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EFFECT OF RIB ATTACK ANGLE ON HEAT TRANSFER AND FRICTION FACTOR IN A SQUARE CHANNEL ROUGHENED BY V SHAPED RIBS WITH A GAP

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ABSTRACT

An experimental study was carried out to investigate the effect of angle of attack on forced convection heat transfer and friction factor of an artificially roughened duct. The test section of the square duct (*AR*=1) was roughened on its top and bottom wall with V shaped square ribs having a gap on its length. The ratio of the width to height of the duct (*W/H*) was 1, the relative roughness height (e/D_h) was 0.060, relative roughness pitch (*p/e*) was 10, relative gap width (*g/e*) was 1and relative gap position (*d/W*) was taken as 1/3. The rib attack angle (α) was varied from 30^0 to 75^0. The air flow rate corresponded to Reynolds number (*Re*) was varied from 5,000 to 40,000. The comparison of heat transfer and pressure drop for various configurations is presented in the form of Nusselt Number and friction factor. The results show that there is significant effect on the Nusselt number and friction factor when the angle of attack has changed. As a result, the maximum enhancement of Nusselt number (*Nu*) and friction factor (*f*) were 4.7 and 8.1 times that of smooth duct, respectively. Maximum heat transfer enhancement was associated with ribs of angle of attack of 60^0 which also produces the maximum friction factor (pressure drop) penalty, in the range of parameters investigated. The thermo-hydraulic performance for all attack angles has also been considered and compared. The result shows that broken V shaped ribs with angle of attack of flow as 60^0 gave the best thermo-hydraulic performance of 2.6 times that of smooth duct.

KEYWORDS: Friction factor, Nusselt number, Angle of attack, thermo-hydraulic performance, V-shaped ribs.

I. INTRODUCTION

Increase in power output and thermal efficiency of gas turbines is possible by increasing the inlet temperature of hot gases through the internal passages. These temperatures exceed the melting point of the turbine components. Thus there is necessity to cool the turbine components by transferring the heat from the surfaces. There are several techniques to cool a modern gas turbine blade and one of them is by artificially roughening the internal flow passage. The surfaces roughened with the ribs cause extra flow resistances. Hence, turbulence has to be created in the area very close to the heat-transferring surface to diminish viscous sub- layer. 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 flow are the main reasons for the increase of the heat transfer [1].

Han et al. [2] 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. [3] reported that the thermal performance of rib arrangements with $\alpha = 60^0$ is better than that with 45^0. Han et al. [4] 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. [5] 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. [6]

investigated the effects of rib 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. [7] 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 (α = 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 reversed V-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 et al. [8] 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 et al. [9] 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 [10]. It is also reviewed that V-shaped ribs pointing upwards (V-shaped) and pointing downwards (ʌ-shaped) will give different Heat transfer and friction characteristics. Han and Zhang [11] 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 [12] 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. [13] 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*) 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 'ʌ' 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 'ʌ' case offered highest flow resistance among all configurations.

A very narrow channel (AR=8:1) with V-shaped, ʌ -shaped, and angled ribs was studied by Gao and Suden [14]. 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. Taslim et al. [15] experimentally investigated the heat transfer and friction in channel roughened with angled, V-shaped and discrete ribs on two opposite walls for Reynolds number raging 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. [16] 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-up 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-down discrete ribs was highest among the three configurations investigated. Karwa [17] 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 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

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respectively. The friction factor ratio for these arrangements was up to 3.92, 3.65, 2.47 and 2.58, respectively. Anil et. al. [18] investigated absorber plate of solar air heater roughened with Multi V shaped ribs with gap. Pitch-to-height ratio was 10, rib height to hydraulic diameter (*e/D*) of 0.0433 while the angle of attack was varied from 30^0 to 75^0. They found that gap equal to rib height results in a substantial improvement in the thermo-hydraulic performance.

