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Numerical Investigation of Heat Transfer and Pressure Drop in a Two-Pass Channel with Criss-Cross Rib Patterns

Arjumand Rasool[1,#], Sarwar Masoodi[2], Raof Ahmad Khan[3]
[1,3]Department of Mechanical Engineering, Govt. College of Engineering and Technology, Kashmir, India-191201
[2]Operations and Maintenance, Reliance Industries Limited, Srinagar, India-190001
[#]arjumand.beigh@yahoo.co.in

Abstract— Present computational investigation deals with the analysis of heat transfer and friction factor for turbulent flow of air through a two-pass square channel, having rib turbulators in a criss-cross pattern formed by 30° and 45° angled ribs. The cases undertaken are numerically investigated by commercial software COMSOL 5.2a with an established turbulence model i.e. Standard k-ε. The emphasis is towards investigating the potential impact of differing angle and comparative roughness pitch (p/ e) of criss-cross ribs. The measurement of heat transfer and friction factor has been carried out for Reynolds number ranging from 30,000 to 60,000. The validation of Nusselt number has been done by comparing with the experimentally and computationally obtained data from earlier studies under similar conditions. The impact of the Reynolds Number on the overall performance of the various criss-cross rib pitches was also investigated. The increment in average Nusselt number over that of the conventional 90° square rib roughened channel was 1.3 and friction factor gets lowered by a factor of 1.1 as compared to 90° ribs normal to the flow respectively. The analysis shows that characteristics of heat transfer distribution and fluid flow in between the ribs are significantly influenced due to rib configuration. The criss-cross ribs with angle 30° and pitch 12 shows better heat transfer and friction factor performance, and therefore guarantee an enhanced thermo-hydraulic performance. The overall averaged heat transfer coefficient along with the measured pressure drop across the test channel is used to calculate thermal performance based on constant pumping power criteria.

Keywords— Convective heat transfer; Turbine blade cooling; criss-cross ribs; Turbulence model; Comsol

I. Introduction

Gas turbine engines play a very important role in today's industrialized society. To combat the increasing demand for power, efficiency of gas turbine engines needs to be increased. One important method of increasing the efficiency of gas turbines is by increasing the turbine inlet temperature. Advanced gas turbines operate at very high turbine inlet temperature (around 1370°C) which is higher than the permissible limit of turbine blade material. Cooling of blade is required to have a reasonable blade life. Cooling is done by circulating air through the internal passages. These internal passages are connected by sharp 180° bend. Cooling air around 650°C is extracted from the compressor and passes through these cooling passages. With this cooling air, temperature of the blades can be lowered to approximately 1000°C, which is permissible for the reliable operation of the engine. The heat transfer augmentation and cooling techniques used in gas turbine are discussed in detail by Han *et al*. [1]. For internal flows, large scale surface roughness including repeated ribs [2-4], vortex generators [5, 6] and baffles/fins [7, 8] has been exhaustively employed for heat transfer enhancement. Rib turbulators on the duct walls are rigorously studied in order to enhance heat transfer rate, though at the cost of pressure penalty. These surface ribs enhance the turbulent fluctuations and could considerably modify the near-wall flow structures by creating the secondary flows and vortices. They penetrate the sub-layers of flow to cause the periodic redevelopment of boundary layers. The overall rib-effects on heat transfer are the improved heat transmission and the modified spatial distributions. In [9], the stagnant vortices behind the 90° transverse ribs raise the fluid temperature in the re-circulating zone and the wall temperature near the rib location. Between the 90°, 45°, V-shaped and discrete ribs, the 45° angled ribs show the highest thermal performance when the friction loss is considered [10]. Maurer *et al.* [11] investigated the heat transfer and pressure drop characteristics of V and W-shaped ribs for different rib pitch-to-rib height ratios. They observed that the thermal hydraulic performance of W-shaped ribs was better than V-shaped ribs. Wright *et al*. [12] investigated the thermal hydraulic performance of different continuous and discrete ribs in a high aspect-ratio rotating channel with Reynolds number ranging from 10,000 to 40,000. They observed that discrete V-shaped and W-shaped ribs performed better than the other configurations studied.

