Performances of solar air heater using polygonal and trapezoidal rib absorber plate for augmentation of heat transfer

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Abstract:
An experimental investigation carried out in a solar air heater (SAH) by implementing an artificial rough surface in the absorber plate to enhance the convective heat transfer coefficient by creating turbulence in the laminar sub-layer. Inclined and sharp edge shape of ribs like the polygonal and trapezoidal shape of ribs were employed in an absorber plate at a relative pitch distance of p/e = 5, 7.5, 10 with Reynolds number in the range of 2000 – 20,000. An empirical equation of Dittus - Boelter and Blausis were taken to validate the performance of Nusselt Number (Nu) and friction factor (f) characterization for a smooth and rough surface. The result reveals that strong turbulence that occurred in SAH due to roughness of the surface in the fluid flow direction leads to heat transfer enhancement and varying pitch distances produce persistent improvement. Significant enhancement found in polygonal rib absorber plate at relative pitch distances p/e = 7.5, has produced a higher Nu number than other shapes and it attained maximum thermo-hydraulic performances (THP) 2.95. An empirical correlation was developed for the Nu and f of experimental values and found good agreement in the range of 8% with predicted values.

Keywords: Polygonal rib, trapezoidal rib, Solar Air Heater, THP.

1. Introduction:

Solar energy was abundantly available to engage the removal of moisture content in agriculture products like crops, grains, fruits, flowers, etc. as a dryer. In that, SAH plays a significant role in drying the product for domestic and industrial purposes. Essentially, the outcome of SAH has lower thermal efficiency due to minimum surface intensity in the smooth surface absorber plate and working fluids. To improve thermal performance, many researchers conducted investigations both experimentally and numerically by employing passive techniques such as ribs, grooves, etc. in the absorber plate for higher heat transfer rate. [1-3] illustrate the procedure for calculation and fabrication guidelines of the experimental setup of SAH. Khushmeet Kumar, et al. [4] and Atul Lanjewar, et al. [5] conducted an experimental investigation in ‘S’ shape arc ribs and ‘W’-shaped at arc angle of 30° to 75° ribs. The researcher reports at an arc angle of 60° produce better thermal enhancement. Giovanni Tanda [6] conducted an experimental analysis for transverse continuous, angled continuous, broken, and discrete V-shaped ribs. Characterization of
f and thermal coefficients were compared with a smooth duct and found better results. Smith Eiamsa-ard, et al. [7] used 3 different rib-groove arrangements with three pitch ratios 6.6, 10, and 13.3 for the experimental analysis. The report reveals that triangular-rib with a triangular-groove-shaped rib produced high heat transfer. Brij Bhushan and Ranjit Singh [8] did experimental studies on Nu and f using a protruded rib and findings are compared with a smooth surface, the author achieved a higher heat transfer rate. P.R. Chandra, et al. [9] carried an experiment to investigate in all four walls one by one in the ranging of Reynolds number 10,000 to 80,000. Researchers reached higher thermal enhancement. Rajendra Karwa, et al. [10] did an experimental analysis using Integral chamfered rib and extended thermal efficiency from 10% to 40% in SAH. Experimental research was done by Aharwal K.R., et al. [11] on the inclined discrete shape of a rib with a gap. The result produced good heat transfer at 1.0 relative gap width in inclined discrete rib. Experimental analysis carried out by Sompol Skullong, et al. [12 - 14] on Combined Wavy-Rib and Groove turbulators rib at a 45°attack angle [12] staggered-winglet perforated tapes rib [13], single attack angle of 45°[14] rib in the flow direction. The researcher reported that the highest thermal efficiency occurred than staggered-winglet perforated tapes rib. An experimental investigation was done by Saini S.K. and Saini R.P. [15] for arc-shaped wire rib for greater Nu and f in SAH. Varun, et al. [16] experimentally investigated for a combination of inclined and transfer ribs. The author stated that at a pitch distance of (p/e) 8 has the highest thermal efficiency. Using Taguchi methods by Patnaik, A et al. [17] investigated again and reported that Taguchi methods can be more effective to find Nu in SAH. Sukhmeet Singh, et al. [18] did an experimental analysis for 5 different ribs roughened plates with a flow-attack-angle of 30° to 75°. The report shows the highest heat transfer occurred at an angle of 60°. Ravi Kant Ravi and Saini R.P. [19] carried out an experimental analysis for the staggered ribs & discrete multi V-shaped ones. Researchers reveal that employing the rib on the side of the plate results in higher heat transfer. The experimental investigation was done by Rajaseenivasan T. et al. [20] for circular turbulators and V-type turbulators for both the inline and the staggered pattern. The result reveals that using a Zigzag arrangement in a circular type of turbulator produces the maximum thermal efficiency. N.K. Pandey, et al. [21] did an experimental investigation for multiple-arc shaped with a gap. The report stated that it achieved 5.86 times of heat transfer and 4.96 times of pressure drop occurred when compared to the smooth duct. Prashant Singh, et al. [22] carried both experimental and numerical investigation for the inline and staggered criss-cross inclined rib, the researcher reveals that the best Nu was achieved in the ranges of 2.7 and 3.1 for inline and staggered methods and the better thermal-hydraulic performance achieved between 1.2 and 1.5. Experimental and numerical simulations work was carried out by Gill R.S, et al. [23], for a broken arc-shaped rib with the Reynolds number ranging from 2000 to 16,000. The result attained the best performance. Vipin B, et al. [24] did a numerical and experimental investigation for reverse L shaped rib. The authors report that thermal efficiency has increased 2.9 times more than the smooth duct. Experimental and numerical research work was done by Sanjay K. Sharma, et al. [25] on the width to height W/H of 10 of the rib in the duct. The author reveals that mid ribs positioned at 3.3% and 6.67% truncation from the sidewalls produced higher thermal enhancement. Varun Kumar B [26] did a review work on various shapes of ribs and positioning of ribs in a duct and found an enhancement in thermal efficiency and regulated friction factor (f).