In view of the above, it can be stated that V shaping of a long angled rib helps in the formation of two secondary flow cells as compared to one in the case of an inclined rib resulting in higher heat transfer rate. Also by producing gap in V rib a considerably large enhancement is bringing out. There is scant data on width and location of the gap which energies the primary flow, particularly in the case of gas turbine blade cooling.

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- shaped rib with a gap along its length. The flow Reynolds number was varied between 5000 and 40000. The variation of the Nusselt number and friction factor as a function of the angle of attack for a fixed value of the other parameters was evaluated and the thermo-hydraulic performance of the channel was also examined. The optimum value of rib attack angle (α) for the range of parameters investigated have been obtained and discussed.

II. EXPERIMENTAL SET-UP AND ROUGHNESS GEOMETRY

A schematic diagram and pictorial view of the test set up is shown in Fig.1. and in Fig.2. respectively [19, 20]. 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.

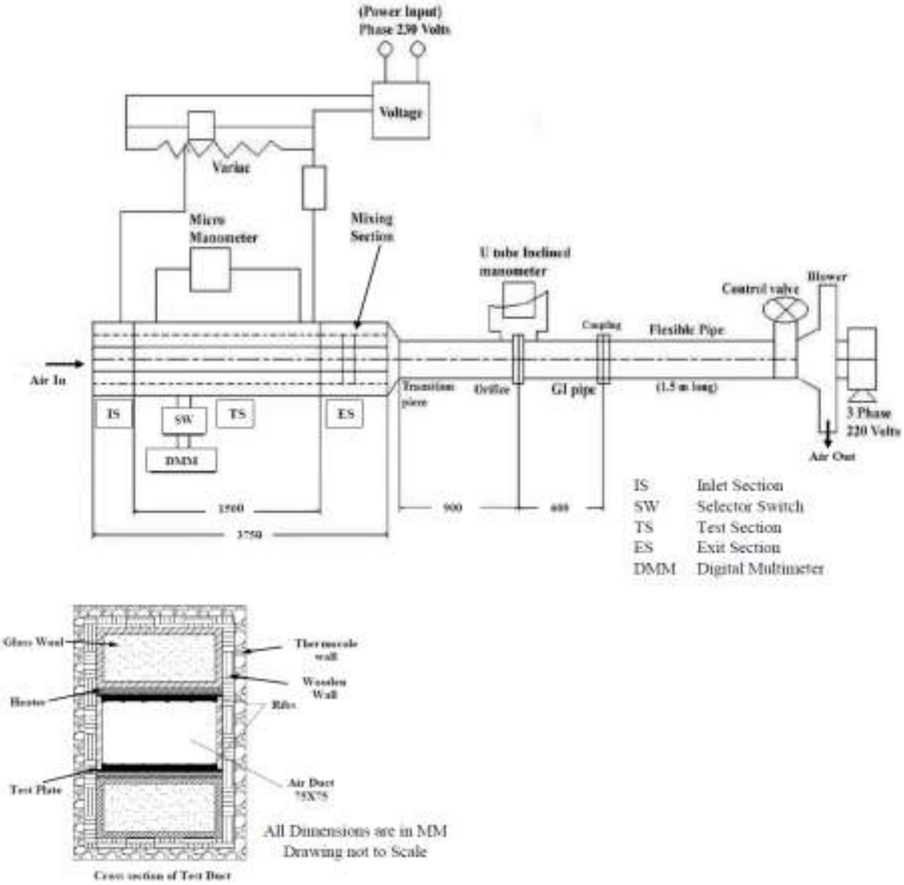

FIG. 1. Schematic diagram of experimental set-up

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The test plates used are the aluminium 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 given heat by means of separate heaters assembly, thus subjected to uniform heat flux of 0-1500 W/m^2. 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 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 5000 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-shaped rib with gap used in the investigation is shown in Fig. 3. In the current investigation, the relative roughness height (e/D_h) and relative roughness pitch (p/e) is kept constant as 0.060 and 10 respectively. Reynolds number was varied in the range of 5000 to 40000 throughout the experimentation. Experimental data were collected for attack angle (α) of 30^0,45^0, 60^0 and 75^0.

