NUMERICAL ANALYSIS FOR EFFECT OF DIFFERENT LENGTH COMBINATION OF TURBULATORS ON HEAT TRANSFER AND PRESSURE DROP IN DOUBLE PIPE HEAT EXCHNAGER

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Abstract - *This article is mainly deals with the use of swirl flow devices with different combinations as passive heat-transfer augmentation technique. In this article, the two different swirl flow devices used are namely twisted tape (TT) and wire coil (WC) turbulator. The present work deals with the counter flow type condition of heat exchanger. Effect of different length combination of these two different turbulator twisted tape and wire coil on the heat transfer, friction factor and pressure drop for Reynolds number ranges from 2000-10000, has studied in double pipe heat exchanger (single pass). There is constant twist ratio 'y' (H/di) 3 of twisted tape and with that combine the wire coil of wire diameter (e) 1 mm and coil diameter (dw) of 23.8 mm with constant pitch (p) 10 mm in all combination and the results are analyzed. The combinations of twisted tape and wire coil are like 25-75 %, 30-70 %, 35-65 %, 40-60 %, 45-55 %, of the full length of heat exchanger respectively were analyzed in the Reynolds number range 2000-10000. The inner pipe inner diameter and outer thickness of the pipe is 25 mm and 1.5 mm respectively. The outer pipe inner diameter and outer thickness of the pipe is 52 mm and 1.5 mm respectively. In this analysis, water used as working fluid in both side inner side and annulus side also. Inserts has inserted in the inner pipe, which increases the heat transfer coefficient and Nusselt number compare to the plain tube. The friction factor is also increased compare to the plain tube. Simulations has carried out using software and the results obtained from the simulation has validated by comparing with the standard results published by the other journals research paper.*

Key Words: Heat transfer enhancement, heat exchanger, twisted tape, wire coil insert, turbulator, passive techniques for heat transfer enhancement.

1. INTRODUCTION

In many engineering applications the good thermal performance of the heat exchanger or thermal systems are needed and for that various types of methods are developed and extensively used to enhance the heat transfer in the system. There are any methods for enhancing the heat transfer from the fluid in convectional heat exchanger by using various augmentation techniques. These techniques enhance the heat transfer rate by creating the following conditions: 1) increased the heat transfer area, 2.) generating of the swirling and/or secondary flows, and 3.) interruption of boundary layer development and rising the degree of turbulence in the flowing fluid. In past, several studies focused on the passive heat-transfer enhancement methods. In passive methods. The swirl flow devices or can say turbulators has inserted into the pipe through which the fluid is flowing to provide an interruption of the boundary layer development, to increase the heat-transfer surface area and to cause enhancement of heat-transfer by increasing the turbulence intensity or the fast and better mixing of the fluid. [1].

Turbulators in the various shapes have proposed. There are many literature reviews are available on using the twisted tape (TT) and for the wire coil turbulators, but from their conclusions is not so high, so that some researchers maybe cannot decide which kind of twisted tape or wire coil and/or combinations of these two turbulators can used for the future work. But this problem can be solved in this analysis to decide which kind of the combination will be select for the achievement of high heat transfer co efficient and higher thermal performance of the heat exchanger. For the last some decades lot of researches with the twisted tape and wire coil devices employed for the augmentation of laminar and turbulent flow heat transfer and the results are discussed [2]. Twisted tape as heat-transfer enhancement devices is another group are widely applied for producing compact heat exchanger and increasing the thermal performance of the existing heat exchanger due to its low cost and ease of manufacture installation [3-6]. Durmus [7] had studied the heat transfer and pressure drop in a heat-exchanger tube fitted with the cut out conical turbulators with different installing angles. Kalczak [8] had studied experimentally the heat transfer for laminar flow of water in an air-cooled copper pipe with twisted tape inserts for various value of pitch. Kiml [9-10] et al. had investigated the effect of elliptical ring (angled/transverse ribs) on the flow structure and circumferential heat-transfer distribution. A Comparison of the thermal and hydraulic performances of twisted tape or wire coil inserts was introduced by Wang and Sunden [11] for both laminar and turbulent flow regions. Sivashanmugam et al.[12] had investigated the heat-transfer and friction factor characteristics of circular tube fitted with right-left helical screw tape inserts of equal length, and unequal length of different twist ratios. They concluded that the heat-transfer enhancement for right-left helical screw tape inserts is higher than that for straight helical twist due to effect of repeated left-right movement of fluid during course of flow through tube attached with left-right twist tape providing efficient mixing in the radial direction.

NOMENCLATURE

A_i	Area of inner pipe, m^2
d_i	(inner pipe) inner diameter, mm
d_o	(inner pipe) outer diameter, mm
D_i	(Outer pipe) inner diameter, mm
D_o	(Outer pipe) outer diameter, mm
D_h	Hydraulic diameter, mm
f	Friction factor
h	Heat transfer co-efficient, W/m^2-K
k	Thermal conductivity, W/m-K
L	Tube length, m
Nu	Nusselt number
Pr	Prandtl number
ΔP	Pressure drop
Q_{avg}	Average heat transfer rate, W
Re	Reynolds number
ΔT_{lmtd}	Logarithmic mean temperature difference, °C
U	Overall heat transfer coefficient, W/m^2-K
V	Velocity, m/s
w	Width of twisted tape, mm
H	Pitch length based on 180°, mm
y	Twist ratio, (dimensionless)
e	Wire Diameter, mm
dw	Coil Diameter, mm
p	Pitch of wire coil, mm
c_p	Specific heat, k-cal/kg-K
h_a	Annulus side heat transfer co-efficient, W/m^2-K
h_i	Inner pipe heat transfer co-efficient, w/m^2-K
m	Mass-flow rate, kg/s
T_h	Hot water temperature,° c
T_c	Cold water temperature,°c

