NUMERICAL ANALYSIS OF LAMINAR HEAT TRANSFER IN A SQUARE DUCT WITH INCLINE DIAGONALLY ANGLES – RIBS

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Abstract

This work presents effect of periodic flow, friction loss and heat transfer in constant temperature-surfaced square duct incline diagonally angled - ribs. The computations are based on the finite volume method, and the SIMPLE algorithm has been implemented. The fluid flow and heat transfer characteristics are presented for Reynolds numbers based on the hydraulic diameter of the channel ranging from 100 to 1200. To generate vortex flows through the tested section, the 20° and 30° ribs were mounted in tandem and in-line arrangement on the opposite walls of the tested section. Effects of different generators heights, BR in range from 0.10-0.20 with single pitch ratio of 1.00 on heat transfer and friction losses in the test section were studied. It was apparent that vortex flows created by ribs existed and helped to induce impinging flows on the walls leading to drastic increase in heat transfer rate over the test section. In addition, the increase in the generators height and attack angle resulted in the rise of Nusselt number and friction loss values. The computational results revealed that the optimum thermal enhancement factor was about 2.24 for incline diagonally angled ribs at height of 0.20 times of the 20° attack angle with the highest Re.
Keywords: square tube, periodic flow, angled ribs, heat transfer

Introduction

The uses of turbulators in the cooling ducts or ducts heat exchanger such as rib, dimples, groove or baffle are often employed in order to increase the convective heat transfer rate leading to the compact heat exchanger and increasing the efficiency. The cooling or heating air is supplied into the channels mostly mounted with several ribs to increase the degree of cooling or heating levels and this configuration is often used in the design of heat exchangers. This is because the use of ribs completely makes the change of the flow field and thus the distribution of the local heat transfer coefficient. Although heat transfer is increased through the rib arrangement, the pressure drop of the ducts flow is also increased due to the decreased flow area effects. Therefore, rib spacing, angle of attack and height are among the most important parameters used in the design of ducts heat exchangers. Closer spacing or larger rib height causes higher heat transfer rate but resulting in poor stream distribution and higher pressure drop. On the other hand, higher baffle spacing or smaller baffle height causes the reduction of the pressure drop but provides more longitudinal flow leading to the decrease of heat transfer. It is, thus, difficult to realize the advantage of rib arrangements or geometry.

The first work on the numerical investigation of flow and heat transfer characteristics in a duct with the concept of periodically fully developed flow was conducted by Patankar et al. [1]. Berner et al. [2] suggested that a laminar behavior for a channel with transverse baffles mounted ontwo opposite walls was found at a Reynolds number below 600 and for suchconditions the flow was free of vortex shed-ding. Webb and Ramadhyani [3] numerically investigated the fluid flow and heat transfer characteristics in a smooth channel with staggered baffles, based on the periodically fully developed flow conditions of Patankar et al. [1]. Kel-lar and Patankar [4] studied the heat transfer behaviors in a channel with staggered baffles and reported that the heat transfer increases with the rise in baffle height and with the decrease in baffle spacing. Their results showed similar be-haviors as results of Webb and Ramadhyani [3]. Cheng and Huang [5] investigated the case of asymmetrical baf-fles and indicated that the friction factor shows a great de-pendence on baffle location, especially for a large height of baffle. Amiri et al. [6] investigated both experimentally and numerically the laminar flow and heat transfer in a two-dimensional channel with packed bed porous media using a two-phase equation model for the transport. A numerical investigation of laminar forced convection in a three-dimensional channel with baffles for periodically fully de-veloped flow and with a uniform heat flux in the top and bottom walls was conducted by Lopez et al. [7].Most of the investigations, cited above, have considered the heat transfer characteristics for transverse ribs/baffles placed repeatedly in square/rectangular channels only. The application of rib attached in square ductwalls has rarely been reported. In the present work, the numerical computations for three dimensional laminar periodic square duct flows over the 20˚ and 30˚ incline diagonal rib mounted periodically on the opposite of the tube wall are conducted with the main aim being to examine the changes in the flow structure and heat transfer behaviors. The application of the 20˚ and 30˚ incline diagonally rib placed periodically on the walls of the tested tube is expected to generate a longitudinalvortex flow through the tube to better mixing of flows between the core and the wall resulting in higher heat transfer rate in the square duct.

