$  \hspace{1cm} (66)

As an example, for acquiring the temperatures of different components of the DPSAH of type C, Eqs. (1), (4), (10), (11), (12), (50), (51), and (52) form a system of eight equations, eight unknowns, which requires to be solved. On the other hand, in the mentioned system of 8 × 8 equations, the radiative heat transfer coefficients are dependent on the temperature of components which are unknown. Hence, an iterative method by regarding initial guesses for the temperatures has been adopted to solve the system of equations. All types of DPSAHs have been solved similarly. The overall procedure having been followed to solve the equations is explained as under,
1. The constant parameters \( m, n, G, L, w, d_1, d_2, A, \beta, \alpha, I, d_p, T_p, p_1, k, a_{\text{gas}}, a_{\text{air}}, a_{\text{mat}}, \rho_{\text{air}}, \rho_{\text{gas}}, \rho_{\text{mat}}, \) and \( \sigma \) are considered as the input variables given to the software.

2. The parameters \( D_{h1}, D_{h2}, D_{h3}, A_{h1}, A_{h2}, A_{h3}, h_{\text{h}}, G_{\text{h}}, P_{\text{h}}, \) and \( \sigma \) are specified through solving the system of equations as the initial values for step 3.

3. The values of \( T_{\text{sh}}, T_{\text{gl}}, T_{\text{sh}}, T_{\text{gl}}, \) and \( T_{\text{sh}} \) are assigned the same values as \( T_p. \)

4. The assumed temperatures, \( \rho_{\text{h}}, \rho_{\text{h} \text{ gl}}, \rho_{\text{h} \text{ sh}}, \rho_{\text{h} \text{ gl} \text{ sh}}, \rho_{\text{h} \text{ gl} \text{ sh} \text{ gl}}, \rho_{\text{h} \text{ gl} \text{ sh} \text{ gl} \text{ sh}}, \rho_{\text{h} \text{ gl} \text{ sh} \text{ gl} \text{ sh} \text{ gl}}, \) \( h_{\text{h}}, h_{\text{h} \text{ gl} \text{ sh}}, h_{\text{h} \text{ gl} \text{ sh} \text{ gl}}, h_{\text{h} \text{ gl} \text{ sh} \text{ gl} \text{ sh}}, h_{\text{h} \text{ gl} \text{ sh} \text{ gl} \text{ sh} \text{ gl}}, \) \( h_{\text{h} \text{ gl} \text{ sh} \text{ gl} \text{ sh} \text{ gl} \text{ sh}}, \) and \( h_{\text{h} \text{ gl} \text{ sh} \text{ gl} \text{ sh} \text{ gl} \text{ sh} \text{ gl}}, \) \( h_{\text{h} \text{ gl} \text{ sh} \text{ gl} \text{ sh} \text{ gl} \text{ sh} \text{ gl}}, \) and \( h_{\text{h} \text{ gl} \text{ sh} \text{ gl} \text{ sh} \text{ gl} \text{ sh} \text{ gl}} \) are obtained.

5. Initially, \( T_{\text{sh}}, T_{\text{gl}}, T_{\text{sh}}, T_{\text{gl}}, \) and \( T_{\text{sh}} \) are assigned the same values as \( T_0. \)

6. The steps 4 and 5 repeat as long as the difference between the stages \( N \) and \( N-1 \) becomes less than 0.000001.

7. Finally, the amounts of \( T_{10 \times 4 \times 4}, T_{12 \times 4 \times 4}, \) and \( T_{13 \times 4 \times 4}, \) for all types of the DPSAHs are obtained; hence, \( Q_{\text{h}}, \eta_{\text{in}}, \) and \( \eta_{\text{out}} \) can be calculated.

3. Results and discussions

In this section, the validation of this analytical study has been conducted. Then, the impacts of different air mass flow rates, various reflux ratios, and different solar radiation rates on the thermal and thermo-hydraulic efficiencies of the DPSAHs have been illustrated to indicate the cost-effective model. Furthermore, the effect of 85%, 90%, and 95% porosity of matrix were taken into account for evaluation of the matrix-contained DPSAHs performances.

