Computational Study of Convective Heat Transfer in Circular Pipe using Trapezoidal Ribs

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Abstract. Perturbations or interruptions provided in the passage of heat exchanger generate the vortices downstream. The formation of these natural vortices, augment local heat transfer abruptly. The effect on convective heat transfer enhancement and friction characteristics by providing trapezoidal ribs inside a circular pipe is computationally investigated in detail. Different variations of height, width and pitch of the ribs are used to optimize the rate of heat transfer through the pipe. Liquid water is employed as the working fluid. Input parameters of Reynolds Number ranging from 5000-60000 with axial flow along the pipe and constant heat flux of 50 W/cm² to the pipe surface is used. After validation with the existing literature, Realizable k-ε turbulent model with enhanced wall function is used in commercial CFD software ANSYS FLUENT. The outcome of the investigation shows that the ribs provided on the inside of the pipe surface enhance the turbulence in the flow and produce recirculation which disturb the thermal boundary layer behind the ribs and thus help in enhancing the rate of heat transfer through the pipe.

1. Introduction

Need of heat transfer enhancement in process industries, chemical industries, evaporators, power plants condensers, air-conditioners and refrigerators, radiators, electrical panels, etc. has always drawn the attention of the researchers. Augmentation of heat transfer through a heat exchanger can broadly be done by three different methods, viz. active, passive and compound method. In active method some external power input is given to enhance the heat transfer rate. In passive method no external power input is required to create turbulence or swirling of the fluid. Instead inserts are provided in the flow tubes or modification in the geometry of the surface of the fins or flow tubes is done to enhance the rate of heat transfer, but this brings pressure loss in the flow field as a drawback [1].

Mohammad et al. [2] in their computational study of transversely corrugated circular tubes with different geometrical parameters observed that Nusselt number (Nu) increases with increasing Reynolds number (Re) and decreasing pitch of the corrugation. On the other hand the, friction factor do not change with alteration in Reynolds number in corrugated tubes but friction coefficient reduces with increase in Reynolds number. A computational comparative study between Reynolds stress transport (RST) and direct numerical simulation (DNS) was put forwarded by Han et al. [3] on an outward convex corrugated steel tube with helium as the fluid. They proposed that RST model matches closely with DNS model numerical results and hence is more suitable than any other Reynolds-Averaged Navier Stokes model for the study of flow and heat transfer characteristics of corrugated tubes. Kathait and Patil [4] experimentally studied the thermo hydraulic performance of a rectangular discrete ribbed copper tube with water as the working fluid. The findings show that the
values of Nusselt number and friction factor increase by increasing the pitch to rib height ratio. The maximum enhanced value of 2.73 for Nusselt number at pitch to rib height ratio of 10 was reported. Liu et al. [5] computationally investigated the heat transfer and flow characteristics of shell side flow of a rod baffle heat exchanger with single, double, triple and quadruple start spirally corrugated tube with water as the fluid. The Nusselt number enhanced to 1.2 times for single start spiral tube at 18000 Reynolds number. The successive increase in number of start of spiral corrugation increased the corresponding heat transfer slightly. A smooth two start spirally corrugated tube with water as the flowing fluid, was analyzed computationally at low Reynolds number from 100-1300 by Kareem et al. [6]. The effect of ratio of corrugation height to tube diameter and corrugation pitch to tube diameter were studied. At Reynolds number greater than 700, increase in friction factor was found to be much greater than the increase in heat transfer. An augmentation in heat transfer of 19%-71% was reported with a particular severity of corrugation.

The literature review presented here suggests that the emphasis is given to the flow with low Reynolds number. The present study employs Reynolds number in the range of 5000-60000 and water as the fluid. Trapezoidal ribs protruded on the inner surface of the pipe are used as the turbulators to enhance the turbulence in the flow. The reverse flow developed by ribs can enhance the heat transfer coefficient and the momentum transfer and has a consequence in the form of higher pressure drop along the flow. The enhancement relies on the strength of the recirculation and the prolongation of the reattachment.

2. Methodology

2.1. Geometrical Model Description

The analysis is performed in 2-D, considering the flow to be axisymmetric about the axis of the pipe. The trapezoidal ribs were protruded circumferentially along the length of the pipe at different pitches for different cases. Figure 1(a) shows the geometry of rib and figure 1(b) shows the schematic of the flow with different boundaries. The diameter of the pipe is 10 mm; entrance length is taken as 200 mm considering the fully developed flow at the test section. Length of the test section is 100 mm while the exit section length is 50 mm.

