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INVESTIGATION OF TUBE SIDE HEAT TRANSFER ENHANCEMENT OF WATER USING PERFORATED DOUBLE COUNTER TWISTED TAPES WITH RECTANGULAR-CUTS

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Abstract- *This research was carried out to study the tube side heat transfer coefficient of water in a circular tube for low turbulence flow using perforated double counter twisted tapes with rectangular-cuts in the periphery. Two counter swirling inserts made up of stainless steel having length 900 mm, width 10 mm and thickness 1.5 mm were used to enhance heat transfer. The perforation diameter, width and depth of the cuts were 4mm, 6 mm and 3 mm respectively. The twist ratio, perforation ratio and depth ratio were 5, 0.4 and 0.3 respectively. The experimental test section, rapping with fiber glass insulation, was a long copper tube of 914 mm length, 26.6 mm inner diameter and 30 mm outer diameter. Flow was varied from 3L/min to 11.5L/min. Reynolds number was varied from 2968 to 11376. From this experiment it is found that heat transfer rate, Nusselt number and heat transfer enhancement efficiency is increased 1.16, 1.43~2.28 and 0.88~2.03 times respectively with the increase in friction factor 27.44~67.37% compared to plain tube.*

Keywords: Perforated Double Counter Twisted Tapes, Heat Transfer Coefficient, Enhancement Efficiency, Heat Transfer Rate, Heat Flux

1. INTRODUCTION

Heat transfer is the exchange process of thermal energy by heat exchangers, evaporators, radiators, heat sources and sinks. These devices are mostly employed for heating and cooling of liquids in power generation plants, air conditioning applications, chemical reactions and solar collector applications, thermal actions in pharmaceutical industries, sensible thermal procedures for milk pasteurization in dairy farms, cooling of mechanical and electrical devices [1]. A great number of research and experiment have been performing to developing the conditions under which heat transfer can be enhanced and to find the suitable heat exchangers for various industrial purposes. It will be great deal if efficient heat exchangers are found which will reduce the cost, energy loss and will save time by the rapid transfer of heat [2]. After the extreme effort and intense devoted experimental analysis, the researchers have found that the heat transfer performance of a heat exchanger can be significantly increased by using various augmentation techniques. Heat transfer augmentation techniques can be categorized into two types-active and passive. In the active enhancement technique, external power is used to surface vibration and rotation, fluid vibration and increasing the fluid flow rate. In the passive enhancement mode, most of the techniques are used for creating the artificial vortex flow, swirl in the vicinity of the tube walls, increasing the turbulence of the fluid flow, destroying continuously the boundary layer formation and increasing the conductivity of the fluid mixing with additives/nanoparticles [3]. At present, effort has been paid to heat transfer enhancement by means of swirl strips, twisted tapes and meshes in heat exchanger tubes. These tube inserts are considered to enhance convective heat transfer by creating one or more these combinations: 1) Continuous interrupting the development of boundary layer of the fluid flow and increasing the degree of flow turbulence. 2) Continuously increasing the effective heat transfer area. 3) Continuously generating secondary flow [4] Inserts improve the heat transfer coefficient capability and minimize the loss of friction factor. Tube inserts are utilizing in heat transfer enhancement and fouling mitigation in different industries such as petroleum refineries and chemical plant for many years. Due to rapid growth of industrialization the research on this topic has been enlarged significantly. Nowadays it is one of the most important topics to enhance heat transfer using inserts. Many experiments have been going on to increase the convective heat transfer for various types of inserts. The modifications of the twisted tape inserts also provide extra advantages in the enhancement of the inserts performance [5]. In the recent years, considerable emphasis has been placed on the development of various augmented heat transfer surfaces and devices. These can be seen from the exponential increase in world technical literature published in heat transfer augmentation devices, growing patents and hundreds of manufacturers offering products ranging from enhanced tubes to entire thermal systems incorporating enhancement technology [6].

