Effect of Alterations in V-down Rib on the Performance of an Artificially Roughened Solar Air Heater Duct

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Received: 28 June 2016; Published online: 1 October 2016

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Abstract

The present study is focused on the selection of a rib roughness geometry having best thermohydraulic performance among various alternatives of the V-down type configuration. The continuous V-down (type I) rib roughness pattern along with its four alternatives viz. V-down rib with two gaps (type II), V-down rib with two gaps combined with staggered ribs (type III), V-down rib with four gaps (type IV) and V-down rib with four gaps combined with staggered ribs (type V) were evaluated by calculating the Nusselt number, friction factor and thermohydraulic performance parameter values. All roughness patterns were applied on one broad wall of a high aspect ratio (12:1) solar air heater duct and data was collected by varying Reynolds number from 4000 to 12000 (5 levels) during the testing of each absorber plate. Present study used fixed values of pitch-to-rib height (P/e) ratio of 12, rib height-to-hydraulic diameter (e/Dₜ) ratio of 0.044, angle of attack (α) of 60°, staggered rib length to height ratio (w/e) of 4.5 and relative staggered rib pitch (p/P) of 0.65. Results revealed that, thermohydraulic performance of type II, III and V were superior as compared to the type I by 8.2-13.8%, 13.7-20.4% and 34.6-54.5% respectively, whereas the thermohydraulic performance of type IV was inferior to the type I by 1.5-14.9% for the range of Reynolds numbers evaluated.

Keywords: Solar air heater; Rib roughness; V-down rib; Staggered rib; Nusselt number; Friction factor; Thermohydraulic performance

1. Introduction

Global focus on enhancing the durability and thermal performance of gas turbines and heat exchangers and adding more functionality along with miniaturization of electronic devices has led to the designing and heat removal from these devices a challenging task. Researchers around the world are working to enhance the heat transfer rates from applicable surfaces in countless
applications requiring efficient cooling. Different methods suit different applications with the same ultimate objective. Hsu et al. (2014) investigated the thermal efficiency and thermal resistance of two-phase closed thermosyphons exhibiting surfaces with various wettabilities in the condenser and evaporator sections. It was reported that, by combining these two modified surfaces, thermal efficiency increased by 2.53% whereas, the thermal resistance of the two-phase closed thermosyphons reduced by 26.1% compared with that by using pure copper.

Application of artificial roughness in various forms viz., ribs, dimples, wings, baffles, twisted tapes, coil shaped wires etc. has proved to be an effective technique of heat transfer enhancement from heated surfaces (Han, 2004; Liu and Sakr, 2013). Presence of roughness on the heated surfaces induces disturbance in the flow in the form of periodic flow separations and reattachments, vortex shedding or generation of secondary flows along the roughness elements. Depending upon the shape of roughness element, generally more than one type of flow disturbances co-exist in the modified flow and these modifications of the flow near the heated surface help to destroy the viscous sub-layer (which acts as barrier to the convective heat transfer (Bhatti and Shah (1987)) and hence improve the heat transfer from the heated surface. One of the major concerns while using the artificial roughness in any form is the associated pressure drop penalty, hence additional pumping power is required to sustain the desired flow rate. Thus, it becomes imperative to evaluate the performance of roughness geometry by taking into consideration the benefit of enhanced heat transfer as well as loss of pressure due to increase in friction factor. Webb and Eckert (1972) introduced the concept of thermohydraulic performance parameter, considering both Nusselt number and friction factor enhancements and stated that any roughness geometry shall be considered useful only if it returns value of thermohydraulic performance parameter more than one.

Out of all the artificial roughness options, fixing of small diameter wires is the most cost effective and easy way to create artificial roughness pattern on the surface to be cooled down. Large amount of literature is available on the use of ribs in various patterns for the enhancement of heat transfer (Han et al., 1991; Taslim et al., 1996; Karwa, 2003). Initial studies (Han et al., 1978; Gupta et al., 1993) used ribs positioned at 90° to the direction of flow for periodic interruption of the primary flow and reported enhancement in the heat transfer with considerable pressure drops. With the aim to further enhancing the rate of heat removal with lesser pressure drop penalty, studies were performed using inclined ribs (Han and Park, 1988; Park et al., 1992) and good amount of improvement in the performance was observed as compared to the transverse ribs. Flow visualization studies through a duct roughened with inclined ribs (Taslim et al., 1996) revealed that, flow moving along the ribs (secondary flow) after striking the end wall of the duct, takes the shape of a recirculation vortex. This secondary flow vortex carries along cool fluid from locations away from the heated surface and forces it to strike the heated surface near the leading edge of the rib, thereby picking heat from it. So, inclined ribs offered very high heat removal rates near the leading edges while poor heat transfer rates near the trailing edges (due to accumulation of hot fluid moving along the rib). Han et al. (1978) evaluated the effect of angle of inclination of a continuous rib on the heat transfer and friction characteristics and reported that angling of the rib from 90° to 45° resulted in steep fall in the friction factor with only 5% decrease in the average Stanton number, which makes angled ribs more suitable for optimum thermohydraulic performance. With the obvious benefit of having two leading edges per rib, Taslim et al. (1996)
experienced with V-shaped ribs and noticed better heat transfer rates as compared to the inclined ribs, due to the generation of double secondary flow vortices.

