Available online at www.sciencedirect.com

ScienceDirect

Renewable Energy 33 (2008) 585–596

RENEWABLE ENERGY

www.elsevier.com/locate/renene

Experimental investigation on heat-transfer enhancement due to a gap in an inclined continuous rib arrangement in a rectangular duct of solar air heater

K.R. Aharwal[a,*], B.K. Gandhi[b], J.S. Saini[b]

[a]*Department of Mechanical Engineering, S.G.S.I.T.S., Indore (MP) 452003, India*
[b]*Department of Mechanical and Industrial Engineering, I.I.T., Roorkee (UA) 247667, India*

Received 15 December 2006; accepted 21 March 2007
Available online 1 June 2007

Abstract

Artificial roughness in the form of repeated ribs has been proposed as a convenient method for enhancement of thermal performance of solar air heaters. This paper presents the experimental investigation of heat transfer and friction factor characteristics of a rectangular duct roughened with repeated square cross-section split-rib with a gap, on one broad wall arranged at an inclination with respect to the flow direction. The duct has a width to height ratio (W/H) of 5.84, relative roughness pitch (P/e) of 10, relative roughness height (e/D_h) of 0.0377, and angle of attack (α) of 60°. The gap width (g/e) and gap position (d/W) were varied in the range of 0.5–2 and 0.1667–0.667, respectively. The heat transfer and friction characteristics of this roughened duct have been compared with those of the smooth duct under similar flow condition. The effect of gap position and gap width has been investigated for the range of flow Reynolds numbers from 3000 to 18,000. The maximum enhancement in Nusselt number and friction factor is observed to be 2.59 and 2.87 times of that of the smooth duct, respectively. The thermo-hydraulic performance parameter is found to be the maximum for the relative gap width of 1.0 and the relative gap position of 0.25.

Keywords: Gap position; Gap width; Reynolds number; Nusselt number; Friction factor; Thermo-hydraulic performance

1. Introduction

The thermal performance of a conventional solar air heater is generally poor because of low convective heat-transfer coefficient between air and the absorber plate. The low value of heat-transfer coefficient is generally attributed to the presence of a viscous sublayer, which can be broken by providing artificial roughness on the heat-transferring surface [1]. However, the artificial roughness results in higher frictional losses leading to excessive power requirement for the fluid to flow through the duct. It is, therefore, desirable that turbulence must be created only in a region very close to the heat-transferring surface to break the viscous sublayer for augmenting the heat transfer, and the core flow should not be unduly disturbed to limit the increase in friction losses. This can be done by keeping the height of the roughness elements small in comparison to the duct dimensions [2]. The application of the artificial roughness in a solar air heater owes its origin to several investigations [3–6] carried out for the enhancement of cooling of the turbine blades' passage. Han et al. [3] investigated the effect of rib shape, angle of attack, and pitch to rib height ratio (P/e) on friction factor and heat-transfer characteristics of a rectangular duct with two side-roughened walls. They reported that the maximum value of heat transfer and friction factor occurs at a relative roughness pitch of 10, and the ribs oriented at a 45° angle. Lau et al. [4] also observed that the replacement of continuous transverse ribs by inclined ribs in a square duct results in higher turbulence at the ribbed wall due to interaction of the primary and secondary flows. Han et al.

*Corresponding author. Department of Mechanical and Industrial Engineering, c/o Dr. B.K. Gandhi M.I.E.D., I.I.T., Roorkee (UA) 247667, India. Tel.: +91 9411121364; fax: +91 1332 285665.
E-mail address: kraharwal@yahoo.com (K.R. Aharwal).

0960-1481/$ - see front matter
doi:10.1016/j.renene.2007.03.023

Nomenclature

A_o	cross-section area of orifice, m^2
A_p	area of absorber plate, m^2
b	width of the rib, m
C_d	coefficient of discharge of orifice
C_p	specific heat of air at constant pressure, J/kg K
D_h	hydraulic diameter of duct, $m[= (4WH/2(W+H))]$
e	rib height, m
e/D_h	relative roughness height
f_s	friction factor of smooth duct
f	friction factor of roughened duct
g	gap width, m
H	depth of duct, m
h	convective heat-transfer coefficient, $W/m^2 K$
$(\Delta h)_o$	difference of manometric fluid levels in U-tube manometer, m
$(\Delta h)_d$	difference of water column levels in micro-manometer, m
k	thermal conductivity of air, W/m K
L	test section length for pressure drop measurement, m
m	mass flow rate, kg/s
Nu	Nusselt number of roughened duct
Nu_s	Nusselt number of smooth duct
P	pitch of the rib, m
P/e	relative roughness pitch
Q_u	useful heat gain rate, W
T_f	mean temperature of air, K $\{=(T_i+T_o)/2\}$
T_i	inlet temperature of air, K
T_o	outlet temperature of air, K
T_p	average plate temperature, K
V	velocity of air, m/s
W	width of duct, m
α	angle of attack , (°)
β	ratio of orifice diameter to pipe diameter
η	efficiency parameter
ρ	density of air, kg/m^3
ρ_m	density of manometric fluid, kg/m^3
ρ_w	density of water, kg/m^3

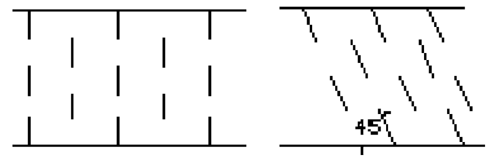
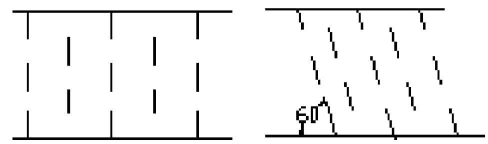
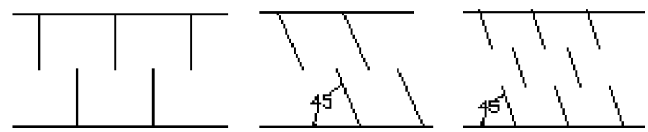
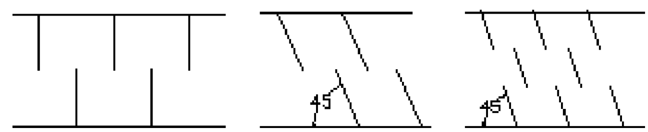
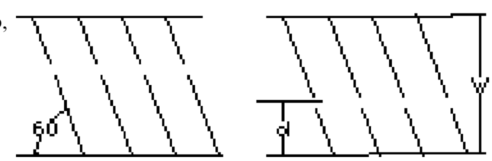
[6] investigated the combined effect of rib angle and channel aspect ratio. They reported that the maximum heat transfer and pressure drop is obtained at an angle of attack of 60° and a square channel provides a better heat-transfer performance than the rectangular channel. Zhang et al. and Kiml et al. [7,8] reported that the thermal performance of rib arrangements with an angle of attack of 60° is better than that with an angle of 45°. For a square duct, Han et al. [9] reported that 45° or 60° V-shaped ribs facing upward show higher heat transfer compared to corresponding V-shaped ribs facing downward. They found that V-shaped ribs facing upward forms two pairs of rotating cells along each divergent axis of rib, while in the case of V-shaped ribs facing downward, two pairs of counter-rotating cells merge resulting in a higher pressure drop and lower heat transfer. Taslim et al. [10] investigated the heat transfer and friction factor characteristics of a channel roughened with angled and V-shaped ribs. They found that V-shaped ribs pointing downward have a much higher heat-transfer coefficient because the warm air being pumped toward the rib-leading region increases the apex region heat-transfer coefficients as compared to that of the leading end region. Goa and Sunden [11] have also reported that the V-shaped ribs pointing downward perform better than the rib pointing upward.