Several numerical studies have been conducted to study the flow and heat transfer characteristics in two-pass ribbed cooling channels. Bonhoff *et al*. [13] numerically studied the flow characteristics in a stationary square ribbed coolant channel with various turbulence models and validated it against PIV measurements. They observed that the results obtained with RSM model were most consistent with the experimental results amongst the models studied. Shevchuk *et al*. [14] investigated the heat transfer distribution in a two-pass varying aspect ratio channel with different divider wall-to-tip wall distances at a Reynolds number of 100,000. They reported good agreement between their computations and PIV

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measurements thereby illustrating that RANS models are capable of predicting the flow and heat transfer in a two-pass channel reasonably.

The objective of the present study is to assess the thermal hydraulic performance of three different criss-cross rib configurations and 90° ribbed channel using numerical simulations. Although 45° ribs have been well documented in the literature, studies pertaining to flow and heat transfer characteristics of criss-cross pattern ribs are sparse and limited to single pass channel only. Furthermore, parametric study of criss-cross ribs in two-pass channels has not been done yet, this study is directed at exploring the performance of the different criss-cross rib patterns at rib pitch-to-height ratios p/e varying from 7 to 12. Also, this study examines the performance of the different rib configurations in a developing channel flow which is common in internal cooling channels

II. Numerical Formulation and Physical Model

The schematic diagram of the computational model for Case-4 is shown in Fig. 1. All the dimensions of the geometry are in mm. The present computational study includes all criss-cross rib arrangements (Case 1-4) as depicted in Table 1. However, only mesh of two-pass channel with 45° criss-cross ribs is presented in Fig. 2. All the four tested cases differ only in rib arrangement keeping the cross-sectional area same. The length (l) of the test section is 1100 mm and the cross-section is 80 mm x 80 mm, and only the bottom wall is heated with a constant temperature of 100°C for all the summary of cases studied. Ribs were periodically mounted on bottom heated wall, eight in each pass. The pitch-to-rib height ratio (P/e) varies from 7-12, and the blockage ratio (e/D_H) of ribs is kept constant equal to 0.1625.

A. Computational Procedure and Validation

The concurrent prevailing partial differential equations subject to the boundary and initial conditions are solved numerically by using conjugate heat transfer (Turbulent k-ε) module of the commercial software COMSOL 5.2a which is based on the finite element method. The steps of the numerical model can be summarized as discretization of the domain and coupled governing equations (elements type and size) and relative and absolute tolerances or errors for the solution to reach convergence, determination of the iteration sequence nonlinear settings and selection of appropriate solver technique.

In the present study a simple linear free triangular mesh was generated in all the regions using Comsol software and Fig. 2 shows the computational grid around the ribs. In order to have homogeneity for all the rib cases studied, similar grid size was used for the different ribs. The distance of nearest nodes from the wall (y^+) was maintained to less than 10 in all the computations.

Current investigation uses the parametric segregated solver, which by means of pressure-correction scheme segregate energy and momentum equations. Earlier investigations delineated that RANS models can accurately predict the heat transfer and fluid flow characteristics in a two-pass channel.

B. Governing Equations

In present investigation, the fluid which is air in this study is supposed to be incompressible. The normal governing equations for different variables are expressed below. Since, the present study uses steady Reynolds averaged Navier-stokes (RANS) equations in the air domain and conductive and convective heat equations for the air and the solid wall. The Turbulent non-isothermal flow predefined multiphysics coupling sets up these application modes together with application couplings, making it easy to model Fluid- Thermal interactions. The general form of Reynolds-averaged Navier–Stokes (RANS) and energy equations can be expressed as follows, according to [15].

