HEAT TRANSFER ENHANCEMENT OF RECEIVER TUBE USING TWISTED TAPE INSERTS

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Abstract: The paper focuses on CFD analysis and experimental investigation of heat transfer enhancement of circular receiver tube fitted with twisted tape inserts. Water is flowing through a circular receiver tube having inner diameter and length of 19 mm and 1000 mm respectively. The tapes are inserted in tube at three different twist ratio(y) of (2, 2.67 and 4) having 45mm, 60mm and 90mm pitch. CFD analysis is performed on plain tube by using standard k-€ and SST-k-ω model. SST-K-ω model was selected for further studies as it is more reliable model and provides more accurate prediction of flow separation. CFD results shows that 45mm pitch twisted tape gives better result as compared to 60mm & 90mm pitch twisted tapes. For further experimentation, 45mm pitch twisted tape is selected. The result obtained from plain twisted tapes are compared with plain tube within the Reynolds number range from 3193.38 to 15962.2. The experimental & CFD results reveals that heat transfer rate and friction factor in a circular tube fitted with twisted tape insert were significantly higher than the plain tube. Based on thermal performance factor, it was observed that 45 mm pitch twisted tape insert gives maximum Π_{th} as compared with 60 mm and 90 mm pitch.

Keywords **- Heat transfer augmentation, Twisted tape insert, Twist ratio (y), Thermal performance factor (ηth)**

I. INTRODUCTION

In many parts of the world, there has been a shortage of non-renewable source of energy and due to this shortage, its price is also increased a lot. India is already facing very serious energy crisis. In 2017, crude oil imports hit a record in India. So, India's crude oil import rises by 1.8% to 4.3 MMbpd (million barrels per day) in 2017 [1]. Presently the non-renewable source of energy had come into its final stage and there comes the need of developing other energy alternatives. Solar energy is one of the most useful alternative to overcome energy crisis problem. Solar energy is the light and heat from the sun that is being harnessed. Solar energy can be used in variety of applications, but the most beneficial solar thermal application is solar water heating which can be used for residential as well as industrial purpose. The performance of solar water heating system improves if the efficiency of its receiver tube is increased keeping all other variables constant. In general, the most significant variable in improving the efficiency of receiver tube are heat transfer coefficient and friction factor.

In general, the heat transfer augmentation techniques are mainly divided into three categories: (1) **active technique** which need external power source (2) **passive technique** which do not need external power source. Some examples of passive techniques include [a] insertion of porous, [b] twisted strips and tapes, [c] wire coil and helical wire inserts, [d] helical screw tapes, [e] regularly spaced twisted tapes and many others, and (3) **compound technique** which is combination of two or more techniques.

To enhance the heat transfer rate various active and passive techniques are available which has been discussed by the Bergles [2] and Bergles and Webb [3]. Suvanjan Bhattacharyya et.al [4] has performed a numerical study of thermal performance factor in a tube equipped with twisted tape swirl generator and it has been observed that the heat transfer value goes on increasing as entrance angle value increases. Keguang Yao et.al [5] has performed a numerical simulation of heat transfer performances and flow of solar water heaters equipped with twisted tape inserts for different initial temperatures and conclude that by using twisted tape inserts it will make the temperature field more uniform and will reduces the velocity magnitude. Eiamsa-ard et.al [6] has performed the 3D numerical simulation of swirling flow induced by means of loose fitted twisted tape inserts and found that by increasing the clearance ratio the value of HT-rate goes on decreasing. Xiaoyu Zhang et.al [7] has studied the heat transfer and friction factor in a tube fitted with helical screw tape without core rod inserts and it has been observed that heat transfer enhancement in a boundary flow of the tube is worse than that of the heat transfer enhancement inside the core flow of the tube. S. R. Shabanain et.al [8] has done the CFD simulation of heat transfer and friction factor by using three tube inserts mainly classical, butterfly and jagged inserts, in the studied range of Reynolds number. Simulation results conclude that as compared to jagged type and twisted type tape inserts, the butterfly type inserts have maximum HT-rate. S. Eiamsa-ard et.al [9] has performed experimentation for thermal performance factor using tube fitted with delta-winglet twisted tape. It has been concluded that the O-DWT is more effective turbulator giving higher heat transfer coefficient than the S-DWT. Pongjet Promvonge et.al [10] has performed experimental investigation of heat transfer and friction factor of a circular tube having wire coils in conjunction with twisted tapes and it has been observed that the heat transfer rate increases by decreasing twist and pitch ratios.

