Numerical modeling of an innovative arc shape rib based solar air heater

Hamid Kazemi Moghadam, Seyed Soheil Mousavi Ajarostaghi and Sébastien Poncet

Abstract
Solar air heating is a solar thermal technology wherein the sun energy is absorbed by an intermediate medium and utilized to heat air. In the present paper, a solar air heater equipped with innovative arc shaped ribs is considered. The heat transfer process in the proposed solar air heater is evaluated numerically for Reynolds numbers between 6000 and 12000. The influences of the rib cross section type, the inter-rib distance, the aspect ratio and the pitch of the ribs on the thermal efficiency of solar air heater are analyzed. Results indicate that firstly, ribs with quadrangular type of cross section exhibit the best coefficient of performance. Moreover, ribs with low pitch (40 mm) and aspect ratio equal to 0.5 show the highest thermal performance. Also, the effect of the distance between the ribs on the thermal efficiency of the considered solar air heater is insignificant. Finally, two correlations are obtained for the average Nusselt number and friction factor based on three dimensionless geometric factors.

Keywords
Solar air heater, numerical simulation, heat transfer enhancement, ribs

Date received: 25 January 2021; accepted: 26 July 2021

Introduction
Renewable energy is the energy obtained from the earth’s natural resources that are not limited or depleted. Renewable energy is an alternative to traditional energy that relies on fossil fuels and appears to do very little harm to the environment. The most popular renewable energy sources currently are solar energy, wind energy, hydro energy, and geothermal energy. A solar thermal equipment, which absorbs the energy of the Sun and uses it to heat air, is known as a solar air heater (SAH). This renewable energy based technology is currently used for its efficiency for heating buildings or in process heat applications including food products, chemical products, manufacturing of non-metallic mineral products and etc.

Heat transfer augmentation is one of the greatest key concerns in a variety of industrial applications and has attracted the interest of researchers. There are two overall approaches to increase heat transfer, namely active and passive techniques. Passive methods are efficient as they do not require any external energy source. Expanded surfaces such as fins or baffles are widely considered in that context to increase the heat transfer area and promote turbulence (see the review in Rashidi et al.). To quantify their benefits, channels are preferably chosen over more complex geometries for their sake of simplicity. Moreover, they represent well the typical geometry of solar air heaters concerning solar energy, cooling devices for electronic equipment, heat exchangers in HVAC systems and bioengineering.

A considerable amount of works has been achieved numerically to quantify the heat transfer enhancement in solar air heaters, channels and ducts. For example, Webb et al. analyzed the heat transfer in a channel with periodic baffles for different geometrical parameters (rib height L/D and rib spacing S/D), Prandtl and Reynolds numbers. The parameters L, S, and D are the rib height, rib spacing, and channel diameter, respectively. The Reynolds number was varied between 340 and 2000, the rib height and rib spacing between 0.0625 and 0.375 and 0.2 and 2,

1DITEN, University of Genoa, Genova, Italy
2Mechanical Engineering Faculty, Babol Noshirvani University of Technology, Iran
3Mechanical Engineering Department, Université de Sherbrooke, Quebec, Canada
Corresponding author: Sébastien Poncet, Mechanical Engineering Department, Université de Sherbrooke, Sherbrooke, QC J1K 2R1, Canada.
Email: Sebastien.Poncet@usherbrooke.ca
respectively. For high Prandtl numbers, heat transfer increases significantly. All in all, the best overall performance (higher Nusselt number and lower friction factor) was achieved for L/D = 0.25 and S/D = 1.5. Kelkar and Patankar evaluated the heat transfer in a channel with parallel plates and fins. The heat transfer rate rises by growing the baffle height and/or by decreasing the distance among two successive baffles. Lopez et al. investigated numerically the heat transfer in a duct with baffles. The Reynolds and Prandtl numbers were varied between 150 and 450, and 0.7 and 7, respectively and four baffle height to channel width ratios (H/D as blockage ratio) were considered, namely 0.5, 0.6, 0.7, and 0.8. A higher H/D ratio causes enhanced heat transfer. Similarly, as the thermal conductivity ratio (wall thermal conductivity over the fluid one) increases from 1 to 10^3, the heat transfer rate increases too. Patankar et al. evaluated the thermal performance of a channel with transverse arrays of baffles under fully developed conditions. The flow field was clearly affected by the produced laminar vortex provoked by the baffles leading to enhanced heat transfer.

Guo and Anand analyzed the thermal performance of a channel with a unique baffle placed in the entrance region. By increasing the baffle height and Reynolds number, a large recirculation appears enhancing the heat transfer rate. Murata and Mochizuki studied the laminar and turbulent fluid enhancement the heat transfer rate. By increasing the baffle height, the heat transfer rate increases too. Patankar et al. evaluated the thermal performance of a channel with baffles. Their main effect is to enhance the bulk flow and wall region. It explains the increase of both the Nusselt number and friction factor. Ho et al. studied the performance of a double-pass SAH equipped with baffles and fins. The collector with the fins and baffles exhibits a higher efficiency for all considered mass flowrates and reflux ratios.

Menasria et al. analyzed the thermal efficiency of a SAH equipped with rectangular baffles for Reynolds numbers between 4000 and 18000. The effects of baffle-pitch spacing ratio and blockage ratio considering sixteen different configurations were quantified. Using baffles in the duct is the most helpful method to enhance the thermal performance of SAH. Four baffle blockage ratios (BR = e/H in which e and H are absolute height of baffle and height of the duct, respectively) ranging between 0.7 and 0.98 and four pitch spacing ratios (PR = P/H where P is the inter-baffle spacing) ranging between 2 and 8 have been considered. The results showed that the configuration with BR = 0.7, PR = 2 and Re = 5000 exhibits the maximum thermo-hydraulic performance factor, named THPF (same definition as η).

Kumar and Goel investigated the impact of different roughness elements including semi-circular, circular, triangular, square, and rectangular on the flow dynamics as well as thermal characteristics of triangular passage SAH. At Re = 18.7 × 10^3, the best thermo-hydraulic performance parameter (THPF) obtained is 2.75 for forward-chamfered rectangular rib (rib aspect ratio e/w = 2, where e and w represent the height and width of the rib, respectively). The maximum value of THPF obtained for various roughness geometries decrease in the order of 2.68, 2.57, 2.45, 2.06, 1.79, and 1.60 for backward-chamfered rectangular ribs (with e/w = 2), rectangular ribs (with e/w = 2), square ribs, semi-circular ribs, and circular ribs, respectively. Kumar et al. studied numerically the heat transfer enhancement in a SAH having triangular cross-sectional passage and square shaped ribs. Two various roughness parameters have been considered in the analysis i.e. relative roughness pitch (P/e) and relative roughness height (e/D) and their values range from 5 to 13 and 0.013 to 0.05, respectively for Reynolds numbers between 3900 and 17900. Better augmentation of heat has been seen in SAH by providing ribs on the absorber plate. The highest heat transfer improvement (97%) was obtained for P/e = 10 and e/D = 0.05 at Re = 17900. The thermo-hydraulic performance factor (THPF) reaches a maximum value of 1.97 under these conditions.

