Hydrothermal performance of humid air flow in a rectangular solar air heater equipped with V-shaped ribs

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Abstract
In this numerical work, heat transfer characteristics of humid air in a rectangular channel as solar air heater equipped with V-shaped ribs has been investigated. The impacts of geometrical parameters of proposed ribs such as height, pitch, and angle relative to air flow direction, as well as relative humidity and Reynolds number of inlet humid air on thermal performance of proposed solar air heater have been evaluated. The numerical results have been achieved using commercial Computational Fluid Dynamics (CFD) code, ANSYS FLUENT 18.2, which works based on finite volume method. Outcomes shows that, over a wide range of operating conditions, utilizing ribs cause thermal boundary layer to separate and reattach, and eventually enhance heat transfer rate. Increasing the rib angle produces stronger secondary flows, leading to higher local velocities and greater heat transfer rate. Moreover, increasing the relative humidity of inlet air from 0% to 50% leads to improve heat transfer by up to 6%. As Reynolds number increases from 4000 to 12,000, average Nusselt number rises by 7.3% and 31.4% for rib pitches of 6 and 16 mm, respectively. The results demonstrate how thermal performance of a solar air heater can be improved by utilizing V-shaped ribs.

KEYWORDS
computational fluid dynamics (CFD), numerical method, renewable energy, rib, solar air heater (SAH), solar energy

1 INTRODUCTION
Renewable energy has been attracting growing attention owing to the increasing threat of climate change and environmental pollution. Various forms of renewable energy are available, such as wind, tidal, solar, and geothermal. Most of these, however, require relatively advanced technology and can be costly to implement at large scale—a key concern of developing countries. The only exception to this is perhaps solar energy, which can be reliably extracted at large scale using simple technology and is also readily available in many parts of the world. This form of energy can be used in various ways, such as to generate electricity via heat plants and...
photovoltaic cells, to directly heat water or air, and to provide passive ventilation.

Solar heaters constitute one of the simplest ways to harness solar energy. They involve using solar energy to heat a fluid (water or air) via a vacuum tube or a collector. Flat-plate solar collectors are one of the simplest devices available for heating such a fluid; owing to their simplicity, they have been seen widespread use around the world. The main advantages of flat-plate solar collectors are multifold: their low cost relative to other collectors, their direct and scattered radiation absorption of the sun, and their ability to operate without solar tracking. Typically, this type of collector contains a dark flat-plate solar absorber, a transparent covering allowing the passage of solar radiation, a fluid (e.g., air, antifreeze, or water) for transferring heat from the absorber, and a thermal insulator on the backside. Most research in this field has focused on increasing the heat transfer and thermal efficiency of flat-plate solar collectors using turbulators and/or reflective surfaces. Some of these works are reviewed below.

Kumar et al. numerically explored the effect of oval-shaped roughness elements in a solar air heater (SAH), with a focus on the roughness height and pitch on the heat transfer and pressure drop. Through the use of finite element simulations, Thakur et al. examined the effect of square-sectioned ribs in different V-shaped and zigzag-shaped configurations. Skullong et al. experimentally and numerically investigated the use of winglet turbulators for Reynolds numbers of 4100 – 25,000. Gawande et al. investigated the effect of inverted L-shaped bands on the heat transfer characteristics of a SAH for Reynolds numbers of 3800 – 18,000 and for a constant heat flux of 1000 W/m² introduced to the absorber. Sharma and Kalamkar numerically and experimentally examined the effect of thin bands in a solar heater at Reynolds numbers of 4000 – 16,000. Application of different band arrangements was associated with a 3%–7% improvement in heat transfer. Manjunath et al. explored the use of spherical turbulators in solar absorber plates, and found that the use of dimples on the absorber plate could increase heat transfer by 24%. Heydari and Mesgarpour numerically and experimentally examined a solar collectors equipped with helical flow paths, and found that the heat transfer rose by up to 15% relative to that of collectors with only straight flow paths. Pramanik et al. evaluated the performance of double-pass solar air collectors equipped with fins, and found up to a 69% increase in heat transfer performance. Singh investigated the use of arched absorber plates with turbulator barriers. In that study, a constant heat flux of 500 W/m² was introduced and it was observed that, for Reynolds numbers of 3000 – 14,000, the use of the plates produced a minor increase in heat transfer. In numerical simulations and experiments, Rajarajeswari et al. explored the use of porous media to create turbulence, which led to heat transfer improvement.