3.1. Validating the present results

So as to verify the present results and show the reliability of them, the obtained thermal efficiencies of the DPSAHs of types C and D against the reflux ratio at the air mass flow rate of 0.01 kg/s and the solar irradiance intensity of 600 W/m² are compared with those given by P. Dhiman and S. Singh [15]. As indicated in Fig. 6, the maximum errors for the types C and D are 3% and 4.5%, respectively.

3.2. Effects of recycling, mass flow rate, and matrix on components temperatures

Table 3 indicates the increase in the temperatures of all components of the DPSAHs of different types. The porosity is regarded as 0.95. As shown in Table 3, the downstream recycling (types A and C) gives a more air temperature rise in the outlet of channel 2 in comparison with the upstream recycling (types B and D). This comparison also indicates that the outlet air temperatures from the channel 3 in the types B and D increased more than those for the channel 1 in types A and C. Comparing the air temperature difference in channel 2 for all types of the DPSAHs results in the fact that this parameter for the DPSAHs of types C and D (including matrix) are higher than that for the DPSAHs of types A and B (without matrix) demonstrating better performances of the DPSAHs.
with matrix. In fact, the matrix leads to increasing the heat transfer area, and turbulence of the air flow. Hence, the matrix leads to an enhancement of the convective heat transfer coefficient; thereby, the air temperature in a channel containing the matrix increases more than that of a non-matrix channel. However, the extra cost of fan power due to a high pressure drop ought to be considered.

Moreover, in the DPSAHs of types A and C regarding a definite air mass flow rate, an increase in the reflux ratio leads to a reduction of the inlet–outlet air temperature difference in the channel 1 and a rise in that of the channel 2. This is because increasing the reflux ratio causes more heated mass flow rate to enter the channel 2 after recycling; hence, the inlet–outlet air temperature difference in channel 2 increases. The highest air temperature rise in the channel 2 occurs at the reflux ratio of 1. In addition, an increase in the air mass flow rate at a definite reflux ratio brings about the reductions in the temperature difference in the channel 1 and 2 because increasing the mass flow rate leads to losing the opportunity for the heat transfer to the fluid. Accordingly, in the DPSAHs of types B and D regarding a definite air mass flow rate, a rise in the reflux ratio leads to reducing the inlet–outlet air temperature difference in the channel 3, and an increase in that of the channel 2. Similar to the previous discussion, the highest fluid temperature rise in the channel 2 occurs at the reflux ratio of 1, demonstrating efficacy of enhancing the reflux ratio than using the matrix to remove the extra cost of electricity due to the pressure drop. Additionally, increasing the air mass flow rate at a definite reflux ratio leads to the reductions in the temperature difference in the channel 2 and 3. Nonetheless, since increasing the fluid mass flow rate, for the most part, enhances the efficiency of solar collectors, the impact of increasing the air mass flow rate on the DPSAH performance must be evaluated.

Fig. 7 shows the air temperature increase in channel 2 for all types of the DPSAHs against the length of the DPSAH at 0.01 kg/s mass flow rate, and the reflux ratio of 0.5. In fact, there exists a certain inlet–outlet air temperature difference in channel 2 because the recycled air and inlet air flows were mixed with each other at \( L = 0 \). As it is indicated, the outlet air temperature from channel 2 in the type C (with porosity of 95%) is higher than that of the other types. The outlet fluid temperature from channel 2 in the type C is 2.69 °C, 8.31 °C, and 3.11 °C more than that of the types A, B, and D, respectively. Moreover, according to Eq. (41), if the collector length is considered as zero, the inlet–outlet temperature difference in channel 2 will also be zero. Furthermore, Fig. 8 shows the air temperature rise in channel 1 for the DPSAHs of types A and C, and the air temperature rise in channel 3 for the DPSAHs of types B and D for different mass flow rates and reflux ratios plotted against the length of collector. As it is illustrated in Fig. 8, the rise in the outlet air temperature from channel 3 for the DPSAH of type B is more than that of the other types. At 0.015 mass flow rate and reflux ratio of 0.5, the outlet fluid temperature from channel 3 for the type B is 3.27 °C more than that of the type D, 4.5 °C more than that from channel 1 of the type C, and 5.36 °C more than that from channel 1 of the type A.