Mathematical Foundation

The numerical model for fluid flow and heat transfer in the square duct was developed under the following assumptions: steady three-dimensional, laminar and periodic incompressible fluid flow and ignoring body forces, viscous dissipation and radiation heat transfer. Based on the assumptions, the tube flow is governed by the continuity, the Navier-Stokes equations and the energy equation. In the Cartesian tensor system these equations can be written as follows:

Continuity equation:

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

Momentum equation:

$$\frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] \tag{2}$$

Energy equation:

$$\frac{\partial(\rho u_i T)}{\partial x_i} = \frac{\partial}{\partial x_j}\left(\Gamma \frac{\partial T}{\partial x_j}\right) \tag{3}$$

where Γ is the thermal diffusivity and is given by

$$\Gamma = \frac{\mu}{\Pr} \tag{4}$$

Apart from the energy equation discretized by the QUICK scheme, the governing equations were discretized by the power law differencing scheme, decoupling with the SIMPLE algorithm and solved using a finite volume approach [10]. The solutions were considered to be converged when the normalized residual values were less than 10^{-5} for all variables but less than 10^{-9} only for the energy equation.Four parameters of interest in the present work are the Reynolds number (Re), friction factor (f), Nusselt number (Nu) and thermal enhancement factor (TEF). The Reynolds number is defined in Eqn. 5 as follows

$$\mathrm{Re} = \rho \bar{u} D / \mu \tag{5}$$

The friction factor, f is computed by pressure drop, Δp across the length of the periodic tube, L shown in Eqn. 6

$$f = \frac{(\Delta p / L) D}{\frac{1}{2}\rho \bar{u}^2} \tag{6}$$

The heat transfer is measured by local Nusselt number which can be written Eqn. 7 as follows

$$Nu_x = \frac{h_x D}{k} \tag{7}$$

The average Nusselt number can be obtained by Eqn. 8.

$$Nu = \frac{1}{A}\int Nu_x \partial A \tag{8}$$

The thermal enhancement factor (TEF) is defined as the ratio of the heat transfer coefficient of an augmented surface, h to that of a smooth surface, h_0, at an equal pumping power and given by Eqn. 9.

$$TEF = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{Nu}{Nu_0}\right|_{pp} = \left(Nu/Nu_0\right)/\left(f/f_0\right)^{1/3} \tag{9}$$

where Nu_0 and f_0 stand for Nusselt number and friction factor for the smooth square channel, respectively.The variation in Nu and f values for the 20°and 30° inclined diagonally ribs at BR = 0.10-0.20 and Re=500 is less than 0.2% when increasing the number of cells from

64,000 to 164,000, hence there is no such advantage in increasing the number of cells beyond this value and thus, the grid system of 128000 cells was adopted for the current computation.

Flow description

Inclined diagonally ribs geometry and arrangement

The system of interest is a square duct with a 20° discrete inclined diagonally ribs inserted into the square duct in tandem as shown in Fig 1. The flow under consideration was expected to attain a periodic flow condition in which the velocity field repeated itself from one cell to another. The concept of periodically fully developed flow and its solution procedure have been described in Ref. [1]. The air enters the channel at an inlet temperature, $\boldsymbol{T}_{in}$, and flows over a 20° discrete inclined diagonally ribs where b is the ribs height, $D = H$ set to 0.05 m, is the channel height and b/H is known as the blockage ratio, BR. The axial pitch, L or distance between the inclined diagonally ribs cell is set to $L = H$ and $2H$ in which L/H is defined as the pitch spacing ratio, $PR = 1$ and the width of the channel, W is equal to H for the square duct. To investigate an effect of the interaction between ribs, the blockage ratio, BR is varied in a range of $BR = 0.10$–0.20 for $\alpha = 20°$ in the present investigation.

Boundary conditions

Periodic boundaries are used for the inlet and outlet of the flow domain. Constant mass flow rate of air with 300K ($Pr = 0.7$) was assumed in the flow direction rather than constant pressure drop due to periodic flow conditions. The inlet and outlet profiles for the velocities must be identical. The physical properties of the air have been assumed to remain constant at average bulk temperature. Impermeable boundary and no-slip wall conditions have been implemented over the channel walls as well as the ribs. The constant temperature of all the channel walls was maintained at 310K while the rib plate was assumed at adiabatic wall conditions.