Saurav and Bartaria studied the effect of the pitch dimension in SAHs equipped with triangular roughness elements. Tsilingiris investigated the use of square roughness elements, with a focus on the effect of their pitch and the Reynolds number. Perwez and Kumar examined the use of dimples in a flat-plate collector, and found that the dimples caused a 151%–164% increase in heat transfer compared with a similar collector without dimples. Komolafe et al. investigated the performance of a flat-plate collector using numerical and experimental methods. The thermal efficiency was found to vary between 14% and 56.5% across different hours of the day. Sivakumar et al. experimentally investigated forced convection in a flat-plate collector equipped with pin fins. The use of pin fins resulted in a 3%–12% increase in thermal efficiency. Shetty et al. investigated the performance of perforated spherical flat-plate solar collectors, with a focus on the effect of the number of perforations in the plate and their diameters. Jousyabi and Lundström examined the effect of a porous thin film over the absorber surface on the thermal performance of a flat-plate solar collector. They found that use of the porous thin film caused a five-fold increase in heat transfer and a two-fold increase in pressure drop.

Yadav and Sain used numerical simulations to examine the use of a fluid jet on the performance of a flat-plate solar collector. Kumar and Layek assessed the effect of winglet turbulators on the thermal performance of a flat-plate collector, and presented a relation for the performance of the collector based on the flow and winglet properties. Baiissi et al. experimentally examined the effect of delta-shaped turbulators on the thermal performance of flat-plate collectors. Ghritlahre, Sahu installed arch-shaped roughness elements in flat-plate collectors, and found that they increased heat transfer by 10% relative to a baseline collector. Olfian et al. evaluated the hydrothermal behavior of a SAH equipped with two various kinds of baffles including angled rectangular baffles and angled V-shaped baffles numerically. The obtained numerical results depicted that in the rectangular model, the pressure drop and average Nusselt number rose 316.67% and 148.15%, respectively. Moreover, in V-shaped angled baffles, the thermal efficiency of β = 90°, 60°, 45°, and 30° were 27%, 18%, and 13% more than no baffle case at Re = 2000, correspondingly. Kazemi Moghadam et al. proposed an innovative arc-shaped ribs for the SAH. The heat transfer process in the proposed SAH was estimated numerically for Re = 6000–12,000. The effects of the rib cross-section shape, the inter-rib distance, the aspect ratio, and the pitch of the ribs on the thermal efficiency of SAH were investigated. The obtained
numerical outcomes showed that the ribs with quadrangular cross-section reveals the highest coefficient of performance.

Maithani et al.\textsuperscript{30} presented an innovative inclined air jet-impingement SAH absorber plate. The impacts of inclined jets on absorber plate for different geometrical factors were investigated by performing experiments. The obtained experimental results depicted that by employing the proposed SAH system, the maximum exergy efficiency of 2.925 could be achieved. Sharma et al.\textsuperscript{31} evaluated numerically the thermal performance of the SAH considering six various geometries of baffles. The obtained numerical outcome depicted that baffles position and shape have considerable impact on the thermal performance of the SAH. The maximum thermo-hydraulic performance by 2.05 belongs to the case with a sine wave-shaped baffles at $Re = 15,000$. Maithani et al.\textsuperscript{32} investigated numerically the heat transfer process of nanofluid flow in a ribbed SAH. The obtained results displayed that the maximum average Nusselt number and friction factor belong to the cases with rib relative height ratio equal to 0.084 and 0.06, respectively. Moreover, the maximum thermo-hydraulic performance belongs to the case with relative slot rib height equal to 0.084.

