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STUDY ON EFFECT OF GROOVES ON TUBE HEAT EXCHANGERS

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Abstract: This study focuses on the Numerical modeling and experimental verification for enhancement in heat transfer in heat exchanger using grooved tube. The objective of the present study was to develop a model of concentric tube heat exchanger having higher heat transfer capacity under specific condition than the similar plain tube heat exchanger. The geometry of the concentric tube heat exchanger was designed in Solidworks® and the meshing or discretization procedures were carried out in ICEM CFD 14.5The computational simulation was carried out by CFX 14.5 solver using Finite Volume Approach. For experimental verification, experimental testing procedure with equipment was established. The result from CFD and Experiment showed the 9.67- and 7.89-times enhancement in heat transfer respectively for varying Reynolds number between 500-5000.

***Index Terms* – Grooved Tube, Heat Transfer, Enhancement, CFD, Swirl.**

1. Introduction

Transfer of heat from one place to another due to temperature difference is known as heat transfer and those devices which is used to transfer heat are known as heat exchangers. These are the most common thermal equipment in large industries and manufacturing processes which are employed for the heat transfer, heat extraction and heat energy re-generation according to the space and time of the industries and manufacturing plants. Due to the considerations of the space, material cost and global economy various method for heat transfer enhancement are being developed. All these methods of heat transfer enhancement can be mainly classified into two major group, namely: active techniques and passive techniques. Active techniques make the utilization of external power for the heat transfer enhancement heat exchanger while in contrast passive techniques uses no direct power for the heat transfer enhancement. Each of these methods of heat transfer enhancement are effective according to the condition and mode of heat of transfer. Generally, heat transfer depends on the material property like thermal conductivity, fluid property like convection heat transfer coefficient, exposed surface area, velocity of fluid movement etc, which are considered constant for a specific case. Properties of heat transfer material and fluid cannot be changed but surface area and the speed of fluid can be altered for the same setup to enhance the efficiency of Heat Exchangers. In comparison with the plain tube, internal spiral groove imparts swirl motion to the axial flow in the near region of the tube wall which induces enhanced heat transfer [1],[2],[3].

Bergles and Morton [4], first studied about various augmentation techniques for convective heat transfer and mentioned various techniques of heat transfer as surface promoters, including roughness and treatment; displaced promoters, such as flow disturbers located away from the heat transfer surface; vortex flows, including twisted-tape swirl generators; vibration of the heated surface or the fluid near the surface; electrostatic fields; and various types of fluid additives. They also summarized the condition under which heat transfer is improved. The performance enhancement criteria for an internally corrugated tube or roughened surface and suggested the heat transfer enhancement factor in terms of Nusselt number and friction factor. One of the notable point in any augmented system is that the friction factor (f) of an enhanced surface in single-phase flow is higher than that of smooth surface when operated at same Reynold's number. This brings about significant pressure drop of the fluid flow. It is the pressure drop constraint that actually contributes to enhanced heat transfer. The increase in heat transfer can described by Nusselt number of the enhanced tube to Nusselt number of smooth tubes [5]. Experimental results showed that, among different kinds of twisted tapes including classic twisted tape, perforated twisted tape, notched twisted tape, jagged twisted tape, and butterfly insert, the Nusselt number and thermal-hydraulic performance of the jagged insert were higher than other ones followed by classic twisted tape, perforated twisted tape, and notched twisted tape. It can be concluded that the holes on the classic twisted tape negatively affected the heat transfer ratio. This trend was also same for the notched one and the results revealed that none of these changes in insert shapes are promising. However, a new designed perforated twisted tape with parallel wings had the heat transfer enhancement up to 208% compared to plain tube [6]. For a 75-start spirally grooved tube with twisted tape insert maximum enhancement in heat transfer in laminar region of flow inside the tube was reported to be around 600% and for the same assembly the heat enhancement in turbulent region was found around 160 %. However, for tube without the twisted tape insert or with only internal corrugation the reduction in heat transfer was noticed over transition of Reynolds numbers [7]. Investigation on friction and compound heat transfer behaviors of a dimpled tube fitted with a twisted tape insert showed that Nusselt number of the dimpled tube with twisted tape insert was 66 to 303% higher than plain tube and 15 to 56% higher than the dimpled tube without twisted tape in all Reynolds numbers. The combined average friction factor raised up to 2.12 times more than the dimple tube acting alone and 5.58 times of that in the plain tube and also the Nusselt number in the tube with the smaller pitch ratio was higher than in the one with the larger pitch ratio [8]. Bilen et al. [9] experimentally investigated the effect of groove geometry on the heat transfer and friction characteristics for the internally grooved tubes. Three geometric groove shapes (circular, trapezoidal and rectangular) were selected to perform the study. It was concluded that heat transfer enhancement was obtained up to 63% for circular groove, 58% for trapezoidal groove and 47% for rectangular groove, in comparison with the smooth tube at Re = 38000. Eiamsa-ard and Promvonge [10] performed a numerical study on the heat transfer of turbulent channel flow over periodic grooves. It was found that the grooved channel provided a considerable increase in heat transfer at about 158% over the smooth channel, and a maximum gain of 1.33 on thermal performance factor was obtained. In the study done by R. Naveenkumar et.al. [11] computational fluid dynamics results indicates that trapezoidal shaped grooving tubes attains the maximum value of drop in pressure, thermal performance and Nusselt number as compared to plain material tube and other grooved tubes pertaining to turbulent flow condition of Re from 5500 to 11500. The output signifies that the highest enhanced value of reduction in pressure attained in trapezoidal, circular and square grooving tubes are 68%, 65% and 63%. The CFD results also indicate that the highest value of increase of heat transfer rate (Q) is attained in trapezoidal grooved tube as 27%. The enhancement in Q value pertaining to circular and square grooved tubes are 21% and 14%.