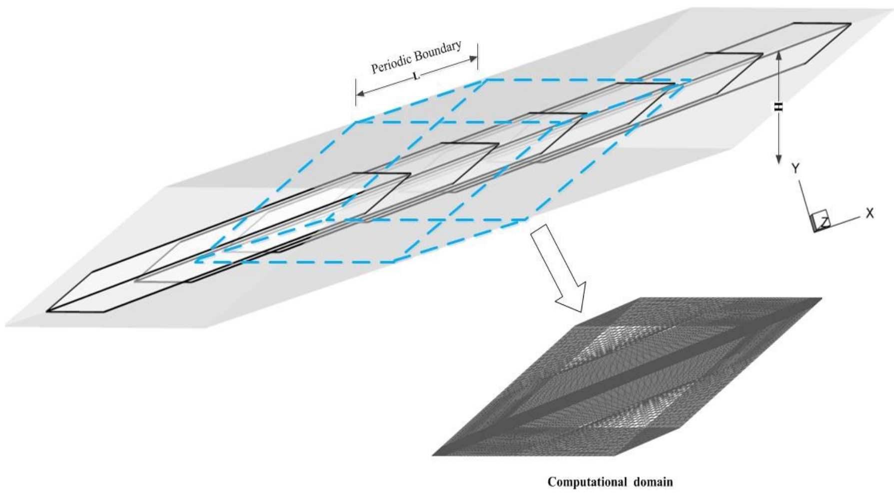


Figure 1 Channel geometry and computational domain of periodic flow

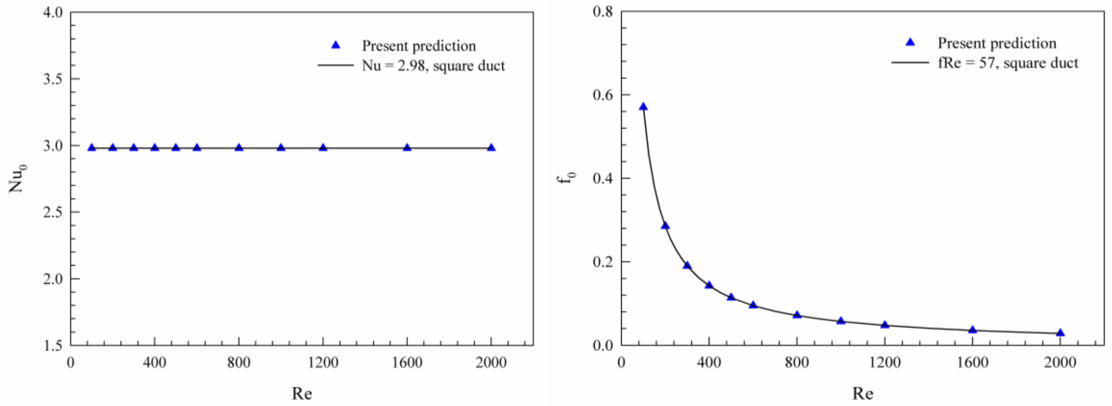


Figure 2 Verification of (a) Nusselt number and (b) friction factor for smooth channel

Verification of smooth tube

Verification of the heat transfer and friction factor of the smooth square duct without ribs was performed by comparing with the previous values under a similar operating condition as shown in Fig 2*a* and *b*, respectively. The current numerical smooth duct result was found to be in excellent agreement with exact solution values for both the Nusselt number and the friction factor, less than ±0.25% deviation. The exact solutions of the Nu and f for laminar flows over the smooth square duct with constant wall temperature are, respectively, $Nu_0 = 2.98$ and $f_0 = 57/\mathrm{Re}$ [9].

