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EXPERIMENTAL INVESTIGATION ON HEAT TRANSFER AND FRICTION FACTOR CHARACTERISTICS OF A STATIONARY SQUARE DUCT ROUGHENED BY V AND Λ-SHAPED RIBS

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ABSTRACT

One of the traditional methods used to improve the efficiency of a gas turbine is to increases the inlet temperature; thereby increasing the power output and in turn, the efficiency. The internal cooling passages of blades are roughened by artificial roughness to improve the cooling performance. The present study investigates the convective heat transfer and friction factor (pressure drop) characteristics of a rib-roughened square duct. The test section of the duct is roughened on its top and bottom wall with V and Λ- shaped square ribs. In the study, the Reynolds number (Re) varied from 10,000 to 40,000, the relative roughness height (e/D_h) is 0.060, relative roughness pitch (p/e) is 10 and rib attack angle (α) is taken as 45^0 and 60^0. The comparison of heat transfer and pressure drop for V and Λ- shaped ribs is presented in the form of Nusselt Number and friction factor. The results show that the Nusselt number enhancement decreases when the Reynolds number increases. The friction factor ratio is found to increases as Reynolds number increases. The thermal performance decreases when the Reynolds number increases. It seems that the ribs disturb the main flow resulting in the recirculation and secondary flow near the ribbed wall. The heat transfer coefficient and friction factor for all the cases are higher than that of the smooth duct. The V-shaped ribs have higher Nusselt number and friction factor than the Λ - shaped ribs. It is also concluded that the V -shaped ribs perform better than the Λ- shaped ribs. Ribs with $\alpha = 60^0$ produces higher heat transfer and friction factor than $\alpha = 45^0$. The results obtained in the study will help the turbine blade designer to design the effective cooling methods to cool the blades.

Keywords: Friction factor; Nusselt number; Reynolds number; thermal performance, V-shaped ribs.

1. INTRODUCTION

It has been made to apply different active and passive methods of heat transfer enhancement in rectangular / square ducts which can be used in compact heat exchangers for the automotive industry, air-conditioning and refrigerant applications, internal cooling for gas turbine blades, electrical circuits in electronic chipsets, etc. Achieving higher heat transfer rates through various augmentation techniques can result in substantial energy savings, more compact and less expensive apparatus with higher thermal efficiency. One of the methods to enhance the heat transfer in a flow passage is to roughen the surfaces with ribs which is popularly used in the design of internal cooling passage of turbine blade. The surfaces roughened with the ribs cause extra flow resistances. Hence to optimize the shape and orientation of rib-roughened surface, it is required to investigate the heat transfer and friction characteristics of various roughened surfaces.

The performance of the heat transfer surface with ribs depends significantly on the parameters of the flow structure, such as reattachment length of the separated streamline and turbulence intensities, as well as the area of the surface and Reynolds number. The ribs increase the level of mixing by turbulence and disturb the laminar sub-layer, also increase the surface area for convective heat transfer, thereby enhance the cooling capacity of the passage.

In general, mixing the main flow, reducing the flow boundary layer, raising the turbulent intensity, creating rotating and secondary flows are the main reasons for the increase of the heat transfer. Vortex

generation is a promising technique for heat transfer augmentation. Geometrical parameters such as channel aspect ratio (AR), rib height-to-passage hydraulic diameter (e/D_h), rib attack angle (α), rib pitch-to-height ratio (p/e), rib shape, discretization of ribs and the manner in which the ribs are positioned relative to one another have considerable effects on heat transfer coefficient and friction factor of the passage (Han *et al.*, 2001). Han *et al.* (1978) observed the effect of rib shape, angle of attack and pitch to rib height ratio on heat transfer and friction factor characteristics of a rectangular duct with two opposite side roughened walls. They observed that the maximum value of heat transfer and friction factor occurs for square ribs, at $p/e = 10$ and $\alpha = 45^0$. Kiml *et al.* (2001) reported that the thermal performance of rib arrangements with $\alpha = 60^0$ is better than that with 45^0. Han and Park (1988) studied heat transfer and pressure losses with 90°, 60°, 45°, and 30° angle ribs in square and rectangular channels. The Reynolds number was from 10000 to 60000. They concluded that the higher thermal performance in the square channel was 30° rib angle and the higher thermal performance in the rectangular channel was 45° rib angle. The higher heat transfer with higher pressure drop in the square channel was 60° rib angle. The square channel in their study showed a larger increase in heat transfer performance than the wide aspect ratio channels.

Lau *et al.* (1991) 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. Metzger *et al.* (1990) investigated the effects of rib

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angle and orientation on local heat transfer in a square channel. It was found that 60° angled ribs giving a two-cell pattern provided the best heat transfer performance. Lau *et al.* (1991) examined the turbulent heat transfer and frictional characteristics in a square channel with V-shaped ribs. The configuration investigated included V-shaped ribs angled at 45°, 60°, 90°, 120° and 135° for a *P/e* ratio of 10.The results indicated that the V-shaped ribs at ($\alpha = 46°$ and 60°) produced a 38%-46% and 47%-66% increase in thermal performance when compared to the 90° full rib. Similarly, when the following ribs were reversed, the heat transfer performance was enhanced by 26%-32% and 39%-48% respectively. This enhancement in thermal performance was accompanied by a 55%-79% increase in the pressure drop for various V-shaped configurations in comparison to the 90° rib. The V-shaped ribs pointing downwards (Λ-shaped ribs) was found to result in poor heat transfer performance. The crossed ribs whose performance was also investigated resulted in poor heat removal. It was also found that doubling the *p/e* ratio lowers the heat transfer augmentation and friction. Liou and Hwang (1992) studied the heat transfer and flow field in the ribbed channels. They found that the (*p/e*) of 10 resulted in the best heat transfer; the heat transfer showed a periodic behavior between consecutive ribs; and both heat transfer and friction factor increased with decreasing rib spacing. Liou and Hwang (1993) conducted experiments to measure the local as well as average heat transfer coefficients to compare the performance of square, triangular and semi-circular ribs and found that the square ribs give best heat transfer performance.

