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An Experimental Study of Heat Transfer Enhancement in Heat Exchangers

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Abstract:
This paper presents an experimental study of heat transfer enhancement and friction factor characteristics in turbulent swirl flow generated by a short helical tape placed at the entrance (inner tube) of double pipe heat exchanger. Experiments were conducted for cold water flow rates in inner tube in the range of 3432≤ Re ≤5108. Experiments were carried out for three helical tapes with different lengths (LH) of 130mm, 270mm, and 370mm. Also, experiments were carried out for two successive helical tapes of length 135 mm at different free space length (Ls). The local and average heat transfer coefficients and pressure drop were measured and compared with the base case of plain tube. The local heat transfer coefficients were found to be increasing to very high values in the region of the helical tape, and then decreasing along the downstream of the helical tape. The thermal performance factor increases with increasing helical tape length at each Reynolds number examined and decreases with increasing Reynolds number in the range of 3432≤ Re ≤5108. The maximum performance factor of η=1.45 was obtained for the length of 370mm at Re=3432. The helical tape of length LH=370 provides the highest average heat transfer rate about 27% over the plain tube but it increased the pressure drop by 2.6 %. To overcome the increasing of pressure drop, different free-spacing ratio (S= Ls/LH) of 1.8, 3.6, and 5.4 between two successive helical tape of length LH=135mm at Re=4083 were examined. It was found that the thermal performance factor of two successive helical tapes of length LH=135mm with free space ratio (S=1.8) is better than the continuous helical tape of length LH=270mm at Re =4083. A performance comparison between a present work and published results of full length helical and twisted tapes inserts has shown superior heat transfer enhancement using short length helical tapes inserted at the cold water tube as compared to full length twisted tapes in the Reynolds number range of Re=3432–5108.

Keywords: *Heat exchanger, heat transfer, pressure drop, swirl flow devices, helical tape inserts, average Nusselt numbers, average friction factor*

1. Introduction

In the past decade, heat transfer enhancement technology has been developed and widely applied to heat exchanger applications; for example, refrigeration, automotive, process industry, solar water heater, etc., Smith and Pongjet [1]. The aim of augmenting heat transfer is to accommodate high heat fluxes (or heat transfer coefficient). To date, there have been a large number of attempts to reduce the size and costs of heat exchangers. The most significant variables in reducing the size and cost of a heat exchanger are basically the heat transfer coefficient and pressure drop. In general, heat transfer enhancing technologies can be divided into two groups; Raj Mitra Manglik [2].One is the passive method, without using any external power such as a surface coating, rough surfaces, extended surfaces, swirl flow devices, the convoluted (twisted) tube and additives for liquid and gases. The other is the active method, which requires extra external power sources, for example, mechanical aids, surface fluid vibration, injection and suction of the fluid, jet impingement, and use of electro static fields.

The swirl flow devices can be classified into two kinds: the first is the continuous swirl flow and the other is the decaying swirl flow. For the continuous swirl flow, the swirling motion persists over the whole length of the tube for example twisted-tape inserts, [3:8] and wire coil insert, Alberto et al. [9].

Liu et al. [10], reported comprehensive review on heat transfer enhancement involving a helical and twisted tape inserts, and conducted that variously developed twisted tape inserts are popular researched and used to strengthen the heat transfer efficiency for heat exchangers. Helical tape inserts. Smith et al. [1], reported that, the enhancement of the heat transfer in a tube with regularly spaced and full-length helical tape swirl generators, and concluded that the full-length helical tape with rod provide the highest heat transfer rate about 10% better than without rod. H.Gul et al. [11], founded that, the enhancement of the heat transfer in circular tubes using helical swirl generator insert at the entrance, and concluded that the helical tape can help to increase the heat transfer rate up to *20%* depending on momentum ratio and *Re* at constant pumping power. Nagarajan et al [12], developed the heat transfer enhancement studies in a circular tube fitted with right-left helical inserts with spacer, and concluded that the performance ratio increases with increasing Reynolds number and decreasing twist ratio. While in the decaying swirl flow, the swirl is generated at the entrance of the tube and decays along the flow path. For example the radial guide vane swirl generator, the tangential flow injection device successfully used tangential injection, to obtain swirl flow for the enhancement of critical heat flux. Change et al. [13], obtained an average enhancement of *35%* to *40%* in heat transfer on a constant pumping power basis. For the decaying swirl flow, the heat transfer coefficient and pressure drop decrease with the axial distance, while for the continuous swirl flow, the heat transfer coefficient and pressure drop are rather constant. P. Sivashnmugam et al [14], presented the experimental studies on heat transfer and friction factor characteristics of turbulent flow through a circular tube fitted with helical screw-tape inserts, and concluded that the performance of the helical twist insert is better than twisted tape performance, and presented a correlation for predicting Nusselt number and friction factor. Lishan et al. [15], developed a computational modeling of laminar swirl flows and heat transfer in circular tubes with twisted – tape inserts, and conducted that the consequent well-mixed helical swirl flow significantly enhances the convective heat transfer.

From the previous experimental work, the twisted-tape insert is extensively used in enhancement heat transfer of heat exchangers, Smith et al. [1]. However, there are restrictions in the case of a helical tape inserts summarized as a follows:

- The electrical heater is used as the heat source; however in practical engineering applications the tubular type heat exchanger is most commonly used.
- The air is used as a test fluid, however in cooling and condensing processes, the water is used.
- The full length of helical tape inserts is used, that is caused high pressure drop. To overcome this, a short length helical tape with free spaces is used.

From the preceding review, it is amply evident there are a large number of experimental works carried out by researchers to investigate the thermo-hydraulic performance of various twisted tapes including the traditional simple twisted tapes, regularly spaced twisted tapes, varying lengths twisted tapes, tapes with different cut shapes, tapes with baffles and tapes with different surface modifications.

There is a need for a comprehensive experimental work covering a wide range of parameters including helical tape length, spaced taps, and the Reynolds number. Also, there is no comparative study of heat transfer enhancement using entrance only and short length helical inserts. To meet this critical need for process heat exchanger applications, the objective of the present study is to investigate the effect of a short helical tape inserts with varying length and different free spaces on heat transfer enhancement and pressure drop. In the present work the tubular type heat exchanger is used with water as a test fluid to simulate a practical engineering applications and Industrial processes, geometrical configuration of helical tape is used as an insert that has low manufacturing cost.

2. Experimental Setup and Procedure

A schematic diagram of the experimental apparatus is shown in Fig 1. The test loops consists of a test section, hot water loop, cold water loop, instrumentation and control valves. The test section and the connections of the piping system are designed such that parts can be changed or repaired easily. The cold water loop consists of a cold water tank of 0.5 m^3 capacity, a circulating pump, two air cooler units in series, and the shell side (annular space) of test section. The flow rates of the cold water is controlled by adjusting the valves on, the main water line and the by-pass line .The hot water closed loop consists of a hot water tank of 0.5 m^3 capacity, a circulating pump, and the tube side of test section. The flow rate of the hot water is controlled by adjusting the valves on the main water line and the by-pass line. In order to provide fine control of the hot water temperature, thermostat is connected with electric heater of 3 KW powers that is immersed inside the hot water tank.

Experiments have been performed using three different helical tape inserts at the entrance of inner tube. The tape inserts lengths are LH = 130, 270, and 370mm. were used. Also, experiments were carried out for two successive helical tapes of length 135 mm at different free spacing's (Ls), as seen in Figs 2 a, b. The experiments were carried out at the same inlet conditions with the Reynolds number, (Re) in a range of 3432–5108. Fig 3. Shows a sketch of a helical tape inserts used in this test. In the experiments, the geometric conditions of the helical tape inserted were kept constant. The helical tape was made of stainless steel and has the geometric dimensions of W= 25mm (0.98 d), P= 10mm (0.39 d), t= 1mm (0.039 d).

