Enhancement of heat transfer using varying width twisted tape inserts

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Abstract : The present work shows the results obtained from experimental investigations of the augmentation of turbulent flow heat transfer in a horizontal tube by means of varying width twisted tape inserts with air as the working fluid. In order to reduce excessive pressure drops associated with full width twisted tape inserts, with less corresponding reduction in heat transfer coefficients, reduced width twisted tapes of widths ranging from 10 mm to 22 mm, which are lower than the tube inside diameter of 27.5 mm are used. Experiments were carried out for plain tube with/without twisted tape insert at constant wall heat flux and different mass flow rates. The twisted tapes are of three different twist ratios (3, 4 and 5) each with five different widths (26-full width, 22, 18, 14 and 10 mm) respectively. The Reynolds number varied from 6000 to 13500. Both heat transfer coefficient and pressure drop are calculated and the results are compared with those of plain tube. It was found that the enhancement of heat transfer with twisted tape inserts as compared to plain tube varied from 36 to 48% for full width (26mm) and 33 to 39% for reduced width (22 mm) inserts.Correlations are developed for friction factors and Nusselt numbers for a fully developed turbulent swirl flow, which are applicable to full width as well as reduced width twisted tapes, using a modified twist ratio as pitch to width ratio of the tape.

Keywords: Enhancement; heat transfer; twisted tape inserts; turbulent; swirling; pressure drop

I. INTRODUCTION

Among many techniques (both passive and active) investigated for augmentation of heat transfer rates inside circular tubes, a wide range of inserts has been utilized, particularly when turbulent flow is considered. The inserts studied included coil wire inserts, brush inserts, mesh inserts, strip inserts, twisted tape inserts etc. Augmentation of convective heat transfer in internal flows with tape inserts in tubes is a well-acclaimed technique employed in industrial practices.

Hsieh and Huang (2000) conducted experimental studies for heat transfer and pressure drop of laminar flow in horizontal tubes with/without longitudinal inserts. They reported that enhancement of heat transfer as compared to a conventional bare tube at the same Reynolds number to be a factor of 16 at Re ≤ 4000, while a friction factor rise of only 4.5. A new method was postulated by (Sarma*et al*., 2003) to predict heat transfer coefficients with twisted tape inserts in a tube in which the wall shear and the temperature gradients were properly modified through friction coefficient correlation leading to heat transfer augmentation from the tube wall. Experimental studies on heat transfer and flow characteristics for turbulent flow of air in a horizontal circular tube with strip type insert (longitudinal and Crossed Strip inserts). They reported that friction factor rise due to inclusion of inserts was typically between 1.1 and 1.5 from low Re (=6500) to high Re (=19500) with respect to bare tube (Hsieh *et al*., 2003). Experimental investigations for the heat transfer characteristics and the pressure drop in horizontal double pipes with twisted tape inserts were carried out (Naphon, 2006). Heat transfer, friction factor and enhancement efficiency characteristics in a circular tube fitted with conical ring turbulators and a twisted-tape swirl generator were investigated experimentally (Promvonge and Eiamsaard, 2007).

Monheit (1987) made a comparative study of the thermal performance of ordinary full-width full-length twisted tapes with tapes having modified surface configurations. Dasmahapatra and Rao (1991) studied augmentation of heat transfer to viscous nonNewtonian fluids in laminar flow using full width interrupted twisted tapes under the uniform wall temperature condition. AlFahed and Chakroun (1996) investigated the effect of tube-tape clearance on heat transfer under fully developed turbulent flow conditions in a horizontal isothermal tube. Investigations were carried out on twisted tape inserts in viscous flow in a horizontal tube (Manglik and Bergles, 1992). Experimental data were obtained for water and ehtylene glycol with snug-fit tape inserts of three different twist ratios, y=3.0, 4.5, and 6.0; the tape thickness in each case was 0.483 mm. In continuation of their research, an extended review of the application of twisted-tape inserts in tubular heat exchangers and their thermal-hydraulic performance was discussed (Manglik and Bergles, 2003). Twisted tapes promote enhanced heat transfer by generating swirl or secondary flows, increasing the flow velocity due to the tube partitioning and blockage. Experiments were conducted on compound heat transfer enhancement in a tube fitted with serrated twisted tape in the Reynolds number range of 5000 to 25000 (Chang *et al*., 2007). Heat transfer and pressure drop characteristics of laminar flow in rectangular and square plain ducts and ducts with twisted-tape inserts were experimentally investigated by (Saha and Mallick, 2005).