Roughness mainly is most useful for the purpose of disrupting or reducing boundary layer flow of the fluid inside heat exchangers. Since no energy is used in this process of heat enhancement these methods are more popular with its low cost of manufacturing. Many studies are done in this sector either by computational approach or by experimental approach and is compared to past literatures. So this study is concerned in both computational and experimental approach to study the effect of grooves in tubes of heat exchangers.

2. METHODOLOGY

2.1 Creating Model

Using SolidWorks® 2014, the model of concentric tube heat exchanger was designed using the given dimensions. Inner tube internal surface was roughened to make it grooved surface

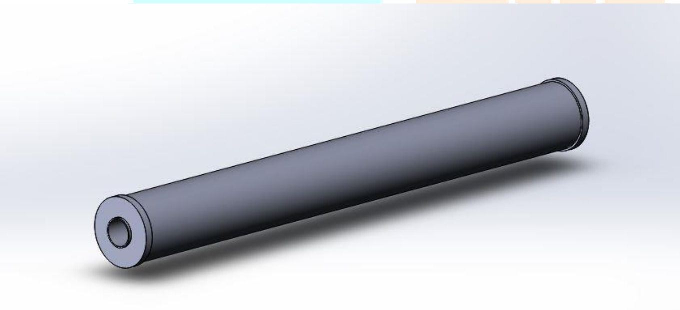

Figure: Conventional Heat Exchanger

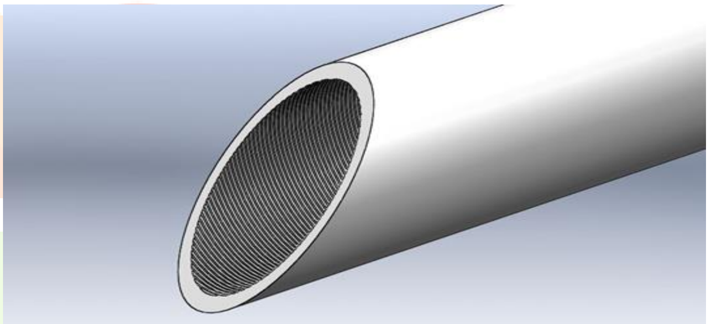

Figure 2: Grooved Tube

Table 1: Dimensions for the Test Specimen

S.N.	Design Parameters	Specifications (in mm)
1	Inner tube (ID)	23.4
2	Inner tube (OD)	25.4
3	Outer tube (ID)	46.8
4	Outer tube (OD)	50.8
5	Length of the specimen	500
6	Pitch of the Helix	37.5
7	Total No. of Revolutions	13.33
8	Height of the Grooves	0.4

2.2. CFX Simulation

The mesh created by ANSYS ICEM CFD v14.5 was imported to CFX and the setup is done. During the setup of the simulation, the fluid properties and the model of equations were chosen and the flow velocity and pressure values are supplied using the boundary conditions at various surfaces. The cell conditions were changed to fluid in the fluid flowing region and as solid in the wall region between two tubes in ICEM CFD itself. Then the solver is run and the necessary Root Mean Square (RMS) value of the residual was set as the convergence criteria. At last, the result was post-processed in CFD-POST, and various necessary parameters were viewed in CFD-POST.

