A Numerical Investigation Based on Heat Transfer and Fluid Flow Characteristics of Air in a Circular Tube Heat Exchanger with Inclined Ribs

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1 Introduction

Saving energy has been the most important engineering concern in the last decade. This can be attributed to the exponential increase in population all over the world and the ever-increasing demand of energy. Innovative design solutions and applications to effectively utilize energy have been used in the industry, with the last few decades having seen the wide use of ribbed wall surfaces. Several industrial processes and applications face thermal energy exchange in fluids with laminar flow. In the case of a laminar flow, there is major thermal resistance in the bulk flow. There also lies a dominant thermal resistance within the thin boundary layer adjacent to the flow. The roughness in the form of continuous or gapped ribs could generate turbulence and disintegrate the viscous sub layer near the heat surface, thereby reducing the thermal resistance. The added advantage of increased heat transfer area is achieved by the number of ribs mounted on the wall (Peng et al. 2011).

In an effort to find viable alternative energy sources people have turned towards the Sun. However a solar air heater's thermal performance is not good because of the poor heat transfer rate observed between the working fluid and absorber plate (Duffie and Beckman 1980). Bhattacharyya et al. (2014) studied low thermo-fluids properties of the fluid in part, and to the viscous sub-layer that appears in the vicinity of the absorber and which is resistant to the heat transfer in the other part. Bhattacharyya et al. (2015) presented the idea that in order to make solar air heaters efficient their thermal performance needs to be improved.

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Many techniques based on both active and passive methods have been proposed to enhance heat transfer in these applications (Boulemtafes 2010). One can find numerous studies on solar air heat enhancement techniques. For example, Varun et al. 2007) reviewed the geometry of roughness used in heat exchangers and reported the optimum geometry of roughness that is adapted in case of solar air collectors. (Hans et al. 2009) studied the basic geometry of roughness element employed by various researchers to enhance the thermal efficiency of solar air heaters. In view of the search for optimal roughness pattern, few decent roughness geometries have been compared on the basis of thermo-hydraulic performance. (Bhushan et al. 2010) presented their attempt to classify and examine the geometry of artificial roughness used in the ducts of solar air collectors. (Chandra et al. 2003) explained the thermal characteristics in a square channel with continuous ribs on one, two, three, and four walls and found that the heat transfer and pressure drop increase with the rise in the number of ribbed walls. Promvonge et al. (2008) represented an approximate idea of the effect of various rib shapes and their thermal behaviours in a tube and showed that the staggered triangular rib performs the best. Currently work on nano-fluids is also becoming popular for enhancing heat transfer. Smithberg and Landis (1964), (Lopina and Bergles 1969), (Manglik and Bergles 1993), (Sarma et al. 2002, 2003), and (Bhattacharyya and Chattopadhyay 2015) have reported heat transfer enhancements with swirl generator and twisted tape inserts of single-phase fluids in a tube.

However, all the studies mentioned above, restricted themselves to just one flow regime. This study has tried to bridge the gap and represent the findings of heat transfer enhancement in all flow regimes—laminar, transition, and turbulent. Furthermore, the arrangement of inclined ribs inside a circular duct could provide better mixing and enhanced heat transfer from the wall that can be easily incorporated within a solar heater. The aim of this work is to put forth the findings of heat transfer and thermal performance of the inclined ribs attachment as shown below, at various attack angles (θ) of 15°, 30°, 45° and 60° studied under a very wide Reynolds number regime (200–20,000).

2 Mathematical Model

In the present investigation, circular tubes with four different attack angles were considered, i.e. 15°, 30°, 45°, and 60°. The geometrical configuration of the circular tube with attack angle discussed in the present work is shown in Fig. 1.

The length of the tube is considered as 20 times the diameter of the tube. It is assumed to be a part of a longer pipe section—one that is typically used in industrial applications. Therefore, a fluid with a fully developed velocity profile is assumed to enter through the inlet at a fixed temperature, and flows through the circular tube fitted with ribs with different attack angles (θ), while a pressure outlet condition was applied at the exit. Air inlet temperature with 300 K was assumed in the flow direction. The wall temperature is 500 K and is constant throughout the experiment. Initially the physical properties of the air at the inlet temperature were