Saha et al. [7] experimentally investigated the effect of regularly spaced twisted tape in heat transfer and pressure drop characteristics in a circular tube. They found that the Reynolds number, Prandtl number, twist ratio, space ratio, tape-width, rod-diameter and phase angle lead the heat transfer characteristics. They concluded that poor heat transfer from tape width reduction and there is no effect of higher than zero phase angle; rather it creates manufacturing complexity. Esmaeilzadeh et al. [8] conducted the experimental study on heat transfer and friction factor characteristics using different thicknesses of twisted tape with γ-Al_2O_3 nanofluid and concluded that both the heat transfer rate and friction factor increase with the increase of thickness of the twisted tape. Piriyarungrod et al. [9] analyzed the effects of taper angle and twist ratio in heat transfer augmentation. They used four different taper angles, θ = 0.0°, 0.3°, 0.6° and 0.9° and three different twist ratios (y/W) of 3.5, 4.0 and 4.5. They reported that both heat transfer performance and friction factor increase with the decreasing of taper angle and twist ratio. Bhuiya et al.[10] worked with the triple helical tapes with different helix angles, α = 9°, 13°, 17°, and 21° and examined heat transfer enhancement for turbulent flow. They reported that Nusselt number, effectiveness and friction factor has increased up to 4.5, 3.45 and 3.0 times, respectively, compared to the plain tube.

Salam et al. [11] carried out the experimental investigation to measure tube side heat transfer coefficient, friction factor and heat transfer enhancement efficiency of water using rectangular cut twisted tape. They varied Reynolds number in the range 10000-19000 and heat flux varied 14 to 22 kW/m2 without insert; 23 to 40 kW/m2 with insert. From the study, they presented that the Nusselt number increased by 2.3 to 2.9 times and friction factor increased by 1.4 to 1.8 times over the smooth tube. They also concluded that the heat transfer performance enhanced with the increase of Reynolds number. Eiamsa-ard et al. [12] investigated the heat transfer, friction loss and thermal performance factor using nine different peripherally-cut twisted tapes. The experiment was performed with constant twist ratio (y/W = 3.0) and different three tape depth ratios (DR = d/W = 0.11, 0.22 and 0.33), each with three different tape width ratios (WR = w/W = 0.11, 0.22 and 0.33). They showed that the peripherally cuts generate higher turbulence intensity in the vicinity of tube wall. The heat transfer rate and friction factor found with the peripherally-cut twisted tapes were significantly higher than those in the tube fitted with the typical twisted tape and plain tube, especially in the laminar flow regime. The obtained results also demonstrated that as the depth ratio increased and width ratio decreased, the heat transfer enhancement increased. Arya et al. [13] investigated the thermal performance of a corrugated plate heat exchanger working with MgO/ethylene glycol nanofluid. They found that the heat transfer was enhanced with the enhancement of flow rate and mass concentration of nanoparticles. They presented that 34% enhancement was achieved by the use of MgO nanoparticles.

Thianpong et al. [14] experimentally investigated the effects of perforated twisted tape with parallel wings in heat transfer and pressure drop characteristics in a heat exchanger. This type of insert involves: (1) the perforation in the center of the insert, reduce the pressure drop and (2) wings produce extra turbulence near the tube walls. By this experiment they revealed that the maximum thermal performance factor 1.32 was increased compared to plain tubes and typical tapes. They also visualized the swirling/flow patterns of the flow stream using dye injection technique. Yaningsih et al. [15] carried out the investigation to check the effect of double-sided delta-winglet tape (DWTs) on heat transfer and friction factor. Three blockage ratios (Rb = winglet-height, b/ inner tube diameter, di) of 0.28, 0.35 and 0.42 were taken to investigation and they presented that Nusselt number and friction factor increased by 364.3% and 15.5 times, respectively, using the DWTs. This work also showed that the thermal performance can be intensified by the increase of Rb and DWTs are proposed as a favorable insert device. Choudhari et al. [16] worked with coil wire inserts to study heat transfer and friction factor characteristics in a horizontal double pipe heat exchanger. Three different materials of copper, aluminum and stainless steel with three different pitches 5, 10, and 15 mm respectively were used. They found that coil wire has enhancement effect on heat transfer and friction factor. They resulted that heat transfer increased for copper, aluminum and stainless steel by 1.58, 1.41 and 1.31 respectively and friction factor increased with the increase of coil wire pitch.