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Experimental investigation of heat transfer and friction factor characteristics of circular tube fitted with full-length helical screw element of different twist ratio, and increasing and decreasing order of twist ratio set have been studied with uniform heat flux (Sivashanmugam and Suresh, 2006). Empirical correlations were developed in terms of twist ratio and Reynolds number. In continuation of their research, investigations on the heat transfer and friction factor characteristics of a circular tube fitted with a full-length helical screw element with different twist ratios (Sivashanmugam and Suresh, 2007). They performed their experiments for various Reynolds numbers and the obtained experimental data were then compared with those previously reported in the literature. They reported higher performance of the helical twisted insert in comparison with the twisted tape insert.

Modification of twisted tape was made by focusing on the increase of heat transfer rate rather than the reduction of friction loss, for example, the jagged (Rahimi*et al.*, 2009) broken (Chang *et al.*, 2007) and serrated (Chang and Jan, 2007) twisted tapes. The tapes in this group were designed to offer stronger swirl flow and better mixing than the typical one. However, the enhanced heat transfer by the use of the twisted tapes in the group was certainly accompanied by the rise of friction factor. In general, the performance factors of twisted tapes in this group were higher.

Experimental studies were carried out on the heat transfer enhancement and pressure drop characteristics in presence of twisted tape inserts, during flow boiling of R-134a inside a horizontal evaporator for plain flow and four tubes with twisted tapes of 6, 9, 12 and 15 twist ratios and four refrigerant mass velocities of 54, 85, 114 and 136 kg/s m^2 for each tape. It has been found that the twisted tape inserts enhance the heat transfer coefficient on relatively higher pressure drop penalty, in comparison to that for the plain tube flow (Kumar *et al.*, 2009). Experimental studies for tubes fitted with Left - Right twisted tape inserts of various twist ratios to enhance the convective heat transfer rates for thermosyphon solar water heater system (Jaisankar*et al.*, 2008). Experimental investigations were carried out on the heat transfer enhancement and pressure drop characteristics in the presence of twisted tape inserts, during flow boiling of R-134a, inside a horizontal evaporator (Akhavan-Behabadi*et al.*, 2009).

Experimental investigations on circular tube fitted with full length twisted tape with trapezoidal-cut were studied for the Reynolds number range of 2000-12000 (Murugesan*et al.*, 2009). A significant increase in heat transfer coefficient and friction factor for tape with trapezoidal-cut was observed. Experimental investigation on heat transfer and friction factor characteristics of circular tube fitted with right-left helical screw inserts of equal length, and unequal length of different twist ratio have been presented

(Sivashanmugam and Nagarajan, 2007). The results shows that the heat transfer coefficient enhancement for right-left helical screw inserts is higher than that for straight helical twist for a given twist ratio. The relationship between the intensity of secondary flow and the intensity of laminar convective heat transfer is studied using a numerical method (Lin *et al.*, 2009). The results reveal that the cross-averaged absolute vorticity flux in the main flow direction can reflect the intensity of secondary.

Experimental studies were conducted in a round tube with short-length twisted tape insert on the mean Nusselt number, friction factor and enhancement efficiency characteristics under uniform wall heat flux boundary conditions (Eiamsa-ard*et al*, 2009). The short-length tape is introduced as a swirling flow device for generating a strong swirl flow at the tube entry while the full-length twisted tape is inserted into the tested tube at a single twist ratio of y/w=4.0. The enhancement efficiency of the tube with the short-length tape insert is found to be lower than that with the full-length one. In continuation, investigations were carried out in a tube equipped with the combined devices between the twisted tape (TT) and constant/ periodically varying wire coil pitch ratio (Eiamsa-ard*et al.*, 2010a). The highest thermal performance factor of around 1.25 is found by using DI-coil in common with the TT at lower Reynolds number. Also experiments were performed using single twisted tapes and full-length dual twisted tapes with three different twist ratios (y/w=3.0, 4.0 and 5.0) and also regularly-spaced dual twisted tapes with three different space ratios (s/D=0.75, 1.5 and 2.25). The effects of major parameters on heat transfer and friction factor are discussed. The result shows that the heat transfer of the tube with dual twisted tapes is higher than that of the plain tube with/without single twisted tape insert (Eiamsa-ard*et al.*, 2010b).