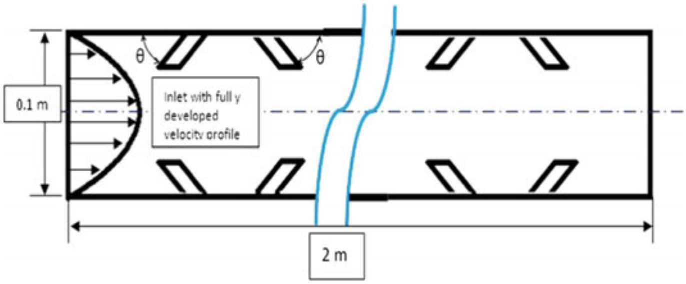

Fig. 1 Circular tube geometry with artificial ribs

used to calculate the mean bulk temperature for each case; after which the correct physical properties at the mean bulk temperature for each individual case was incorporated and all cases were reiterated. This was done until very little variation (less than 1.0 %), i.e. convergence, in the result values was achieved.

The mathematical modelling for this simulation was based on the following assumptions:

- Steady axis-symmetric fluid flow and heat transfer.
- The flow is incompressible.
- Heat transfer by radiation, body forces, and viscous dissipation has been neglected.

The three dimensional governing equations of continuity, momentum, and energy equations are solved using Transient SST model modified and proposed by (Abraham et al. 2009). Ansys Fluent 15 is used to solve the following governing equations.

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

$$\rho\left(u_i \frac{\partial u_j}{\partial x_i}\right) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i}\left((\mu + \mu_{turb})\frac{\partial u_i}{\partial x_i}\right), \quad j = 1, 2, 3 \tag{2}$$

$$\left(u_i \frac{\partial \Theta}{\partial x_i}\right) = \frac{\partial}{\partial x_i}\left(\left(\alpha + \frac{v_{turb}}{\mathrm{Pr}_{turb}}\right)\frac{\partial \Theta}{\partial x_i}\right) \tag{3}$$

The values of turbulent Prandtl number ($\mathrm{Pr_{turb}}$) are taken from Abraham et al. (2009).

$$\frac{\partial(\rho u_i \kappa)}{\partial x_i} = \gamma \cdot P_\kappa - \beta_1 \rho \kappa \omega + \frac{\partial}{\partial x_i}\left(\left(\mu + \frac{\mu_{turb}}{\sigma_\kappa}\right)\frac{\partial \kappa}{\partial x_i}\right) \tag{4}$$

$$\frac{\partial(\rho u_i \omega)}{\partial x_i} = A\rho S^2 - \beta_2 \rho \omega^2 + \frac{\partial}{\partial x_i}\left(\left(\mu + \frac{\mu_{turb}}{\sigma_\omega}\right)\frac{\partial \omega}{\partial x_i}\right) + 2(1 - F_1)\rho \frac{1}{\sigma_{\omega 2}\omega}\frac{\partial \kappa}{\partial x_i}\frac{\partial \omega}{\partial x_i} \tag{5}$$

3 Meshing Arrangement

Computations were carried out for θ = 15°, 30°, 45°, and 60°. The diameter of the tube was 100 mm. The height of each rib was 30 mm and the space between each rib was 100 mm. Each rib was 5 mm thick. For reducing calculations we have assumed only the top half and run the model in the axis-symmetric setting. The work has been validated with the computational work of (Tanda 2011) on angular ribs. A preliminary mesh independence study is done. Grid containing 125,807 numbers of elements was sufficient to arrive at results of good resolution with only 1 % change in result parameters if the grid as further increased.

4 Results and Discussion

All the results obtained from computation for plain tube, heat transfer and pressure penalty characteristics are verified in terms of Nusselt number and friction factor. The Nusselt number and friction factor obtained from the present numerical computation compared with those from the proposed correlations by (Ozisik 1985) and proposed correlation by (Ozisik 1985) respectively.

The data obtained from the computational analysis for the plain duct are in good agreement with the predicted results from the proposed correlations with data range of +3.02 % to +4.12 % and +0.8 % to +1.6 % for the Nusselt number and friction factor, respectively as shown in Figs. 2 and 3.

