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Heat Transfer Enhancement of Concentric Heat Exchanger using Compound Techniques

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Abstract: In this present study, numerical modelling and experimental verification of heat transfer enhancement using compound technique of a concentric tube heat exchanger is done. Laminar flow is considered for both the numerical and experimental study. Among the active and passive technique, passive techniques were employed with surface roughened and insertion of twisted tape. A Computational Fluid Dynamics approach was used for numerical analysis which was further verified experimentally. The geometry of the concentric tube heat exchanger was designed in SolidWorks® and the discretization was carried out in ICEM CFD 14.5. The computational simulation was carried out by ANSYS CFX 14.5 solver. The experimental verification was conducted in the same Reynold's Number range used in the numerical analysis, of 500-5000. The heat transfer enhancement was calculated through Performance Evaluation Criteria (PEC) which was found to be 9.67 and 7.89 for CFD analysis and Experiment respectively.
Keywords: CFD, Grooves, Heat Enhancement, Swirl, Twisted Tape (TT).

I. INTRODUCTION

Among various uses of heat exchangers in our daily life, they are extensively used in industries, automobiles, HVAC etc. in order to extract heat from the steam or to cool the engine or in rejecting heat to the surrounding. Industries strive to increase the amount of heat transfer in the heat exchanger, which requires cost effectiveness to be considered extensively. Active enhancement technique of heat transfer requires external power source like electricity, vibration, etc. whereas passive enhancement techniques are solely dependent on the geometry of the heat exchange tube. Passive techniques are ideal than active ones, due to its cost-reduction, maintenance and its size reduction. Since a single passive heat transfer enhancement technique has a limited scope in enhancement, a compound approach to two or more passive techniques is able to produce many of the time added effect on heat transfer enhancement. Some of the example of the compound techniques are internal corrugation with external fin, external fin with helical baffle, internal corrugation with twisted tape and so on with superimposed two or more above described active and passive method of heat transfer enhancement [1].

Several studies had been conducted on the field of heat transfer enhancement since 1960, it is when Bergles, Webb and other leading pioneers had contributed significantly to develop the field of heat transfer enhancement. In heat exchangers, the efficiency is decreased due to the boundary layer at the solid–liquid interface. To improve heat exchange rate between the fluids several methods are implemented which includes (a) surface modification of the tube wall [2], [3] (b) use of nanofluid as the working fluid [4], [5], [6] and (c) use of inserts in the internal flow [7], [8], [9], [10]. Bergles and Morton 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 [11]. Webb extends the previous work of Bergles et al., and defined the performance enhancement criteria for heat transfer enhancement for different conditions such as reduced heat exchanger material, increased heat duty, reduced logarithmic-mean temperature difference (LMTD) [12],[13]. Study shows the increase in heat transfer can be described by Nusselt number of the enhanced tube to Nusselt number of smooth tube [14]. Study on converging-diverging tube with evenly spaced twisted tape concluded that the best performance among the four types of tested twisted tape a can be expected from the one with twist ratio (H/D) = 4.72 [15]. 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 [16].

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 [2]. In case of tube involving twisted tape only, different geometry of twisted tapes were studied by different researchers. For peripherally cut twisted tape inside the copper tube the enhancement in heat was found to be about 86% while in case of serrated twisted tape the mean heat transfer rate was increased up to 72.2%. The thermal performance factor of serrated twisted tape was increased with increasing depth ratio and serration depth over twisted tape width and decreased with decreasing serration width ratio and ratio of peripherally cut tape width over twisted tape width [17]. 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 [18]. Similarly, as that of the passive technique involving only twisted tape, some studies had been done in another method of passive heat transfer enhancement. M.M. Rahman et al. (2013) performed numerical simulation of fluid flow and heat transfer in inner grooved copper tube, in which tube model with inner grooved tube was generated using SOLIDWORKS while the mesh generation and mesh refinement was done in GAMBIT and followed by simulation in fluent solver with designed boundary conditions. When the results from the simulation was compared with the existing literature, they found that heat transfer enhancement in the range of 649.66% to 917.22% of inner grooved tube compares to plain tube [19].