From the detailed literature survey, it can be concluded that an artificial roughness of the surface will enhance heat transfer better than a flat plate. Many researchers also endorse the fact that the sharp edge and inclined angle of the ribs produce more turbulence in the fluid flow direction, which results in higher Nu and nominal f. It was noted that most of the researchers have
focused only on single side inclination in the rib without modifying pitch distances. No attempt had hitherto been made to make an investigation by varying the shapes of the rib along with varying pitch distances. Taking this research gap into account, it was proposed to conduct the experimental analysis in SAH using novel shapes of rib, inclined on both sides with a sharp edge such as polygonal and trapezoidal shapes with various relative pitch distances.

The main objective of our work is to study the augmentation of Nu, characteristics of $f$, and Thermo Hydraulic Performance (THP) for proposed geometry. Also, the empirical correlation was developed for Nu and $f$ for the above parameters and compared with experimental values.

2. Experimental setup

A schematic diagram of the experimental setup of SAH is shown in Fig [1]. It has a) the rectangular duct in 3 portions, b) heater plate, c) digital thermocouple, d) U tube manometer, e) inclined manometer g) insulating materials, h) centrifugal blower with a control valve for regulating inlet airflow based on Reynolds number and connecting pipes

2.1 Rectangular Duct

A rectangular wooden duct consists of 2390 mm length (L) with a cross-section of 300 mm width (W) and 25 mm height (H). The aspect ratio (W/H) has kept 12 and pictorial views of SAH & rib are shown in Fig [2]. The entry section of the duct has an 890mm length, exit section of the duct has 500mm length with plenum and the test section has 1000mm long. Based on ASHRAE guideline [1] entry, exit section length was calculated by $5 \times (\sqrt{W \times H})$, and $2.5 \times (\sqrt{W \times H})$ for experimental investigation The inner portion of SAH has 1mm thickness of laminated mica sheet in sidewalls, entry and exit section. Wooden plank thickness in the sidewall is 18mm act was insulated and a total set up was roofed by thermo-form sheet to arrest heat losses.