Heat transfer, flow friction and thermal performance factor characteristics in a tube fitted with delta-winglet twisted tape, using water as working fluid are investigated experimentally for oblique delta-winglet twisted tape (O-DWT) and straight delta-winglet twisted tape (S-DWT) arrangements over a Reynolds number range of 3000–27,000 by considering three twist ratios (y/w = 3, 4 and 5) and three depth of wing cut ratios (DR = d/w = 0.11, 0.21 and 0.32) (Wongcharee*et al.*, 2010). The results show that Nusselt number and mean friction factor in the tube with the delta-winglet twisted tape increase with decreasing twisted ratio (y/w) and increasing depth of wing cut ratio.

Patil (2000) studied the frictional and heat transfer characteristics of laminar swirl flow of pseudo plastic type power law fluid in a circular tube using varying width twisted tapes under a uniform wall temperature condition. Reduced width twisted tape inserts gave 18%−56% lower isothermal friction factors than the full width tapes. Nusselt numbers decreased only slightly by

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5% and 25 %, for tape widths of 19.7 and 11.0 mm, respectively inside a 25mm diameter tube. Also, the reduced width tapes offered 20% – 50% savings in the tape material as compared to the full width tapes, which is more economical.

The present investigation is aimed at studying the frictional and heat transfer characteristics in turbulent region using varying width twisted tape inserts under constant wall heat flux. The objective of using varying (reduced) width twisted tapes is to reduce the pressure drops associated with full width twisted tapes without seriously impairing the heat transfer augmentation rates and to achieve material savings.

II. MATERIALS AND METHODS

The apparatus consists of a blower unit fitted with a pipe, which is connected to the test section located in horizontal orientation. Nichrome bend heater encloses the test section to a length of a 40 cm. Four thermocouples T2, T3, T4 and T5 at a distance of 5cm, 15cm, 25cm and 35cm from the origin of the heating zone are embedded on the walls of the tube and two thermocouples are placed in the air stream, one at the entrance (T1) and the other at the exit (T6) of the test section to measure the temperature of flowing air as shown in Figure 1. The pipe system consist a valve, which controls the airflow rate through it and an orifice meter to find the volume flow rate of air through the system. The diameter of the orifice is 14mm with coefficient of discharge 0.64.
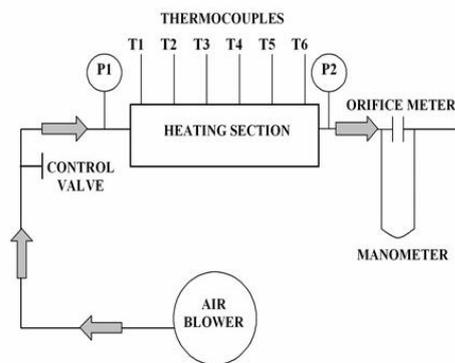
Figure 1: Experimental Set up Layout

Two calibrated pressure gauges (P1 and P2 range: 0-1000mm of water column) are provided to measure the pressure drops across the test section. Input to heater is given through dimmer stat. The inner tube of the heating part, which is the test tube with inside diameter 27.5mm, is made of 3.2 mm thick copper plate. Display unit consists of voltmeter, ammeter and temperature indicator. The circuit was designed for a load voltage of 0-220 V, with a maximum current of 10A. Difference in the levels of manometer fluid represents the variations in the flow rate of air. The velocity of airflow in the tube is measured with the help of orifice plate and the water manometer fitted on board.

2.1 Materials

The layout of tubes containing full width and the reduced width twisted tapes is shown in Figure 2. The tapes consist of uniformly long aluminium strips of desired width, which have been twisted about the longitudinal axis. Twisted tapes of five different widths (26, 22, 18, 14 and 10 mm) were fabricated for each of the three different pitches viz. 82.5, 110, and 137.5 mm as shown in Figure 2. For the case of reduced width tapes, the gap between the tube wall and the tape was maintained constant throughout the tube length. The tapes of full width (26 mm) were inserted as is into a 27.5mm-dia tube to give a sliding fit.