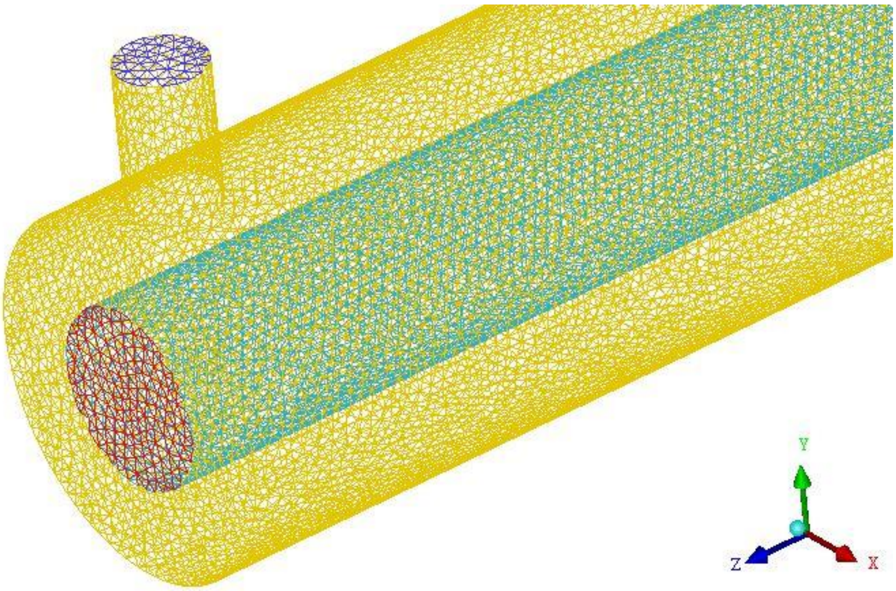

Figure 3: Mesh Generation during Simulation

2.3. Setup Parameters

For this study setup parameters were kept as below:

Table 2: Setup Parameters

Model	Laminar
Material	Water and copper (With 1 mm thickness) as the wall material
Viscosity	Constant
Reynolds Number	500-5000

2.4. Equations Involved

- Laminar model:

The equation involved is the Navier-Stokes equations of conservation of mass and momentum equation. The equations are as follows:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \dots\dots (1)$$

$$\rho\frac{DV}{Dt}=\rho g+\nabla\cdot\tau-\nabla p \dots\dots (2)$$

- Thermal Energy model:

$$P\frac{\partial K}{\partial t}+\rho\nabla\cdot(UK)=-U\cdot\nabla p+U\cdot(\nabla\cdot\tau) \dots\dots (3)$$

2.5. Experimental Setup

An open loop test rig with water as working fluid was constructed for experiment. Cold water from ground tank was supplied continuously to overhead tank and was heated with the help of heaters to the required temperature (nearly 80^0C). Heated water in overhead tank was passed through the test section containing smooth tube/grooved tube and data was noted. A U-tube manometer was placed to note pressure drop in water after passing through test section. The circulated water from test section was then collected in collector tank.

Meanwhile, the cold water was fed from the ground tank via centrifugal pump. (Analogous to the connection in hot water tube section, the pump was followed by a T-Joint to allow bypass (for the measurement of flow rate) and the Ball Valve to regulate the flow of cold water to the outer tube of the concentric heat exchanger. The cold-water discharge was passed to collector tank and then to the drain.