![Figure. 1 Schematic diagram of the solar air heater](image-url)
2.2 Absorber Plate

A 0.8mm thickness of the aluminum sheet as an absorber plate was taken for investigation. The polygonal and trapezoidal-shaped rib at pitch distance $p = 20\text{mm}$. 30mm and 40mm were positioned one by one separately for further investigation. Roughness shapes are shown in Fig [3] A 6mm thickness of a nickel-chromium heated plate has stood on the top surface of the absorber plate to supply heat energy as solar radiation at heat flux $1000 \text{ W/m}^2$. The positioning of electric resistances has 60-volt energy is shown in Fig [4]. A voltage regulator is used to control and regulate the energy supply to the heater plate. Glass wool and 4cm thickness of thermo-form were used to protect the entire experimental setup as an insulating material to avoid heat losses.
2.3 Instrument

Orifice and U tube manometer are connected between the plenum and centrifugal blower to measure the mass flow rate and pressure drop across orifice meter. The inclined manometer [butyl alcohol with a density of 800 kg/m³ taken for accuracy] was used to measure the pressure drop across the test section. Calibrated copper constantan 16 thermocouples were located in the SAH as shown in Fig. [5] to measure the temperature of the various location.

3. Experimental procedure

An experimental setup was fabricated as per the guideline of ASHRAE [1]. At first, the centrifugal blower switched ON to runs the experiment for 30 minutes to bring the fluid to steady-state conditions inside the duct. The U-tube manometer was used to checks the mass flow rate at a constant velocity. All the instruments are calibrated, checked for proper mounting to avoid leakage, and confirm mass flow rate is constant in the duct. Electric energy was supplied to the heater plate at a constant heat flux of 1000 W/m² mounted on the top surface of the absorber plate. A flat absorber plate was initially placed in the test section for investigation, later it was replaced by a rough plate one by one. All values are compared to the plot for the augmentation of thermal performance. The various parameters are measured from the investigations.
a) 12 different temperatures recorded in the test section ($t_p$), inlet ($t_i$), outlet ($t_o$), and ambient temperature ($t_a$) are measured for analysis
b) Pressure drop readings are observed at the exit section of the duct ($\Delta p$).
c) $f$ measured by the pressure drop across the test section($\Delta p_o$).

The experiment was run continuously to record readings in smooth and roughness surfaces of the SAH. Dittus Boelter and Blausis empirical equation used to validate the experimental values of the smooth surface of Nu and $f$ was further extended for roughness surface. From that, the empirical correlation was developed for the experimental Nu and $f$ from the regressive analysis for further study.

4. Data Reduction

The above parameter is used to calculate the Nu and $f$ by using the following equation. Nusselt number is determined by using the equation

$$\text{Nu} = \frac{hD_h}{k}$$  \hspace{1cm} (1)

a) $D_h = \frac{4 \times A}{P}$ the hydraulic diameter of the duct and $k$ thermal conductivity (W/m K).
b) $h$ = convective heat transfer coefficient of the test section was determined by absorber plate and room temperature using the equation

$$h = \frac{q}{A_c(t_p - t_a)}$$  \hspace{1cm} (2)

a) $A_c$ is Area of absorber temperature, $t_p$ absorber mean temperature & $t_a$ ambient temperature
b) $q$ = rate of heat transfer from the artificial roughness plate to ambient air by equation

$$q = m \cdot C_p (t_o - t_i)$$  \hspace{1cm} (3)

a) $C_p$, Specific heat of air at constant pressure (J/Kg. K), $t_o$ outlet temperature ($^\circ$C) & $t_i$ inlet temperature ($^\circ$C).
b) $m$ = mass flow rate of air calculated from orifice plate ($\Delta p_o$) by equation

$$m = C_d A \left[\frac{2 \rho \cdot (\Delta p_o)}{1 - \beta^4}\right]^{0.5}$$  \hspace{1cm} (4)

a) $C_d$ Coefficient of discharge, $A$ cross-sectional area of the duct, $\beta$ is the ratio of throat diameter of orifice to the inner diameter of the orifice.
b) The pressure drop of the friction factor was determined across the length of the test section by equation

$$f = \frac{2 \cdot \delta p \cdot D_h}{4 \rho \cdot L \cdot V^2}$$  \hspace{1cm} (5)

where $L =$ length of the duct (m) & $v =$ velocity of the duct (m/s)

Validation of the experimental values of smooth surface, an empirical correlation of Dittus-Boelter equation for Nu and Blasius equation for $f$ is calculated by given equation,
\[ Nu_s = 0.024Re^{0.8}Pr^{0.4} \]  \tag{6}

Where Pr, Prandtl number of air, Nu, Nusselt number of smooth duct

\[ f_s = 0.085 Re^{-0.25} \]  \tag{7}

The effectiveness of roughness plate in SAH measured by thermo-hydraulic performances (THP) should be greater than one, it is calculated by

\[ THP = \left( \frac{Nu_f}{Nu_s} \right) \left( \frac{f_r}{f_s} \right)^{1/3} \]  \tag{8}

5. Results and discussion

Figure [6 -7] illustrates the comparison studies of experimental values of smooth surface absorber plate with an empirical equation of Dittus Boelter for Nu number and Blausis equation for f with various Reynolds numbers. It is clear that in SAH a performance of Nusselt number (Nu) increases with increasing Reynolds number, whereas the characterization of friction factor (f) decreased with increasing Reynolds number. The experimental values are found closer to the empirical equation reading and it establishes confidence to extend further in experimental work.