Lau et al. [4,5] investigated the heat-transfer and friction factor characteristics of fully developed flow in a square duct with transverse and inclined discrete ribs. They reported that a five-piece discrete rib with 90° angle of attack shows 10–15% higher heat-transfer coefficient as compared to the 90° continuous ribs, whereas inclined discrete ribs give 10–20% higher heat transfer than that of the 90° discrete rib. Hu et al. [12] investigated the effect of inclined discrete rib with and without groove and reported that discrete rib arrangement without groove shows better performance than that of the discrete rib with groove. Han et al. [7] carried out experiments to study the heat transfer and pressure drop characteristics of a roughened square channel with V-shaped broken rib arrangement with the angle of attack of 45° and 60° and reported that 60° V-shaped broken rib arrangement gives better performance than 45° V-shaped broken rib arrangement.

Chao et al. [13] examined the effect of angle of attack and number of discrete ribs, and reported that the gap region between the discrete ribs accelerates the flow, which increases the local heat-transfer coefficient. In a recent study, Cho et al. [14] investigated the effect of a gap in the inclined ribs on heat transfer in a square duct and reported that a gap in the inclined rib accelerates the flow and enhances the local turbulence, which will result in an increase in the heat transfer. They reported that the inclined rib arrangement with a downstream gap position shows higher enhancement in heat transfer compared to that of the continuous inclined rib arrangement. Table 1 summarizes the various arrangements of discretizing the inclined ribs employed by these investigators.

Most of the investigations carried out so far have applied the artificial roughness on two opposite walls with all four walls being heated. It is noted that for the application of this concept of enhancement of heat transfer in the case of solar air heaters, roughness elements have to be considered only on one wall, which is the only heated wall comprising the absorber plate. These applications make the fluid flow and heat-transfer characteristics distinctly different from those found in case of two roughened walls and four heated wall duct. In the case of solar air heaters, only one wall of

Table 1
Discretizing arrangements of inclined ribs

Investigators	Roughness parameters	Roughness geometry
Lau et al. [4]	$(P/e) = 10$, $(e/D) = 0.625$ $W/H = 1.0$ $\alpha = 90^\circ$ and 45° $Re = 1000$–80000	
Lau et al. [5]	$(P/e) = 10$ $(e/D) = 0.625$ $W/H = 1.0$ $\alpha = 90^\circ$ and 60° $Re = 10000$–80000	
Han et al. [7]	$(P/e) = 10$ $(e/D) = 0.625$ $W/H = 1.0$ $\alpha = 90^\circ$, 60° and 45° $Re = 15000$–90000	
Cho et al. [13]	$(P/e) = 8$, $(e/D) = 0.0743$ $W/H = 2.04$ $\alpha = 90^\circ$ and 45° $Re = 25000$–70000	
Cho et al. [14]	$(P/e) = 8$, Gap position $= W/3$ and $2W/3$, gap width = width of rib, $(e/D) = 0.8$, $W/H = 1.0$ $\alpha = 60^\circ$ $Re = 25000$–70000	

the rectangular air passage is subjected to uniform heat flux (insolation) while the remaining three walls are insulated. Many investigators [15–21] have employed artificial rib roughness in various forms on the airflow side of the absorber plate to enhance the thermal performance of solar air heaters. Studies carried out by these researchers have shown that the geometry of the rib, namely shape, pitch, angle of attack and height, affects significantly the heat transfer and friction characteristics of the duct.

In view of the above, it can be stated that discrete inclined or V-shaped rib arrangement can yield better performance as compared to continuous rib arrangement. However, investigations have not been carried out so far to optimize the gap width between the rib elements to form the discrete rib and also to locate the optimum position of this gap. The present investigation was therefore taken up to determine the optimum location and width of gap in an inclined rib to form a discrete rib. This study will help in determining the gap size and position while discretizing the inclined (non-transverse) ribs for enhancing the performance as compared to non-discretized ribs. In the present work, experimental investigation on the performance of solar air heater ducts, having the absorber plate with artificial roughness in the form of inclined rib, provided with and without a gap, has been carried out. The flow Reynolds number has been varied between 3000 and 18,000. The variations of Nusselt number and friction factor as a function of roughness parameters including gap position and gap width have been evaluated to examine the thermo-hydraulic performance of the system to ascertain the benefit of this selected roughness geometry.

2. Experimental program

An experimental test facility has been designed and fabricated to study the effect of gap position and gap width of inclined rib geometry on the heat transfer and fluid flow characteristics of a rectangular duct. A schematic diagram of experimental setup is shown in Fig. 1. It consists of an entry section, test section, exit section, transition section, a flow measuring orifice-meter and a centrifugal blower with a control valve. The wooden rectangular duct has an internal size of 2600 mm × 182 mm × 31 mm, which consists of an entrance section, a test section and an exit section of length 800, 1200 and 600 mm, respectively, in accordance with the recommendation of ASHARAE standard 93–77 [22]. In the exit section, three equally spaced baffles are provided for the purpose of mixing the delivered air. A 6 mm-thick heated aluminum plate is used as the top broad wall of the test section whereas the top wall of entry and exit sections of the duct were made of 12 mm-thick plywood. All other sides of the entry, test and exit sections are 50 mm-thick wooden walls. The absorber plate is heated from the top by supplying uniform heat flux