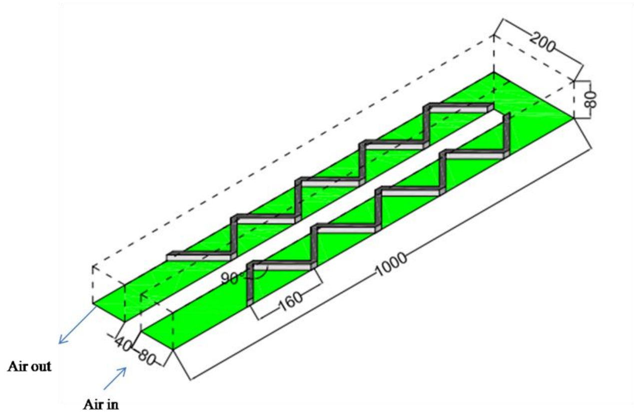
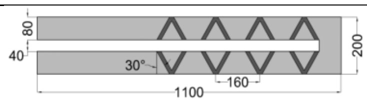
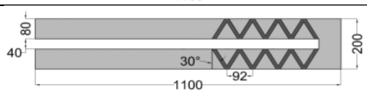
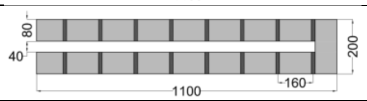
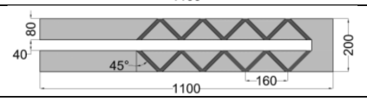


Fig. 1 Geometry of a computational domain

TABLE I. Construction details of different criss-cross rib configurations for current study

Test Cases	Rib Configuration	Rib Angle	P/e
Case 1		30°	12
Case 2		30°	7
Case 3		90°	12
Case 4		45°	12

Continuity equation:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

Momentum equation:

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$$\frac{\partial}{\partial x_i}(u_i u_j) = -\frac{1}{\rho}\frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_i}\left[\nu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_k}{\partial x_k}\right)\right] + \frac{\partial}{\partial xi}\left(-\overline{u_i' u_j'}\right) \quad (2)$$

where,

$$-\overline{u_i' u_j'} = \nu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}k\delta_{ij} \quad (3)$$

$$\nu_t = C_\mu k^2 / \varepsilon \quad (4)$$

Energy equations:

$$\frac{\partial}{\partial x_i}\left[u_i(\rho E + p)\right] = \frac{\partial}{\partial x_j}\left[\left(K + \frac{c_p \mu_T}{\mathrm{Pr}_T}\right)\frac{\partial T}{\partial x_j} + u_i(\tau_{ij})_{eff}\right] \quad (5)$$

$$(\tau_{ij})_{eff} = \mu_{eff}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}\left(\mu_{eff}\frac{\partial u_k}{\partial x_k}\right)\delta_{ij} \quad (6)$$

C. Boundary Conditions

Air with constant properties is selected as the working fluid. The flow is assumed to be steady and turbulent. Inlet temperature is considered to be uniform at 300K. Reynolds number is determined by the inlet fluid velocity. The turbulence intensity at inlet was fixed at 5% (air with absolutely no fluctuations in air speed or direction would have a turbulence intensity value of 0%). 7% of the hydraulic diameter of channel is taken as the length scale. The inlet boundary condition was set as velocity inlet and Boundary condition (outflow) was imposed at the outlet. A constant temperature boundary condition (100°C) was used for the ribbed bottom surface with other surfaces being adiabatic and no-slip.

D. Grid Independence Test

In order to get assured about the results obtained from the computation are not dependent on the mesh size, a grid independence study was done. The study of grid independence has been carried out with two-pass channel (smooth walls) at Reynolds number of 52000 with three grid sizes consisting of 200000, 250000 and 330000 elements. The results for the stream-wise variation of span-wise averaged normalized Nusselt number obtained by using various grids are compared and shown in Fig. 3. The variation in normalized Nusselt number for smooth channel using fine and extremely coarse mesh is 6.3%. In order to maintain balance in computational time and efficiency, a grid size consisting of 250000 elements has been selected for the present computational method. In some areas the mesh is finer for a better observation of the results. These are mostly near the bottom heated wall and ribs.