It is revealed from the literature that discrete ribs were performed outstanding as compared to continuous angled or V-shaped ribs (Cho *et al.*, 1992). It is also reviewed that V-shaped ribs pointing upwards (V-shaped) and pointing downwards (Λ-shaped) will give different Heat transfer and friction characteristics. Han and Zhang (2000) carried out experiments to study the heat transfer and pressure drop characteristics of a roughened square channel with various angled and V-shaped broken rib arrangement with rib attack angle of 45^0 and 60^0, the Reynolds number was from 15000 to 90000. They were reported that 60^0 V-shaped broken rib arrangements give better performance than 45^0 V-shaped broken ribs. They also concluded that broken ribs create heat transfer enhancement level of 2.5 to 4, while the enhancement created by the continuous ribs is only 2 to 3. Tanda (2004) investigated the heat transfer enhancement for one wall-ribbed rectangular channel of *AR* = 5:1 with continuous, 90° and V-broken ribs and found that the enhancement of the 90° broken ribs is around 1.8 times higher than that of continuous ribs. Han *et al.* (1991) presented experimental results on pressure drop and heat transfer in a square channel with ribs on two walls for nine different rib configurations. Regionally averaged heat transfer and friction factor were reported for rib pitch-to-height ratio (*p/e*) of 10 and the rib height to hydraulic diameter (*e/D_h*) of 0.0625. They concluded that the angled ribs and 'V' ribs provide higher heat transfer augmentation compared to continuous ribs. It was observed that the heat transfer augmentations and the friction factor were highest for the 60^0 orientation compared to 45^0 and 90^0 orientation amongst the angled ribs. The 60^0 'V' and 60^0 'Λ' offered higher heat transfer augmentation compared to the corresponding 45^0 orientation. The crossed rib configuration offered least heat transfer augmentation. The friction factor was found to be highest for 60^0 angled ribs compared to 45^0 and 90^0 configuration. Continuous rib configuration offered the least resistance and the 60^0 'Λ' case offered highest flow resistance among all configurations.

A very narrow channel (*AR*=8:1) with V-shaped, λ-shaped, and angled ribs was studied by Gao and Suden (2001). Using a liquid crystal technique, they too confirmed that V-shaped ribs result in the highest heat transfer enhancement and the highest frictional losses. They concluded that the V-shaped ribs yield the best overall thermal performance. Kim *et al.* (2001) also investigated heat transfer enhancement mechanisms in rectangular channels with V- and λ -shaped ribs using a flow visualization technique to examine the

secondary flow behaviours created by the V-shaped ribs. Taslim *et al.* (1996) experimentally investigated the heat transfer and friction in channel roughened with angled, V-shaped and discrete ribs on two opposite walls for Reynolds number ranging from 5,000 to 30,000. The results showed that the 90° transverse ribs produced the lowest heat transfer performance. The 45° angled V-shaped ribs produced the highest heat transfer performance in comparison to other rib configurations. For V-shaped ribs facing downstream of flow, the one with lowest blockage ratio had better heat removal rate. The discrete ribs also produced better performance in comparison to the transverse ribs.

Muluwork *et al.* (1998) have investigated the effect of a staggered discrete V- apex up and down on the thermal performance. The Stanton number for V-down discrete ribs was higher than the corresponding V-upwards and transverse discrete roughened surfaces. The Stanton number ratio enhancement was found to be 1.32 to 2.47 in the range of parameters covered in the investigation. Further for the Stanton number, it was seen that the ribbed surface friction factor for V-downwards discrete ribs was highest among the three configurations investigated. Karwa (2003) has investigated and revealed the effect of transverse, inclined, V-continuous and V-discrete patterns on heat transfer and the friction factor in a rectangular duct. The ribs in the V-pattern were tested for both pointing upstream (V-upwards) and downstream (V-downwards) to the flow. The angle of inclination of the ribs in inclined and V-pattern was 60°. The enhancement in the Stanton number over the smooth duct was up to 137%, 147%, 134% and 142% for the V-up continuous, V-down continuous, V-up discrete and V-down discrete rib arrangement respectively. The friction factor ratio for these arrangements was up to 3.92, 3.65, 2.47 and 2.58, respectively. Based on the equal pumping power, V-down discrete roughness gave the best heat transfer performance.

From above, it can be stated that inclined and V-shaped ribs when arranged in a systematic manner produces excessive enhancement in convective heat transfer, however the friction factor also increases. Some of the researchers investigated that V-shaped ribs give higher heat transfer than λ-shaped ribs, whereas, some observed the reverse results.

