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Thermo-Hydraulic Performance of a Roughened Square Duct Having Inclined Ribs with a Gap on Two Opposite Walls

S. Gupta[1], A. Chaube[2], P. Verma[3]
[1]Corresponding Author, Department of Mechanical Engineering, Rajiv Gandhi Technological University, Bhopal-462036 (M.P.), India.
[2]Department of Mechanical Engineering, Jabalpur Engineering College, Jabalpur-482011 (M.P.), India.
[3]Department of Industrial and Production Engineering, Jabalpur Engineering College, Jabalpur-482011 (M.P.), India.

Abstract:- Experimentation has been carried out to find the influence of a gap provided in ribs on thermo-hydraulic performance of a square duct roughened with discrete inclined ribs. The two opposite walls of the square duct are roughened with ribs having attack angle (α) of 45^0 and a gap on its length. The investigation has been performed for relative roughness pitch (p/e) of 10, relative roughness height (e/D_h) of 0.060 and Reynolds number is varied in the range of 5000-40,000. The other rib parameters; relative gap position (d/W) and relative gap width (g/e) is varied in the range of 1/4 – 2/3(4 steps) and 0.5 – 1.5 (3 steps) respectively. The comparative study of various cases shows that the ribs with a gap considerably enhance the value of thermo-hydraulic performance for the range of parameters taken for the present investigation. Presence of inclined ribs with a gap yields about 2.1-fold enhancements in thermo-hydraulic performance as compared to smooth duct. The maximum value of thermo-hydraulic performance parameter has been observed for relative gap width of 1.0 and the relative gap position of 1/3.

Keywords:- Relative gap width, Relative gap position, Reynolds number, Thermo-hydraulic performance.

I. INTRODUCTION

Gas turbines are extensively used for aircraft propulsion, land-based power generation, and industrial applications. According to the thermodynamics concept, gas turbine performance is improved when the gas temperature, which exit from combustion chamber, increases. This high gas temperature can melt turbine blades and cause thermal stress. Thus 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. Therefore, 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. Lewis [1] proposed a thermo-hydraulic performance parameter known as efficiency parameter *'η'*, which evaluates the enhancement of heat transfer of a roughened duct compared to that of smooth duct for same pumping power requirement and is defined as;

$$\eta = \frac{\left(Nu_r / Nu_o\right)}{(f_r / f_o)^{1/3}}$$

Thermo-hydraulic performance *(η)* having a value higher than unity ensures the fruitfulness of using enhancement in heat transfer without much friction penalty.

The heat transfer coefficient and friction factor are considerably affected by various 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 [2]. Han et al. [3] investigated 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 a relative roughness pitch of 10 and rib angle of attack of 45^0. Han et al. [4] studied heat transfer and pressure losses with different angle ribs (90°, 60°, 45°, and 30°) in square and rectangular channels. 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. Johnson et al. [5], Taslim [6], Han [7], and Wagner [8] 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. [9] and Kiml et al. [10] showed that the thermal performance of rib arrangements with an angle of attack of 60^0 is better than that with an angle of 45^0. Lau et al. [11] 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. [12] 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. 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. [13] study the effect of p/e and e/D_h on the heat transfer and friction losses for 90° sharp angle ribs, and 90° round angle ribs in square duct. The e/D_h was 0.133, 0.167 and 0.25 and p/e was 5, 7, 8.5, and 10. They found that relative roughness pitch of 8.5 and 10 has the highest thermal performance.

In the majority of cooling channels, discrete ribs were shown to outperform the continuous angled or V-shaped ribs [14]. Lau et al. [15] 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 90^0 angle of attack shows 10-15% higher heat transfer coefficient as compared to the 90^0 continuous ribs, whereas inclined discrete ribs give 10-20% higher heat transfer than that of the 90^0 discrete ribs. Tanda [16] 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. [17] 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. [18] 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 1.0 and at relative gap position of 0.25.Thakur et al. [19] 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.