### 3.3. Thermal and thermohydraulic investigations

In this part, the useful heat gain, the pressure drop, the thermal and thermohydraulic efficiencies of DPSAHs at different mass flow rates and reflux ratios are analytically evaluated. Moreover, the effect of the porosity of matrix on the thermal and thermohydraulic efficiencies of the DPSAHs are investigated.

#### 3.3.1. The useful heat gain of DPSAHs

The useful heat gains of all types of the DPSAHs with regard to the different reflux ratios and various air mass flow rates are indicated in Fig. 9. As shown in this figure, increasing the reflux ratio and the mass flow rate leads to enhancing the useful heat gain of collector. It is indicated that the useful heat gain of type C (with porosity of 95%) is
more than that of the other types. On average, at each reflux ratio and at 0.01 kg/s mass flow rate, the useful heat gain of type C is respectively 10 W, 40 W, and 60 W more than that of type D (with porosity of 95%), type B, and type A. The main point behind this phenomenon is that without a matrix it is the temperature rise in the air passing through the channel 3, which plays a crucial role in specifying the performance of DPSAH because temperatures of the absorber plate and the back plate are higher than those of the upper and the lower glasses implying the better performance of type B compared to type A. On the other hand, as indicated in Fig. 9, as the air mass flow rates increases, the useful heat gain of type B approaches the useful heat gain of the identical DPSAH containing the matrix (type D). It can be inferred that in the case of an upward recycling pattern, enhancement of the air mass flow rate can compensate for using the matrix. Table 3 demonstrates that the temperatures of the matrices

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**Fig. 9.** (a) Variation of useful heat gain of the DPSAH against the reflux ratio, (b) Changing useful heat gain of the DPSAH against the air mass flow rate.

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**Fig. 10.** Impact of the solar irradiance intensity variation on the useful heat gains of the DPSAHs of all models.
(\(T_m\)) are higher than those of other components, such as the absorber plate and the back plate in both the types C and D. Hence, it can be concluded that when matrix is used, it is the air temperature rise in channel 2 denoting the efficacy of the DPSAH; therefore, the type C gives better performance in comparison with the type D. All the related information can be gathered from the temperatures report in Table 3.

Moreover, the effect of the solar irradiance intensity on the useful heat gain of the DPSAHs at mass flow rate of 0.01 kg/s is displayed in Fig. 10. It is indicated that increasing the solar radiation intensity brings about the higher useful heat gain for all types.

3.3.2. Thermal efficiency analysis

Fig. 11 (a), (b), and (c) illustrate the thermal efficiencies of DPSAHs against variation of the reflux ratio at mass flow rates of 0.01, 0.015, and 0.025 kg/s. It is indicated that increasing the reflux ratio leads to the enhancement of thermal efficiencies of all types of the DPSAHs. The highest thermal efficiency of 78.98% is assigned to the type C, the reason for which is the existence of matrix as previously explained. Furthermore, the thermal efficiencies of the DPSAHs against variation of the mass flow rate at reflux ratios of 0.1, 0.5, and 1 are displayed in Fig. 11 (d), (e), and (f). It is shown that increasing the mass flow rate also leads...
to the enhancement of thermal efficiencies of all types of the DPSAHs. Overall, the thermal efficiency of type C is respectively 5%, 7%, and 10% higher than that of the types D, B, and A. Another intriguing point of this project is that if a non-matrix DPSAH is preferable, the upstream recycling would be the best choice of the designing. In other words, it can be deduced that the energy efficiency of type B can approach the energy efficiency of type D through enhancing the mass flow rate, and the reflux ratio. In this efficient way, the less electrical power is required for the fan.

