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Research Article

Augmented Heat Transfer in Square ducts with Transverse and Inclined Ribs with and without a gap

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Abstract

The effect of the rib angle orientation and influence of a gap provided in integral ribs on heat transfer and pressure drop in a square duct with two opposite in-line ribbed walls is investigated. The experimental investigation has been performed for continuous ribs (with no gap) and ribs with a gap having relative roughness pitch (p/e) of 10, relative roughness height (e/D_h) of 0.060 and rib attack angle of 90^0 and 60^0 for Reynolds number from 5000 to 40,000. Discrete ribs with relative gap position (d/W) of 1/5 and relative gap width (g/e) of 1.0 are investigated to compare their heat transfer performance with corresponding continuous ribs (without gape). The enhancement in heat transfer and friction factor of this roughened duct is also compared with smooth duct under similar flow condition. The results show that inclined ribs performs better than transverse ribs for both the cases i.e. for continuous ribs and ribs with a gap. The 60^0 ribs with a gap yields about 3.8-fold enhancements in Nusselt number and about 7.4-fold increase in the friction factor compared with smooth duct and about 1.1 times and 1.2 times that of 60^0 continuous ribs (without gaps) for the entire range of parameters investigated. However for 90^0 ribs the enhancement in heat transfer and factor is very low as compared to that of corresponding 60^0 ribs.

Keywords: *Nusselt number, friction factor, relative gap width, relative gap position, Reynolds number.*

1. Introduction

Gas turbines are extensively used for aircraft propulsion, land-based power generation, and industrial applications. Thermal efficiency and power output of gas turbines increases with increasing turbine rotor inlet temperature (RIT). The level and variation in the temperature within the blade material must be limited to achieve reasonable durability goals. Also the temperatures are far above the permissible metal temperature due to which there is a need to cool the blades to operate without failure.

One of the methods to cool the blades internally is by extracting the air from the compressor of the engine, which routed through serpentine channels within the blades and extracted the heat from the outsides of the blades. Internal cooling passages are mounted with ribs on channel walls. These ribs, which are also known as turbulators, increases the level of mixing by turbulence and disturb the laminar sub-layer, also increases the surface area for convective heat transfer, thereby enhances the cooling capacity of the passage.

The use of ribs, in addition to enhancing heat transfer coefficient considerably, results in higher frictional penalty. These ribs, which are also known as turbulators, increases the level of mixing by turbulence and disturb the laminar sub-layer, also increases the surface area for convective heat transfer, thereby enhances the cooling capacity of the passage. The use of ribs, in addition to enhancing heat transfer coefficient considerably, results in higher frictional penalty. Hence, it is essential to optimize the geometrical parameters of the artificial roughness (ribs) in order to achieve the maximum possible enhancement in heat transfer with minimum frictional penalty.

Geometrical parameters such as channel aspect ratio (AR), rib height-to-passage hydraulic diameter (e/Dh), 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 (Han et al., 2001). Han et al., (1978) investigated the effect of rib shape, angle of attack and pitch to rib height ratio on heat transfer and friction factor characteristics of a

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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 a relative roughness pitch of 10 and rib angle of attack of 450. Han et al., (1988) studied heat transfer and pressure losses with different angle ribs in square and rectangular channels. The rib angles for both square and rectangular channels were 90°, 60°, 45°, and 30°. The Reynods number was from 10000 to 60000. The higher heat transfer with higher pressure drop in the square channel was 60° rib angle.

Johnson et al., (1993), Taslim (1991), Han (1985), and Wagner (1992) studied the heat transfer and friction characteristics in rib-roughened passages with different rib arrangements. They focused on the effects of the Reynolds number and rib geometry on the heat transfer and pressure drop in the fully developed region of a uniformly heated square and rectangular channel. All these studies showed that angled ribs provide better heat transfer enhancement than transverse ribs. Zhang et al. (1992) and Kiml et al. (2001) reported that the thermal performance of rib arrangements with an angle of attack of 600 is better than that with an angle of 450. 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. Park et al. (1992) studied heat transfer and pressure losses with different angle ribs in square channel and rectangular channels with aspect ratio 1/4, 1/2 ,1, 2 and 4. The rib angles for both square and rectangular channels were 90°, 60°, 45°, and 30°. The Re was from 10000 to 60000. The low aspect ratio channels have the highest thermal performance. For low aspect ratio, the 45° and 60° have the highest thermal performance. For square channel, the 60° and 45° have the highest thermal performance. For large aspect ratio, 30° and 45° have the highest thermal performance. Taslim et al. (1997) study the effect of p/e and e/Dh on the heat transfer and friction losses 90° sharp angle ribs, and 90° round angle ribs in square duct. The e/Dh was 0.133, 0.167 and 0.25 and p/e was 5, 7, 8.5, and 10. Relative roughness pitch of 8.5 and 10 has the highest thermal performance. Han and Park (1998) observed that as the relative roughness pitch is reduced to the value below 8 the flow is not likely to reattach before it reaches the successive rib. On the other hand, an increase in relative roughness pitch above 10 the number of reattachment points per unit length will reduce as compared to those with the relative roughness pitch of 10.

