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Experimental Studies on Heat Transfer and Friction Factor Through Isothermal Square Duct with Twisted Tape Inserts

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***Abstract* – This work addresses Experimental investigations of heat transfer and friction factor characteristics in a concentric double pipe heat exchanger (square duct inner and circular tube outer) using full length twisted tapes of different twist ratios. The data were taken for Reynolds number well in the laminar region (*Re* =30-1100) with twisted tapes of twist ratios (y=2.66 and y=3.55). Experiments were carried out for constant wall temperature boundary condition using Ethylene glycol as working fluid; the results demonstrate that as twist ratio decreases, the twisted tape gives better heat transfer enhancement. Isothermal friction factors were found to be 6 to13 times the plain duct values. Mean Nusselt number for the twisted tapes are higher than those for the plain duct around 6.0 and 5.30 times for y=2.55 and y=3.66 respectively. The experimental result shows that Nusselt numbers are found to be 5.44 -7.49 and 2.46- 4.88 times the plain square duct forced convection values based on constant flow rate and constant pumping power criteria respectively, for y =2.66.**

***Keywords* – Isothermal Square duct, heat transfer enhancement, twisted tapes, laminar flow, twist ratio**

I. INTRODUCTION

Laminar flow heat transfer occurs in many process industries and it has particular importance where viscous liquids are being heated and cooled .since heat transfer coefficients in this special type of flow are low, there is need for enhancement. The process of improving the performance of a heat transfer system or increase in heat transfer coefficient is referred to as heat transfer augmentation or enhancement. This leads to reduce size and cost of heat exchanger. An increase in heat transfer coefficient generally leads to additional advantage of reducing temperature driving force, which increases second law efficiency and decreases entropy generation. General techniques for enhancing heat transfer can be divided in two categories. One is passive method such as twisted tapes, helical screw tape inserts, rough surfaces, extended surfaces, additives for liquid and gases. The other is active method, which requires extra external power, for example mechanical aids, surface fluid vibration, use of electrostatic fields. Passive methods are found more inexpensive as compared to other group. Twisted tape is one of the most important members useful in laminar flow from this group. Many studies were conducted previously to analyze heat transfer and pressure drop of both swirl generators and finned tubes.Hong and Bergles[1] studied the performance of twisted tape inserted tubes for laminar flow heat transfer and found that the Nusselt number depends on the Reynolds number .They reported that as much as threefold improvement in heat transfer rate using twisted tape insert in a tube.

Manglik and Bergles [2] correlated heat transfer and pressure drop for twisted tape inserts for uniform wall temperature conditions using water and ethylene glycol as working fluid, for laminar flow condition. Agarwal and Raja Rao[3] studied heat transfer augmentation laminar flow viscous flow by means of twisted tape inserts. The Nusselt were found to be 2.28 -5.35 and 1.21-3.7 times the plain tube forced convection values based on constant flow rate and constant pumping power respectively. They proposed correlation representing effect of friction factor on heat transfer for practical application. Pinjala and Raja Rao [4] studied pseudo-plastic type power law fluid for laminar flow in a circular pipe under constant wall temperature condition and proposed a predictive correlation to fit well all available data.

Patil [5] studied the friction factor and heat transfer characteristics of laminar swirl flow of pseudo plastic type power law fluid in a circular tube using varying width full length twisted tapes under uniform wall temperature conditions.He found that, from the considerations of enhanced heat transfer and savings in heat transfer and savings in pumping power and in tape material cost, reduced width twisted tapes are better for enhancing laminar swirl flow heat transfer. He has observed that 17-60% reduction in friction factor and 5-24 % reduction in Nusselt number for 15-50%reduction in tape-width.

Sivashanmugam and Suresh [6] also studied the laminar heat transfer and friction factor characteristics in a circular tube fitted with full-length helical screw-tapes with different twist ratios under constant heat flux conditions, including the increasing and decreasing order of twist ratio sets.They reported on a Significant improvement of the heat transfer rate for using the tape inserts and also found that there is not much change in the magnitude of heat transfer coefficient enhancement between using the increasing and decreasing twist ratio sets.