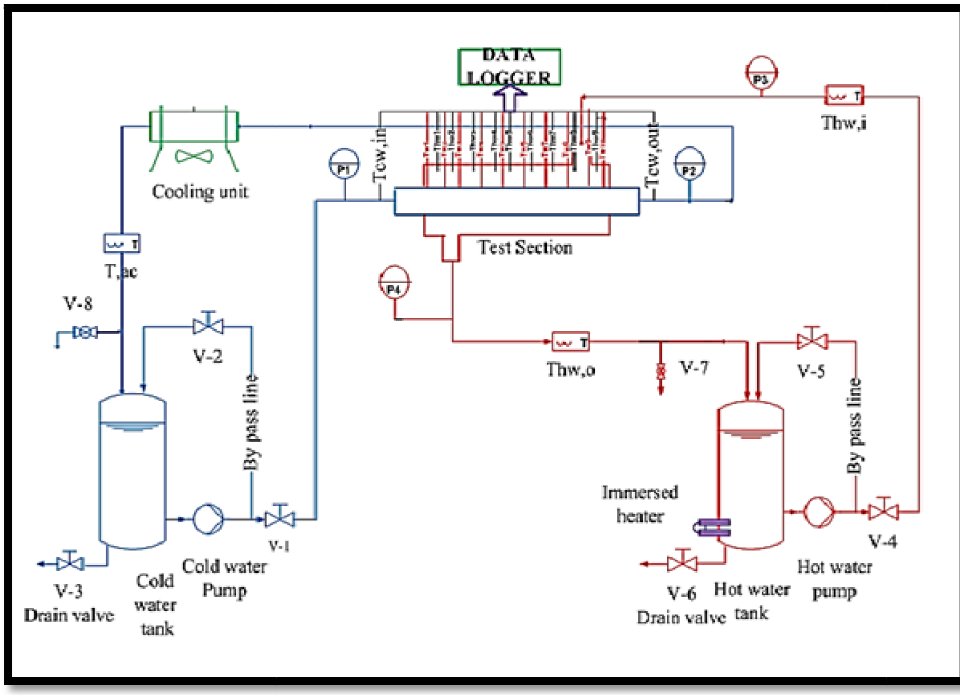

Figure 1: Schematic Diagram of Experimental Apparatus

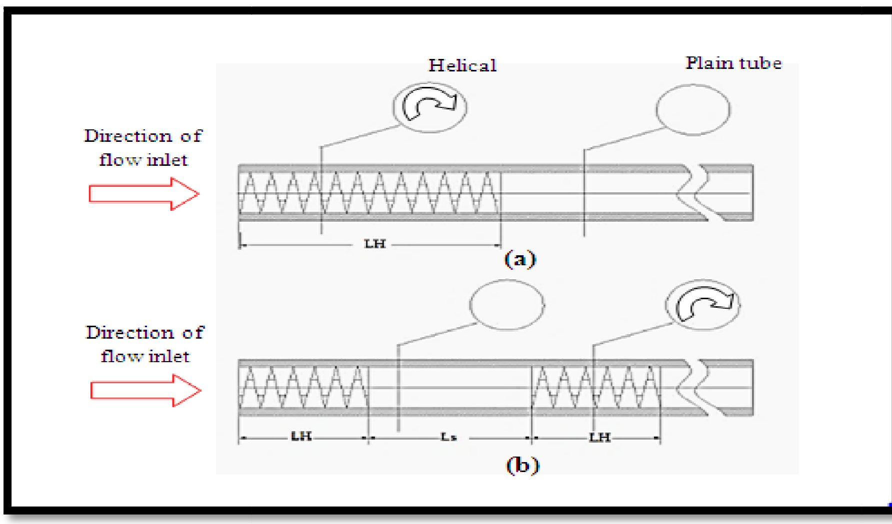

Figure 2: Helical Tapes Insert Geometry: (A) Single Insert; (B) Free Spaced Helical Tape Inserts

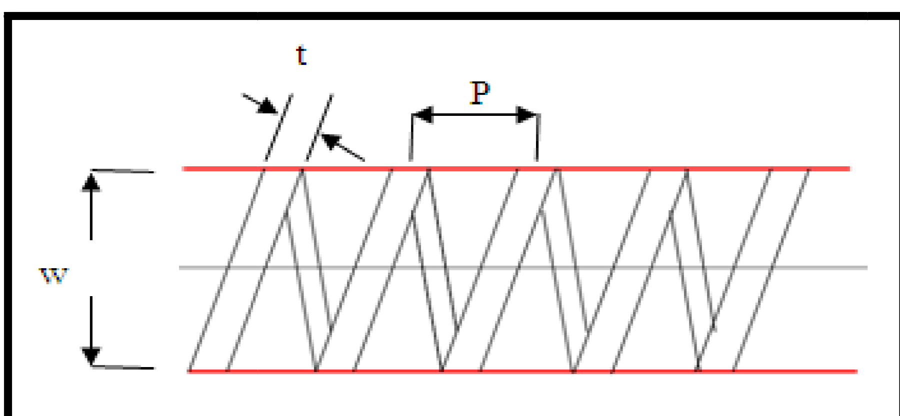

Figure 3: Sketch of Helical Tape Fitted Inside the Inner Tube

Test section details are depicted in Fig 4. The test section is double tube counter flow heat exchanger where the hot water flows through the shell (outer tube) and the cold water flows inside the inner tube. Copper and carbon steel tubes were employed for the inner and outer tubes, respectively. Outer tube surface and piping system are thermally insulated by asbestos to minimize heat loss to surroundings. Ten and nine types K - thermocouples are installed to measure the local wall and hot water temperatures, respectively as shown in Fig 5.

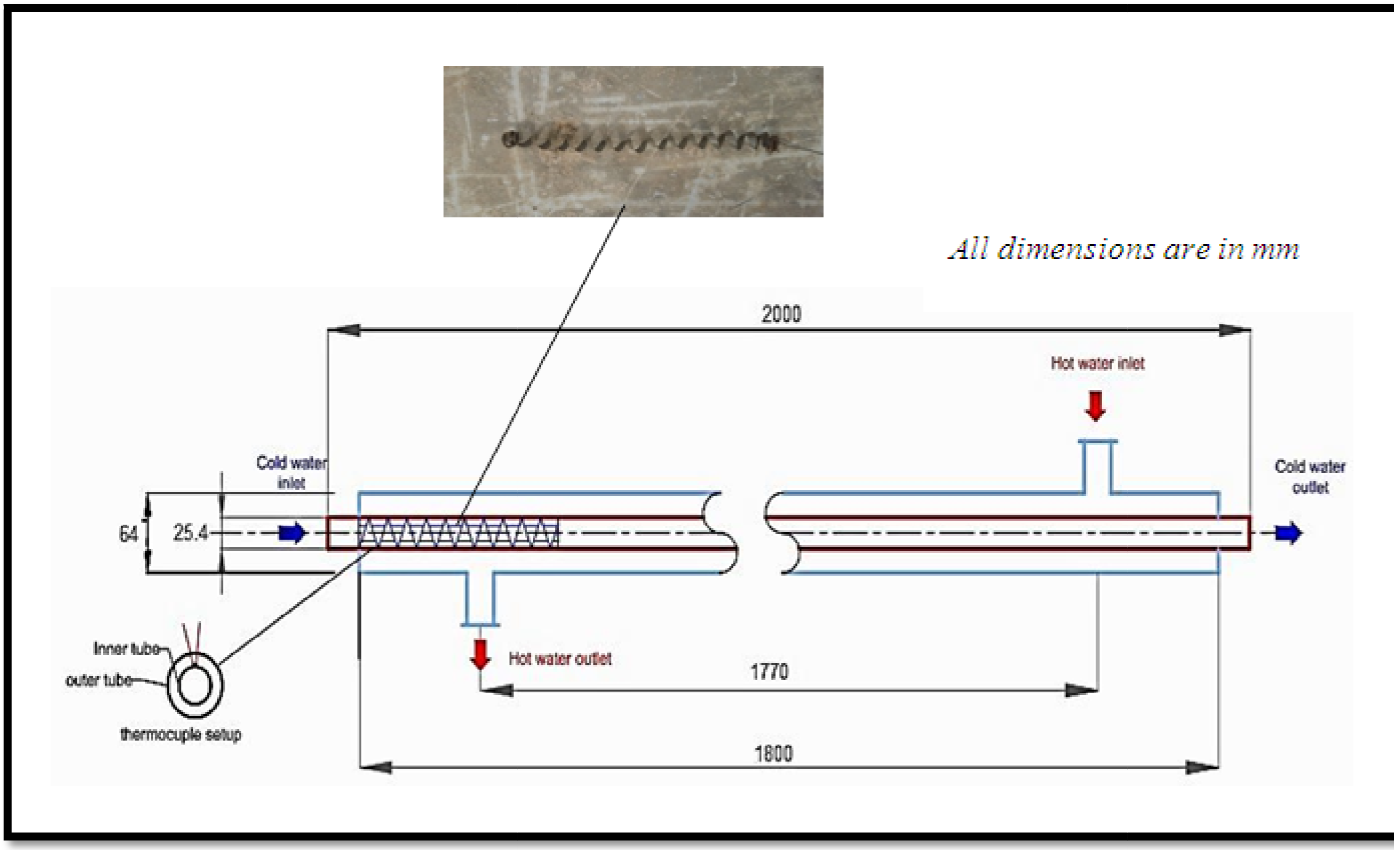

Figure 4: A Concentric Tube Heat Exchanger Fitted With a Helical Tape

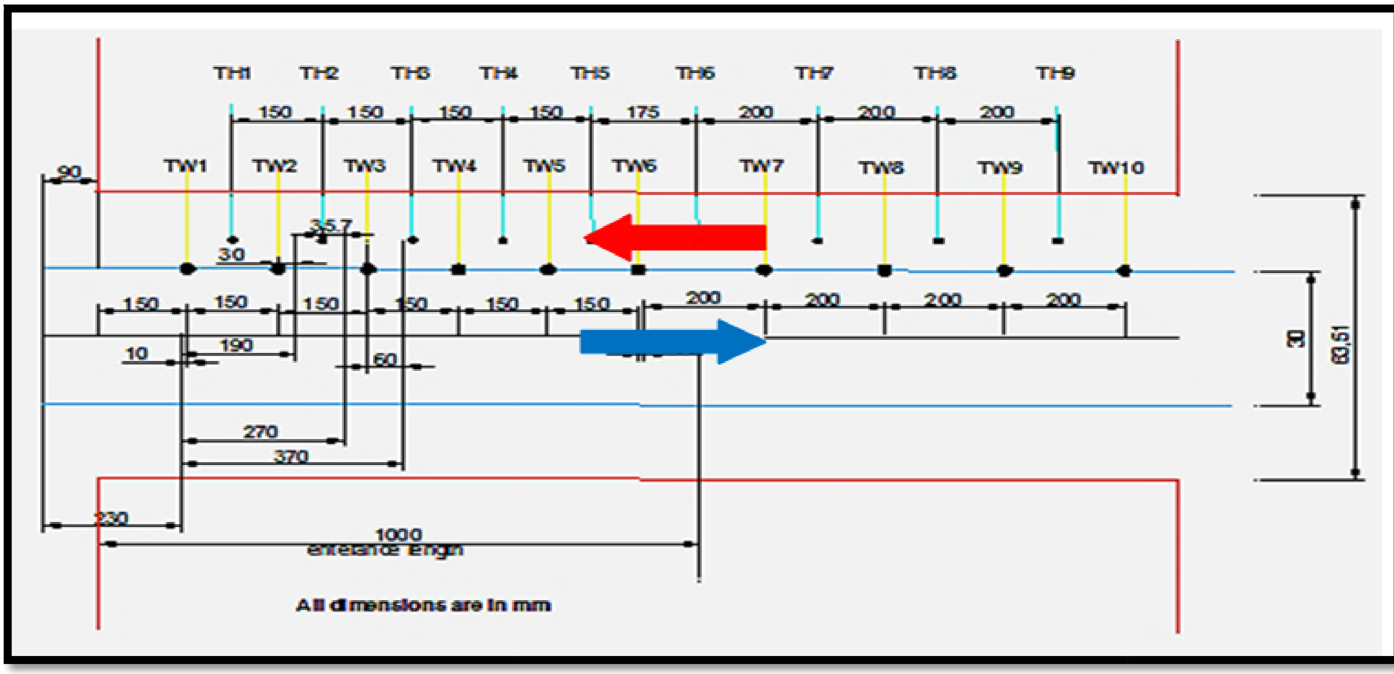

Figure 5: Thermocouple Locations

The detail of control volumes through the test section is depicted in Fig 6. Digital temperature recorder (6 channels) is used to measure the temperatures at various locations in hot water loop *(T_{h-i} , T_{H-1}... T_{H-9} ,T_{h-o})*, inlet, outlet cold water stream *($T_{c,i}$, $T_{c,o}$)* and inner tube wall *(Tw1.... Tw10)* the temperature sensors (thermocouples type K) are connected to temperature selector and other end inserted into wells in both hot and inlet, outlet cold water stream. A small quantity of thermal oil inserted in the well to ensure good thermal contact between the well and the thermocouple bead. Calibrated pressure gauges (bourdon tube with dial output) are mounted on the inlet and outlet of the tube and shell side of the test section through a handle valve to measure their pressure drop. The flow rates of the hot and cold water are controlled by adjusting the valves and are measured by weight measurements (using digital balance capable of reading ± 2 gm resolution) during defined period (using a stop watch capable of reading ± 0.01 sec resolution) . Experiments were conducted with various flow rates of cold water entering the test section. In the experiments, the cold water flow rate was increased in small increments, while the hot water flow rate, inlet cold water and hot water temperatures were kept constant.