Kumar and Goel\textsuperscript{33} presented an innovative SAH considering various kinds of rib shapes. The proposed numerical model was verified properly with the experimental results. The obtained numerical outcomes depicted that among the evaluated models, the ribs with square and rectangular displayed the better thermal performance. Moreover, the maximum thermo-hydraulic performance value of 2.68 is achieved in this study. Goel et al.\textsuperscript{34} proposed an innovative kind of SAH equipped with hemispherical dimples as an efficient passive heat transfer augmentation method. The obtained results of experiments and numerical simulations depicted that the maximum enhancement in the heat transfer rate is 5.33 in comparison with the plain SAH. Moreover, the highest thermal performance in the proposed SAH was obtained by 38.42%. Kumar et al.\textsuperscript{35} investigated numerically the heat transfer process in a triangular cross-sectional type of SAH with circular-shaped corners. The obtained numerical outcomes showed that presence of rounded corners leads to higher velocity through the channel in comparison with the channel with sharp corners. The maximum augmentation on average Nusselt number by 191\% was achieved at $Re = 17,500$.

According to the above literature review and also the other numerous works in the field of SAH,\textsuperscript{36–40} it should be noted that the impact of the humidity of the humid air has not been considered and analyzed yet on the thermal performance of a SAH. Moreover, as the above literature review is shown, the use of ribs as artificial roughness to enhance the heat transfer rate in the solar systems such as SAH, solar collectors, and so forth has been the interest of various researchers. However, to date, the effect of environmental conditions, such as the inlet humid air temperature and relative humidity, has yet to be comprehensively explored. Changes in the air humidity can change the thermo-physical properties of the heat transfer fluid (humid air here), thus affecting the thermal performance of the overall SAH. In view of this point, the effects of inlet air humidity and Reynolds number on the heat transfer characteristics of the proposed SAH as well as the geometrical parameters of the V-shaped ribs have been explored here numerically.

\section{GEOMETRY OF THE PROPOSED MODEL}

The base geometry of the proposed model has been chosen based on the work of Thakur et al.\textsuperscript{8} The total length of the channel is $L = 200$ mm, which includes inlet and outlet sections (each 50 mm long) upstream and downstream of the central ribbed section (see Figure 1). Figure 1 shows the geometry of the proposed SAH and the geometrical parameters of the system. Thus, the length of the central ribbed section is 100 mm. The cross-section of the channel is rectangular with dimensions of $W = 50$ mm (width) by $H = 20$ mm (height). In this study, the effects of
geometrical parameters such as; the rib angle ($\alpha$) (see Figure 2), the rib height ($h$) (see Figure 1), and the rib pitch ($p$) (see Figure 1) as well as the Reynolds number and relative humidity of the heat transfer fluid (air) have been evaluated numerically. According to Figure 2 (which is top-view of proposed SAH), it is shown that five various values for the rib angle ($\alpha$) were considered including 0°, 15°, 30°, 45°, and 60°.

3 | BOUNDARY CONDITIONS

The finite volume method (FVM) was used, by utilizing a commercial CFD code (ANSYS FLUENT 18.2), to numerically simulate the heat transfer and fluid flow of humid air over the ribs and through the rectangular channel as proposed SAH. Here, the boundary conditions of the computational domain are described in Figure 3. At the inlet section, the Velocity-Inlet type boundary condition is employed in which a constant velocity condition is imposed, with an inlet temperature of 300 K. A constant heat flux of 1000 W/m$^2$ is imposed on the absorber plate with a no-slip wall boundary condition. At the outlet section, the outlet pressure is set to zero as Pressure-Outlet type boundary condition. The heat transfer fluid is humid air with constant thermo-physical properties which are changed based on the inlet temperature of the inlet air. In other words, the impact of the humidity of the inlet humid air is considered in the thermo-physical properties of the air.

Humid air enters the inlet of the computational domain at a Reynolds number of 8000 and a relative humidity of 50%. The properties of humid air have been previously studied by Tsilingiris. The thermo-physical properties of humid air for relative humidity values ranging from 0% to 100% at working temperatures between 0°C and 100°C can be expressed as follows (Equations 1–4):

$$k_m = SK_0 + SK_1 T + SK_2 T^2 + SK_3 T^3 + SK_4 T^4,$$

$$c_{pm} = SC_0 + SC_1 T + SC_2 T^2 + SC_3 T^3 + SC_4 T^4 + SC_5 T^5,$$

where the numerical constants ($SD_i$, $SV_j$, $SK_k$, and $SC_m$ in which $i = 0–3$, $j = 0–4$, $k = 0–4$, and $m = 0–5$) and coefficients of determination (COD) for the proposed polynomial fit expressions as Equations (1–4) can be used according to the Reference [41].