2.2 Experimental Procedure

Supply is given to the blower motor and the valve is opened slightly. A heat input of 40w is given to the nichrome heating wire wound on the test section by adjusting the dimmer stat. Thermocouples 2 to 5 are fixed on the test surface and thermocouples 1 to 6 are fixed inside the pipe. Thermocouples used for experimentation are J-type with an accuracy ±0.2%. The readings of the thermocouples are observed every 5 minutes until they show constant values. Under steady state condition, the readings of all the six thermocouples are recorded. The experiment is repeated for different openings of the valve, thus varying the airflow rate. The fluid properties were calculated as the average between the inlet and the outlet bulk temperature. It took 110 minutes to reach steady state conditions. Experiment was carried out at constant heat flux conditions and constant heat input of 40 w at different mass flow rates.

Initially the experiment is carried out without any insert (plain tube experiment). The working fluid air flows through the pipe section with least resistance. The experiment is carried out in similar fashion with straight tape inserts, with twisted tape inserts with twist ratios (pitch/diameter) 3, 4 and 5 for widths of 26, 22, 18, 14 and 10 mm. the inserts are made of aluminium. The mass flow rates considered for the constant heat input of 40 w in terms of water level difference in U-tube water manometer are 2 inch, 3 inch, 4 inch and 5 inches (air mass flow rate varying from 0.0033 to 0.0055 kg/sec). Different types of inserts used are shown in Table 1. Each insert in table 1 is experimentally investigated for three different mass flow rates of air. Each insert is taken and inserted into the test section axially. It is taken care that the strip doesn't scratch the inner wall of the pipe and get deformed. The presence of the insert in the pipe causes resistance to flow and increases turbulence. For the case of reduced width tapes, the gap between the tube wall and the tape was maintained constant throughout the tube length by brazing metal pins to the edges of the tape.

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Figure 2: Photographs of twisted tape inserts

Table 1. Shows the different types of inserts used

S. No	Twist Ratio (T.R)	Width of Tape insert (mm)
1	3, 4 and 5	26, 22, 18, 14 and 10
2	Straight tape	26, 22, 18, 14 and 10
3	Plain tube	-------------------

2.3. Experimental Uncertainty

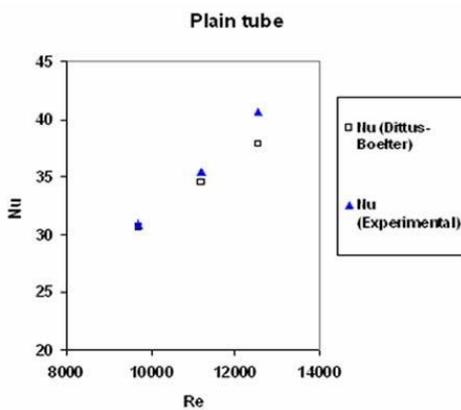Experiments were conducted initially for plain tube without inserts for different mass flow rates of air.

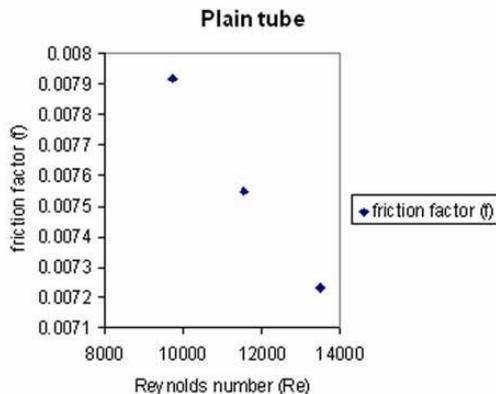**Figure 3**: Comparison of Nusselt number for plain tube

Figure 4: Variation of friction factor with Reynolds number for plain tube

The Nusselt number obtained from experimental work is compared with the value obtained using Dittus-Boelter equation (theoretical) (Figure 3). It is seen that the experimental results are in good agreement with aforementioned studies and Nu (experimental) is less than Nu (Dittus-Boelter). Actual heat carried away by air passing through the test section is the combination of convective and radiative heat transfers. As the heat transferred by convection alone is considered while performing numerical calculations (Eq. 8), it can be expected that Nu (experimental) is less than Nu (Dittus-Boelter).The experimental uncertainty is found as 7.6% for Nusselt number. From Figure 4. Friction factor is observed to reduce with increase in Reynolds number for plain tube.

