ISSN(Online): 2319-8753
ISSN (Print): 2347-6710

International Journal of Innovative Research in Science, Engineering and Technology

(A High Impact Factor, Monthly Peer Reviewed Journal)

Vol. 5, Issue 4, April 2016

Heat Transfer Intensification in U-bend Double Pipe Heat Exchanger using Twisted Tape Inserts

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Abstract: Generating a swirl flow in the flow field is widely employed passive heat transfer augmentation or intensification technique. In present work, twisted tapes are inserted into the inner pipe side of Counter flow U-bend Double Pipe Heat Exchanger for creating a swirl flow and thus Enhancing the Heat Transfer. Twisted tapes of different twist ratios (H/D) 5,10,15,20 and strip insert (i.e without any twist) are employed to enhance the Heat Transfer. Turbulent swirl flow with forced convection is considered for experimental study. For different twist ratios, mass flow rate of cold fluid in inner pipe side is varied, by maintaining the constant mass flow rate in the annulus side. Experimental data is validated using available correlations. The results show that, for mass flow rate of 8 LPM the enhancement in Heat Transfer is 55.69% compared with that of plain tube and corresponding Pressure Drop is only 20%.

Keywords: Heat Transfer Intensification, Heat Transfer Enhancement, U-bend Double Pipe Heat Exchanger, Inserts, Twisted Tapes.

I. INTRODUCTION

Heat Transfer is unavoidable process that nature has gifted to Humankind. In engineering applications wherever energy conversion is involved, Heat transfer finds a great importance. Heat Transfer Enhancement or Intensification aims at increasing the rate at which Heat transfers from one body to another or from one medium to another. Improved Heat Exchange over and above the usual and standard practice can significantly improve thermal efficiency in such engineering applications as well as the economics of their design and operation can be optimised [1]. This fact gives lot of scope for Research in this area of Heat Transfer Enhancement.

S.S. Hsieh, I.W. Huang[2] investigated Heat Transfer Enhancement in Laminar Flow with Longitudinal inserts. For these inserts, Thermal Entry Length was found and it is correlated with in terms of Reynolds Number. Based on same hydraulic Diameter, Enhancement of Heat Transfer compared to bare tube was found to be 16 times more at Reynolds Number less than 400, while Friction Factor rise is just 4.5 times.

S. K. Saha, A. Dutta[3] observed that for higher twists, twisted tapes of short length are performing better than full length twisted tapes. Lower values of Nusselt Number and Friction Factor are obtained for short length twisted tapes compared with full length counter parts.

Under Uniform Wall Heat Flux boundary condition, S. K. Saha, P. Langille[4] conducted experimental and theoretical studies on Laminar flow through a circular tube. For short length strips & regularly spaced strip elements, Friction Factor & Nusselt Number are less in comparison with the full length elements

Anil Singh Yadav[5] found that heat transfer coefficient is found to increase by 40% with half-length twisted tape inserts when compared with plain heat exchanger. It was found that on the basis of equal mass flow rate, the heat

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ISSN (Print): 2347-6710

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transfer performance of half-length twisted tape is better than plain heat exchanger, and on the basis of unit pressure drop the heat transfer performance of smooth tube is better than half-length twisted tape.

P. V. Durga Prasad, A. V. S. S. K. S. Gupta[6] reported that on equal mass flow rate basis, Heat Transfer performance is enhanced with twisted tapes . Whereas, on unit pressure drop basis, Heat Transfer performance is better on smooth tube compared with twisted tapes. They also reported significant increase in heat transfer coefficient, friction factor, and found that the thermal Performance of smooth tube is better than the full length twisted tape by 2.0–2.2 times.

Based on experimental flow visualization and computational modelling of single-phase laminar flows, Raj M. Manglik, Arthur E. Bergles[7] considered a fundamental scaling of the cross-sectional vortex structure and a parametric analysis of the primary enhancement mechanisms in single-phase flows are delineated.

B. V. N. Ramakumar, J. D. Arsha, Praveen Tayal[8] employed Tapered Twisted Tape Inserts for Enhancing Heat Transfer. They performed simulations using ANSYS FLUENT 14.0. They reported that overall enhancement ratio is greater than that of classical twisted tapes under all Reynolds Number Studied. An increase of 17% in overall enhancement is predicted with taper angle of 0.5.

S.N. Sarada, A.V.S.R. Raju and K.K. Radha[9] conducted experimental and theoretical studies on turbulent flow heat transfer enhancement in a horizontal circular tube using mesh inserts. They reported an increase of Nusselt number 2.15 times and pressure drop of only 1.23 times compared to that of plain tube.