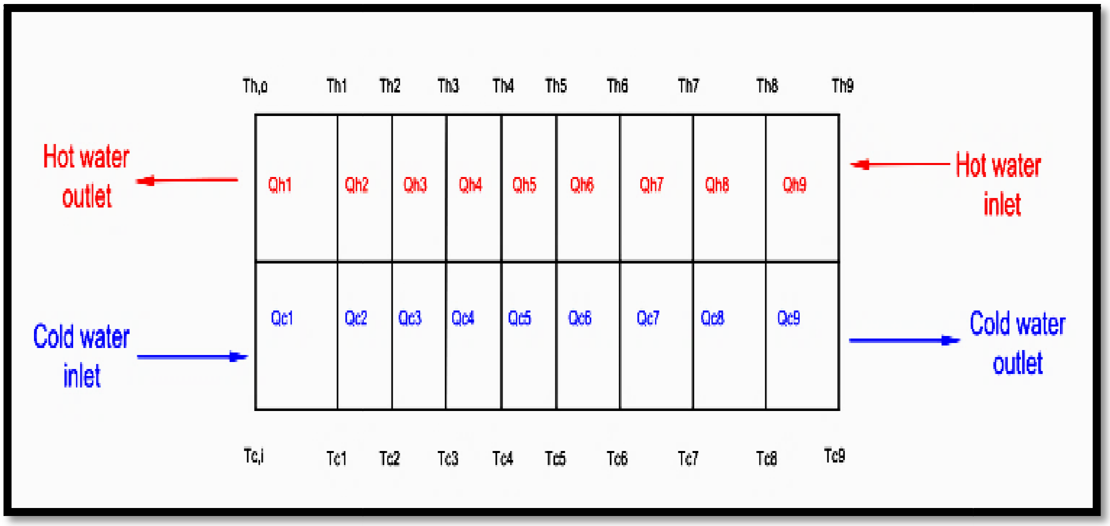

Figure 6: Sketch of Control Volumes through a Test Section

2.1. Governing Equations

For fluid flows in a concentric tube heat exchanger, the local heat transfer rate of the hot fluid (water) in the outer tube can be expressed as:

$$Q_{h,x} = \dot{m}_h Cp_{,h,av}(T_{h,x+1} - T_{h,x}) \quad (1)$$

Where:

$Q_{h,x}$ = local heat transfer rate of the hot fluid at each control volume, (watt)

$\dot{m}_h$ = mass flow rate of the hot water, (kg/s)

$Cp_{,h,av}$ = average specific heat of hot fluid at each control volume, (J/Kg °C)

$T_{h,x+1}$, $T_{h,x}$ = local temperature of the hot water at each control volume (°C)

While the local heat transfer rate of the cold fluid (water) for the inner tube is:-

$$Q_{c,x} = Q_{h,x} = \dot{m}_c Cp_{c,x-1}(T_{c,x} - T_{c,x-1}) = h_{c,x}A_{,x}(Tw_{av,x} - T_{c,x}) \quad (2)$$

Where:

$Tw_{,av,x}$ = average wall temperature of the inner tube, (°C) = $\frac{(Tw_{,x}+Tw_{,x+1})}{2}$,

$A_{,x} = \pi d L_{,x}$ = heat transfer area, (m^2)

$L_{,x}$ = inner tube length, (m)

The local Nusselt Number can be expressed as:

$$Nu_{c,x} = \frac{h_{c,x} D_h}{k_{c,x}} \quad (3)$$

Where:

$h_{c,x}$ = convection heat transfer coefficient of cold fluid, (W/m^2 °C)

D_h = hydraulic diameter = d (for circular tube), (m)

$k_{c,x}$ = thermal conductivity of cold water, (W/m °C)

And *x* denoted to location of each control volume in the inner tube and Twis the local wall temperature and it is measured at the depth of *0.5 mm* from the outer surface of the inner tube. The average wall temperatures are calculated from 10 points, lined between the inlet and exit of the inner tube.

The average Nusselt number is estimated as follows:

$Nu = Average\ (Nu_{c,x})$ (4)

The Reynolds number is given by $Re = \frac{4\,\dot{m}_c}{\pi\ d\,\mu_c}$

Where:

$\dot{m}_c$ = mass flow rate of cold water, (kg/s)
d = inner diameter of the inner tube, (m)
μ_c = cold fluid dynamic viscosity, (kg/m.s)

Average friction factor can be written as:

$f = \frac{2\,\Delta p\, d}{\rho v^2 L}$ (5)

Where:

Δp = pressure drop between tube entry and exit, (N/m^2)
d = inner diameter of the inner tube, (m)
ρ = fluid density, (Kg/m^3)
v = mean axial velocity, (m/s)
L= inner tube length, (m)

Thermal performance factor (η) of the test tube with a helical tape insert was evaluated as following, W.K, J.F. Fan, and Ding [16].

$\eta = \frac{(Nu/Nu^*)}{\left(f/f^*\right)^{1/3}}$ (6)

where:

Nu = average Nusselt number in inserts
Nu^* = average Nusselt number in plain tube
f = average friction factor in inserts
f^* = average friction factor in plain tube

2.2. Experimental Uncertainty

The evaluation of experimental uncertainties for the present data reduction process was conducted, the propagation of error method given by J.P Holman [17] has been used. Temperature and pressure drop measurements were the major sources for the uncertainties of Nu and f. With the differences in temperature of cold water and pressure drop measurements in the ranges of *Δt (3.4–5.7 °C)* and *ΔP (0.034 -0.1455 bar)*, the maximum uncertainties for Nu, Re and f were about *2.57% and1.56 % and 3.65%,* respectively.

3. Results and Discussions

3.1. Experimental Verification

The present experimental results on heat transfer and fluid flow characteristics in a tube without the helical tape insert were first reported in the form of Nusselt number and friction factor, f. The results of the plain tube were compared with the correlations of (Dittus and Boelter, 1930 [18]; Petukhov, 1970 [19]); for heat transfer; (Petukhov, 1970 [19]; Moody, 1944 [20]) for friction factor as shown in Figs 7 and 8.

3.1.1. Nusselt Number Correlations

- Correlation from Dittus and Boelter, 1930 [18] is of the form:

$Nu = 0.023\ Pr^{0.3}\ Re^{0.8}$ *for turbulent flow* (7)

- Correlation from Petukhov, 1970 [19] is of the form:

$$Nu = \frac{(f/8)\,\mathrm{Re}\,\mathrm{Pr}}{1.07 + 12.7(f/8)^{0.5}(\mathrm{Pr}^{2/3} - 1)}$$ *for $3000 \le Re \le 5x$ 10^6* (8)

Where:

Pr = Prandtle number
Re = Reynolds number

3.1.2. Friction Factor Correlations

- Correlation from Moody, 1944 [20]is of the form:-

$f = 0.316\ Re\text{-}1/4$ *for $Re \le 2x10^4$* (9)

- Correlation from Petukhov,1970,[1] is of the form:-

$f = (0.790\ log_{10}\ (Re) - 1.64)^{-2}$ *for $3000 \le Re \le 5x10^6$* (10)

Where:

f = friction factor

Figs 7 and 8. Shows comparison between the present experimental work and the past correlations from previous work. In the figures, the present work agrees well with the available correlations developed by Moody and Petukhov for the friction factor, and Dittus–Boelter, and Petukhov for the Nusselt number.