4 | GOVERNING EQUATIONS

The flow is governed by the standard conservation of mass and momentum equations (Navier–Stokes)\textsuperscript{42,43}:

$$\frac{\partial (\rho \vec{u}_i)}{\partial x_i} = 0,$$

$$\rho \left( \frac{\partial \vec{u}_i}{\partial x_j} + \frac{\partial \vec{u}_j}{\partial x_i} \right) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_i} \left( \mu \frac{\partial \vec{u}_i}{\partial x_i} + \rho \mu_t \vec{u}_i \cdot \vec{u}_j' \right).$$

Assuming negligible work done, the energy equation including temperature terms can be written as

$$\frac{\partial (\rho T)}{\partial x_i} + \frac{\partial (\rho u_i T)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \frac{k}{c_p} \frac{\partial (T)}{\partial x_i} \right).$$

Turbulence is modeled using the standard k-ε two-equation model\textsuperscript{44–46}:

$$\nabla \cdot (\rho \vec{u} k) = \nabla \cdot \left( \left( \mu + \frac{\mu_t}{\sigma_k} \right) \nabla k \right) + P_k - \rho \varepsilon,$$

$$\nabla \cdot (\rho \vec{u} \varepsilon) = \nabla \cdot \left( \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \nabla \varepsilon \right) + \frac{\varepsilon}{\kappa} (C_{1\varepsilon} P_k - C_{2\varepsilon} \rho \varepsilon).$$

The average Nusselt number is shown in the following equation:

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure2.png}
\caption{The schematics of the proposed SAH with various values of rib angle ($\alpha$). SAH, solar air heater}
\end{figure}
where $h_{avg}$ is the mean heat transfer coefficient:

$$h_{avg} = \frac{q_{avg}}{T_w - T_{avg}}.$$  

(11)

Mean temperature ($T_{avg}$) inside the SAH duct in Equation (11), is obtained from the following equation:

$$T_{avg} = \frac{\int uTdT}{\int u\,dA}.$$  

(12)

The friction factor is shown in Equation (13):

$$f = \frac{2d_h\Delta P}{\rho v^2L}.$$  

(13)

And finally, thermo-hydraulic performance parameter is defined as follows:

$$\eta = \frac{Nu_f/Nu_s}{(f_f/f_s)^{1/3}}.$$  

(14)

In the above equation, $Nu$ is the average Nusselt number, $f$ is Darcy’s friction factor, and $f$ and $s$ in the subscript indicate frictional and smooth ducts, respectively. Higher thermo-hydraulic performance parameter, $\eta$, represents better system performance. Table 1 summarizes the settings used for the numerical simulations which have been performed using the commercial FVM code, ANSYS Fluent 18.2.

### 5 | MESH INDEPENDENCE AND VALIDATION ANALYSES

The dimensions of the numerical model are chosen based on the work of Thakur et al.8 The SAH is taken to be a flat absorber plate with a rectangular cross-section of 100 mm (width) by 20 mm (height). The mesh is generated with tetrahedral elements. Given the turbulent state of the humid air flow and to improve solution accuracy, seven layers of boundary layer mesh are created with an initial layer thickness of 0.2 mm and a growth rate of 1.2. Overall, the numerical model contains around 2.7 million cells. Figure 4 shows two sample views of the mesh details for a rib angle of 30°.

Figure 5 shows the average Nusselt number and the friction factor as a function of the number of mesh elements. When the number of elements in the mesh reaches 2.7 million, any further increases in mesh density lead to only minor fluctuations (<1%) in both the average Nusselt number and the friction factor, thus indicating mesh independence.