FIG. 2. Pictorial view of experimental set-up

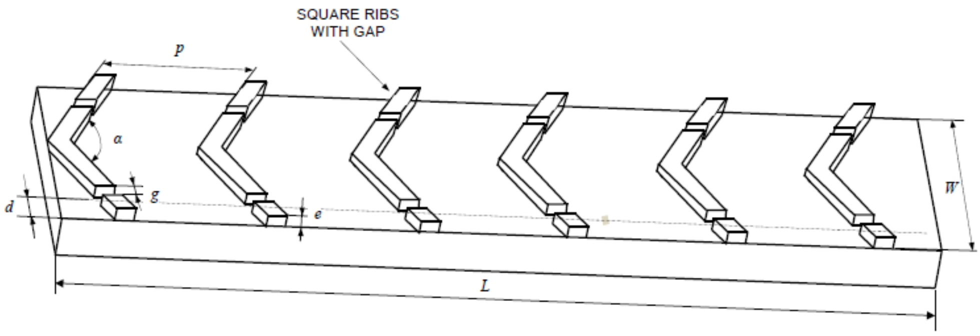

FIG. 3. Schematic of roughness geometry

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III. 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

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

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

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

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:

$$Q_u = m\, C_p\, (T_o - T_i) \tag{3}$$

$$h = \frac{Q_u}{A_p (T_p - T_f)} \tag{4}$$

Then, average Nusselt number, Nu, is written as

$$Nu = \frac{h D_h}{k_a} \tag{5}$$

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.

$$f = \frac{2\,(\Delta P)_d\, D_h}{4\,\rho_a\, L_f V^2} \tag{6}$$

The thermo-hydraulic performance can be calculated by

$$\eta = [(Nu_r/Nu_o)/(f_r/f_o)^{1/3}] \tag{7}$$

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 [21].

IV. 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 [22]. 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.

V. RESULTS AND DISCUSSION

The values of Nusselt number and friction factor for V-shaped rib with gap 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. The values of Nusselt number, and Nusselt number ratio for fixed values of relative gap position (d/W) of 1/3, relative gap width (g/e) of 1, relative roughness height (e/D_h) of 0.060, relative roughness pitch (p/e) of 10 and different values of angle of attack (α) are presented in Fig. 4 and Fig. 5, respectively. It is clear from the figures that the Nusselt number and Nusselt number ratio increase with an increase in the angle of attack up to about 60^0, beyond which they decrease with

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increase in the angle of attack. Fig. 6. clearly depicts the effect of angle of attack and shows that Nusselt number was highest for an angle of attack of 60^0. The heat transfer enhancement was about 3.1 to 4.7 times than that of smooth duct. The formation of secondary flow by the ribs is mainly responsible for the increase in heat transfer coefficient along with the strengthening of primary flow due to gap. The strength of secondary flow along the rib changes with a change in angle of attack.

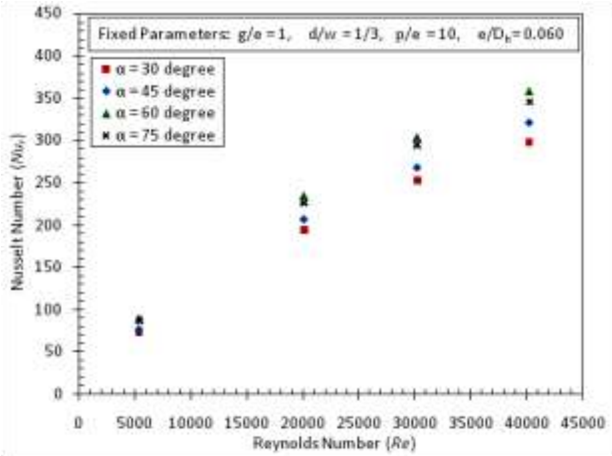

FIG. 4. Variation of Nusselt number with Reynolds number for different angle of attack.