![Figure 6](image1.png)

**Figure. 6** Comparison of experimental values with a Dittus Boelter equation of Nusselt number Vs Reynolds number

![Figure 7](image2.png)

**Figure. 7** Comparison of experimental values with a Blausis equation of Friction factor Vs Reynolds number

Figure [8] shows the performances of Nu by varying Reynolds number for different relative pitch distances of polygonal, trapezoidal, and smooth absorber plates. It illustrates that implementing the roughness surface in SAH has increased the heat transfer coefficient. Fig [8] it is evident that the polygonal shape of the rib at p/e =7.5 shows greater heat transfer rate than others due to strong turbulence induced in fluid flow direction through the presence of sharp edge and inclination in both sides of the rib with proper pitch distances results better performance.
Figure 8 Nusselt number Vs Reynolds number for smooth and proposed roughness shape

Figure [9] illustrates the characterization of $f$ in the smooth surface along with proposed surfaces. It is observed that $f$ decreasing at a higher Reynolds number among these the polygonal rib plate has greater shear stress near to the wall surface and causes a higher pressure drop in the flow direction.

5.1 Thermo hydraulic performance (THP)

Figure [10] shows the THP performances of the proposed roughness surface with a smooth surface. In the experimental investigation, it is significant to study the enhancement of heat transfer by thermal performance and pressure drop by the hydraulic performance of SAH. It is calculated by using the equation [8]. It found that polygonal ribs of p/e = 7.5 and it has achieved 2.95 times THP more than a smooth surface and shown in Table.1.
From the experimental work, it is clear that the enhancement of Nusselt number is higher in polygonal rib with nominal friction factor than others. It elucidates that Fig [8-10] implementing roughness surface in SAH has produced a higher heat transfer rate at higher Reynolds numbers.

6. Correlations for Nusselt number and friction factor

The development of empirical correlation for experimental investigating is significant to predict the data which were used to extend the study for further work. Correlation of Nusselt number was developed from the effect of the various parameter are;

a. Nusselt number increase with increasing Reynolds number, Re

b. Nusselt number increase with the relative pitch distances, p/e

Where the Nusselt number is a function of Reynolds number, Re and relative pitch distances p/e, that can be written as

\[
Nu = f(Re, p/e)
\]  

(9)
Figure 11 Experimental values of a) Nu Vs. Re Number and b) Nu Vs. p/e

In experimental analysis performance of the roughness absorber plate of Nu was calculated by Reynolds's number for heat transfer coefficient as shown in Figure [11]. Regression analysis used to plot a straight line in experimental Nu values were shown in Figure [12] and equations are

\[
\ln (\text{Nu}) = n \ln (\text{Re}) + A_1
\]

Equation (10) can be written as

\[
\text{Nu} = A_0 \cdot \text{Re}^n
\]

Where \( A_0 = \frac{\text{Nu}}{(\text{Re})^n} \)  

The \( n \) values calculated by straight-line regression analysis were shown in Fig 12. (a) and values are in detail in Table 2.

Table. 2 values of slope and intercept of straight lines

<table>
<thead>
<tr>
<th>P/e</th>
<th>Slope value 'n'</th>
<th>Values of A0</th>
</tr>
</thead>
<tbody>
<tr>
<td>Polygonal p/e 5</td>
<td>0.879</td>
<td>-3.6858</td>
</tr>
<tr>
<td>Polygonal p/e 7.5</td>
<td>0.796</td>
<td>-2.8449</td>
</tr>
<tr>
<td>Polygonal p/e 10</td>
<td>0.8185</td>
<td>-3.0727</td>
</tr>
<tr>
<td>Trapezoidal p/e 5</td>
<td>0.8713</td>
<td>-3.7175</td>
</tr>
<tr>
<td>Trapezoidal p/e 7.5</td>
<td>0.8118</td>
<td>-3.0318</td>
</tr>
<tr>
<td>Trapezoidal p/e 10</td>
<td>0.8211</td>
<td>-3.1363</td>
</tr>
</tbody>
</table>
a) Figure. 12 Plot values of a) Ln (Nu) Vs. Ln (Re) and b) Ln (Ao) Vs. Ln (p/e)

$A_o$ is constant for the dependent function of the parameter $p/e$, where it illustrates relative roughness pitch distances combined to expose the effect of Nusselt number.