3.3.3. Consideration of the pressure drop

The overall pressure drop along the channels of each DPSAH was theoretically calculated through Eqs. (63–66). As shown in Fig. 12 (a) and (b), the pressure drop along the channels increases, and raising the reflux ratio leads to an increase in the pressure drop. Moreover, the maximum value of pressure drop for the types A and B is 3.6 Pa; whereas this figure for the types C and D is 123 Pa. Thereby, the pressure drop for the types C and D is, on average, 40 times higher than that for the types A and B. This difference is due to existence of the matrix in the types C and D leading to a huge pressure drop along channel 2. A smooth channel creates a small drop in the pressure. In addition, Fig. 12 (c) and (d) display the values of pressure drops in all types of the DPSAHs at mass flow rates of 0.01, 0.015, and 0.025 kg/s considering the reflux ratio of 1. It is similarly indicated that increasing the mass flow rate causes more pressure drop along the channels. Taking the aforementioned discussions into account, it is true that using the matrix can improve the DPSAH performance to some amounts; however, the concepts of sustainability and green energy become undermined through coming across a remarkably extra electricity cost to overcome the pressure drop.

3.3.4. Thermohydraulic efficiency analysis

In order to make a thorough comparison among the thermal, and thermohydraulic efficiencies of DPSAHs, both types of the efficiencies have been illustrated in one graph. Fig. 13 (a), (b), (c), and (d) show the thermal and thermohydraulic efficiencies of types A, B, C, and D, successively. As it can be noticed, the both efficiencies for the types A and B are roughly the same because as to these types the channels are smooth; thereby the pressure drop is negligible. However, in the types C and D the pressure drop is significant due to the matrix. In each of these types, the thermohydraulic efficiency is about 2% less than the thermal efficiency. Additionally, there exist the jumps in Fig. 13 (a), (c), and (d) due to the turbulence of airflow.

Furthermore, Table 4 compares the present research findings with those of the previous similar works. It is indicated that the obtained energy efficiency of type B is roughly less than that of type C (about 7%); nonetheless, it presents the performance value within the previous investigated matrix-based SAHs. The great advantage is to supply the required hot air demand in a costly effective manner.

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**Fig. 12.** Investigation of the amounts of overall pressure drops in the DPSAHs along the channels, (a) for the types A and B at the 0.025 kg/s mass flow rate and reflux ratios of 0.1, 0.5 and 1, (b) regarding the types C and D at the 0.025 kg/s mass flow rate and reflux ratios of 0.1, 0.5 and 1, (c) as to the types A and B at the reflux ratio of 1 and mass flow rates of 0.01, 0.015, and 0.025 kg/s, (d) as to the types C and D at the reflux ratio of 1 and mass flow rates of 0.01, 0.015, and 0.025 kg/s.
3.3.5. On the role of porosity

It is shown that enhancing the porosity causes better performance for the DPSAHs. The thermal efficiency of a DPSAH with matrix, whose porosity is 95%, is about 3% higher than that of the DPSAH containing the matrix with porosity of 85% (see Fig. 14 (a) and (b)). Furthermore, the impact of different porosities of the matrix in channel 2 on the thermohydraulic efficiency of the DPSAH is investigated. Fig. 14 (c) demonstrates that at a 0.01 mass flow rate, increasing the porosity and the reflux ratio lead to enhancing the thermohydraulic efficiency of DPSAH. Also, Fig. 14 (d) indicates that at mass flow rate of 0.025, a rise in the reflux ratio up to 0.8 shows the same trend; however, raising the reflux ratio from 0.8 to 1 leads to a reduction in the thermohydraulic efficiency of the DPSAH. It is because increasing the mass flow rate and the reflux ratio cause the pressure drop enhancement along the channels. The interesting point is that the pressure drop for porosity of 95% is higher than that for the porosities of 85% and 90%; nevertheless, the porosity of 95% presents better thermohydraulic efficiency of the DPSAH. The reason for this result is that increasing the porosity causes more heat transfer to the air passing through channel 2 leading to enhancement of the useful heat gain of DPSAH. This rise in the useful heat gain is more than the heat loss due to the pressure drop, which results in giving higher thermohydraulic efficiency for the collector.