![Figure 1. Trapezoidal rib along the fluid flow.](image)

2.2. Discretization

A structured grid was adopted for the analysis. As wall gradient has a significant role in heat transfer problems a finer mesh is adopted by proper biasing factor near walls to capture the properties of flow in viscous sub layer. The nearest wall distance governed by $y^+$ approach was considered which lies below 5. A grid independent test was performed to verify the results. Grid test was performed with width of the ribs 0.5mm, height of the ribs 0.25mm and pitch of the ribs 5mm. Grid sizes employed was 64929, 172959 and 458976. Pressure and Velocity distribution was considered for grid independency test. Grid with 458976 numbers of elements was adopted for the simulation after the grid independent test as the $y^+$ value lied below 1 for this.

2.3. Boundary Conditions and Material Selection
Inlet was given selected as velocity inlet with different velocities to account for the desired Reynolds number. For turbulent specification hydraulic diameter of 10mm and the turbulent intensity of 3% were selected. At outlet zero gauge pressure outlet was selected. The upper edge was selected as wall with no slip condition and the test section length was given a constant heat flux of 50 W/cm$^2$. With centerline as axis the axisymmetric solver was selected. Properties of water are shown in table 1.

| Properties                  | Specifications |
|-----------------------------|----------------|
| Density (kg/m$^3$)          | 998.2          |
| Specific Heat (J/kg·K)      | 4182           |
| Thermal Conductivity (W/m·K)| 0.6            |
| Viscosity (kg/m·s)          | 0.001003       |

2.4. Solver Settings
For pressure velocity coupling SIMPLE scheme was used. Pressure equation has been discretized using Standard. Momentum, Turbulent Dissipation Rate, Turbulent Kinetic Energy and Energy equations has been discretized using second order upwind scheme.

2.5. Governing Equations
Following are some important equations for the fluid flow in a pipe with heat exchange in focus [7].

Reynolds number $Re = \frac{\rho V_{in} D_h}{\mu}$,

Nusselt number $Nu = \frac{h D_h}{\frac{k}{d_p}}$,$k_{min}$

Friction factor $f = \frac{\sqrt{2 \Delta p}}{\rho V_{in}}$

Newton’s law of cooling: $q_{conv} = h (T_s - T_c)$

Continuity equation in two dimensions: \( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \)

Navier Stokes Momentum equation:

X-momentum: $u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + \frac{1}{\rho} \frac{\partial p}{\partial x} = \partial^2 u_{x} + \partial^2 u_{y}$

Y-momentum: $u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \frac{1}{\rho} \frac{\partial p}{\partial y} = \partial^2 u_{x} + \partial^2 u_{y}$

Transport equation for $k$: $\frac{\partial}{\partial x} \left[ \rho u (\mu + \mu \frac{\partial k}{\partial x}) \right] = \partial (x_j + \mu \frac{\partial k}{\partial x}) + G_k - \rho \varepsilon$

Transport equation for $\varepsilon$: $\frac{\partial (\rho \mu)}{\partial x_j} = \partial (x_j + \mu \frac{\partial \varepsilon}{\partial x_j}) + \rho C_1 \varepsilon - \rho C_2 \frac{\varepsilon^2}{k + \sqrt{4 \varepsilon}}$

Energy Equation $\frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \partial (\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2})$

2.6. Validation
Some of the cases of variation in dimensions of the ribs as discussed by Mohammed et al. [6] were adopted for comparing their average Nusselt number with the present simulated results. The percentage error obtained in the different cases gives the clarification of the validated results with the Mohammed et al. [6]. For validation rib with width 0.5mm, pitch 5mm and height 0.25mm similar to the Mohammed et al. [6], was generated. Figure 2 shows a minor deviation in results of Average Nusselt number to Reynolds number reported from the referenced work, owing to the difference in the grid size, $y^*$ value and the turbulence model selected.
Figure 2. Validation of the CFD code.

3. Results and Discussion

Table 2 shows the different combinations that have been used to carry out the parametric investigation. Trapezoidal ribs were tested for five variations.