P.Murugesan, K.Mayilsamy, S.Suresh[10] investigated Heat Transfer Enhancement using square cut twisted tapes with three twist ratios 2.0, 4.4, 6.0. Over the range considered, the Nusselt number is 1.03 to 1.14 times, friction factor is 1.05 to 1.25 times and thermal enhancement factor is 1.02 to 1.06 times in a tube with Square cut Twisted Tape of those in tube with Plain Twisted Tapes.

Twin counter twisted tapes and twin co-twisted tapes are employed by S. Eiamsa-ard, C. Thianpong, P. Eiamsa-ard[11] for generating counter swirl flow and co-swirl flow respectively. They reported that Twin counter twisted tapes improve Heat Transfer around 12.5–44.5% and 17.8–50% higher than those with the twin co-twisted tapes and single twisted tape, respectively.

S. V. Patil & P. V. Vijay Babu[12] employed twisted tapes in square Duct and Circular Tube. Working Fluid is Ehylene Glycol. Based on constant flow rate and constant pumping power criteria, Nusselt numbers were found to be 5.44–7.49 and 2.46–4.87 times that of plain square duct forced convection values respectively.

Numerical Simulation Studies are conducted by Sami D. Salman, Abdul Amir H. Kadhum, Mohd S. Takriff, Abu Bakar Mohamad[13] on Circular Tube Fitted with Alternative Axis Twisted Tape under a Constant Heat Flux. Among the various twist ratios and alternative angles, the twist ratio of y = 2.93 and alternative angle β = 90° offered a maximum heat transfer enhancement.

II. EXPERIMENTAL SETUP AND PROCEDURE

2.1 Description of Experimental Setup:

The Counter Flow U-bend Double Pipe Heat Exchanger considered for this Experimental study consists of two essential parts

1. Inner Pipe of Inner Diameter(d_i) =19mm, Outer Diameter (d_0) = 25mm
2. Outer Pipe of Inner Diameter (Di) = 50mm, Outer Diameter (Do) = 56mm

Both these Pipes are mounted concentrically as shown in the Figure 1.

Outer pipe is insulated using an asbestos sheet throughout its length. Inner pipe which is not covered by annulus is also insulated with asbestos sheet.

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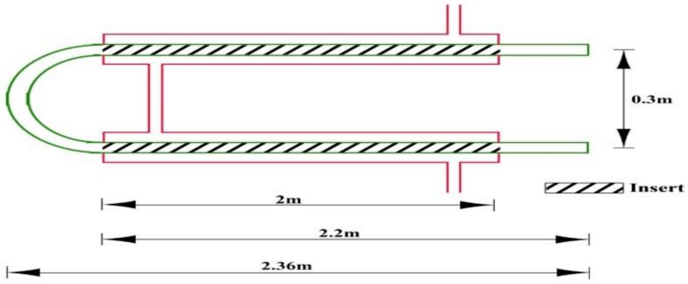

Figure1: Dimensions of U-Bend Double Pipe Heat Exchanger.

Experimental setup primarily comprises of two circuits.

1. Cold Water Circuit
2. Hot Water Circuit

In Cold Water Circuit, indicated with green colour in Figure 1 and Figure 2, the cold water in the cold water stainless steel tank is pressurised using 0.5HP Motor Pump. At the exit of the Pump, the cold water is allowed to flow through a Rotameter, which is used for regulating the Discharge or Rate of Flow. Then water is made to enter the inner pipe side of Heat Exchanger. At exit, cold water is drained.

- In this cold water circuit, two k- type thermocouples are placed at the inlet and outlet of inner pipe of Heat Exchanger, which are used for indicating the temperatures in the separately connected Digital meter.
- Inlet and outlet of the inner pipe are connected to a U-Tube Mercury Manometer for measuring Pressure Drop.

In Hot Water Circuit, indicated with green colour in Figure 1 and Figure 2, the water in the insulated stainless steel tank is heated using a 0.375kW Heater. After obtaining the required hot water temperature, the hot water is pressurised using a 0.5HP Motor Pump. At the exit of the pump, a Rotameter is arranged for regulating the Discharge or Rate of Flow. Then the water is made to flow through the annulus of the Heat Exchanger.

- In this Hot water circuit also, two k- type thermocouples are placed at the inlet and outlet of Annulus of Heat Exchanger, which are used for indicating the temperatures in the separately connected Digital meter.