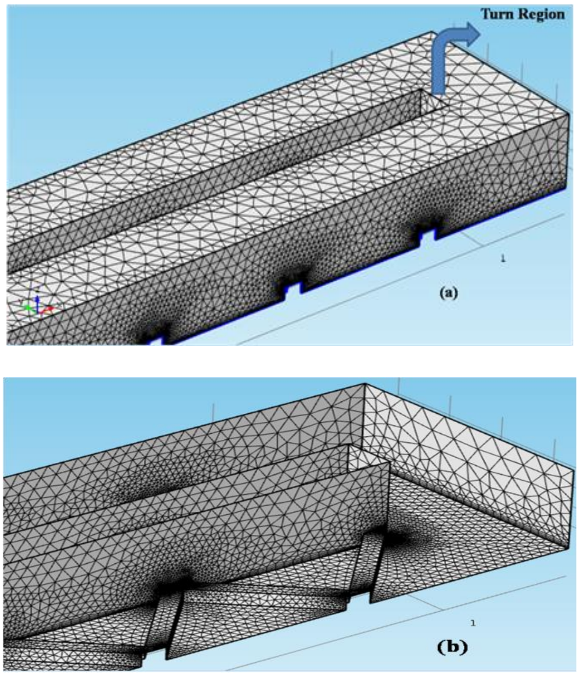

Fig. 2 (a) Overall view of the mesh and (b) Cut-section view of turn region for 45° criss-cross ribs at p/e 12

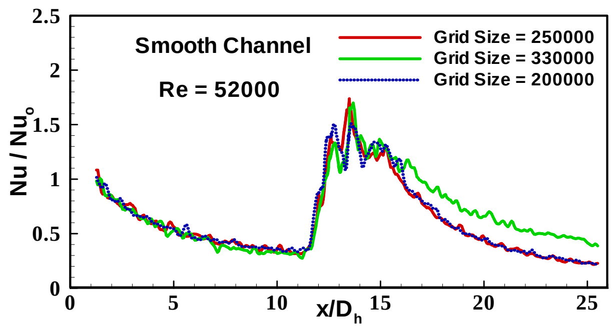

Fig. 3 Centerline variation of normalized Nusselt number along Stream-wise direction for three different grids (Re = 52000)

III. Data Reduction

The general formula for determining heat transfer coefficient for ribbed walls is

$$h = \frac{q''}{(T_w - T_f)} \quad (7)$$

Since, fluid velocity and turbulence variations near the wall are very small and can even vanish close to the wall. heat conduction by molecules in the viscous sub-layer carries the thermal, which makes the temperature to linearly vary with distance from the wall [26]. Thus, the heat flux can be determined by temperature gradient as

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$$q'' = k\frac{\partial T}{\partial n} \tag{8}$$

In order to determine the local bulk fluid temperature, the mass averaged fluid temperature was calculated at a number of cross-sections in the stream-wise direction along the length of the channel. A linear fit was then used to compute the local bulk fluid temperature.

$$Nu = \frac{hD_H}{k} \tag{9}$$

Normalization of Nusselt number is done by Dittus-Boelter equation

$$Nu_o = 0.023(\mathrm{Re})^{0.8}\,\mathrm{Pr}^{0.4} \tag{10}$$

Friction factor (f) has been determined by

$$f = \frac{\Delta p D_H}{2 l u_{in}^{\;2} \rho} \tag{11}$$

Friction factor is normalized by Blausius equation

$$f_o = 0.316/(\mathrm{Re})^{0.25} \tag{12}$$

A. Performance Evaluation

The performance evaluation of heat transfer enhancement is aimed at determining when it is useful to employ an augmentation case instead of an unaugmented case. In present investigation augmentation cases involve use of surface mounted ribs in criss-cross pattern at different angles and p/e ratio. A high value of this parameter indicates greater enhancement of heat transfer at small pressure drop. The thermal performance index (η) is given by

$$\eta_o = \frac{Nu/Nu_o}{(f/f_o)^{1/3}} \tag{13}$$

IV. Computational Results

Normalized Nusselt number contours are shown for all criss-cross rib configurations at Reynolds number range considered. Different configurations have been compared for their heat transfer characteristics through averaged Nusselt number enhancement Nu/Nu_o. Fluid flow in the duct has been discussed in order to understand the heat transfer enhancement mechanisms by the criss-cross rib turbulators. In the end, overall averaged Nusselt number enhancement, friction factor and thermal hydraulic performance is reported and compared with existing rib turbulator designs.