Let $\ln (A_0) = n_1 \ln\left(\frac{e}{d}\right) + A_1$  \hspace{1cm} (13)

Where $A_0 = A_1 \left(\frac{p}{e}\right)^{n_1}$  \hspace{1cm} (14)

Equation 14 can be written as $\frac{\nu}{R_e^n} \eta = A_1 \left(\frac{p}{e}\right)^{0.204}$  \hspace{1cm} (15)

From regression analysis using second order polynomial, their plotted values are shown in Figure [12.b] and $n_1$ values are 0.204 where above equation is written as

$\nu = A_1 \left(\frac{R_e}{p/e}\right)^{n_1} \exp\left(m_1 \ln^2\left(\frac{p}{e}\right)\right)$  \hspace{1cm} (16)

Where

$\nu = 0.0251 \left(\frac{R_e}{p/e}\right)^{0.833} \left(\frac{p}{e}\right)^{0.218} \exp\left[0.01318 \ln^2\left(\frac{p}{e}\right)\right]$  \hspace{1cm} (17)
The developed correlation of Nusselt number [Nu] values is compared with experimental values as shown in Figure [13]. It was good agreement with predicted values and found ±8% deviation.

b) Correlation for friction factors

Correlation of friction factor was developed from the effect of the parameters are
a. Friction factor decreases with increasing Reynolds number Re.
b. Friction factor increases with relative pitch distances, p/e
c. The function of the friction factor strongly dependent on Reynolds number and relative pitch distance, p/e were the equation is

\[ f = f(\text{Re}, \frac{p}{e}) \]  \hspace{1cm} (18)

The same procedure followed in Nusselt number correlation was repeated in friction factor to develop a correlation from regression analysis and data are plotted in Fig 14.

\[ f = B_0 (\text{Re})^m \]  \hspace{1cm} (19)
Figure. 14 Experimental values of a) $f$ Vs. Re Number and b) $f$ Vs. p/e

$B_0$ is the dependent parameter of p/e were relative roughness pitch distances have produced greater pressure drop and effect of $f$ as shown in Figure 15. Final equation of $f$ derived using the second-order polynomial from regression analysis and equation are

$$f = 4.06 \times 10^{-6} \left(\text{Re}\right)^{-0.225} \left(\frac{p}{e}\right) \times \exp\left[-0.0117\left(\ln\left(\frac{p}{e}\right)^2\right)\right]$$

(20)

Figure.15 Plot values of a) $\ln(f)$ Vs. $\ln(\text{Re})$ and b) $\ln(Bo)$ Vs. $\ln(p/e)$

The developed correlation of friction factor values is compared with experimental values as shown in Figure [16]. It was good agreement with predicted values and found a $\pm 8\%$ deviation.
An experimental investigation of heat transfer and friction factor for polygonal and trapezoidal rib in SAH with varying pitch distances was studied. It was found that enhancement of Nu occurred at increasing Reynolds number and $f$ decreasing at increasing Reynolds number. From experimental analysis, the finding is reported that

a) Implementing a roughness surface in the absorber plate has significant improvement in Nusselt number.

b) Among the polygonal rib at $p/e = 7.5$ has produced a higher Nusselt number at 20,000 Reynolds number.

c) Nominal friction factor occurred at the polygonal rib of absorber plate due to strong function

d) Thermo hydraulic performances (THP) of the proposed shapes have achieved the highest values 2.9 at polygonal shape at relative pitch distance $p/e = 7.5$.

e) An empirical correlation was developed for Nu and $f$ from the experimental values and it has maximum accuracy between experimental and calculated values.

**References:**


Figure 16 Comparison of experimental and predicted values of friction factor

7. Conclusion