The literature review reveals that discrete inclined or V-shaped rib arrangement can yield better performance as compared to continuous rib arrangement. Only few researchers have been given the information about the number of discretized parts of ribs in to which it should be broken so as to give the maximum thermal performance. 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 optimize the gap width between rib elements to form the discrete rib and also to locate the optimum position of this gap, particularly in case of two opposite rib roughened walls which is used in gas turbine blade internal cooling. Therefore, present study has been performed to determine the optimum location and width of gap in an inclined rib to form a discrete rib for maximum thermal performance of a roughened square duct.

II. EXPERIMENTAL PROGRAM

A schematic diagram of the test set-up used in the present study 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 ($20D_h$), 1500 mm ($20D_h$) and 750 mm ($10D_h$) respectively [9]. 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.

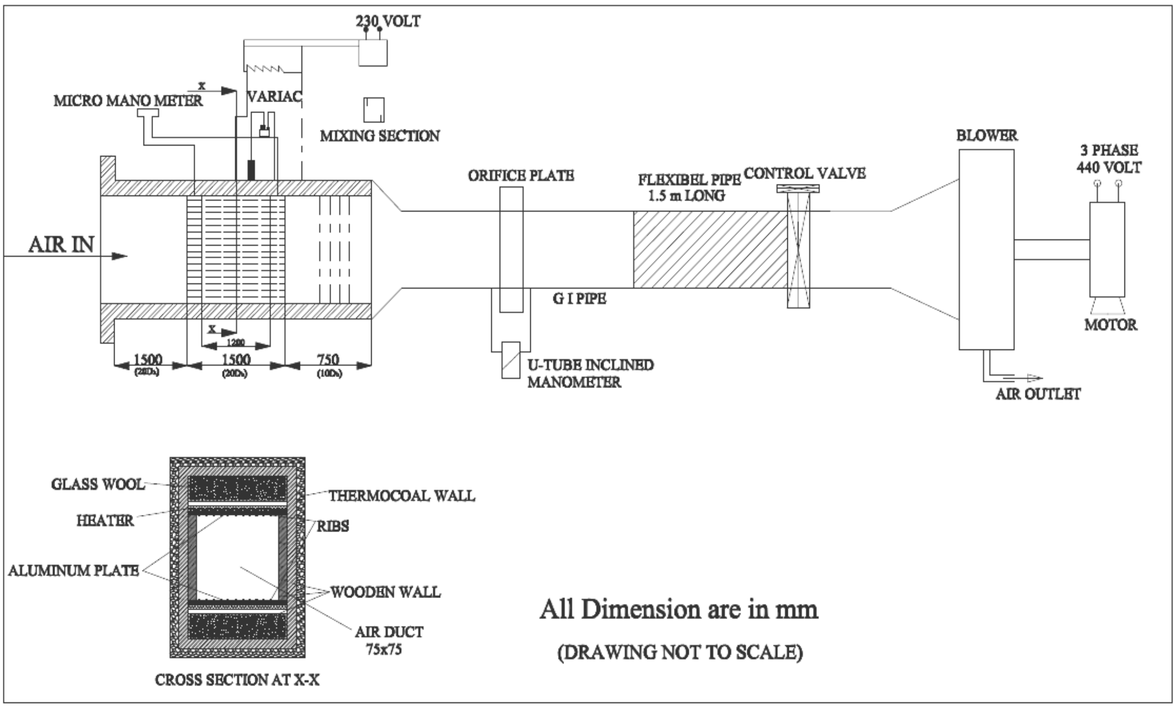

FIG. 1: Schematic diagram of experimental set-up

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 are constructed 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 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 temperatures of the test plate are 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 20^0C - 30^0C above to that of mean bulk air temperature, to minimize the error.

III. ROUGHNESS GEOMETRY AND RANGE OF PARAMETERS

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. The values of system and operating parameters are selected on the basis of related literature [4, 6, 9, 12, 13, 15, 17, 18, 19] and are given in Table I.