Referring to the existing literature, it has been observed that a lot of work has been done in the field of heat transfer enhancement by using many passive technique devices. Some of them has performed experimental and some have done numerical analysis, but

the combined study of experimental and numerical analysis is done limited. So, the present work focuses on numerical and experimental simulation of heat transfer rate and friction factor in a circular shape receiver tube equipped with twisted tape inserts.

NOMENCLATURE

Q	Heat transfer rate (KJ/sec)
Cp	*Specific heat of water {J/ (kg. °C)}*
m	Mass flow rate (Kg/sec)
$A_{C/S}$	Area of cross-section (m²)
D_H	Hydraulic Diameter (m)
L	Length of Receiver tube (m)
K	Thermal conductivity {W/ (m.°C)}
Δh	Head loss (mm)
l	Pitch of the tape (m)
w	Width of the tape (m)
Nu	Nusselt Number
Pr	Prandtl Number
f	Friction factor
Re	Reynold Number
ηth	Thermal Performance Factor

Greek letter

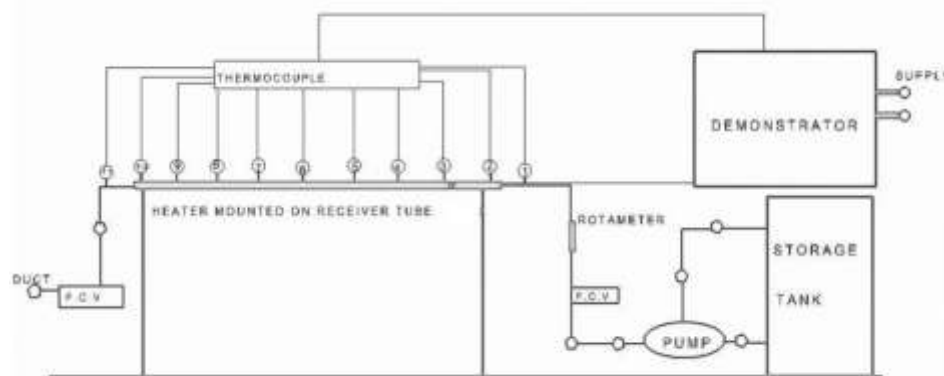|	
ρ	Density *(kg/m³)*
ϑ	Kinematic Viscosity (m2/s)
μ	Dynamic viscosity *(W/mK)*

II. EXPERIMENTAL APPARATUS

Figure 1: Schematic diagram of an experimental setup

Above figure shows a schematic view of an experimental set up in which circular geometry of receiver tube is used. Water is flowing through circular tube having length and diameter of 1000 mm and 19 mm respectively. Cold-water storage tank was used to pass the fluid flow from the receiver tubes. Flow rate varies from 2lpm to 10lpm, and to control the fluid flow rate inside the tube rotameter is used. The heater is mounted on receiver tube which can be started by applying current and voltage on the demonstrator. The hot water flows from the outlet of the tube. Thermocouples are placed at entry, exit and nine thermocouples are placed on the surface of tube to measure the inlet, outlet and wall temperatures. Experiments were performed for plain tube and for tube having twisted tape inserts within the Reynolds number range from 3193.38 to 15962.2. U-tube manometer is used to determine friction factor by means of pressure difference at the hot water side.
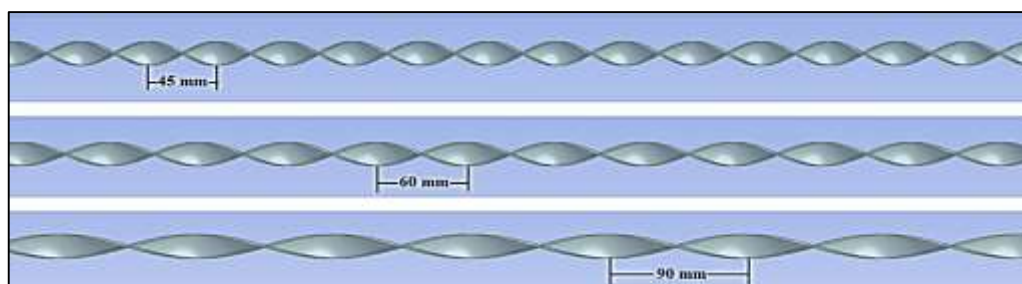

Figure 2: Geometry of twisted tape inserts having 45 mm, 60 mm and 90 mm pitch

Above figure show geometry of twisted tape inserts having 45 mm, 60 mm and 90 mm pitch used for heat transfer enhancement. The twisted tape used are manufactured from aluminum strip with 2 mm thickness, 18 mm width and 1000 mm length. The tapes are inserted in a tube at three different twist ratio(y) of (2, 2.67 and 4) having 45 mm, 60 mm and 90 mm pitch. To enhance the heat transfer rate, twisted tape creates turbulence in a tube which increases the tangential flow established by the inserts.