In the majority of cooling channels, discrete ribs were shown to outperform the continuous angled or V-shaped ribs (Cho et al., 2000). Lau et al. (1991) 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 900 angle of attack shows 10-15% higher heat transfer coefficient as compared to the 900 continuous ribs, whereas inclined discrete ribs give 10-20% higher heat transfer than that of the 900 discrete ribs. Han and Zhang (1992) 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 the angle of attack of 450 and 600, the Reynolds number was from 15000 to 90000. They were reported that 600 V-shaped broken rib arrangements give better performance than 450 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 over the continuous ribs. Cho et al. (2003) examined the effect of angle of attack and number of discrete ribs in a square duct and observed that the gap region between the inclined discrete ribs accelerates the flow and enhances the local turbulence, which will results in an increase in the heat transfer. They also reported that the inclined rib arrangement with a downstream gap position shows higher enhancement in heat transfer compared to that of continuous inclined rib arrangement. Aharwal et al. (2008) experimentally investigated the heat transfer enhancement due to a gap in an inclined continuous rib arrangement in rectangular duct of solar air heater and observed the optimum performance for relative gap width of 1 and at relative gap position of 0.25. Thakur et al. (2009) performed study on absorber plate of solar air heater duct roughened with inclined discrete ribs. They reported that the maximum heat transfer enhancement occurs for the relative roughness pitch of 12, relative gap position of 0.35 and relative roughness height of 0.0498.

From above, it can be stated that discrete inclined or V-shaped rib arrangement can yield better performance as compared to continuous rib arrangement. Sufficient work has been done for discretized inclined ribs, but only few investigations have been given the information about the number of discretized parts of ribs in to which it should be broken so as to get the maximum enhancement in heat transfer. Some of the researchers given the results of providing gaps in inclined ribs in leading and trailing edge of the ribs. However, investigations have not been carried out so far to compare the results of transverse ribs with a single gap to the corresponding angled ribs so as to see the effect of inclination of rib along with the gap, particularly in case of two opposite rib roughened walls and for the parameters considered in the present study.

The present investigation has been carried out to compare the results of transverse ribs to that of corresponding inclined ribs for heat transfer and friction factor characteristics in a square duct with two opposite roughened walls. This study will help to show that, the inclined ribs with a gap perform far better than the transverse ribs.