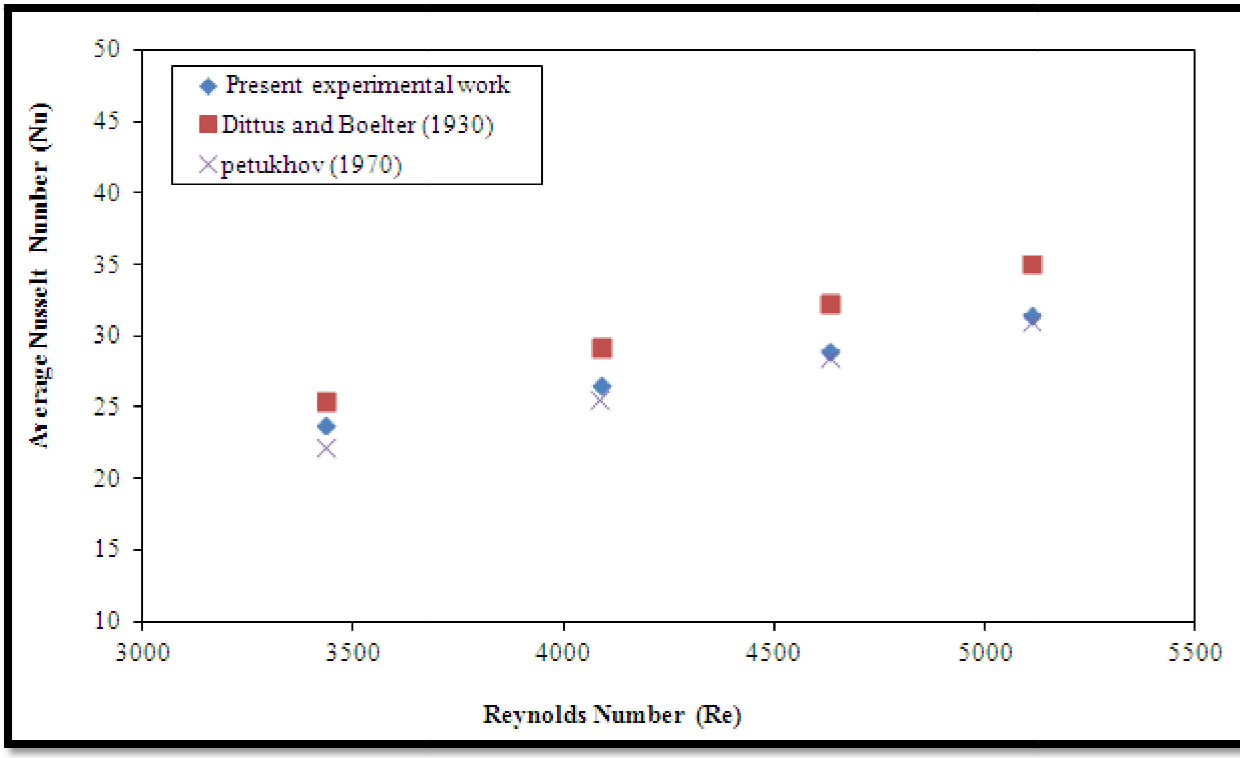

Figure 7: Experimental Data Verification of Nusselt Number of the Plain Tube versus Reynolds Number

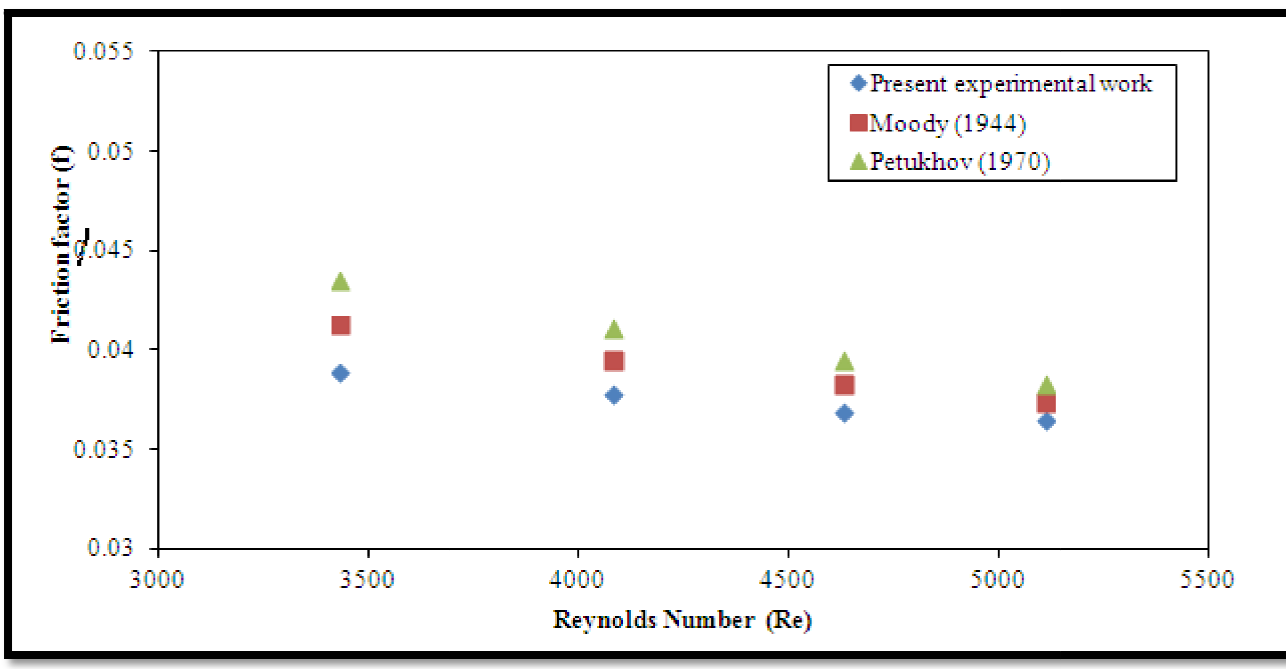

Figure 8: Experimental Verification of Friction Factor in Plain Tube

3.2. Effects of the Helical Tape Length

3.2.1. Heat Transfer Measurements

Table1 and Figure 9 shows the axial distributions of Nusselt number along the tube fitted with a helical tape of lengths (a) 130mm, (b) 270mm, (c) 370mm located at the entrance of the tube at Reynolds numbers (Re) of 3432, 4083, 4628, and5108. With each length examined, the local Nusselt numbers systematically increase as Re increases.Table1 and Fig 9, also shows the well-known approach to the fully developed flow from the flow entrance where the hydraulic and thermal boundary layers are developed due to the abrupt entrance. For each Re and length tested, Nusselt numbers in the entry region exponentially decay to the fully developed value at the axial location about x/d= 32.5 downstream of the flow entrance. In the final axial region of x/d ≥ 55.1, local Nusselt numbers gradually rise in the downstream direction due to the usual exit loss effect. The end tube values approach the fully developed constant value within ± 3% attributed to end loss effects. The Nusselt number data collected from all the axial stations are averaged to generate the average Nusselt number (Nu).

Configuration	**x/d**	**Re = 3432**	**Re = 4083**	**Re= 4628**	**Re = 5108**
plain tube		Nux	Nux	Nux	Nux
	8.9	306.0	373.4	409.9	430.1
	14.8	86.8	91.9	102.9	108.2
	20.7	51.7	57.2	60.2	70.5
	26.6	35.3	43.7	47.3	49.9
	32.5	3.3	3.8	3.9	4.2
	39.4	2.2	2.9	3.3	3.6
	47.2	2.4	2.8	3.2	3.6
	55.1	1.8	2.7	3.0	3.1
	63	7.8	7.9	8.0	8.9
LH=130mm	8.9	437.0	457.6	459.9	463.2
	14.8	105.0	105.3	112.5	117.6
	20.7	60.2	64.6	65.5	72.5
	26.6	39.6	47.8	52.7	54.0
	32.5	4.7	4.8	4.9	4.8
	39.4	3.1	3.1	3.6	3.5
	47.2	2.8	2.9	3.4	3.5
	55.1	2.6	2.8	3.3	3.3
	63	9.0	9.2	9.3	9.1
LH=270mm	8.9	450.6	471.5	481.0	490.6
	14.8	112.1	113.8	117.3	123.0
	20.7	65.7	72.1	73.3	80.9
	26.6	40.7	54.8	58.1	57.9
	32.5	7.2	7.2	7.3	7.4
	39.4	4.5	4.9	5.0	5.0
	47.2	3.7	3.8	4.2	4.2
	55.1	3.4	3.5	3.8	4.0
	63	9.8	9.6	11.0	11.0
LH=370mm	8.9	461.6	482.6	493.0	508.8
	14.8	117.9	118.1	123.3	124.7
	20.7	69.5	75.7	78.1	84.5
	26.6	43.1	59.9	64.7	67.7
	32.5	10.7	12.1	13.2	13.8
	39.4	5.0	5.8	6.7	7.0
	47.2	4.0	4.4	5.4	5.6
	55.1	3.7	4.0	4.8	5.0
	63	9.8	9.8	11.5	12.8

Table 1: Variation of Axial Nu along the Tube with the Helical Tape Inserts of Length 130, 270, and 370mm