SUBSCRIPT

I	Inner
O	Outer
h	Hydraulic/hot water side
c	Cold water side
p	Plain tube
avg	Average
lmtd	Logarithmic mean temperature difference
b	Bulk temperature

GREEK SYMBOLS

ρ	Fluid Density, kg/m^3
δ	Thickness of twisted tape, mm
μ	Fluid Dynamic viscosity, kg/m-s
η	Thermal performance factor
ϑ	Kinematic viscosity of fluid, m^2/s

Wire coil turbulator are the another type of the turbulator which are used in the heat transfer devices for the enhancement of the heat transfer from the fluid. Garcia et.al. [13]had studied experimentally the thermo-hydraulic behaviour of round tube with helical-wire coil inserts in laminar, transition and turbulent regimes at different Prandtl numbers. They reported that within the transition region, by fitting the wire-coils inside a smooth tube heat exchanger, the heat-transfer rate increased up to 200%. Prof.Shashank S. Choudhari et al.[14]had studied the heat transfer characteristics and friction factor of horizontal double pipe heat-exchanger with coil wire inserts made up of different materials. Behabadi et al. [15] studied experimentally heat-transfer augmentation, pressure-drop by wire coil inserts at the time of heating the engine oil inside a horizontal tube and they developed two empirical correlation within an error band of ± 20%. S.Biswas et.al.[16]had studied experimentally for measuring tube side heat transfer co-efficient, friction factor, heat-transfer enhancement efficiency of air for turbulent flow in a circular tube-fitted with wire-coil insert. They reported that at comparable Nusselt numbers in tube with wire-coil insert enhanced by 1.5 to 2.3 times at the cost of increase of friction factors by 3 to 3.5 times compared to that of smooth tube. Gunes et.al.[17]had studied experimentally with coiled wire inserted into the tube in the turbulent regime. They used equilateral triangle cross-sectioned wire coil inserts and reported that at low Reynolds number regime were the best operating regime for wire-coil insert. C.Nithiyesh Kumar et al.[18]had studied different heat transfer augmentation techniques used to increase rate of heat-transfer without affecting much the overall performance of the system. They mainly focuses on the twisted tape heat-transfer enhancement and its design modification towards the enhancement of heat-transfer and saving the pumping power. V. Kongkaitpaiboon et al.[19]had performed experimental studied of convective heat-transfer and pressure loss in a round tube fitted with circular ring turbulators. They reported the effect of circular-ring turbulator (CRT) on the heat-transfer and fluid-friction characteristics in heat exchanger tube. Results revealed that heat-transfer rates in the tube fitted with CRTs were augmented around 57% to 195% compared to that in the plain tube, depending upon operating conditions. A. E. Zohir et al.[20] had investigated the enhancement of heat transfer co-efficient by inserting the coiled-wire around the outer surface of the inner tube of double-pipe heat exchanger. They performed for the turbulent water flow in double pipe heat exchanger. They reported that heat transfer co efficient for turbulent flow increases for all coiled wire pitches, with the highest enhancement of about 450 % for counter flow and 400 % for the parallel flow.

2. Present CFD analysis facility and TT and WC Details:

Present analysis work had been carried out in the CFD fluent application. Geometry of the double pipe and the inserts (twisted tape and wire coil) has shown in figure below. Specifications of the heat exchanger and inserts used in this analysis are, the inner diameter of inner pipe (di) was 25 mm, outer thickness of inner pipe is 1.5 mm, inner diameter of the outer pipe are (Di) is 52 mm, outer thickness of the outer pipe is 1.5 mm. the length of the test section (L) is 1500 mm and the twisted tape and wire coil inserts bonded together has inserted into this heat-transfer test tube.
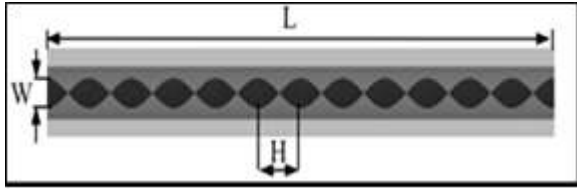

Fig 1. Typical twisted tape [2].

Fig 2. Twisted tape configuration

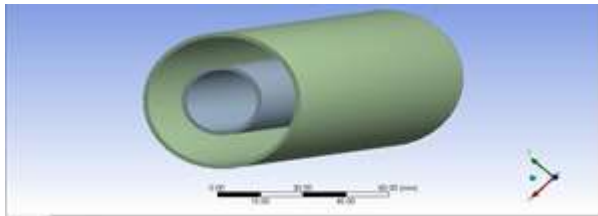**Fig 3. Wire coil turbulator [14].**

Fig 4. Double pipe geometry.