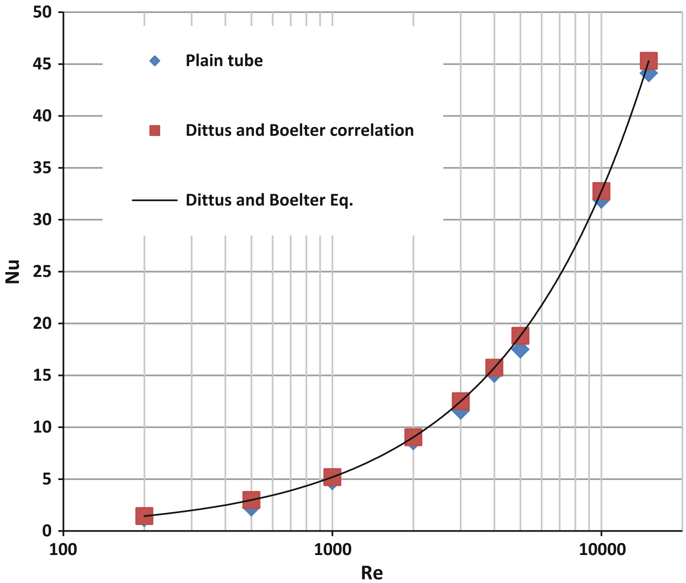

Fig. 2 Correlation of Nu_0 for plain tube with Dittus-Boelter equation

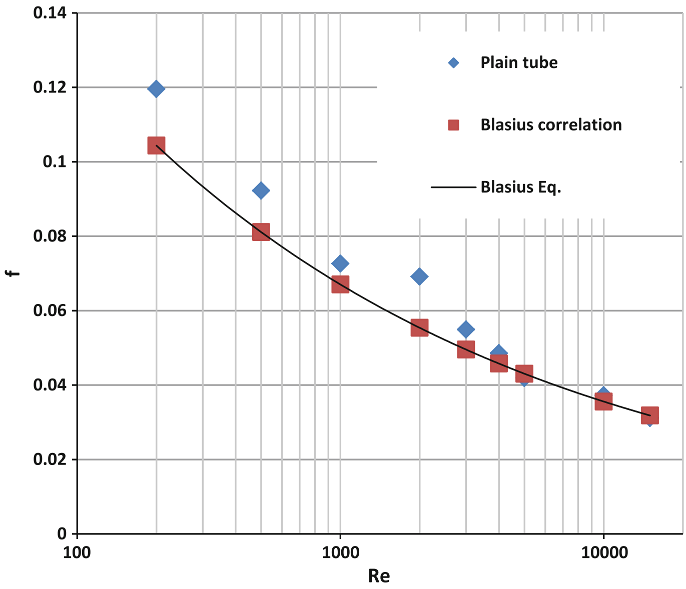

Fig. 3 Correlation of f_0 for plain tube with Blasius equation

It can be observed in Fig. 4 that the Nusselt number increases with the increase of Reynolds number. From Fig. 4, one can notice that the attack angle $\theta = 60°$ yields higher heat transfer than the other attack angles. The rate of heat transfer increases with the increase of attack angles. The better performance of the inserted artificial inclined ribbed tube can be attributed to the swirl flow created by the artificial ribs, leading to more effectiveness in disruption of the boundary layer compared to those caused by the individual device. Friction factor results are presented in Fig. 5. It is seen from Fig. 5 that as the Reynolds number is increased, with the decrease of friction factor. The friction factor values for all the artificial inclined ribs have higher value than the base case. Friction factors are higher at lower Reynolds number and then decreases slowly at higher Reynolds numbers. This is because the attributed to the use of inclined ribs with a higher attack angle which led to a higher viscous loss near the duct wall regions caused by a stronger swirl flow or turbulence flow and long residence time in the tube.

From Fig. 6 it can be observed that Nu/Nu_0 shows a slightly opposite trend for all cases as Nusselt number shows—it shows an increase in the transition region with Reynolds number ranging between 2000 and 4000 and then decreases at the downstream end of the graph. This enhancement is due to a secondary flow induced by the artificial inclined ribs, which promotes radial mixing of bulk flow with

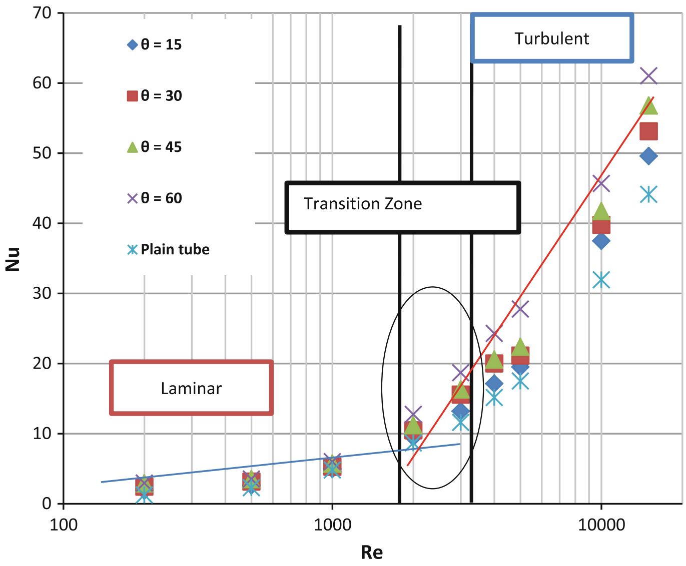

Fig. 4 Variation of Nusselt number with Reynolds number for different angle of attacks