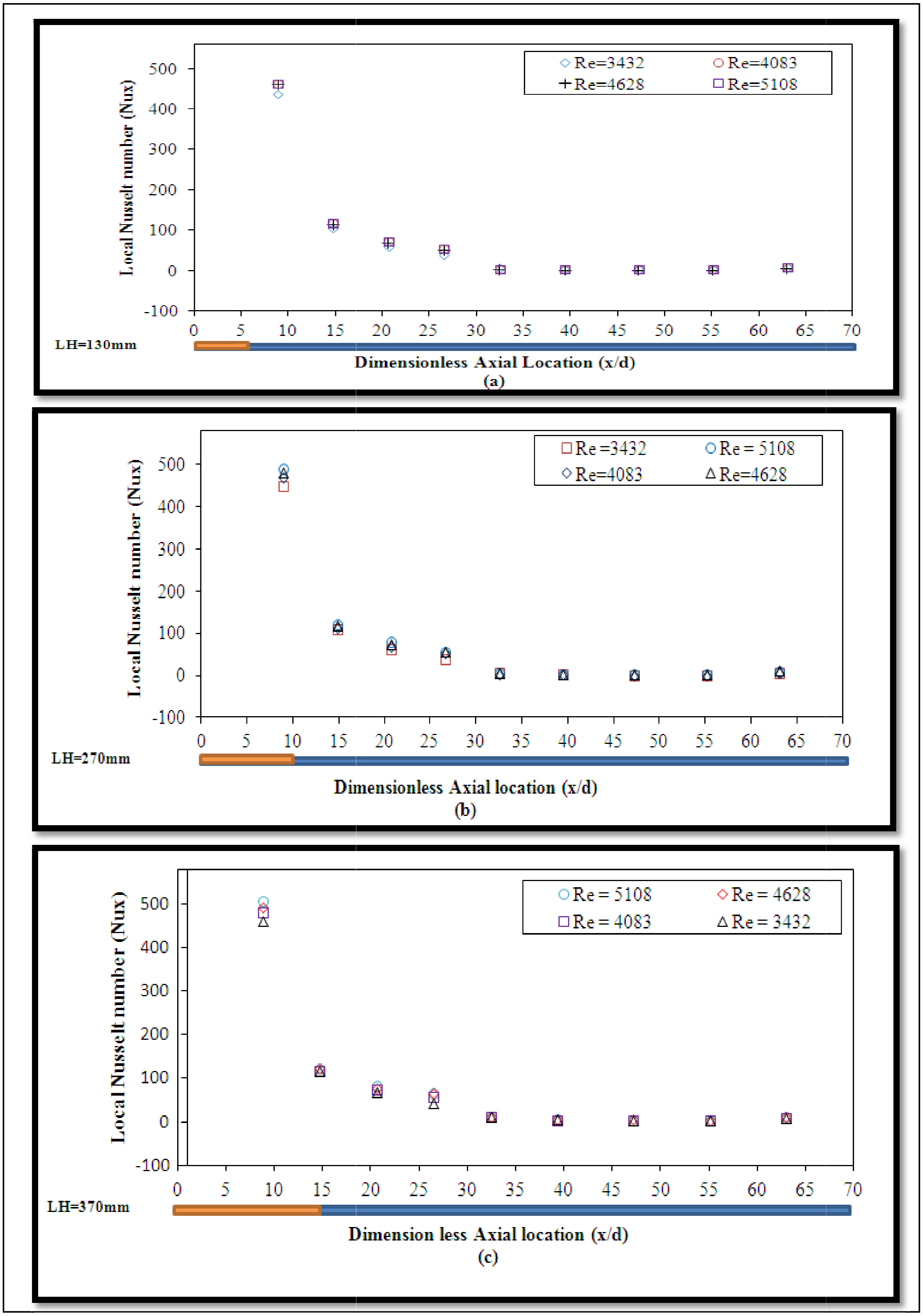

Figure 9: Variation of Axial Nu Number along the Tube with the Helical Tape Insert of Length (A) 130mm, (B) 270mm, And (C) 370mm

All the axial *Nu* distributions generated by present study are collected in Fig 9. This consistently shows the exponentially axial decay toward the fully developed value. This axial *Nu* profile along the tube with the helical tape inserts is also typical for the plain tube as well as the enhanced tube fitted with the helical tapes of length *LH =130,270* and *370mm* located at the entrance of the tube. And it is found that the effect of the heat transfer enhancement limited by a helical tape inserts location. It can be attributed that the use of helical tape insert of different lengths at the entrance of the tube can cause the swirl and pressure gradient in the radial direction. The boundary layer along the tube wall through this distance would be thinner with the increase of radial swirl and pressure resulting in more heat flow through the fluid. Furthermore, the swirl enhances the flow turbulence, which led to even better convection heat transfer.

Figure 10 present the effectiveness of heat transfer augmentation in the tube fitted with the helical tape insert relative to the length of helical tape insert. The average Nusselt number *Nu* in the tube fitted with the short helical tape is evaluated by averaging the local Nusselt numbers measured along the tube axis. The average Nusselt number *Nu* for the tubes fitted with the short helical tape increased in the turbulent flow regions as *Re* increases, *Nu* increases systematically with the increase of helical tape length as indicated in Fig 10. This can be attributed to effective length (swirling flow region) of the helical tapes and long residence time in the tube, in the *Re* range of *3432-5108.* The maximum *Nu* of *40* was obtained for the length of *370mm* at *Re = 5108*.

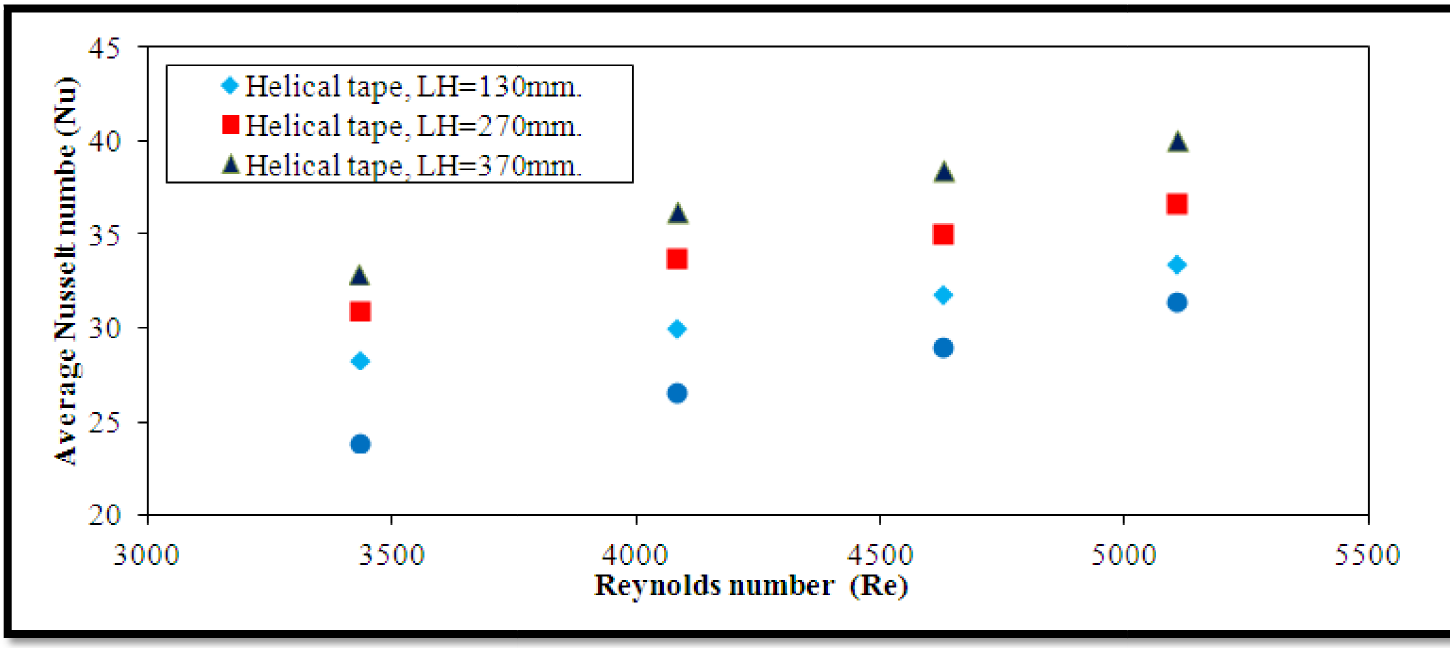

Figure 10: Heat Transfer Enhancement in Tube Side Cold Water Flows Due to Helical-Tape Insert with LH = 130mm, 270mm, and 370mm

3.2.2. Empirical Heat Transfer Correlation

The empirical heat transfer equations that correlate all the data points collected in Fig 10 are derived using *Re* and *LH/d* as the controlling variables since there is no significant change in the coolant's Prandtl number *Pr* for the range of temperatures covered by present experimental work. Here the Prandtl number of coolant (cold water) is treated as an invariant. A general mathematically structure for each *LH/d*- controlled *Nu* profile as shown in Fig 10 is assumed in compliance with the Correlation reported by Dittus and Boelter ,1930 [18] as,

$$Nu = B\,Re^{n} \qquad (11)$$

Where:

B = coefficient of intercept.

n = exponent.

The dependency of *Nu* on *Re* for each *(LH/d)* tested is initially examined by plotting the average Nusselt number against *Re* as indicated in Fig 11. The coefficient *B* and exponent *n* have their physical implications describing the heat transfer characteristics in a tube with the short length helical tape insert. It is interesting to note that all the average Nusselt number measurements obtained with Re ≥ 3432 (turbulent flow) along each test tube are well correlated by $Re^{0.4217}$,$Re^{0.4213}$, and $Re^{0.497}$ with the *(LH/d)* of 5.12, 10.63, and 14.57, respectively. As *(LH/d)* increases from 5.12 to 14.57, the exponent *(n)* of *Re* in the *Nu* correlation increases toward the asymptotic value of 0.497 as shown in Fig. 12 The variation manner of the *Re* exponent *(n)* against *(LH/d)* is well correlated by Eq. 12.

$$n = 0.0075(LH/d) + 0.3715 \qquad (12)$$

where:

LH = helical tape length, (mm)
d = inner diameter of the inner tube, (mm)

Also, the coefficient *B* of intercept features the heat transfer level in terms of *LH/d* for turbulent flow as indicated in Fig 13. It is interesting to note that coefficient *B* decreases consistently as *LH/d* increases that is opposed to the increased n exponent as *LH/d* increases. Therefore the largest *B* coefficient as indicated in Fig13 does not necessarily reflect the largest *Nu* value. In the quest of deriving the heat transfer correlation for design applications, the equation of coefficient *B* with the ratio *LH/d* as the controlling parameter is derived.

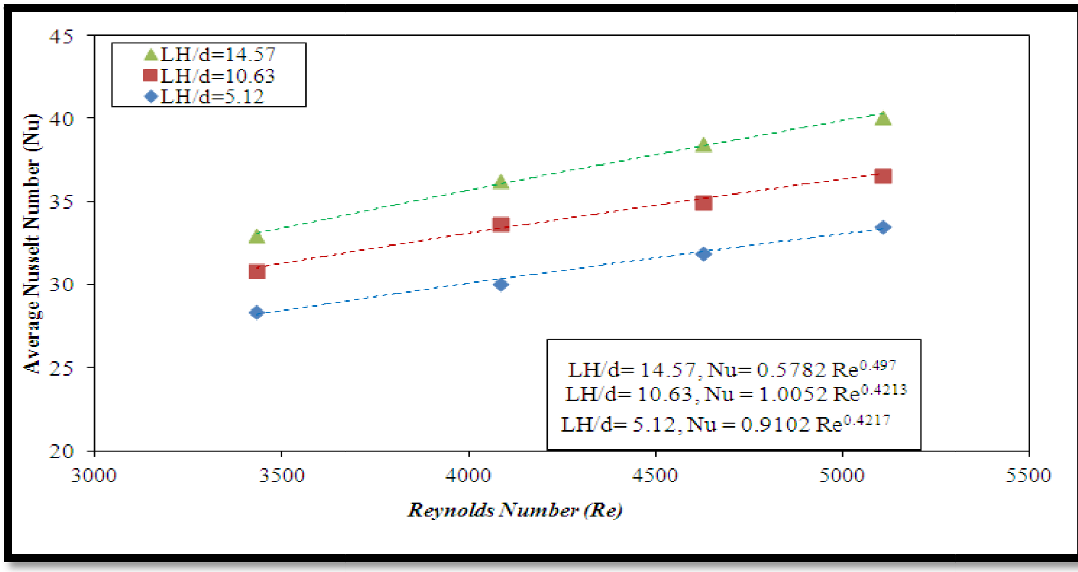

Figure 11: Variations of Average Nusselt Number against Re for the Tube Fitted with the Short Length Helical Tape of LH/D= 5.12, 10.63, And 14.57