In the present work, experimental investigations have been carried out on an opposite side roughened square duct having artificial roughness in the form of V and Λ-shaped rib, to determine the enhancement in heat transfer and friction factor to find out which one give the highest heat transfer performance. The optimum value of rib attack angle (*α*) for both types (V and Λ- shaped) of ribs have been obtained and discussed.

2. EXPERIMENTAL PROGRAM AND ROUGHNESS GEOMETRY

A schematic diagram of the test set up is shown in Fig.1. which was used by Gupta *et al.* (2013) and Chaube *et al.* (2014). The wooden square duct has an internal size of 3750 mm x 75 mm x 75 mm, which consists of an entrance section, a test section and an exit section of length 1500 mm ($20D_h$), 1500 mm ($20D_h$) and 750 mm ($10D_h$) respectively. The exit end of the duct is connected to 81 mm internal diameter G. I. (galvanized iron) pipe provided with a calibrated orifice plate through a square to circular transition piece. The outside of entire set-up from inlet to the orifice plate, were covered with 25 mm thick thermocole (foamed polystyrene) sheet, so that the heat losses from the test section can be minimized.

The aluminium plates are used as test plates which are made by gluing square aluminium ribs (4.5 mm x 4.5 mm) in a required distribution to serve as top and bottom ribbed walls of the test section. The plates are heated by means of separate heaters assembly, thus subjected to uniform heat flux of 0-1500 W/m². To measure mass flow rate of air through the duct, a calibrated orifice-meter is used which is connected with an inclined U-tube manometer with a control valve in the pipeline. Twenty thermocouples are affixed on backside of the test plate as shown in Fig. 2 to record the plate temperature at different

locations. The bulk air temperature at the entrance section is measured by providing thermocouple at the entrance section, whereas bulk temperature of the air at the exit is recorded by three thermocouples mounted spanwise at the exit section after mixing (depthwise). To measure the pressure drop across the test section length a digital micro-manometer (Fluke-922) is connected between the required points. The airflow rate was varied to give the flow having Reynolds number in the range of 10000 to 40,000. Relevant data were noted under the steady-state condition for constant surface heat flux, which was assumed to have reached when the plate and air temperatures shows negligible

variation for about 10- minute duration. The steady state for each test run was obtained in about 1.5 to 2 hours. In order to minimize the percentage error in measurement of temperatures, minimum heat flux value is so selected as to maintain the temperature of roughened plate around 20^0C - 30^0C above to that of mean bulk air temperature.

The V and Λ-shaped ribs used in the investigation are shown in Fig. 3. The relative roughness height (e/D_h) and relative roughness pitch (p/e) are kept constant as 0.060 and 10 respectively. Reynolds number was varied in the range of 5000 to 40000. Experimental data for both types of ribs were collected for attack angle (α) of 45^0 and 60^0.
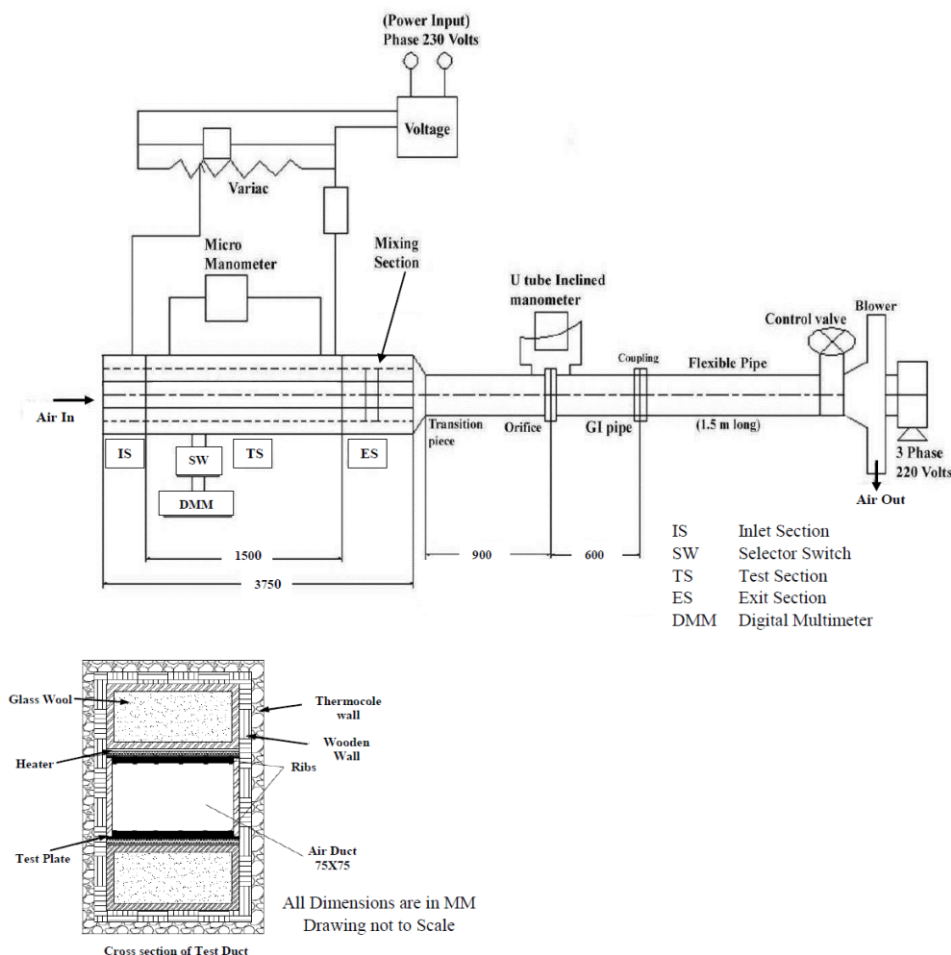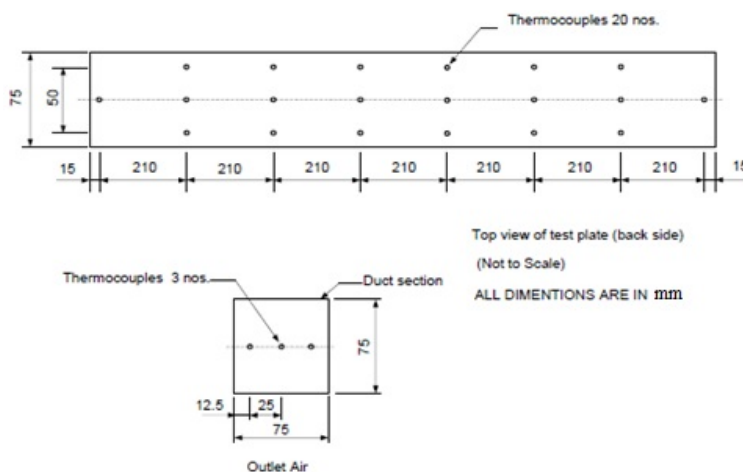