A majority of heat exchangers used in multiple applications are simple and conventional concentric tube heat exchangers with counter fluid flow. A coiled or long plain tube with smooth surface texture and material of high thermal conductivity has been used so far to regulate the heat flow. The design is widely celebrated due to its simplistic yet effective approach. However, recent developments and innovations present some intriguing facts that this design possess lots of potential without any major modification in its design constraints. In other words, it could perform a lot better if we concentrate on few minor details of the design specifications. Integration of passive enhancement techniques including corrugated surface texture and insertion of twisted tapes is a promising approach. Combining these methods not only ensure an improved heat transfer rate but also reduce cost, effort and active components that might be required to attain a similar level of heat transfer efficiency.

II. METHODOLOGY

A. *Modelling of Geometry*

The Solid Geometry for Simulation was generated using SolidWorks® 2014. Each of the parts, Twisted Tape, Grooved Tube, Outer Tube, Smooth Tube etc. were modelled separately and assembled to form the geometry.

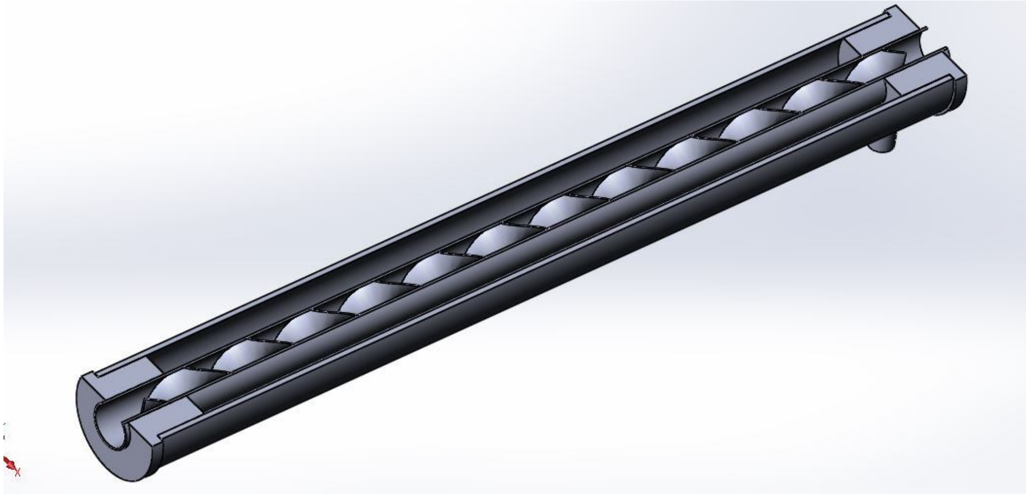

Fig. 1 Sectional View of the main Test section

TABLE I
Specification of Test section

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	Length of the TT	500
7	Width of the TT	23.4
8	Thickness of the TT	1.00
9	Twist Ratio (H/D)	1.71
10	Pitch of the Helix	37.5
11	Total No. of Revolutions of Grooves	13.33
12	Height of the Grooves	0.4

B. Simulation Procedure

1) *Governing Equations and Boundary Conditions:* The equation involved is the continuity equation, Navier-Stokes equations and energy equation. The equations are as follows

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

$$\rho\frac{Dv}{Dt} - \rho g + \nabla\cdot\tau - \nabla P \quad \text{.....(2)}$$

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

For each of the computational domain, velocity inlet and pressure outlet boundary condition was provided to inlet and outlet section respectively. Wall, with no-slip boundary condition was imposed on the wall surface. An interface between the copper tube and outer PVC tube was defined with the material as copper.