2.1 DATA REDUCTION

The mean Nusselt number and Friction factor are based on the inside diameter of the plain tube.

Nusselt Number:
Experimentally Nusselt Number is calculated as,

$$Nu_{exp} = \frac{H_W \times D_H}{K}$$

Nusselt Number from Dittus-Bolter correlation

$$Nu = 0.023 \times Re^{0.8} \times Pr^{0.4}$$

Friction factor (f)
Experimentally Friction factor is calculated as,

$$f = \frac{\Delta h \times 2 \times g \times D_H}{L \times V^2}$$

Friction factor petukhov correlation for plain tube

$$f = \frac{1}{(0.79 \, In \, Re - 1.64)^2}$$

III. CFD MODELLING

CFD analysis was carried out in ANSYS WORKBENCH 18.2 to predict the actual behaviour of the fluid flow inside the tube using twisted tape inserts. CFD was carried out to solve the PDE using the governing equations. Using digital computers, these governing equations are solved.

In the present study standard k-€ model and SST-k-ω model are selected for the simulation purpose. The governing equation set for both models are [11],

Standard k-€ model

$$\frac{\partial}{\partial t}(\dot{\rho}k) + \frac{\partial}{\partial xj}(\rho kui) = \frac{\partial}{\partial xj}\left[\left(ч + \frac{чt}{6k}\right)\frac{\partial k}{\partial xj}\right] + Gk - \beta1k\omega$$

$$\frac{\partial}{\partial t}(\dot{\rho}\omega) + \frac{\partial}{\partial xj}(\rho\omega ui) = \frac{\partial}{\partial xj}\left[\left(ч + \frac{чt}{6\omega}\right)\frac{\partial \omega}{\partial xj}\right] + G\omega - \beta2k\omega^2$$

SST-k-ω model

Turbulent Kinetic Energy:

$$\frac{\partial k}{\partial t} + uj\frac{\partial \omega}{\partial xj} = Pk - \beta^*k\omega + \frac{\partial}{\partial xj}[(v + \sigma kvt)\frac{\partial k}{\partial xj}]$$

Specific Dissipation Rate:

$$\frac{\partial \omega}{\partial t} + uj\frac{\partial \omega}{\partial xj} = \alpha s^2 - \beta\omega^2 + \frac{\partial}{\partial xj}\left[(v + \sigma\omega vt)\frac{\partial \omega}{\partial xj}\right] + 2[(1 - F1)\sigma\omega^2\frac{1}{\omega}\frac{\partial k}{\partial xi}\frac{\partial \omega}{\partial xi}]$$

Based on the above model used, SST-k-ω model is selected for the further studies as this model is not overly sensitive to inlet boundary condition like the standard k-€ model. The utilization of a k-ω formulation in the interior regions of the boundary layer makes the model directly usable all the way down to the wall through the viscous sub-layer, hence the SST-k-ω model can be used as a Low-Re turbulence model, without any extra damping functions. Literature shows that this model provides more accurate prediction of flow separation over another RANS model.

Here to resolve the link between velocity and pressure field, SIMPLE algorithm is chosen under the solution scheme. The residual value is set to 10^{-6} for all the variables in the present simulation. Here when the addition of normalized residuals values was below the set value of residuals, then the solution is set to be converged.

3.1 DATA REDUCTION

During CFD investigation the value of wall shear stress is determined at the outlet of the tube. Based on that, the value of friction factor is calculated at each flow rate for a turbulent flow, the friction factor for a turbulent flow in a tube is given by [12],

$$f = \frac{8\tau_w}{\rho v^2}$$

The Nusselt's number in turbulent flow is related to the friction factor through **the Chilton- Colburn analogy** expressed as,

$$Nu = 0.125 * f * Re * Pr^{\frac{1}{3}} \ldots\ldots \text{(Turbulent flow)}$$

Above equations are used for calculation of Nusselt's number and friction factor for the numerical simulation of plain tube and the tube equipped with twisted tape inserts.

3.2 CFD RESULTS

[A] RESULTS FOR PLAIN CIRCULAR TUBE

Above figure 3 shows the CFD results for a plain circular tube using standard k-€ model and SST-k-ω model. Fig 4 shows the comparison of these results with the Dittus-Boelter correlation (Theoretical).