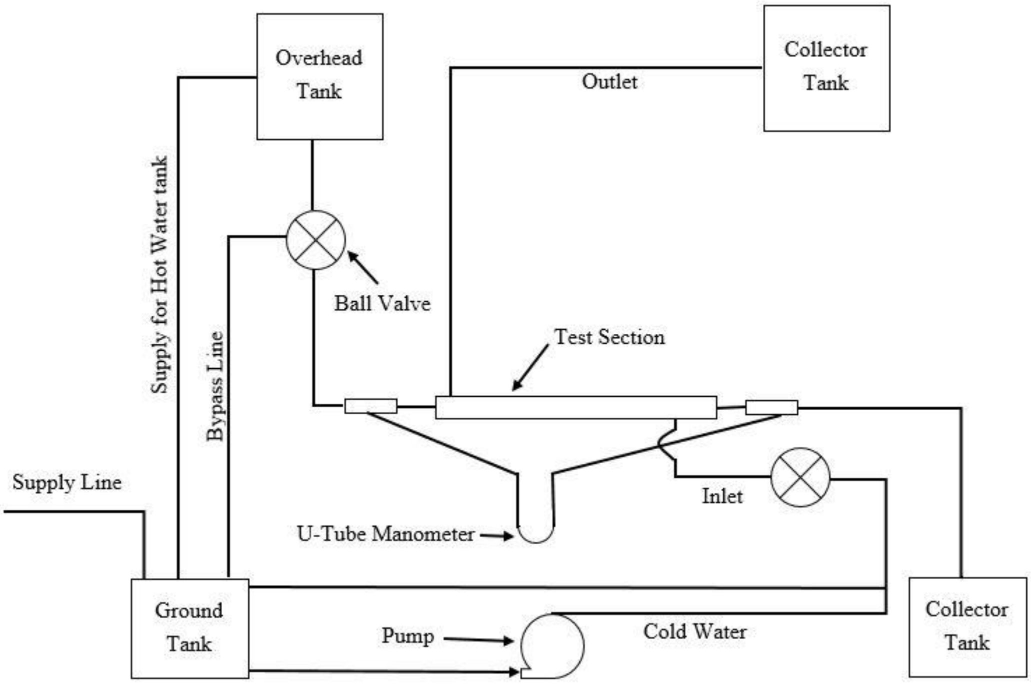

Figure 4: Experimental Routing Setup

Table 3: Orientation of Test section during testing

S.N	Orientation Parameter	Height from datum (m)
1	Height of the heat exchanger tube	0.58
2	Height of the base of the overhead tank	1.7
3	Height of the top of the overhead tank	1.48
4	Height of the CG of the overhead tank	1.59

2.6. Performance Enhancement Criteria

PEC is a figure of merit to determine numerically the enhancement achieved by incorporating passive enhancement techniques in heat exchanger. Mathematically the performance enhancement criteria is given by: [12]

$$PEC = \frac{\frac{Nu}{Nu_p}}{\left(\frac{f}{f_p}\right)^{0.291}}$$

2.7. Grid Independence Test

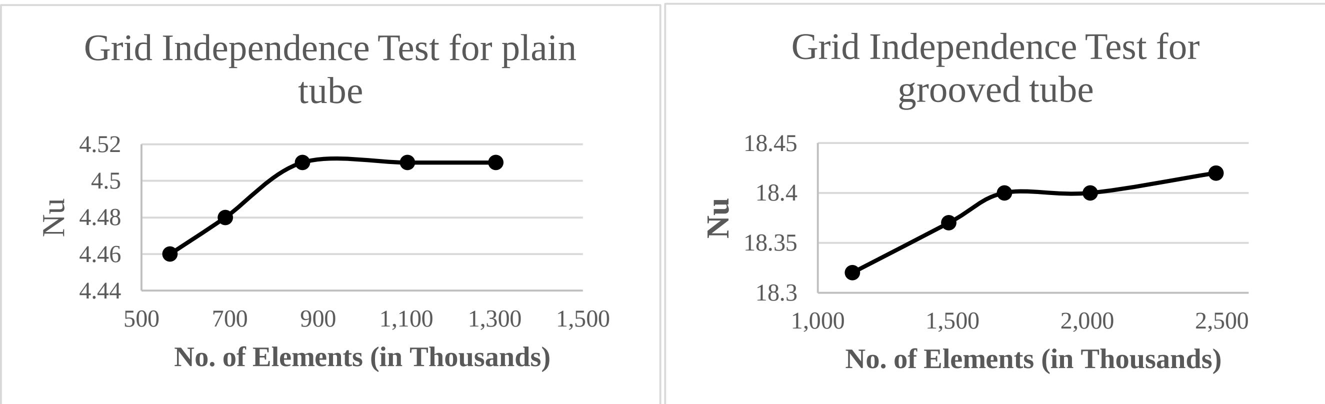

Figure 5: Mesh Independence Test for smooth tube and Grooved Tube

Grid independence test is performed to eliminate/reduce the influence of the number of grids/grid size on the computational results. The Grid Independence Test was performed in each of the cases respectively. For the case of smooth tube and grooved tube the grid independence test result is shown below. Here 864,272 number of elements were chosen for smooth tube and 2,011,598 number of elements were chosen for the case of grooved tube.

3. RESULTS AND DISCUSSION

3.1 CFD Results

Obtained data from CFD analysis is plotted For the presentation of results, the basic parameters chosen are friction-factor (f) and Nusselt Number (Nu). The variation of friction factor and Nusselt number is observed with respect to the varying Reynolds Number (Re). Below graphs are plotted to express and compare result for smooth Tube and Grooved Tube.