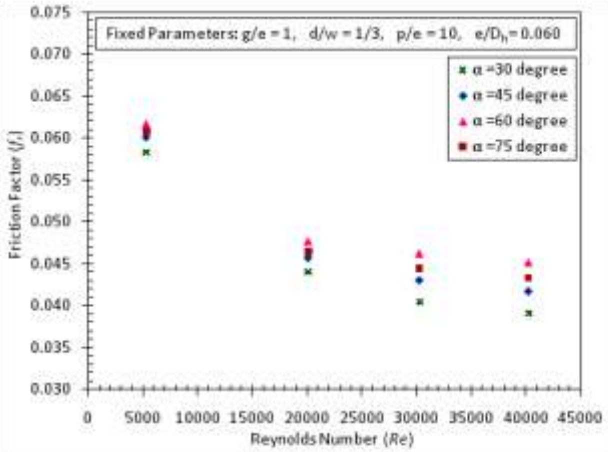

FIG. 7. Variation of friction factor with Reynolds number for different angle of attack.

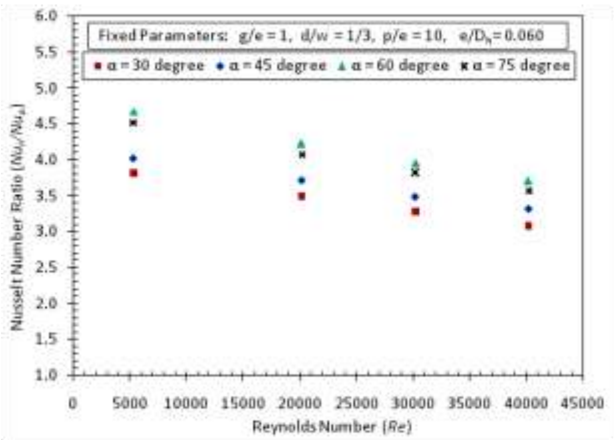

FIG. 5. Variation of Nusselt number ratio with Reynolds number for different angle of attack.

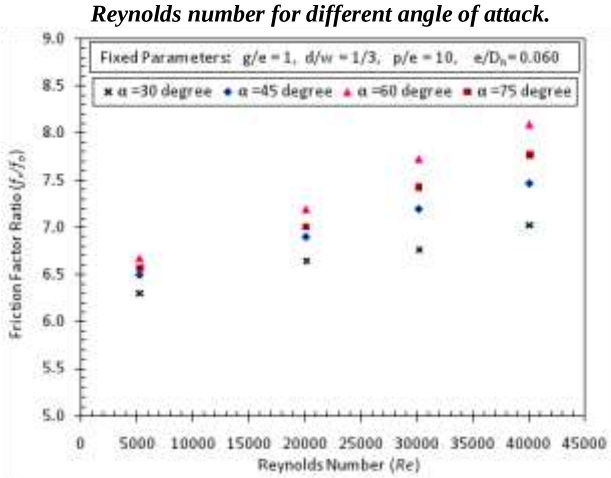

FIG. 8. Variation of friction factor ratio with Reynolds number for different angle of attack.

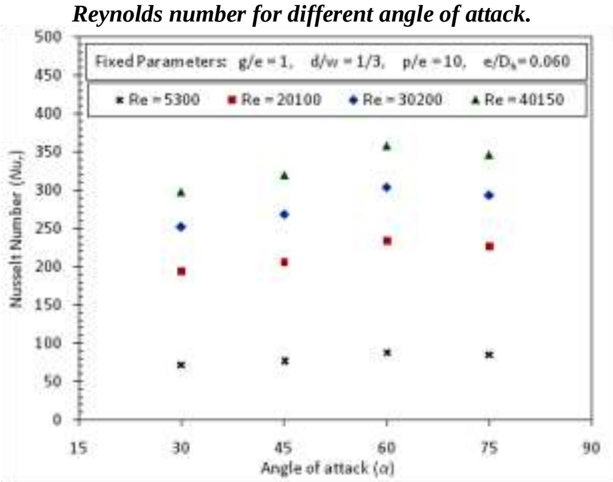

FIG. 6. Variation of Nusselt number with angle of attack for selected Reynolds number.