In forced convection, the average Nusselt number is a function of the Reynolds number and Prandtl number. Various relations have previously been proposed to predict the average Nusselt number for turbulent flow with various boundary conditions, such as presented correlations from Gnielinski, Dittus—Boelter, and Sieder—Tate. Owing to its versatility, the Dittus—Boelter correlation (Trinh48) is used here to determine the average Nusselt number of straight tubes in turbulent flow:

$$Nu_D = 0.023Re_D^{4/5}Pr^n,$$  

(15)

where $D$ represents the hydraulic diameter of the flow cross-section, $Pr$ is the Prandtl number of the fluid, and $n$ is the exponent for heating (0.4) or cooling (0.3). This relation

| Selected method | Title | No |
|-----------------|-------|----|
| Steady, segregated | Solver | 1 |
| Second order upwind | Momentum Discretization method | 2 |
| Presto | Pressure | |
| Second order upwind | Turbulence kinetic energy | |
| Second order upwind | Turbulence loss rate | |
| Second order upwind | Energy | |

### TABLE 1 Settings used in the numerical simulations
enjoys good accuracy for relatively small temperature differences between the fluid and the wall. It is applicable to the conditions including $0.6 \leq Pr \leq 160$, $Re_D \geq 10,000$, and $\frac{L}{D} \geq 10$. The validation of $f$ (friction factor) can be done based on the following equation and the Moody chart, as correlation of Petukhov:

$$f = (0.79 \ln(Re_D) - 1.64)^{-2}.$$  \hfill (16)

The results of the validation analysis are presented in Figure 6. Accordingly, it can be seen that two parameters including average Nusselt number and friction factor have been employed for verifying the present numerical work which show the accuracy of the used numerical method. Figure 6A compares the results of the present numerical framework (average Nusselt number parameter) against that of the Dittus–Boelter correlation as well as the experimental results of Thakur et al.\(^8\) The difference between the results of the present framework and those of Dittus–Boelter correlation and the experimental findings is usually less than 5%. As shown in Figure 6A, the maximum error between the results of the present numerical work (average Nusselt number) and the experimental results of the Thakur et al.\(^8\) is by about 6.89% which is acceptable for a validation analysis with the experimental results. Also, according to Figure 6A, the maximum error between the present numerical results and the results of the Dittus–Boelter correlation is by about 1.56% which shows a good agreement. Also, the presented comparison analysis in Figure 6B show good agreement between the present numerical work and the references correlations. As shown in Figure 6B by considering the friction factor, the maximum error between the present numerical results and the results of the Petukhov correlation is by about 3.79% which is acceptable. It is worth mentioning that among the experimental results of the Thakur et al.\(^8\) for friction factor (Figure 6B), one of that at $Re = 6000$ is out of trend because of the possible errors in the experimental setup. Except the result of this Reynolds number ($Re = 6000$), the maximum error between the results of the present numerical work and the experimental results of the Thakur et al.\(^8\) at $Re = 8000$ is by about 3.33% which is acceptable for a validation analysis with the experimental results.

### 6 RESULTS AND DISCUSSION

In the following, the results on the effect of the (i) rib angle, (ii) rib height, (iii) rib pitch, (iv) inlet air relative humidity, and (v) inlet air Reynolds number on the thermal performance of the proposed SAH are presented. Three parameters are selected to analyze the obtained numerical results including the average Nusselt number ratio (for the heat transfer performance), the friction factor (for the pressure drop), and the thermo-hydraulic performance.
6.1 Effect of rib angle

In this section, the influence of the rib angle on the thermal performance of the proposed SAH is studied. Five values of rib angle ($\alpha$) are considered here including $\alpha = 0°$, 15°, 30°, 45°, and 60° and the comparison analysis is presented. A relative humidity of 50% is considered for inlet humid air here. Figure 7 shows the impact of the rib angle on the heat transfer parameters (average Nusselt number ratio), pressure drop (friction factor ratio), and the thermo-hydraulic performance.