In an effort to further enhance the heat transfer in ribbed channels, discrete ribs in various configurations were tested and it was observed that broken ribs performed better than their continuous counterparts in all cases (transverse, inclined, V-shaped) with comparatively smaller pressure loss penalties (Park et al., 1992; Fann et al., 1994; Gupta et al., 1997; Momin et al., 2002; Mulluwork, 2000). These studies reported that, airflow over a staggered array of angled discrete ribs separates not only at the top edges of the ribs but also at the ends of the ribs. The secondary flow near the wall resulting from flow separation at the ends of the ribs interrupts the growth of the boundary layers downstream of the nearby reattachment zones. This secondary flow around the ends of the ribs along with the secondary flow caused by the orientation of the ribs interacts with the primary flow separated from the top edges of the ribs at the reattachment position on the wall. As a result of the vigorous mixing of the flowing air near the wall and slight increase in the rib surface area, angled discrete ribs are believed to enhance the heat transfer to the air stream more than angled full ribs.

Many researchers dedicated their studies to the effect of rib roughness on the performance of high aspect ratio (10 to 12:1) solar air heater ducts and observed that, application of the rib roughness geometries viz. transverse ribs (Gupta et al. (1993)), inclined ribs (Gupta et al. (1997)), V-shaped ribs (Momin et al. (2002)), staggered discrete V-shaped ribs (Mulluwork (2000)), V-up and V-down discrete ribs (Karwa (2003)), inclined rib with gap (Aharwal et al. (2009)), V-rib with gap (Singh et al. (2011)) and multi V-rib with gaps (Anil et al. (2013)) resulted in enhancement of the performance of solar air heaters. Patil et al. (2012) experimented with a variation of the V-rib with one gap geometry by fixing a staggered rib piece in front of each gap and reported improvement in heat transfer and thermohydraulic performance. Deo et al. (2016) studied the effect of relative roughness pitch, angle of attack and relative roughness height on the performance of multigap V-down rib combined with staggered ribs roughness and reported improvement in thermohydraulic performance as compared to the V-down rib with gap (Singh et al. (2011)) and broken V-rib roughness combined with staggered ribs (Patil et al. (2012)) geometries.

On the basis of literature survey, it can be concluded that V-down rib is the best choice as a basis for a roughness pattern to enhance the performance of a solar air heater and some modifications in it can further substantially enhance the performance. Since experiments on various versions of the V-rib geometries were performed on different test rigs at different locations with even different duct aspect ratio and rib geometry parameters (corresponding to best performance) in many cases so, a new researcher often faces difficulty while comparing the performance of various versions of the V-rib based roughness geometries. Keeping in view the convenience of the readers, it was decided to conduct experiments on five versions of the V-down rib roughness patterns viz. continuous V-down rib, V-down rib with two gaps (one in each limb), V-down rib with two gaps combined with staggered ribs, V-down rib with four gaps (two in each limb) and V-down rib with four gaps combined with staggered ribs, using a fixed set of duct and rib geometry parameters under similar environmental conditions, so as to obtain an authentic comparison on the basis of their relative performance.
Following sections of the paper explain the V-rib based roughness geometries along with applicable roughness geometry parameters, experimental details (set up, procedure, conditions and uncertainty analysis), data processing, set up validation, roughness effect on undisturbed flow and discussion of results from this study.