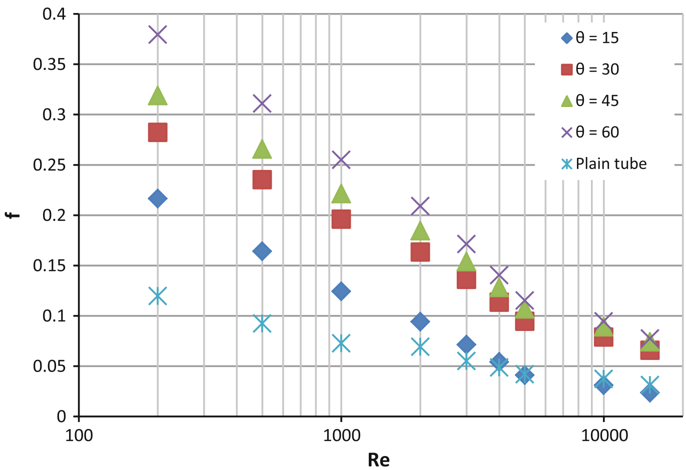

Fig. 5 Variation of friction factor with Reynolds number for different angle of attacks

near-wall flow. At similar operating condition, the circular tube with artificial ribs of different attack angle yields higher heat transfer rate than the plain tube acting alone.

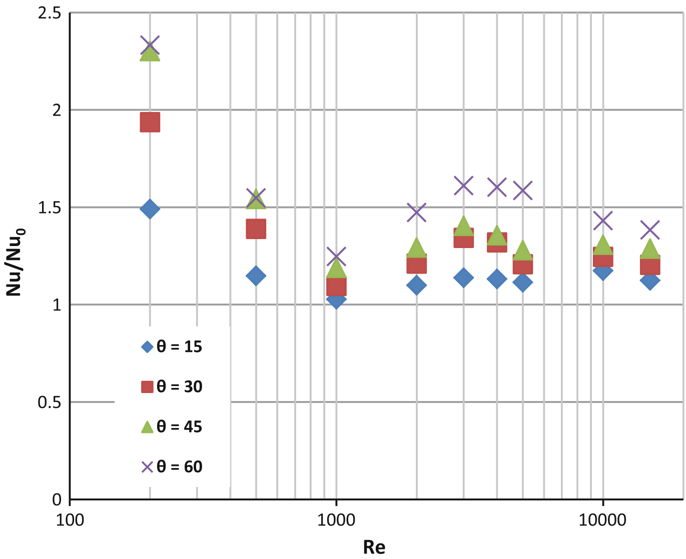

Fig. 6 Variation of Nu/Nu_0 with Reynolds number for different angle of attacks

Figure 7 shows the turbulent intensity contour for different attack angles of the ribbed tube for (Re = 3000). The vortices are formed in the section due to the flow separation, recirculation and rib-induced flow. From Fig. 4 one can see that a number of vortices are formed when the attack angle is highest (θ = 60°). The vector distributions around the artificial roughness are for observing more closely the main recirculation zones that appear when the flow is separated. When the fluid turbulent intensity is about 9–10 % more than the normal inlet fluid velocity, the second recirculation zone generates. Near the wall, there is a recirculation zone which is a reattachment area.

Overall performance evaluation is important for comparing the performances of different rib configurations. Figure 8 shows the thermal enhancement factor (PEC) with variations of rib attack angles.

This parameter is called the thermal performance factor, which means that the comparison is made based on constant pumping power. The computational analysis reveals that the PEC decreases at laminar region and increase with an increase in Reynolds number at turbulent region. This shows the significant role of the ribs at different attack angles in increasing the turbulence intensity at higher velocities. Also, the values of PEC vary between 0.8 and 1.75 for all cases. From Fig. 8, it is evident that the attack angle θ = 15° and θ = 60° shows the most-promising results in the transition zone and turbulent region, while θ = 30° has the lowest performance.