In this present analysis, five different combinations of the two different turbulators named twisted tape and wire-coil turbulators were used and results of heat transfer friction factor and the thermal performance factor for each combination of twisted tape and wire coil was reported. These all five different combinations of twisted tape and wire-coil has analysed with the specified boundary conditions in the heat exchanger. The technical detail of the present analysis work given in table 1 below:

Table 1 : Technical detail of the present work analysis and test conditions

SR.NO	NAME	SPECIFCATION / MATRIAL
A.) Detail of the double pipe heat exchanger		
1	Inner Pipe(Inner Dia) (d_i)	25 mm
2	Inner Pipe(Outer Dia) (d_o)	26.5 mm
3	Outer Pipe(Inner Dia) (D_i)	52 mm
4	Outer Pipe(Outer Dia) (D_o)	53.5 mm
5	Test tube length	1500 mm
6	Material of the inner pipe	copper
7	Material of the outer pipe	aluminium
B.) Twisted tape		**copper**
8	Width (w)	23.8 mm
9	Thickness (δ)	1 mm

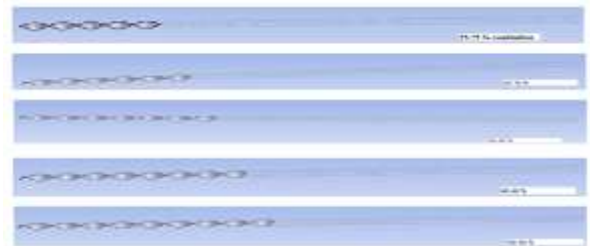| 10	180° Tape pitch length (H)	75 mm
C.) wire coil		**copper**
11	wire diameter (e)	1 mm
12	coil diameter (d_w)	23.8 mm
D.) Test condition		
13	Reynolds number	2000-10000
14	Hot water inlet temperature	45 °C
15	Cold water inlet temperature	25 ° C

Fig 5 Different combinations of twisted tape and wire coil.

The five different combinations are 25-75%, 30-70%, 35-65%, 40-60%, and 45-55% length of tube of twisted tape and wire coil respectively has shown in figure Above.

Boundary conditions used in the analysis are tabulated in the below table 2.

Table 2: Boundary conditions

	Boundary Condition Type	Mass flow rate value (Kg/s)	Temperature (°C)
Inner Inlet	Mass-flow inlet	0.03,0.04,0.06, 0.08,0.1	45
Inner Outlet	Pressure Outlet	-	-
Outer Inlet	mass-flow inlet	0.166	25
Outer Outlet	Pressure Outlet	-	-

2.1 Solution Methods:

1. Scheme = Simple.
2. Gradient = Least Square Cell Based.
3. Pressure = Second Order.
4. Momentum = Second Order Upwind.
5. Turbulent Kinetic Energy = Second Order Upwind.
6. Turbulent Dissipation Rate = Second Order Upwind.
7. Energy = Second Order Upwind.

2.2 Solution Control and initialization:

1. Pressure = 0.3 Pascal
2. Density = 1 kg/m3
3. Body forces = 1 kg/m2s2
4. Momentum = 0.7 kg-m/s
5. Turbulent kinetic energy = 0.8 m2/s2
6. Turbulent dissipation rate = 0.8 m2/s3
7. Turbulent viscosity = 1
8. Energy = 0.95

2.3 Measure of convergence :

It is tried to have a nice convergence throughout the simulation and hence criteria are made strict, so to get an accurate result. For this reason, residuals given as per the table 3 that follows.

Table 3: Residuals

Variable	Residual
x-velocity	0.001
y-velocity	0.001
z-velocity	0.001
Continuity	0.001
Specific dissipation energy/ dissipation energy	0.001
Turbulent kinetic energy	0.001
Energy	1.00E-06

2.4 Theoretical analysis / Data Reduction :

The data reduction of the measured result summarized as follows:

The average Nusselt number and the friction factor has based on the inner diameter of the test tube. Heat absorbed by the cold water in the annulus side, Q_w, c can be written by:

$$Q_{w,c} = m_c * C_p * (T_{c,out} - T_{c,in})$$

Where m_c is the mass flow rate of cold water; C_p, w is the specific heat of water; T_c, in and T_c, out are the inlet and outlet cold-water temperatures, respectively.

The heat supplied from the hot water, Q_h can be determined by:

$$Q_{w,h} = m_h * C_{p,h} * (T_{h,in} - T_{h,out})$$

Where m_h is the hot water mass flow rate; T_h, in and T_h, out are the inlet and outlet hot water temperatures, respectively. The heat supplied by the hot fluid in the test tube has found to be 3 % to 8 % difference of the heat absorbed by the cold fluid for the thermal equilibrium due to the convection and radiation heat-losses from the test-section to the

surroundings. Thus, the average value of heat transfer rate, supplied and absorbed by both fluids, has taken for internal convective heat-transfer coefficient calculation.

$$Q_{avg} = \frac{Q_{w,c} + Q_{w,h}}{2}$$

For fluid flows in a concentric tube heat exchanger, the heat transfer co-efficient, h_i calculated from:

$$Q_{avg} = U_i * A_i * \Delta T_{LMTD}$$

Where,

$$A_i = \Pi * d_i * L$$

From this we can calculate the overall heat transfer co efficient U_i,

$$U_i = \frac{Q_{avg}}{A_i * \Delta T_{LMTD}}$$

The annulus side heat-transfer coefficient (h_a) calculated using,

$$\frac{h_a * D_h}{K} = 0.023 * \mathrm{Re}^{0.8} * \mathrm{Pr}^{0.4}$$

The tube side heat-transfer coefficient is determined using,

Thus,

$$Nu_i = \frac{(h_i * d_i)}{K}$$
$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{1}{h_a}$$

Friction factor,

$$f = \frac{\Delta P}{\left[\left(\frac{1}{d_i} \right) * \left(\frac{\rho V^2}{2} \right) \right]}$$

For a constant pumping power,

$$(V * \Delta P)_p = (V * \Delta P)_t$$

Where index t indicates twisted tape and index p indicates plain tube.