A. Validation with Experiment

Before carrying out the simulation for a ribbed channel, Validation has been performed to determine the heat transfer coefficient distribution for smooth channel (without ribs). Fig. 4 shows the comparison of current computational results with results of Jang *et al*. [16], Erille *et al*. [17] and Ekkad *et al*. [18]. The normalized Nusselt number trent of the present investigation show good agreement with the experimental results. The variations between the results become prominent behind the turn towards the outlet of the channel because now the flow conditions also depend on the width (thickness) of the divider wall. Also a number of studies [19, 20] have revealed that the simple eddy viscosity models are not capable of correctly anticipating the complex turbulent flow physics in the bend and associated regions in the second pass and therefore under predicts the Nusselt number than the actual. In general the matches between experimental and computational results were found to be satisfactory and therefore selected turbulence model was chosen for the computation.

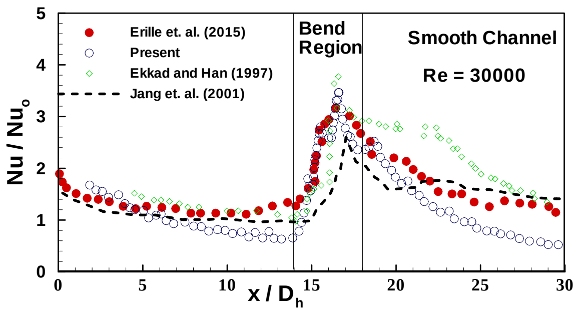

Fig. 4 Comparison of normalized Nusselt number with experimental results (Smooth channel Re = 30,000)

B. Ribbed Two-pass Channel: heat transfer and Fluid Flow Characteristics

In order to study the flow and heat transfer characteristics of ribbed two-pass channels, it is important to understand the innate behavior of the rib induced secondary flows for the different rib configurations. Fig. 5 shows the secondary flows induced by different rib configurations and the flow structure in the inter-rib region close to the bottom wall. To understand the mechanism of heat transfer in the inter-rib region of criss-cross ribs, it is required to analyze the fluid flow characteristics, which are responsible for this variation. The flow pattern and turbulence intensity are two major factors influencing the local heat transfer coefficient [21]. Many investigators have observed that the peak in local heat transfer coefficient occurs at reattachment point [22 –25]. The effects of criss-cross ribs on the overall heat transfer and pressure drop characteristics in the two-pass channel are first discussed with the Reynolds number varying from 30000 to 60,000.

Fig. 6 shows the detailed measurement of normalized Nusselt number Nu/Nu_o for the criss-cross configuration. The heat transfer enhancement mechanism of rib turbulator is through the increase in near wall shear due to rib induced secondary flows and the increase in turbulent mixing leading to energy exchange between enclosure walls and coolant. The secondary

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flows get induced due to the angle of attack of the ribs with respect to flow.

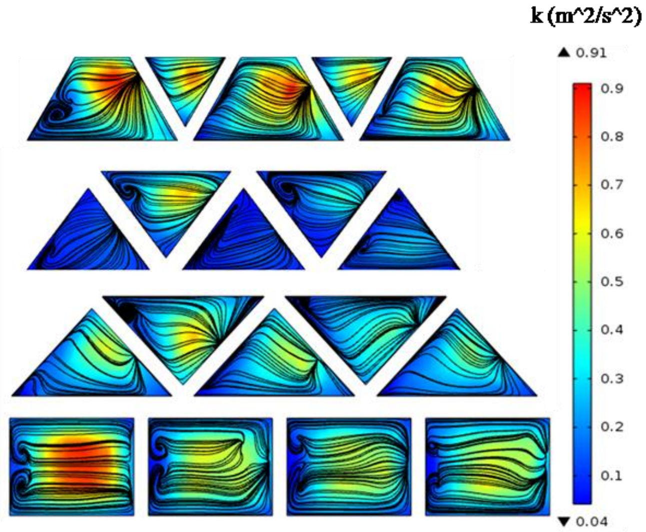

Fig. 5 Streamlines superimposed with normalized Turbulent kinetic energy (k) plotted at x-z plane for various rib configuration