III. HEAT TRANSFER CALCULATIONS

$$Ts = (T2+T3+T4+T5)/4 \qquad (1)$$

$$Tb = (T1+T6)/2 \qquad (2)$$

Equivalent height of air column, $h_{air} = (\rho w * h_w)/ \rho a$ (3)

Discharge of air, $d = C_d A_p A_o \sqrt{(2gh_{air})}/\sqrt{(A_p2 - A_o2)}$ (4)

Velocity of air flow, $U = d/Ap$ \qquad (5)

Reynolds number, $Re = UD/\nu$ \qquad (6)

(To calculate Re while using tape inserts, D_h instead of D is used)

$$Nu_{the} = 0.023\ Re^{0.8}Pr^{0.4} \qquad (7)$$

$$Q = m*C_p*(T1-T6) \qquad (8)$$

$$Qr = \sigma*A*\varepsilon_C*(Ts^4 Tb^4) \qquad (9)$$

$$H = (Q- QR)/(A (TS - TB)) \qquad (10)$$

$$Nu = h\ D/K \qquad (11)$$

(To calculate Nu while using tape inserts, Dh instead of D is used)

$$f_{the} = 0.25\ (1.82 * \log_{10}Re_D -1.64)^{-2} \qquad (12)$$

$$f = \Delta P / ((L/D)\ (\rho_a U^2/2)) \qquad (13)$$

$$\eta = (Nu_i/Nu)/ (f_i/f)^{0.333} \qquad (14)$$

IV. RESULTS AND DISCUSSION

4.1 Heat Transfer and Overall Enhancement Characteristics

The variation of Nusselt number with Reynolds number for straight tape inserts is shown in Figure 5. Highest Nusselt number was obtained for full width straight tape. The mean heat transfer gain for straight tape inserts varied from 4 to 24% compared to plain tube. This is due to strong turbulence intensity generated by tape inserts leading to rapid mixing of the flow causing heat transfer enhancement.
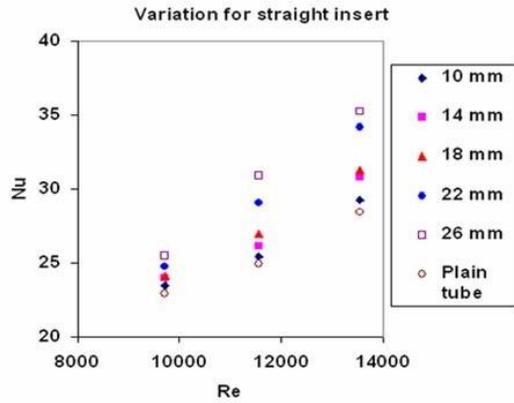
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Figure 5: Variation of Nusselt number with Reynolds number for straight tape inserts

The variations of friction factor with Reynolds number for straight tape inserts are presented in Figure 6. It is observed that the friction factor gradually reduced with rise in Reynolds number. It is observed to be maximum for full width straight tape insert.
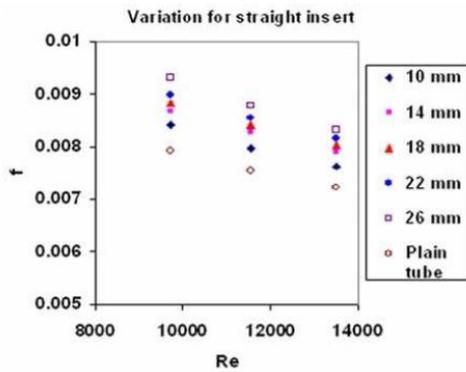

Figure 6 : Variation of friction factor with Reynolds number for straight inserts
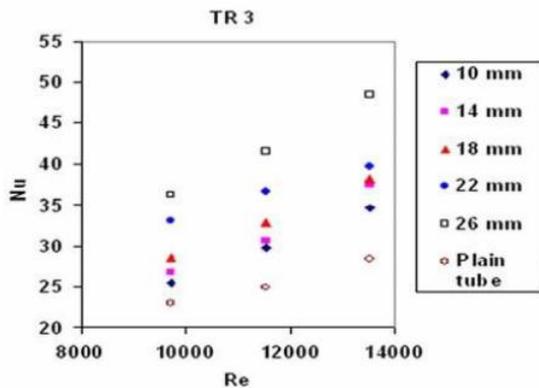