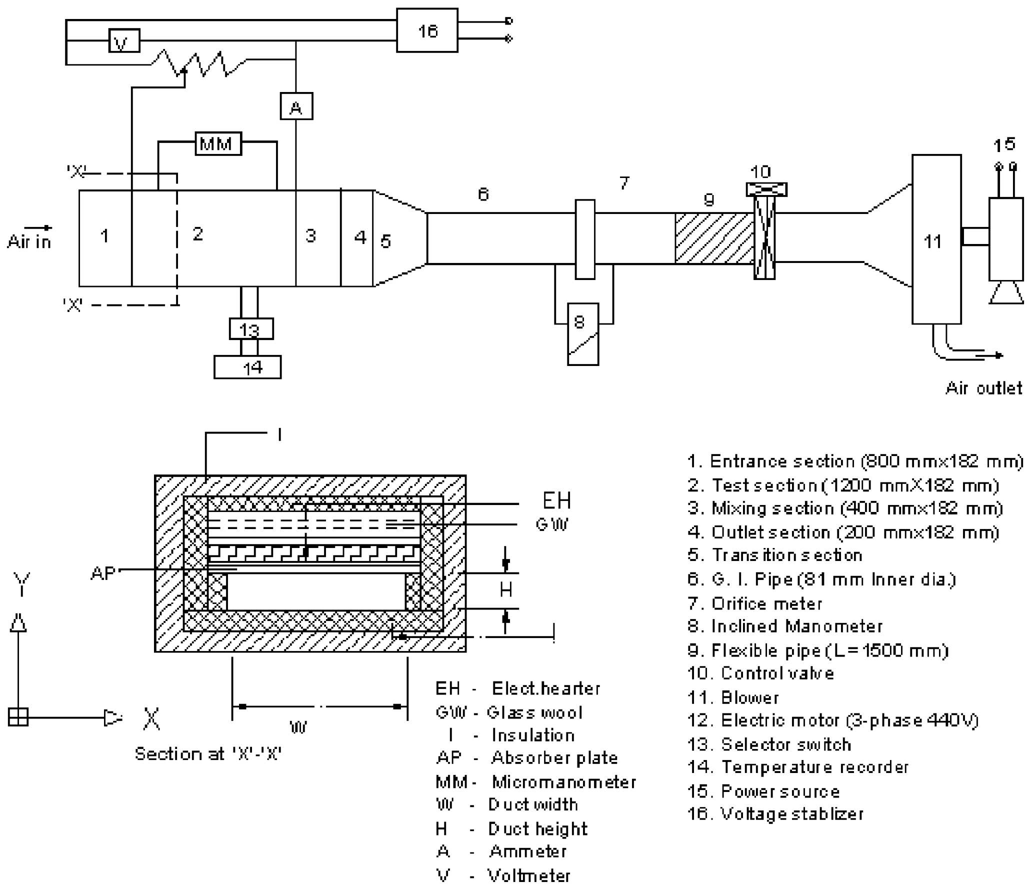


Fig. 1. Schematic diagram of experimental set-up.

(1000 W/m^2) by means of an electrical heater and is insulated with 50 mm-thick glass wool topped with 12 mm-thick plywood. A calibrated orifice-meter connected with a U-tube manometer using kerosene as manometric fluid is used to measure the mass flow rate of air. A control valve is provided in the pipe used to connect the blower to regulate the flow rate. Calibrated thermocouples, prepared by butt welding of 0.36-mm-diameter copper–constantan wires, have been used for temperature measurement. Sixteen thermocouples were provided along the axial center line of the absorber plate by drilling about 2.0 mm-diameter holes around 3–4 mm deep at the back of the plate to measure the temperature. Five thermocouples are arranged span wise in the duct to measure the air temperature after the mixing section and one thermocouple is used to determine the inlet temperature of the air.

To minimize the percentage error in measurement of temperatures, minimum heat flux value is so selected as to raise the temperature of air by about 8 °C in the test section. The temperature difference between the heated plate and the bulk air was observed to be above 20 °C. The airflow rate was varied to give the flow Reynolds number in the range of 3000–18,000.

Data were noted under the steady-state condition, which was assumed to have reached when the plate and air temperatures showed negligible variation for around a 10-min duration. The steady state for each test run was obtained in about 1.5–2 h.

Table 2
Values of parameters

Sr. no.	Parameters	Values
1	Relative roughness pitch (P/e)	10
2	Rib height (e) and width (b)	2 mm
3	Relative roughness height (e/D_h)	0.0377
4	Duct aspect ratio (W/H)	5.87
5	Reynolds number (Re)	3000–18000
6	Angle of attack (α)	60°
7	Relative gap position (d/W)	0.167–0.5 (4 steps)
8	Relative gap width (g/e)	0.5–2 (4 steps)

3. Roughness geometry and range of parameters

The values of system and operating parameters of this investigation are listed in Table 2. The relative roughness pitch (P/e) value is selected as 10, based on the optimum value of this parameter reported in the literature [3]. Similarly, the value of the angle of attack is chosen as 60°, to achieve maximum enhancement of heat transfer [6].

The arrangements of ribs on the absorber plate are shown in Fig. 2(a–f). In order to investigate the effect of the

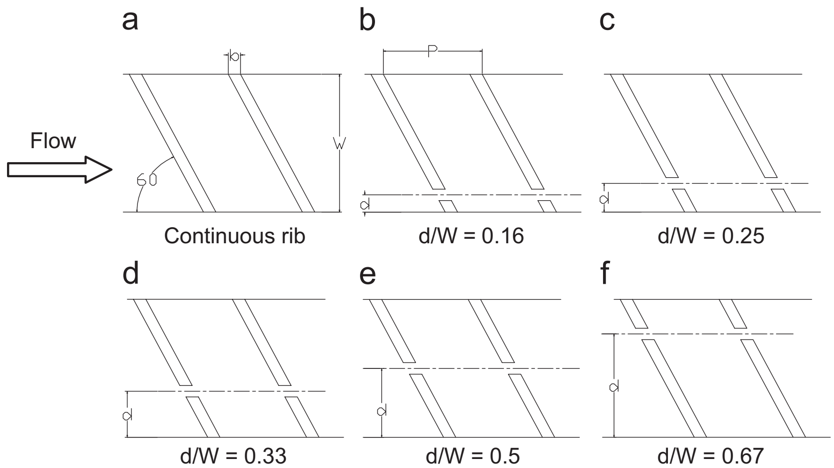


Fig. 2. Variation of gap position in an inclined rib arrangement.

gap position and gap width on the enhancement of heat transfer and friction factor, the relative gap position (d/W) is varied from 1/6 to 2/3 of the width of duct from the trailing edge of the rib, while the relative gap width (g/e) is varied from 0.5 to 2.0. The range of Reynolds number and relative roughness height (e/D_h) has been chosen based on the requirement of the solar air heater [2].

4. Data reduction

The following procedure is applied for the determination of heat-transfer coefficient 'h', useful heat gain 'Q'$_u$, Nusselt number 'Nu', Reynolds number 'Re', friction factor 'f', and collector thermo-hydraulic performance parameter 'η'.

The heat-transfer coefficient for the heated section was calculated from the equation

$$h = \frac{Q_u}{A_p(T_p - T_f)}, \tag{1}$$

where the rate of heat gain by the air 'Q_u' is given by,

$$Q_u = mC_p(T_o - T_i). \tag{2}$$

The electrical heating gives a condition of uniform heat flux boundary condition and the plate temperature varies in the direction of the airflow. In all calculations, a mean plate temperature 'T_p' is calculated as a weighted mean of the plate temperature measured at different locations and 'T_f' is the bulk mean air temperature.

Mass flow rate, m, has been determined from the pressure drop measurement across the orifice plate.

$$m = C_d \times A_o[2\rho\rho_m g(\Delta h)_o/(1-\beta^4)]^{0.5}, \tag{3}$$

where C_d is the discharge coefficient and is evaluated as 0.603 by calibration.

Heat-transfer coefficient has been used to determine the Nusselt number using the equation:

$$Nu = \frac{hD_h}{k}, \tag{4}$$

where $D_h = (4WH/2(W+H))$ is the hydraulic diameter of the rectangular duct.

The Reynolds number was determined from the value of the mass velocity G, using the equation:

$$Re = \frac{GD_h}{\mu}, \tag{5}$$

where $G = (m/WH)$.