4. Conclusion

In the present study, the performance of a DPSAH was improved through the upstream recycling, the downstream recycling, and the matrix. In addition, the impacts of some performance parameters; such as changing the air mass flow rate, altering the reflux ratio, changing the solar radiation intensity on the collector surface, variation of the matrix porosity, and the pressure drop due to the matrix in the second channel on the DPSAH thermal and thermohydraulic performances were investigated. The main findings of the present research were as follows:

- Generically, escalation of the air mass flow rate and the reflux ratio leads to a rise in the thermohydraulic efficiency of a DPSAH. However, as to types C and D, increasing the reflux ratio up to 0.8 leads to enhancing the thermohydraulic performance, and any subsequent rise in the reflux ratio causes a reduction in the thermohydraulic efficiency of the DPSAH due to the high amount of pressure drop caused by the matrix.
- Enhancing the air mass flow rate and the reflux ratio causes more pressure drop. The highest pressure drops at the mass flow rate of 0.025 kg/s and the reflux ratio of 0.8 to 1 is 3.6 Pa for the types A and B (without matrix), and is 123 Pa for the types C and D (with matrix).
- From an economical viewpoint, a non-matrix upward recycling DPSAH can present the same efficiency of a similar matrix-based DPSAH operating at high mass flow rates and reflux ratio of 0.8 to 1. In this way, the trouble of matrix preparation, the cost of matrix, and the additional fan power cost will be eliminated.
- If cost is not of a significant concern, increasing the air mass flow rate, the reflux ratio, the porosity, and the solar irradiance intensity cause higher useful heat gain, and better performance of the DPSAH. On average, the thermal efficiency of type C is 5% higher than that of the type D, 7% higher than that of the type B, and 10% higher than that of the type A.

![Fig. 13. The thermal and thermohydraulic efficiencies of the DPSAHs against the reflux ratio variation at 0.025 kg/s mass flow rate, (a) for type A, (b) for type B, (c) for type C, (d) for type D.](image-url)
Table 4
Making a comparison between the present work and previous similar researches.