2. Experimental Program and Range of Parameters

The experimental set up used for investigations of

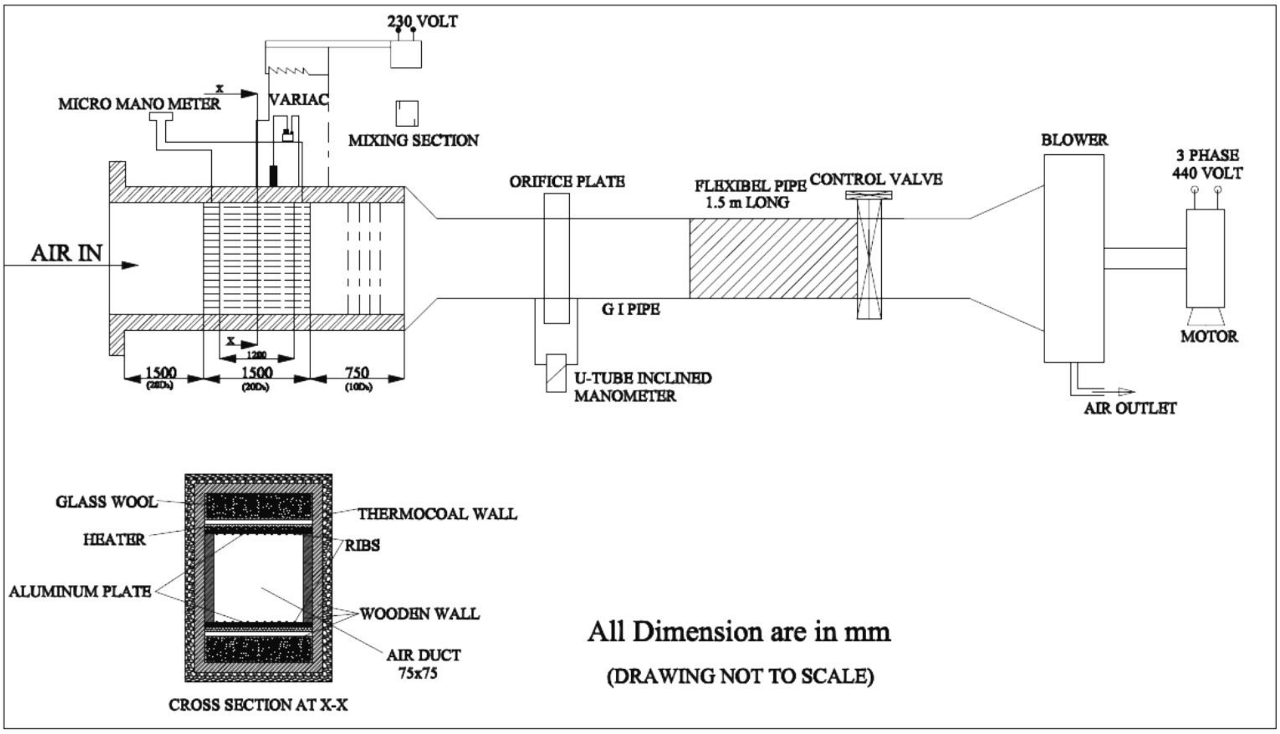

Figure 1: Schematic diagram of experimental set-up

convective heat transfer coefficient and friction factor consists of a square duct with entrance, test and exit sections, a blower, a hand operated control valve, a calibrated orifice meter, electrical heating arrangement and temperature measuring device. A schematic diagram of the test set-up is shown in Figure 1. 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 (20Dh), 1500 mm (20Dh) and 750 mm (10Dh) respectively (Han and Zhang (1992)). The entrance unheated duct serves to establish hydro-dynamically fully developed flow at the entrance to the test duct and unheated exit section is used downstream of the test section in order to reduce the end effect in the test section. The exit end of the duct is connected to 81 mm internal diameter G. I. 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 sheet, so that the heat losses from the test section can be minimized.

The entrance and exit sections are made by 50 mm thick polished wooden walls. The square test duct consists of 6 mm thick heated aluminium plate on its top and bottom walls. The other two walls of test section areconstructed by 50 mm thick polished wood. The ribbed aluminium plates are made by gluing square aluminium ribs (4.5 mm x 4.5 mm) to the finished aluminium plate surface in a required distribution to serve as top and bottom ribbed walls of the test section. The thin layer of glue (5 min epoxy) provides electrical isolation but thermal conduction to ribs. The plates are heated from outside by means of separate heaters assembly, thus subjected to uniform heat flux (0-1500 W/m^2) and are insulated with 50 mm thick glass wool topped with 12 mm thick plywood. A calibrated orifice-meter connected with an inclined U-tube manometer used to measure the mass flow rate of air through the duct. A control valve is provided in the pipe line to connect the blower to regulate the flow rate of air. The temperature of the test plate is measured by 24 copper-constantan calibrated thermocouples, distributed along the length and across the span by drilling about 2 mm diameter holes around 3-4 mm deep at the back side of the plate. Four thermocouples are arranged span wise in the duct to measure the air temperature at exit of the test section and 2 thermocouples are used at inlet to the test section, to measure the entry temperature of air. A digital micro-manometer (Fluke-922) is used to measure the pressure drop across the test section.

Experimental data is collected under steady- state condition for different mass flow rate of air to give the flow Reynolds number in the range of 5000 - 40,000. The heat flux was set and kept constant for each run so as to maintain the temperature of roughened plate around 20^{0}C- 30^{0}C above to that of mean bulk air temperature, to minimize the error. The dimensionless roughness parameters are determined by rib height (*e*), rib pitch (*p*), gap position (*d*) and gap width (*g*). The schematic of the geometry of inclined discrete rib used in the present study is shown in Figure 2. In the present study, the relative roughness height (e/D_h) and relative roughness pitch (*p/e*) is kept constant as 0.060 and 10 respectively. The range of Reynolds number was maintained from 5000 to 40000 throughout the experimentation. Square ribs of cross section 4.5 mm X 4.5 mm are selected for the study. The relative gap position (d/W) and relative gap width (g/e) is taken as 1/5 and 1.0 respectively. Experimental data is collected for rib attack angle (α) = 90^0 and 60^0.