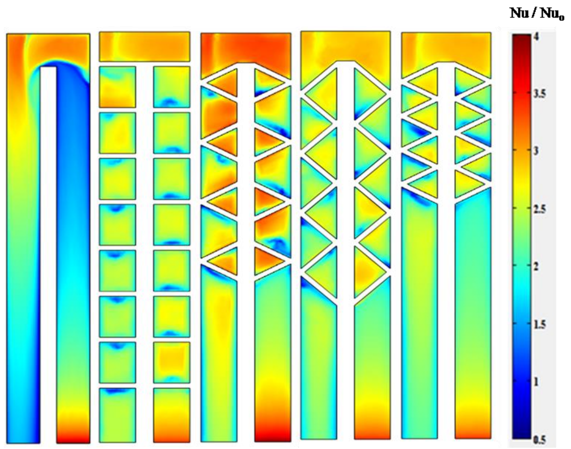

Fig. 6 Detailed Nusselt number ratio Nu/Nuo contour for the criss-cross configuration at Re = 40000

The coolant while passing over the ribs gets attached with the end wall slightly downstream of the rib. The space between the reattachment location and the rib undergoes fluid recirculation, which leads to relatively lower heat transfer. Figure 5 shows a unique pattern of secondary flows developed due to interaction of bulk flow and the ribs arranged in a new format. The secondary flows originated from both corners of the side walls and travelled inwards towards the channel centerline. The secondary flows induced by the ribs directly resulted in increase in near wall shear stress which lead to enhancement of heat transfer on the end walls featuring rib turbulators. Just downstream of the rib, lower levels of heat transfer were observed due to separation of coolant from the top of the rib, hence resulting in recirculation. The ribs were placed at an angle of attack of 45°and 30°, hence the flow near the wall was expected to travel at an angle to the bulk flow. However, due to the interaction of secondary flows and the core coolant flow, the resultant flow directions near the wall can be seen from the streamlines. Heat transfer enhancement decreases with increase in Re for all rib configurations, which is a well-known behaviour for ribbed channels [1] and is shown for case 2 in Fig. 7.

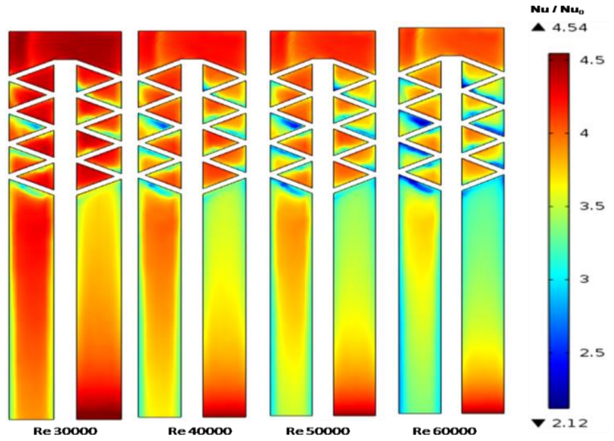

Fig. 7 Detailed Nusselt number ratio Nu/Nuo contour for the criss-cross configuration at different Reynolds number

C. Normalized Nusselt Number friction factor and thermal hydraulic performance

Fig. 8 shows the overall averaged Nusselt number for the four configurations with respect to Reynolds Number. Case1 shows slightly higher heat transfer enhancement compared to other two criss-cross configurations and 90° ribbed configuration. The enhancement in heat transfer for Case 1 is about 1.5 times compared to that of Case 4 for Reynolds Number 30000 and it decreased to about 1.2 times for the Reynolds number of 50,000. It is clear that the criss-cross ribs have higher levels of heat transfer enhancement compared to the continuous 90° ribbed channel. Fig. 9 and 10 shows the plot of normalized friction factor and thermal hydraulic performance for the cases studied. The friction factor is expected to increase with the Reynolds number. Case 4 shows higher friction factor than other configuration.

The overall thermal hydraulic performance factors for the various configurations have been compared in Fig. 10. To assess the best criss-cross configuration in the various test cases, the thermal performance factor was calculated for four different Reynolds numbers using Eq. (13). Case 1 shows best thermal hydraulic performance while case 4 shows poor performance. The reason for higher performance is attributed to uniform heat transfer distribution.