Beside the numerical investigations, some experimental works were presented to optimize solar air heater. Akpinar and Kocyigit performed experimental tests to study the thermal behavior of a SAH with different obstacles placed on the absorber plate. Three different arrangements of obstacles have been considered. By increasing the mass flow rate, the collector efficiency increases. Also, using obstacles mounted in the collector duct has a major impact on its collector efficiency η = Q/(A_c. I) (Q, A_c, and I are useful heat rate, surface area of the collector, and solar radiation, respectively) by a 128.57%
enhancement of the heat transfer. El-Khawajah et al.\textsuperscript{34} evaluated experimentally the heat transfer process in a double pass SAH with fins. The maximum performances obtained for the 2, 4 and 6 fins configurations were 75.0, 82.1 and 85.9\% correspondingly. Furthermore, the collector with 6 fins exhibited the highest temperature change (among inlet and outlet ports). Promvonge et al.\textsuperscript{35} analyzed the turbulent heat transfer in a channel with 30\° inclined baffles. For Reynolds numbers within the range (4000-23000), they investigated the influence of the fin pitch to channel height ratio (\(\text{PR} = \text{P/H}\)) and the fin blockage ratio (\(\text{BH} = \text{b/H}\)). The results demonstrated that lower PR and greater BR causes great heat transfer amount.

One of the more interested passive methods consists of utilizing V-shaped inserts in the channel (or SAH) for heat exchange augmentation. Some of the relevant works are noted as follows. Han et al.\textsuperscript{36} focused on square ducts with rows of angular ribs. V-shape ribs led to greater heat transfer at a 60\° angle. In another work, Promvonge et al.\textsuperscript{37} considered the laminar channel flow and related heat transfer for periodic staggered V-baffles having an inclination of 45\°. The heat transfer (ratio of average Nusselt numbers (\(\text{Nu}/\text{Nu}_0\)) in which 0 index refers to the channel without baffles) may be increased by 100 up to 1100\%. But it is also accompanied by a drastic increase of the pressure drop, 2 to 90 times more than for the plain channel. Jin et al.\textsuperscript{38} examined numerically the thermal efficiency of a SAH with V-shaped ribs for Reynolds number between 8000-20000. They create helical vortexes, which enhance mixing between the warmer and colder streams close to the absorber wall. They obtained a maximum value of the thermal performance factor \(\eta\) equal to 1.93. Tamna et al.\textsuperscript{39} evaluated a SAH equipped with V-shaped baffle ribs which were placed with a 45\° angle of attack and 0.25 m baffle height. Impact of various baffle pitch to duct height ratios were investigated on the heat transfer. Smallest value of \(\text{PR}\) presented maximum pressure drop and heat transfer rate. Rana et al.\textsuperscript{40} studied numerically the influence of V-shaped ribs on the thermal performance of SAH. The range of the efficient parameters are \(\text{P/e} = 6-12\) (as relative roughness pitch), \(\text{Re} = 3800-18000\), and \(\text{\(\chi\)} = 30-75\°\) (\(\text{\(\chi\)}\) the angle of attack). The maximum thermal performance belonged to the case with 60\° and \(\text{P/e} = 10\). Rajaseenivasan et al.\textsuperscript{41} evaluated experimentally a SAH equipped with circular and V-shape baffles. Six various schematics of baffles have been tested. The overall system efficiency increases when increasing either the number of turbulators in the absorber plate or the inlet Reynolds number. The obtained maximum average Nusselt number was 210 at \(\text{Re} = 11615\). Kumar et al.\textsuperscript{42} evaluated experimentally the impact of V-pattern dimpled obstacles on the thermal performance of SAH. It enhanced the thermal performance of the proposed SAH by 7\% compared to the plain case without obstacle. Moreover, some correlations have been presented for the Nusselt number and pressure drop in the proposed SAH with a± 10\% deviation. Maithani and Saini\textsuperscript{43} presented a new design of SAH integrating V-ribs with symmetrical gaps. The maximum enhancement in terms of Nusselt number was obtained for a relative gap width of 4 and 3 gaps. The thermo-hydraulic improvement achieved between 154 and 239\%. Moreover, in another work,\textsuperscript{44} the same authors showed experimentally that the ribs increased by a factor 2.59 (resp. 2.87) the Nusselt number (resp. friction factor) compared to the plain SAH. Furthermore, the relative gap width of 4 and the use of 3 gaps lead to the highest value of the thermal performance parameter. Kumar et al.\textsuperscript{45} experimentally analyzed the heat transfer and fluid flow in a SAH integrated with the broken multi type V-baffles. It became apparent that the thermal performance of the proposed SAH is better than the performance of plain SAH.

Another efficient method for heat exchange improvement in the air channel of SAH is by inserting porous structures. Bayrak et al.\textsuperscript{46} analyzed experimentally a SAH equipped with porous baffles. They investigated separately the influences of some efficient operational parameters on the performance of the SAH. Two thicknesses, 6 and 10 mm, have been considered for the porous baffles, which have been placed in the SAH with two different arrangements; sequentially and staggered. The experimental tests have been done at two mass flowrates of air, namely 0.016 and 0.025 kg/s. The configuration using closed-cell aluminum foams (as porous baffles (PBs)) with thickness of 6 mm exhibits the maximum collector efficiency and growth of air temperature at an air mass flowrate equal to 0.025 kg/s. Ko et al.\textsuperscript{47} considered a channel with porous baffles to increase the thermal behavior of the system. These porous baffles increased the heat transfer by 300\% compared to the case with the plain channel. In addition to the above mentioned works, there are also some studies on arc-oriented roughness elements placed in a SAH to enhance heat transfer. It is worth mentioning that these elements, though having the same denomination as in the present work, have completely different shapes. Saini and Saini\textsuperscript{48} presented an experimental study to investigate the heat transfer augmentation in a solar air heater having roughened air duct with arc-shape parallel wires as vortex generators. The impacts of relative roughness height (\(e/d\)) and arc-angle (\(\text{\(\chi\)}/90\)) were evaluated on the average Nusselt number (\(\text{Nu}\)) and friction factor (\(f\)) in the range of Reynolds number (\(\text{Re} = 2000-17000\)). The maximum augmentation in terms of average Nusselt number was 280\% for a relative arc angle \(\text{\(\chi\)}= 0.3333\) at a relative roughness height of 0.0422. It was accompanied by an increase of the friction factor by 75\%. Ghritlahre et al.\textsuperscript{49} investigated
experimentally the influence of arc-oriented roughness elements on the performance of a SAH. The experiments were performed for mass flow rates between 0.007 and 0.022 kg/s. The maximum efficiency achieved for apex up was 73.2% at a mass flow rate of 0.022 kg/s. The arc shape roughness with relative roughness pitch, relative roughness height, rib roughness and arc angle were 10, 0.0395, 2.5 and 60°, respectively, have been chosen. The maximum efficiency obtained for apex up and apex down air flow arc shaped roughened SAH were 73.2% and 69.4% respectively, at the maximum mass flow rate. Kumar et al.\textsuperscript{50} studied the heat transfer augmentation in multiple curved arcs (with gap) ribbed solar air heater. Experimentation was designed using response surface methodology. ANOVA technique was utilized to obtain optimized thermal performance of the SAH. Quadratic correlations for the Nusselt number and friction factor were developed. The optimum value obtained for apex up and apex down air flow arc shaped roughened SAH were 73.2% and 69.4% respectively, at the maximum mass flow rate. Pandey and Bajpai\textsuperscript{51} presented an experimental study on the heat transfer and friction factor in a rectangular channel with multiple-arcs with gaps as roughness elements. The maximum augmentations in Nusselt number (\(Nu\)) and friction factor (\(f\)) were 5.85 and 4.96 times in comparison to the smooth duct. Singh et al.\textsuperscript{52} used artificial roughness on the underside of the absorber plate. The experimental study was undertaken to analyze the impact of various geometrical parameters for multiple arc shape ribs on the heat transfer and friction characteristics. The Nusselt number and friction factor were multiplied by a factor of 5.07 and 3.71, respectively for multiple arc-shaped roughness elements as compared to the smooth case.