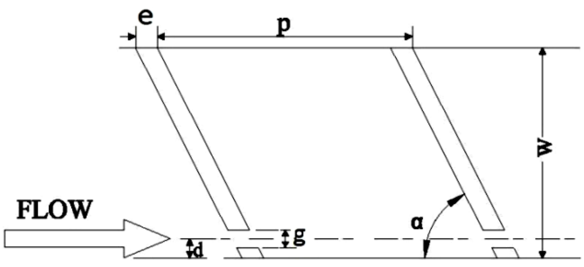

Figure 2: Schematic of roughness geometry of an inclined rib

4. Data Reduction

To determine heat transfer coefficient 'h', useful heat gain 'Q_u', Nusselt number 'Nu', Reynolds number 'Re' and friction factor 'f', the following procedure is adopted ;

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

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

where, C_d is the coefficient of discharge which is determined as 0.610 by calibration.

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

$$(\Delta P)_o = 9.81 \; X \; (\Delta h)_o \; X \; \rho_m \; X \; sin\theta \qquad (2)$$

The heat-transfer coefficient for the heated section was calculated as;

$$h = \frac{Q_u}{A_p (T_p - T_f)} \qquad (3)$$

where, heat transfer rate, Q_u to the air is given by

$$Q_u = m \; C_p \; (T_o - T_i) \qquad (4)$$

where T_p and T_f are average temperature values of test plate and fluid respectively. The average value of plate temperature (T_p) is calculated as a weighted mean of the plate temperature measured at different locations.

The convective heat transfer coefficient is then used to obtain Nusselt number, Nu , as

$$Nu = \frac{hD_h}{k_a} \qquad (5)$$

The Reynolds number was determined from the value of velocity of air through the duct, using equation:

$$\mathrm{Re} = \frac{\rho_a \; V D_h}{\mu_a} \qquad (6)$$

where,

$$V = \frac{m}{\rho_a .W.H} \qquad (7)$$

The friction factor was 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.

$$f = \frac{2\,(\Delta P)_d \; D_h}{4\,\rho_a \; L_f V^2} \qquad (8)$$

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

$$(\Delta P)_d = 9.81 \; X \; (\Delta h)_d \; X \; \rho_m \qquad (9)$$

The uncertainties in the calculated values of Reynolds number, Nusselt number and Friction factor are estimated as ±1.65 %, ±1.94 %, ±3.22 % respectively (Holman, 2004).

5. Validation of Experimental Data

The values of Nusselt number and friction factor determined from the experimental data were compared with the values obtained with the standard Dittus-Boelter equation for the Nusselt number and Blasius equation 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 figure 3 and in figure 4 respectively. The average deviation between the predicted and experimental values has been found to be ±3.4 % and ±5.9 % for Nusselt number and friction factor respectively. This shows a good match between the two values, which ensures the accuracy of the experimental data with the present experimental set-up.

6. Results and Discussion

The effect of various flow and roughness parameters on the heat transfer and friction factor characteristics in case of an artificially rib-roughened square duct has been investigated. Results have also been compared with those of the smooth duct under similar experimental conditions to determine the enhancement in the heat transfer coefficient and friction factor.

Figure 5 shows the variation of the Nusselt number as a function of Reynolds numbers for various rib roughness cases. For all the cases considered, Nusselt number increases as Reynolds number increases. It is seen that the Nusselt number have maximum value for 60^0 inclined ribs having a gap. The variation in Nusselt number ratio with Reynolds number is shown in figure 6, which shows that the maximum heat transfer enhancement is in the range of 3.0 to 3.8, occurs in the case of 60^0 rib (with gap) for the entire range of Reynolds number. Also the gap doesn't produce significant increase in heat transfer enhancement in 90^0 rib case. These results are comparable with the results of Cho et al. (2003) and Aharwal et at. (2008) which show that the gap in ribs will accelerate the flow and increases the heat transfer.