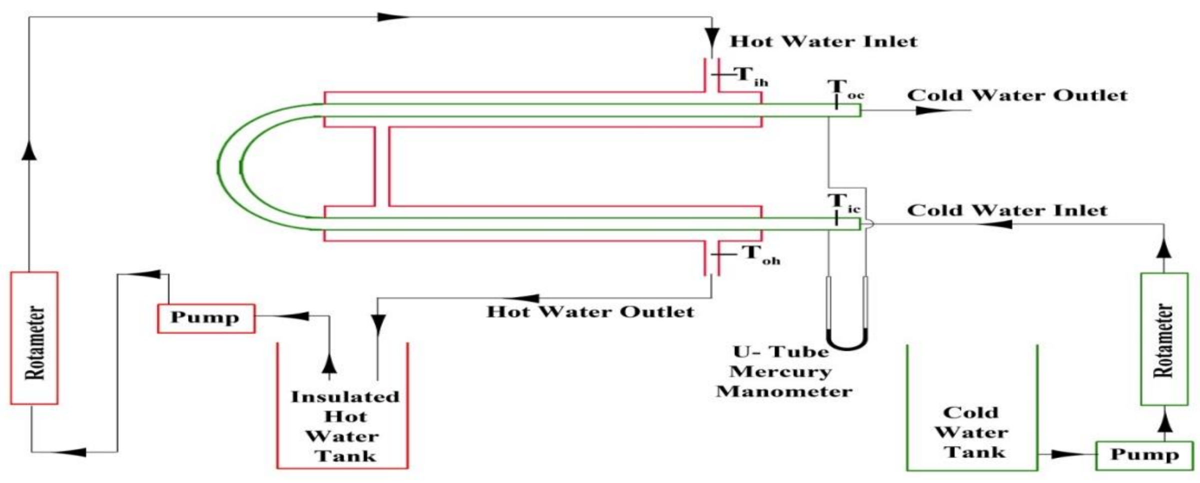

Figure2: Experimental Setup of Counter flow U-bend Double Pipe Heat Exchanger.

2.2 Twisted Tape Geometry:

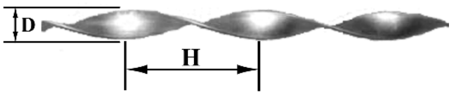

Figure 3: Geometry of Twisted Tape insert.

The Geometrical features of the twisted tape insert are represented in Figure 3. A aluminium strip of 1mm thickness is twisted into required twist ratios. Here 'H' represents the pitch of Twisted Tape and 'D' represents the width of Twisted Tape, measured across diameter. Table1 shows the Dimensions of Twisted Tapes with Different Geometries. H/D ratio is used to name each twisted tape insert.

Table 1: Dimensions of Twisted Tape Insert.

H/D Ratio	H (in meter)	D (in meter)
5	0.09	0.018
10	0.18	0.018
15	0.27	0.018
20	0.36	0.018

2.3 Experimental Procedure:

Initially experiments are conducted on plain tube i.e without any insert. At the beginning of every experiment, the flow rates of cold water and hot water are set at required rates using Rotameter. For all experiments, the cold water flow rate is fixed at 8LPM. The Hot water flow rate is varied from 8LPM to 12LPM for the different experiments.

After heating the water in insulated Hot water tank to the required temperature, the hot water is made to flow through the annulus. Simultaneously, the cold water is made to flow through the inner pipe. Steady state is reached in 20 to 30 minutes of time. After reaching Steady State, the temperatures are recorded. Difference of heights of mercury in Manometer is also recorded.

Twisted tape insert of 2m length is inserted into the inner pipe of Heat Exchanger such that it covers the length of annulus and same experimental procedure is followed.

III. DATA REDUCTION PROCEDURE

Heat transfer on annulus side i.e on hot water side is calculated using

$Q_h=\dot{m}_h*C_{ph}*\Delta T_h$ (1)

Heat transfer on inner pipe side i.e on cold water side is calculated using

$Q_c=\dot{m}_c*C_{pc}*\Delta T_c$ (2)

Average Heat Transfer is calculated using

$Q_{avg}=(Q_h+Q_c)/2$ (3)

Reynolds number is calculated using

$Re=v*D_e/\nu$ (4)

where Kinematic Viscosity 'ν' is measured at Bulk Mean Temperature.

Nusselt Number on Hot water Annulus side is calculated from Dittus Boelter Equation

$Nu=0.023*(Re)^{0.8}*(Pr)^{0.3}$ (5)

Heat Transfer Coefficient is measured from

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$h_{annulus}$=Nu*k/D_e (6)

where Thermal Conductivity 'k' is measured at Bulk Mean Temperature.
Overall Heat Transfer Coefficient is measured from
U=Q_{avg}/(A_{avg}*LMTD) (7)

Experimental Heat Transfer Coefficient is measured from
h_{exp}=1/((1/U)-(1/$h_{annulus}$)) (8)

Friction Factor is Calculated from
f = (2*ΔP*D_i)/(ρ*L*v^2) (9)

IV. DISCUSSION ON RESULTS

4.1 Validation of Plain Tube Data:

4.1.1.Heat Transfer:

