Journal of Engineering and Development, Vol. 16, No.2, June 2012 ISSN 1813- 7822

Heat Transfer and Pressure Drop in Turbulent Flow through an Eccentric Converging-Diverging Tube with Twisted Tape Inserts

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Abstract

The aim of this study is to investigate the heat transfer and pressure drop characteristics in an Eccentric Converging-Diverging Tube (ECDT) with twisted tape inserts. Experiments were conducted with tape inserts of three different twist ratios. Cold and hot water are used as working fluids in shell and tube sides, respectively. The effect of the twist ratio and other parameters on heat transfer characteristics and pressure drop are considered. The experimental data for the plain tube and (ECDT) without inserts are compared with (ECDT) with twist tape insert. The results show that the twist tape insert has significant effect on the enhancement of heat transfer. The Nusselt numbers for the (ECDT) are found to be 15% to 45% higher than of the plain tube while for the (ECDT) combined with twisted tape insert is found 52% to 280% higher and pressure drop is found 6.8 times the plain tube. Moreover, (ECDT) combined with a twisted tape insert gives higher heat transfer rate and pressure drop than the (ECDT) alone around 23% to 35% and 98% to 125%, respectively.

Keywords: heat transfer, twisted-tape, enhancement, turbulent flow, pressure drop

الخلاصة

هدف هذه الدراسةِ التحقق من خصائص انتقال الحرارةَ وهبوط الضغط في Eccentric Converging-Diverging Tube (ECDT) بوجود الشريط الملتويِ. أجريت التجارب بنسب ليّ مختلفة وبعدد ثلاث.تم استخدام الماء البارد والحار في المبادل الحراري نوع Double Pipe Heat Exchanger، تم الأخذ بعين الاعتبار تأثير نسبة الليّ والعوامل الأخرى على خصائص انتقال الحرارة و وهبوط الضغط . تم إجراء التجارب للأنبوب الدائري و (ECDT) بدون وجود الشريط الملتويِ و مقَارنتها مع (ECDT) بوجود الشريط الملتوي وقد أظهرت النتائج تأثير الشريط الملتوي على تحسينِ انتقالِ الحرارة. ان رقم نسلت ل(ECDT) كَانَ 15 % إلى 45% أعلى من ذلك للأنابيب الدائرية بينما ل(ECDT) وبوجود الشريطِ الملتويِ كانت 52% الى 280% وهبوط الضغط كان أعلى بـ 6.8 مرةِ من التي للأنابيب

الدائرية. علاوة على ذلك، (ECDT) بوجود الشريط الملتوي حقق زيادة في معدل انتقال الحرارة والهبوط في الضغط أعلى من (ECDT) بدون الشريط الملتوي حوالي % 23 إلى 35% و% 98 الى % 125 على التوالي.

Introduction

Enhancement of heat transfer intensity in all types of thermo technical apparatus is of great significance for industry. Beside the savings of primary energy, it also leads to a reduction in size and weight. Several heat transfer enhancement techniques have been developed. One of them is using turbulators. Hosni [1] (2004), studied the comparison of heat transfer enhancement methods in heat exchangers. Ahmed et. al. [2] (2005), investigated experimentally heat transfer and pressure drop characteristics in a circular tube fitted with twisted tape inserts. Their results indicated that average heat transfer coefficient is about 1.3 to 3 times higher than that of the plain tube. Kittisak et. al. [3] (2006), studied the turbulators by using nozzles placed inside the inner test tube with various pitch ratios of PR = 2.0, 4.0, and 7.0. They found that Nusselt numbers for using the enhancement devices with PR = 2.0, 4.0 and 7.0 are 374%, 342% and 309% respectively, in comparison with the plain tube. Hong et. al. [4] (2007), investigated experimentally Pressure drop and compound heat transfer characteristics of a converging-diverging tube with evenly spaced twisted-tapes (CD-T tube). They found that At Reynolds number ranging from 3400 to 20000, when space ratio 48.6, the heat transfer efficiency index, which increases as the Reynolds number increases, is 0.85—1.21 and 1.07—1.15 compared to that of a plain circular tube and a (CD-T tube)without twisted-tape inserts, respectively. Somsak et. al.[5] (2007) studied experimentally the Effects of the louvered strip insertion in a parallel-flow concentric double pipe heat exchanger on heat transfer performance and flow friction. They found that the mean Nusselt Number was increased up to about 246% in comparison with the plain tube, while approximately 167% of the friction factor was increased, comparative to the plain tubes. Eiamsa et.al [6](2007), investigate experimentally the heat transfer and turbulent flow friction characteristics in a circular wavy-surfaced and constant heat-flux tube with a helical-tape insert. They found that the Nusselt numbers and friction factors are found to be, respectively, 3.0 and 50 times over the plain tube for the tube with wavy- surfaced wall alone and to be 4.2 and about 110 times for the tube with combined wavy-surfaced wall and the helical-tape. The wavy-surfaced tube combined with the helical-tape provides higher heat transfer rate and friction factor than the wavy-surfaced tube alone around 57% and 125%, respectively. Paisarn Naphon [7](2007), studied Effect of coil-wire insert on heat transfer enhancement and pressure drop of the horizontal concentric tubes. He found that the effect of coil-wire insert on the enhancement of heat transfer tends to decrease as Reynolds Number increases. Promvonge et. al. [8] (2007), studied heat transfer enhancement in a tube with combined conical-nozzle inserts and swirl generator. They found that each application of the conical nozzle and the snail can help to increase considerably the heat transfer rate over that of the plain tube by about 278% and