The relation between Reynolds number and friction factor can written as below:

$$(f * \mathrm{Re}^3)_p = (f * \mathrm{Re}^3)_t$$

$$(\mathrm{Re})_P = (\mathrm{Re})_t * \left(\frac{f_t}{f_p} \right)^{1/3}$$

The thermal enhancement factor (η), defined as the ratio of the, h_a of an augmented surface to that of a smooth surface, h_p, at a constant pumping power written as:

$$\eta = \left(\frac{h_a}{h_p}\right)_{pp} = \left(\frac{Nu_a}{Nu_P}\right)_{pp} = \frac{\left(\dfrac{Nu_a}{Nu_P}\right)}{\left(\dfrac{f_a}{f_P}\right)^{1/3}}$$

3. Result and Discussion:

3.1 Validation of plain tube:

Prior to the actual data collecting, and the confidence of the data simulated and the simulation method was evaluated by comparing the data obtained from the present simulated results with those calculated from the standard correlations including the two equations for the Nusselt number and friction factor given below.

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

$$f = 0.6165 * \mathrm{Re}^{-0.317}$$

The mean absolute percentage deviation (or error) of the present simulated Nusselt number is 2.78 % from the value predicted from the Nusselt number equation, and the average absolute percentage deviation (or error) of the present simulated friction factor is 9.7% from the value predicted by friction factor equation. This shows that a sufficient accuracy of the simulated data obtained from the present simulation.
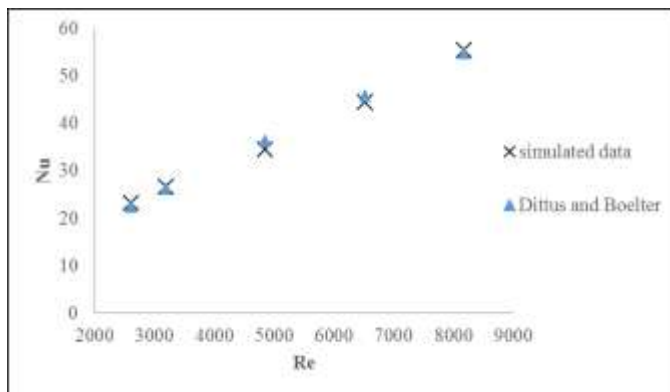
Fig 6. Validation of plain tube-Nusselt number.
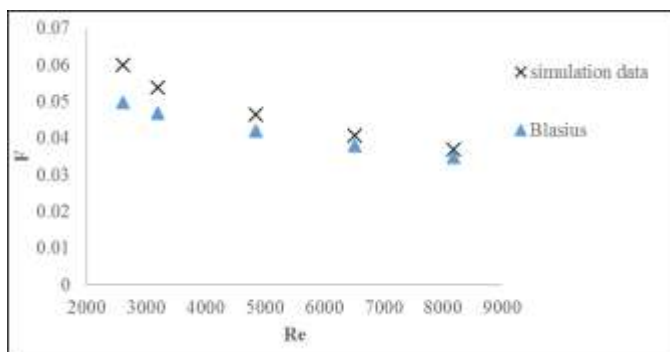
Fig 7. Validation of plain Tube-Friction factor.

3.2 Hot water inlet from twisted tape side:

3.2.1 Nusselt number result (hot water inlet from twisted tape side):-

Heat transfer in terms of the Nusselt number depicted in figure 8. It is clear from the figure that Nusselt number increasing with the increasing of the Reynolds number. Figure 8, shows the result of Nusselt number for the five different combination of turbulators that is 25-75 %, 30-70 %, 35-65 %, 40-60 %, and 45-55 % and the hot fluid inlet is from the twisted tape side. Nusselt number result of the smooth tube also presented in the figure for the comparison purpose. From that, we can say that heat transfer from the fluid increasing as the Reynolds number increasing.

As shown in the figure 8, Nusselt number is increasing in all cases that is with the different combinations of the turbulators inserts and the heat exchanger without inserts (plain tube). However, as shown in figure the 25-75 % combination of turbulator gives the high value of Nusselt number this combination gives higher heat transfer compare to the other combinations and plain tube (without inserts).
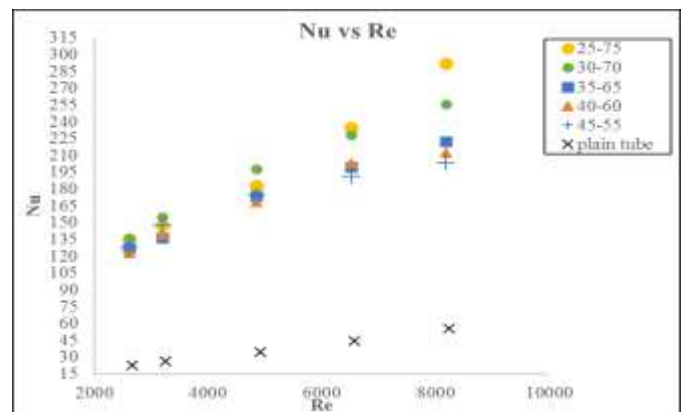

Fig 8. Variation of Nusselt number with Reynolds number.

Heat transfer ratio that is the ratio of the augmented Nusselt number (Nu_a) and the plain tube Nusselt number (Nu_p) {(Nu_a/Nu_p)}. The second parameter that heat transfer ratio (Nu_a/Nu_p) increases as the Reynolds number decreases. This has shown in figure 9 below.
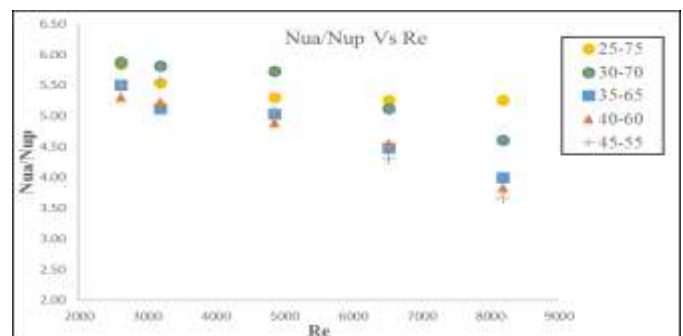

Fig 9. Variation of Nua/Nup with Reynolds number.