Abdolbaqia et al. [17] experimentally carried out the investigation to explore the influence of counter twisted tapes (CTT) and co-twisted tapes (CoTT) on heat transfer rate (Nu), friction factor (f) and thermal enhancement index (η). The investigation showed that Nusselt number (Nu), friction factor (f) and thermal enhancement index (η) was increased with decreasing of twist ratio. The CTT is more effective than the CoTT and plain tube for heat transfer enhancement and heat transfer enhanced by 22.5% and 61% compared to CoTT and plin tube respectively. Everts et al. [18] presented that the increasing of surface roughness in the transitional flow regime also increase the heat transfer. Sreedhar Rao et al. [19] performed the experimental investigation to enhance heat transfer using corrugated rough surface of three different angles 300, 400 and 500. They concluded that the heat transfer was increased with the increasing of corrugation angle.

In this experiment, a new configuration of inserts named perforated double counter twisted tapes with rectangular cuts in periphery is used to enhance heat transfer of water in circular tube.

2. METHODOLOGY

2.1 EXPERIMENTAL SETUP

This experiment was dealt with two perforated counter twisted tapes with periphery cuts. The length of the twisted tapes was 900 mm, width was 10 mm and thickness was 1.5 mm. The perforation diameter was 4 mm and the width and depth of the periphery cuts were 6 mm and 3 mm respectively. The twist ratio, perforation ratio and depth ratio were 5, 0.4 and 0.3 respectively. Figure 2.1 shows the twisted tapes.

Fig 2.1: Double counter twisted tapes

The test section was 914 mm long copper tube having 26.6 mm internal diameter and 30 mm outer diameter of which 900 mm was used for this experiment. A continuous cold water supply was maintained by a centrifugal pump (capacity 0.5 hp) from a reservoir tank. A voltage regulator was used to maintain the constant heat and 120 volt was used for heating the test section. The heating wire was covered with fiber glass for insulation. Five T-type thermocouples (four was functioning) were placed on five equally spaced points of the test section to measure the outer surface temperature of the tube. Inlet and outlet pressure drop was measured by a manometer and the flow rate of water through the tube was measured by a rotameter whose capacity was 27.5L/min. Inlet and outlet temperature of water was measured by two thermometers.

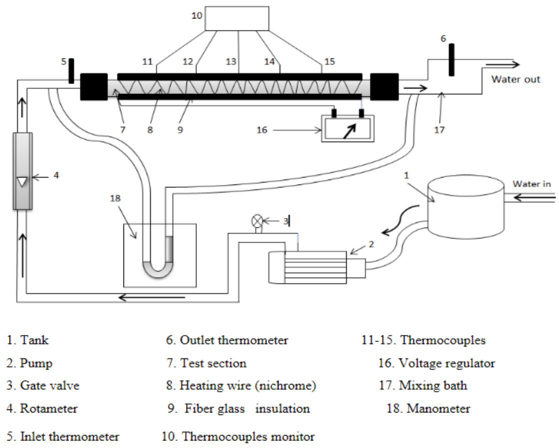

Fig 2.2: Schematic diagram of experimental setup

All the data were taken at steady state condition and it was taken about 8-15 minutes to get the steady state condition. The same procedures were applied for taking data in plain tube as well as with twisted tapes.

Fig 2.3: Photographic view of experimental setup

2.2 DATA COLLECTION

Table 2.1: Rotameter, manometer and temperature readings for plain tube

Ro	Ma	T_i	T_o	T_1	T_2	T_3	T_4
3	0.15	31.5	34.5	46	47	47	47
3.3	0.17	31.5	34.4	46	46	46	46
3.7	0.2	31.5	34.3	45	45	46	46
4.1	0.22	29.2	31.9	42	43	43	43
4.5	0.24	29.2	31.8	41	42	42	43
5.5	0.28	29.2	31.5	41	41	41	42
6.5	0.31	29.2	31.2	39	40	40	41
7.5	0.34	29.2	31	38	39	39	40
8.5	0.38	29.2	30.8	37	38	38	39
9.5	0.43	29	30.5	36	38	37	38
10.5	0.49	29	30.4	36	37	37	37
11.5	0.55	29	30.3	35	36	36	37