Fig. 1. Schematic diagram of experimental set-up

Fig. 2. Detailed locations of thermocouples

(a) V-shaped (V- upstream) rib

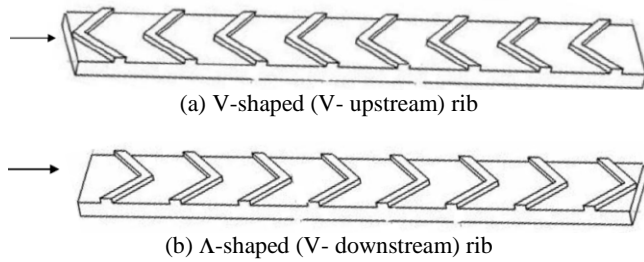(b) Λ-shaped (V- downstream) rib

Fig. 3. Schematic of roughness geometry

3. DATA REDUCTION

The objective of this experiment is to investigate the Nusselt number and friction factor in the channel. The Reynolds number based on the channel hydraulic diameter, D_h, is given by

$$Re = \frac{\rho_a V D_h}{\mu_a} \tag{1}$$

where,

$$V = \frac{m}{\rho_a \cdot W \cdot H} \tag{2}$$

The mass flow rate, m, of air through the duct has been calculated from pressure drop measurement across the orifice plate.

$$m = C_d \cdot A_o \left[\frac{2 \cdot \rho_a \cdot (\Delta P)_o}{1 - \beta^4} \right]^{0.5} \tag{3}$$

The pressure drop $(\Delta P)_o$ across the orifice plate is given by

$$(\Delta P)_o = 9.81 \text{ x } (\Delta h)_o \text{ x } \rho_m \text{ x } sin\theta \tag{4}$$

The average heat transfer coefficient, h, is evaluated from the measured temperatures and heat inputs. With heat added uniformly to fluid (Q_u) and the temperature difference of plate and fluid ($T_p - T_f$), the average heat transfer coefficient will be evaluated from the experimental data via the following equations:

$$Q_u = m \, C_p \, (T_o - T_i) \tag{5}$$

$$h = \frac{Q_u}{A_p (T_p - T_f)} \tag{6}$$

The average value of plate temperature (T_p) is calculated as a weighted mean of the plate temperature measured at different locations and mean bulk air temperature (T_f) is calculated from the following equations.

$$T_f = \frac{T_i + T_o}{2} \tag{7}$$

$$T_p = \sum_{n=1}^{20} \frac{T_n}{20} \tag{8}$$

Then, average Nusselt number, Nu, is written as

$$Nu = \frac{h D_h}{k_a} \tag{9}$$

The friction factor is determined from the measured values of pressure drop, $(\Delta P)_d$ across the test section length, between the two points located 1.2 m apart.

$$f = \frac{2 \, (\Delta P)_d \, D_h}{4 \, \rho_a \, L_f V^2} \tag{10}$$

where, $(\Delta P)_d$ is the pressure drop across the duct and is given by

$$(\Delta P)_d = 9.81 \text{ x } (\Delta h)_d \text{ x } \rho_m \tag{11}$$

A parameter known as thermo-hydraulic performance parameter 'η' is used to evaluate the effectiveness of artificially roughened surfaces accounting for the enhancement of Nusselt number and friction factor [Lewis, M. J., 1975]. It can be calculated by

$$\eta = [(Nu_r/Nu_o)/(f_r/f_o)^{1/3}] \tag{12}$$

Based on the calculations of the errors in the experimental measurements by various instruments used, the uncertainties in the calculated values of Reynolds number, Nusselt number, friction factor and thermo-hydraulic performance parameter are estimated as ± 1.65%, 1.94 %, ± 3.22 % and 2.25 % respectively (Holman, 2004).