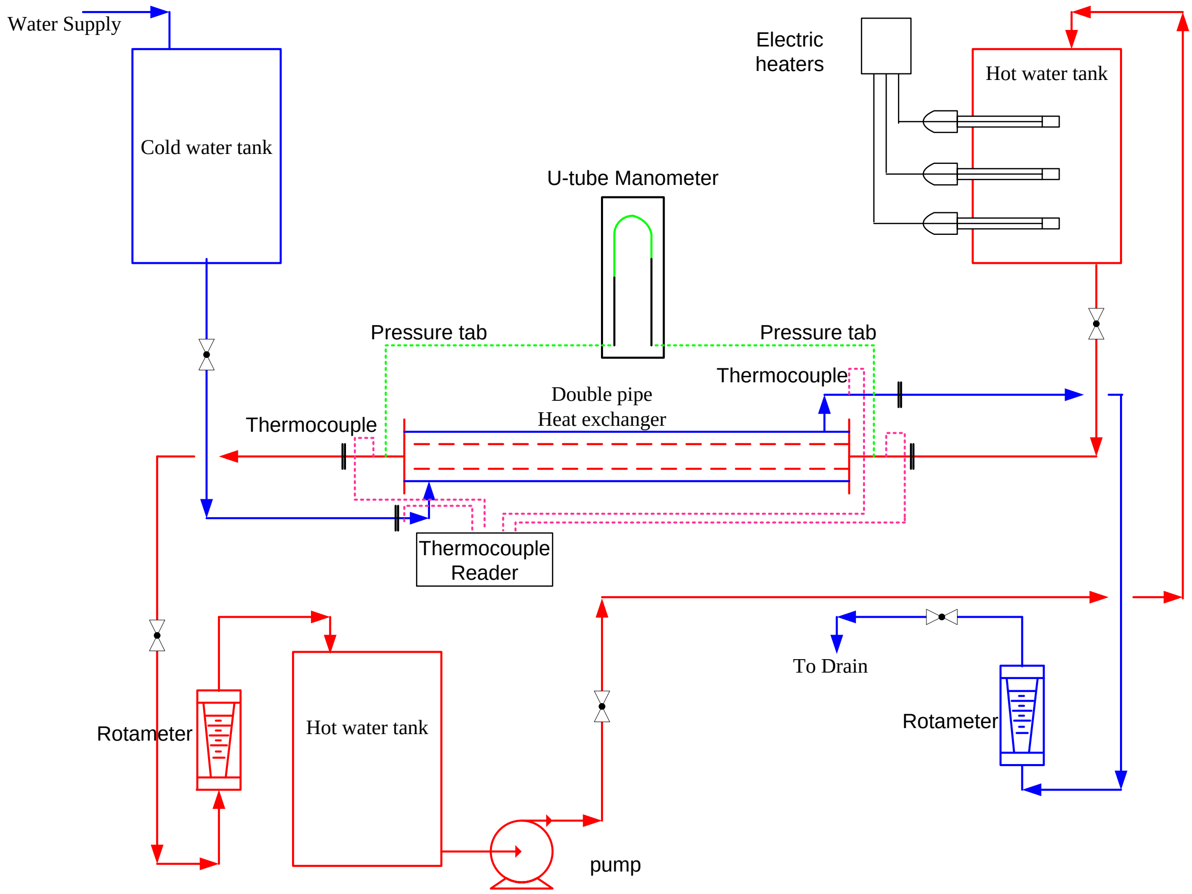
206%, respectively. The use of the conical nozzle in common with the snail leads to a maximum heat transfer rate that is up by 316%.