According to Figure 7, it can be realized that when the rib angle increases from 0° to 30°, the average Nusselt number ratio increases monotonically. It then remains nearly constant up to a rib angle of 45°, before eventually dropping as the rib angle increases to 60°. The average Nusselt number ratio, however, reaches a maximum at a rib angle of 30°. Meanwhile, the friction factor ratio starts off at a relatively high value, but then it decreases by increasing the rib angle. As a result, the thermo-hydraulic performance curve behaves similarly to the average Nusselt number ratio curve, reaching a maximum at a rib angle of 30°.

Employing ribs on the surface of the collector’s absorber leads to produce weak separation and secondary flows, enhancing the heat transfer rate via a thinner boundary layer. The increase in the heat transfer rate by increasing rib angle can be attributed to the secondary flows formed. The zero-angle rib producing a lower local flow velocity (see Figure 8B). However, by using non-zero angled ribs, the secondary flows formed are accelerated, as can be seen in the streamlines shown in Figure 8A.

Figure 9 shows the velocity magnitude contours at 2 mm away from the absorber plate ($Y = 2$ mm), as well as the temperature contours on the absorber plate itself, at rib angles of $\alpha = 0°$, 15°, 30°, and 60°. As observed in the velocity magnitude contours, employing zero-angled ribs leads to widespread boundary layer separation. However, it reduces the local velocity along the absorber plate. As the rib angle increases, an oblique flow is developed between the ribs toward the outlet, increasing the local velocity along the absorber plate and thus enhancing the heat transfer rate. When the rib angle increases to 60°, the fluid velocity between the ribs augments. Nevertheless, it still causes a dramatic reduction in the velocity at the junction of the ribs as
FIGURE 8  Streamlines for two different rib angles: (A) 60° and (B) 0° at a Reynolds number of 8000 and a relative humidity of 50%.

FIGURE 9  Velocity magnitude and temperature contours within 2 mm from the absorber plate ($Y = 2$ mm) at a Reynolds number of 8000 and a relative humidity of 50%.
well as an increase in the surface temperature in those areas, reducing the temperature difference between the surface and fluid, thus impairing the heat transfer. Hence, the combination of these two effects causes the best heat transfer to occur at a rib angle of around 30° to 45° (among the evaluated rib angles).

6.2 Effect of rib height

In this section, the effect of rib height on the heat transfer characteristics of the proposed SAH is evaluated numerically. Six different values of the rib height are considered ($h = 0.5, 1, 1.5, 2, 3,$ and $5\,\text{mm}$), while the other parameters remain constant: the rib angle is 30°, the rib pitch is 10 mm, the Reynolds number is 8000, and the relative humidity is 50%. Figure 10 shows the average Nusselt number ratio, friction factor ratio, and the thermo-hydraulic performance as a function of the normalized rib height ($h/H$). Accordingly, as

![Figure 10](image1.png)

**Figure 10** Average Nusselt number ratio, friction factor ratio, and the thermo-hydraulic performance versus the normalized rib height at a Reynolds number of 8000 and a relative humidity of 50%.

![Figure 11](image2.png)

**Figure 11** Comparison of the velocity, temperature, and pressure distribution for two different rib heights; (A) $h = 5\,\text{mm}$ and (b) $h = 2\,\text{mm}$ at a Reynolds number of 8000 and a relative humidity of 50%.
the normalized rib height increases, both the average Nusselt number ratio and the friction factor ratio augment monotonically. Meanwhile, the thermo-hydraulic performance first increases when the rib height rises to 2 mm (normalized rib height of 0.1), but then it remains relatively constant with further increases in the rib height. This trend is consistent with the changes in slope observed for the average Nusselt number ratio and the friction factor ratio.

Figure 11 illustrates the contours of velocity magnitude, temperature, and pressure distribution at 25 mm from the channel wall for two representative rib heights: \( h = 2 \) and 5 mm. Accordingly, as the rib height increases, the cross-sectional area of the channel decreases, increasing the local flow velocity owing to conservation of mass; the flow velocity is around 4 m/s for a rib height of 2 mm, but rises to 6 m/s for a rib height of 5 mm. As the flow velocity increases locally in the region above the rib itself, the secondary flow in the inter-rib region is strengthened. Based on Figure 11, it can be seen that this increases the flow velocity adjacent to the absorber plate by approximately 100%, promoting mixing and heat transfer in the fluid adjacent to the absorber plate, thus reducing the surface temperature of the absorber itself. Moreover, the higher velocities for the taller rib \((h = 5 \text{ mm})\) can be seen to reduce both the thickness of the thermal boundary layer and the temperature of the fluid next to the wall, as a result of enhanced heat transfer (a higher Nusselt number).