In the present paper, innovative arc shaped ribs are used to increase the thermal performance of the solar air heater. The numerical simulations are performed for Reynolds numbers ranging between 6000 and 12000. The effects of the type of the rib cross section, height of the ribs (a), the distance between the ribs (e), their aspect ratio (AR) and pitch (P) on the thermal performance of SAH are evaluated numerically. To the best of the author knowledge, such a detailed analysis considering arc shaped ribs for a SAH has never been considered so far in the literature.

**Numerical modeling**

**Geometrical modeling**

Numerical simulations are performed to investigate the heat transfer rate in a solar air heater (SAH) equipped with innovative arc shaped ribs on the SAH duct bottom surface. Schematic views with the boundary conditions and relevant geometrical parameters are shown in Figure 1. The channel length (L), width, and height are 800 mm, 150 mm, and 25 mm, respectively. The ribs are characterized by five important parameters including \(r_1\), \(r_2\), P, e, and a. One defines a dimensionless parameter, namely the aspect ratio, as \(AR = r_1/r_2\). In the present paper, the influences of the type of the rib cross section, the distance between the ribs (e), the aspect ratio of the ribs (AR), the pitch of the ribs (P), and the rib height (a) on the thermal performance of the SAH are investigated numerically.

**Governing equations**

Among the various turbulence models of the k–\(\varepsilon\) family, the RNG and realizable k–\(\varepsilon\) models are known to perform better when dealing with swirling flows.\textsuperscript{53–56} Also, according to the literature review, the realizable k–\(\varepsilon\) model exhibited better performance than the RNG one and so it will be used in the present study to close the momentum equations in the turbulent regime.

The natural way to treat wall boundaries is to make the grid sufficiently fine so that the sharp gradients prevailing there are resolved. In the present work, an enhanced wall treatment is used to model correctly the heat transfer process near the wall under constant heat flux. It consists of a near-wall modeling method, which combines a two-layer model (each layer having its own wall function) and a blending function to smooth the transition between these two layers.

The fluid flow and heat transfer are then governed by the following equations for a 3D steady-state process:

\[
\frac{\partial \vec{u}_i}{\partial x_i} = 0 \quad (1)
\]

**Momentum equation:**

\[
\rho \vec{u}_i \frac{\partial \vec{u}_i}{\partial x_j} = -\frac{\partial P}{\partial x_j} + \frac{\partial}{\partial x_j} \left( \mu \left( \frac{\partial \vec{u}_i}{\partial x_j} + \frac{\partial \vec{u}_j}{\partial x_i} \right) - \rho \vec{u}_i \vec{u}_j \right) \quad (2)
\]

where the Reynolds stress tensor \(-\rho \vec{u}_i \vec{u}_j\) is evaluated through the Boussinesq approximation as:

\[
-\rho \vec{u}_i \vec{u}_j = 2\mu S_{ij} - \frac{2}{3} \rho k \delta_{ij} \quad (3)
\]

with \(\delta_{ij}\) the Kronecker delta.

Transport equations for the realizable k–\(\varepsilon\) model:

\[
\frac{\partial (\rho k \vec{u}_i)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_e} \right) \left( \frac{\partial k}{\partial x_j} \right) + G_k + G_b - \rho \varepsilon \right]
- Y_M + S_k \quad (4)
\]
\[
\frac{\partial(\rho u_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu}{\sigma} \right) \frac{\partial \epsilon}{\partial x_j} \right] + \rho C_1 S \epsilon \\
+ \rho C_2 \frac{\epsilon^2}{k + \sqrt{\epsilon}} + C_1 \frac{\epsilon}{k} C_3 G_b + S \epsilon 
\]  
(5)

where

\[
C_1 = \max \left( 0.43, \frac{\beta}{\beta + 5} \right) 
\]  
(6)

\[
\beta = S \frac{k}{\epsilon} 
\]  
(7)

\[
S = \sqrt{2S_{ij}S_{ij}} 
\]  
(8)

Equation (4) calculates the turbulence kinetic energy (k). Equation (5) calculates the dissipation rate of the turbulence kinetic energy (\( \epsilon \)). \( G_b \) is the generation of turbulence kinetic energy owing to the buoyancy effect, \( G_k \) is the generation of turbulence kinetic energy because of the mean velocity gradients, \( \mu_t \) is turbulent viscosity, \( S_b \) and \( S_e \) are user defined source terms. \( Y_M \) refers to the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate. \( C_2 \) and \( C_{1e} \) are constant fixed to 1.9 and 1.44, respectively. \( \sigma_k \) and \( \sigma_\epsilon \) are turbulent Prandtl numbers for \( k \) and \( \epsilon \), equal to 1 and 1.2, correspondingly. \( S_{ij} \) represents components of the deformation rate (strain rate magnitude) calculated as follows:

\[
S_{ij} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) 
\]  
(9)

The turbulent or eddy viscosity is computed from:

\[
\mu_t = \rho C_{\mu} \frac{k^2}{\epsilon} 
\]  
(10)

**Energy equation:**

\[
\frac{\partial (\rho u_j T)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \frac{2}{c_p} \frac{\partial T}{\partial x_j} \right) 
\]  
(11)
The turbulent heat flux is assumed to be negligible in the present case. It is noteworthy that the thermo-physical properties of air are kept constant throughout all the simulations. They are listed in Table 1.

### Numerical procedure and boundary conditions

The three-dimensional steady-state numerical simulations are performed using the commercial CFD code ANSYS Fluent 18.2 based on the finite volume method utilizing a commercial CFD code. The spatial discretization of the mass, momentum, turbulence kinetic energy, turbulence dissipation rate and energy equations has been achieved by a second-order upwind scheme. The velocity-pressure coupling has been overcome by the SIMPLE algorithm. To calculate the gradients, Green-Gauss cell-based method has been utilized. The convergence criteria were set to $10^{-6}$ for all residuals except for the energy equation ($10^{-8}$).

Air enters the channel with constant temperature and velocity. Four Reynolds numbers based on the hydraulic diameter of the channel including 6000, 8000, 10,000, and 12000 are considered here. The inlet temperature of air is 300 K. The turbulence intensity is fixed to 5% at the inlet but has no noticeable influence on the results. A constant heat flux equal to 1000 W/m² is imposed on the bottom surface of the SAH. It should be noted that to keep constant the surface area of the heater, the top wall of the ribs as well as the other part of the bottom surface without ribs are submitted to a heat flux, which is adjusted depending on the rib configuration to keep the overall heat flux received by the channel constant. Pressure is imposed at the outlet of the channel. No slip conditions are imposed on all walls.