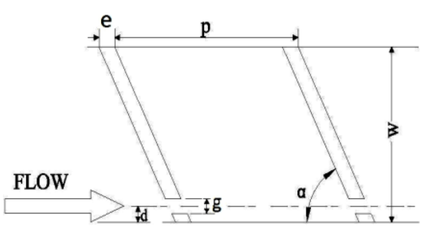

FIG. 2: Schematic of roughness geometry of an inclined rib

Table I: Range of parameters

S. No.	**Parameters**	**Range of parameter**
1	Reynolds number, (*Re*)	5000 - 40000
2	Relative roughness height, (e/D_h)	0.060
3	Relative gap width, (*g/e*),	0.5, 1, 1.5 (3 steps)
4	Relative gap position, (*d/W*)	1/4,1/3,1/2 and 2/3 (4 steps)
5	Rib attack angle, (α)	45^0
6	Relative roughness pitch (*p/e*)	10

IV. 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.

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

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

$$(\Delta P)_o = 9.81 \ X \ (\Delta h)_o \ X \ \rho_m \ X \ \sin\theta \tag{2}$$

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

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

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

$$Q_u = m \ C_p \ (T_o - T_i) \tag{4}$$

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

$$Nu = \frac{hD_h}{k_a} \tag{5}$$

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

$$\mathrm{Re} = \frac{\rho_a V D_h}{\mu_a} \tag{6}$$

where,

$$V = \frac{m}{\rho_a . W . H} \tag{7}$$

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.

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

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

$$(\Delta P)_d = 9.81 \ X \ (\Delta h)_d \ X \ \rho_m \tag{9}$$

The uncertainty analysis as proposed by Kline and McClintock [20] was used for the prediction of uncertainty associated with the experimental results based on the observations of the scatter in the raw data used in calculating the results. The uncertainties in the calculated values of Reynolds number, Nusselt number and Friction factor are estimated as ± 1.65 %, ± 1.94 %, ± 3.22 % respectively.

V. VALIDATION OF EXPERIMENTAL DATA

The values of Nusselt number and friction factor determined from the experimental data are compared with the values 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 [21]. 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.

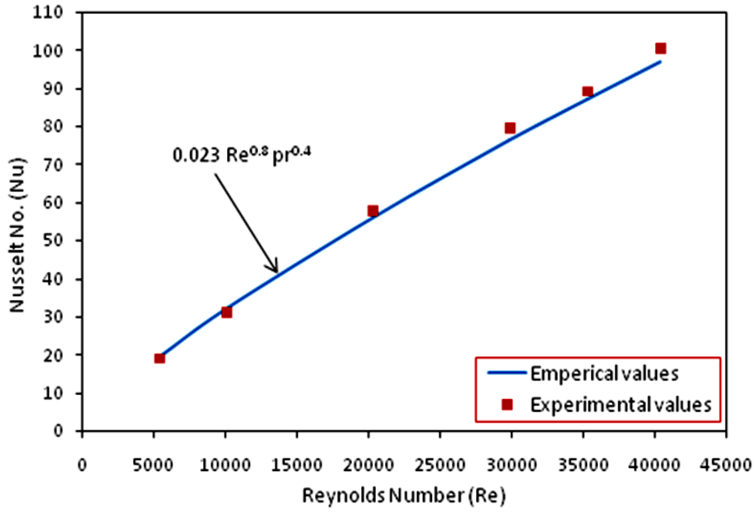

FIG. 3: Comparison of experimental and predicted values of Nusselt number for smooth duct

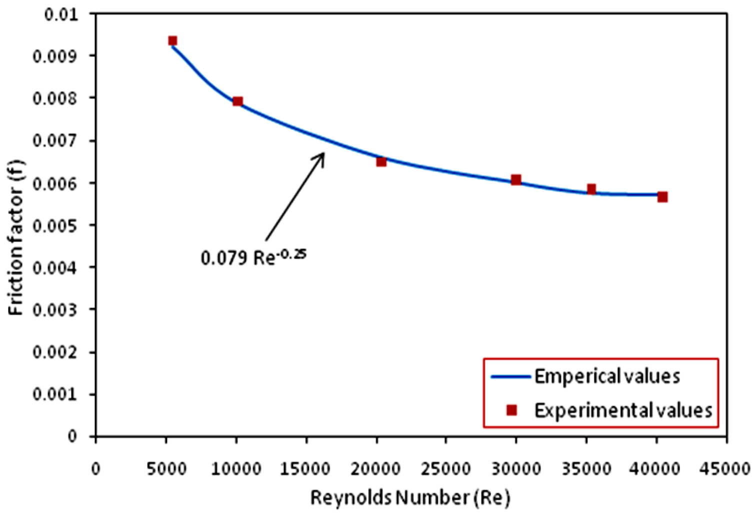

FIG. 4: Comparison of experimental and predicted values of friction factor for smooth duct