2. Details of Roughness Geometries

Type of roughness pattern and the values of various flow and roughness parameters employed for this study are listed in Table 1. In order to elaborate the details of the rib geometry parameters; Fig. 1 shows the schematic of type V rib roughness pattern. The geometry of the pattern can be explained by the values of rib height (e), V-rib pitch (P), angle of attack (α), number of symmetrical gaps (n) in the V-rib, gap width (g), length of the staggered rib (w) measured along the width of the absorber plate and distance of the staggered rib from the V-rib (p). These parameters have been expressed in the form of dimensionless roughness parameters, viz., relative roughness pitch (P/e), relative roughness height (e/Dh), relative gap width (g/e), relative staggered rib size (w/e) and relative staggered rib position (p/P). Roughness pattern was created by fixing V shaped circular Aluminum wires with an adhesive to the absorber plate followed by the cutting of gaps and fixing of staggered ribs. Values of P/e, e/Dh and (α) were selected based upon their optimization in our previous study (Deo et al. (2016)), whereas values of (w/e) and (p/P) were estimated based upon the data available in the literature (Mulluwork, 2000; Patil et al., 2012).

Table 1 Details of various roughness patterns along with geometric and flow parameters.

<table>
<thead>
<tr>
<th>S No.</th>
<th>Name of roughness pattern</th>
<th>Type</th>
<th>Fixed duct and rib geometry parameters &amp; values</th>
<th>Reynolds number</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Continuous V-down rib</td>
<td>I</td>
<td>W/H: 12, P/e: 12, e/Dh: 0.044, α: 60°</td>
<td>4000, 6000, 8000, 10000, 12000 (5 levels)</td>
</tr>
<tr>
<td>2</td>
<td>V-down rib with two gaps</td>
<td>II</td>
<td>W/H: 12, P/e: 12, e/Dh: 0.044, α: 60°, n: 2, g/e: 1</td>
<td>4000, 6000, 8000, 10000, 12000 (5 levels)</td>
</tr>
<tr>
<td>3</td>
<td>V-down rib with two gaps combined with staggered ribs</td>
<td>III</td>
<td>W/H: 12, P/e: 12, e/Dh: 0.044, α: 60°, n: 2, g/e: 1, w/e: 4.5, p/P: 0.65</td>
<td>4000, 6000, 8000, 10000, 12000 (5 levels)</td>
</tr>
<tr>
<td>4</td>
<td>V-down rib with four gaps</td>
<td>IV</td>
<td>W/H: 12, P/e: 12, e/Dh: 0.044, α: 60°, n: 4, g/e: 1</td>
<td>4000, 6000, 8000, 10000, 12000 (5 levels)</td>
</tr>
<tr>
<td>5</td>
<td>V-down rib with four gaps combined with staggered ribs</td>
<td>V</td>
<td>W/H: 12, P/e: 12, e/Dh: 0.044, α: 60°, n: 4, g/e: 1, w/e: 4.5, p/P: 0.65</td>
<td>4000, 6000, 8000, 10000, 12000 (5 levels)</td>
</tr>
</tbody>
</table>
3. Experimental Details

3.1 Test set-up

An indoor experimental test facility (simulating the outdoor solar air heater) was created to study the effect of different rib geometry patterns on the heat transfer and fluid flow characteristics of a rectangular duct. A schematic diagram of the experimental setup is shown in Fig. 2. It consists of an entry section, a test section, an exit section, a flow meter, a flow control valve and a motor operated centrifugal blower. The cross-section of 3000 mm long rectangular duct is also shown in the Fig. 2. It has a flow cross-section of 300 mm x 25 mm. Lengths of the test section, entry section, and exit section measuring 1400 mm, 800 mm (9.2√WH), and 800 mm (9.2√WH), respectively were decided as per the guidelines available in ASHRAE standard 93-77. The referred standard recommends minimum entry and exit lengths of (5√WH) and (2.5√WH), respectively; therefore with the current selection of entry section length, the flow can be assumed to be fully developed before entering the test section. Provision of longer exit section (800 mm length) is made to minimize the end effects due to convergence of rectangular section into a small circular cross section in the transition section. The top side of the duct is made from 1 mm thick galvanized iron (GI) sheet duly roughened with the pattern selected for experimentation in the test section part. The length of the GI sheet is 2400 mm. It extends 500 mm on both sides of the test section, in the entry and exit sections. The remaining topside of the entry (300 mm) and exit lengths (300 mm) of the duct is of a 1 mm thick smooth sunmica sheet. The inner surfaces of the remaining three sides of the duct are also made from smooth sunmica. An electrical heater is placed above the absorber plate and it supplies uniform heat flux (1000 W/m²) to the absorber plate. To avoid top heat losses from electric heater, a 12 mm thick asbestos sheet is placed above the electric heater assembly. It is further insulated on the top with 75 mm thick glass wool layer sandwiched between 12 mm thick plywood sheets.
A calibrated orifice meter connected with an inclined U-tube manometer was fitted to measure the mass flow rate of the air sucked through the duct by the blower fitted on the exit side of the setup. Calibrated thermocouples, prepared by butt welding of 0.3-mm-diameter copper constantan wires, were used for temperature measurement. Sixteen thermocouples were used to measure the temperature of the absorber plate at different locations, as shown in Fig. 3. Air leaving the exit section has to pass through a mixing section having baffles to ensure uniform temperature in the entire cross-section. Five thermocouples were employed to measure the outlet air temperature after the mixing section and one thermocouple was used to determine the inlet air temperature. A data logger interfaced to a computer was used to record the output of all the thermocouples. The pressure drop across the test section was measured by a digital micromanometer of least count 0.1 Pa.
Fig. 3. Locations of the thermocouples on absorber plate (all dimensions are in mm)