The friction factor was determined from the flow velocity 'V' and the head loss 'Δh_d' measured across the test section length of 1 m using the Darcy–Weisbach equation as

$$f = \frac{2(\rho_w g \Delta h_d)D_h}{4\rho L V^2}. \tag{6}$$

The Stanton number was calculated from the following equation:

$$St = \frac{Nu}{RePr}, \tag{7}$$

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

Based on the analysis of the errors in the experimental measurements through various instruments employed, the uncertainties in the calculated values of Reynolds number, Nusselt number, friction factor Stanton number and thermo-hydraulic performance parameter are estimated as $\pm 1.8\%$, $\pm 2.45\%$, $\pm 3.25\%$, $\pm 3.06\%$ and $\pm 2.04\%$, respectively [23].

5. Validation of experimental data

The values of Nusselt number and friction factor determined from experimental data for smooth duct have been compared with the values obtained from Dittus–Boelter equation for the Nusselt number and modified Blasius equation for the friction factor.

The Nusselt number for a smooth rectangular duct is given by the Dittus–Boelter equation as

$$Nu_{\mathrm{s}} = 0.023Re^{0.8}P_{\mathrm{r}}^{0.4}. \quad (8)$$

The friction factor for a smooth rectangular duct is given by the modified Blasius equation as

$$f_{\mathrm{s}} = 0.085Re^{-0.25}. \quad (9)$$

The comparison of the experimental and estimated values of the Nusselt number and friction factor as a function of the Reynolds number is shown in Fig. 3(a) and (b), respectively. The average deviation of experimental values of the Nusselt number is $\pm 2.8\%$ from the values predicted by Eq. (8), and the average deviation of experimental values of the measured friction factor is $\pm 2.3\%$ from the values predicted by Eq. (9). Thus, reasonably good agreement between the two sets of values ensures the accuracy of the data being collected with the experimental setup.

6. Results and discussion

6.1. Heat transfer

The effect of gap width and its location in case of an inclined rib for an artificially rib-roughened duct has been investigated on heat transfer and friction characteristics of the duct. The values of the Nusselt number and friction factor of the roughened ducts as a function of the Reynolds number have been compared with those of the smooth duct under similar experimental conditions. The thermo-hydraulic performance of the roughened duct has also been evaluated.

Fig. 4 shows the values of the Nusselt number as a function of relative gap width (g/e) for different relative gap positions (d/W) for a 60°-inclined rib-roughened duct, at few selected Reynolds numbers. It is observed that at any relative gap width, the Nusselt number is the highest for the relative gap position of 0.25 for all Reynolds numbers. Similarly, the variation of Nusselt numbers with relative gap width shows that the value of the Nusselt number is the maximum for the relative gap width of 1.0 for all Reynolds numbers.

To compare the enhancement of the Nusselt number of the roughened duct with and without gap for the inclined rib arrangement with that of the smooth duct, the values of the Nusselt number ratio (Nu/Nu_{s}) for different values of the relative gap position and relative gap width is presented in Figs. 5 and 6, respectively.

Fig. 5 shows the effect of relative gap width (g/e) on the Nusselt number ratio at the relative gap position (d/W) of

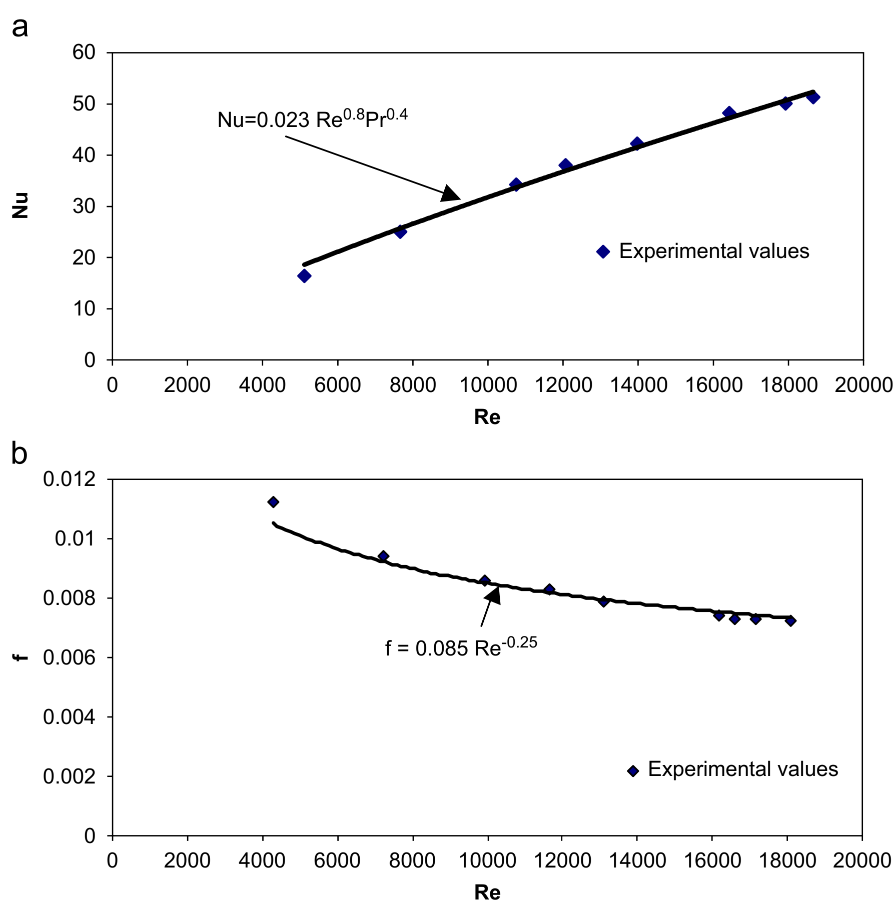


Fig. 3. (a) Comparison of experimental and estimated values of Nusselt number of smooth duct. (b): Comparison of experimental and estimated values of friction factor of smooth duct.

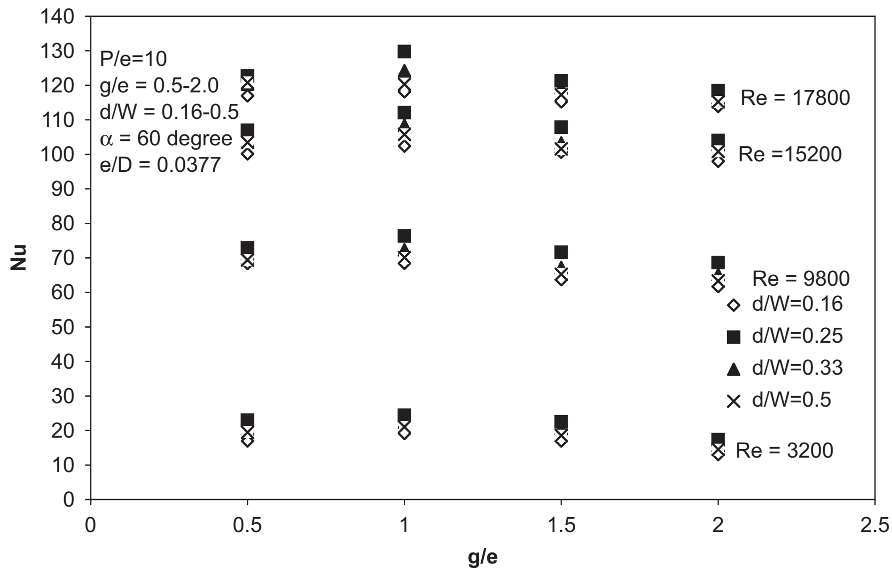


Fig. 4. Effect of relative gap width and relative gap position on Nusselt number at selected Reynolds numbers.