2) *Setup Parameters:* For this study setup parameters were kept as below

Table III
Setup Parameters

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

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3) *Grid Independence Test:* Grid independence test is performed to eliminate/reduce the influence of the number of grids/grid size on the computational results. The grid independence for one geometry will be applicable only for that geometry. The Grid Independence Test was performed in each case.

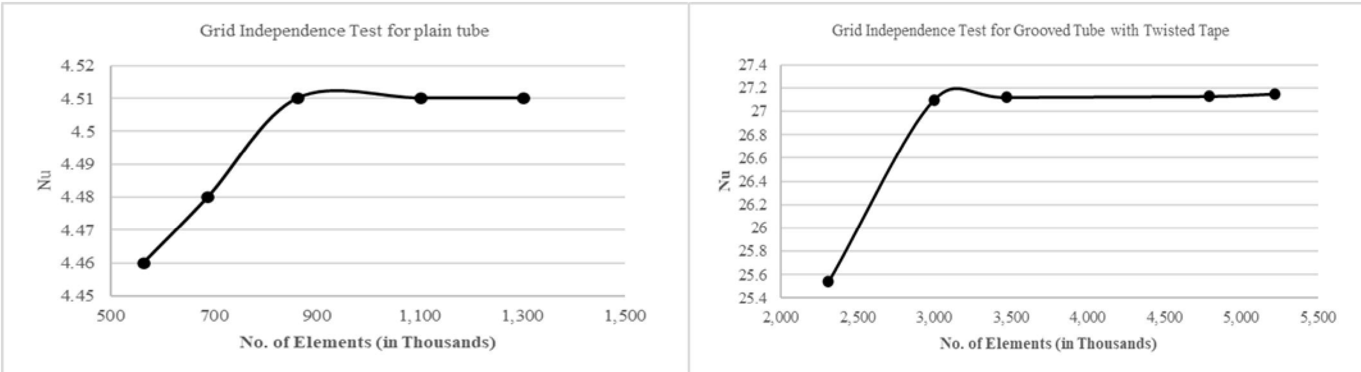

Fig. 2 Grid Independence Test for Plain and Grooved Tube with TT insert

867272 number of elements were chosen for smooth plain tube and 3473492 number of elements were chosen for grooved tube with twisted tape insert.

4) *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.

C. Experiment Procedure

1) Mathematical Procedure for the Experiment

a) Working Fluid: The experiment was carried out on single-phase liquid flow with water as the working fluid.

Pressure drop measurement: In order to evaluate f–Re relationship for the grooved tube with and without twisted tape, the pressure drop Δp and average velocity U were measured. The friction factor is defined as:

$$f = \frac{1}{2}\,\frac{\Delta P}{L}\,\frac{D_i}{\rho\,U^2} \quad \text{..........(4)}$$

Since Δp recorded using manometer at two end of the test piece, Di, L, ρ is known and the average velocity was evaluated from U = Q/A and nominal cross-sectional area A=πDi/4, The friction factor can then be evaluated. Then since the Reynold's numbers on the basis of nominal diameter is defined as:

$$Re = \frac{\rho\,U\,D_i}{\mu} \quad \text{..........(5)}$$

We can plot f-Re graph as measured from the experimentation for both the plain and enhanced tube. Now to evaluate the Nusselt number we use length averaged Nusselt number using the formula:

$$Nu = \frac{h\,D_i}{k} \quad \text{..........(6)}$$

where, $h = \frac{Q}{\pi \times D_i \times L \times (T_{w,i} - T_{w,o})}$

Where k is the thermal conductivity of the water. Hence, we calculated the Nusselt number for different conditions of the experimentation and then we plot the Nu-Re graph for each condition of the experimentation.

2) *Experiment Setup:* Experiments were conducted in an open loop test-rig with water as the working fluid. Water was continuously supplied from a ground tank to an overhead tank (25 liters bucket) manually without a pump where it was heated to desired temperature (nearly 800C) using three heater coils (each with capacity of 1500 watts). The heated water was then drawn to the main line via T-Joint (allowing bypass for the measurement of flow rate) and then to the ball valve to regulate the desired level of flow to inner tube (hot section) of the heat exchanger. The discharged hot water was then collected to the Collector tank (25 liters bucket) which was later fed back to the overhead tank for heating.