<table>
<thead>
<tr>
<th>No.</th>
<th>Authors</th>
<th>Investigation procedure</th>
<th>Work description</th>
<th>Maximum efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Chouksey and Sharma [27]</td>
<td>Theoretical</td>
<td>Employed a blackened matrix in an SAH at 0.035 kg/s mass flow rate and under 1000 W/m² solar radiation</td>
<td>0.52</td>
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<tr>
<td>2.</td>
<td>Verma and Varshney [28]</td>
<td>Theoretical</td>
<td>Investigating a packed bed DPSAH at 0.02 kg/s mass flow rate and regarding 950 W/m² solar radiation intensity</td>
<td>0.66</td>
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<tr>
<td>3.</td>
<td>Dhiman and Singh</td>
<td>Analytical</td>
<td>Using a matrix in a DPSAH at 0.025 kg/s mass flow rate and regarding 600 W/m² solar radiation</td>
<td>0.74</td>
</tr>
<tr>
<td>4.</td>
<td>Rajarajeswari et al. [29]</td>
<td>Numerical and Experimental</td>
<td>Utilized a 0.84 porosity matrix in an SAH at 0.03 airflow rate and 800 W/m² average solar radiation</td>
<td>0.74</td>
</tr>
<tr>
<td>5.</td>
<td>Ghritlahre and Prasad [30]</td>
<td>Analytical</td>
<td>The performance of an SAH with an indirect airflow throughout the matrix was modelled regarding 0.022 kg/s mass flow rate and 963 W/m² solar insolation</td>
<td>0.64</td>
</tr>
<tr>
<td>6.</td>
<td>Singh et al. [31]</td>
<td>Experimental</td>
<td>Considering a matrix between the absorber plate and the back plate in a DPSAH at 0.023 kg/s mass flow rate and under 800 W/m² average solar radiation</td>
<td>0.84</td>
</tr>
<tr>
<td>7.</td>
<td>S. Singh [14]</td>
<td>Numerical and experimental</td>
<td>Applied a 0.93 porosity wavy matrix with a 7.5 cm wavelength to a DPSAH at 0.03 kg/s airflow rate and under 900 W/m² solar radiation</td>
<td>0.8</td>
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<tr>
<td>8.</td>
<td>Present research</td>
<td>Analytical</td>
<td>Investigating the performance of a DPSAH regarding a downward recycle containing a matrix with 0.95 porosity at 0.025 kg/s flow rate and under 600 W/m² solar radiation</td>
<td>0.79</td>
</tr>
<tr>
<td>9.</td>
<td>Present research</td>
<td>Analytical</td>
<td>Examining the efficacy of a DPSAH possessing an upward recycling pattern without using any matrix at 0.025 kg/s mass flow rate and under 600 W/m² solar radiation intensity</td>
<td>0.72</td>
</tr>
</tbody>
</table>

Fig. 14. (a) The thermal efficiencies of the DPSAHs containing the matrix with porosities of 0.85, 0.9 and 0.95 with regard to the reflux ratio at 0.01 kg/s mass flow rate, (b) the thermal efficiencies of the DPSAHs containing the matrix with porosities of 0.85, 0.9 and 0.95 with regard to the air mass flow rate at the reflux ratio of 0.5, (c) the thermohydraulic efficiencies of the DPSAHs containing the matrix with porosities of 0.85, 0.9 and 0.95 with regard to the reflux ratio at 0.01 kg/s mass flow rate, (d) the thermohydraulic efficiencies of the DPSAHs containing the matrix with porosities of 0.85, 0.9 and 0.95 with regard to the reflux ratio at 0.025 kg/s mass flow rate.
The thermal and thermohydraulic efficiencies of types A and B are approximately equal to each other; however, the thermohydraulic efficiencies of the types C and D are roughly 2% lower than their thermal efficiencies due to the pressure drop created by means of the matrix in the second channel.

The downstream recycling leads to more air temperature rise in the second channel of a DPSAH. Thereby, the temperature rise in the outlet fluid from the second channels of types A and C is more than that of the types B and D. Moreover, the matrix causes more air temperature rise. The outlet air temperature rise from channel 2 of the type C at the mass flow rate of 0.01 kg/s and the reflux ratio of 0.5, is 2.69 °C more than that of the type A, 3.11 °C more than that of the type D, and 8.31 °C more than that of the type B.

Raising the reflux ratio at a certain mass flow rate, leads to reducing the inlet-outlet air temperature difference in channel 1 for the types A and C, and in channel 3 for the types B and D. Additionally, increasing the air mass flow rate at a certain reflux ratio resulted in the reduction of the inlet-outlet air temperature difference in all channels for all types.

Future studies can be focused on transient state evaluation of the present work. Moreover, the effect of considering the matrix in channels 1 and 3 on the thermal characteristics of DPSAH can be regarded. Furthermore, the influence of vacuum between the upper and the lower glass covers on the DPSAH performance is worth being investigated.

CRediT authorship contribution statement

Ali Ahmadkhani: Conceptualization, Methodology, Resources, Software. Gholamabass Sadeghi: Writing - original draft, Results interpretation, Validation. Habibollah Safarzadeh: Supervision.

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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