Figure 7 : Variation of Nusselt number with Reynolds number for twist ratio=3

It is evident from Figures 7, 8 and 9 that when a twisted tape is inserted into a plain tube there is a significant improvement in Nusselt number because of secondary flow, with greater enhancement being realized at higher Reynolds numbers and lower twist ratios. This enhancement is mainly due to the centrifugal forces

resulting from the spiral motion of the fluid and partly due to the tape acting as fin. It is observed that the reduction in tape width causes reduction in Nusselt numbers as well as reduction in pressure drop. From Figure 7, the percentage increase in Nusselt numbers for reduced width tapes compared to plain tube are about 11–22%, 16–31%, 24–34% and 39–44% respectively for tape widths of 10, 14, 18 and 22 mm respectively for twist ratio =3. For full width tapes, the percentage increase is observed to be 58 to 70% compared to plain tube. From Figure 8, the percentage increase in Nusselt numbers for reduced width tapes compared to plain tube are about 5–12%, 9–22%, 13–30% and 23–36% respectively for tape widths of 10, 14, 18 and 22 mm respectively for twist ratio =4. For full width tapes, the percentage increase is observed to be 36 to 42% compared to plain tube.
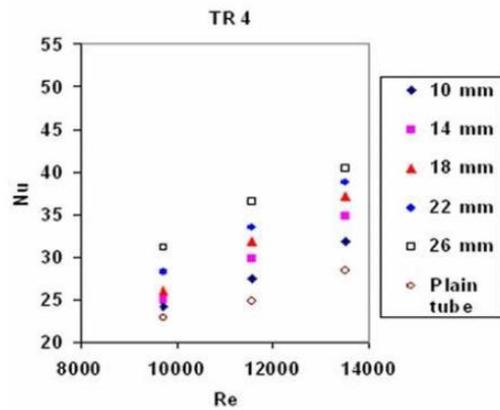

Figure 8 : Variation of Nusselt number with Reynolds number for twist ratio = 4
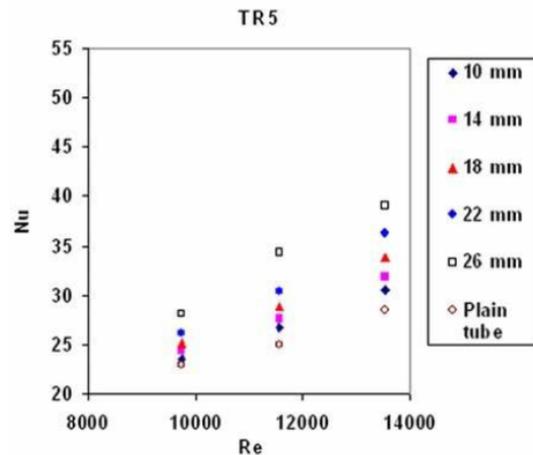

Figure 9 : Variation of Nusselt number with Reynolds number for twist ratio =5

From Figure 9, the percentage increase in Nusselt numbers for reduced width tapes compared to plain tube are about 2–8%, 6– 12%, 9–19% and 14–27% respectively for tape widths of 10, 14, 18 and 22 mm respectively for twist ratio =5. For full width tapes, the percentage increase is observed to be 22 to 37%

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compared to plain tube. The overall enhancement ratio is useful to evaluate the quality of heat transfer enhancement obtained over plain tube at constant pumping power. It is found to be more than unity for all the tape inserts used. Variations of over all enhancement ratio η against Reynolds number for twist ratios 3, 4 and 5 are shown in Figures 10, 11 and 12 respectively. It is observed that overall enhancement ratio tended to decrease gradually with the rise of Reynolds number for all twist ratios. The maximum value of over all enhancement ratio is 1.62 for full width twisted tape insert with twist ratio equal to 3.
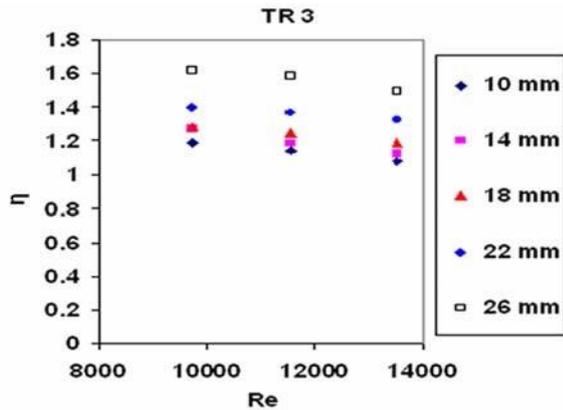