### Grid independence study

Four different structured grids composed of quadrilateral elements were considered. The average Nusselt number is illustrated in Figure 2 as a function of the Reynolds number. It corresponds to a SAH with ribs having a quadrangular cross section. The geometrical parameters are fixed to $P = 50$ mm, $AR = 1$, $e = 0$ mm and $a = 2$ mm.

Grids 3 and 4 provide undistinguishable results such that grid 3 with 3,243,000 cells will be used in the following sections to save computational efforts. To calculate accurately the heat transfer near the bottom wall and ribs under a constant heat flux, the grid was refined using 10 prismatic layers in these regions with a refinement ratio equal to 1.1. It leads to a maximum value of the wall coordinate $y+$ within the channel equal to 0.82, which is well suited when using enhanced wall treatment to correctly model the viscous sublayer ($y+<1$).

### Validation of the solver

There are so many correlations to predict turbulence heat transfer coefficient inside an empty duct and for given ranges of Nusselt and Prandtl numbers. The Dittus-Boelter correlation predicts the mean Nusselt number as a function of the Reynolds number for fully developed flows in an empty duct. Figure 3(a) indicates the present results follow the Dittus-Boelter correlation with a maximum difference of 7.6%. For a constant wall temperature, the correlation of Seban and Shimazaki is also chosen to further validate the present numerical results. This correlation was recommended for $Pe.D \geq 100$. As shown in Figure 3(b), the agreement between the present results and the Seban and Shimazaki’s correlation is fairly good with a maximum difference of 9.2%.

In order to verify the numerical method for modeling the heat transfer in the proposed solar air heater, the present CFD results are compared to the experimental data of Thakur et al. for a square duct equipped with hyperbolic ribs. The absorber plate area is kept constant and fixed to $100 \times 100$ mm². The hyperbolic ribs are characterized by $e = 1$ mm and $P = 10$ mm. The comparison is displayed in Figure 4 in terms of the averaged Nusselt number variation with the inlet Reynolds number. The Reynolds Re and Nusselt Nu numbers are defined as:

$$Re = \frac{\rho u_{in} D_h}{\mu}$$

### Table 1. Thermo-physical properties of air.

| Property                      | Value  |
|-------------------------------|--------|
| Density, [kg.m⁻³]             | $\rho$ 1.165 |
| Specific heat, [J.kg⁻¹.K⁻¹]   | $C_p$ 1006.96 |
| Dynamic viscosity, [kg.m⁻¹.s⁻¹] | $\mu$ 1.8566 × 10⁻³ |
| Thermal conductivity, [W.m⁻¹.K⁻¹] | $\lambda$ 0.0264 |
where $D_h$ is the hydraulic diameter of the channel and $u_\text{in}$ the average velocity magnitude imposed at the inlet. The average Nusselt number can be obtained by:

$$
\bar{Nu} = \frac{1}{L} \int N_u \text{d}x
$$

The achieved outcomes by the present numerical work show pretty decent agreement with the experiments of Thakur et al.$^{57}$ with an average deviation of 4.28%. The discrepancies besides do not vary too much with the Reynolds number.

**Results and discussion**

The influence of adding innovative arc shaped ribs in the SAH bottom surface on the heat transfer enhancement is numerically investigated. The influences of the type of the rib cross section, the distance between the ribs ($e$), the aspect ratio and the pitch of the ribs ($P$) on the thermal performance of SAH are quantified in the following subsections. The results are discussed in terms of the average Nusselt number ($\bar{Nu}$), pressure drop ($\Delta P$), friction factor ($f$) and coefficient of performance (COP).

The friction factor is given by:

$$
f = \frac{2D_h\Delta P}{\rho u_\text{in}^2 L}
$$

with $\Delta P$ the pressure gradient across the channel.

These two parameters can be combined to form a kind of coefficient of performance (COP) in the following way:

$$
\text{COP} = \left( \frac{\bar{Nu}}{\bar{Nu}_0} \right)^{\frac{1}{f_0}}
$$

where $\bar{Nu}_0$ and $f_0$ are defined as the Nusselt number and friction factor of the simple channel of the SAH without ribs. This COP corresponds to the thermal hydraulic performance factor denoted $\eta$ or THPF by past authors.$^{58-60}$

**Influence of the shape of the rib cross section**

Three different types of the rib cross section including quadrangular, triangular and circular shapes are considered here. In this section, the influence of the shape of the rib cross section for different rib heights is quantified in detail by keeping all the other parameters constant ($AR = 1$, $P = 40$ mm, $e = 5$ mm). The schematics of the investigated models are illustrated in Figure 5. The rib height is varied between 1.5 and 3 mm. The average temperature of the wall and the average Nusselt ($\bar{Nu}$) number are displayed in Figure 6(a) and (b), correspondingly, as a function
of the rib height for the three types of cross section. The results are compared to the base case without rib.

According to Figure 6(a), whatever the rib height, the circular type exhibits a higher wall temperature whereas quadrangular and triangular types lead to the lowest ones with only small differences. At the same time, the circular shape has the lowest heat transfer rate. Figure 6(b) shows that ribs whatever their shape provide much higher average Nusselt numbers compared to the base case without ribs. Quadrangular and triangular types have a greater heat transfer in comparison with the circular type. Also increasing the rib height leads to higher Nusselt number for all considered shapes.

The pressure drop is displayed in Figure 7 as a function of the rib height and for the three types of cross section. Consequently, the results are obtained at a constant Reynolds number (Re = 10,000), so based on equation (15), the friction factor can be directly linked to the pressure drop. As expected, these two quantities increase by increasing the rib height in all cases. The highest and lowest pressure
drops (or friction factor) are obtained for the triangular and circular types, respectively.

The ratio of average Nusselt numbers ($\frac{\text{Nu}}{\text{Nu}_0}$) and the ratio of friction factors ($\frac{f}{f_0}$) are shown in Figure 8(a) and (b), correspondingly, as a function of the rib height (denoted $a$) and for the four configurations. It is worth mentioning that index 0 refers to the SAH without ribs. Figure 8(a) shows that adding ribs whatever their shape and height enhances the average Nusselt number in comparison with the simple SAH. More precisely, this enhancement is higher for the triangular shape whatever the rib height. The maximum and minimum improvements in average Nusselt number are obtained for the triangular and quadrangular cross-sections with $a = 3$ mm and $a = 1.5$ mm, respectively.

Figure 8(b) shows that the ratio of friction factors for ribs with any kinds of cross section is higher than for the simple SAH. Also, the maximum and minimum improvements in terms of the friction factor ratio in comparison with the simple SAH are obtained for the triangular ($a = 3$ mm) and quadrangular ($a = 1.5$ mm) cases, respectively. Finally, the variations of the coefficient of performance (COP) are illustrated in Figure 8(c). COP is a main parameter to analyze the thermal performance of any heat transfer system as it combines both the heat transfer enhancement and the pressure drop (related to the required pumping power). Whatever the rib shape, COP increases by increasing the rib height and reaches a maximum for $a = 3$ mm. The highest and lowest COP improvements are obtained for the circular shaped rib with $a = 3$ mm and the triangular shaped one with $a = 1.5$ mm, reaching 112% and 96.6% enhancements, respectively.