This has clarified that the thermal boundary becomes thicker as Reynolds number decreases and therefore provides an effective boundary layer destruction by the tapes. In order to provide a net energy gain, the dimensionless heat transfer ratio (Nua/Nup) must be greater than unity for all cases. Apparently, in the present all cases the heat transfer ratios are consistently higher than the unity. This verifies the beneficial gain of the all combination of the turbulators over the plain tube for heat transfer enhancement.

3.2.2   Friction factor result (hot water inlet from twisted tape side):-

Figure 10 shows the effect of different length combination of turbulators on the friction characteristics. The variation of the friction factor with the Reynolds number has shown in the figure below. The variation of the pressure drop has shown in terms of friction factor with Reynolds number. In the figure, it is apparent that the presence of the bonded combined twisted tape and wire coil turbulator increase in friction factor over the smooth (plain tube) tube. The increase in the friction factor due to the reverse/swirl turbulent flow is higher than that with the axial flow (plain tube). This has attributed to the dissipation of dynamic pressure of the fluid due to the higher surface area and the act caused by reverse/swirl flow.
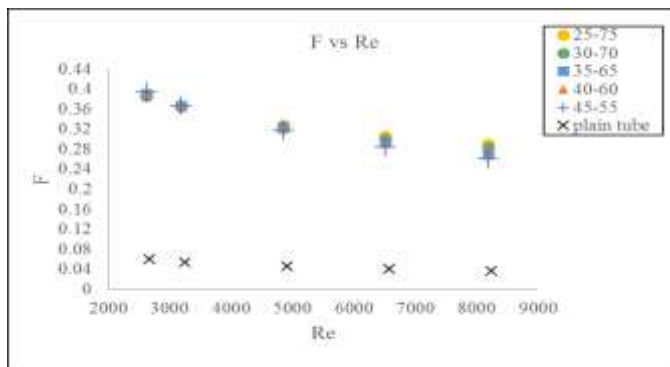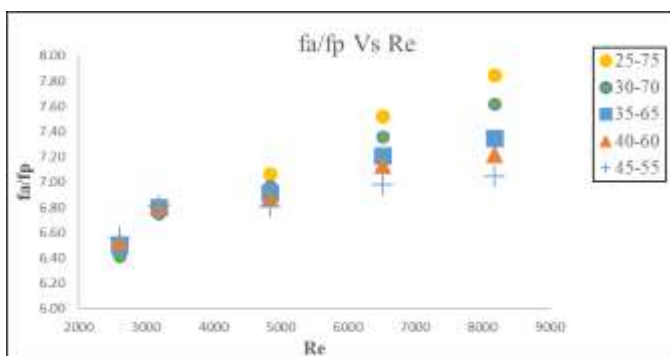

Fig 10. Variation of Friction factor with Reynolds

number.
Fig 11. Variation of fa/fp with Reynolds number

Figure 11 depicts the variation of the friction factor ratio (fa/fp) {which is the ratio of the friction factor when tube equipped with combined twisted tape and wire coil (fa) to the friction factor of plain tube (without turbulators)} with

the Reynolds number for the different combination of twisted tape and wire coil turbulators.

Opposite trend has observed in the case of the friction factor ratio when we are comparing it with the Nusselt number ratio. Figure shows that, Friction factor ratio (fa/fp) increases with the increase of Reynolds number.

3.3   Hot water inlet from wire-coil side:

3.3.1   Nusselt number result (hot water inlet from wire coil side):-

The effect of hot fluid inlet from the wire coil turbulator on the Nusselt number for the different length combination of the twisted tape and wire coil turbulators. Figure 12 shows the result of Nusselt number variation with the Reynolds number when the hot fluid inlet from the wire coil turbulators.

The trend of the graph is very much same as the result of Nusselt number variation in the case of the hot fluid inlet from the twisted tape turbulators. However, from the figure 12, observed that the result of the Nusselt number in the case of hot inlet from the wire coil is lower than hot inlet from the twisted tape.
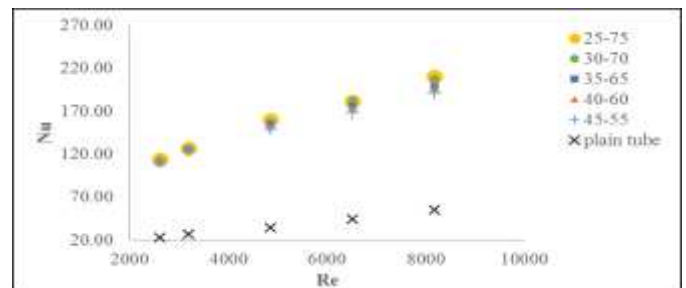

Fig 12. Variation of Nusselt number with Reynolds number.