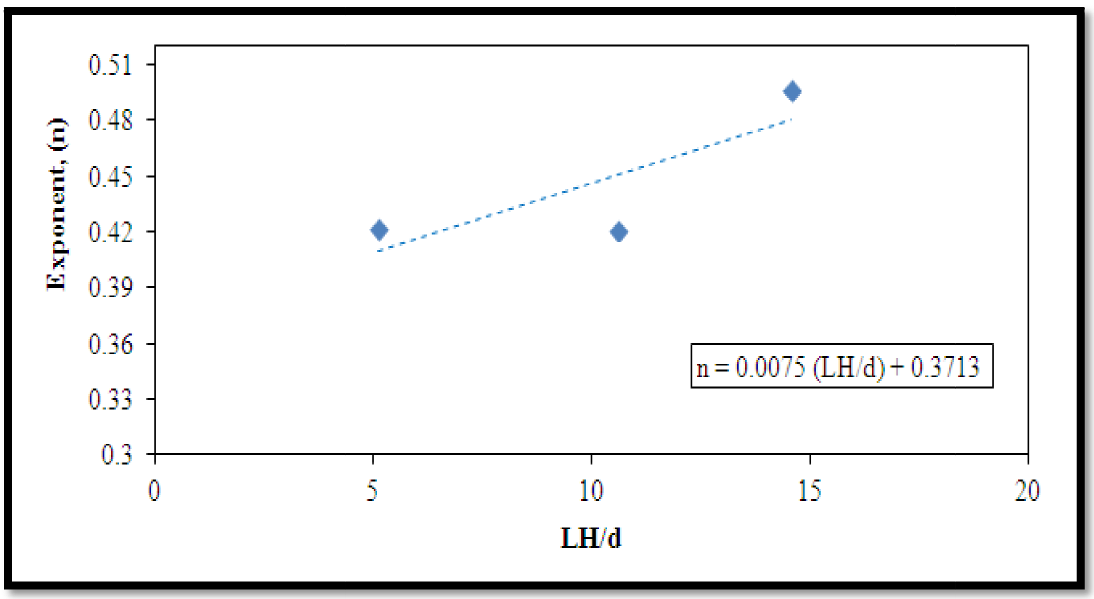

Figure12: Variation of Exponent (N) Against Ratio (LH/D)

Fig 13 depicts the variation manner of coefficient *B* against ratio *LH/d*. As seen in Fig13, coefficient *B* can be well fitted by the linear function expressed as Eq 13.

$$B= - 0.032\ (LH/d) + 1.155 \qquad (13)$$

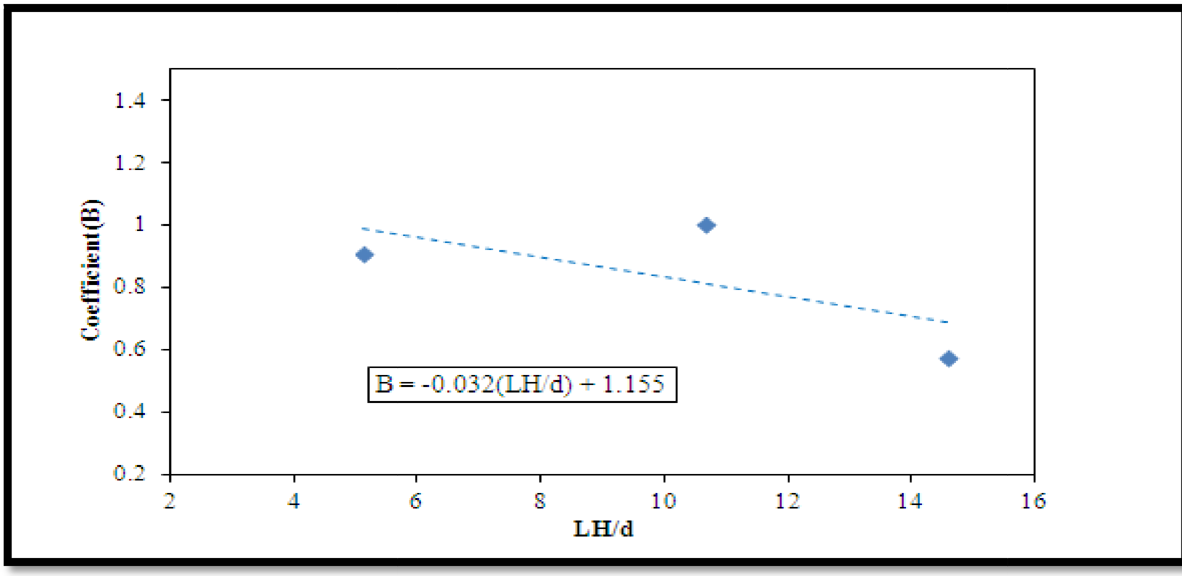

Figure 13: Variation of Coefficient (B) against Ratio LH/D

Eq. 14 represents the generalized correlation for predicting the average Nusselt number *Nu* along the tube with the short length helical tape insert located at the entrance of the tube of $5.12 \le LH/d \le 14.57$ with $3432 \le Re \le 5108$

$$Nu = (-0.032*(LH/d) + 1.155)\ Re^{\left(0.0075\left(\frac{LH}{d}\right)+0.3713\right)} \qquad (14)$$

Figure.14. Shows the discrepancies between the experimental measurements and the results obtained from these heat transfer correlation Eq.14. The maximum discrepancies between the experimental measurements and correlation results are ± 5%.

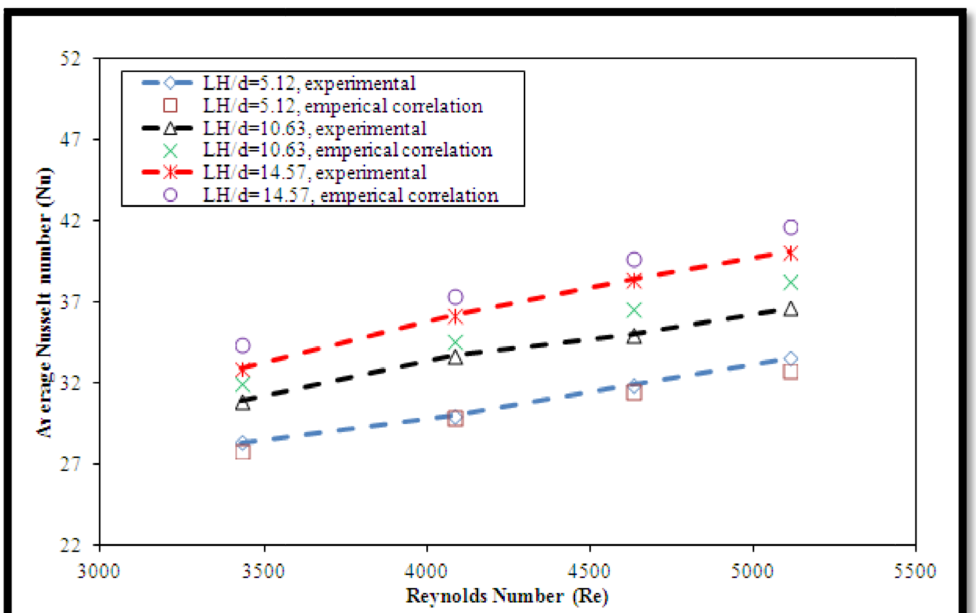

Figure 14: Discrepancies between Experimental Measurements and Empirical Heat Transfer Correlation for the Tube with Short-Length Helical Tape Inserts

3.2.3. Pressure Drop Measurement and Thermal Performance Factors

The pressure drop coefficients evaluated from the pressure differences across the tube fitted with the helical tape are defined as the mean friction factors f. The augmentation of pressure drop coefficient from the plain tube level due to the helical tape insert is indexed by the ratio of f/f* where the pressure drop coefficients for the plain tube f*. The variations of average friction factors in the tube fitted with the helical tape of length LH=130, 270, and 370mm along with their f /f* ratios against Re are displayed in Fig 15a and b, respectively. As shown in Fig 15a, the mean friction factors f in each tube fitted with the helical tape insert of LH= 130, 270, and 370mm decrease exponentially with a tendency of approaching the asymptotic value as Re increases from 3432 to 5108. It can be seen that the trend of friction factor f is similar for both the axial flow (plain tube) and the swirling flow (helical tape inserts). The friction factor for the tube with helical tape is substantially higher than that for the plain tube because of a higher surface area and the dissipation of dynamic pressure of the fluid at high viscosity loss near the tube wall. Moreover, the friction factor had high possibility to occur by the interaction of the pressure forces with inertial forces in the boundary layer. Also, the flow velocity is larger since the motion is not in an axial direction. At each Re tested, the mean friction factor f consistently increases as LH increases. The heat transfer and pressure drop characteristics revealed in Figs 10 and 15a, respectively, indicate that the flow in the tube with the helical tape insert belongs to the boundary layer type flow except for the dashed contacts along the helices of the tape. However, the shear layers downstream these helices interact with the swirling mainstream in the tube fitted with the helical tape considerably amplify the turbulent intensity and the vorticity. These interactive flow mechanics in the tube fitted with the helical tape generate the aforementioned heat transfer augmentations with the penalty of increased f factors to considerable extent. The friction factor ratios in terms of f/f* for the present tubes fitted with the helical tape of length LH =270,370mm constantly operate at the higher values than those in the helical tape of length 130mm as shown in Fig 15b. From the results of the tape with length 130mm, and 270mm friction factor could be reduced around 1.45%, and 0.28 % respectively in comparison with the helical tape of length 370mm.