Available correlations for plain tube are

1. Dittus Boelter Equation[15]

$$Nu = 0.023.(Re)^{0.8}.(Pr)^{0.3} \qquad \{ 0.7 \le Pr \le 160; Re > 10{,}000\}$$

2. Petukhov Equation[15]

$$Nu = \frac{\left(\frac{f}{8}\right).Re.Pr}{1.07 + 12.7\left(\frac{f}{8}\right)^{0.5}\left(Pr^{\frac{2}{3}} - 1\right)} \qquad \begin{array}{l} 0.5 \le Pr \le 2000 \\ 10^4 \le Re \le 5 \times 10^6 \end{array}$$

3. Gnielinski Equation[15]

$$Nu = \frac{\left(\frac{f}{8}\right).(Re - 1000).Pr}{1 + 12.7\left(\frac{f}{8}\right)^{0.5}\left(Pr^{\frac{2}{3}} - 1\right)} \qquad \begin{array}{l} 0.5 \le Pr \le 2000 \\ 3 \times 10^3 \le Re \le 5 \times 10^6 \end{array}$$

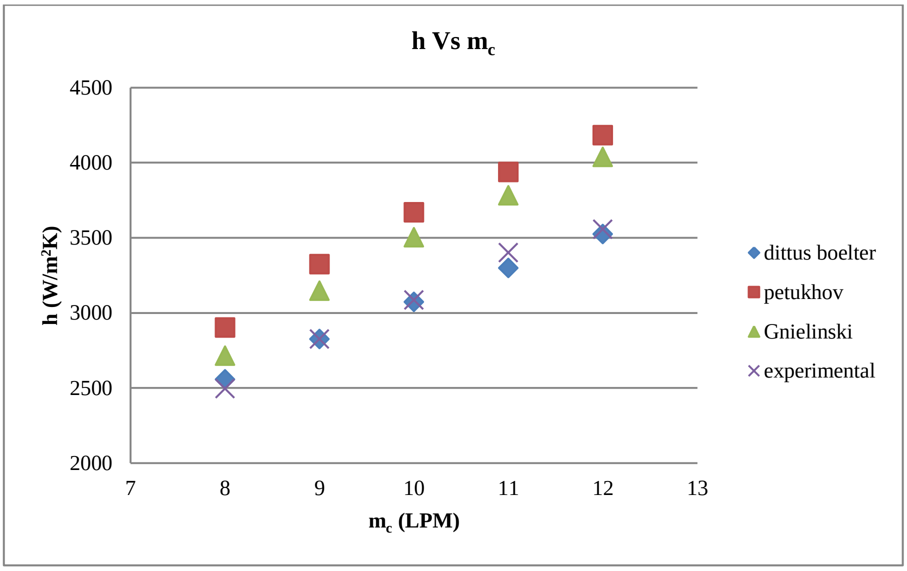

Figure 4: Comparison of present experimental Heat Transfer data with available correlations.

ISSN(Online): 2319-8753
ISSN (Print): 2347-6710

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(A High Impact Factor, Monthly Peer Reviewed Journal)

Vol. 5, Issue 4, April 2016

In Figure 4, the value of h, Heat Transfer Coefficient obtained in experiment is compared with the available correlations given by Dittus Boelter, Petukhov, Gnielinski. From this it has been observed that the experimental Heat transfer coefficient is in agreement with available correlations.

4.1.2. Friction Factor

Available correlations for plain tube are

1. Blasius Equation[15]

$$f = \frac{0.3614}{Re^{0.25}}$$

2. Petukhov Equation[15]

$$f = \frac{1}{(0.790 \ln(Re) - 1.64)^2} \qquad 3000 \leq Re \leq 5 \text{ x } 106$$