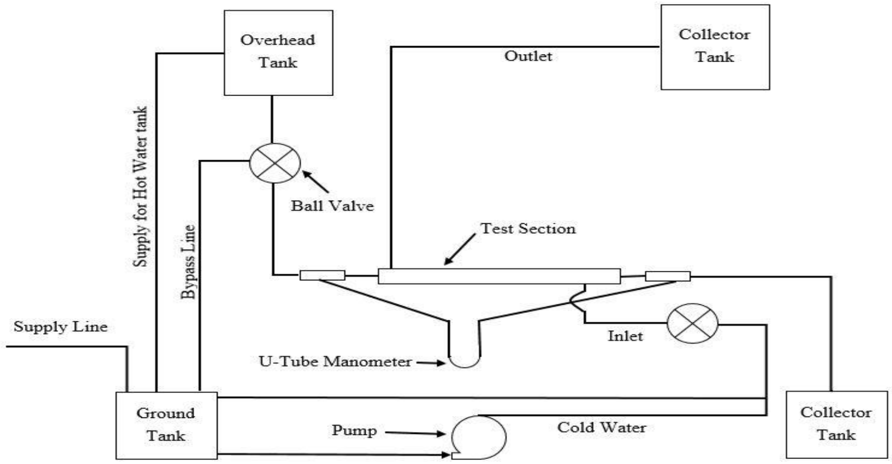

Fig. 3 Experimental Test Rig

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.

D. 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:[14]

$$PEC = \frac{\frac{Nu}{Nu_p}}{\left(\frac{f}{f_p}\right)^{0.291}} \quad \text{……………………………………………………………..….(7)}$$

III.RESULTS AND DISCUSSION

A. CFD Results

For the presentation of results, the basic parameters chosen are friction-factor (f) and Nusselt Number (Nu). The variation of these parameters is observed with respect to the varying Reynolds Number (Re). These parameters are used to express result for the cases of Smooth Tube and Grooved Tube with twisted tape insert.