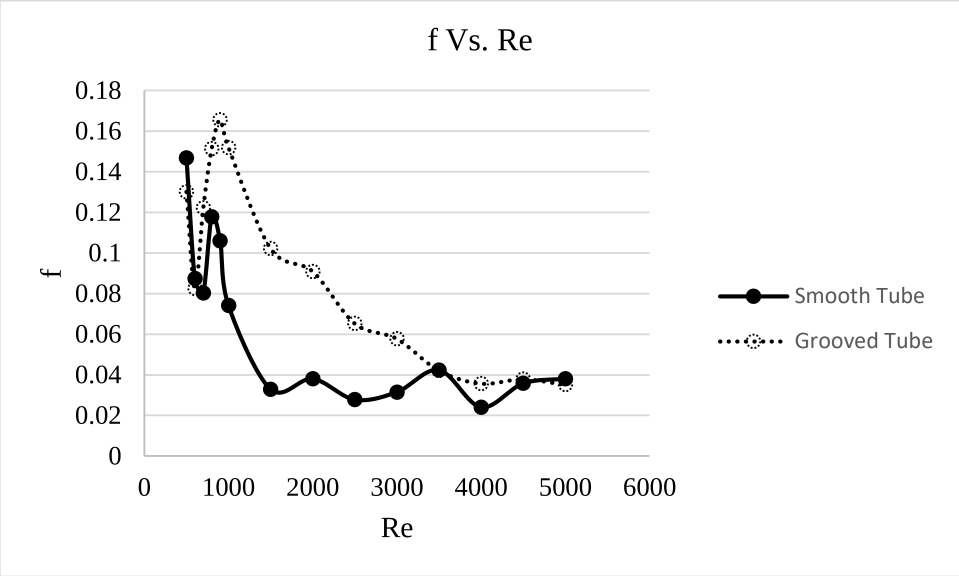

Figure 6: Plot for friction factor versus Reynold's number

As sown in above figure friction factor decreases with the increase in Reynolds number. It is due to the fact that the grooved tubes produce more turbulence than the smooth tube, which then increases the heat transfer area. They also provide periodic redevelopment of the boundary layers and cause a more effective heat transfer. Comparing to the flow of fluid in smooth tube, the grooved tube flow's thermal boundary layers become thinner, and secondary vortices of the grooves contributed to the enhancement of the heat transfer.

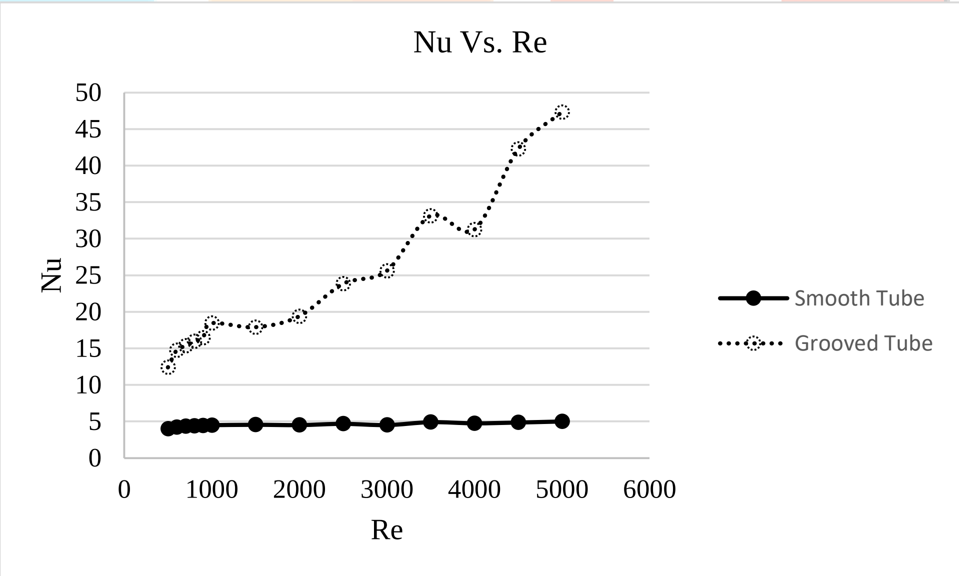

Figure 7: Plot for Nusselt Number Vs Reynolds number

It can be seen in above figure that the Nusselt number increases as there is increase in Reynolds number. The increment of Nusselt number in Grooved tube is noticeably high than that of smooth tube which signifies the enhancement of heat transfer. Grooved surface in the tube helps to reduce thermal boundary layer and also provide large contact surfaces between fluid and wall so as to increase the heat transfer between inner and outer fluid.