Eiamsa et. al. [9] (2008), studied the Enhancement of heat transfer in a tube with regularly-spaced helical tape swirl generators. They found that experimental results confirmed that the use of helical tapes leads to a higher heat transfer rate over the plain tube. The full-length helical tape with rod provides the highest heat transfer rate about 10% better than that without rod but it increased the pressure drop. Murugesan et. al. [10] (2009), investigated experimentally the heat transfer and friction factor characteristics of circular tube fitted with full length twisted tape with trapezoidal -cut for the Reynolds number range of 2000-12000. They found that the heat transfer enhancement of trapezoidal-cut twisted tape is reasonable since the performance ratio was more than unity. Subsequently an empirical correlation has been formulated to base on experimental results with ±5% variation in Nusselt number and ±6% in friction factor. Nagarajan et al. [11] (2009), simulate the heat transfer augmentation in a circular tube fitted with; Right-left helical tape inserts with 100 mm spacer in laminar flow conditions by CFD. They found that the data obtained by simulation are matching with the literature value for plain tube with the discrepancy of less than ±5% for Nusselt number and friction factor. Naga Sarada et. al [12] (2009), studied experimentally the augmentation of turbulent flow heat transfer in a horizontal tube by means of mesh inserts with air as the working fluid. They found that the enhancement of heat transfer by using mesh inserts when compared to plain tube at the same mass flow rate is more by a factor of 2 times where as the pressure drop is only about a factor of 1.45 times. Anil Singh Yadav [13] (2009) studied the effect of half length twisted-tape turbulators on heat transfer and pressure drop characteristics inside a double pipe U-bend heat exchanger. The heat transfer coefficient is found to increase by 40% with half-length twisted tape inserts when compared with plain heat exchanger, and that the thermal performance of plain heat exchanger is better than half length twisted tape by 1.3-1.5 times. Kapatkar et. al. [14](2010) investigated heat transfer and friction factor of a plain tube fitted with full length twisted tape inserts for laminar flow. They found that, for the flow in plain tubes, full length twisted tapes yield improvement in average Nusselt number, for Reynolds number range of 200 to 2000.For Aluminum tapes, the maximum improvement in Nusselt number range from 50% to 100%; for Stainless steel tapes, maximum improvement in Nusselt number range from 40% to 94% and for insulated tapes, maximum improvement in Nusselt number range from 40% to 67%. The isothermal friction factor for the flow with the twisted tape inserts are 340% to 750 % higher as compared with those of smooth tube flow, in the given range of twist ratios.

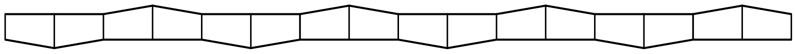
In this study is to investigate the heat transfer and pressure drop characteristics in an Eccentric Converging-Diverging Tube (ECDT) with twisted tape inserts.

Experimental Procedure

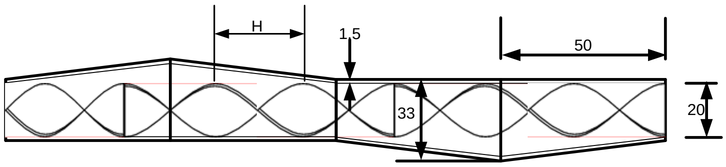
The experimental investigation was conducted to reach a more practical look on the influences of using (ECDT) in conjunction with a twisted tape for both heat transfer enhancement and pressure loss in a double pipe heat exchanger. Three twisted tape inserts of different twisted ratios were used for comparison with the standard plain tube. A schematic drawing of a double pipe heat exchanger and the (ECDT) combined with a twisted tape is given in Figures 1 , 2, and 3 while the details of the test conditions are also presented in Table 1. The (ECDT) were made of stainless steel with length of 1600 mm and installed in a double pipe heat exchanger as an inner tube using hot water as working fluid. The twisted tapes were made of straight aluminum tape with thickness (δ) of 0.5 mm, width (w) of 20 mm, and twist ratios y/w=3, 6 and 9. The twist ratio is defined herein as the ratio of two twist length (y, 180°) to the tape width (w). The experiments were done in Dura Electricity station.