A   B
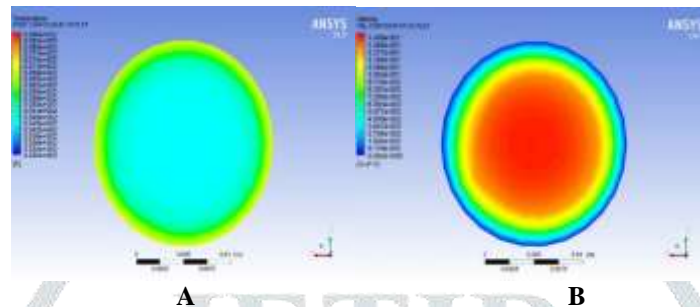
Figure 3: Figures A & B above shows the temperature & velocity contour for plain circular tube
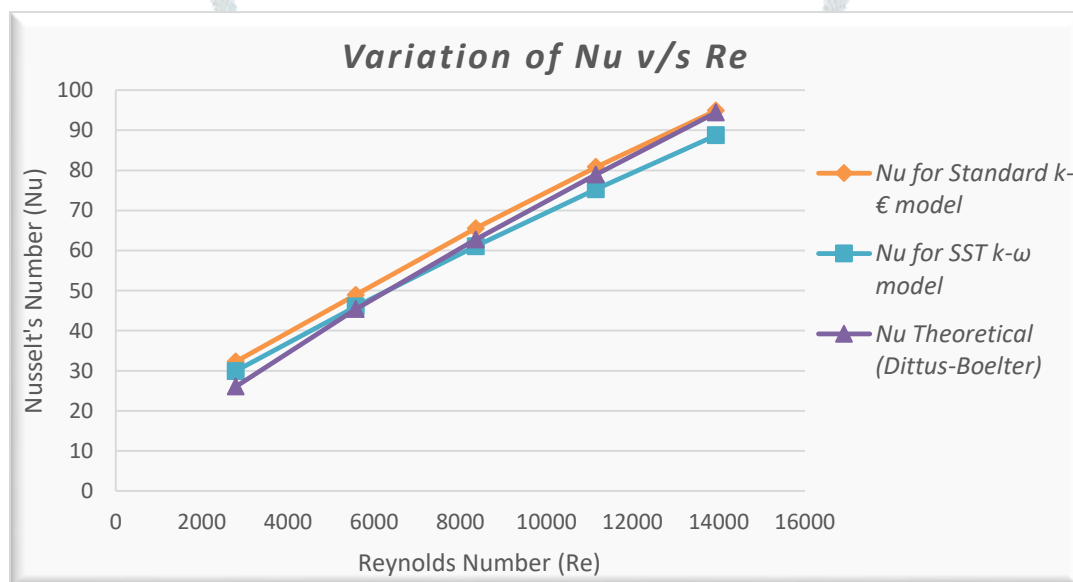

Figure 4: Variation of *Nu v/s Re* for plain circular tube using SST k-ω, Standard k-€ model and Dittus-Boelter correlation

Above figure shows that SST-k-ω model are in better agreement with the Dittus-Boelter equation hence this model is selected for performing the further simulations of heat transfer enhancement. Above figure shows that the readings obtained by CFD and Dittus-Boelter equation varies ±14.78% for the Nusselt's number.

[B] RESULTS FOR TUBE EQUIPPED WITH PLAIN TWISTED TAPE INSERTS

Above figure 5 shows the CFD results for a tube equipped with plain twisted tape inserts having 45 mm, 60 mm and 90 mm pitch using SST-k-ω model. Figures 6, 7 & 8 shows the comparison of these results with the plain tube.
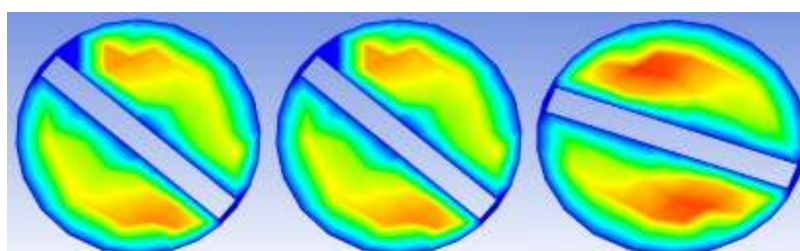
A   B   C

Figure 5: Figures A, B, C shows the velocity contours of 45 mm,60mm and 90 mm pitch twisted tape using SST-k-ω model

Plots for the twisted tape having 45mm, 60mm and 90mm pitch:

Results obtained from twisted tape inserts having 45mm, 60mm and 90mm pitch is compared with plain tube as shown below. Thermal performance factor is also obtained for twisted tape inserts having 45mm, 60mm and 90mm pitch.
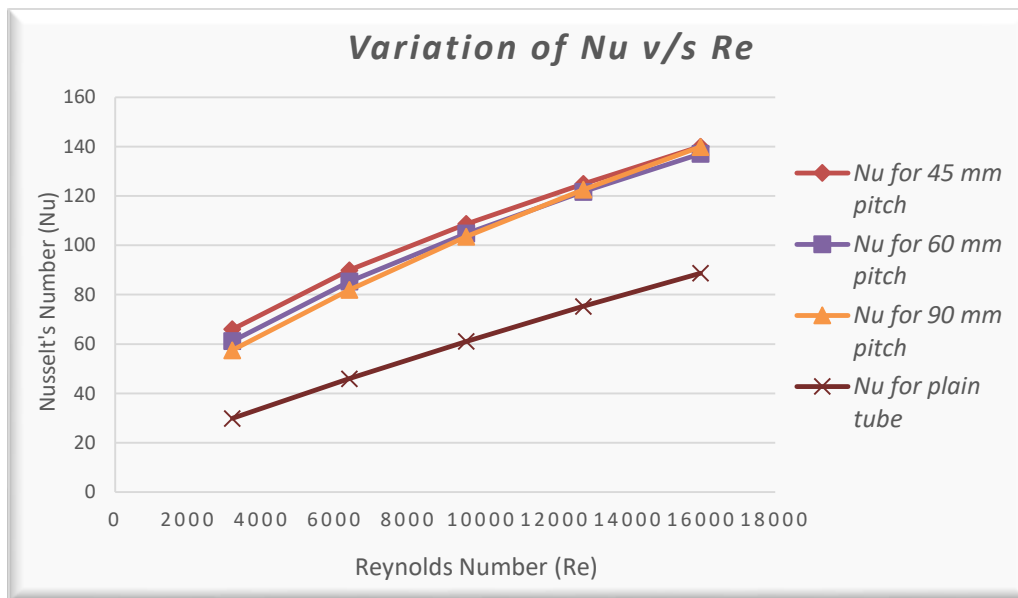

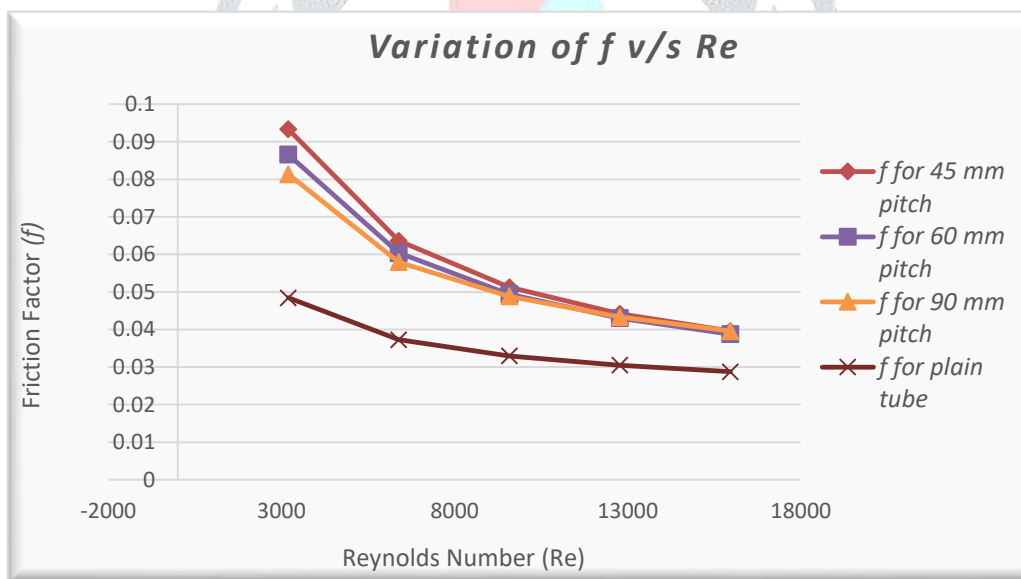**Figure 6: Variation of *Nu v/s Re* of classical twisted tapes with plain tube**