Saha and Mallik[7] reported an experimental investigation of the heat transfer and pressure drop characteristics of laminar flow of viscous oil through horizontal rectangular and square plain ducts and ducts inserted with full-length twisted tapes, short length twisted tapes, and regularly spaced twisted-tape elements, under constant heat flux boundary conditions They have studied effect of twist ratio (y), aspect ratio and tape length on heat transfer enhancement. Performance evaluation shows that short-length twisted tapes are worse and regularly spaced twisted-tape elements are better than the full-length twisted tapes.

Most of earlier works were mainly deals with heat transfer enhancement through circular tube under constant wall and heat flux boundary conditions, with Passive techniques. Little work is available on heat transfer studies through square duct under constant heat flux conditions. To date no attempt has been made on heat transfer enhancement through square duct under constant wall temperature conditions. Hence, this paper presents the enhancement of heat transfer studies in laminar flow using full length twisted tapes of different twist ratio with Ethylene glycol as working fluid.

The ranges of parameters studied for this system are as follows:

Reynolds number (*Re*) = 30-1100;
Prandlt number (*Pr*) = 60-83;
Twist ratio (*y*) = 2.66 and 3.55

Twist ratio (y) is defined as the ratio of length of one twist to the hydraulic diameter of duct.

II. EXPERIMENTAL SET UP

Details of test inserts and the experimental set up of a concentric tube heat exchanger (square duct inner and circular annulus outer) are presented in Figs.1 and 2 respectively. The double pipe heat exchanger consists of two concentric tubes; the inner square duct for Ethylene glycol flow and the outer tube for hot water flow in counter current fashion. The inside diameters of the inner and outer tubes were 21.41 and 56mm, respectively. The tubes were 2000mm long and 2 mm thick. Copper duct and steel tube were employed for the inner and outer tubes, respectively. The outer tube surface was wrapped with insulation to minimize heat loss to surroundings. Teflon gaskets are used in between flanges, with 8 mm thickness to prevent heat conduction loss to calming section and mixing section. The pressure drops across the test section were measured by using vertical U-tube manometer filled with CCl_4 (specific gravity = 1.590).Calibrated RTD PT 100 type (Teflon coated) temperature sensors of 0.1°C accuracy with digital indicator is used to measure Inlet, outlet temperature of Ethylene glycol, hot water, and wall temperatures respectively. Twisted tape was made of stainless steel strips of thickness 1 mm (t) and width 19 mm, were fabricated by twisting a straight tape, about its longitudinal axis, while other end being held under tension. This was done with the help of lathe.

III. EXPERIMENTAL PROCEDURE

In the experiments, heater was switched on for hot water tank and heated until the temperature reached 70 °C. Then, the centrifugal pump was switched on, and water flow rate to the annulus section was adjusted using by-pass valve, and test liquid flow rate was varied (1×10^{-3} m^3 /min to 15×10^{-3} m^3 /min) through a Rota meter and the inner square tube of a double pipe heat exchanger. The hot water at 0.221 Kg /sec is allowed to flow through jacket side to maintain constant wall temperature condition.

Steady state was attained within 1 hour for first run, and 20 minutes for subsequent runs. Temperatures of the inlet and outlet of the cold and the hot waters were recorded throughout the experiments and isothermal pressure drop was also measured by U-tube manometer. Typically 10 min were required to settle flow after each change of flow rate. The mass flow rate was measured by collecting the test fluid in a specified time with the help of weighing machine. Then, the experiment was conducted for plain square duct, with full length twisted tapes of, different twist ratios set. Tapes were inserted within duct at angle of 45^0 for maximum utilization of swirl effect.