4. VALIDATION OF EXPERIMENTAL DATA

Before taking the experimental data of artificially roughened ducts, the experimental set up was validated by keeping all surface as smooth surfaces and the results obtained in the form of Nusselt number and friction factor were compared with the standard correlations for Nusselt number and friction factor obtained with the standard Dittus-Boelter equation ($Nu_o = 0.023 \, Re^{0.8} \, Pr^{0.4}$) for the Nusselt number and Blasius equation ($f_o = 0.079 \, Re^{-0.25}$) for friction factor (Bhatti and Shah 1987).

The comparison of the experimental and predicted values of Nusselt number and friction factor as a function of Reynolds number is shown in Fig.4 and in Fig.5 respectively. It is observed that the average deviation between the predicted and experimental values has been ±5.9 % and ±7.3 % for Nusselt number and friction factor respectively. Thus the results obtained and the predicted values are in good agreement, which ensures the accuracy of the experimental data with the present experimental set-up.
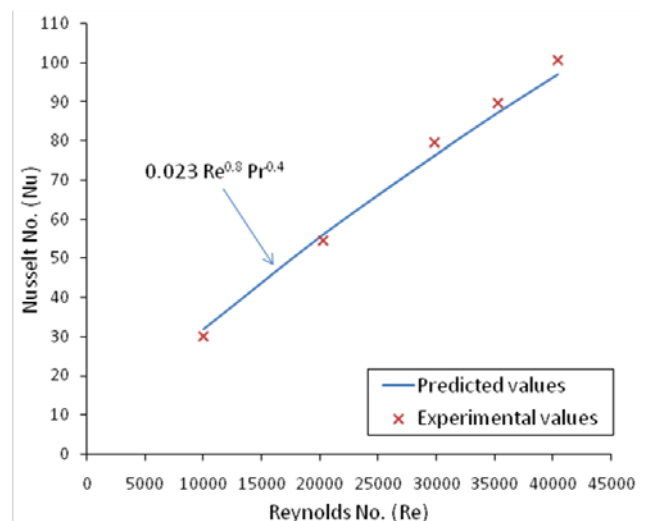
Fig. 4. Comparison of experimental and predicted values of Nusselt number for smooth duct
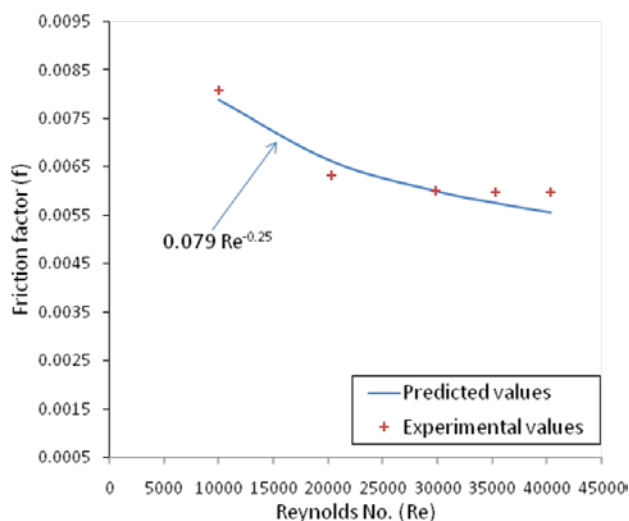
Fig. 5. Comparison of experimental and predicted values of friction factor for smooth duct

5. RESULTS AND DISCUSSION

The values of Nusselt number and friction factor for Λ-shaped and V-shaped rib geometry were computed on the basis of experimental data collected for various flow and roughness parameters. The effects of various parameters on Nusselt number and friction factor are presented in this section.

Figure 6 and Fig. 7 shows the values of Nusselt number, and Nusselt number ratio for V-shaped ribs respectively. It is clear from the figures that the Nusselt number increases as Reynolds number increases and the heat transfer enhancement is about 2.75 to 3.8 times than that of smooth duct. The formation of secondary flow by the ribs is mainly responsible for the increase in heat transfer coefficient.

The V-shaped ribs are responsible for the spanwise heat transfer coefficient. Gao and Sunden [23] suggested the four cell pattern of secondary flow. The swirling motion created by the secondary flow helped to generate a high overall average heat transfer coefficient. The heat transfer enhancement is higher for rib attack angle $\alpha= 60^0$ than $\alpha= 45^0$. The flow travels along the ribs from side wall and meets at the apex of the rib whereas for V ribs the flow deflects from the apex and then travels along the ribs to generate secondary flows. (as shown in Figure 8).
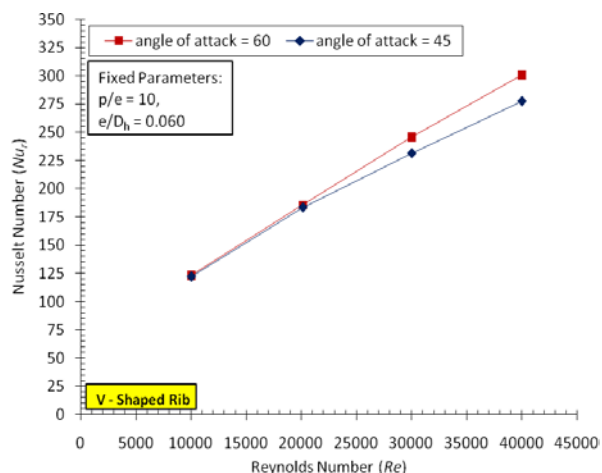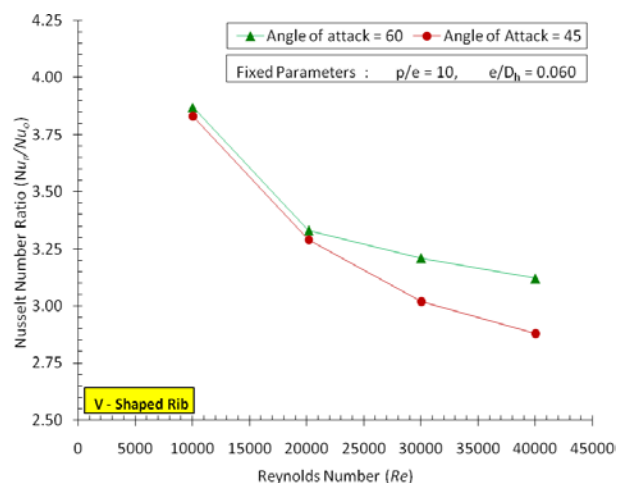