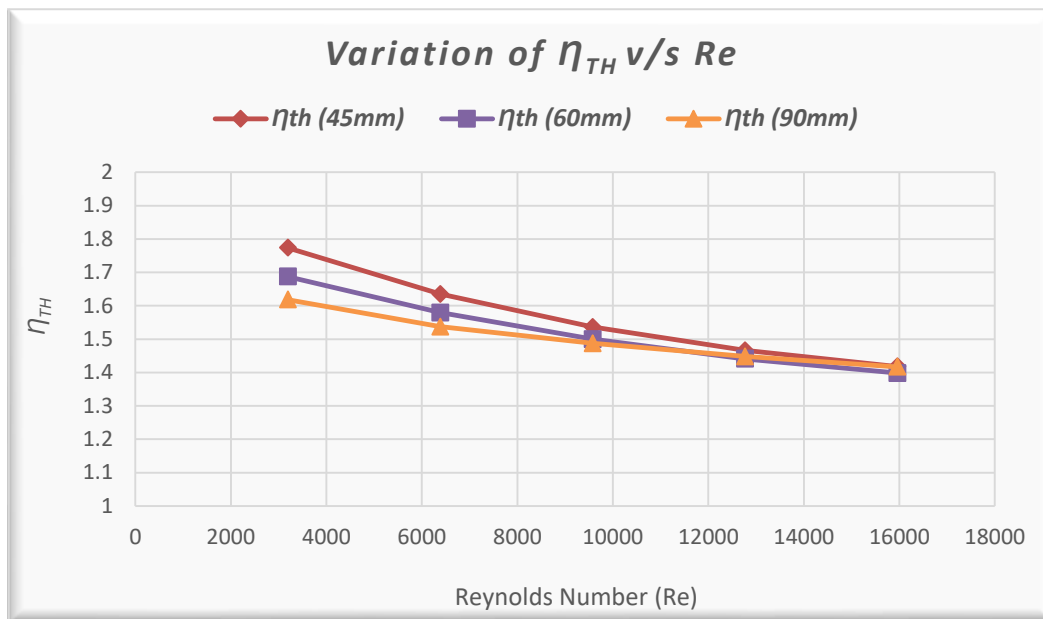**Figure 7: Variation of *f v/s Re* of classical twisted tapes with plain tube**

Figure 8: Variation of thermal performance factor for twisted tapes of 45, 60 and 90 mm pitch

3.3 RESULTS AND DISCUSSION

CFD results revealed that the Nusselt's number increases by 43.14%, 40.99% & 40.44% for the 45 mm, 60mm & 90mm pitch twisted tapes with respect to plain tube, simultaneously the Friction factor increased by 1.9 times, 1.8 times & 1.7 times for 45, 60, and 90 mm tapes and the thermal performance factor is determined, and the maximum performance is obtained for 45 mm pitch twisted tape insert, which is 1.7736 for the Reynolds number ranging from 3193 to 15962. As 45mm pitch twisted tape insert shows better heat transfer enhancement, it is selected for the further simulations.

IV. EXPERIMENTAL RESULTS

[A] RESULTS FOR PLAIN CIRCULAR TUBE

Above figure 9 shows the experimental result for a plain circular tube which is validated in terms of Nusselt's number and comparison of this result with the Dittus-Boelter correlation (Theoretical).
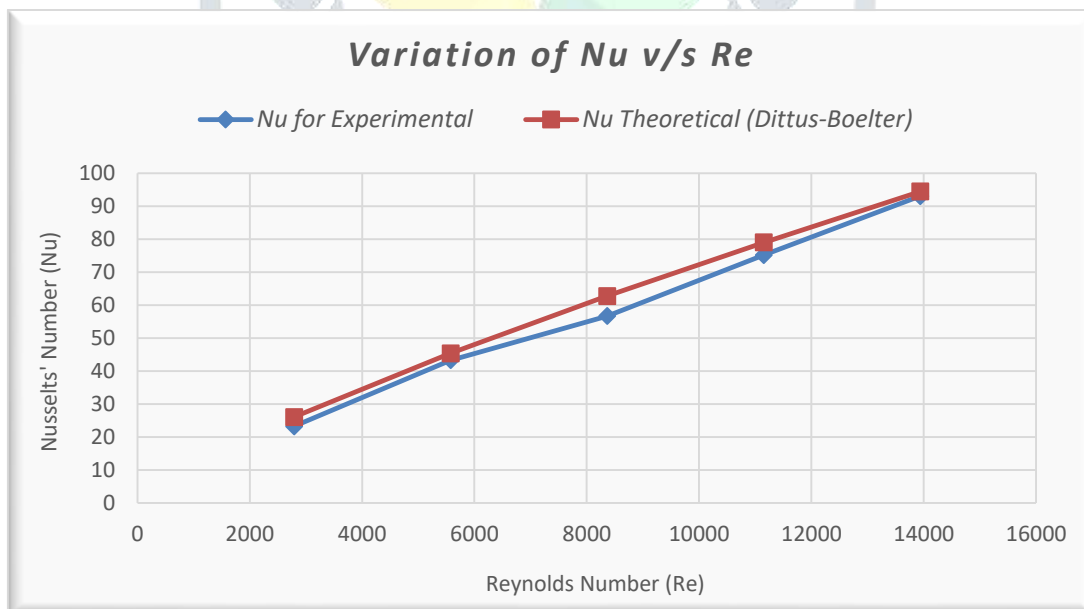

Figure 9: Variation of Nu v/s Re for the plain tube (Expt. And Dittus-Boelter)

Above figure shows that the readings obtained by experiment and Dittus-Boelter equation varies ±12. 04% for the Nusselt's number.