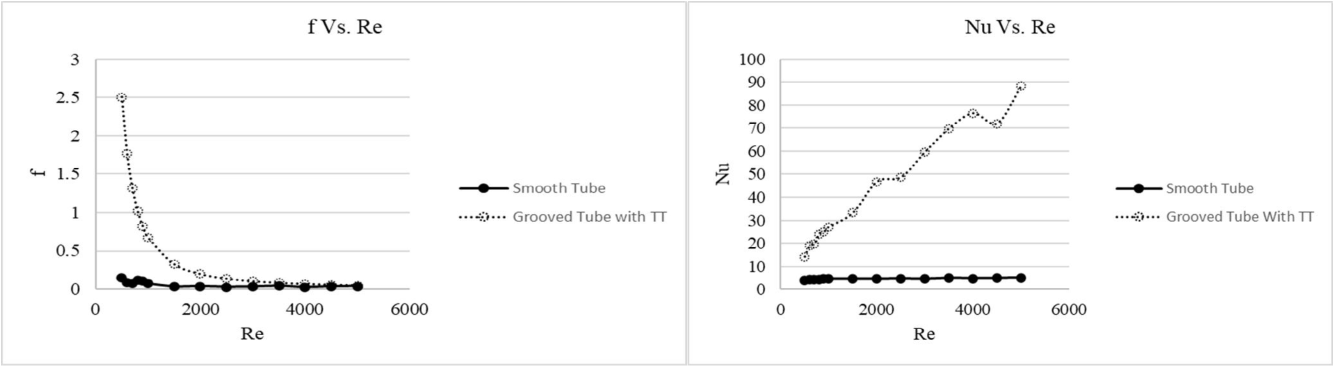

Fig. 3 Plot of friction factor, Nusselt Number Vs. Reynolds Number for CFD

Above figure shows the decrease in friction factor as increase in Reynolds number. It is in the fact with that, friction factor decreases when there is increase in velocity of fluid flow. Furthermore, Nusselt number is increased with the increase in Reynolds number. The rise in Nusselt number is considerably high in the grooved tube with twisted tape insert compared to smooth tube. The combination of grooved tube and twisted tape insert was found to be more effective than the smooth tube. This is due to the combined effect of both techniques which enable for better mixing of fluid and increased effective length inside the tube.

B. *Experiment Results*

Experiments were conducted with pure water in plain tube under laminar flow conditions 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. The variation of Nusselt number and friction factor with Reynold's number is shown in figure.

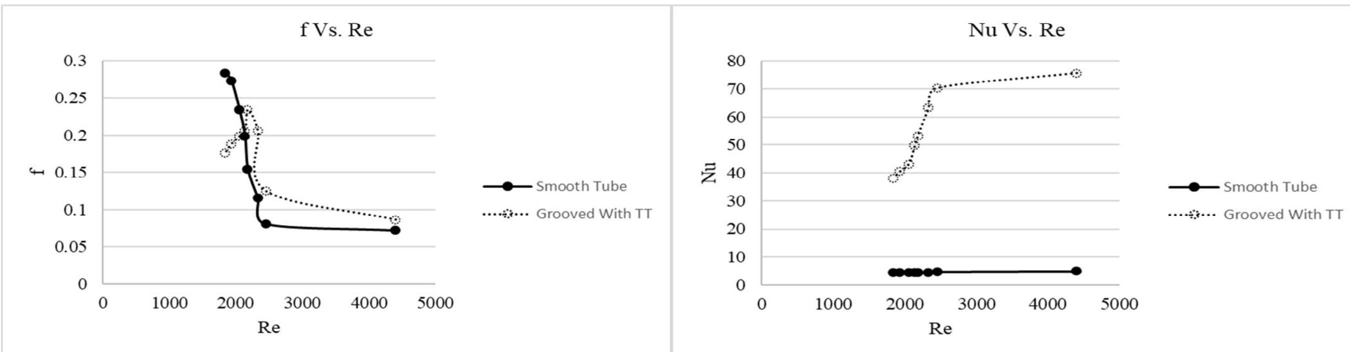

Fig. 4 Plot of friction factor, Nusselt Number Vs. Reynolds Number for Experiment

Heat transfer enhancement associated by the applications of grooved tube with twisted tape insert is found to be more effective than smooth tube. This can be the consequence of the combined mechanisms induced by both techniques. The presence of both grooved tube and twisted tape possibly promotes the dispersion and random movement of the particles, resulting in a better mixing between the core fluid and the tube wall. In addition, both grooved tube and twisted tape provide large contact surfaces between fluid and wall and thus, heat transfer area.

C. *Correlations*

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 IIIII
Correlations Obtained From Experiments

Experiment Correlations		
Parameter	Correlations	Range
Friction Factor(f)	$f = 3192.27 \times Re^{-1.251}$	$500 \leq Re < 5000$
Nusselt Number(Nu)	$Nu = 0.643 \times Re^{0.5715}$	$500 \leq Re < 5000$

Table IVII
Correlations Obtained From Cfd

CFD Correlations		
Parameter	Correlations	Range
Friction Factor(f)	$f = 263476.8 \times Re^{-1.8628}$	$500 \leq Re < 1500$
	$f = 32486.17 \times Re^{-1.5796}$	$1500 \leq Re \leq 5000$
Nusselt Number(Nu)	$Nu = 0.1499 \times Re^{0.74788}$	$500 \leq Re < 1500$
	$Nu = 0.13687 \times Re^{0.75745}$	$1500 \leq Re \leq 5000$

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D. Comparison Between CFD and Experimental Data

For the verification of the result, results obtained from CFD and Experiment were compared.