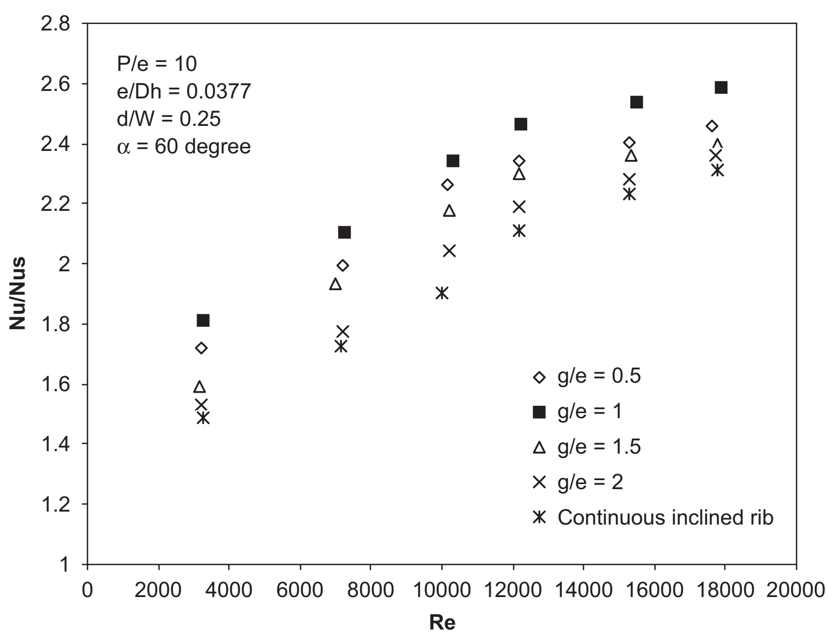


Fig. 5. Effect of relative gap width on Nusselt number at the relative gap position of 0.25.

0.25. This figure shows that the Nusselt number ratio increases with increase in the relative gap width up to about 1.0, beyond which it decreases with increase in the relative gap width. The Nusselt number ratio for inclined rib with gap lies in the range of 1.52–2.58 for the range of Reynolds numbers of the present investigation. The value of Nusselt number ratio is higher for a relative gap width of 1.0 and lower for the relative gap width of 2.0. This may be due to fact that as we increase the relative gap width beyond 1.0, the flow velocities through the gap will reduce, which may not be strong enough to accelerate the flow through the gap and hence the heat transfer due to this flow may not be increased significantly in comparison to that of the continuous ribs. Whereas when we reduce the relative gap width lower than 1.0 (i.e. $g/e = 0.5$), it may leave very little space for flow of the fluid through it, which results in low turbulence and hence reduce the enhancement of heat transfer. It is clear from this discussion that for the enhancement of heat transfer, the width of the gap should be maintained such that it can increase the velocity of the fluid passing through it in order to create the local turbulence. By keeping the relative gap width of $g/e = 1.0$ in the present study, the maximum heat transfer is obtained for a 60°-inclined square rib-roughened duct. This may be the optimum relative gap width for this rib configuration and flow conditions.

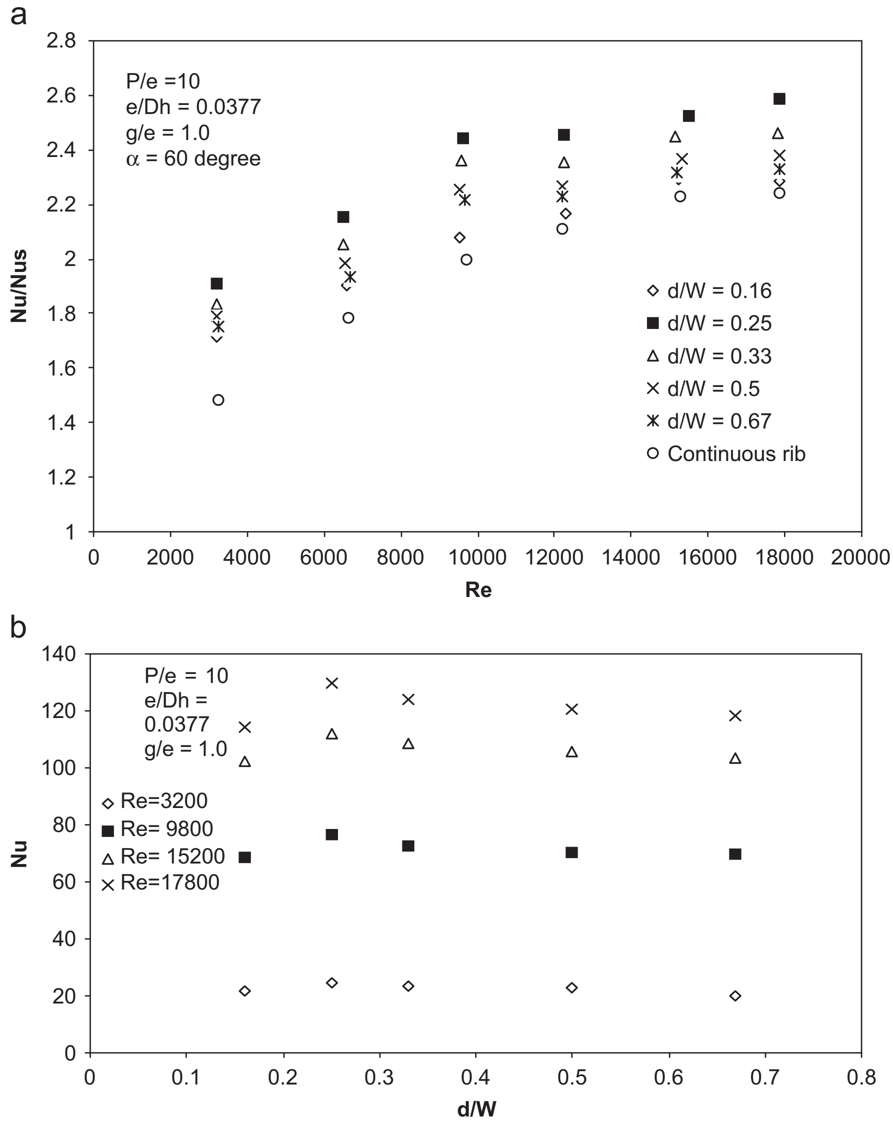


Fig. 6. (a) Effect of relative gap position on Nusselt number ratio. (b) Effect of relative gap position on Nusselt number.