3.2 Experiment Results

Experiment was performed as described in methodology with the record of volume of fluid flow temperatures of hot water, cold water in inlet and outlet of test section and different portion of test section. Similarly, for the experiment part plot of friction factor Vs. Reynolds number and Nusselt number Vs. Reynolds number was plotted. As in CFD result friction factor for smooth tube and grooved tube decreases while increase in Reynolds number. It is due to the increase in velocity which reduces the formation of boundary layer which then decreases the friction factor.

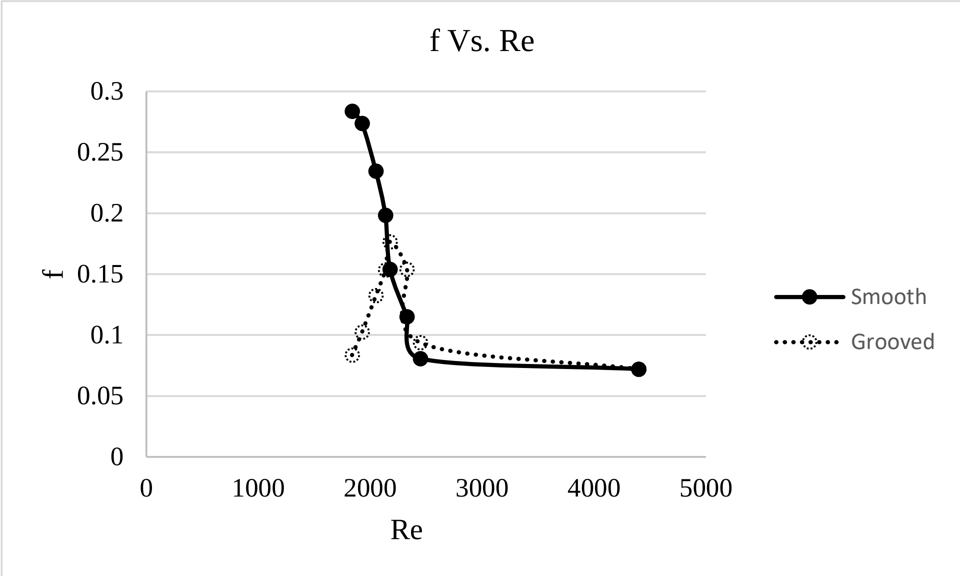

Figure 8: Experimental data Plot for friction factor vs Reynolds number

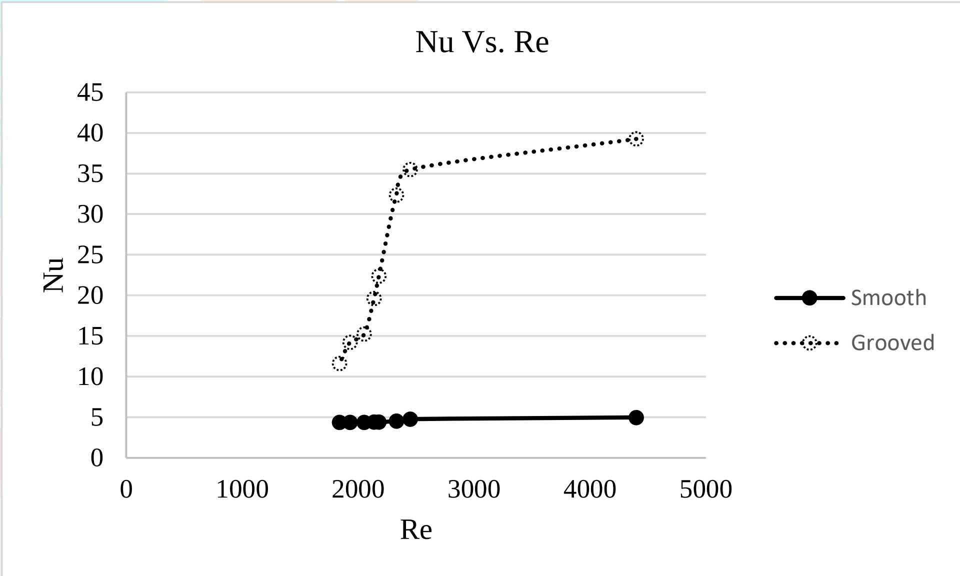

Figure 9: Experimental data Plot for Nusselt Number vs Reynolds number

Similarly, for the experimental result Nusselt number is considerably high than that of smooth tube with increase in Reynolds number. Roughness inside the tube helps to reduce thermal boundary layer and helps to increase the heat transfer.