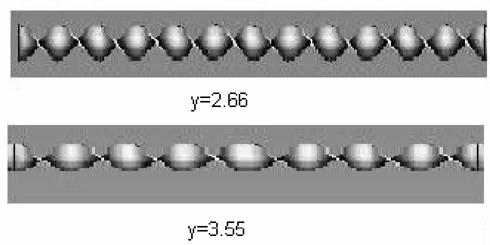

Fig. 1 Schematic diagram of twisted tape inserts

IV. HEAT TRANSFER CALCULATIONS

The heat transfer rate in the test section was calculated as

$$Q_c = m_c C_{pc} (T_2 - T_1) \quad (1)$$

The heat transfer from hot water was calculated as

$$Q_h = m_h C_{ph} (T_3 - T_4) \tag{2}$$

The average heat transfer was calculated as follows

$$Q_{avg} = 0.5(Q_c + Q_h) \tag{3}$$

Under constant wall temperature, the heat transfer rate was calculated as

$$Q_{avg} = h_i A_i (\Delta T)_{\ln} \tag{4}$$

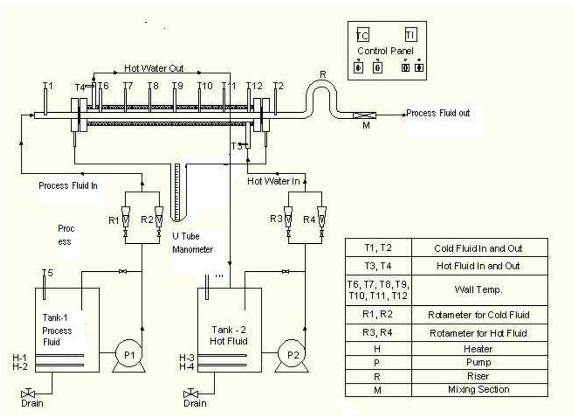

Fig. 2 Schematic diagram of experimental set up

$$h_i = \frac{Q_{avg}}{A_i (\Delta T)_{\ln}} \tag{5}$$

$$A_i = 4 D_h L \tag{6}$$

$$(\Delta T)_{\ln} = ((T_w - T_1) - (T_w - T_2)) / \ln((T_w - T_1)/(T_w - T_2)) \tag{7}$$

Mean bulk temperature (T_b) was calculated as

$$T_b = \frac{(T_1 + T_2)}{2} \tag{8}$$

$$T_w = \sum T_{wall} / 8 \tag{9}$$

Where, T_{wall} is the local wall temperature evaluated at outer wall surface of duct.

The internal convective heat transfer coefficient hi was calculated by substituting Eqs. (4), (6) and (7) in Eq. (5) and Nusselt number was calculated as

$$Nu = \frac{h_i D_h}{k} \tag{10}$$

All thermo physical properties of fluid were determined at mean bulk temperature from Eq. (8)

Percentage heat loss in the heat transfer experiments were calculated as

$$\%\text{ heat loss} = ((Q_h - Q_c)/Q_h)*100 \tag{11}$$

The heat supplied by the hot fluid into the plain square duct is found to be 3-8% higher than the heat absorbed by Cold fluid; this is due heat loss by convection and radiation from the test section to surrounding.

V. FRICTION FACTOR CALCULATION

The friction factor was calculated using vertical U –tube manometer. The fully developed friction factor was calculated from following equation

$$f = \left(\frac{\Delta p}{L}\right)\left(\frac{D_h}{2\rho V^2}\right) \tag{12}$$

Where ΔP in pressure drop across length L

The uncertainty in the data calculation was based on Steel and Coleman [8] and Kline and McClintock [9]. The maximum uncertainties of non-dimensional parameters were ± 5 % for Nusselt number and ±14% for friction factor and ± 3% in Reynolds number respectively. The uncertainty in the mass measurement has a corresponding estimated uncertainty of ± 3%. The experimental results obtained were reproducible within these uncertainty ranges.

VI. RESULTS AND DISCUSSION

Comparison between the present experimental work and correlation from the previous work of plain square duct for turbulent flow in order to standardize experimental set up are presented in Figs. 3 and 4.

A. *Validation of experimental set up*

The results obtained during experimental investigations are presented and discussed in this section. Fig. 3 shows laminar flow, isothermal friction factor for the plain duct were Compared with analytical equation, $f = 14.227/R_e$ (both f and R_e defined on the basis of hydraulic diameter).Experimental data of friction factor are matching with ± 5%.discripancy.

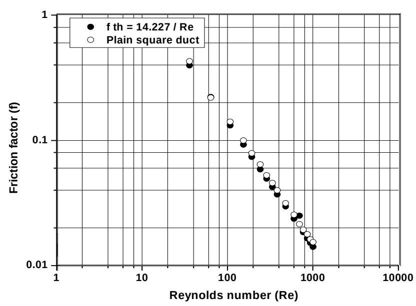

Fig. 3 Data verification of friction factor of plain duct

B. *Heat transfer result*

Fig.4 shows variation of Nusselt number with Reynolds number for plain duct for turbulent flow. Experimental results were found matching with the Dittus–Boelter equation with ± 9 %.discrepancy.