3.2 Experimental conditions

The experiments were performed in accordance with recommendation of ASHRAE standard 93-77 for testing of solar collectors operating in the open loop flow mode. In order to ensure the experimental conditions in accordance with the actual operation of the outdoor solar air heater, voltage supply to the electric heater was adjusted with the help of variac to maintain input power of 1000 W/m², which is equivalent to the solar insolation available in this part of the earth on a sunny day. Since bulk fluid temperature keeps on varying whenever mass flow rate was varied until steady state was reached (due to constant heat flux value throughout the experiment) so, values of thermo-physical properties of the air at the inlet of the duct were used to calculate density and viscosity to set the Reynolds number at the beginning of each run of the experiment by adjusting the flow control valve. At the start of each set of experiment, it was ensured that all instruments were working properly and there was no leakage at the joints. All the data were collected under steady state conditions. For each set, from start it took about 1.5–2.0 hours to reach a steady state. The steady state was assumed to have been attained when no considerable variation in plate temperature and outlet air temperatures was observed over a period of 10 minutes.

3.3 Experimental procedure

Procedure to conduct the experiment and recording of data can be explained with the steps given below:

Step 1: Before the fitting of absorber plate, whole duct was cleaned with the help of a cotton cloth and portable blower to make it free from any type of flow obstructions.

Step 2: All the thermocouples were physically inspected for any damage to the bead and then checked to ensure same temperature reading at the ambient temperature. U-tube Manometer was also checked to have same level of the fluid in both limbs.

Step 3: Absorber plate was painted black from the side facing the heater and all the sixteen thermocouples were soldered to the test section of the absorber plate at the locations shown in Fig. 3 and all beads were covered with the putty to avoid direct contact with the heater. Micromanometer was also attached to two pressure tappings attached to the test section. Heater assembly was placed over the absorber plate and whole length of the duct is finally wrapped in a thermocole sheet to avoid any heat loss by conduction to the environment.
Step 4: Electric supply to all the instruments is turned ‘ON’ and flow rate is adjusted to give Reynolds number value of 12000. Supply to the heater is adjusted from variac to keep input energy at 1000 W/m². Rise in the temperature of all the thermocouples attached to the data logger is observed at regular intervals.

Step 5: Upon reaching the steady state as explained in the experimental conditions; inlet air temperature, outlet air temperature at five points in the span-wise direction of the duct, temperature of the heated absorber plate at 16 locations, pressure drop across the orifice plate, and pressure drop across the test section were recorded.

Step 6: Flow rate is varied to set the Reynolds number at the next level and upon reaching the steady state all the observations listed in step 5 were recorded again. This was repeated for all levels of the Reynolds number.

Step 7: After recording the reading for all levels of the Reynolds number for a particular absorber plate, heater supply was turned ‘OFF’ and setup was run at full opening of the flow control valve so as to cool down the absorber plate quickly. When temperature of the thermocouples attached to the test section comes close to the ambient temperature then blower is also turned ‘OFF’ and heater is lifted from the duct for removing the absorber plate. Steps 1 to 7 were repeated for all absorber plates listed in Table 1 above.

In order to compare the results of the roughened duct with that of smooth duct, a smooth plate operating under similar flow conditions was also tested.