To well recognize the influence of the rib shape, the corresponding velocity vectors are presented in Figure 9. The ribs persuade swirl flows with a recirculation in the rib wake, which enhances the heat transfer between the heater (bottom wall) and the

![Figure 8](image_url)

**Figure 8.** (a) Ratio of Nusselt numbers ($\frac{\text{Nu}}{\text{Nu}_0}$), (b) ratio of friction factors ($\frac{f}{f_0}$) and (c) the coefficient of performance (COP) versus three different rib heights and for different rib shapes at $Re = 10,000$. 

Kazemi Moghadam et al.
fluid. It is noticeable that the recirculation behind the circular rib is smaller compared to the other cases provoking a lower pressure drop. However, it is very important to mention that the use of ribs with any geometry has a significant impact on the thermal performance of SAH because of the produced secondary flows behind the ribs (see Figure 9).

**Influence of the distance between two ribs (e)**

In this section, the effect of the distance between the ribs (e) on the thermal efficiency of the present SAH is studied numerically. Three various distances between the ribs including $e = 0$, $5$, and $10$ mm are selected. The simulations are done for four Reynolds numbers ranging between 6000 and 12000. The aspect ratio of the ribs (AR), the pitch of the ribs (P), and height of the ribs (a) are kept constant as 1, $40$ mm, and $2$ mm, respectively. The schematics of the considered models with different distances between the ribs are shown in Figure 10.

The pressure drop ($\Delta P$) and average Nusselt number ($\text{Nu}$) variation with various Reynolds numbers for different aspect ratios of the ribs are shown in Figure 11(a) and (b), respectively. The differences in terms of $\Delta P$ between the considered models remain very low (Figure 11(a)). However, whatever the Reynolds number, the highest pressure drop is obtained for $e = 5$ mm while its minimum appears for $e = 0$ mm. According to Figure 11(b), the highest average Nusselt number belongs to the case with $e = 5$ mm but its minimum value varies with the inter-rib distance, namely $e = 0$ mm at $\text{Re} = 6000$ and $e = 10$ mm at $\text{Re} = 12,000$. As expected, as the Reynolds number rises, the average Nusselt number augments too due to the forced convection and the trend is the same for all values of the distance between the ribs (e).
To better understand the influence of the distance between the ribs (e) on the heat transfer amount, temperature contours are displayed in Figure 12 at the bottom surface of the SAH for Re = 12,000, AR = 1, P = 40 mm, and a = 2 mm. The case with e = 5 mm exhibits the lowest area of hot spots behind the ribs, which means that more heat transfer happens from the bottom surface to the fluid. It should be noted that the differences between the considered models remain very small. However, among the evaluated models, the case with e = 5 mm exhibits the better thermal performance.

The ratios of the average Nusselt number (Nu/Nu₀) and the friction factor (f/f₀), and the COP (equation (16)) are depicted in Figure 13(a) to (c), respectively, as a function of the Reynolds number for three distances between the ribs (e). Figure 13(a) illustrates that for considered Reynolds numbers, the case with e = 5 mm exhibits a higher ratio of average Nusselt number. However, the difference between cases with e = 0 and 10 mm are very small. Also, based on Figure 13(b), the case with e = 5 mm has a higher ratio of friction factor than the other cases. Figure 13(c) depicts that at low Reynolds number (Re = 6000), the case with e = 5 mm provides the highest COP by a 121% improvement in comparison with the case without ribs (COP = 1). It has to be compared to cases with e = 0 mm and e = 10 mm, which lead to a 119.2% and 118.8% improvement, respectively.

Figure 14 shows the contours of the velocity magnitude for different distances between the ribs (e) at Re = 12,000. Its influence remains weak for this range of e values. Nevertheless, one can notice that the blue regions representing the secondary flows behind the ribs for e = 5 mm are larger (even of the differences with the other cases are small) leading to higher pressure drop and also heat transfer rate. It can be concluded that in the case with e = 5 mm, the proposed geometry of the arc-shaped ribs causes more secondary flows behind the ribs and consequently more heat exchange rate.
Influence of the rib pitch (P)

In this section, the effect of the pitch (P) of the ribs on the thermal performance of the SAH is investigated numerically. Three different pitch values, namely P = 40, 50, and 60 mm are selected. The distance between the ribs (e), the aspect ratio (AR) and the rib height (a) are kept constant as 0 mm, 1 and 2 mm, correspondingly. The schematics of the considered models with different aspect ratios are displayed in Figure 15.

The average Nusselt number (Nu) and pressure drop (ΔP) are displayed in Figure 16(a) and (b), respectively, variation with the Reynolds number for different aspect ratios of the ribs. According to Figure 16(a), the highest average Nusselt number is obtained for P = 40 mm and the minimum one for P = 60 mm. Also, growth in Reynolds number causes an augmentation in average Nusselt number because of the forced convection and the trend is the same for all models with different pitches.

Figure 16(b) shows that at low Reynolds number (Re = 6000), there is no noticeable difference when varying P. As the Reynolds number increases, the difference gets higher. Whatever the Reynolds number, the maximum pressure drop is obtained for P = 40 mm and it decreases by increasing the pitch. To better highlight the heat transfer amount and the influence of the rib pitch, temperature contours at the bottom wall of the SAH are illustrated in Figure 17 for the corresponding cases.

Figure 17 depicts that hot spots appear behind the first row of arc shaped ribs. The surface covered by the hot spots is a good indicator of the heat transfer performance of the SAH. Small hot spot areas are related to higher heat transfer from the bottom wall to the fluid (air). For P = 40 mm, the hot spots behind the arc shaped ribs are smaller compared to the other cases, which means that it can be considered as the best configuration. The size of the hot spots (resp. the heat transfer) increases (resp. decreases) with the pitch.

The ratios of the average Nusselt number (Nu/Nu₀) and of the friction factor (f/f₀) and the resulting COP (see equation (16)) are displayed in Figure 18(a) to (c), respectively, as a function of the Reynolds number (Re) and for various pitch values (P).

Figure 18(a) shows that for all Reynolds numbers, the highest and lowest ratios of the averaged Nusselt number (Nu/Nu₀) are obtained for the cases with P = 40 mm and P = 60 mm, respectively. Also, the case with P = 40 mm and Re = 8000 depicts the highest ratio of average Nusselt number by a 145% improvement in comparison with the simple SAH without ribs. Moreover, the case with P = 60 mm and Re = 12000 presents the lowest ratio of average Nusselt number by a 104% enhancement compared with the plain SAH.
According to Figure 18(b), all the investigated cases present higher friction factors compared to the plain SAH. On the other hand, the ratio of friction factor ($f/f_0$) is greater than one for all models. Also, it can be concluded that the highest and lowest ratios of friction factor ($f/f_0$) belong to cases $P = 40\text{ mm}$ (at $Re = 12000$) and $P = 60\text{ mm}$ (at $Re = 6000$) by 63 and 18% improvements (compared to plain SAH), respectively.

Figure 13. Influence of the Reynolds number on (a) the average Nusselt number ratio ($Nu/Nu_0$), (b) the ratio of friction factor ($f/f_0$), and (c) coefficient of performance (COP) for different distances between the ribs ($e$) and Reynolds numbers.