3. Simple Power Law Equation[15]

$$f = \frac{0.184}{Re^{0.2}}$$

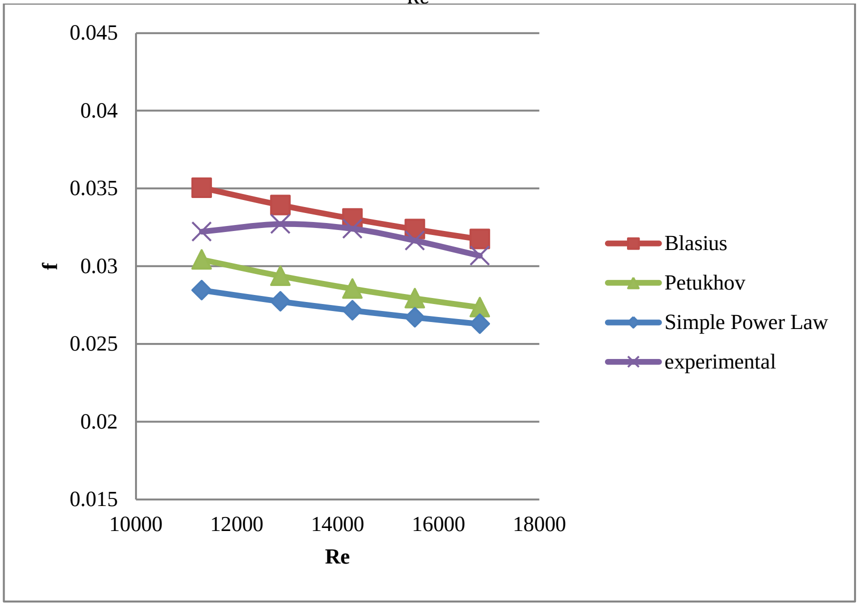

Figure 5: Comparison of present experimental friction factor with available correlations.

In Figure 5, the value of f, friction factor obtained in experiment is compared with the available correlations Blasius Equation, Petukhov Equation, Simple Power Law Equation. From this figure it is observed that the experimental friction factor is in agreement with available correlations

4.2 Validation of Twisted Tape Data:

Available Nusselt Number correlations for twisted tapes are

1. Sarma et.al Correlation

$$Nu = 0.1012(1+(D/H))^{2.065}(Re)^{0.67}(Pr)^{0.3}$$

2. Manglik and Bergles Correlation[16]

$$Nu = 0.023\, Re^{0.8} Pr^{0.4}(1+(0.769(2D/H)))\, \Phi_2$$

$$\text{Where, } \Phi_2 = \left[\frac{\pi}{\pi - 4t/D}\right]^{0.8} \left[\frac{\pi + 2 - 2t/D}{\pi - 4t/D}\right]^{0.2}$$

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Vol. 5, Issue 4, April 2016

a) H/D 5

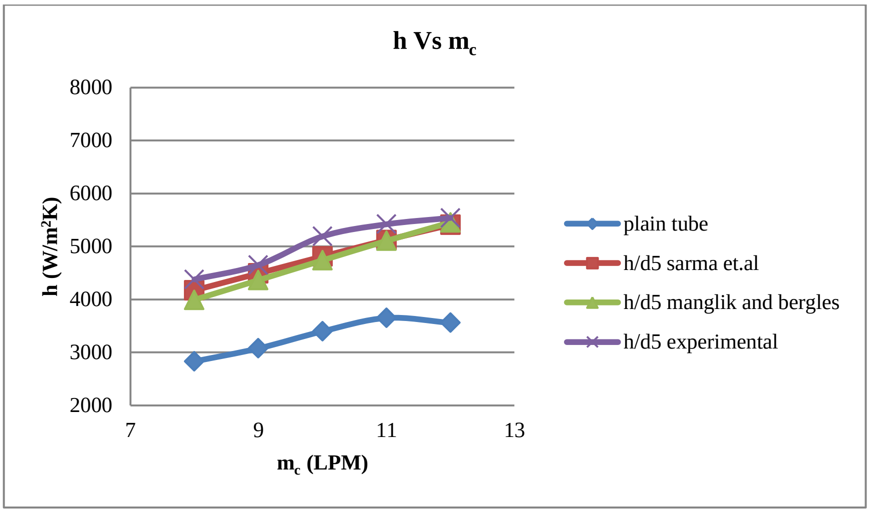

Figure 6: Comparison of experimental heat transfer coefficient with available correlations-twisted tape insert H/D5.

In Figure 6, the value of h, Heat Transfer Coefficient obtained in experiment is compared with the available correlations given by Sarma et.al and Manglik & Bergles. In this Plot, it has been observed that the experimental Heat transfer coefficient is in agreement with available correlations with considerable maximum variation of +8%. When twisted tape insert of twist ratio H/D 5 is used, Compared with Plain Tube, Highest Heat Transfer Enhancement of 55.69% is observed at 12LPM (Re=16988). Corresponding Pressure Drop is 20%, Compared with Plain Tube.

b) H/D 10

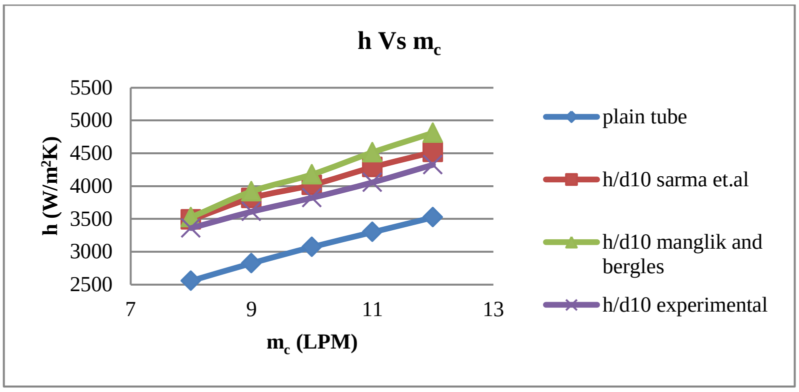

Figure 7: Comparison of experimental heat transfer coefficient with available correlations-twisted tape insert H/D 10.