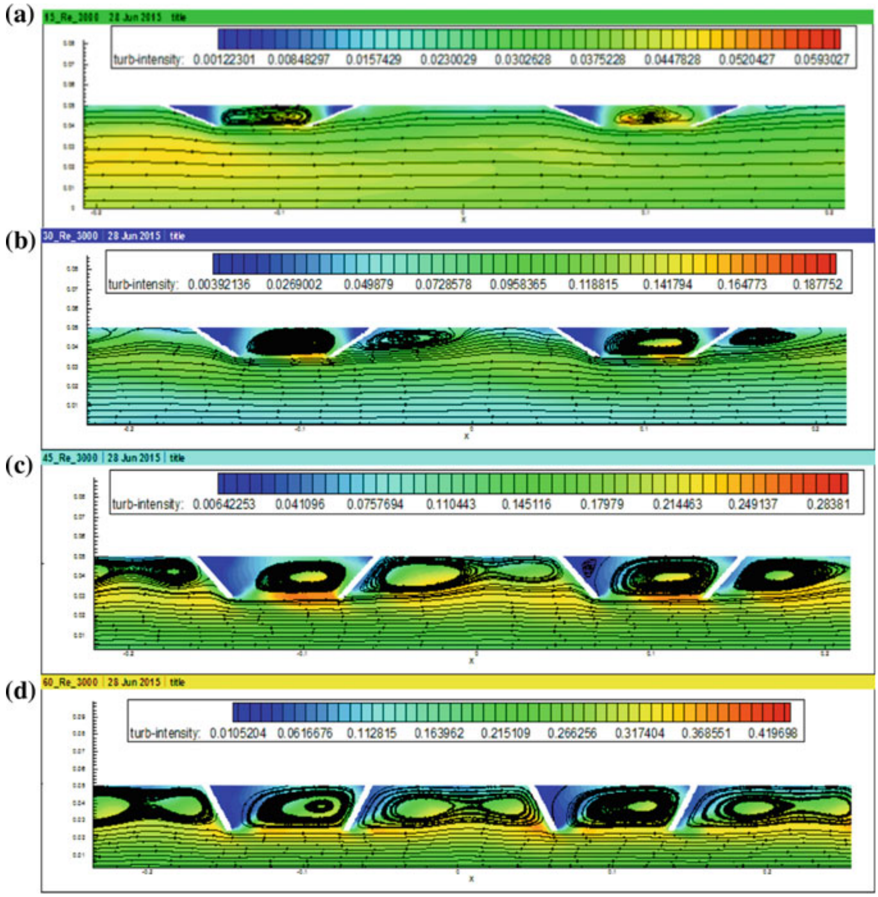

Fig. 7 Turbulent Intensity contours of u-components for different attack angles

5 Conclusion

In the present study, the thermal performance factor of different artificial attack angles has been compared with plain smooth tube. The studies of artificial inclined angles of 15°, 30°, 45°, and 60°, were placed in a circular tube of the solar air heater/heat exchanger. Effects of the artificial attack angles (θ) on local convective heat transfer, friction factor, and thermal performance characteristics are also reported. Smooth, plain tube consistently provides poorer heat transfer than tube with ribs inserts over the range studied. The attack angle of θ = 15° and θ = 60° yields better thermal performance than the other two attack angles (θ = 30° and 45°)

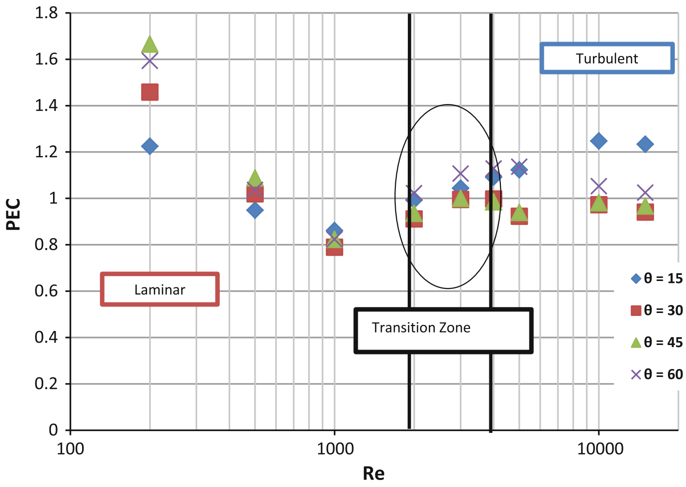

Fig. 8 Comparison of thermal enhancement factor for different rib attack angles

in varied Reynolds number range. In addition to those good performances, the artificial inclined ribs are also easy to fabricate. Thus, it is a promising work which can be widely used in heat transfer enhancement of turbulent flow.

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