Fig. 7. Variation of Nusselt number ratio with Reynolds number at $\alpha=60^0$ and 45^0 for V-shaped ribs.
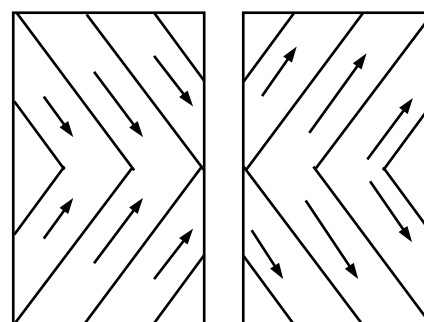

Fig. 8. Secondary flow generated in Λ-shaped and V-shaped ribs

Figure 9 and 10 illustrates the variation of friction factor with Reynolds number for different rib attack angles of V-shaped ribs. It is observed from Figures 9 and 10 that the friction factor decreases as Reynolds number increases and it remains almost constant at higher Reynolds numbers. The increase in friction factor ranges from 7 to 8.8. The friction factor enhancement is more for 60^0 ribs as compared to 45^0 ribs.
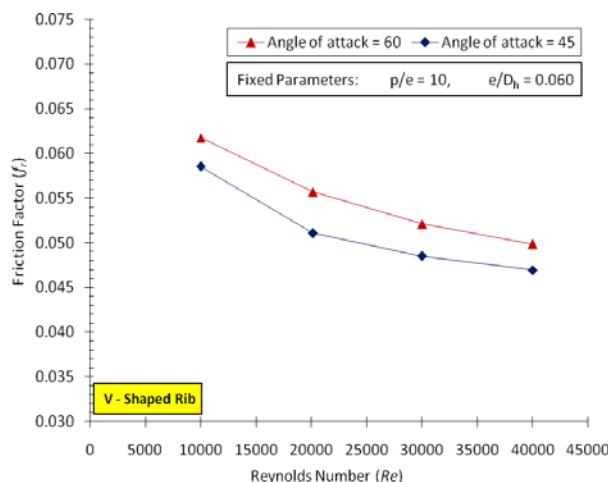

Fig. 6. Variation of Nusselt number with Reynolds number at $\alpha=60^0$ and 45^0 for V-shaped ribs.

Fig. 9. Variation of friction factor with Reynolds number at $\alpha=60^0$ and 45^0 for V-shaped ribs.
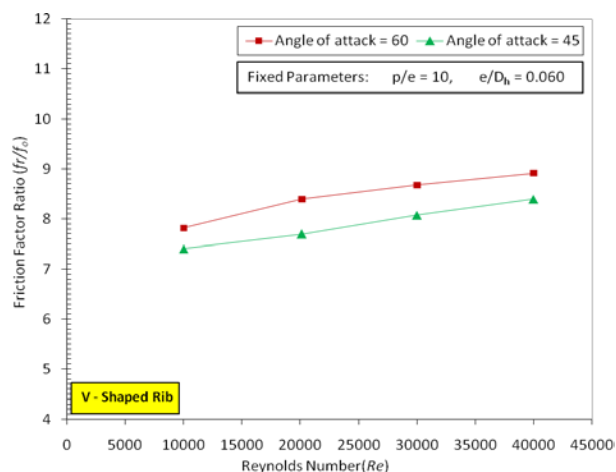
Fig. 10. Variation of friction factor ratio with Reynolds number at $\alpha=60^0$ and 45^0 for V-shaped ribs.