VI. RESULTS AND DISCUSSION

The effect of flow and roughness parameters on the thermal-hydraulic performance of an artificially rib-roughened square duct has been investigated and presented in Figs. 5- 9. Fig. 5 to Fig. 8 shows the effect of relative gap width on thermo-hydraulic performance for different relative gap positions at few selected Reynolds numbers. The thermo-hydraulic performance of roughened duct considerably increases due to gap in ribs as compared to smooth duct. As expected, the thermo-hydraulic performance of roughened duct decreases as the Reynolds number increases due to high turbulence effect at high Reynolds number for all the cases. It is observed that the maximum value of thermo-hydraulic performance occurs at relative gap width of 1 for all gap positions and also for all Reynolds numbers. This result conforms the result of Aharwal et. at. [17].The enhancement in thermo-hydraulic performance varies in the range of about 1.3 to 2.1 times of smooth duct (without ribs) in the range of flow parameter (Re varies from 5000 to 40000).

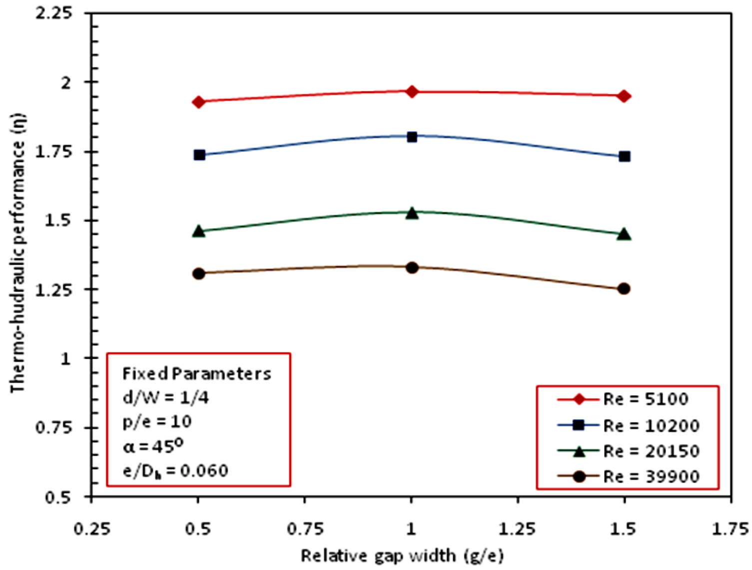

FIG. 5: Effect of relative gap width on thermo-hydraulic performance for d/W = 1/4

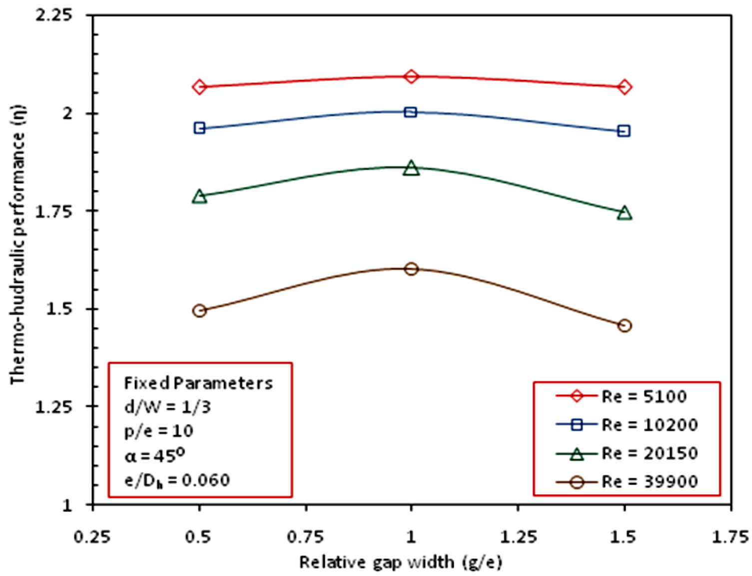

FIG. 6: Effect of relative gap width on thermo-hydraulic performance for d/W = 1/3