Fig. 6(a) shows the effect of relative gap position (d/W) on the Nusselt number ratio for a fixed relative gap width (g/e) of 1.0. It can be seen that for any Reynolds number, the Nusselt number ratio is higher for a gap in the continuous rib as compared to that of the rib without gap and that the Nusselt number ratio increases with increase in relative gap position from 0.16 to 0.25, attains a maxima at a gap position of 0.25 and thereafter it decreases with an increase in the relative gap position. The Nusselt number ratios lie in the range of 1.71–2.59, while the corresponding values for continuous ribs lie in the range 1.48–2.26 under similar operating conditions.

To bring out the effect of gap position clearly, the variation of the Nusselt number with relative gap position is presented in Fig. 6(b) for the selected Reynolds number values where a clear maxima in plot can be observed. This is explained by the flow phenomenon as discussed below.

A continuous inclined rib in a rectangular duct gives rise to secondary flow along the rib length, which allows the working fluid to travel from leading edge to trailing edge of the rib. The flow along the rib is gradually heated and the boundary layer grows thicker. The flow turns downward from the sidewall and it completes the recirculation loop as shown in Fig. 7(a). The increase in the heat-transfer rate in this case is higher than that of the transverse rib for a similar rib cross-section and operating conditions due to additional heat transfer by the secondary flow recirculation [24].

The introduction of a gap in the inclined ribs allows release of secondary flow and main flow through the gap. The main flow is a developed flow with thicker boundary layer, and due to the presence of viscous sublayer, it leads to a low amount of heat transfer. In fact, the ribs are introduced to break this retarded flow and let it reattach again with the surface to enhance the heat transfer [1]. However, in case of gap in the inclined rib, the secondary flow along the rib joins the main flow to accelerate it, which energizes the retarded boundary layer flow along the

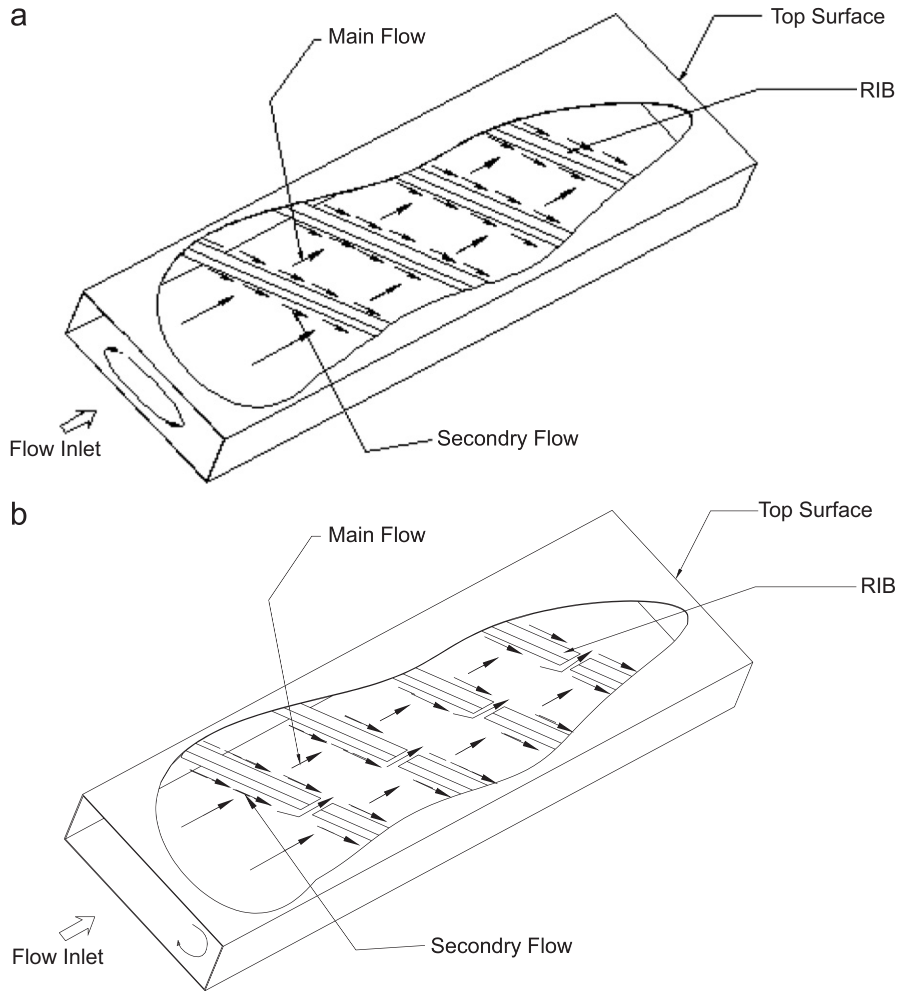


Fig. 7. (a) Flow pattern of secondary flow for inclined continuous rib. (b) Flow pattern of secondary flow for inclined discrete rib.

surface. This increases the heat transfer through the gap width area behind the rib.

If a gap is created near the leading edge (say at $d/W = 0.67$), the strength of the secondary flow may not be sufficient to energize the main flow passing through the gap and this gap position does not lead to significant increase in heat transfer as compared to that of the continuous ribs (see Fig. 6a). Lowering the relative gap position signifies shifting of the gap toward the trailing edge. This increases the strength of the secondary flow and heat transfer increases with decrease in the relative gap position up to 0.25 (see Fig. 6a). However, further decrease in the relative gap position results in decrease in heat transfer, because the fluid flowing along the rib is continuously heated and this reduces the heat-transfer rate as the gap position shifts toward the trailing edge of the rib. Further, the secondary flow in the remaining part of the rib develops a flow recirculation loop as shown in Fig. 7(b), whose strength decreases with decrease in the relative gap position. For a relative gap position (d/W) of lower than 0.25, the contribution of this recirculation loop may be insignificant. Thus it appears that at a relative gap position of 0.25, the contribution of secondary flow in energizing the main flow through the gap and recirculation loop in the remaining part of the rib is the maximum, which results in the highest values of Nusselt number.

To further investigate the mechanism of enhancement of heat transfer as a result of creating a gap, investigations have also been carried out on transverse rib-roughened surface with and without a gap in the ribs. For this study, two relative gap positions, namely $d/W = 0.25$ and 0.5 with the relative gap width of 1.0, have been selected to compare the enhancement in heat transfer for the case of continuous rib. The variation of Nusselt number with Reynolds number for the transverse ribs with and without a gap is shown in Fig. 8. It is seen from this figure that there is no significant change in Nusselt number for the continuous ribs and ribs with a gap. This shows that the creation of a gap in the transverse rib does not result in significant increase in the heat transfer in case of the transverse ribs. Thus, the increase in the heat transfer in case of inclined ribs can be fully attributed to the presence of secondary flow as discussed earlier.

However, Lau et al. [4] reported that a 90° discrete rib arranged in a staggered manner on the roughened walls, with a gap between the rib elements, enhances the heat transfer by 10–15% compared to that of the 90° continuous rib arrangement. It is reasoned that this enhancement of heat transfer occurred due to the separation of fluid through the end of the discrete rib.