Figure 14. Contours of the velocity magnitude for different distances between the ribs ($e$) at $Re = 12000$ and $Y = 1.5\text{ mm}$.
So, the best parameter to analyze the thermal performance of the SAH by considering both ratios of average Nusselt number and friction factor appears to be the coefficient of performance (COP) (Figure 18(c)). Accordingly, all the proposed models depict COP values higher than one so thermal performance of all models is greater than for the plain SAH. From Figure 18(c), the maximum COP belongs to the model with \( P = 40 \text{ mm} \) at \( \text{Re} = 6000 \) by a 119% enhancement compared to the plain SAH. Furthermore, the model with \( P = 60 \text{ mm} \) at \( \text{Re} = 12000 \) has the lowest COP by an 87% improvement in comparison with the case without ribs.

Figure 19 presents the contours of the velocity magnitude for three pitch values in the plane \( \text{Y} = 1.5 \text{ mm} \) and for \( \text{Re} = 12000 \). For the lowest pitch \( P = 40 \text{ mm} \), the spatial extension of the red region characterized by a higher velocity magnitude is small. The fluid flow has then a lower velocity, which causes a higher heat transfer rate. On the other hand, for case \( P = 60 \text{ mm} \), the red region extends between the ribs, meaning a higher velocity. The time for heat transfer between the heater and the fluid is then reduced and as a consequence, the heat transfer rate decreases.

The velocity vectors for various pitches of the ribs are illustrated in Figure 20 for \( \text{Re} = 12000 \). At low pitch (here \( P = 40 \text{ mm} \)), the distance between the ribs is small, inducing weaker velocity in the gap between two ribs. So the time for heat transfer between the heater (bottom wall of the solar air heater) and air is sufficient, leading to a high heat transfer rate. On the contrary, as the pitch of the ribs increases from \( P = 40 \) to \( P = 60 \text{ mm} \), the high velocity region between the ribs extends and the velocity magnitude increases. Time is then not sufficient for a good heat transfer between the air and heater, and so the heat transfer rate decreases, which confirms the above results (Figures 16 to 19).

**Influence of the rib aspect ratio (AR = \( r_1/r_2 \))**

In this section, the effect of the aspect ratio of the ribs on the thermal performance of the SAH is
Figure 17. Temperature contours for three pitch values of the ribs (P) for \( \text{Re} = 12,000 \).

Figure 18. Influence of the Reynolds number on the (a) ratio of average Nusselt number (\( \text{Nu}/\text{Nu}_0 \)), (b) ratio of friction factor (\( \text{f}/\text{f}_0 \)), and (c) coefficient of performance (COP) for three pitch values.
Figure 19. Contours of the velocity magnitude in the plane $Y = 1.5\,\text{mm}$ for three pitch values at $Re = 12,000$.

Figure 20. Velocity vectors for different rib pitches: (a) $P = 40\,\text{mm}$, (b) $P = 50\,\text{mm}$ and (c) $P = 60\,\text{mm}$. Results obtained for $Re = 12000$ and $Y = 1.5\,\text{mm}$.

Figure 21. Schematics of the models with different aspect ratios.
investigated numerically for three aspect ratios including 2, 1, and 0.5 and Reynolds numbers between 6000 and 12000. The distance between the ribs (e), the pitch of the ribs (P) and their height (a) are kept constant as 0, 40 mm and 2 mm, correspondingly. The schematics of the considered models with different aspect ratios are shown in Figure 21.

The average Nusselt number (\(Nu\)) and pressure drop (\(\Delta P\)) are illustrated versus the Reynolds number for different aspect ratios (AR) in Figure 22 (a) and (b), respectively. Figure 22(a) presents that the maximum average Nusselt number belongs to the case with aspect ratio equal to 1 (AR = 1), while its lowest value is found for AR = 2. As expected, growth

Figure 22. Variations of (a) the average Nusselt number along the bottom wall with constant heat flux and (b) average Nusselt number (\(Nu\)) with the Reynolds number for three aspect ratios AR.

Figure 23. Temperature contours for different aspect ratios (AR) at Re = 12000 and \(Y = 0\) mm.
in Reynolds number causes the augmentation in average Nusselt number owing to the forced convection. Figure 22(b) shows that the lowest pressure drop appears for \( \text{AR} = 0.5 \), while there is noticeable difference between the cases \( \text{AR} = 1 \) and \( \text{AR} = 2 \).

To better realize the effect of the aspect ratio (AR) on the heat transfer process, the corresponding temperature contours are shown in Figure 23 for different aspect ratios at \( \text{Re} = 12,000 \). For \( \text{AR} = 0.5 \), regions with lower temperatures are wider compared to the other aspect ratio values, which confirms that a higher heat transfer rate is obtained for that configuration.

The ratios of average Nusselt number \((\text{Nu}/\text{Nu}_0)\) and of friction factor \((f/f_0)\) are depicted in Figure 24 (a) and (b), respectively. It should be noted that index 0 refers to the simple solar air heater (plain SAH) without any ribs. In order to evaluate the effect of the rib pitch (P) on the thermal performance, the calculated dimensionless parameter (COP) versus different Reynolds numbers (Re) is illustrated in Figure 24(c).

Figure 24(a) shows that the case with \( \text{AR} = 1 \) has the highest average Nusselt number ratio and the lowest one is obtained for \( \text{AR} = 2 \). According to Figure 24(b), the highest friction factor ratio appears for \( \text{AR} = 2 \), while the case with \( \text{AR} = 0.5 \) exhibits the lowest friction factor ratio. Figure 24(c) shows that at relatively high Reynolds numbers (\( \text{Re} = 8000-12000 \)), the highest and lowest COP are observed for \( \text{AR} = 0.5 \) and \( \text{AR} = 1 \), respectively. The contours of the velocity magnitude for various aspect ratios (AR) at \( \text{Re} = 12000 \) and \( Y = 1.5 \text{mm} \) are displayed in Figure 25. Accordingly, the secondary flows for cases \( \text{AR} = 1 \) and \( 2 \) are higher than for the case \( \text{AR} = 0.5 \), which induces a higher heat transfer rate.

**Correlations for the friction factor and Nusselt number**

Based on the obtained results from the previous sections, the Nusselt number and friction factor are functions of three dimensionless geometrical parameters and Reynolds number. The considered dimensionless

![Figure 24. Variations of the (a) average Nusselt number ratio, (b) friction factor ratio, and (c) coefficient of performance (COP) for different aspect ratios (AR) of the ribs and Reynolds numbers.](image-url)
geometrical parameters include the aspect ratio (AR or \(r^*\)), the dimensionless distance between the ribs (\(z'_1\)), and the dimensionless pitch (\(z'_2\)). The general form of the regression function for the Nusselt number and friction factor are as follows:

\[
 f(Re, r^*, z'_1, z'_2) = aRe^b r^c z'_1^d z'_2^e
\]

where \(r^* = \frac{r_1}{r_2}\), \(z'_1 = \frac{z_1}{r_1}\) and \(z'_2 = \frac{z_2}{r_1}\).