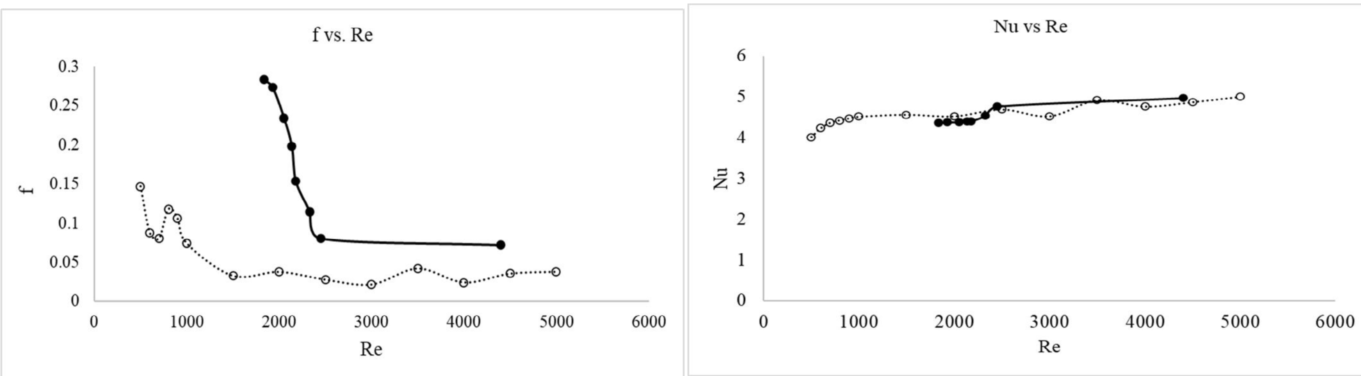

Fig. 4 Plot of friction factor, Nusselt Number Vs. Reynolds Number of Smooth Tube

As sown in above figure friction factor decreases with the 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. Increase in Nusselt number is also observed with the increase in Reynolds number which indicates the increase in performance enhancement criteria

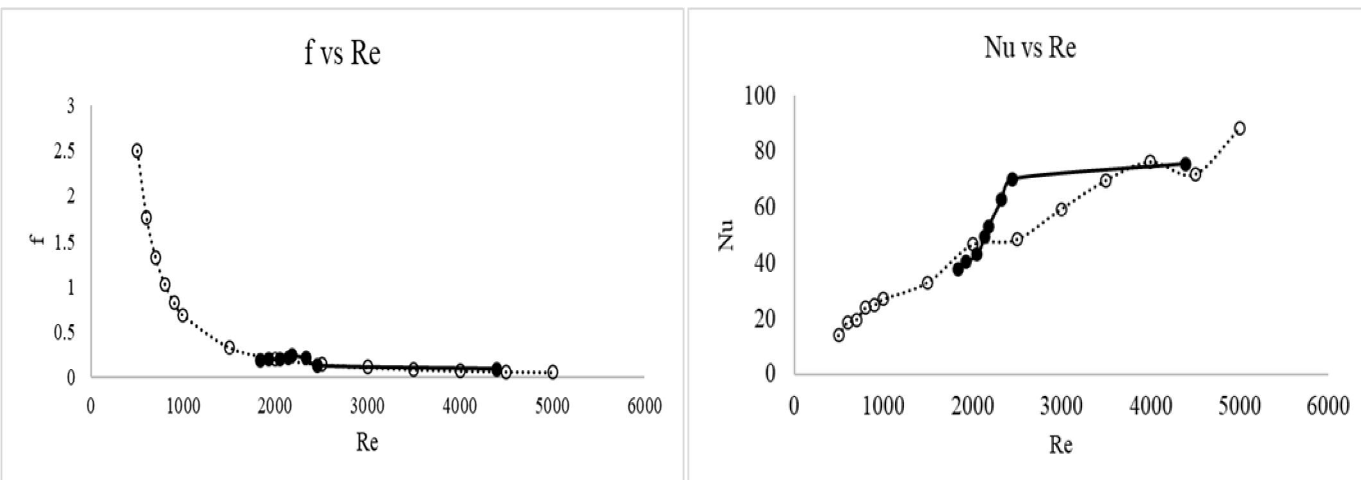

Fig. 5 Plot of friction factor, Nusselt Number Vs. Reynolds Number of Grooved Tube with Twisted Tape Insert.