Results and discussion

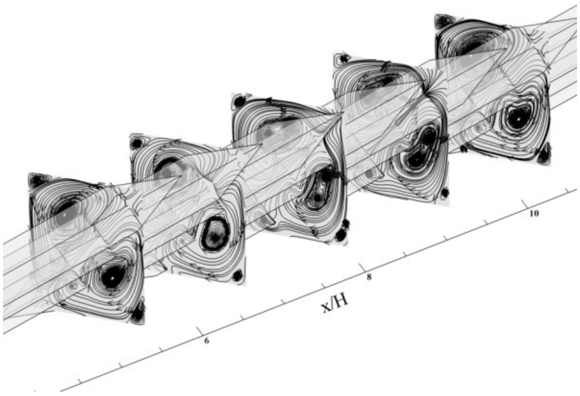
Flow structure

The flow structure in the duct inserted diagonally with a tape mounted periodically with double-sided angled ribs can be visualized by the streamline plots as depicted in Fig. 3a to b. Here the streamlines of the rib modules are presented for Re=1000 at BR=0.2 and PR=1. Fig. 3a and b shows the streamlines in transverse planes along the duct with the 20°and 30˚angle, In Fig.3a and b, it is visible that two counter rotating vortices or longitudinal vortex flows created by the angled ribs appear on both the upper and lower triangular parts of the duct, as expected and the vortex center is moving diagonally in the inter rib cavity from the rib leading end to the next downstream rib trailing end. The appearance of the longitudinal vortex flows can help to increase considerably the heat transfer rate in the duct because of stronger fluid filament mixing. It is concluded that the presence of the ribbed tape creates two counter-rotating vortices (Fig.3) leading to longer flow path, high vortex strength and impingement flows.

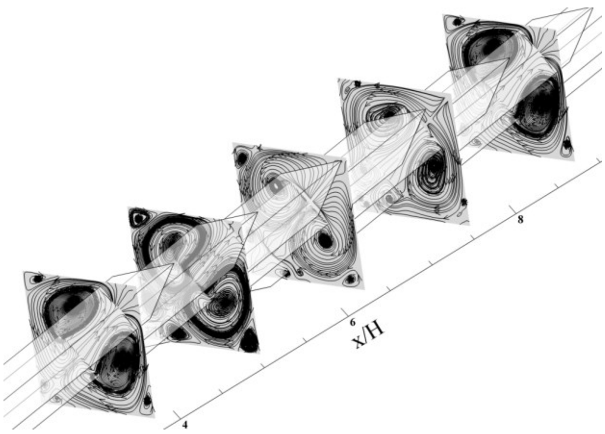
Heat transfer and friction loss

Fig. 4a and bdisplays the contour plot of temperature in transverse planes for the ribbed tape with PR=1 and BR=0.2 at Re=1000. It was found that there was a major change in the temperature field over the duct cross section. The core and the near-wall temperatures showed nearly the same values (blue region). This meant that the vortex flow provided a significant influence on the temperature field, because it can induce stronger fluid flow mixing between the wall and the core regions, leading to a high temperature gradient over the heating duct wall.Fig. 5a and b exhibit local Nu_x contours of the tube wall for (a) $\alpha = 20^o$ and

(b) α = 30°of the incline diagonally ribs with PR = 1.00 at Re = 1000, BR = 0.20. It is visible in the figure that a larger area of high Nu_x values can be observed for this case.The variation of the average Nu/Nu_0 ratio with Re for the incline diagonally ribs with various BRs and incline diagonally ribs angle is presented in Fig. 6a.In the figure, it is visible that the Nu/Nu_0 value tends to increase with the rise of Re for all BR values.The use of higher BR value leads to the increase in Nu/Nu_0 while α = 30° provides the higher of Nu/Nu_0 in comparison with α = 20° for all BRs. The angled incline diagonally ribs with BR = 0.20 provides the highest Nu/Nu_0 value.The maximum Nu/Nu_0 values are found to be about 5.62 and 6.10 for α = 20° and 30°, respectively. A closer examination reveals that the use of the incline diagonally ribs with BR ranges studied yields average heat transfer rate of about 1.00-5.60 times over the square duct with no incline diagonally ribs, depending on the BR, Re and α values. Fig. 6*b* displays the variation of the friction factor ratio, f/f_0 with *Re* for various *BR* and *α* values. In the figure, it is noted that the f/f_0 tends to increase with the rise of *Re*, *BR* and *α* values.The incline diagonally ribs with *BR* = 0.20 and *α* = 20° gives the highest f/f_0 value.The friction factor for the angled incline diagonally ribs appears to be about 1.00-15.00 times above that for the smooth tube with no incline diagonally ribs.Thus the flow blockage due to the presence of the incline diagonally a rib is a vital factor to cause a high pressure dropin the tube