In equation (17), \(a\), \(b\), \(c\), \(d\) and \(e\) are constants. Finally, the correlation for the Nusselt number is given with a maximum error of 3.6%:

\[
 Nu = 0.105Re^{0.736} r^{-0.154} z'_1^{-0.258} z'_2^{0.015}
\]

The correlation for the friction factor is obtained with a maximum error of 5.8%:

\[
 f = 0.17Re^{-0.108} r^{-0.06} z'_1^{-0.338} z'_2^{0.004}
\]

The above equations are applicable for the following geometrical and operational conditions:

\[
 Pr \cong 0.71, \ 6000 \leq Re \leq 12000, \ 0.5 \leq r^* \leq 2, \ 0.5 \leq z'_1 \leq 1 \text{ and } 2 \leq z'_2 \leq 4.
\]

**Conclusions and future scope**

Solar air heating is a solar thermal technology in which the energy from the sun is absorbed by a medium and utilized to heat air. In the present paper, a SAH equipped with innovative arc-shaped ribs was considered. Actually, the objective of the present work was the evaluation of a solar air heater thermal performance in the presence of innovative arc-shaped ribs, which have never been considered previously in the literature. The heat transfer process in the studied SAH was investigated numerically using a commercial CFD code. The results were obtained for Reynolds numbers between 6000 and 12000. The effects of the rib cross section, the inter-rib distance, their aspect ratio and pitch on the thermal performance of the solar air heater were quantified. Results indicated that:

- Ribs with a quadrangular cross section exhibit the best coefficient of performance.
- Ribs with low pitch (here 40 mm) and aspect ratio equal to 0.5 show the highest thermal performance.
- The effect of the distance between the ribs on thermal performance of the proposed SAH is negligible.
- Two correlations are obtained for the average Nusselt number and friction factor based on the three geometrical parameters.

Although this study has provided new insight into the flow and thermal characteristics of a solar air heater combining innovative arc shape ribs, additional research is needed in order to demonstrate the viability of this design in more realistic engineering situations by the development of an experimental prototype.
Acknowledgements

S.P. acknowledges the NSERC chair on industrial energy efficiency established at Université de Sherbrooke in 2019 with the support of Hydro-Québec (laboratoire des technologies de l’énergie), Resources Canada (CanmetEnergy-Varennes) and Emerson Commercial and Residential Solutions.

Declaration of Conflicting Interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Funding

The author(s) received no financial support for the research, authorship, and/or publication of this article.

References

1. Olfian H, Ajarostaghi SSM and Ebrahimnataj M. Development on evacuated tube solar collectors: a review of the last decade results of using nanofluids. Int J Sol Energy 2020; 211: 265–282.
2. Olfian H, Zabihi Sheshpoli A and Mousavi Ajarostaghi SS. Numerical evaluation of the thermal performance of a solar air heater equipped with two different types of baffles. Int J Heat Transf Asian Res 2020; 49: 1149–1169.
3. Ackermann T and Söder L. Wind energy technology and current status: a review. Int J Renew Sust Energ 2000; 4: 315–374.
4. Rehman S, Al-Hadrami LM and Alam MM. Pumped hydro energy storage system: a technological review. Int J Renew Sust Energ 2015; 44: 586–598.
5. Javadi H, Ajarostaghi SS, Rosen MA, et al. Performance of ground heat exchangers: a comprehensive review of recent advances. Energy 2019; 178: 207–233.
6. Javadi H, Mousavi Ajarostaghi SS, Rosen MA, et al. Comprehensive review of backfill materials and their effects on ground heat exchanger performance. Sustainability 2018; 10: 4486.
7. Dbouk T. A review about the engineering design of optimal heat transfer systems using topology optimization. Appl Therm Eng 2017; 112: 841–854.
8. Rashidi S, Eskandarian M, Mahian O, et al. Combination of nanofluid and inserts for heat transfer enhancement – gaps and challenges. J Therm Anal Calorim 2019; 135: 437–460.
9. Noorbakhsh M, Mousavi Ajarostaghi SS, Zaboli M, et al. Thermal analysis of nanofluids flow in a double pipe heat exchanger with twisted tapes insert in both sides. Int J Therm Anal 2021; 1–12.
10. Hashemi Karouei SH and Mousavi Ajarostaghi SS. Influence of a curved conical turbulator on heat transfer augmentation in a helical double-pipe heat exchanger. Int J Heat Transf Asian Res 2021; 50: 1872–1894.
11. Kazemi Moghadam H, Mousavi Ajarostaghi S and Poncet S. Extensive numerical analysis of the thermal performance of a corrugated tube with coiled wire. Int J Therm Anal 2020; 140: 1469–1481.
12. Saini RP and Singal SK. A review on roughness geometry used in solar air heaters. Int J Sol Energy 2007; 81: 1340–1350.
13. Hans VS, Saini RP and Saini JS. Performance of artificially roughened solar air heaters – a review. Int J Renew Sust Energ 2009; 13: 1854–1869.
14. Alam T, Saini RP and Saini JS. Heat and flow characteristics of air heater ducts provided with turbulators – a review. Int J Renew Sust Energ 2014; 31: 289–304.
15. Boutina L and Bessafi R. Numerical simulation of mixed convection air-cooling of electronic components mounted in an inclined channel. Int J Appl Therm Eng 2011; 31: 2052–2062.
16. O’Connor D, Calautit JKS and Hughes BR. A review of heat recovery technology for passive ventilation applications. Int J Renew Sust Energ 2016; 54: 1481–149.
17. Mohammadi S, Mousavi Ajarostaghi SS and Pourfallah M. The latent heat recovery from boiler exhaust flue gas using shell and corrugated tube heat exchanger: a numerical study. Int J Heat Transf Asian Res 2020; 49: 3797–3815.
18. Khashan SA, Dagher S, Alazzam A, et al. Microdevice for continuous flow magnetic separation for bioengineering applications. J Micromech Microeng 2017; 27: 055016.
19. Webb BW and Ramadyhani S. Conjugate heat transfer in a channel with staggered ribs. Int J Heat Mass Transf 1985; 28: 1679–1687.
20. Kelkar KM and Patankar SV. Numerical prediction of flow and heat transfer in a parallel plate channel with staggered fins. Int J Heat Transfer 1987; 109: 25–30.
21. Lopez JR, Anand NK and Fletcher LS. Heat transfer in a three-dimensional channel with baffles. Numer Heat TR A-Appl 1996; 30: 189–205.
22. Patankar SV, Liu CH and Sparrow EM. Fully developed flow and heat transfer in ducts having streamwise-periodic variations of cross-sectional area. Int J Heat Transfer 1977; 99: 180–186.
23. Guo Z and Anand NK. Three-dimensional heat transfer in a channel with a baffle in the entrance region. Numer Heat TR A-Appl 1997; 31: 21–35.
24. Murata A and Mochizuki S. Comparison between laminar and turbulent heat transfer in a stationary square duct with transverse or angled rib turbulators. Int J Heat Mass Transf 2001; 44: 1127–1141.
25. Naphon P. On the performance and entropy generation of the double-pass solar air heater with longitudinal fins. Int J Renew Energy 2005; 30: 1345–1357.
26. Mousavi SS and Hooman K. Heat and fluid flow in entrance region of a channel with staggered baffles. Energy Convers Manag 2006; 47: 2011–2019.
27. Promvonge P, Sripattanapipat S and Kwankaomeng S. Laminar periodic flow and heat transfer in square channel with 45 inclined baffles on two opposite walls. Int J Therm Sci 2010; 49: 963–975.
28. Promvonge P, Sripattanapipat S, Tamna S, et al. Numerical investigation of laminar heat transfer in a square channel with 45 inclined baffles. Int. Commun Heat Mass 2010; 37: 170–177.
29. Ho CD, Chang H, Wang RC, et al. Performance improvement of a double-pass solar air heater with fins and baffles under recycling operation. Appl Energy 2012; 100: 155–163.
30. Menasria F, Zedairia M and Moummi A. Numerical study of thermohydraulic performance of solar air heater duct equipped with novel continuous rectangular baffles with high aspect ratio. *Energy* 2017; 133: 593–608.