In Figure 7, the value of h, Heat Transfer Coefficient obtained in experiment is compared with the available correlations given by Sarma et.al and Manglik & Bergles. In this Plot, it has been observed that the experimental Heat transfer coefficient is in agreement with available correlations with considerable maximum variation of +4%. When twisted tape insert of twist ratio H/D 10 is used, Compared with Plain Tube, Highest Heat Transfer Enhancement of 38% is observed at 12LPM (Re=16988). Corresponding Pressure Drop is 23%, Compared with Plain Tube.

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ISSN(Online): 2319-8753
ISSN (Print): 2347-6710

International Journal of Innovative Research in Science, Engineering and Technology

(A High Impact Factor, Monthly Peer Reviewed Journal)

Vol. 5, Issue 4, April 2016

c) H/D 15

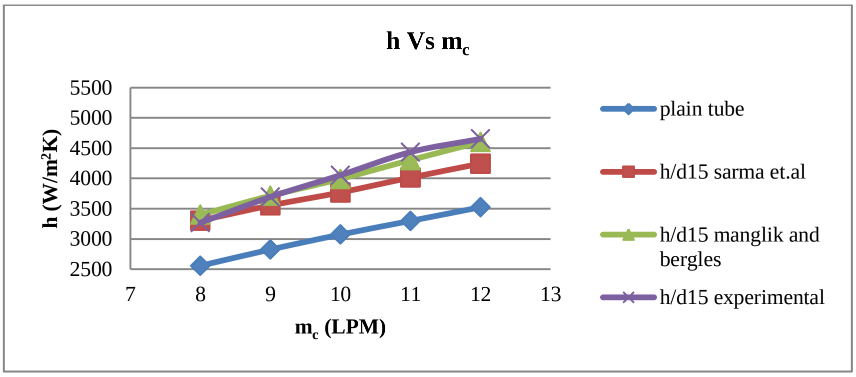

Figure 8: Comparison of experimental heat transfer coefficient with available correlations-twisted tape insert H/D 15.

In Figure 8, the value of h, Heat Transfer Coefficient obtained in experiment is compared with the correlations given by Sarma et.al and Manglik & Bergles. In this Plot, it has been observed that the experimental Heat transfer coefficient is in agreement with available correlations with considerable maximum variation of +9%. When twisted tape insert of twist ratio H/D 10 is used, Compared with Plain Tube, Highest Heat Transfer Enhancement of 24.8% is observed at 12LPM (Re=17026). Corresponding Pressure Drop is 26.66%, Compared with Plain Tube.

d) H/D 20

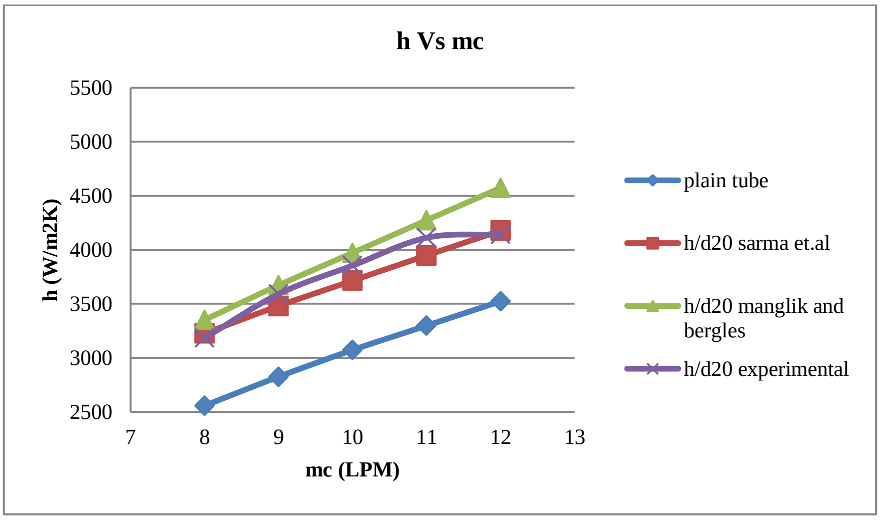

Figure 9: Comparison of experimental heat transfer coefficient with available correlations-twisted tape insert H/D 20.