In case of grooved tube with twisted tape the nature of friction factor versus Reynolds number graph shows that the friction factor in case of CFD decreases drastically up to the 1000 Reynolds number. This is the due to the fact that combined effect of twisted tape and corrugation imparts very strong swirl to the flow and boundary layer reduction. Hence the frictional component of the flow has very little effect on the flow. Both results obtained from CFD and Experiment shows similar trend and somewhat deviation from each other. It is due to the fact that, there was no loss of heat in CFD analysis compared to experiment. Further the grooves in tubes were made using sand paper which may not be accurate compared to the CFD model which was modelled using Solidworks software.

E. 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 shows the enhancement obtained at different Reynolds number using Grooved tube with twisted tape insert. The maximum enhancement obtained during this study using CFD and Experiment was 16.48 and 11.36 respectively.

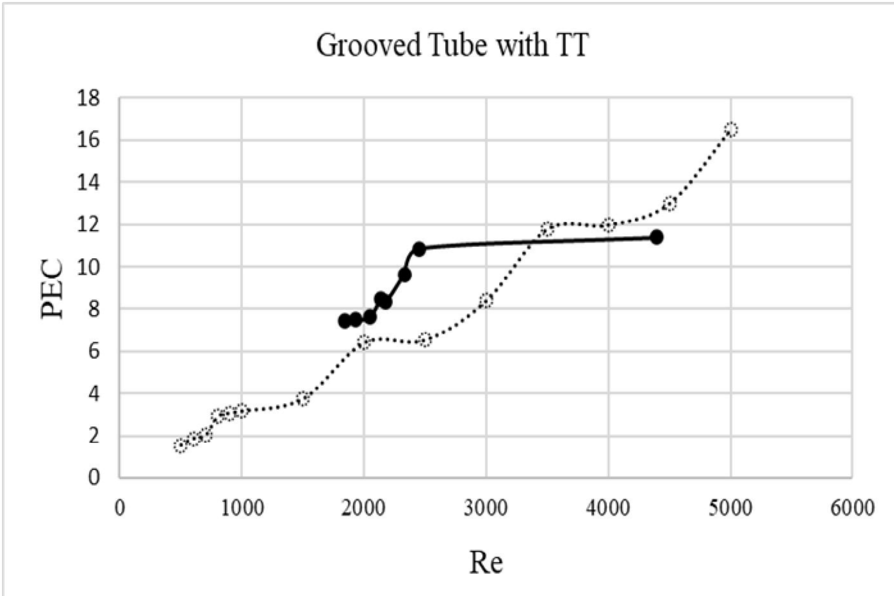

Fig. 6 Plot PEC Vs. Reynolds Number obtained from Grooved Tube with Twisted Tape Insert.

IV. CONCLUSIONS

The comparative study of the Plain and Grooved Tube with the twisted tape inserts concludes that, heat transfer is better for the grooved tube with the twisted tape inserts. 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. The twisted tape imparts the necessary swirl to the axial flow to the incoming fluid. This swirl component provides more time of contact between the tube and the water. Grooves and Twisted tape together further synergize the enhancement given by both techniques to give better heat transfer. Thus, this study concludes that the heat transfer for tube heat exchangers is increased by the use of Twisted Tape inserts and by the use of roughened tube. The numerical modelling is performed at various range of Re and is compared with that of the experimental results. The maximum enhancement obtained from CFD and experiment in this study was 9.67 and 7.89 respectively.