31. Kumar R and Goel V. Unconventional solar air heater with triangular flow passage: a CFD based comparative performance assessment of different cross-sectional rib roughnesses. *Int J Renew Energy* 2021; 172: 1267–1278.

32. Kumar R, Kumar A and Goel V. Performance improvement and development of correlation for friction factor and heat transfer using computational fluid dynamics for ribbed triangular duct solar air heater. *Int J Renew Energy* 2019; 131: 788–799.

33. Akpinar EK and Komurcu A. Experimental investigation of thermal performance of solar air heater having different obstacles on absorber plates. *Int Commun Heat Mass* 2010; 37: 416–421.

34. El-Khawajah MF, Aldabbagh LBY and Egeliov GL F. The effect of using transverse fins on a double pass flow solar air heater using wire mesh as an absorber. *Int J Sol Energy* 2011; 85: 1479–1487.

35. Promvonge P, Skullong S, Kwankaomeng S, et al. Heat transfer in square duct fitted diagonally with angle finned tape — part 1: experimental study. *Int Commun Heat Mass* 2012; 39: 617–624.

36. Han JC, Zhang YM and Lee CP. Augmented heat transfer in square channels with parallel, crossed, and V-shaped angled ribs. *Int J Heat Transfer* 1991; 113: 590–596.

37. Promvonge P and Kwankaomeng S. Periodic laminar flow and heat transfer in a channel with 45 staggered V-baffles. *Int Commun Heat Mass* 2010; 37: 841–849.

38. Jin D, Zhang M, Wang P, et al. Numerical investigation of heat transfer and fluid flow in a solar air heater duct with multi V-shaped ribs on the absorber plate. *Energy* 2015; 89: 178–190.

39. Tamna S, Skullong S, Thianpong C, et al. Heat transfer behaviors in a solar air heater channel with multiple V-baffle vortex generators. *Int J Sol Energy* 2014; 110: 720–735.

40. Rana J, Silori A, Maithani R, et al. Computational fluid dynamics analysis of a V-rib with gap roughened solar air heater. *Therm Sci* 2018; 22: 963–972.

41. Rajaseenivasan T, Srinivasan S and Srithar K. Comprehensive study on solar air heater with circular and V-type turbulators attached on absorber plate. *Energy* 2015; 88: 863–873.

42. Kumar A, Kumar R, Maithani R, et al. Correlation development for Nusselt number and friction factor of a multiple type V-pattern dimpled obstacles solar air passage. *Int J Renew Energy* 2017; 109: 461–479.

43. Maithani R and Saini JS. Heat transfer and fluid flow behaviour of a rectangular duct roughened with V-ribs with symmetrical gaps. *Int J Ambient Energy* 2017; 38: 347–355.

44. Maithani R and Saini JS. Performance evaluation of solar air heater having V-ribs with symmetrical gaps in a rectangular duct of solar air heater. *Int J Ambient Energy* 2017; 38: 400–410.

45. Kumar A, Kumar R, Maithani R, et al. An experimental study of heat transfer enhancement in an air channel with broken multi type V-baffles. *Heat Mass Transfer* 2017; 53: 3593–3612.

46. Bayrak F, Oztop HF and Hepbasli A. Energy and exergy analyses of porous baffles inserted solar air heaters for building applications. *Int J Energy Build* 2013; 57: 338–345.

47. Ko KH and Anand NK. Use of porous baffles to enhance heat transfer in a rectangular channel. *Int J Heat Mass Transf* 2003; 46: 4191–4199.

48. Saini SK and Saini RP. Development of correlations for nusselt number and friction factor for solar air heater with roughened duct having arc-shaped wire as artificial roughness. *Int J Sol Energy* 2008; 82: 1118–1130.

49. Ghrilahre HK, Sahu PK and Chand S. Thermal performance and heat transfer analysis of arc shaped roughened solar air heater – an experimental study. *Int J Sol Energy* 2020; 199: 173–182.

50. Kumar R, Goel V, Singh P, et al. Performance evaluation and optimization of solar assisted air heater with discrete multiple arc shaped ribs. *Int J Energy Storage* 2019; 26: 100978.

51. Pandey NK and Baipai VK. Experimental investigation of heat transfer augmentation using multiple arcs with gap on absorber plate of solar air heater. *Int J Sol Energy* 2016; 134: 314–326.

52. Singh AP. Heat transfer and friction factor correlations for multiple arc shape roughness elements on the absorber plate used in solar air heaters. *Int J Exp Therm Fluid Sci* 2014; 54: 117–126.

53. Ansys I. *ANSYS fluent theory guide*. Canonsburg, PA: Ansys Inc., 2020.

54. Saedodin S, Zaboli M and Ajarostaghi SSM. Hydrothermal analysis of heat transfer and thermal performance characteristics in a parabolic trough solar collector with turbulence-inducing elements. *Int J Sustain Energy Technol Assess* 2021; 46: 101266.

55. Mousavi Ajarostaghi SS, Aghanazhad M, Davudi H, et al. Numerical evaluation the heat transfer enhancement in a tube with curved conical turbulator insert. *Int J Ambient Energy* 2021; 1–23.

56. Abbaspour M, Mousavi Ajarostaghi SS, Hejazi Rad SA, et al. Heat transfer improvement in a tube by inserting perforated conical ring and wire coil as turbulators. *Int. J. Heat Transf Asian Res.* 2021; 50: 6164–6188.

57. Thakur DS, Khan MK and Pathak M. Solar air heater with hyperbolic ribs: 3D simulation with experimental validation. *Int J Renew Energy* 2017; 113: 357–368.

58. Zaboli M, Saedodin S, Mousavi Ajarostaghi SS, et al. Numerical evaluation of the heat transfer in a shell and corrugated coil tube heat exchanger with three various water-based nanofluids. *Int. J. Heat Transf Asian Res.* 2021; 50: 6043–6067.

59. Afsharpanah F, Sheshpoli AZ, Pakzad K, et al. Numerical investigation of non-uniform heat transfer enhancement in parabolic trough solar collectors using dual modified twisted-tape inserts. *Int J Thermal Eng* 2021; 7: 133–147.

60. Outokesh M, Ajarostaghi SSM, Bozorgzadeh A, et al. Numerical evaluation of the effect of utilizing twisted tape with curved profile as a turbulator on heat transfer enhancement in a pipe. *J Therm Anal Calorim* 2020; 140: 1537–1553.