6.2. Friction factor

The secondary flow exerts a measurable effect in disturbing the axial flow profile, which increases the friction coefficient in non-circular ducts [1]. A lower value of the friction factor has been reported for discrete ribs compared to the continuous ribs [18], which is probably due to the diminished secondary flow cells. The effect of the relative gap width on the friction factor of roughened ducts with Reynolds number at a relative gap position of 0.25 is presented in Fig. 9 in terms of friction factor ratio (f/f_s). It is seen that the friction factor ratio increases with increase in Reynolds number due to increase in flow turbulence, while this ratio increases with an increase in the relative gap width up to 1.0 and decreases with further increase in the relative gap width up to 2.0. The friction factor ratio for a relative gap width of 2.0 lies close to that observed for continuous inclined ribs, which could have resulted due to very weak flow through this large gap (i.e. $g/e = 2$). The range of the friction factor ratio is between 2.03 and 2.9 for the range of Reynolds number and gap position investigated.

The maximum value of friction factor is found at a relative gap width of 1.0 with a relative gap position of 0.25 for which the maximum value of Nusselt number was observed. It is therefore essential to consider the thermo-hydraulic performance of the roughened duct with inclined ribs of different gap widths to evaluate the optimum arrangement.

6.3. Thermo-hydraulic performance

Study of the heat transfer and friction characteristics of the roughened ducts shows that an enhancement in heat transfer is, in general, accompanied with friction power

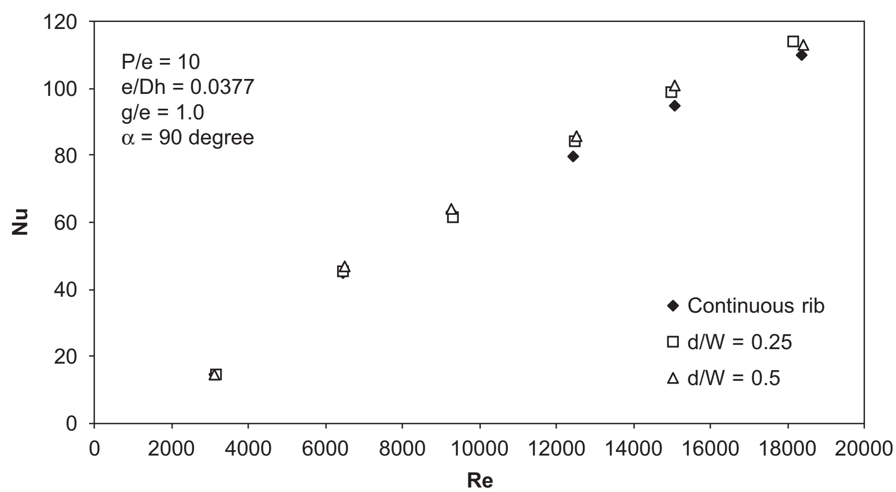


Fig. 8. Effect of gap position on Nusselt number for 90° transverse.

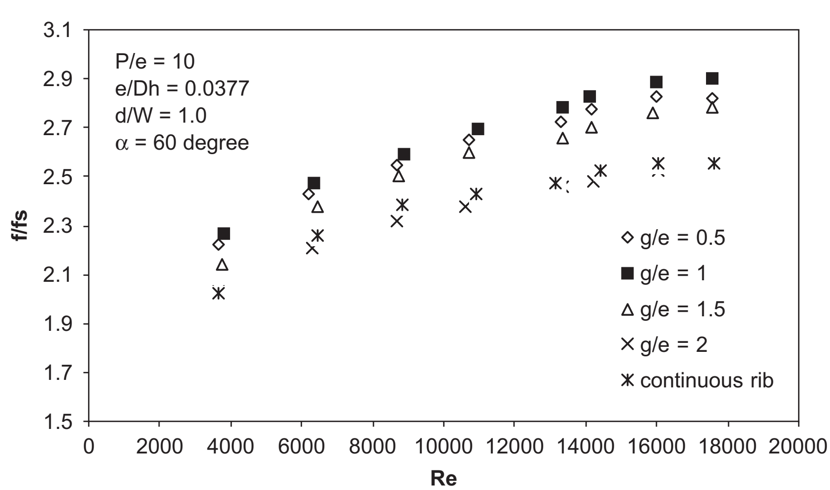


Fig. 9. Effect of gap widths on friction factor ratio with Reynolds number.

penalty due to a corresponding increase in the friction factor. Therefore, it is essential to determine the geometry that will result in maximum enhancement in heat transfer with minimum friction power penalty. In order to achieve this objective of simultaneous consideration of thermal as well as hydraulic performance, i.e., thermo-hydraulic performance, Lewis [25] proposed a thermo-hydraulic parameter known as efficiency parameter 'η', which evaluates the enhancement in heat transfer of a roughened duct compared to that of the smooth duct for the same pumping power requirement and is defined as

$$\eta = (S_t/S_{ts})/(f/f_s)^{1/3}.$$

A value of this parameter higher than unity ensures the fruitfulness of using an enhancement device and can be used to compare the performance of number of arrangements to decide the best among these. The value of this parameter for the roughness geometries investigated in this work has been shown in Fig. 10. It is seen that the value of parameter is generally higher for discrete ribs as compared to that for a continuous rib under similar conditions. The value of this parameter is seen to increase with increase in relative gap width up to about 1.0 and then decreases with further increase in the relative gap width at all values of the Reynolds number, thus attaining a maxima at a relative gap width of about 1.0.

Fig. 11 shows the comparison of the thermo-hydraulic performance of the roughness geometries employed by [4,7] with that of the present study. It is seen from this that the thermo-hydraulic performance of the roughness geometry selected for the present study is the best. It may be pointed out that the friction factor for the present roughness geometry is low as compared to that of the geometries employed by [4,7]. The heat transfer coefficient in the cases of Han et al. [7] and Lau et al. [4] may be higher because of discretization but the combined effect of the heat transfer and friction, i.e., thermo-hydraulic performance of the present geometry is seen to be higher. The data used for above comparison have been generated form the correlations available in the literature. It can therefore be

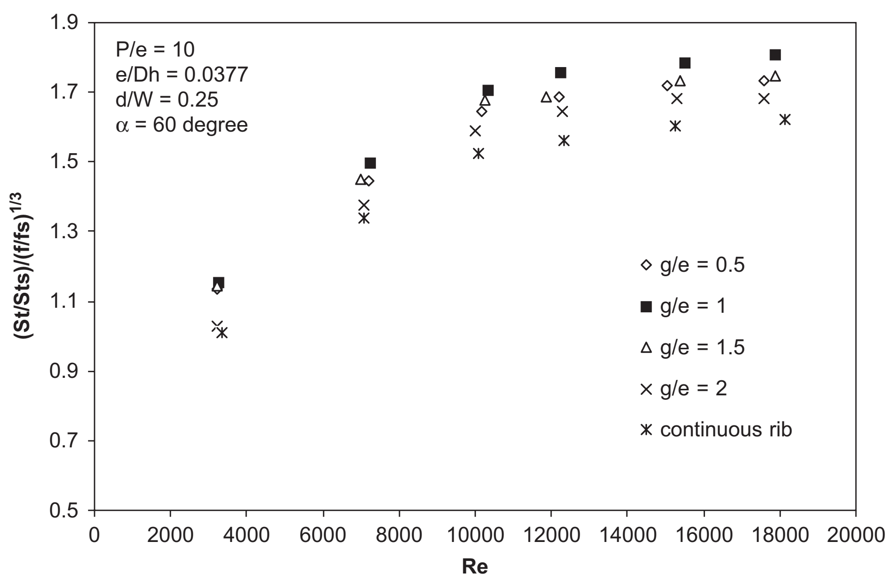


Fig. 10. Effect of parameter as a function of Reynolds number for different values of relative gap width (g/e).