Figure 10 also suggests that, as the rib height increases, the friction coefficient ratio, which is a proxy for the pressure drop, increases as well, much as the Nusselt number ratio does. According to Figure 11, this increase in pressure drop occurs owing to the drastic reduction in the channel's cross-sectional area and the increased pressure at the inlet. From the pressure distribution shown in Figure 11, it can be seen that as the rib height increases from Figure 11, it can be seen that as the rib height increases from 2 to 5 mm, the inlet pressure nearly doubles.

6.3 Effect of rib pitch

The impact of the rib pitch on the hydrothermal behavior of humid air in the proposed SAH is presented in this section. The considered rib pitches here are \( p = 4, 6, 8, 10, 12, \) and 16 mm. The rib height is 2 mm, the rib angle is 30°, the Reynolds number is 8000, and the relative humidity is 50%. Figure 12 shows the Nusselt number ratio, the friction factor ratio, and thermo-hydraulic performance as a function of the rib pitch normalized by the rib height \((p/h)\). Initially, at low values of the normalized rib pitch, the Nusselt number ratio, friction factor ratio, and thermo-hydraulic performance start off at high values, but they all subsequently decrease as the normalized rib pitch increases. Figure 13 shows the velocity magnitude and temperature distributions for two different values of the rib pitch: \( p = 6 \) and 16 mm.

According to Figure 13, the local flow velocity in the inter-rib region is fairly uniform, suggesting that the rib pitch does not have a marked effect on the flow velocity adjacent to the wall and hence on the heat transfer either. However, Figure 13 shows that the rib pitch has a more noticeable effect on the growth of the thermal boundary layer near the wall. As the rib pitch decreases (i.e., as the number of ribs increases), the thermal boundary layer becomes thinner, enhancing heat transfer. As observed in Figure 12, at lower values of the rib pitch (i.e., with a larger number of ribs), the average Nusselt number increases, which is consistent with the above observations.

6.4 Effect of the inlet air relative humidity

In this section, the impact of the inlet air temperature and relative humidity on the thermal characteristics of the proposed SAH is examined. Two specific values of the relative humidity are considered: 0% (dry air) and 50% (humid air). Figure 14 shows the effect of the relative humidity ratio on the heat transfer parameters as a function of the inlet air temperature. It can be seen that, as the inlet air temperature rises from 300 to 330 K, the thermo-hydraulic performance decreases monotonically. This occurs because the growth in the inlet air temperature cause an increase in the local fluid temperature. Consequently, the absorber surface temperature also rises which leads to
augmentation in the average Nusselt number subsequently. The average Nusselt number also increases with relative humidity (humid air vs. dry air); this increase becomes larger as the inlet air temperature augments, ranging from around 1% to 6%.

6.5 | Effect of Reynolds number

This section explores the effect of Reynolds number on the heat transfer rate and pressure drop. Two different values of the rib pitch are considered: \( p = 6 \) and 16 mm. The rib angle is 30°, and the rib height is 2 mm, while the relative humidity is kept by 50%. Figure 15 shows the average Nusselt number and pressure drop as a function of the Reynolds number. As the Reynolds number increases, the Nusselt number rises monotonically for both rib pitch values because of the forced convection effect. As mentioned in the previous section, ribs with a smaller pitch tend to improve heat transfer. When the Reynolds number increases, this improvement in heat transfer becomes more pronounced. When the Reynolds number increases from 4000 to 12,000 (growth by 200%), the average Nusselt number for \( p = 6 \) mm increases by about 7.3%, but, that for \( p = 16 \) mm increases by about 31.4%. Also, it is found that within the Reynolds number range examined here, the rib pitch does not have a noticeable influence on the pressure drop.