Figure 12 shows that when the hot fluid inlet from the wire coil side it also shows the higher heat transfer result compare to the plain tube. Therefore, these two case of hot fluid inlet, first when the hot fluid is inlet from the twisted tape and second hot fluid is inlet from the wire coil shows better result compare to the plain tube (without turbulators).
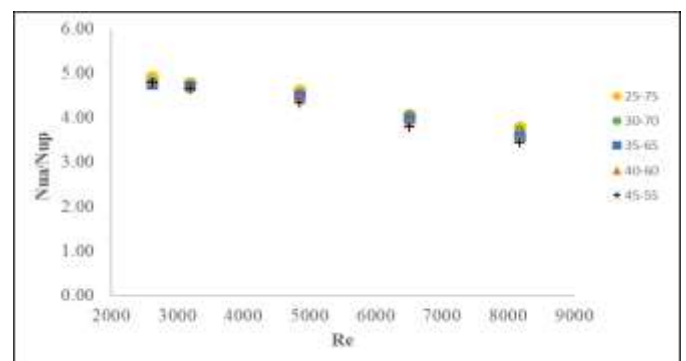

Fig 13. Variation of Nua/Nup with Reynolds number

The trend of the heat transfer ratio (Nua/Nup) for the case of hot water inlet from the wire coil turbulator has shown in the figure 13.

From the figure 13, it has observed that graph trend of the heat transfer ratio is follows same as heat transfer ratio in the case of hot water inlet from the twisted tape. However, in this second case when the hot water is entering the tube from the wire coil side the Nusselt number value is somewhat lower compare to the Nusselt number result when hot water enters from the twisted tape side. So from this Nusselt number results it concluded that heat transfer is higher in the of hot water inlet from twisted tape side.

Therefore, the conclusion is the hot fluid inlet from the twisted tape gives the better heat transfer result compare to the hot fluid inlet from the wire coil turbulator.

3.3.2 Friction factor result (hot water inlet from wire coil side):-

When the hot fluid inlet from the wire-coil turbulator friction factor is also decrease with the increasing of the Reynolds number as in the first case of hot fluid inlet from the twisted tape for the all-different length combination of twisted tape and wire coil turbulators.
From the figure 14, it observed that the friction factor result is very much similar to the friction factor result in the case of hot fluid inlet from the twisted tape.

As the variation in the Pressure drop has shown in terms of friction factor, we conclude that there is no larger difference in pressure drop in both different hot fluid inlet condition.
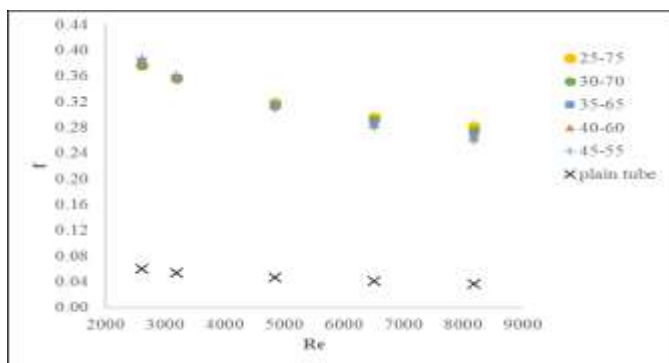

Fig 14. Variation of Friction factor with Reynolds number.

Pressure drop has shown in terms of friction factor, we conclude that there is no larger difference in pressure drop in both different hot fluid inlet condition.

Figure 15 shows the result of the friction factor ratio (fa/fp) with the Reynolds number for the different combination of twisted tape and wire coil turbulators for the hot water inlet from the wire coil turbulator.
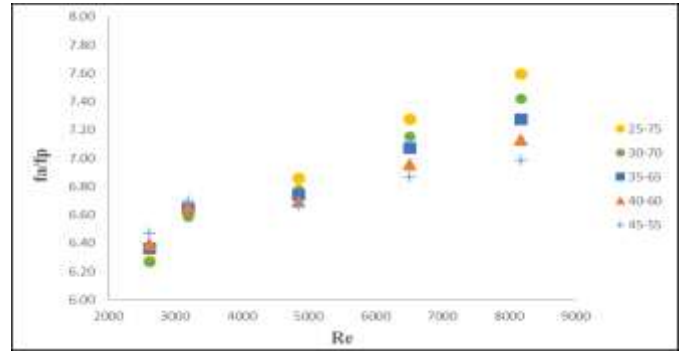

Fig 15. Variation of fa/fp with Reynolds number.

The trend of this graph is very much similar to the figure 11, shows the result of friction factor ratio when hot water inlet from the twisted tape side. The difference between these two results is not very large, so we can consider that the hot water inlet from the twisted tape has good agreement when we are considering the heat-transfer and friction factor.

3.4 Thermal performance factor:

3.4.1 Thermal performance factor (hot water inlet from twisted tape):-

The thermal performance factor is a most significant parameter in order to determine potential of a combined twisted tape and wire coil turbulators for the industrial applications. The thermal performance factor (η) based on the constant pumping power criteria was calculated by Eq.(14) in order to evaluate the effectiveness of the tube with different combination of two turbulators (i.e. twisted tape and wire coil).

Figure 16 shows the relation between the Reynolds number (Re) and thermal performance factor (η) for different twisted tape and wire coil length combination inserted into the tube. Obviously, thermal performance factor tends to decrease with the increasing of the Reynolds number in all different combined cases.
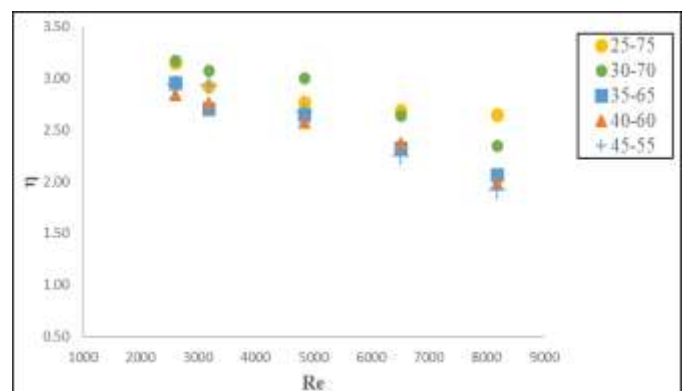

Fig. 16. Variation of thermal performance factor with Reynolds number, (Inlet from twisted tape).