Figures 11 to 14 show the Nusselt number and friction factor results for V – ribs pointing downstream (∧- shaped ribs). In contrast to the V the ∧- shaped ribs produces less heat transfer and more friction factor. It is also observed that at low Reynolds numbers, the ∧-ribs produces lower heat transfer coefficient. It seems that a much weaker secondary flow is generated than the case of V-shaped ribs. Gao and sunden (2001) shows that in addition to the main flow V ribs also created two double cell counter rotating vortices but the direction is reversed as in the case of ∧-shaped ribs. Liou *et al.* (2000) investigated the secondary flow patterns for 45^0 V-shaped ribs pointing upstream and downstream in a square duct. The results conforms the results of present investigation and but opposite results are observed as compared to the results of Gao and Sunden (2001). The reason is that the aspect ratio in present case is 1 which produces the different kind of secondary flow as compared to the rectangular ducts. The wall confinement reduces the regions where secondary flows are directed towards the bottom wall and therefore curtails the heat transfer enhancement for the case of V- ribs pointing downstream. The friction factor enhancement is also somewhat reduces as compared to the case of V ribs pointing upstream.
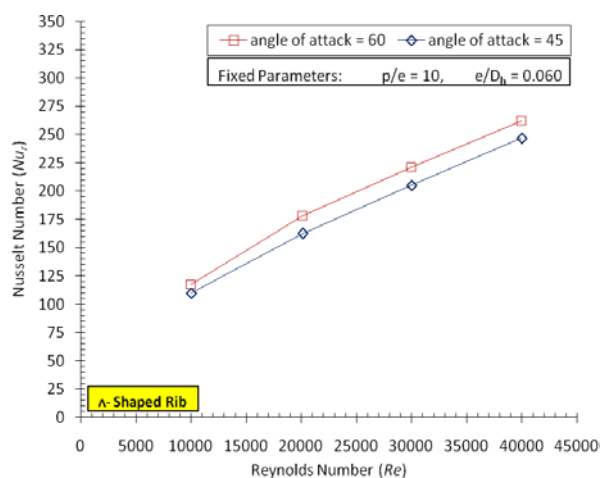

Fig. 11. Variation of Nusselt number with Reynolds number at $\alpha=60^0$ and 45^0 for ∧-shaped ribs.

The ∧-shaped ribs produce less heat transfer whereas pressure penalty is greater than the V-shaped ribs. Thus the best rib geometry can be decided by considering the heat transfer and friction characteristics simultaneously. A parameter that facilitates the simultaneous consideration of thermal and hydraulic performance

known as thermo-hydraulic performance (Lewis, 1975) is evaluated for both the cases and plotted as in Figure 15. It is observed from the figure that the thermo hydraulic performance for V-shaped ribs is higher than the ∧-shaped ribs . For all the cases studied the performance is decreases as Reynolds number increases. V-shaped ribs with $\alpha=45^0$give comparable results as compared to the results of V-shaped ribs with $\alpha=60^0$.
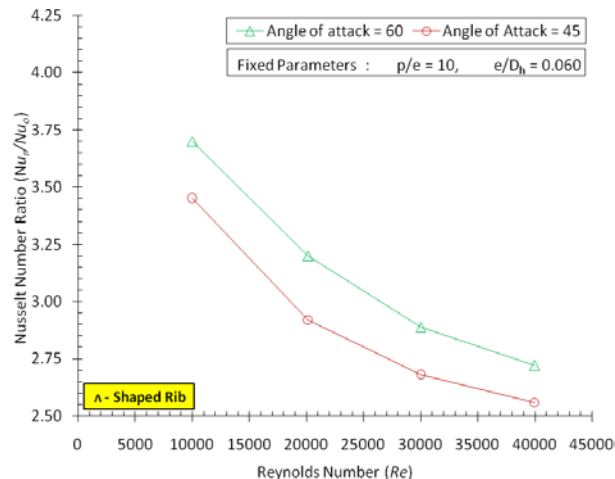

Fig. 12. Variation of Nusselt number ratio with Reynolds number at $\alpha=60^0$ and 45^0 for ∧-shaped ribs.
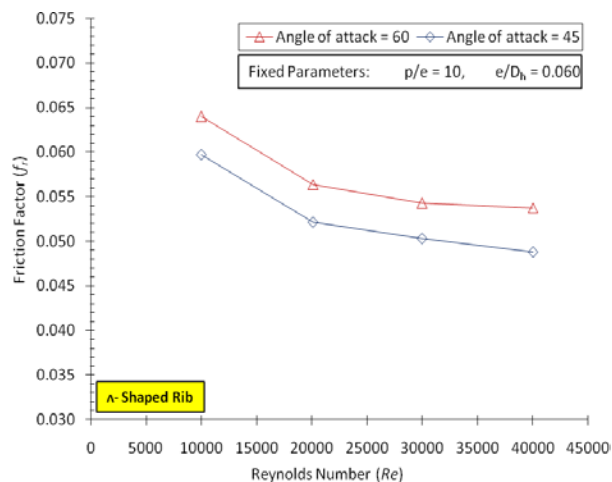

Fig. 13. Variation of friction factor with Reynolds number at $\alpha=60^0$ and 45^0 for ∧-shaped ribs.
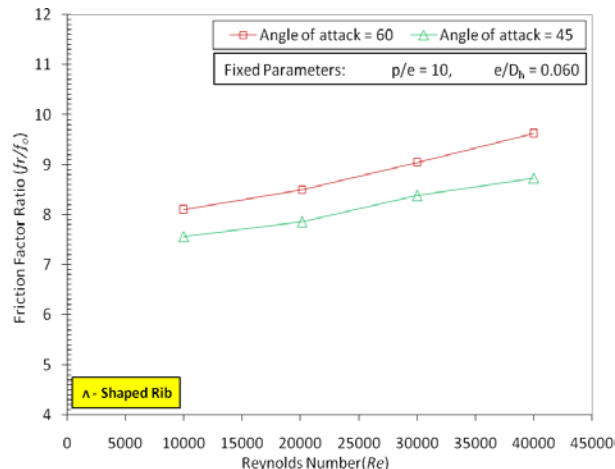