a) *α* =20˚

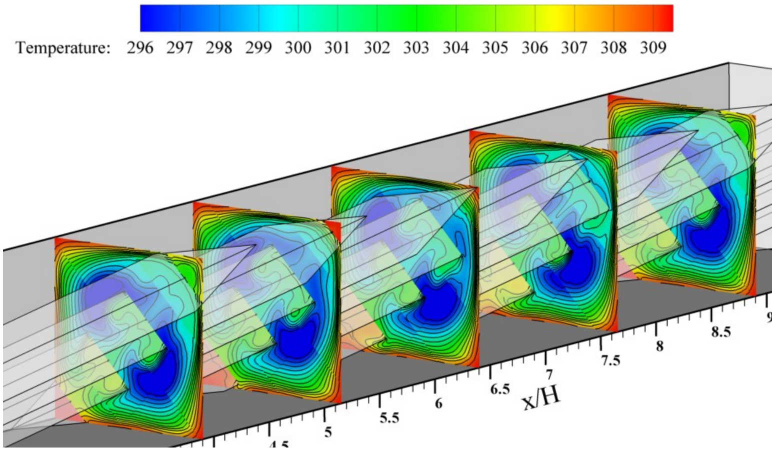


b) *α* =30˚

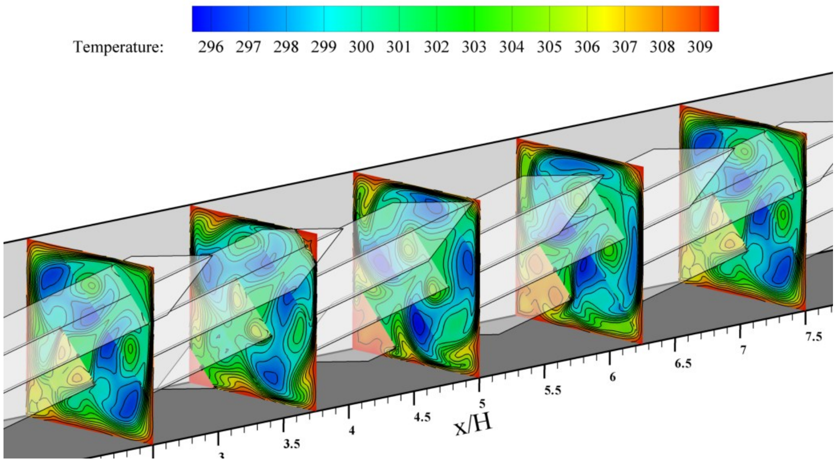
Figure 3 Streamlines in transverse planes for 20˚and30° at *BR* = 0.20, *Re* = 1000 and *PR*= 1

Performance evaluation

Fig. 7 shows the variation of thermal enhancement factor (TEF) for air flowing in the incline diagonally ribs tube. In the figure, the enhancement factor of using the incline diagonally ribs tends to increase with the rise in Re values. It is found that the incline diagonally ribs with BR = 0.20 and α = 20˚ provides the highest enhancement factor of 2.24The enhancement factor of the incline diagonally ribs is seen to vary between 1 and 2.24, depending on the BR, α and Re values.



a)α =20˚



b)α =30˚

Figure 4 Temperature contour in transverse planes for 20˚and30° at *BR*=0.2, *Re*=1000 and *PR*=1