3.4.2 Thermal performance factor (hot water inlet from wire coil):-

Figure 17 shows the relation between the Reynolds number (Re) and η for different twisted tape and wire coil length combination inserted into the tube when hot water inlet from the wire coil turbulator.

As shown in the figure the thermal performance factor is decreasing with the increasing of the Reynolds number as same trend in the case of hot water inlet from the twisted tape. In this hot water inlet condition, in all different length combination the thermal performance factor is higher than the unity, which shows the good thermal performance of the all-different combination with the hot fluid inlet condition.
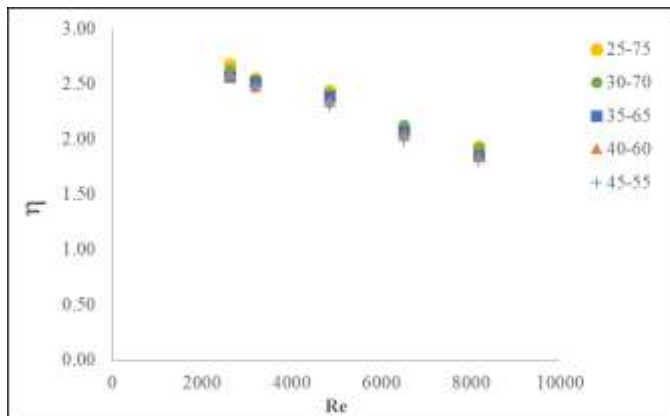

Fig. 17. Variation of thermal performance factor with Reynolds Number (inlet from wire coil).

4. Comparison of the present work with the work of other Researchers:

For the comparison of my present work analysis results, solved the classified papers published by researchers and compare my results with their results, and my result follows same trend as the researchers, my result follows with error in the range of ± 6 to ± 10 %.

To check the methodology, analyzed double pipe plain tube heat exchanger with the specified boundary condition and calculate Nusselt number and friction factor and compare it with results of other researcher's work and its nearly match with their results.

From this comparison, verification of my methodology of analysis has completed and the result shows that my methodology of the present work is right for analyze the double pipe heat exchanger with insert and/or without inserts

4.1 Comparisons of results:-

4.1.1 Comparison for Smooth Tube (Without Inserts) with the work of P. Murugesan, K. Mayilsamy, Suresh, P.S.S. Srinivasan, (ELSEVIER) [21].
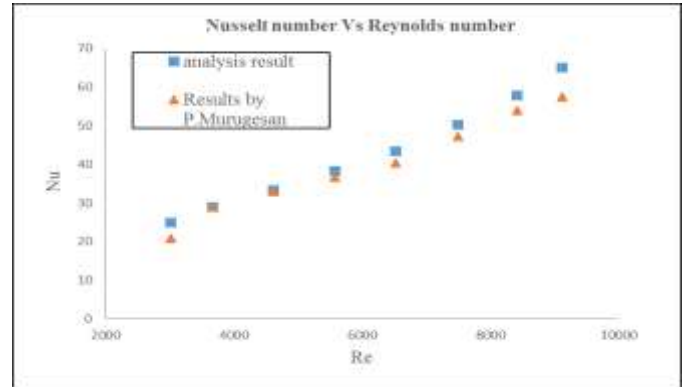

Fig. 18. Comparison of Nusselt Number-Smooth tube.

Figure 18 shows the comparison of analyzed result of Nusselt number for the plain tube with the results of the research's P. Murugesan, K. Mayilsamy, Suresh, and P.S.S. Srinivasan. It shows that the trend of Nusselt number of the present work is same as the researcher's results. The reasons for getting this trend of Nusselt number with increasing Reynolds number is when Reynolds number increases the flow will change from laminar to turbulent or the more fluid layer will come in contact with the surface and it will increase the heat transfer and for that it gives increasing trend of Nusselt number.
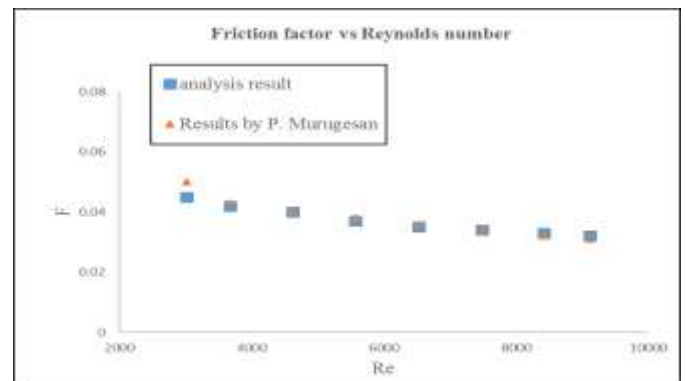

Fig. 19. Comparison of Friction Factor-Smooth tube.

Figure 19 shows the comparison of friction factor result with the results of the research's P. Murugesan, K. Mayilsamy, Suresh, and P.S.S. Srinivasan. As figure shows that, the trend of the results of friction factor decreases as Reynolds number increases is due to at higher Reynolds number because as increasing Reynolds number turbulent create in the flow will reduce the friction between the fluid and surface. The trend of the graph is same as the friction factor results P. Murugesan, K. Mayilsamy, Suresh, and P.S.S. Srinivasan.