Figure 10: Variation of Overall enhancement with Reynolds number for twist ratio=3

It is seen in Figure 10 that, for tapes of widths 26, 22, 18, 14 and 10 mm curves are of decreasing order (η = 1.0–2.0) for a pitch of 82.5, 110 and 137.5 mm in the range of Reynolds number from 8000 to 14000. The tapes of width 22 mm perform more or less like full-width tapes for twist ratio equal to 3 (82.5 mm pitch). The reduction in Nusselt number for 22 mm width tape was found to be 8% compared to that of full width tape. This is comparable with the experimental results obtained by (Patil, 2000) where percentage reduction in Nusselt numbers varied from 5-9% for reduced width twisted tape inserts.

It is observed from Figure 11 and 12, that even the tapes of 10 mm width have an η value of about 1.08–1.18 times over the plain tube. The gradual reduction of these curves basically represents the same performance of reduced width tapes as that of the fullwidth tape. Thus the same performance can be achieved using reduced width tapes with 15%–61% material saving at higher Reynolds number and/or lower twist ratios. For 61% material savings (tapes of width 22 mm), the performance is 1.32–1.39 times better than for a plain tube. This is comparable with results obtained by (Patil, 2000) in which, material savings varied from 20 to 50%.
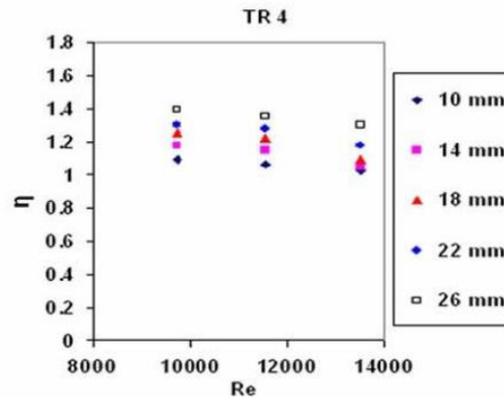

Figure 11 : Variation of Overall enhancement with Reynolds number for twist ratio=4
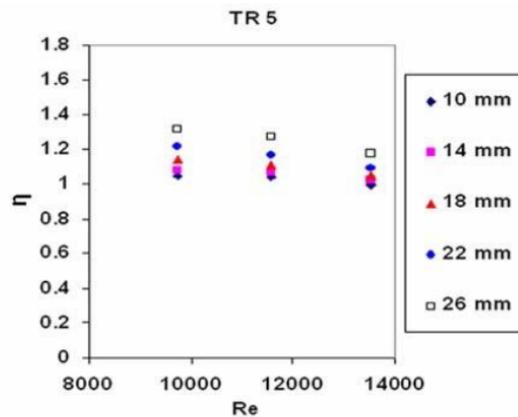

Figure 12 : Variation of Overall enhancement with Reynolds number for twist ratio=5

4.2 Heat Transfer Correlations:

The present experimental results of the Nusselt number and friction factor are correlated in terms of Reynolds number and modified twist ratio as follows:

$Nu = 0.4141*10\text{-}4 * Re0.9591 [(0.001+ H/w)\text{-}0.04645] *(Dh/L)\text{-}1.411$ (15)

Average deviation = 6.246% and standard deviation = 7.983%

$f = 0.01391* Re\text{-}0.1374 [(0.001+H/w)\text{-}0.003] *(De/L)\text{-}0.2097$ (16)

Average deviation = 2.216% and standard deviation = 2.692%

Eq. (15) and (16) are applicable when 3.17 $<H/w<$ 61, 0.02 $<D_e/L<$0.03 and 6000 $<Re<$ 13500 When D_h =D and H/w=0; Eq. 15 and 16 are applicable to plain tube also.