C. Heat transfer and Friction factor

Effects of insertion of full length twisted tapes of various twist ratios on friction factor and Nusselt number for square duct are presented in Figs. 5and 6. Fig.5 shows the variation of isothermal friction factor with Reynolds number for full length twisted tapes in square duct. The isothermal friction factors were found to be 6 to13 times the plain duct values. They were found to decrease with the increase in Reynolds number, and increases with decreasing twist ratio. Insertion of twisted tape of low twist, produces high swirling flow, increases flow velocity, as flow velocity is larger since motion is not in axial direction, and hence increases residence time in duct.

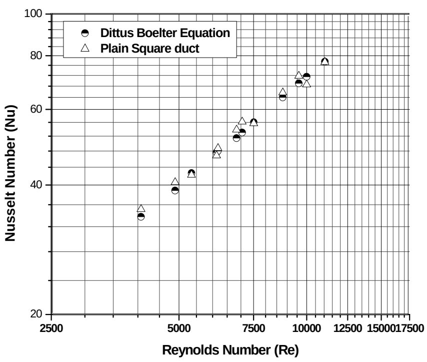

Fig. 4 Data verification of Nusselt number of plain duct

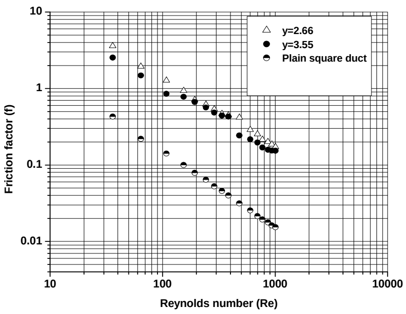

Fig. 5 Variation of friction factor with Reynolds number

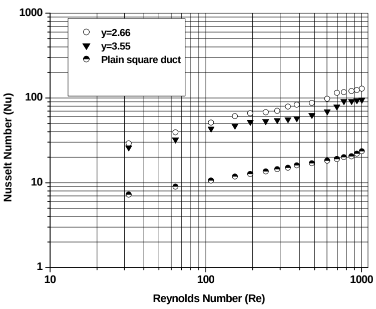

Fig. 6 Variation of Nusselt number with Reynolds number

Variation of Nusselt number with Reynolds number for various twist ratios is shown in Fig.6. It has been found that increasing Reynolds number increases Nusselt number. Increase in flow, increases forced convective current, enhancing heat transfer coefficient. For a certain Reynolds number Nusselt number increases, with decrease in twist ratio.

This can be due to swirling effect generated from the use of twisted tape, causing high temperature gradient in radial direction. Due to generated swirl flow at low twist, reduces flow cross sectional area and centrifugal force generated has significantly enhanced hear transfer rate. Mean Nusselt number for the twisted tapes are higher than those for the plain duct around 6.0 and 5.30 times for y=2.66 and y=3.55 respectively.

VII. THERMAL PERFORMANCE RATIO

The performance ratios R_1 and R_3 were then calculated using equations (13) and (14) respectively, based on actual experimental data for Nu_a. [3]

$$R_1 = \left(Nu_a / Nu_0\right)_{m,L,N,\Delta T,T_1,D_h} \tag{13}$$

$$R_3 = \left(Nu_a / Nu_o\right)_{P,L,N,\Delta T,T_1,D_h} \tag{14}$$

Variation of Thermal performance ratios with augmented Reynolds number (Re_a) for full length twisted tapes of different twist ratios are presented in Fig.7 The thermal performance ratio (R_1 at constant flow rate) increases with increasing Reynolds number for both twisted tapes. For twisted tape of y=2.66, after a Reynolds number of about 959, and for y=3.55 after Reynolds number of 980, R_1values decreases sharply. Maximum performance yields 7.49 and 6.29 times the plain square duct values., for twisted tape inserts with y=2.66 and y=3.55, at corresponding Reynolds number of 959 and 948 respectively.

On constant pumping power basis, thermal hydraulic performance ratio R_3 increases with increase in Re_a. Maximum performance yields 4.88 and 4.33 times the plain square duct values at corresponding Reynolds number of 547 and 664 for twisted tape inserts of twist ratios y=2.66 and y=3.55 respectively, further increase in Re_a, declines R_3 sharply.

Fig. 8 represents that variation of R_3 with equivalent plain duct Reynolds number (Re_o). It has been observed that with twisted tape insert of y=2.66 and y=3.55, yields maximum enhancement of 4.88 and 4.33 at corresponding equivalent plain duct Reynolds number (Re_o) 1788 and 1887 respectively, further rise in Re_o sharply declines R_3 values, due to appearance of turbulence in transition region, this may leads to increased value of Nu_o.