5. CONCLUSION:

1.) Nusselt number is increasing with the Reynolds number is increasing. From the Nusselt number graph in both case first when the hot water inlet from the twisted tape and second inlet, hot water inlet from the wire coil turbulators, in both case with different turbulators Nusselt number increasing as Reynolds number increasing, which shows that in all cases heat transfer is increasing.

2.) When compare the plain tube (without turbulators) with the inserted tube the result of Nusselt number is very much higher in the case of inserted tube than the plain tube (without inserts). From this, we can conclude that the turbulators give higher heat transfer than plain tube.

3.) Now, when comparisons between different length combinations of two turbulators, graph shows that 25-75 % and 30-70% combination of the twisted tape and wire coil turbulator gives the highest heat transfer result with less pressure drop. Therefore, this two combination are good compare to the other combinations.

4.) From the comparison of hot water inlet position, concluded that the Nusselt number results higher in the case of the hot water inlet from the twisted tape side than the hot water inlet from wire coil side in all the different combination of twisted tape and wire coil turbulators. Therefore, the 25-75 % and 30-70 % combination with the hot water inlet from twisted tape gives better heat transfer compare to others and between this two combinations 25-75 % and 30-70%, the heat transfer is higher in the case of 25-75%.

5.) One fact is, if in any case of combined twisted tape and wire coil, the thermal performance factor is under unity, that is to say, the friction factor penalty is more dominant than the heat transfer ratio in this particular case. However, in present all cases, thermal performance factor is above the unity, which shows that all the cases shows the good thermal performance.

6. FUTURE SCOPE RELATED TO THIS PROJECT:

1.) In this project work, the turbulators used are made of copper material, further study possible by changing the material of the turbulators.

2.) By using different turbulators expect this two twisted tape and wire coil, experiments will perform.

3.) In this study, the turbulators (twisted tape and wire coil) not insulated and here the fin effects of turbulator is utilize for cooling the fluid. In future, turbulators will be insulate by insulating material and what are the effect of insulation on turbulator will find out.

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Appendix A. Different Contours of Heat Transfer Process:

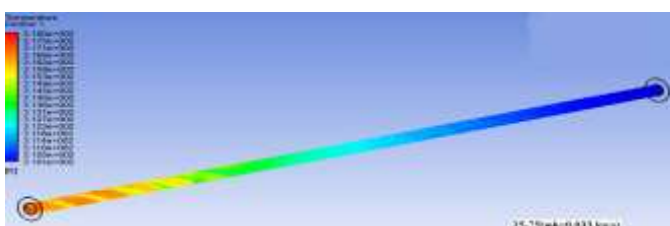A.1. **Contour of Hot fluid temperature profile:**

Figure A-1 Contour of hot fluid temperature profile

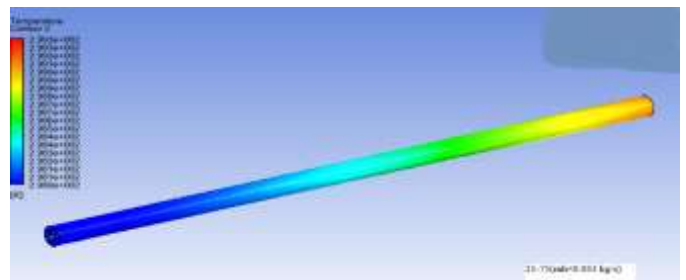**A.2 Contour of Cold fluid temperature profile:**

Figure A-2 Contour of cold fluid temperature profile.

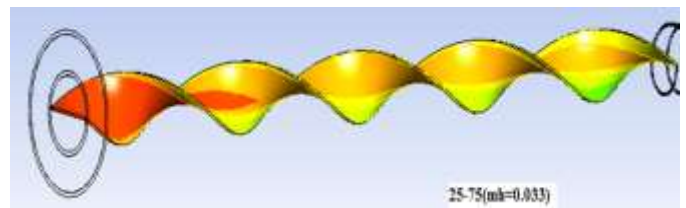**A.3 Contour of Temperature Profile on Twisted tape:**

Figure A-3 Contour of temperature profile on twisted tape. (Fin effect of twisted tape).

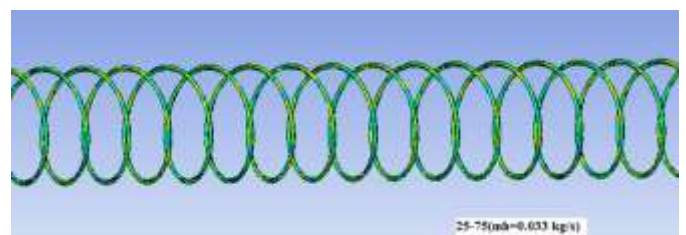**A.4 Contour of Temperature Profile on Wire-Coil:**

Figure A-4 Contour of temperature profile on wire-coil. (Fin effect of wire coil).

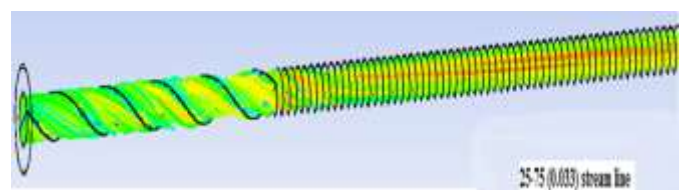**A.5 Streamline of velocity Profile of hot fluid:**

Figure A-5 Streamline of velocity Profile of hot fluid.