Table 2.2: Rotameter, Manometer and Temperature Readings for Inserts

Ro	Ma	In	Out	T_1	T_2	T_3	T_4
3	0.25	29	32.5	41	42	42	43
3.3	0.28	29	32.3	40	41	41	41
3.7	0.32	29	32.1	39	40	41	40
4.1	0.36	29	32	38	39	40	40
4.5	0.39	29	31.8	37	38	38	39
5.5	0.45	29	31.5	36	37	38	38
6.5	0.5	29	31.2	36	36	36	37
7.5	0.56	29	31	35	35	35	36
8.5	0.6	29	30.8	34	34	35	35
9.5	0.63	29	30.7	33	34	34	35
10.5	0.67	29	30.6	33	33	34	34
11.5	0.7	29	30.5	32	33	33	34

3. MATHEMATICAL FORMULAS

Data for heat transfer enhancement of water using inserts in circular tube were calculated by using the following equations:

Outer surface area, $A_o = \pi\, d_o\, L$ (3.1)

Inner surface area, $A_i = \pi\, d_i\, L$ (3.2)

Cross sectional area, $A_c = \pi d_i^2/4$ (3.3)

Friction factor, $f_{ex} = (2\Delta p d_i)/(\rho L u_m^2)$ (3.4)

Mean velocity, $u_m = \dot{m}/A_c$ (3.5)

Pressure difference, $\Delta p = \Delta h \times \rho \times g \times 13.6$ (3.6)

Added heat, $Q = \dot{m} C_p (T_o - T_i)$ (3.7)

Reynolds Number, $Re = \rho u_m d_i/\mu$ (3.8)

Nusselt number, $Nu_{ex} = h d_i/k_w$ (3.9)

Gnielinski correlation for theoretical Nusselt number,

$$Nu_{th} = \frac{\frac{f}{8}\times(Re-1000)\times Pr}{1+12.7\times(\frac{f}{8})^{\frac{1}{2}}\times(Pr^{\frac{2}{3}}-1)} \qquad (3.10)$$

Prandtl number, $Pr = \mu C_p/k_w$ (3.11)

Heat transfer coefficient, $h = Q/A_i(Twi-T_b)$ (3.12)

Heat flux, $q = Q/A_i$ (3.13)

Petukhov correlation for theoretical friction factor,
$f_{th} = (0.79 \ln Re - 1.64)^{-2}$ (3.14)

Bulk temperature, $Tb = (T_i + T_o)/2$ (3.15)

Outer wall temperature, $Two = \sum_{k=1}^{5} \frac{T_{wo,k}}{5}$ (3.16)

Inner wall temperature, $Twi = Two - Q\frac{\ln(d_o - d_i)}{2\pi k_{cu} L}$ (3.17)

Efficiency, $\eta = \left|\frac{Nu_i}{Nu_p}\right| = \left|\frac{h_i}{h_p}\right|$ (3.18)

4. RESULTS AND DISCUSSION

4.1 HEAT FLUX

At first, the Nusselt number was determined and compared with theoretical Nusselt number by using Gnielinski correlation Eq. (3.10). The average error for plain data was found 18% compared to Gnielinski correlation. The Reynolds number was calculated using Eq. (3.8) and was varied in the ranges 2968 to 11376 for both the plain tube as well as inserts.