The thermo-hydraulic performance increases with increase in gap width from 0.5 to 1.0 beyond which it decreases with increase in gap width throughout the range of Reynolds number. This is prompted as by creating a gap in rib, it promotes local turbulence and flow mixing along the gap flow region while the rib induced secondary flow is usually maintained in the duct. It may, therefore, reasoned that the widening the gap width beyond 1 reduces the flow velocities through the gap and hence local turbulence, whereas too small gap width will not allow sufficient amount of secondary flow of fluid to pass through it and hence the turbulence level will remain low which ultimately results in low Nusselt number ratio (Nu_r/Nu_o) as compared to g/e =1.

To bring out the effect of relative gap position clearly, the value of thermo-hydraulic performance is plotted as a function of relative gap position and is shown in Fig.10 at few selected Reynolds number and for relative gap width of 1.0. It is clear that thermo-hydraulic performance attains a maximum value at relative gap position of 1/3. It is note that the inclination of rib creates a high heat transfer region at the leading edge and a low heat transfer region at the trailing edge [22]. The gap in trailing edge region helps in improving the Nusselt number ratio and is observed till the gap position is about 1/3 of the duct width. The gap at a position close to duct wall (d/w = 1/4) is not likely to produce similar effect as lateral boundary layer near to the wall may plays a dominant role. Thus placing the gap closer to the trailing edge side of inclined rib reckons poor results. Whereas, if, a gap is created near the leading edge (d/w = 2/3) the secondary flow weakens and thus this gap position does not lead to significant increase in thermo-hydraulic performance.

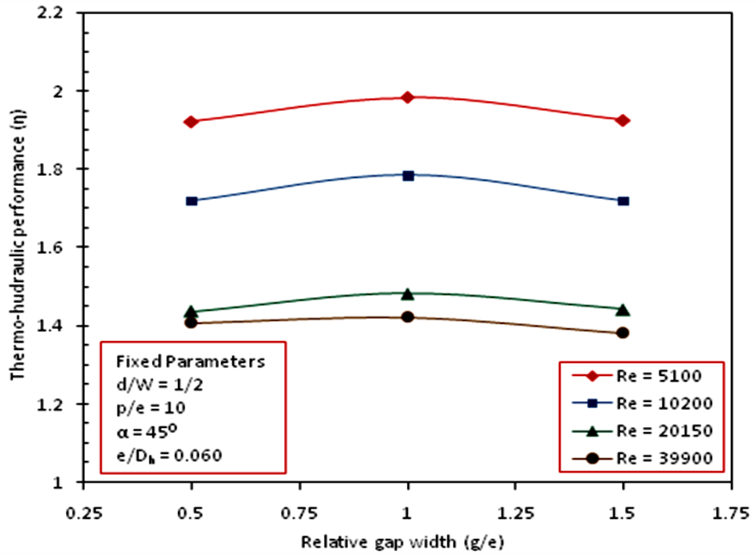

FIG. 7: Effect of relative gap width on thermo-hydraulic performance for d/W = 1/2

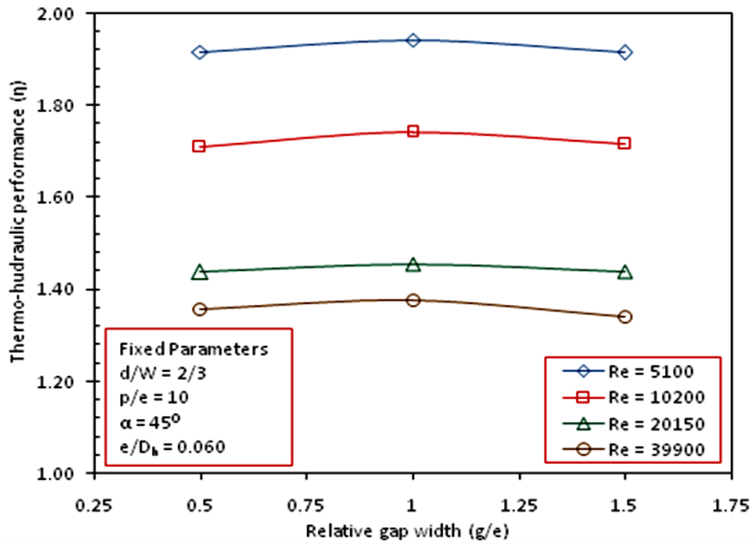

FIG. 8 Effect of relative gap width on thermo-hydraulic performance for d/W = 2/3