(Figure: 1) Schematic diagram of the experimental apparatus



(a) Eccentric Converging-Diverging Tube (ECDT)



(b) Dimensions of Eccentric Converging-Diverging with twisted tape (in mm)

(Figure: 2) (ECDT) configuration

(Figure: 3) Photograph of Twisted Tape with different twist ratio

Table 1: Detail of test conditions

(a) Reynolds number, (Re)	5000 to 50,000
(b) Working fluid	Cold/hot water
(c) Inlet cold water temperature,	(T_{c1}) 20 °C
(d) Inlet hot water temperature,	(T_{h1}) 70 °C
Double pipe heat exchanger	
(a) Inner tube diameter (plain tube),	(d_i) 30 mm
(b) Outer tube diameter,	(d_o) 50 mm
(c) Test tube length	1600 mm
(d) Test tube thickness	1.5 mm
(e) Material of inner tube	Stainless Steel
(f) Material of outer tube	Stainless Steel
Twisted Tape	
Width	20 mm
Thickness	0.5 mm
Twist ratio	3,6,9
Tape pitch length	60, 120, 180 mm
Material	Aluminum

The average Nusselt number and the friction factor are based on the inner diameter of the test tube. Heat absorbed by the cold water in the annulus, Q_c can be written by

$$Q_c = \dot{m}_c C_{pw} \left(T_{c2} - T_{c1}\right) \qquad (1)$$

Where $\dot{m}_c$ is the mass flow rate of cold water; $C_{p,w}$ is the specific heat of water; T_{c1} and T_{c2} are the inlet and outlet cold water temperatures, respectively. The heat supplied from the hot water, Q_h can be determined by

$$Q_h = \dot{m}_h C_{pw} (T_{h1} - T_{h2}) \qquad (2)$$

Where $\dot{m}_h$ is the hot water mass flow rate; T_{h1} and T_{h2} are the inlet and outlet hot water temperatures, respectively. The heat supplied by the hot fluid into the test tube is found to be 2 to 4% higher than the heat absorbed by the cold fluid for thermal equilibrium due to convection and radiation heat losses from the test section to surroundings. Thus, the average value of heat transfer rate, supplied and absorbed by both fluids, is taken for internal convective heat transfer coefficient calculation.

$$Q_{average} = \frac{Q_c + Q_h}{2} \qquad (3)$$

For fluid flows in a concentric tube heat exchanger, the heat transfer coefficient (h_i) is calculated from

$$Q_{average} = UA\,\Delta T_{LMTD} \qquad (4)$$

Eq. (5) defines the overall heat transfer coefficient U, taking into consideration the convective thermal resistance and conductive thermal resistance

$$U^{-1} = \frac{1}{h_o} + \frac{d_o}{h_i d_i} + \frac{d_o}{2k} \ln \frac{d_o}{d_i} \qquad (5)$$

Where h_o is the annulus side heat transfer coefficient and can be calculate from [15]

$$Nu_o = \frac{h_o d_h}{k} = 0.023\ Re^{0.8}\ Pr^{0.3} \qquad (6)$$

Then

$$h_o = \frac{k}{d_h} Nu_o \qquad (7)$$

Where

$$d_h = d_o - d_i \qquad (8)$$

The local thermal conductivity (k) of the fluid is calculated from fluid properties at the local mean fluid temperature [15]. The logarithmic temperature difference by Eq. (9)

$$\Delta T_{LMTD} = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln\left(\frac{(T_{h1} - T_{c2})}{(T_{h2} - T_{c1})}\right)} \tag{9}$$

Thus

$$U = \frac{Q_{average}}{A\dfrac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln\left(\frac{(T_{h1} - T_{c2})}{(T_{h2} - T_{c1})}\right)}} \tag{10}$$