In Figure 9, the value of h, Heat Transfer Coefficient obtained in experiment is compared with the correlations given by Sarma et.al and Manglik & Bergles. In this Plot, it has been observed that the experimental Heat transfer coefficient is in agreement with available correlations with considerable maximum variation of -9%. When twisted tape

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International Journal of Innovative Research in Science, Engineering and Technology

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Vol. 5, Issue 4, April 2016

insert of twist ratio H/D 10 is used, Compared with Plain Tube, Highest Heat Transfer Enhancement of 16.4% is observed at 12LPM (Re=17677). Corresponding Pressure Drop is 30%, Compared with Plain Tube.

e) Strip Insert (without any twist)

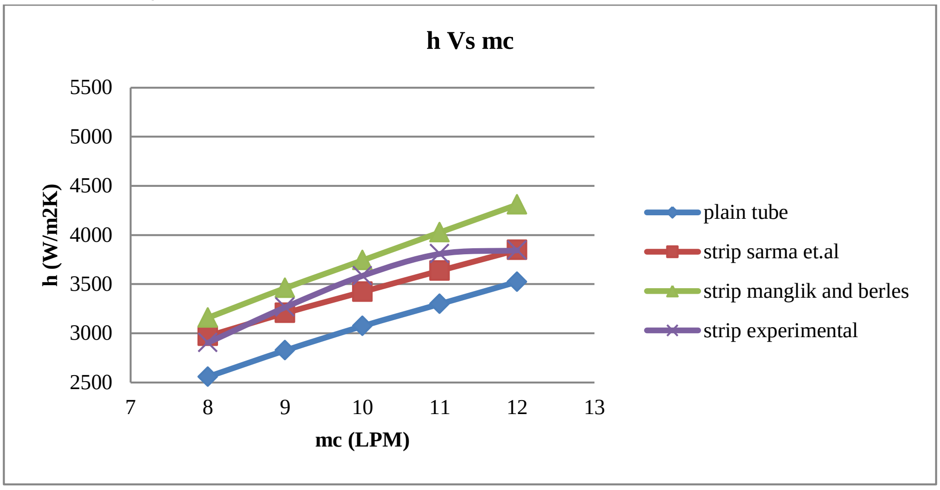

Figure 10: Comparison of experimental heat transfer coefficient with available correlations for strip insert.

In Figure 10, the value of h, Heat Transfer Coefficient obtained in experiment is compared with the correlations given by Sarma et.al and Manglik & Bergles. In this Plot, it has been observed that the experimental Heat transfer coefficient is in agreement with available correlations with maximum variation of -13%. When twisted tape insert of twist ratio H/D 10 is used, compared with Plain Tube, Highest Heat Transfer Enhancement of 8% is observed at 12LPM (Re=17677). Corresponding Pressure Drop is 36%, Compared with Plain Tube.

4.2.1 Effect of Twist Ratio on Heat Transfer Enhancement

Twist Ratio has significant effect on the amount of Heat Transfer Enhancement. From experiments it has been observed that more is the twist i.e lesser the Twist Ratio, Higher is the Heat Transfer Enhancement. Further it is observed that negative effect of Pressure Drop is reduced significantly at Higher Twists and Lower Twist Ratios.

4.2.2 Mass Flow Rate based Analysis

At Cold Water mass flow rate of 12LPM, it has been observed that the highest Heat Transfer Enhancement is achieved in all cases. Also, it is observed that Heat Transfer Enhancement increases with increase of Mass Flow Rate. This mass flow rate refers to Reynolds Number in Non Dimensional Similarity Analysis. Higher is the Reynolds Number, Higher is the Heat Transfer Enhancement. Also we may conclude that Highly Turbulent Flow Fields have Higher chances of Heat Transfer Enhancement, in the range considered for Experiments.

4.3 Pressure Drop with inserts

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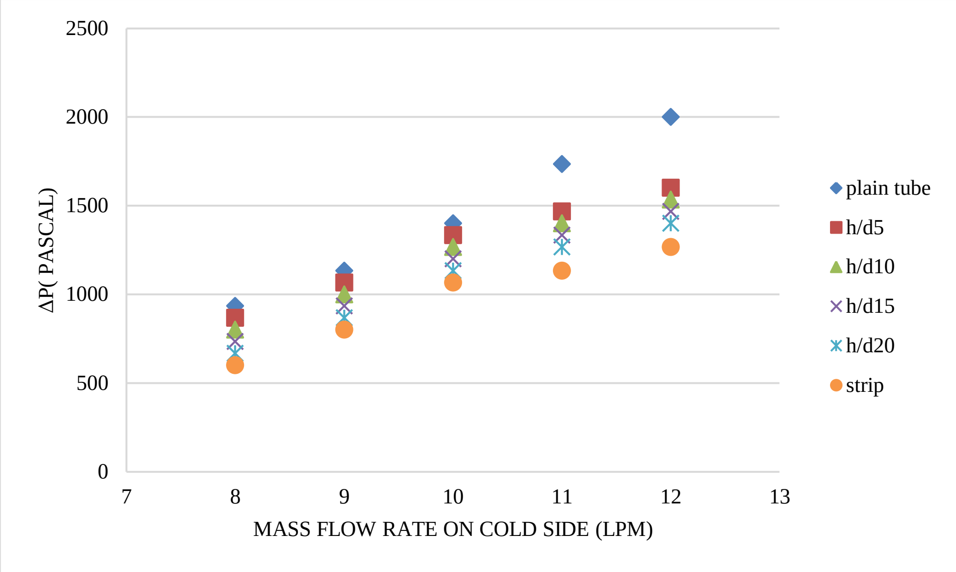