Table 2. Parametric description of the profiles (Dimensions in mm).

| Case | Width | Height | Pitch |
|------|-------|--------|-------|
| A    | 0.5   | 0.25   | 5     |
| B    | 0.5   | 0.25   | 10    |
| C    | 0.5   | 0.25   | 15    |
| D    | 0.5   | 0.75   | 15    |
| E    | 0.5   | 1      | 15    |

Figure 3 and 4 show the effects on average Nusselt number and the friction factor against variation in Reynolds number by varying the pitch of the ribs (5, 10, 15 mm), keeping width (0.5 mm) and height (0.25 mm) constant. It can be seen at higher Reynolds number the incoming velocity increases which disturbs the flow resulting in better heat transfer. With an increase in the periodicity of the ribs along the flow in pipe, the thermal boundary layer tends to get thinner and thinner as compared to flow in a plain pipe. This thinning of thermal boundary layer helps greatly in enhancing the rate of heat transfer. It is evident from figure 3 that Nusselt number increases in Nusselt number with a decrease in the pitch of the ribs and increasing the Reynolds number. On the contrary, friction factor reduced with the rise in Reynolds number for all the cases. By decreasing the pitch of the ribs the roughness of the flow field increased and consequently pressure drop and friction factor increased quite predictably.

Figure 3. Variation of $\text{Nu}$ with $\text{Re}$.

Figure 4. Variation of $f$ with $\text{Re}$.

Figure 5 and 6 show the effect on average Nusselt number and friction factor at different Reynolds number by varying rib height (0.25, 0.75, 1 mm) keeping width (0.5 mm) and pitch (15 mm) constant.
It is evident that with an increase in Reynolds number the average Nusselt number also increased. Similar trend for Nusselt number was seen by increasing the height as were reported by decreasing the pitch of the ribs. Furthermore, increasing the height of the ribs disturbs the whole flow field which causes enhanced heat transfer at the cost of enhanced drop in pressure. As shown in figure 6, the friction factor is increasing with an increase in the height of the ribs. Case E being the most prominent case as the height of the rib is highest in it.

Figure 5. Variation of Nu with Re.  
Figure 6. Variation of f with Re.  
Figure 7. Streamlines plot behind ribs.

With the change in Reynolds number, change in friction factor is seen to be little compared to variation in Nusselt number. Figure 6 shows the height of the ribs is a prominent factor considering the magnitude of the friction factor. Figure 7 illustrates the streamlines of the flow field. With an increase in height of the ribs more pronounced recirculation is taking place behind the ribs as shown in figure 7 (d) and 7 (e). Figure 8 shows the axial velocity distribution contours which demonstrated the presence
of negative axial velocity which marked the presence of recirculation zone behind the ribs in the flow. Consequently, it can be clearly observed that the ribs having larger height shows bigger wake region behind ribs.

![Figure 8](image)

**Figure 8.** Axial velocity distribution.

The overall observation of the variations of pitch and height in the geometry of the ribs confirm that the change in pitch gives better results than changing the height of the ribs as far as rate of heat transfer and the friction factor are concerned. For the same increase in Nusselt number at a given Reynolds number, the friction factor increases by as much as 400% by increasing the height compared to decreasing the pitch of the ribs. However a quantitative analysis shows that the case which is showing maximum heat transfer enhancement (i.e. case C) shows 56% increase in Nusselt number at 10000 Reynolds number.

### 4. Conclusion

The immergence of a recirculation zone just behind the ribs disturbs the thermal boundary layer which enhances the rate of heat transfer. Nusselt number enhances with reduction in the pitch and by increasing the height and width of the ribs, resulting augmentations in convective heat transfer. Friction factor decreases with increase in pitch and reducing the height and width of the ribs. Friction factor varies very little as compared to Nusselt number by increasing Reynolds number.

### References

[1] Wcbb R L 1994 *Principles of Enhanced Heat Transfer* (New York-Wiley).

[2] Mohammed H A, Abdalrazzaq K A and Sheriff J M 2013 *Int. Comm. Heat Mass Transfer* **44**, 116-26.

[3] Han H Z, Li B X, Li F C and He Y R 2013 *Computers & Fluids* **91**, 107-29.

[4] Kathait P S and Patil K 2014 *Applied Thermal Engineering* **66**, 162-70.

[5] Liu J J, Liu Z C and Liu W 2014 *International Journal of Thermal Sciences* **89**, 34-42.

[6] Kareem Z S, Jaafar M N M, Lazim T M, Abdullah S and Wahid A F A 2015 *Alexandria Engineering Journal* **54**, 415-22.

[7] Incropera F P, DeWitt D P, Bergman T L and Lavine A S 6th Ed. *Introduction to heat transfer* (New Jersey-John Wiley & Sons, Inc).