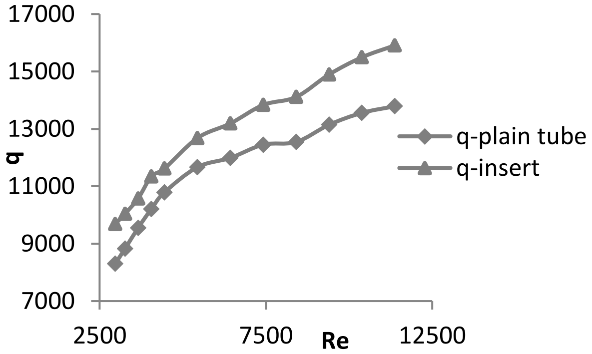

Fig 4.1: Variation of heat flux with Re for with & without inserts

It is found that heat transfer rate for plain tube varies in the range 624 ~ 1037 W and for inserts it is seen in 728 ~ 1196 W. Heat flux was calculated using Eq. (3.13) and Figure 4.1 shows the variation of heat flux with Reynolds numbers for plain tube as well as inserts. It is found that heat flux increases with the increasing of Reynolds number for both cases. For Reynolds number 2968~11376, heat flux varies from 8300 ~ 13788 W/m^2 for plain tube and 9684 ~ 15909 W/m^2 for insert. Heat flux is increased average 12% in insert compared to plain tube and it is found that with the increasing of Reynolds number heat flux increased more rapidly for using insert. Convective heat transfer coefficient was determined from Eq. (3.12) and found in the range 605 ~ 2191 W/m^2 k for plain tube and 863 ~ 4998 W/m^2 k for inserts. Heat transfer coefficient was increased 1.4 to 2.16 times for inserts compared to plain tube.

4.2 NUSSELT NUMBER

Experimental Nusselt numbers were obtained from Eq. (3.9) and Figure 4.2 shows the variation of Nusselt number for this experimental study. Experimental Nusselt number for plain tube is compared with the Nusselt number calculated by Gnielinski correlation. Nusselt number for plain tube varied average 18% with theoretical values. Nusselt number for plain tube is increased from 26 to 94 with the increasing of Reynolds number 2968 to 11376.

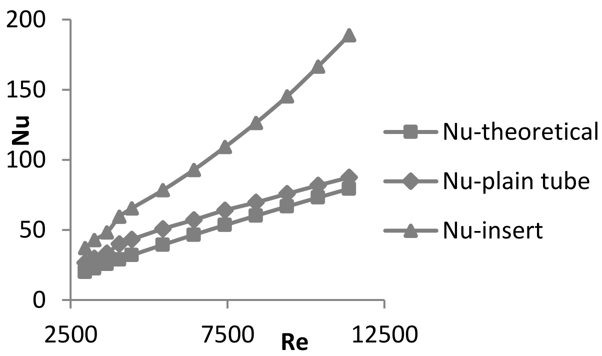

Fig 4.2: Variation of Nu with Re for with & without inserts

In insert, Nusselt number is found 37 for Re 2968 and increased to 214 when Reynolds number is increased to 11376. Nusselt number is increased 1.4 to 2.2 times with the increasing of Reynolds number. Nusselt number is increased average 109% in insert compared to plain tube. In the transition region (Reynolds number 3000-4000) Nusselt number is increased average 85% and after transition period Nusselt number is increased average 116%.

4.3 FRICTION FACTOR

The theoretical friction factor was determined from Petukhov correlation and Figure 7.3 shows the variations in friction factor with Reynolds number. Friction factor decreases with increasing Reynolds number.

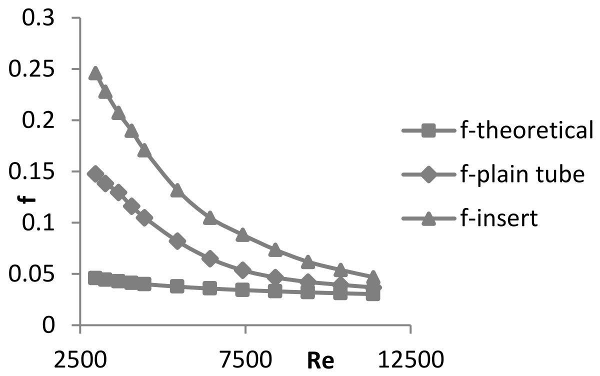

Fig 4.3: variation of friction factor with Re for with & without inserts

Friction factor varies for plain tube from 0.147 to 0.0368 and for insert 0.246 to 0.0469 with Reynolds number. In insert friction factor is increased 27.44% to 67.37% compared to plain tube.