Reynolds number is based on the different flow rate at the inlet of the test section.

$$Re = \frac{\rho u d}{\mu} \tag{11}$$

Where μ is the dynamic viscosity of the working fluid.
Friction factor (*f*) can be written as:

$$f = \frac{\Delta P}{\left(\frac{L}{d}\right)\left(\rho \frac{u^2}{2}\right)} \tag{12}$$

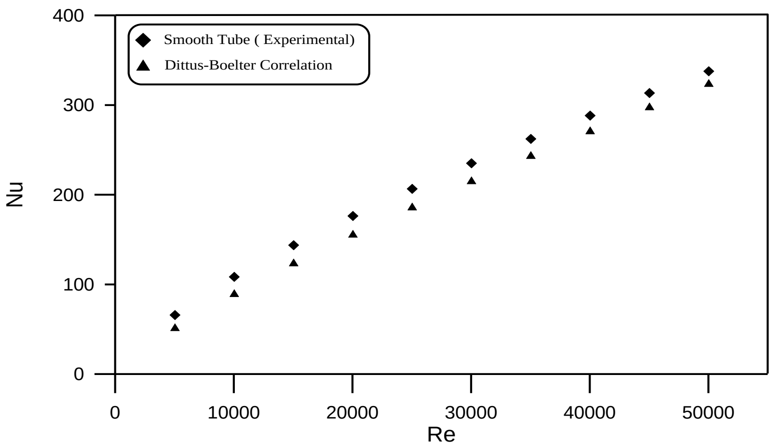
Results and discussion

In this section the following results are respectively presented, the validity of plain tube, the effect of (ECDT) without twisted tapes insert, and with twisted tapes insert with different twist ratio.

Confirmatory tests

The present experimental results on heat and fluid flow characteristics in a smooth tube were first reported in the form of (1) Nusselt number and (2) flow (friction factor, *f*). The results of the smooth tube were compared with the correlations of Dittus – Boelter, Blasius, and Petukhov (for the fully developed turbulent flow in circular tubes.

The comparisons for Nusselt Number and friction factor are shown in **(Figure 4)** and **(Figure 5)** respectively. Obviously, the experimental data are in good agreement with existing correlation, which are Dittus-Boelter correlation, Petnkhov Correlation, and Blasius Correlation. It is noted that the Nusselt Number differs by about 10% from Dittus-Boelter correlation, while the friction factor differs by about 15% from Blasius correlation.



(Figure: 4) Comparison of experimental data and empirical correlation of the plain tube for Nusselt number

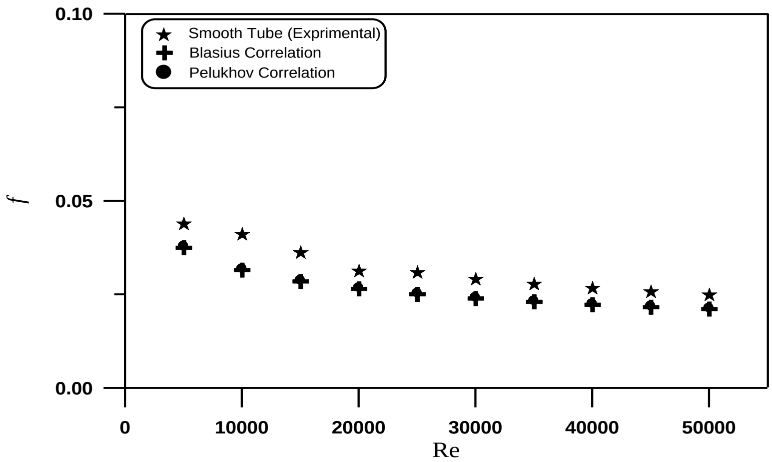
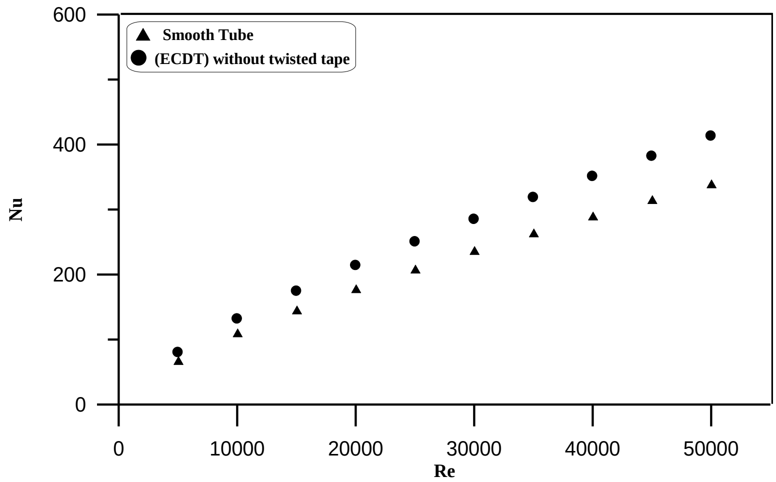