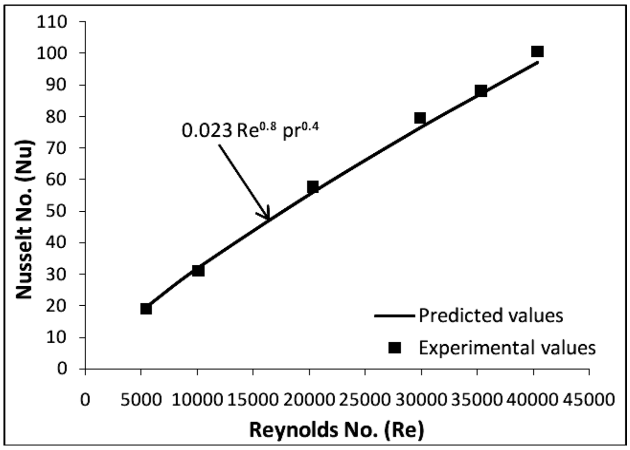

Figure 3: Comparison of experimental and predicted values of Nusselt number for smooth duct

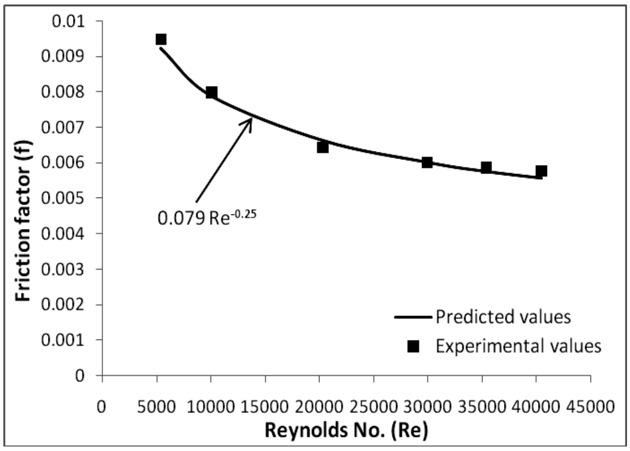

Figure 4: Comparison of experimental and predicted values of friction factor for smooth duct

Experimental values of friction factor have been plotted with Reynolds number (Figure 7). The variation of friction factor ratio as a function of Reynolds number for all the cases considered in the present study is shown in Figure 8. It is observed that for all the cases studied, the friction factor decreases as Reynolds number increases for the entire range of parameters investigated. Also the friction factor ratio increases as Reynolds number increases. Ribs with $\alpha=60^0$ (with gape) imposes the maximum frictional penalty. The maximum enhancement in friction factor is noted in the range of 6.6 to 7.4 in this case.

The thermo-hydraulic performance parameter is used to evaluate the effectiveness of artificially roughened surfaces accounting for the enhancement of Nusselt number and friction factor and is expressed as $(Nu_r/Nu_o)/(f_r/f_o)^{1/3}$ (Lewis, 1975). On the basis of results of heat transfer and friction characteristics of the different cases, it is observed that an enhancement in heat transfer is always accompanied with friction penalty.

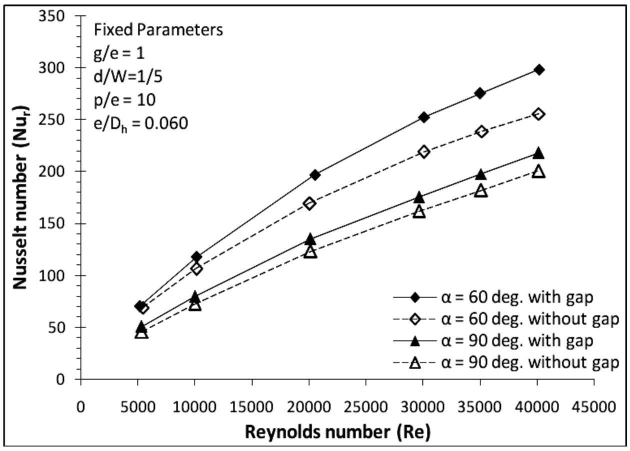

Figure 5: Varition of Nusselt number with Reynolds number for different cases

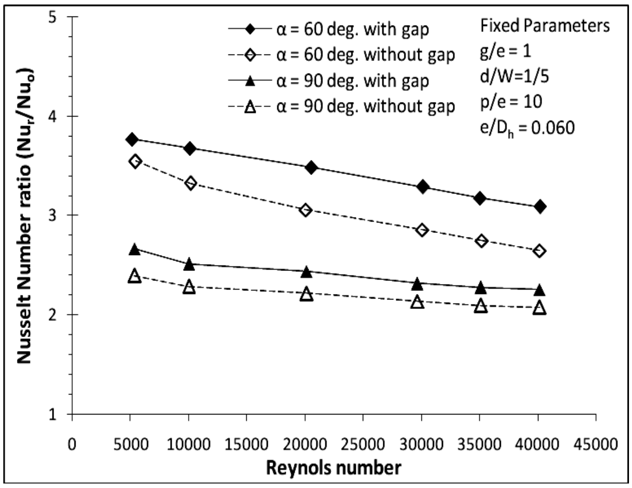

Figure 6: Varition of Nusselt number ratio with Reynolds number for different cases