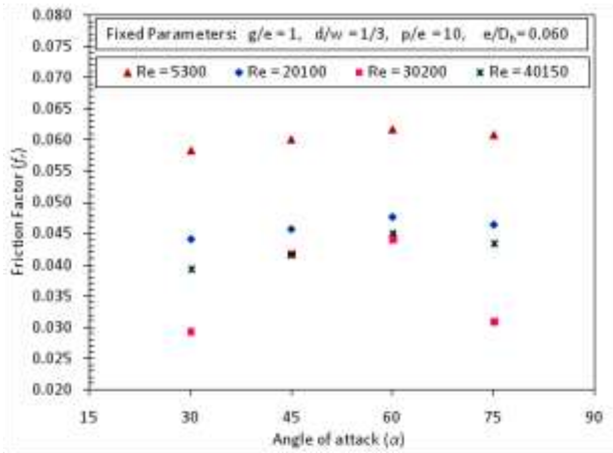

FIG. 9. Variation of friction factor with Angle of attack for selected Reynolds numbers

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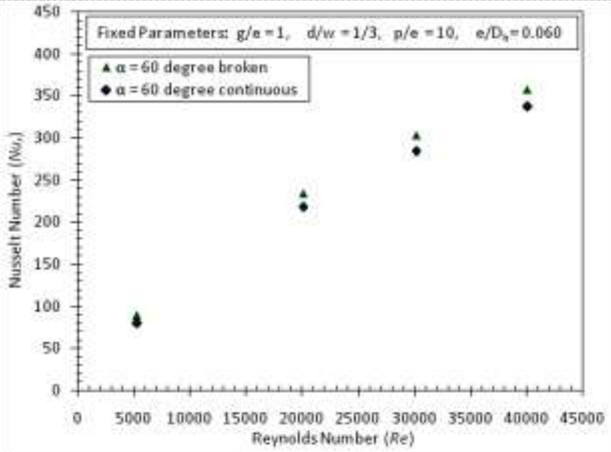

FIG. 10. Variation of Nusselt number with Reynolds number for V-shaped rib with gap and continuous V-shaped rib.

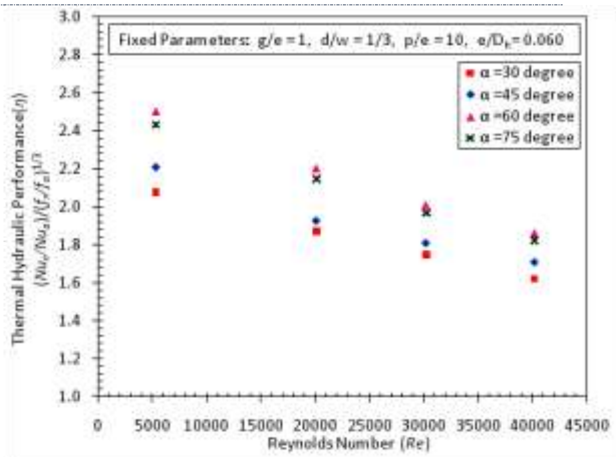

FIG. 12. Variation of thermo-hydraulic performance with Reynolds number for different angle of attack.