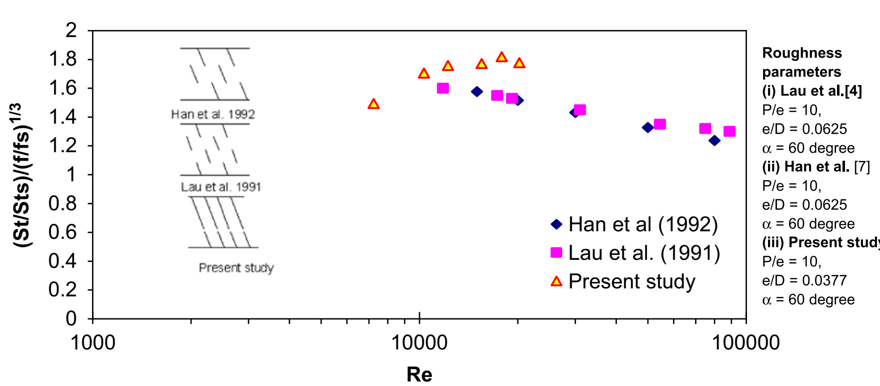


Fig. 11. Comparison of thermo-hydraulic performance with previous investigations.

concluded that a descritization based on the optimization of gap width and gap position can lead to a thermo-hydraulically superior arrangement.

7. Conclusions

Based on this experimental investigation on 60°-inclined square rib-roughened ducts with and without a gap, the following conclusions can be drawn.

(1) A gap in the inclined rib arrangement enhances the heat transfer and friction factor of the roughened ducts. The increase in Nusselt number and friction factor is in the range of 1.48–2.59 times and 2.26–2.9 times of the smooth duct, respectively, for the range of Reynolds numbers from 3000 to 18,000.
(2) The maximum values of Nusselt number and friction factor are observed for a gap in the inclined repeated ribs with a relative gap position of 0.25 and a relative gap width of 1.0.
(3) The thermo-hydraulic performance analysis of roughened ducts shows that the relative gap width of 1.0 and a relative gap position of 0.25 results in a higher value of efficiency parameter.

References

[1] Bhatti MS, Shah RK. Turbulent and transition flow convective heat transfer handbook of single-phase convective heat transfer. New York: Wiley; 1987.
[2] Saini JS. Use of artificial roughness for Enhancing Performance of Solar air heater. Proceedings of XVII national and VI ISHME/ASME heat and mass transfer conference, January 05–07, 2004. IGCAR, Kalpakkam, India.
[3] Han JC, Glicksman LR, Rosenow WM. Investigation of heat transfer and friction for rib-roughened surfaces. Int J Heat Mass Transfer 1978;21:1143–56.
[4] Lau SC, McMillin RD, Han JC. Turbulent heat transfer and friction in a square channel with discrete rib turbulators. Trans ASME, J Turbo Machinery 1991;113:360–6.
[5] Lau SC, McMillin RD, Han JC. Heat transfer characteristics of turbulent flow in a square channel with angled rib. Trans ASME, J Turbomachinery 1991;113:367–74.
[6] Han JC, Park JS. Developing heat transfer in rectangular channels with rib turbulators. Int J Heat Mass Transfer 1988;3(1):183–95.
[7] Han JC, Zhang YM. High performance heat transfers ducts with parallel broken and V- shaped broken ribs. Int J Heat Mass Transfer 1992;35(2):513–23.
[8] Kiml R, Mochizuki S, Murata A. Effects of rib arrangements on heat transfer and flow behavior in a rectangular rib roughened passage. J Heat Transfer 2001;123:675–81.
[9] Han JC, Zhang YM, Lee CP. Augmented heat transfer in square channels with parallel, crossed, and V-shaped angled ribs. Trans ASME, J Heat Transfer 1991;113:590–6.
[10] Taslin ME, Li T, Rercher DM. Experimental heat transfer and friction in channels roughened with angled, V-shaped and discrete ribs on two opposite walls. J Turbomachinery 1996;118:20–8.
[11] Gao X, Sunden B. Heat transfer and pressure drop measurements in rib roughened rectangular ducts. Exp Thermal Fluid Sci 2001;24:25–34.
[12] Hu Z, Shen J. Heat transfer enhancement in a conversing passage with discrete ribs. Int J Heat Mass Transfer 1996;39(8):1719–27.
[13] Cho HH, Wu SJ, Kwon HJ. Local heat/mass transfer measurement in a rectangular duct with discrete ribs. J Turbomachinery 2000;122:579–86.
[14] Cho HH, Kim YY, Rhee DH, Lee SY, Wu SJ. The effect of gap position in discrete ribs on local heat/mass transfer in a square duct. J Enhanced Heat Transfer 2003;10(3):287–300.
[15] Gupta D, Solanki SC, Saini JS. Thermo-hydraulic performance of solar air heaters with roughened absorber plates. Sol Energy 1997;61(1):33–42.
[16] Bhagoria JL, Saini JS, Solanki SC. Heat Transfer coefficient and friction factor correlation for rectangular solar air heater duct having transverse wedge shaped rib roughness on the absorber plate. Renew Energy 2002;25:341–69.
[17] Karwa R, Solanki SC, Saini JS. Heat transfer coefficient and friction factor correlation for the transitional flow regime in rib-roughened rectangular duct. Int J Heat Mass Transfer 1999;42:1597–615.
[18] Karwa R. Experimental studies of augmented heat transfer and friction in asymmetrically heated rectangular ducts with ribs on the heated wall in transverse, inclined, V-continuous and V-discrete pattern. Int J Heat Mass Transfer 2003;30(2):241–50.
[19] Bhagoria JL, Sahu MM. Augmentation of heat transfer coefficient by using 90° broken transverse ribs on absorber plate of solar air heater. Renew Energy 2005;25:2057–73.
[20] Momin AME, Saini JS, Solanki SC. Heat transfer and friction in solar air heater duct with V-shaped rib roughness on absorber plate. Int J Heat Mass Transfer 2002;45:3383–96.
[21] Muluwork KB. Investigations on fluid flow and heat transfer in roughened absorber solar heaters. PhD Dissertation I.I.T. Roorkee (2000).
[22] ASHARAE Standard 93–77. Method of testing to determine the thermal performance of Solar Air Heater, New York 1997; 1–34.
[23] Holman JP. Experimental methods for engineers. New Delhi: Tata McGraw-Hill; 2004.
[24] Han JC, Fu WL, Wright LM. Thermal performance of angled V-shaped and W-shaped rib turbulators in a rotating rectangular cooling channels. Trans ASME, J Heat Transfer 2005;126:604–14.
[25] Lewis MJ. Optimizing the thermo hydraulic performance of rough surface. Int J Heat Mass Transfer 1975;18:1243–8.