3.3 Correlation between various parameters

The correlation for Nusselt number and friction factor was developed using least square method of regression of regression analysis. The relation is valid for laminar and transitional flow of i.e for Re<5000.

Table 4: Correlations obtained from Experiment

Experiment Correlations		
Parameter	Correlations	Range
Friction Factor(f)	$f = 417.77 \times Re^{-1.03}$	$500 \leq Re < 5000$
cNusselt Number(Nu)	$Nu = 0.018 \times Re^{0.92061}$	$500 \leq Re < 5000$

Table 5: Correlations Obtained from CFD

CFD Correlations		
Parameter	Correlations	Range
Friction Factor(f)	$f = 0.0802 \times Re^{0.0677}$	$500 \leq Re < 1500$
	$f = 160.73 \times Re^{-0.9983}$	$1500 \leq Re \leq 5000$
Nusselt Number(Nu)	$Nu = 1.7573 \times Re^{0.3278}$	$500 \leq Re < 1500$
	$Nu = 0.04336 \times Re^{0.8095}$	$1500 \leq Re \leq 5000$

3.4 Comparison between CFD and Experimental Results

3.4.1 Smooth Tube

The plot of friction factor vs Reynolds number and Nusselt number Vs. Reynolds number obtained from CFD and Experimental results is compared. Although there is some difference in obtained results but both results show the similar trend i.e., friction factor decreases with increase in Reynolds number and Nusselt number increases with increase in Reynolds number indicating increase in heat transfer. It is due to the fact that increase in velocity reduces the formation of boundary layer which then decreases the friction factor. Increase in Nusselt number is also observed with the increase in Reynolds number which indicates the increase in performance enhancement criteria.