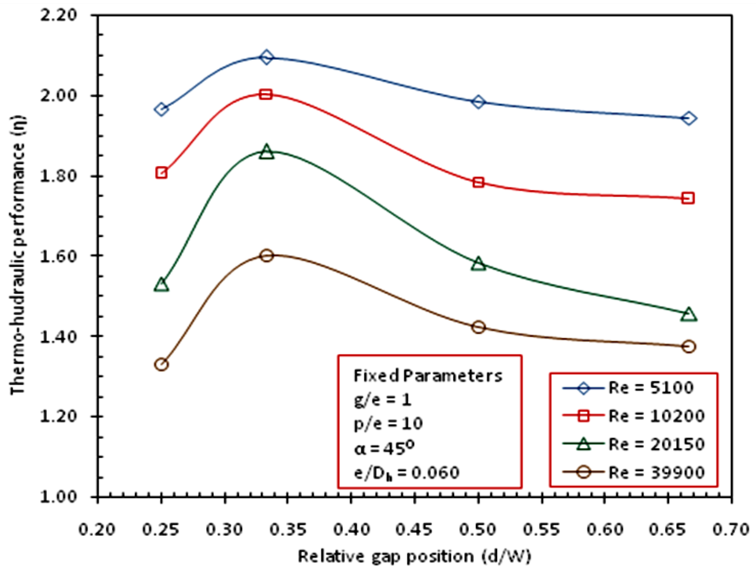

FIG. 9: Effect of relative gap position on thermo-hydraulic performance for g/e = 1

VII. CONCLUSION

On the basis of experimental results it is found that by providing gap in the inclined rib, there is considerable enhancement in thermo-hydraulic performance of a square opposite side artificially roughened duct. The main findings are:

1. The thermo-hydraulic performance of roughened surface having inclined ribs with a gap yields an increase of about 2.1 times as compared to that of smooth duct in the range of Reynolds number from 5000 to 40000.
2. The value of thermo-hydraulic performance decreases with the increase in Reynolds number.
3. The relative gap position of 1/3 and the relative gap width of 1 give the highest value of thermo-hydraulic performance parameter.

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 const. pressure, J/kg K
D_h	Channel hydraulic diameter of duct, m (= 4 W H /2(W+H))
D_p	Inside diameter of pipe, m
d/W	Relative gap position
e	Rib height, m
e/D_h	Relative roughness height
f	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 to calculate 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, ^0C or K
T_i	Air inlet temperature, ^0C or K
T_o	Air outlet temperature, ^0C or K
T_p	Mean plate temperature, ^0C 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
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}

REFERENCES

[1] M. J. Lewis, "Optimizing the thermo hydraulic performance of rough surface", *international Journal of Heat and Mass transfer*", 18, (1975), pp. 1243-1248.

[2] J. C. Han, S. Dutta, and S. V. Ekkad, "*Gas Turbine Heat Transfer and Cooling Technology*", Taylor & Francis, Inc., New York, (2001).

[3] J.C. Han, L.R. Glicksman, and W.M. Rohsenow, "An Investigation of Heat Transfer and Friction for Rib-Roughened Surfaces," *International Journal of Heat and Mass Transfer*, 21, (1978), pp. 1143-1156.

[4] J.C. Han and J.S. Park, "Developing Heat Transfer in Rectangular Channels with Rib Turbulators," *International Journal of Heat and Mass Transfer*, 31,(1988), pp. 183-195.

[5] B. V. Johnson, J. H. Wagner, G. D. Steuber, and F. C. Yeh, ''Heat Transfer in Rotating Serpentine Passages With Selected Model Orientations for Smooth or Skewed Trip Walls,'' ASME Paper 93-GT-305 (1993).