[B] RESULTS FOR TUBE EQUIPPED WITH PLAIN TWISTED TAPE INSERTS

From CFD analysis, it has been observed that 45 mm pitch twisted tape insert shows better results than 60mm and 90mm pitch. Therefore, for experimentation only 45mm pitch twisted tape insert is selected. Below figure shows the experimental results for a tube equipped with 45 mm pitch twisted tape inserts and comparison of these results with the plain tube.
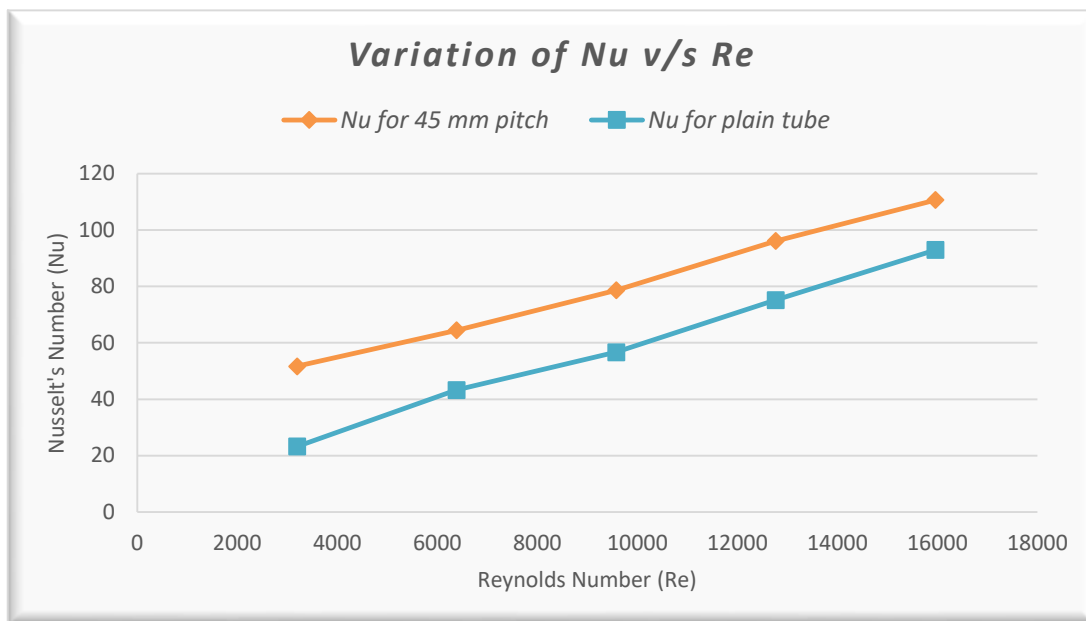

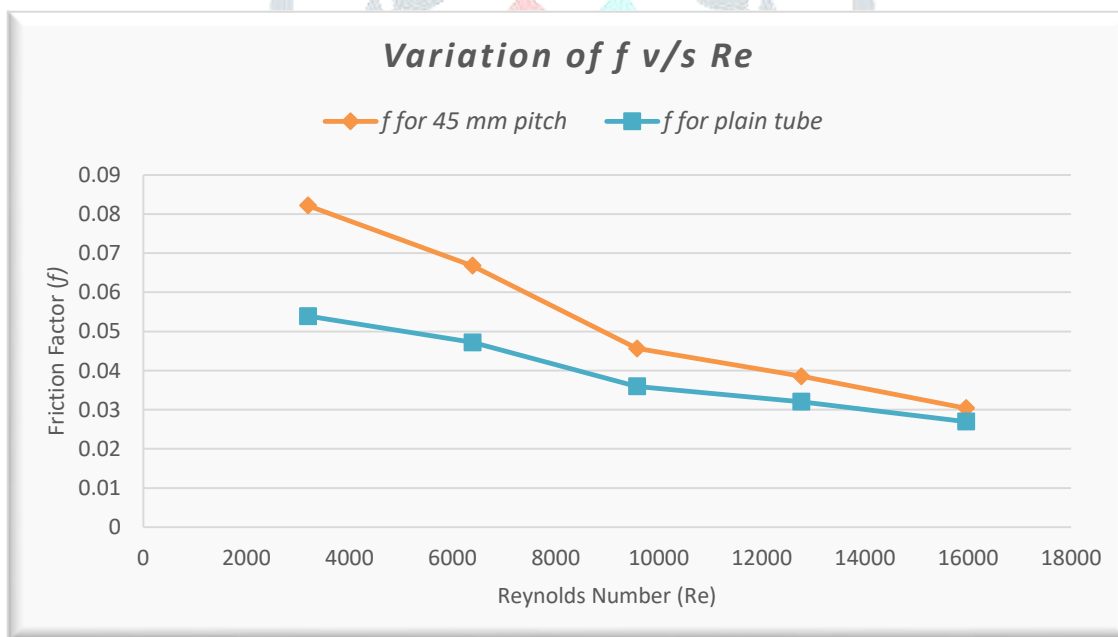**Figure 10: Variation of Nu v/s Re of 45 mm pitch twisted tape insert with plain tube**