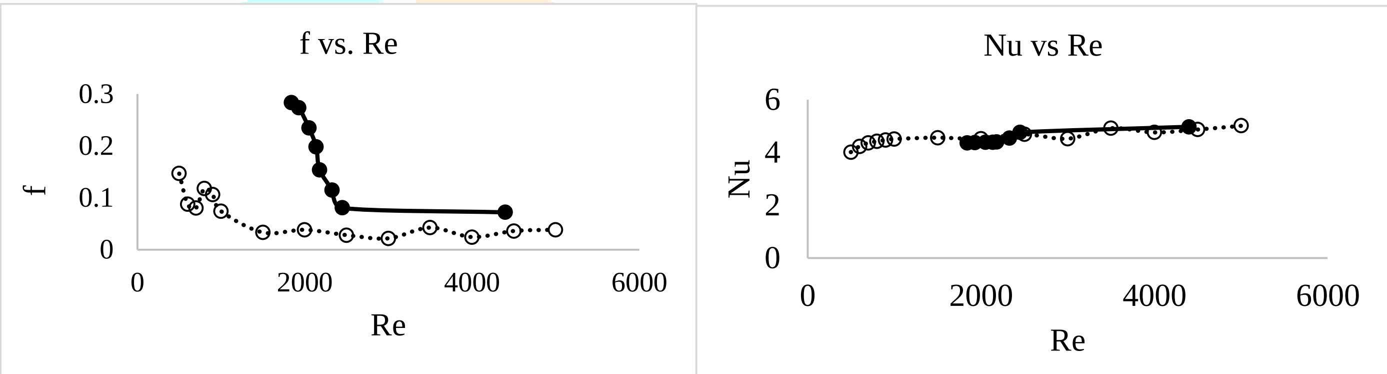

Figure 10: Comparison graph for experimental and CFD results for Smooth Tube

3.4.2 Grooved Tube

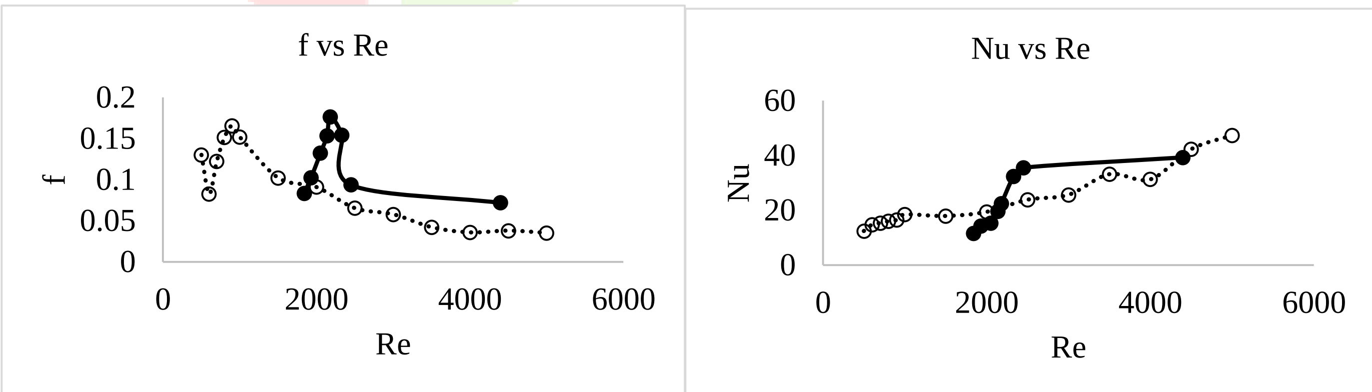

Figure 11: Comparison graph for experimental and CFD results for grooved tube.

Some deviations in results are observed in CFD and Experiment as plotted in above graph. In case of CFD analysis there is no loss of heat inside the test section, while during experimentation there occurs losses of heat of hot water from hot water container, connecting pipes and test section which had not been considered during calculation of the results. Corrugation in copper tube used in experiment was done using sand paper which may not be regular whereas in case of CFD same was modeled in Solidworks with exact dimension.

3.5 Performance Enhancement Parameter

To gain the significance of the data obtained from the CFD simulation, the results were compared against the result obtained by Experiment at the Re-range of 500-5000. The defining parameter for the heat transfer enhancement is called Performance Enhancement Parameter (PEC).

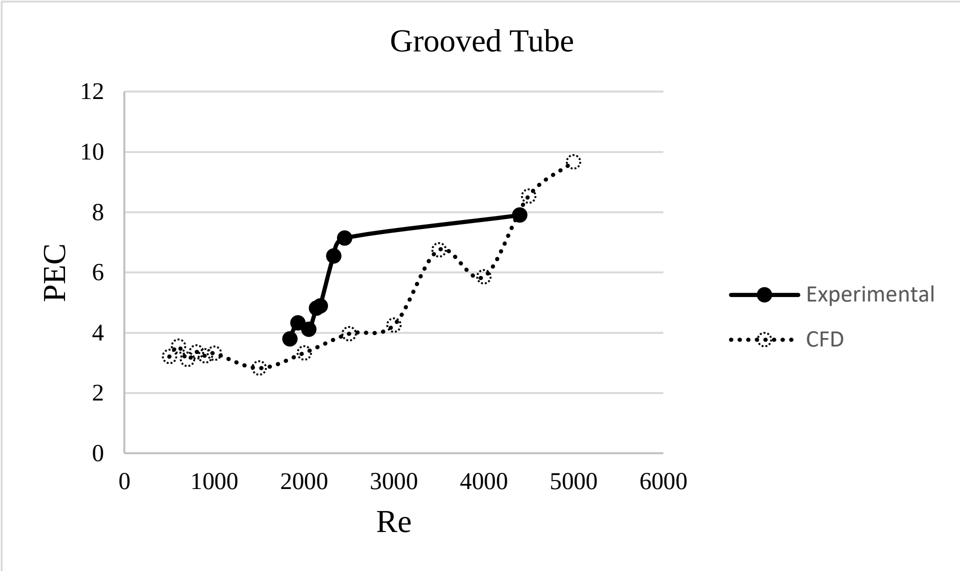

Figure 12: Performance Enhancement Parameter Vs. Reynolds Number for Tube with Twisted Tape

From above figure it is shown enhancement of heat transfer is obtained by using grooved tube. The maximum enhancement obtained during this study using CFD and Experiment was 9.67 and 7.89 respectively.

4. Conclusion

From this study it concludes that, heat transfer is better for the grooved tube with the twisted tape inserts. The grooved tubes provide more increase in heat transfer augmentation due increase in the heat transfer area. It is with the fact that the grooved tube reduces the boundary layer thickness which in turn increases the heat transfer rate between the inner tube and outer tube. They also provide periodic redevelopment of the boundary layers and cause a more effective heat transfer. In this study, numerical modelling and experiment are performed at various range of Re and are compared. From the study of various parameters like f and Nu, it leads to the conclusion that, heat transfer is indeed enhanced with the use of grooved tube rather than plain tube. The results obtained from the study showed Maximum enhancement of 9.67 using CFD and 7.89 using Experiment procedure.

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