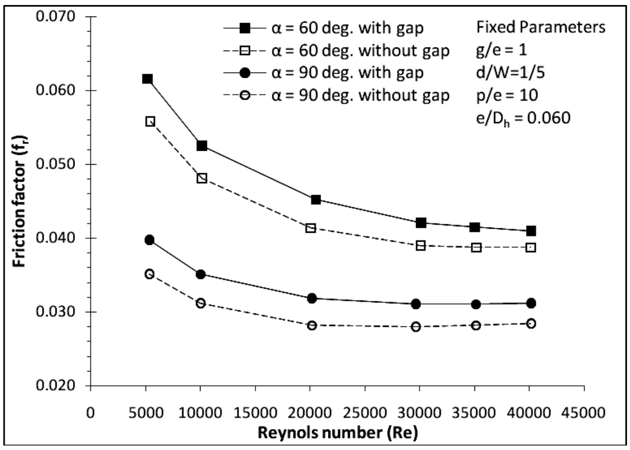

Figure 7: Varition of friction factor with Reynolds number for different cases

Figure 9 have been drawn between thermo-hydraulic performance parameter (η) and Reynolds number for all the cases, which shows that the inclined ribs with a gap provide higher thermo-hydraulic performance compared to that of the corresponding continuous rib arrangement for the entire range of Reynolds number. It is also seen that the value of this parameter decreases with increase in Reynolds number.

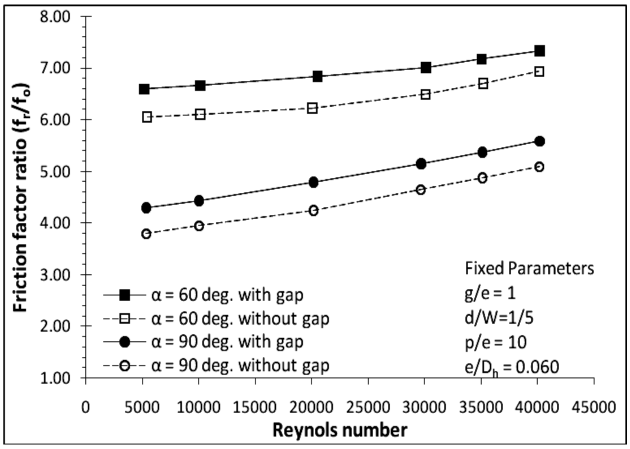

Figure 8: Varition of Friction factor ratio with Reynolds number for different cases

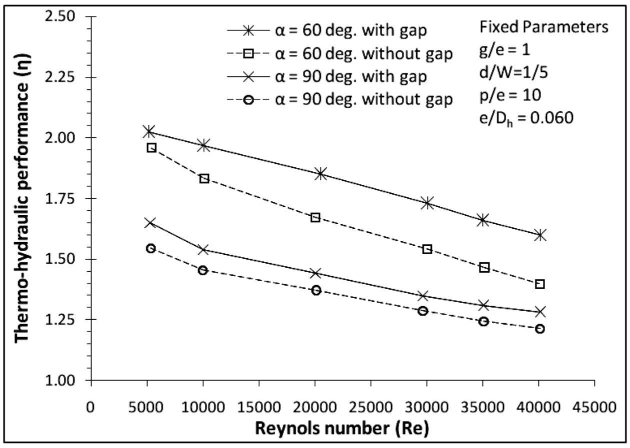

Figure 9: Varition of Thermo-hydraulic performance with Reynolds number for different cases

6. Conclusion

On the basis of experimental results of heat transfer and friction factor characteristics, it is found that by adducing gap in the inclined rib, there is considerable enhancement in heat transfer coefficient and friction factor of a square opposite side artificially roughened duct. The main findings are:

1. A gap in the inclined rib arrangement enhances the heat transfer and friction factor of the square roughened duct. Whereas, for transverse ribs ($\alpha = 90^0$) the enhancement in heat transfer and friction factor is substantially less as compared to that of inclined ribs.
2. The maximum values of Nusselt number, friction factor and thermo-hydraulic performance parameter are observed for a gap in an inclined ($\alpha = 60^0$) rib in the entire range of Reynolds number.

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