3.4 Experimental uncertainty

Uncertainties in final experimental results have been calculated based upon the analysis of errors in experimental measurements proposed by (Kline and McClintock, 1953). The uncertainties in the values of Nusselt number, Reynolds number and friction factor were calculated using the least count values of the instruments used for the measurement of linear dimensions, plate and air temperatures, pressure drop across the orifice plate and test section, atmosphere pressure, current and voltage. Relations given in equations 1, 2 and 3 were used for the estimation of uncertainty range of the Nusselt number, Reynolds number and friction factor respectively:

\[
\frac{\delta Nu}{Nu} = \left[ \left( \frac{\delta h}{h} \right)^2 + \left( \frac{\delta D_h}{D_h} \right)^2 + \left( \frac{\delta k}{k} \right)^2 \right]^{0.5}
\]  
\[
\frac{\delta Re}{Re} = \left[ \left( \frac{\delta V}{V} \right)^2 + \left( \frac{\delta \rho_{air}}{\rho_{air}} \right)^2 + \left( \frac{\delta D_h}{D_h} \right)^2 + \left( \frac{\delta \mu}{\mu} \right)^2 \right]^{0.5}
\]  
\[
\frac{\delta f}{f} = \left[ \left( \frac{\delta V}{V} \right)^2 + \left( \frac{\delta \rho_{air}}{\rho_{air}} \right)^2 + \left( \frac{\delta D_h}{D_h} \right)^2 + \left( \frac{\delta L}{L} \right)^2 + \left( \frac{\delta (\Delta P)_d}{(\Delta P)_d} \right)^2 \right]^{0.5}
\]

The maximum uncertainties in the estimation of Nusselt number, Reynolds number and friction factor were found to be ±3.08%, ±1.78% and ±4.56% respectively.
4. Data Reduction

The mass flow rate \(m\) of air has been determined from the pressure drop across the orifice plate using the following relation:

\[
m = C_d A_i \left[ \frac{2 \rho_{air, i} (\Delta P_o)}{1 - \beta^4} \right]^{0.5}
\]  

(4)

where, \(C_d\) is determined as 0.612 through calibration.

The heat transfer rate to the air is calculated as:

\[
Q_a = m C_p (T_o - T_i)
\]  

(5)

The average heat transfer coefficient for the test section is calculated as:

\[
h = \frac{Q_a}{A_p (T_{pm} - T_{fm})}
\]  

(6)

The average Nusselt number is calculated from the heat transfer coefficient \((h)\) using the following relations:

\[
N_u = \frac{h D_h}{k_{air}}
\]  

(7)

The friction factor \((f)\) is determined from the measured value of pressure drop across the length \((L_f)\) of 1.2 m in the test section using the following equation:

\[
f = \frac{2(\delta P) \rho_{air} D_h}{4L_f G_{air}}
\]  

(8)

Application of the rib roughness on the absorber plate of the SAH is supposed to increase the Nusselt number as well as friction factor. To evaluate the performance of the SAH for equal pumping power for smooth as well as roughened plates, a thermo-hydraulic performance parameter \((\eta)\) considering both Nusselt number and friction factor enhancements, as defined by Webb and Eckert (1972) is used as given in the following equation:
\[ \eta = \left( \frac{Nu}{Nu_s} \right) \left( \frac{f/f_s}{0.3} \right)^{1/3} \]  

(9)

The thermo-physical properties of the air used in the calculations correspond to bulk mean air temperature.

5. Results and Discussion

The results from the experiments performed as per the procedure given in section 3.3 to study the effect of various design modifications in the V-down rib on the heat transfer enhancement, friction factor and thermohydraulic performance of the solar air heater duct are discussed in the forthcoming sections.

5.1 Setup validation

For validation of experimental setup, Nusselt number and friction factor for the smooth duct were determined from experimentation on smooth plate and their values were compared with the values calculated from the correlations given by Dittus–Boelter equation (Rohsenow et al. (1998)) and modified Blasius equation (Bhatti and Shah (1987)) for smooth duct available in the literature.

Dittus–Boelter equation: \[ Nu_s = 0.023Re^{0.8}Pr^{0.4} \]  

(10)

Modified Blasius equation: \[ f_s = 0.085Re^{-0.25} \]  

(11)

Figure 4 shows the comparative variation of the experimental and predicted values of the Nusselt number and friction factor for various values of Reynolds numbers. The average absolute deviation between the experimental and predicted values of the Nusselt number and friction factor were
calculated to be 4.44% and 4.06% respectively. Low values of the deviations establish the accuracy of the experimental set up for the collection of heat transfer and flow friction data.