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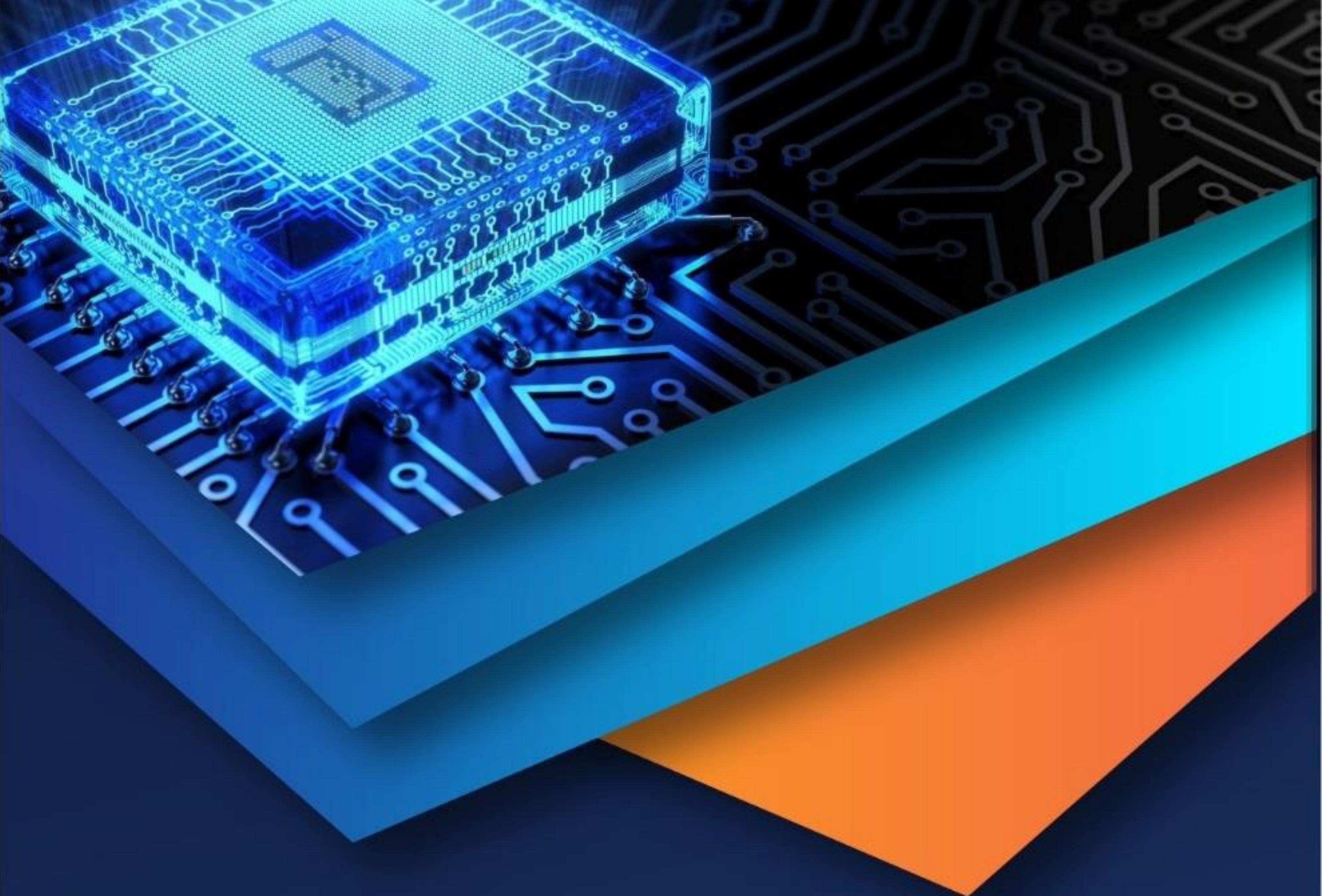

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