Fig16. Shows the thermal performance factor Vs Re number for different lengths of helical tapes. The thermal performance factor (η) factors consistently increase as LH increases due to the accompanying decreases in f factors. The thermal performance factor (η) factors in each tube fitted with the helical tape of LH = 130, 270, 370mm consistently decrease with the Re increases.

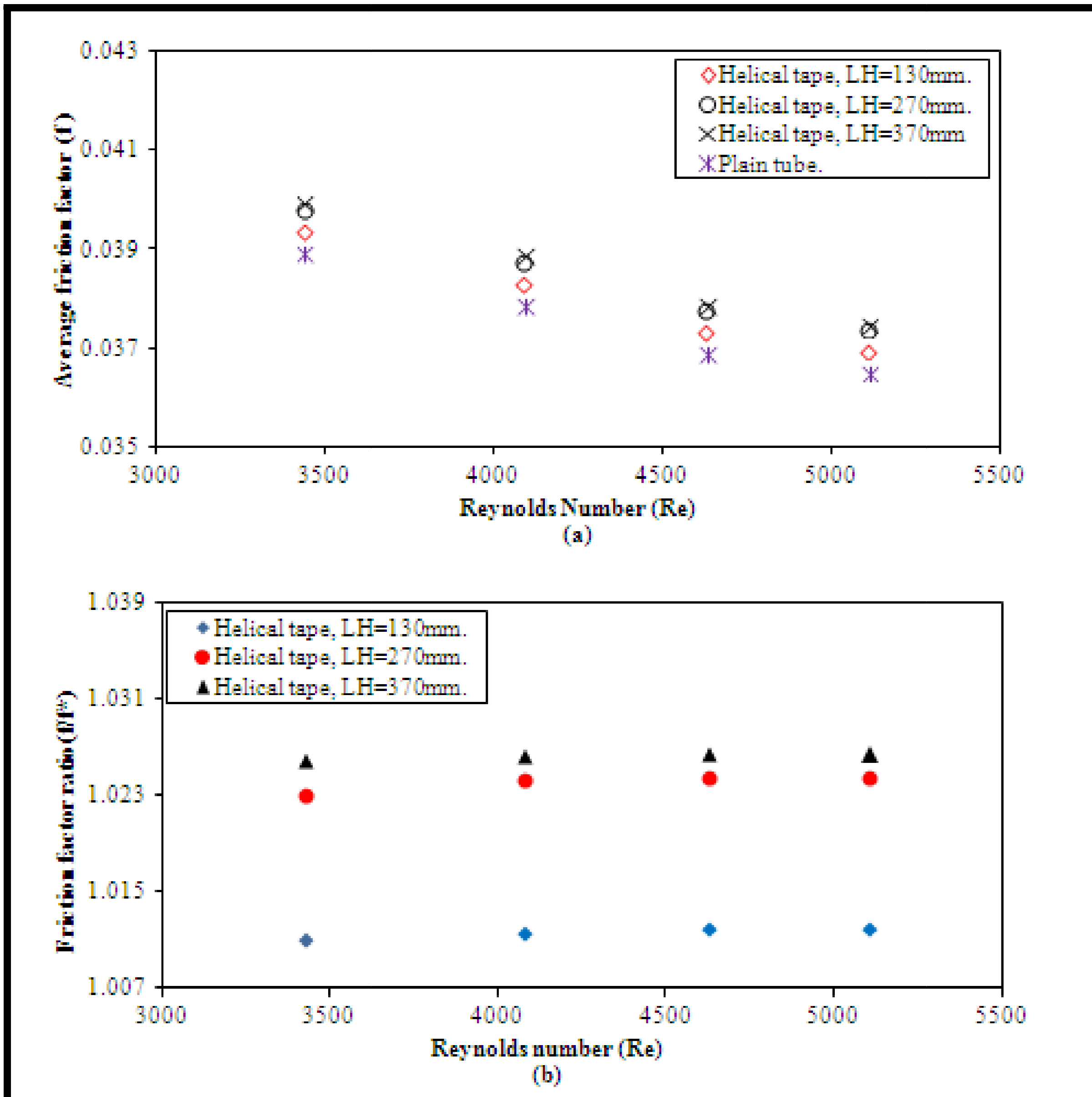

Figure 15: (A) Average Friction Factor (F) Versus (Re) and (B) Friction Factor Ratio (F/F) Versus (Re) for Tubes Fitted with Helical Tape of LH=130, 270, and 370mm*

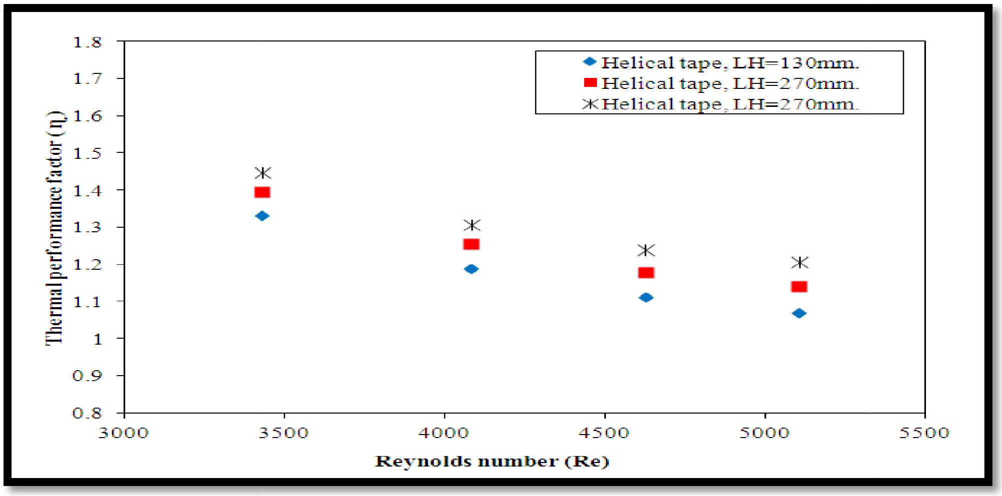

Figure 16: Thermal Performance Factor in Tube Fitted with Short Length Helical Tapes

3.3. Effects of Free Spaced Helical Tape

3.3.1. Heat Transfer Measurements

The experimental heat transfer results for in-tube swirl flows are presented in Fig 17 as well as Table 2, for three different space to tape ratio *S= 1.8, 3.6, and 5.4, LH= 135mm* Data were taken for cold water as test fluid, at *Re = 4083* and data are presented in the form *Nux* versus *x/d*. Fig 17. Shows variation of average Nusselt number with the helical tape insert configurations, *LH = 135 mm* with different Space ratio, *S = 1.8, 3.6, and 5.4 at Re =4083.* In the figure, the use of small value of space ratio *(S)* yields a higher heat transfer rate than that of large value of space ratio (S). The large free-spacing between the two helical tape of length *135mm for S = 5.4* is found to be not sufficient to maintain the swirl intensity to extent to the next helical tape. The swirling flow decays faster in the large free-spacing before reaching the next helical tape to build up swirling flow again. The optimum value of the space ratio should be less than *3.6* to get benefit of a higher heat transfer rate and reduction of pressure drop in comparison with the plain tube. Throughout the experimental results, it is visible that using the smaller free space ratio *(S = 1.8)* yields the higher values of heat transfer than the greater space ratio *(S = 3.6 and 5.4)*as shown in Table 2. This can be attributed to a strongly swirling flow remain for the small space ratio while decaying swirling flows for the large space ratio. In the figure, the spaced helical tape of length *130mm for S = 1.8, 3.6,and 5.4*, gives the maximum heat transfer rate at about *142%, 138%, and 137%* respectively, all compared with the plain tube. It is observed that at the free-space ratio, *S = 3.6*, there are slight differences in the heat transfer rate when compared with the free-space ratio, *S = 5.4.*

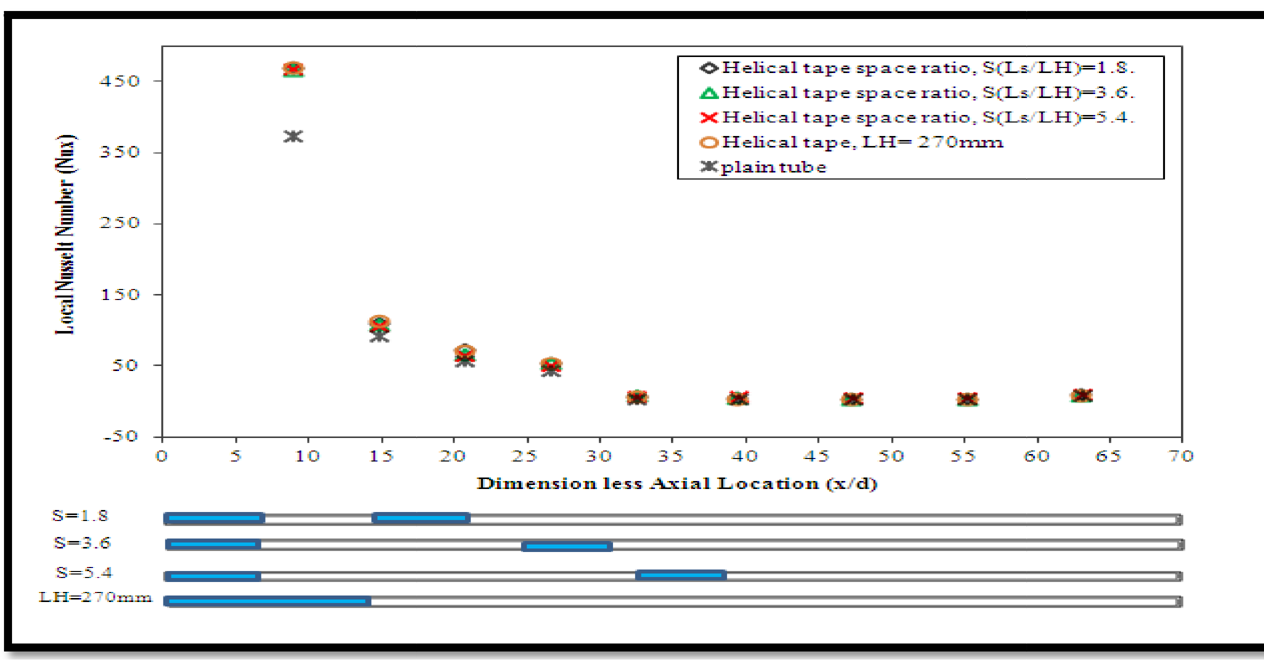

Figure 17: Variation of Axial Nu Number along the Tube with the Helical Tape Insert, LH = 135mm with Different Space Ratio, S (Ls/LH) = 1.8, 3.6, and 5.4 at Re =4083