5.2 Flow patterns in duct roughened on one broad side

In order to have an idea about the anticipated flow disturbances caused by various versions of the V-down rib type roughness pattern, perceived sketches have been drawn in Figs. 5(a-e). Figure 5a shows the flow disturbances caused by the application of continuous V-down rib. As explained by Taslim et al. (1996) and Lei et al. (2012), application of V-down rib causes a portion of the flow to move along the rib (called secondary flow) in the shape of counter-rotating vortices towards the apex of the V-rib. Counter-rotating vortices coming from two leading edges of the V-rib interfere with each other near the apex and a portion of them move toward the depth of the duct to form a pair of vertical loops, while the remaining portion jumps the rib along with the primary flow. These loops carry along relatively cool fluid from the middle region of the duct and force it to the absorber plate near the leading edges, causing local heat transfer enhancement. On the other hand, sweeping of hot fluid near the surface along the ribs by counter-rotating vortices causes poor heat transfer near the trailing edge region. So, a continuous V-rib roughness is characterized by high rates of heat transfer near the leading edges and poor rate of heat transfer near the trailing edges.

As shown in Fig. 5b, cutting of 2 gaps in the V-rib causes refreshment of secondary flow moving along the ribs, thereby releasing the hot fluid accumulated along the rib through the gap near the trailing edge. This causes improvement in the rate of heat transfer near the trailing edge. In addition to this accelerated flow through the small gap also prevents the redevelopment of boundary layer beyond the reattachment point (approximately at P/e = 4 to 5) and hence leads to further improvement in the heat removal rate. Application of two staggered rib pieces in front of these gaps (as shown in Fig. 5c) causes additional disturbance to the flow after the reattachment point and as a result can further enhance the rate of heat removal in the span between two adjacent V-ribs. Increasing the number of gaps to four (as shown in Fig. 5d) will increase the benefits from the refreshment of the piled up flow along the ribs at multiple locations at the cost of reducing the strength of secondary flow vortices. Further in the absence of staggered rib pieces, four gaps geometry will allow obstacle free passage to considerable amount of primary flow, which will reduce the flow friction and at the same time may also have adverse effect on the heat removal rate from the absorber plate. Application of four gaps combined with the staggered ribs roughness is supposed to result in very complicated flow patterns near the wall as shown in Fig. 5e). It is believed that flow patterns responsible for the enhancement of heat transfer are 1) reattachment of the primary flow behind the V-rib 2) reattachment of the primary flow behind the staggered rib piece 3) movement of the secondary flow along the V-rib and its refreshment through the gap 4) acceleration of the primary flow through the gaps 5) generation of the counter rotating secondary flow vortices across the cross section of the duct and 6) generation of vortices at the edges of staggered rib pieces. On the other hand, presence of the recirculation region behind the V-ribs and decrease in the length of the main V-rib for higher number of gaps configurations; lowers the heat transfer from the absorber plate. All these alterations to the flow are supposed to enhance the rate of heat transfer along with some pressure drop penalty.
**Fig. 5.** Flow patterns at various locations for different types of ribs arrangements a) continuous V-rib (type I) b) V-rib with two gaps (type II) c) V-rib with two gaps combined with staggered pieces (type III) d) V-rib with four gaps (type IV) and e) V-rib with four gaps combined with staggered pieces (type V)

### 5.3 Effect on heat transfer

Figure 6 shows the plots between Nusselt number and Reynolds number for various variations of the V-down rib. It is evident from the figure that, introduction of two small gaps (one on each leg of the V-rib) in the continuous V-rib, results in enhancement of the Nusselt number. This enhancement is primarily due to the refreshment of the air moving along the rib (secondary flow) through the gaps near the trailing ends of the V-rib. Taslim et al. (1996) observed that, heat transfer from the heated surfaces roughened with V-down ribs is low near the trailing ends, due to heating up of the air as it moves along the ribs from leading to trailing ends. Aharwal et al. (2009) and Singh
et al. (2011) also concluded that introduction of a gap near the trailing end of the inclined and V-shaped ribs respectively, enhances the rate of heat transfer from the absorber plate of a solar air heater. As shown in the figure, fixing of a small staggered rib piece in front of each gap at a distance of $2/3$ of the pitch, further improves the rate of heat transfer. The observation is in agreement with the findings of Patil et al. (2012), who performed extensive experiments with this type of geometry and stated that, fixation of a staggered rib piece in front of the gap in V-down rib increases the level of turbulence in the span between two adjacent V-ribs and causes enhancement of heat transfer from this region.