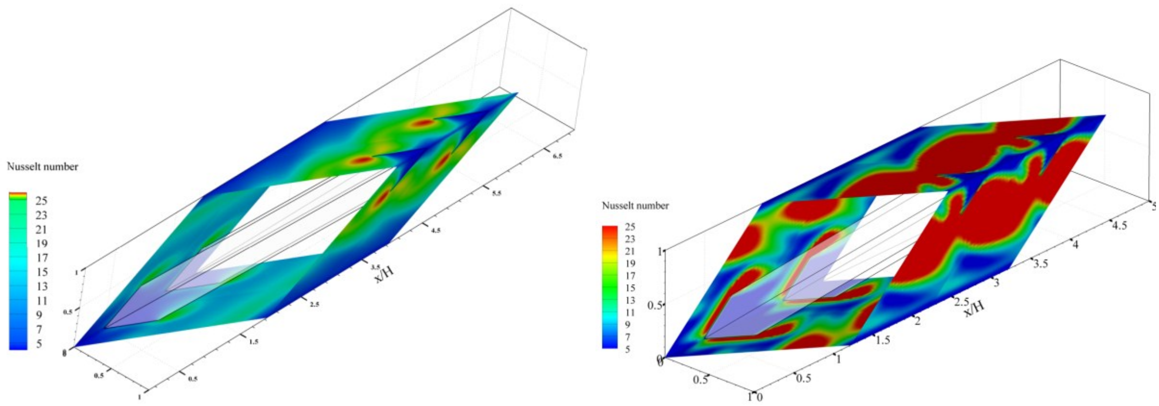


Figure 5 Nu_x contours (a) α = 20˚and (b) α = 30˚ for incline diagonally ribs, at *BR* = 0.20, *Re* = 1000 and *PR* = 1.00.

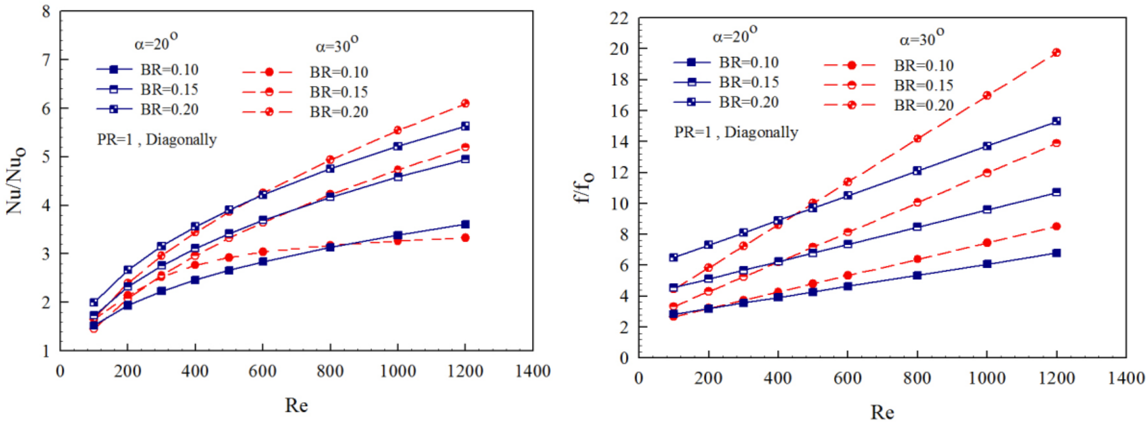


Figure 6 (a) Nu/Nu0 and (b) f/f0 versus Reynolds number at various incline diagonally ribs BRs and α.

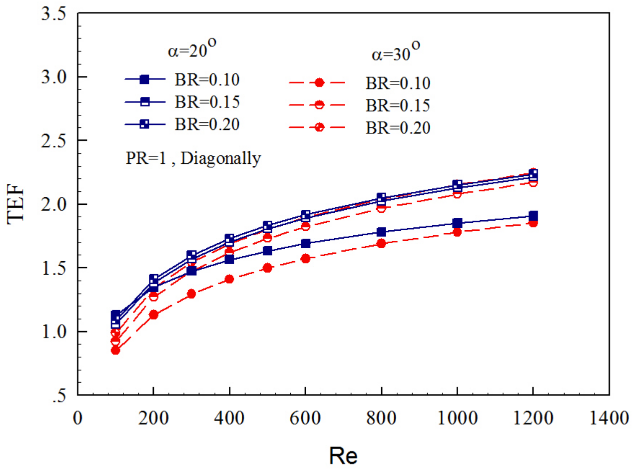


Figure 7 Thermal enhancement factor at various incline diagonally ribs BRs and α.

Conclusions

Numerical computations of laminar periodic flow and heat transfer characteristics in a square duct fitted with in-line angled ribs in tandem on the tube wall are performed. The main vortex flow created by the incline diagonally ribs exists and helps to induce impingement flows over the tube wall leading to drastic increase in heat transfer in the square duct. The order of heat transfer enhancement is about 1.00-5.60 times for using the incline diagonally ribs with BR = 0.1-0.20 at PR = 1.00 and Re vary from 100 to 1200. The enlarged pressure loss is in a range of 1 to 15.00 times above the smooth square duct. Thermal enhancement factor for the incline diagonally ribs is higher than unity and its maximum value is about 2.24 at BR = 0.20, α = 20°and Re = 1200, indicating higher performance over the smooth square duct.

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