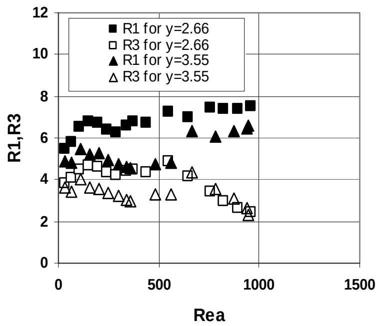

Fig. 7 Variation of R_1 and R_3 with Re_a

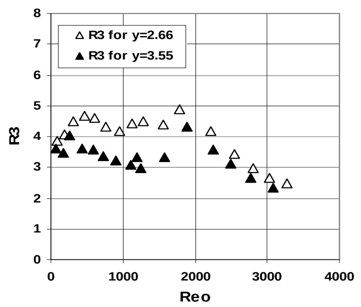

Fig. 8 Variation of R_3 with Re_o

VIII. CONCLUSION

Experimental studies on heat transfer and friction factor characteristics through isothermal square duct with twisted tape inserts having different twist ratios y=2.66 and y=3.55 were presented in this paper. Experiments were carried out for constant wall temperature condition using Ethylene glycol as working fluid; the results demonstrate the twisted tape with low twist ratio (y=2.66) not only gives better heat transfer enhancement but also increases friction factor. Isothermal friction factors with twisted tape inserts were found to be 6 to 13 times higher than plain duct values. The experimental result shows that Nusselt number were found to be 5.44 -7.49 and 2.46 - 4.88 times the plain square duct forced convection values based on constant flow rate and constant pumping power criteria respectively, for y =2.66.

NOMENCLATURE

A_0	plain duct flow cross sectional area, $(W.D)$, m^2
C_P	specific heat at constant pressure, $KJ/Kg.K$
D	depth of duct cross section, mm
D_h	hydraulic diameter of test duct $=4A_0/2(W+D)$, mm
D_0	outside diameter of duct, mm
D_1	inner diameter of annulus, mm
f	fanning friction factor, dimensionless
g	acceleration due to gravity, m^2/s
h_i	average convective heat transfer coefficient, $W/m^2\ ^oC$
Δh	differential height of vertical manometer, mm
H	axial distance for 180^o rotation of the tape, mm
k	thermal conductivity of the test liquid, W/m.k
L	length of test section, m
m_c	mass flow rate of cold liquid, Kg/s
m_h	mass flow rate of hot liquid, Kg/s
N	number of heat transfer tubes
Nu	Nusselt number (defined on hydraulic diameter), $N_u = h_i D_h/k$, dimensionless
Nu_a	augmented Nusselt number, dimensionless
Nu_0	equivalent plain duct Nusselt number, dimensionless
P	pumping power, W
ΔP	pressure drop across test section, $\Delta h(\rho_m - \rho)g$, N/m^2
Pr	Prandlt number, $C_P\mu/k$, dimensionless
Q_c	heat transfer rate of test liquid, kW
Q_h	heat transfer rate of hot water, kW
Q	average heat transfer rate, kW
R_e	Reynolds number (based on hydraulic diameter), $D_h V\rho/\mu$, dimensionless
Re_a	augmented Reynolds number, dimensionless
Re_o	equivalent plain duct Reynolds number, dimensionless
R_1	thermal performance ratio, (Nu_a/Nu_0), on constant mass flow rate, dimensionless.
R_3	thermal performance ratio, (Nu_a/Nu_0), on constant Pumping power, dimensionless.
T_1	temperature of test liquid at inlet of test section, ^oC
T_2	temperature of test liquid at outlet of test section, ^oC
T_3	temperature of hot liquid at inlet, ^oC
T_4	temperature of hot liquid at inlet, ^oC
T_w	average wall temperature of the duct, ^oC
$(\Delta T)_{ln}$	log mean temperature difference, ^oC
V	average velocity of test liquid, m/s
y	twist ratio of the twisted tape, (H/D_h), dimensionless
W	width of duct, as well as insert, mm
	Greek symbols
ρ_m	density of manometer fluid, Kg/m^3
ρ	density of test of liquid, Kg/m^3
μ	viscosity of test liquid, Kg/m.s

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