X/D	Local Nusselt Number (Nux) At Re = 4083				
	Plain Tube	Lh=270mm	S(Ls/Lh)=1.8	S(Ls/Lh)=3.6	S(Ls/Lh)=5.4
8.9	373.4	471.5	470.8	468.1	467.9
14.8	91.9	113.8	110.2	109.5	107.0
20.7	57.2	72.1	73.9	67.7	64.3
26.6	43.7	54.8	54.9	54.7	50.4
32.5	3.8	7.2	7.6	10.2	7.8
39.4	2.9	4.9	5.1	5.6	7.4
47.2	2.8	3.8	3.9	4.0	5.2
55.1	2.7	3.5	3.6	3.8	4.5
63	7.9	9.6	9.7	9.8	11.3

Table 2: Variation of Axial Nu Number along the Tube with the Helical Tape Insert, LH = 135mm with Different Space Ratio, S (Ls/LH) = 1.8, 3.6, and 5.4 at Re =4083

3.3.2. Pressure Drop Measurements and Thermal Performance Factor

Table 3 shows the relationship between the friction factor ratio and various free spaced helical tapes inserts in comparison with experimental data of the helical tape of length 270mm and plain tube. In this arrangement of the helical tapes, the swirling flow can happen continuously in the free-spacing but at different swirl levels. The presence of the free-spacing region is to reduce the surface areas that affect considerably the pressure loss of fluid flow. Many space ratios are introduced to investigate flow behaviors. In Table 3, the large pressure drop occurring from the free space helical tape inserts =1.8 can be decreased substantially by using the free spaced helical tape. It can be seen that for the space ratio of S=1.8, S=3.6, and S=5.4, gives the reduction in mean Nusslet number at about 0.22%, 1.07%, and 1.85% respectively, all compared with the helical tape of length 270mm. Furthermore the reduction in friction factor is found to be about 1.26 %, 1.27%, and 1.29 %, respectively. The spaced helical tape inserts at S=1.8 yields the highest Nusselt number which is about 26 % above the plain tube. Table 3. Shows the maximum performance factor of 1.28 was obtained for the free space ratio (S=1.8) at Re= 4083. It was found that the thermal performance factor of two helical tapes (LH=135mm) with free space ratio (S=1.8) are better than the continuous helical tape of length (LH=270mm).

Inserts Configuration	Nu	Nu/Nu*%	f/f*%	η
Helical tape of length, LH=270mm	82.36	126.5	101.4	1.25
Spaced helical tapes of length, LH=135mm, S=1.8	82.18	126.2	101.09	1.28
Spaced helical tapes of length, LH=135mm, S=3.6	81.48	125.1	101.08	1.25
Spaced helical tapes of length, LH=135mm, S=5.4	80.66	123.8	101.06	1.23

Table 3: Relationship between the Average Nu, Nu/Nu Ratio, F/F*Ratio, and Various Free Spaced Helical Tape Inserts at Re = 4083*

4. Comparison with Previous Studies

The enhancements that the short, full length helical and the twisted tape inserts produce in Re range (3432–5108) have been compared. Fig 18 and Table 4 shows the Nusselt number ratios versus Reynolds number results for a short length helical tape (Present Experimental Work), compared with the predictions of Smith and Pongjet [1], For full length helical with and without centered a rod and twisted tapes, y=2.5. The helical tape was made of stainless steel and has the geometric dimensions of W=17mm (0.95D), P=18mm (0.95D), t=1mm (0.05D). The predictions have been obtained for a 2 m long horizontal copper tube, with inside diameter 19 mm under the flow conditions Re =2300 - 8800 and using cold water and hot air as the test fluids. Fig 18 Shows that the helical tapes (present work), using hot and cold water as the test fluids have higher heat transfer coefficients than the full length helical and twisted tapes using cold water and hot air as the test fluids in the Reynolds number range Re=3432-5108. The difference is essential: whereas the thermal conductivity and specific heat capacitance of the cold water is greater than hot air. Also can be attributed to a strongly swirling flow remain for the helical tape (present work) with a small pitch, P=10mm while decaying swirling flows for the helical tape, Smith and Pongjet [1] with large pitch, p=18mm and twisted tape, y=2.5.

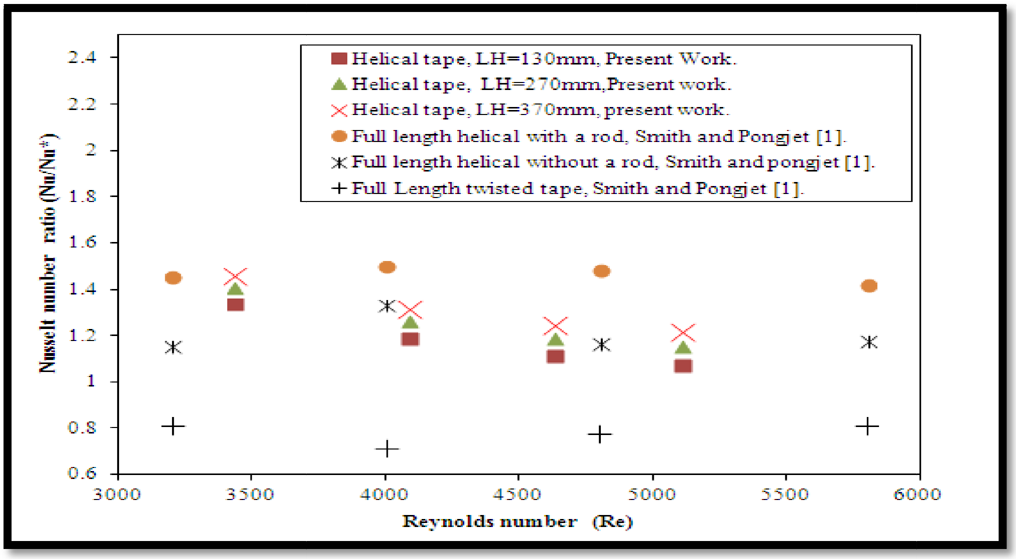

Figure 18: Nusselt Number Ratio Vs Reynolds Number, Helical Tapes with LH=130,270, and 370mm, Full Length Helical and Twisted Tapes with Y=2.5

Present Experimental Work			Smith And Pongjet [1]		
Configuration	Re	Nu/Nu*	Configuration	Re	Nu/Nu*
Short length helical tape, LH=130mm	3432	1.3	Full length helical tape with rod	3200	1.5
	4083	1.2		4000	1.5
	4628	1.1		4800	1.5
	5108	1.1		5800	1.4
Short length helical tape, LH=270mm	3432	1.4		5900	1.5
	4083	1.3	Full length twisted tape	3200	0.8
	4628	1.2		4000	0.7
	5108	1.1		4800	0.8
Short length helical tape, LH=370mm	3432	1.5		5800	0.8
	4083	1.3		5900	1.3
	4628	1.2	Full length helical tape without rod	3200	1.2
	5108	1.2		4000	1.3
________	---------	---------		4800	1.2
	---------	---------		5800	1.2
	---------	---------		5900	1.5

Table 4: Nusselt Number Ratio Vs Reynolds Number, Helical Tapes with LH=130,270, and 370mm, Full Length Helical and Twisted Tapes with Y=2.5

5. Conclusions

An experimental study has been conducted to investigate heat transfer enhancement in process heat exchangers by means of helical tape insert with manufacture cost is low of length LH=130 mm, 270 mm, and 370 mm, and different configurations S=1.8, 3.6, and 5.4 located at the inlet of inner tube in a double pipe heat exchanger using In tube flows of cold water in horizontal tubes. This is representative of typical conditions encountered in practical applications.From the experimental results; it can be concluded as follows:

- The use of helical tapes leads to a higher heat transfer rate over the plain tube. It is noticed that, in the range of (3432≤ Re ≤ 5108), local Nusselt numbers and friction factors increase as the length increases.
- The use helical tape insert of length LH =370 mm provides the heights heat transfer rate which is about 27% above the plain tube at Re = 3432.

- The local heat transfer coefficients were found to be increasing to very high values in the region of the helical tape, and then decreasing along the downstream of the helical tape.
- The thermal performance factor increases with increasing helical tape length at each Re examined and decreases with increasing Reynolds number in the range of 3432≤ Re ≤5108 with a constant helical tape length.
- The maximum performance factor of η =1.45 was obtained for the length of 370mm at Re=3432. And η =1.27 for the space to tape ratio (S=1.8) at Re= 4083. The thermal performance factor of two helical tapes with free space ratio (S =1.8) are found to be better than the continuous helical tape of length (LH=270mm), at Re =4083. It was found that the space ratio value should be ≤ 1.8 at Re = 4083.
- A performance comparison between short length helical tapes that is inserted at the entrance (using water as a tested fluid) and full length helical and twisted tapes inserts (using air as tested fluid) is presented. It is found that, the short length helical tapes inserts perform better than full length helical tapes with and without a rod and full length twisted tapes in the range of Re = 3432–5108.
- In general , It is found that enhancing heat transfer with passive method using different short lengths of helical tapes construction in the inner tube of a concentric double pipe heat exchanger can improve the heat transfer rate efficiently and the friction factor also increased . The maximum average Nusselt number Nu may be increased by 134% for the length LH=130 mm, 140 % for the length LH=270 mm, 146% for the length 370 mm , and 126.17% for the free spaced helical tape , S=1.8 at Re=4083 ,all compared with the plain tube .

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