7 | CONCLUSIONS

In the present work, the thermal performance of the humid air flow inside a SAH with V-shaped ribs has been evaluated numerically using a commercial FVM code. The objective of this study is that instead of dry air flow which has been considered in the previous works, the humid air flow has been applied here. In the
numerical simulations, the thermo-physical and transport properties of moist air are a function of mixture temperature with relative humidity as a parameter. In this study, the impacts of the rib angle, rib height, rib pitch, inlet air relative humidity, and inlet air Reynolds number on the thermal behavior of the proposed SAH has been evaluated. This study provides useful insight into how the heat transfer performance of the humid air flow inside a SAH can be improved by employing V-shaped ribs. The major obtained outcomes are as follows:

- Employing the V-shaped ribs causes the thermal boundary layer to separate and reattach, enhancing heat transfer.
- At a rib angle of 30°, both of the thermo-hydraulic performance and the average Nusselt number ratio reach to their maximum value.
- Concerning the impact of the rib height on the thermal performance of the proposed SAH, the maximum thermo-hydraulic performance belongs to the case with normalized rib height of 0.1–0.15.
- By increasing the rib pitch, the thermo-hydraulic performance of the proposed SAH decline. Accordingly, among the considered values of the rib pitch, the maximum thermo-hydraulic performance belongs to the case with rib pitch of 4 mm.
- As the relative humidity increases, the average Nusselt number augments and this increase becomes larger as the inlet air temperature augments, ranging from around 1% to 6%. In other words, Increasing the relative humidity of the inlet humid air from 0% to 50% is found to improve the heat transfer rate by up to 6%.
- When the Reynolds number increases by 200%, the average Nusselt number for p = 6 mm increases by about 7.3%, but, that for p = 16 mm increases by about 31.4%. Also, it is found that within the Reynolds number range examined here, the rib pitch does not have a noticeable influence on the pressure drop.

**NOMENCLATURE**

| Symbol | Description |
|--------|-------------|
| A      | area (m²)   |
| CP     | specific heat capacity (kJ/(kg · K)) |
| d      | diameter (m) |
| f      | friction factor (nd) |
| H      | height of the SAH duct (m) |
| h      | rib height (m) |
| k      | thermal conductivity (W/(m · K)) |
| L      | length of the SAH duct (m) |
| P      | pressure (Pa) |
| p      | rib pitch (m) |
| T      | temperature (°C) |
| t      | rib thickness (m) |
| v      | velocity (m · s⁻¹) |
| W      | width of the SAH duct (m) |

**GREEK AND SYMBOLS**

| Symbol | Description |
|--------|-------------|
| ρ      | density, kg/m³ |
| μ      | viscosity, kg/m · s |
| τ      | stress tensor |
| σ      | turbulent Prandtl number |
| η      | thermo-hydraulic performance (nd) |
| α      | rib angle (°) |
| k      | turbulent kinetic energy per unit mass (J/kg) |
| ε      | energy dissipation rate per unit mass (W/kg) |

**DIMENSIONLESS PARAMETER**

| Symbol | Description |
|--------|-------------|
| Nu     | Nusselt number, $Nu = \frac{hd}{k}$ (nd) |
| Re     | Reynolds number, $Re = \frac{ud}{\mu}$ (nd) |

**INDEX**

| Symbol | Description |
|--------|-------------|
| S      | smooth |
| avg    | mean/average |
| h      | hydraulic |
| f      | frictional |
| m      | mixture |
| t      | turbulence |
| w      | wall |

**ACKNOWLEDGMENT**

This study received no external funding.
CONFLICTS OF INTEREST
The authors declare no conflicts of interest.

DATA AVAILABILITY STATEMENT
The data sets generated during and/or analyzed during the current study are available from the corresponding author on reasonable request.

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**How to cite this article:** Nouri Kadijani O, Karemi Moghadam H, Mousavi Ajarostaghi SS, Asadi A, Saffari Pour M. Hydrothermal performance of humid air flow in a rectangular solar air heater equipped with V-shaped ribs. *Energy Sci Eng*. 2022;10:2276-2289. doi:10.1002/ese3.1136