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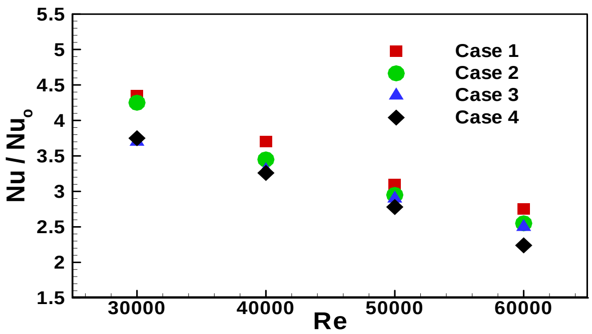

Fig. 8 Comparison of overall averaged Nusselt number enhancement versus Reynolds number

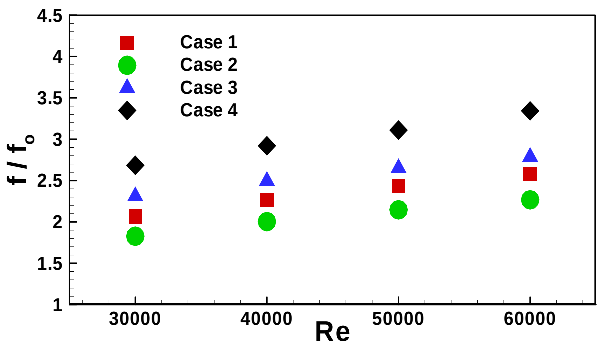

Fig. 9 Comparison of normalised friction factor

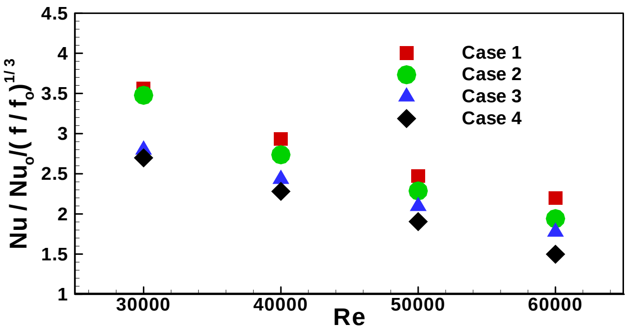

Fig. 10 Comparison of thermal hydraulic performance

V. CONCLUSIONS

In this work, numerical simulations were carried out in a stationary two-pass square duct with one wall heated. The three-dimensional turbulent fluid flow characteristics and Nusselt number ratio are presented for four different criss-cross rib configurations (Cases-1, 2, 3, and 4). The average values of normalized Nusselt number ratios and friction factor ratios are also reported for four different Reynolds numbers (i.e., 30,000, 40,000, 50,000 and 60,000). The following conclusions were obtained from the present study:

1. The current numerical simulation is able to predict the secondary flow patterns induced due to angular ribs, the strength of which are found to vary with different criss-cross rib configuration.
2. The overall normalized Nusselt number ratio decreases while friction factor ratio increases with an increase in Reynolds number for all test cases.
3. The heat transfer enhancement Nu/Nu_o is 2.2 to 4.4, for Reynolds number range 30,000 to 60,000. The enhancement level for case 1 is higher than the other configurations.
4. The friction factor for the case 1 configuration was relatively lower than the other criss-cross configurations.
5. The thermal hydraulic performance of case 1 is higher than that for other criss- cross configurations with values from 1.2 to 1.5 for Reynolds Number range 30,000 to 60,000.

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NOMENCLATURE

D_H	Hydraulic diameter	[m]
e	Rib height	[m]
f	Friction factor	
H	Heat transfer coefficient	[W/m^{2}K]
k	Thermal conductivity	[W/mK]
l	Channel length	[m]
Nu	Nusselt number	
p	Rib pitch	[m]
P	Pressure	[Pa]
q″	Heat flux	[W/m2]
Re	Reynolds number	[$\rho\, u_{in}\, D_H / \mu$]
u	Fluid velocity	[m/s]

Subscripts

o	From Dittus-Boelter Coorelation
f	Fluid
w	wall
in	inlet

Greek symbols

η	Thermo-hydraulic performance	
μ	Dynamic viscosity	[kg/ms]

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