Figure 11: Variation of f v/s Re of 45 mm pitch twisted tape insert with plain tube
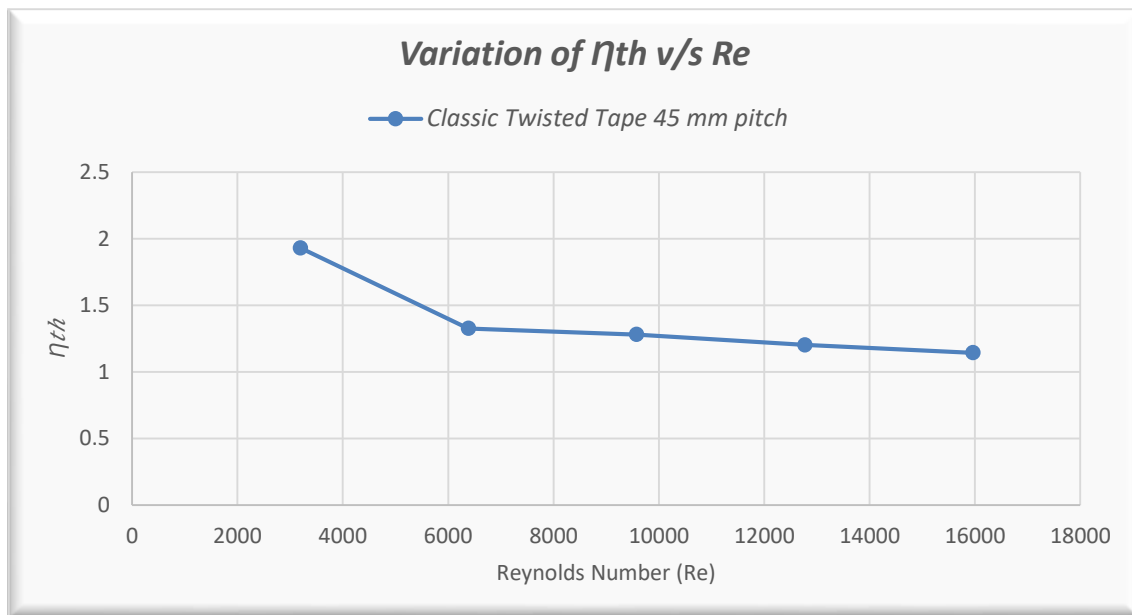
Figure 12: Variation of thermal performance factor v/s Re for 45mm pitch twisted tape

4.1 RESULTS AND DISCUSSION

Experimental results revealed that the Nusselt's number increases by 37.84% for 45 mm pitch twisted tape when compared with that of the plain tube, simultaneously the friction factor for 45 mm pitch tape is increased by 1.52 times as compared with the plain tube and the thermal performance factor is determined for a plain twisted tape and the maximum thermal performance is obtained from 45 mm pitch twisted tape which is 1.9315 for the Reynolds number ranging from 3193 to 15962.

V. CONCLUSIONS

In the present paper both the CFD and Experimental investigation of heat transfer, friction factor and thermal performance factor of a circular receiver tube is done. By using CFD analysis, the maximum heat transfer rate is obtained by using 45 mm pitch twisted tape insert at an expense of increase in friction factor by 1.9 times as compared with that of the plain tube within the Reynolds number set between 3193 to 15962. The maximum thermal performance factor is 1.7736 for 45 mm pitch twisted tape insert. By using Experimental analysis, the maximum heat transfer rate is obtained by using 45 mm pitch twisted tape insert at an expense of increase in friction factor by 1.52 times as compared with that of the plain tube within the Reynolds number set between 3193 to 15962. The maximum thermal performance factor is 1.9315 for 45 mm pitch twisted tape insert.

VI. ACKNOWLEDGMENT

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