[6] M. E. Taslim, ''An Experimental Investigation of Heat Transfer Coefficients in a Spanwise Rotating Channel With Two Opposite Rib-Roughened Walls,'' *ASME Journal of Turbomachinery*, 113, (1991), pp. 75–82.

[7] J. C. Han, ''Heat Transfer Enhancement in Channels With Turbulence Promoters,'' *ASME J. of Engg. Gas Turbines Power*, 106, **(**1985), pp. 774–781.

[8] J. H. Wagner,‘‘Heat Transfer in Rotating Serpentine Passages With Trips Normal to the Flow,’’ *ASME Journal of Turbomachinery*, 114, (1992), pp. 847–457.

[9] J.C. Han and Y.M. Zhang, “High Performance Heat Transfer Ducts with Parallel and V-Shaped Broken Ribs,” *International Journal of Heat and Mass Transfer,* 35, (1992), pp. 513-523.

[10] R. Kiml, S. Mochizuki, A. Murata ‘‘Effects of rib arrangements on heat transfer and flow behavior in a rectangular rib roughened passage , *Journal of heat transfer*, 123,(2001), pp. 675-681.

[11] S.C. Lau, R.D. McMillin, J.C. Han, “Heat Transfer characteristics of turbulent flow in a square channel with angled rib”, Trans. ASME, *Journal of Turbomachinery*, 113, (1991), pp 367-374.

[12] J.S. Park, J.C. Han, Y. Huang, S. Ou, “Heat transfer performance comparisons of five different rectangular channels with parallel angled ribs”, *International Journal of Heat Mass Transfer,* 35 (11), (1992), pp. 2891–2903.

[13] M.E. Taslim , and C.M. Wadsworth, "An experimental investigation of rib surface- average Heat Transfer Coefficient in Rib-Roughened Square Passage", *Journal of Turbomachinery*, 119(2),(1997), pp. 381-389.

[14] H.H. Cho, S.J. Wu, and H.J. Kwon, “Local Heat/Mass Transfer Measurements in a Rectangular Duct with Discrete Ribs,” *ASME Journal of Turbomachinery,* 122 (2000), pp. 579-586.

[15] S.C. Lau, R.D. McMillin, J.C. Han, “Turbulent heat transfer and friction in a square channel with discrete rib turbulators”, Trans ASME, *Journal of Turbomachinery,*113, (1991), pp. 360-366,.

[16] G. Tanda, “Heat transfer in rectangular channel with transverse and V-shaped broken ribs”, *Int. Journal of Heat and Mass Transfer*, 47, (2004), pp. 229–243.

[17] H.H. Cho, Y.Y. Kim, D.H. Rhee, S.Y. Lee, S.J. Wu, “The effect of gap position in discrete ribs on local heat/mass transfer in a square duct”, *Journal of Enhanced Heat Transfer*, 10 (3), (2003), pp. 287-300.

[18] K. R. Aharwal, B. K. Gandhi, J. S. Saini, “Experimental investigation on heat-transfer enhancement due to a gap in an inclined continuous rib arrangement in a rectangular duct of solar air heater”, *Journal of Renewable Energy*, 33 (2008), pp. 585-596.

[19] S. K. Thakur, V Mittal, N. S. Thakur, and A. Kumar, “Heat Transfer and Friction Factor Correlations for rectangular Solar Air Heater Duct Having 60° Inclined Continuous Discrete Rib Arrangement”, *British Journal of Applied Science & Technology,*1(3), (2011), pp. 67-93.

[20] J. P. Holman, Experimental methods for engineers, *Tata Mcgraw Hills, New Delhi*, 2004.

[21] M. S. Bhatti, R. K. shah, “Turbulent and transition flow convective heat transfer, Handbook of Single-phase Convective Heat Transfer, *wiley, New York*, 1987.

[22] J. C. Han, W. L. Fu, L. M. Wright, “ Thermal performance of angled, V-shaped and W-shaped rib tabulators in rotating rectangular cooling channel”, Trans. ASME, *Journal of Turbomachinery,* (2004),126, pp. 604-614.