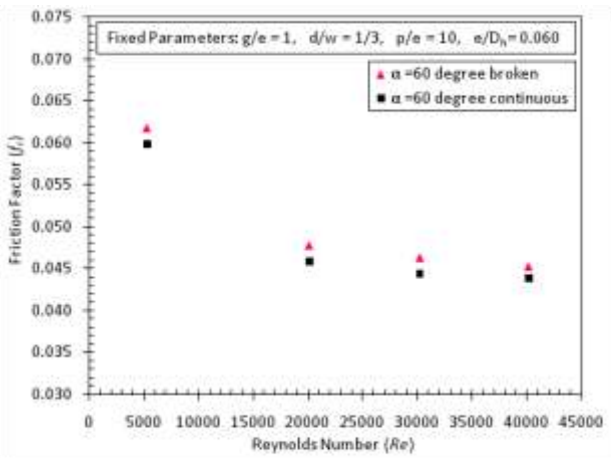

FIG. 11. Variation of friction factor with Reynolds number for V-shaped rib with gap and continuous V-shaped rib.

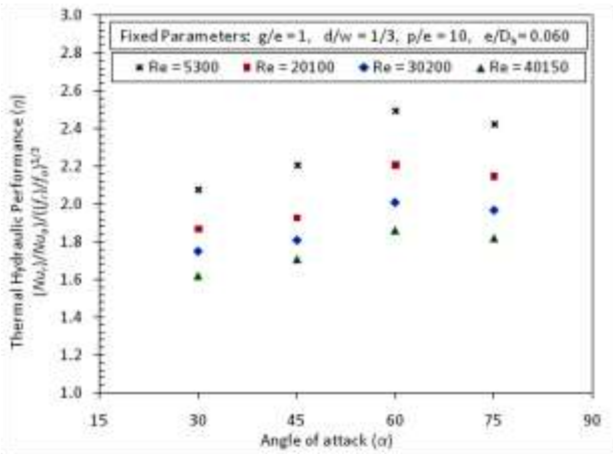

FIG. 13. Variation of thermo-hydraulic performance with angle of attack for selected Reynolds number.

Figure 7 and 8 illustrates the variation of friction factor and friction factor ratio with Reynolds number for different rib attack angles. It is observed that the friction factor and friction factor ratio increases as attack angle increases up to $60^0$and decreases with further increase in angle of attack. Friction factor remains almost constant at higher Reynolds numbers. The increase in friction factor ranges from 6.3 to 8.1. The friction factor enhancement is highest for 60^0 ribs and lowest for an an angle of attack of 30^0 (as shown in Fig. 9). Figure 10 and 11 clearly depicts the association of higher Nusselt number and friction factor as compared to that of the continuous V-shaped rib.

Analysis of heat transfer and friction factor characteristics shows that an augmentation in heat transfer is, in general, accompanied with a friction power penalty. Therefore, to select the best rib geometry the heat transfer and friction characteristics should be considered simultaneously. A parameter that facilitates the simultaneous consideration of thermal and hydraulic performance known as thermo-hydraulic performance [23] is evaluated for both the cases and plotted as in Figure 12 and 13. It is observed from the figures that, Thermo-hydraulic performance decreases with an increase in Reynolds number. In general, the thermo hydraulic performance improves with increase in angle of attack. The maximum value of this parameter occurs with an angle of attack of 60^0, which is about 2.6 times that of smooth duct, in the range of values investigated in the present study.

VI. CONCLUSIONS

The following conclusions can be drawn from the present 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.

(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 Nusselt number and friction factor of the roughened duct duct are a strong function of the angle of attack.

(4) The value of Nusselt number and friction factor increase with an increase of angle of attack and found maximum at an angle of attack of $60^{0.}$

(5) A gap equal to the height of the rib in both legs results in a considerable improvement in the thermo-hydraulic performance.

(6) The maximum enhancement in Nusselt number and friction factor values for V Rib with gap compared to smooth duct are of the order of 4.7 and 8.1 respectively.

(7) The thermal performance decreases when the Reynolds number increases. The V shaped ribs with gap at an angle of attack of 60^0 gave the highest thermo- hydraulic performance.

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