Introduction of four gaps (two on each leg of the V-rib) resulted in lowering the Nu than the continuous V-rib. It is presumed that, four gaps in the V-rib cause considerable amount of air to pass through without encountering the ribs, thereby depriving it from the heat transfer enhancement benefits of the rib roughness. It seems that refreshment of the secondary flow at multiple locations through the gaps is not enough to enhance the rate of heat transfer. Further division of each leg of the V-rib into three pieces reduces the strength of the secondary flow vortices (Lee et al. (2009)), thereby diluting the effect of V-shaping the transverse rib on the enhancement of heat transfer. Next variation of the V-rib consisted of fixation of staggered ribs in front of each of the four gaps. Experiments with this type of geometry resulted in huge enhancement of Nu as compared to all other geometries tested here. It seems that this type of geometry optimally combines the benefits of gaps and staggered ribs by effectively decreasing the role of factors (redevelopment of boundary layer after flow separation from ribs, low heat transfer near trailing ends of the V-ribs etc.) responsible for reducing the heat transfer and enhancing the contribution of factors (refreshment of secondary flow through gaps to enhance the heat transfer near the trailing ends and stopping the redevelopment of boundary layer, increase in the turbulence in the region between adjacent V-ribs by vortex shedding from staggered ribs) responsible for increasing the heat transfer.

![Variation of Nusselt number with Reynolds number](image.png)

**Fig. 6.** Variation of Nusselt number with Reynolds number for various alterations of the V-down rib
Results of Fig. 6 have been rearranged in Fig. 7, to clearly show the changing trend of Nusselt number with the type of rib geometry pattern at various Reynolds numbers. Figure 7 clearly shows increase in Nusselt number by transformation of rib pattern from type I to type II & III respectively. For type IV, Nusselt number plunges considerably, but addition of the staggered ribs in the type V pattern enhance the Nusselt number by 52.3-96.2% as compared to the type IV geometry.

5.4 Effect on friction factor

Figure 8 shows the plots between friction factor and Reynolds number for various variations of the V-down rib. In general, friction factor decreases with increase in the Reynolds number for all type of rib geometries tested here. This behavior is in agreement with the literature available on average skin friction coefficient for fully developed turbulent flow over roughened surfaces (Nikuradse (1958)), where it is stated that, increase in the Reynolds number (i.e. flow velocity) reduces the thickness of the viscous sub-layer (major contributor to the friction factor for moderate relative roughness height) which in effect lowers the friction factor. It is seen that cutting of two gaps in the continuous V-rib (one on each side of the apex), increases the overall turbulence over the absorber plate (mainly due to acceleration of the primary flow through the gaps and subsequent disturbance to the flow in the span between upstream and downstream ribs) and hence higher friction factor. Addition of a small rib in-front of each gap further increases the turbulence level due to the separation of the flow over the staggered ribs and generation of wake region vortices. These vortices are supposed to move from the leading to the trailing edges of these rib pieces and hence creating flow disturbance near the trailing edge of the main V-rib. This increase in the friction factor for V-rib with gaps and V-rib with gaps combined with staggered rib pieces also complements the increase in Nusselt number as shown in figure 6 above.

Further experiments were performed by cutting two more gaps in the V-rib, totaling to 4 gaps (two on each side of the apex) and it lead to lowering the friction factor as compared to the continuous V-rib. It is believed that cutting of four gaps caused a considerable amount of flow to pass through
without striking the roughened elements and hence overall flow resistance came down. It seems that, in case of more than two gaps, the effect of increase in the flow friction due to disturbance of the flow in the span between two adjacent V-ribs by the accelerated flow coming through the gaps is dominated by the decrease in flow friction on account of resistance free passing of flow through the gaps. Decrease in flow friction also causes decrease in the Nusselt number values for four gaps geometry as compared to the continuous V-rib as shown in figure 5. In the next variation of the V-rib geometry, staggered rib pieces were fixed in front of each gap in case of four gaps configuration. As a result, friction factor increased by 20-25 % as compared to the continuous V-rib configuration. This happened despite the fact that friction factor for four gaps only configuration was lower than the continuous V-rib geometry. It seems that staggered rib pieces played a major role in enhancing the turbulence in the flow by obstructing the accelerated flow coming through the gaps and associated vortex shedding from the trailing edges of the staggered rib pieces. Increase in friction factor for this configuration is also complemented by huge enhancement (44-64 %) of Nusselt number for this configuration as compared to the continuous V-rib shown in figure 7 above.

**Fig. 8.** Variation of friction factor with Reynolds number for various variations of the V-down rib
Fig. 9. Variation of friction factor with type of rib geometry for various Reynolds numbers

Results of Fig. 8 have been reproduced in Fig. 9, to clearly show the changing trend of friction factor with the type of rib geometry pattern at various Reynolds numbers. The changing trend of Fig. 9 is in accordance with the trend shown in figure 7. This proves that increase or decrease of Nusselt number for various versions of the V-down rib roughness pattern depends upon the change in friction factor as per the level of disturbance in the flow. Figure 8 clearly shows increase in friction factor by transformation of rib pattern from type I to type II & III respectively. For type IV, friction factor plunges considerably, but addition of the staggered ribs in the type V pattern enhance the friction factor by 38.8-26.4% as compared to the type IV geometry as Reynolds number increases from 4000 to 12000.