Fig. 14. Variation of friction factor ratio with Reynolds number at $\alpha=60^0$ and 45^0 for ∧-shaped ribs.
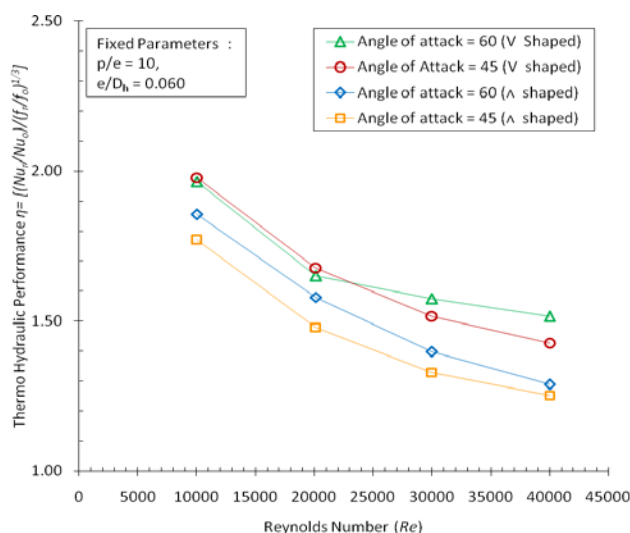
Fig. 15. Variation of Thermo hydraulic performance with Reynolds number at $\alpha=60^0$ and 45^0 for V and Λ-shaped ribs.

6. CONCLUSIONS

An experimental investigation has been carried out with the aim to compare V and Λ-shaped rib geometry for a square duct to found out rib geometry that gives maximum heat transfer with high value of thermo-hydraulic performance.

The following conclusions can be drawn from this work:

(1) In general, Nusselt number increases and friction factor decreases with an increase of Reynolds number. Nusselt number and friction factor are significantly higher as compared to those obtained for smooth duct. This is due to the distinct change in the fluid flow characteristics as a result of V and Λ–shaped roughness that causes flow separations, reattachments and the generation of secondary flows.

(2) The Nusselt number enhancement decreases when the Reynolds number increases. The friction factor ratio is found to increases as Reynolds number increases and becomes constant at high Reynolds numbers.

(3) The maximum value of Nusselt number has been found for V-shaped ribs with $\alpha=60^0$ corresponds to relative roughness height (e/D_h) of 0.06 and relative roughness pitch (p/e) of 10.

(4) The rib attack angle plays an important role for desirable fluid flow characteristics. The maximum value of Nusselt number for both the cases is obtained for 60° due to stronger secondary flow.

(5) The maximum enhancement in Nusselt number and friction factor values compared to smooth duct are of the order of 3.8 and 8.8 respectively.

(6) The thermal performance decreases when the Reynolds number increases. The V shaped ribs with $\alpha = 60^0$ give the highest thermal hydraulic performance for most of the Reynolds numbers. It produces somewhat less thermal hydraulic performance as compared to $\alpha = 45^0$ for a narrow range of Reynolds number due to less value of friction factor at these Reynolds numbers.

(7) The V shaped ribs with $\alpha = 45^0$ shows the comparable results as compared to the results of $\alpha = 60^0$.

NOMENCLATURE

A	Area of duct cross section, m^2
A_p	Area of roughened plate, m^2
A_{pipe}	cross section area of pipe, m^2
AR	Channel aspect ratio (W/H)
C_d	Coefficient of discharge for orifice meter
C_p	Specific heat of air at constant pressure, J/kg K
D_h	Channel hydraulic diameter of duct, m
D_p	Inside diameter of pipe, m
d/W	Relative gap position
e	Rib height, m
e/D_h	Relative roughness height
f	Fanning friction factor
f_o	Friction factor for smooth circular duct
f_r	Friction factor for roughened duct
f_r/f_o	Friction factor ratio
g/e	Relative gap width
H	Depth of duct, m
h	Convective heat transfer coefficient, W/m^2 K
k_a	Thermal conductivity of air, W/m K
L	Test section length, m
L_f	Duct length for calculation of friction factor, m
L/D_h	Test length to hydraulic diameter ratio
m	mass flow rate, kg/s
Nu	Nusselt number
Nu_o	Nusselt number for smooth circular duct
Nu_r	Nusselt number for roughened duct
Nu_r/Nu_o	Nusselt number ratio
p	Rib pitch, m
Pr	Prandtl number
$(\Delta P)_d$	Pressure drop in the test channel, N/m^2
$(\Delta P)_o$	Pressure drop across the orifice plate, N/m^2
p/e	Relative roughness pitch
Q_u	Useful Heat gain, W
Re	Reynolds number
T_f	Bulk mean air temperature, ^{0}C or K
T_i	Air inlet temperature, ^{0}C or K
T_o	Air outlet temperature, ^{0}C or K
T_p	Mean plate temperature, ^{0}C or K
V	Velocity of air, m/s
W	Width of duct, m

Greek Symbols

α	Rib angle of attack, degree
β	Ratio of orifice diameter to pipe diameter
θ	Angle of inclination of U-tube manometer, degree
μ_a	Dynamic viscosity of air, N s m^{-2}
ρ_a	Density of air, kg m^{-3}
ρ_m	Density of manometer fluid, kg m^{-3}

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