4.4 ENHANCEMENT EFFICIENCY

The heat transfer enhancement efficiency (ɳ) is calculated for equal pumping power [11]. Using equal pumping power,

$$(\bar{Q}\times\Delta P)_{plain} = (\bar{Q}\times\Delta P)_{insert} \quad (4.1)$$

$\bar{Q}$ = volume flow rate

ΔP = pressure difference

$$(f\times Re^3)_{plain} = (f\times Re^3)_{insert} \quad (4.2)$$

The correlation for experimental Nusselt number and friction factor for plain tube are-

$$Nu_p = 0.028Re_p^{0.8}Pr^{0.32} \quad (4.3)$$

$$f_p = 425Re_p^{-0.9999} \quad (4.4)$$

The error for Nusselt number compared to plain tube and correlation is in the range of -10.8% to 10% having r.m.s value 5.46% and for friction factor in the range of -10.2% to 8.5% having r.m.s value 6.69%.

The correlation for experimental Nusselt number and friction factor for inserts are-

$$Nu_i = 0.00062Re_i^{1.36}Pr^{0.067}(\frac{y}{w})^{-0.034} \quad (4.5)$$

$$f_i = 22.5Re_i^{-1.142}(\frac{y}{w})^{2.86} \quad (4.6)$$

The error for Nusselt number compared to insert and correlation is in the range of -7.3% to 7.5% having r.m.s value 5.94% and for friction factor in the range of -10.4% to 11.7% having r.m.s value 7.13%.

The enhancement efficiency,

$$ɳ = 0.064Re_i^{0.619}Pr^{-0.253}(\frac{y}{w})^{-1.178} \quad (4.7)$$

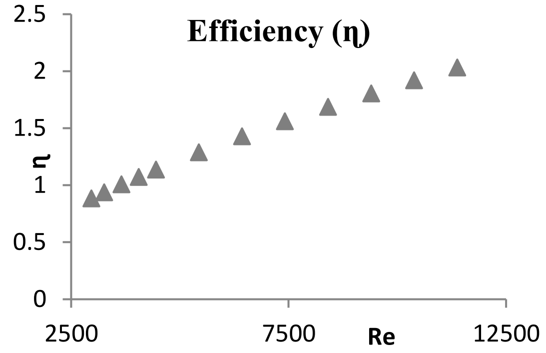

Fig 4.4: Thermal enhancement efficiency

The heat transfer enhancement efficiency is determined using Eq. (4.7). The enhancement efficiency is increased with the increase in Reynolds number. Figure 4.4 shows the enhancement efficiency for insert. Enhancement efficiency is increased 0.88 to 2.03 times with the increase in Reynolds number from 2968 to 11376.

5. CONCLUSION

From this experimental study following results are found for inserts compared to plain tube:

- Nusselt number is increased 2.2 times for double counter twisted tapes compared to plain tube
- The friction factor is increased 67.37%
- Heat flux has increased 12%
- The heat transfer enhancement efficiency is increased 2.03 times

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7. NOMENCLATURE

Symbol	Meaning	Unit
d_i	Tube inner diameter	m
d_o	Tube outer diameter	m
L	Tube length	m
Ro	Rotameter reading	L/min
Ma	Manometer reading	m
A_o	Tube outer surface area	m^2
A_i	Tube inner surface area	m^2
A_c	Tube cross sectional area	m^2
T_{wo}	Tube outer wall temperature	K
T_{wi}	Tube inner wall temperature	K
T_i	Cold water temperature	K
T_o	Outlet temperature	K
T_b	Bulk temperature	K
$T_{1,2,3,4}$	Thermocouples temperature	K
$\dot{m}$	Mass flow rate of water	Kg/s
$\bar{Q}$	Volume flow rate	m^3/s
U_m	Mean velocity	m/s
k	Thermal conductivity	W/mK
ρ	density	Kg/m^3
μ	Dynamic viscosity	Kg/m-s
C_p	Specific heat	J/kgK
Δp	Pressure difference	N/m^2
Q	Heat transfer rate	W
q	Heat flux	W/m^2
h	Heat transfer coefficient	W/m^2K
Nu	Nusselt number	Dimensionless
Pr	Prandtl number	Dimensionless
f	Friction factor	Dimensionless
Re	Reynolds number	Dimensionless
η	Enhancement efficiency	Dimensionless