5.5 Effect on thermohydraulic performance

Since objective of the enhancement of the performance of SAH is to maximize the rate of heat transfer from the absorber plate with minimum increase in pressure drop (friction factor), so comparison of the thermohydraulic performance of various alterations of the V-down rib is an appropriate performance appraisal tool. Figure 10 shows the relative values of the thermohydraulic performance parameter for various V-rib configurations. It is observed that, thermohydraulic performance increased by 8.2-13.8% and 13.7-20.4% for V-rib with two gaps and V-rib with two gaps combined with staggered ribs patterns respectively as compared to the continuous V-rib. It is also noted that, rate of increase of thermohydraulic performance parameter was sharp till Reynolds number of 6000, whereas after that it remained almost constant with slight decrease for higher Reynolds number values. This is due to continuously decreasing rate of increase of Nusselt number after Reynolds number of 8000 for these rib configurations as shown in Fig. 6. Better performance of these rib configurations is in agreement with the findings of Singh et al. (2011) and Patil et al. (2012) as reported in their studies with experimental investigations with similar geometries.
Thermohydraulic performance parameter values of V-rib with four gaps geometry decreased by 1.5-14.9% as compared to the continuous V-rib geometry. This is due to lower Nusselt number values for this geometry as compared to the continuous V-rib geometry as shown in Fig. 6. Further fixation of staggered ribs in front of the four gaps resulted in 34.6-54.5% increase in the thermohydraulic performance as compared to the continuous V-down rib. Another difference of the performance of this version of the modified V-rib geometry as compared to the
other modifications is the continuous increase in the value of thermohydraulic performance parameter in the entire range of the Reynolds numbers evaluated, thus making this geometry suitable for good performance across entire range of Reynolds numbers (3000-15000) commonly encountered in solar air heater applications (Karwa, 2003; Gupta et al., 1993; Singh et al., 2011). In order to have better clarity about the effect of changes in the continuous V-down rib geometry on the thermohydraulic performance, variation of thermohydraulic performance with type of V-down rib roughness pattern is shown in Fig. 11. It is clear that thermohydraulic performance increases up to type III and then it goes down for type IV rib pattern, but again jumps to the highest level for type V roughness pattern by 36.6-81.6% as compared to the type IV geometry.

6. Conclusions

Present study is aimed at the performance appraisal of some design variations of the V-down rib roughness pattern, when applied to one broad wall of a rectangular solar air heater duct under constant heat flux conditions. Findings from the detailed experimental investigations can be concluded as given below:

1) Some careful modifications in the already good performing V-down rib roughness geometry can further enhance its performance.
2) Cutting of small symmetric gaps in the continuous V-down rib, do not always result in the enhancement of its performance, but fixation of small staggered rib pieces in front of the gaps always ensure enhancement in its performance, e.g. cutting of two gaps in the V-down rib resulted in 8.2-13.8% increase in the thermohydraulic performance but, cutting of four gaps resulted in 1.5-14.9% decrease in performance in the range of Reynolds numbers investigated. On the other hand fixing of staggered ribs in case of two gaps geometry resulted in 13.7-20.4% and in case of four gaps geometry 34.6-54.5% improvement in the performance respectively.
3) Maximum values of the Nusselt number, friction factor and thermohydraulic performance parameter were observed for type V (four gaps combined with staggered rib pieces) rib roughness pattern, whereas, minimum values for corresponding performance indices were noted for type IV (V-down rib with four gaps) rib roughness pattern.
4) Four gaps combined with staggered ribs geometry resulted in considerable enhancement in the thermohydraulic performance (18.3-28.3%) as compared to the two gaps combined with staggered ribs geometry.
5) V-down rib having four gaps combined with staggered ribs geometry returned thermohydraulic performance parameter value of more than 2 across the entire range of Reynolds numbers investigated and it continued to increase with increase in the Reynolds number.

Nomenclature

\[ A_o \] orifice cross-sectional area, m\(^2\)
\[ A_p \] absorber plate area, m\(^2\)
\[ AR \] duct aspect ratio
\[ C_d \] orifice discharge coefficient
Acknowledgements

Authors convey their sincere gratitude to Dr. BR Ambedkar National Institute of Technology for their financial support for executing the experimental study.
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