Figure 11: Variation of Pressure Drop for all Inserts.

It has been observed from Figure 11, that the Pressure Drop is also increasing with increase in mass flow rate. But the Negative Effect of this increased Pressure Drop can be compromised with higher Heat Transfer Enhancement at Higher Twists i.e at Lower Twist Ratios.

V. CONCLUSION

The following conclusions are drawn from the experimental investigations carried out in the Counter flow U-bend Double Pipe Heat Exchanger by the means of strip and twisted tape inserts.

- From experiments it is observed that enhancement of heat transfer is maximum when twisted tape insert with H/D ratio= 5 is used. Also it is observed that more is the twist or the swirl of the insert, more is the Heat transfer Enhancement.
- Enhanced Heat Transfer obtained is 55.696% with insert of H/D ratio =5 at 12LPM and Re = 16,988 compared with that of plain tube and corresponding Pressure Drop is only 20%. Swirl motion is primarily responsible for the Enhancement of Heat Transfer, it is obvious from the fact that more is the twist ratio, more is the Enhancement.

Acknowledgement

Corresponding Author gratefully acknowledges the warm and kind support of Dr.A.V.S.S.K.S.Gupta, Professor, JNTUHCEH & Dr.P.Bhramara, Associate Professor, JNTUHCEH during this research work.

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NOMENCLATURE

A	Area of heat transfer m^2	**$h_{inner\ pipe}$**	Heat transfer coefficient on Cold Inner pipe, W/m^2-K
d_0	Outer Diameter of Inner Pipe, m	**ΔP**	Pressure Drop across Cold Water Inner Pipe Side, Pa
D_o	Outside diameter of annulus pipe, m	**v**	Velocity of water on Cold Water Side, m/s
H/D	Twist ratio = Pitch of insert/width of insert{ H/d_i}	**LMTD**	Log Mean Temperature Difference
d_i	Inner Diameter of Inner Pipe, m	**Re**	Reynolds Number
D_i	Inside diameter of annulus pipe, m	**Pr**	Prandt Number
B	Equivalent Diameter $\{(D_i^2-d_o^2)/ d_0\}$,	**Nu**	Nusselt Number
L	Length of tube	**K**	Thermal conductivity of fluid, W/m-K
t	Thickness of Insert	**f**	Friction factor
ΔP	Pressure drop, mm of Hg	**U**	Overall heat transfer coefficient, W/m^2-K
m_c	Mass flow rate on Cold Water in Inner Pipe Side, LPM	**Greek Symbols**	
m_h	Mass flow rate on Hot Water in Annulus Sides, LPM	**α**	Thermal diffusivity, m^2/s
Q	Rate of heat transfer, W	**μ**	Dynamic viscosity of fluid, Pa-s
C_p	Specific heat of fluid, kJ/kg-K	**ρ**	Density, kg/m^3
T_{ih}	Inlet temperature of hot water, °C	**ν**	Kinematic viscosity of fluid, m^2/s
T_{oh}	Outlet temperature of hot water, °C	**π**	Equivalent to (22/7)
ΔT_h	Temperature Difference of Hot Water, °C	**Subscripts**	
T_{avgh}	Average Temperature of Hot Water, °C	**h**	Hot water
T_{ic}	Inlet temperature of cold water, °C	**c**	Cold water
T_{oc}	Outlet temperature of cold water, °C	**i**	Inlet
ΔT_c	Temperature Difference of Cold Water, °C	**o**	Outlet

ISSN(Online): 2319-8753
ISSN (Print): 2347-6710

International Journal of Innovative Research in Science, Engineering and Technology

(A High Impact Factor, Monthly Peer Reviewed Journal)

Vol. 5, Issue 4, April 2016

$\mathbf{T}_{avgc}$ Average Temperature of Cold Water, °C $\mathbf{h}_{annulus}$ Heat transfer coefficient on hot annulus side, W/m^2-K	