V. CONCLUSIONS

The study presents an experimental investigation of the potential of reduced width twisted tape inserts to enhance the rate of heat transfer in a horizontal circular tube with inside diameter 27.5mm with air as working

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fluid. The Reynolds number varied from 6000 to 13500. The effects of parameters such as modified twist ratio, Reynolds number on the heat transfer and overall enhancement ratio are studied.

The following conclusions can be drawn:

• The enhancement of heat transfer with twisted tape inserts as compared to plain tube varied from 36 to 48% for full width and 33 to 39% for reduced width-22 mm inserts. This enhancement is mainly due to the centrifugal forces resulting from the spiral motion of the fluid.

• Reduction in tape width causes reduction in Nusselt numbers as well as friction factors. The maximum friction factor rise was about 18% for 26mm and only 17.3% for reduced width inserts compared to plain tube.

• The over all enhancement ratio of the tubes with full width twisted tape inserts is 1.62 for full width–26mm and 1.39 for reduced width-22mm twisted tape insert.

• 61% material savings could be obtained for reduced width-22 mm and the performance is 1.32–1.39 times compared to plain tube.

• Nusselt numbers decreased by a maximum of 8% and 29%, for tape widths of 22 and 10 mm, respectively compared to full width twisted tape inserts. The present work is comparable with the experimental results obtained by (Patil, 2000) on laminar flow heat transfer enhancement using reduced width twisted tape inserts

• Correlations for the heat transfer coefficient and friction factor are proposed based on the present experimental data. The agreement between the results obtained from the experimental and those obtained from the proposed correlations is reasonable.

Thus the same performance can be achieved using reduced width tapes with 15% material saving at higher Reynolds number and/or lower twist ratios. Even for 61% material savings twisted tapes (of width 10.0 mm) the performance is 1.08–1.18 times better than for a smooth tube. Thus, from the considerations of enhanced heat transfer and savings in pumping power and in tape material cost, reduced-width tape inserts are seen to be attractive for enhancing turbulent flow heat transfer in a horizontal circular tube.

Future work may be extended to:

• Change the tape material from Aluminium to Copper

• Compound enhancement techniques maybe applied i.e., the tape inserts can be coupled with coil wire inserts for better enhancement

• Other reduced width twisted tapes along with variation in Reynolds numbers; and

• Develop further correlations by considering lower Reynolds numbers

Nomenclature

A convective heat transfer area (πDL), (m^2)

A_0 area of orifice, (m^2)

A_p test section inner tube area, ($\pi/4\ D^2$) (m^2) C_p specific heat of air, (J/kg K) d air discharge through test section (m^3/sec)

D_h hydraulic diameter ($4A/P$), (m)

D Inner diameter of test section, (m)

Hpitch , (mm) w width of tape insert,(mm)

H/D twist ratio

H/w modified twist ratio

f_{the} friction factor(theoretical) for plain tube f friction factor(experimental) for plain tube f_i friction factor obtained using tape inserts h experimental convective heat transfer coefficient, (W/m^{2}K) h_w manometer level difference, (m) h_{air} equivalent height of air column, (m) k thermal conductivity, (W/mK)

L length of test section, (m)

$\dot{m}$ mass flow rate of air, (Kg/sec)

Nu_i Nusselt number (experimental) with tape inserts, (hD_h/k)

Nu Nusselt number (experimental) for plain tube

Nu_{the} Nusselt number for plain tube (theoretical)

Pr Prandtl number

p pitch, (m)

P wetted perimeter, (m)

ΔP pressure drop across the test section, (Pa)

Q total heat transferred to air ($Qc + Qr$), (W)

Qc heat transferred to air by convection, (W)

Qr heat transferred to air by radiation, (W)

Re Reynolds number, ($\rho u\ D/\mu$)

$T1$, $T6$ - air temperature at inlet and outlet, (°C)

$T2$, $T3$, $T4$, $T5$ - tube wall temperatures, (°C)

T_s average Surface temperature of the working fluid, (°C)

Tb bulk temperature, (°C)

U air velocity through test section, (m/sec)

Greek symbols

v Kinematic viscosity of air, (m^2/sec)

εC emissivity of Copper

μ dynamic viscosity, (kg/m s)

η Over all enhancement ratio

ρw 3 density of water, (Kg/m)

ρa 3 density of air (Kg/m)

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