Figure: 5) Comparison of experimental data and empirical correlation of the plain tube for friction factor

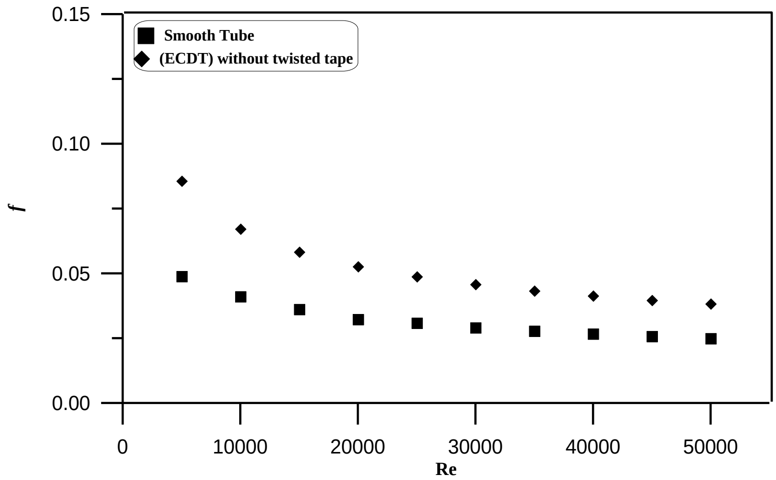
Effect of (ECDT) without twisted tape

Experimental results of the Nusselt Number (Nu) and friction factor (*f*) characteristics in (ECDT) without twisted tapes insert are presented in **(Figure 6)** and **(Figure 7)**. The Nusselt Number and friction factor of plain tube and the (ECDT) are plotted for comparison. The data show that the Nusselt Number, (therefore heat transfer coefficient) increases with increasing Reynolds Number (Re). The Nusselt Number of (ECDT) without twisted tape insert is 18%

higher than for the plain tube. **(Figure 7)** shows the influence of a (ECDT) on pressure loss, which indicates the friction in a heat exchanger. The increment of friction factor in the (ECDT) is of about 1.75 times of that of the plain tube.



(Figure: 6) Comparison of experimental data of the plain tube and (ECDT) without twisted tape insert for Nusselt number

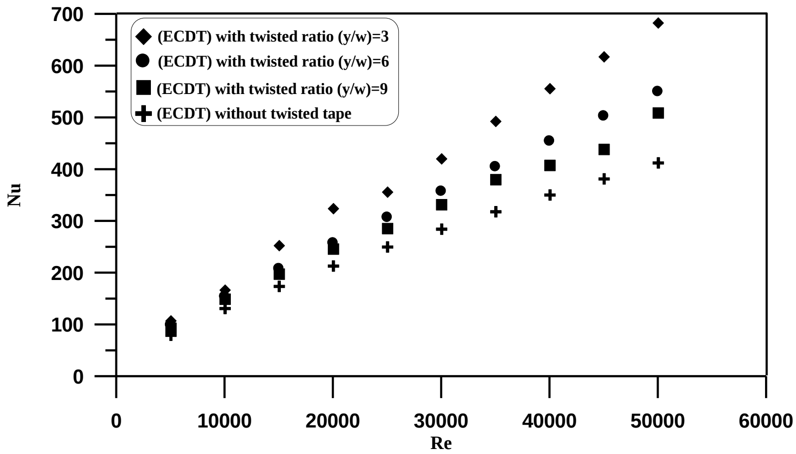


(Figure: 7) Comparison of experimental data of the plain tube and (ECDT) without twisted tape insert for friction factor

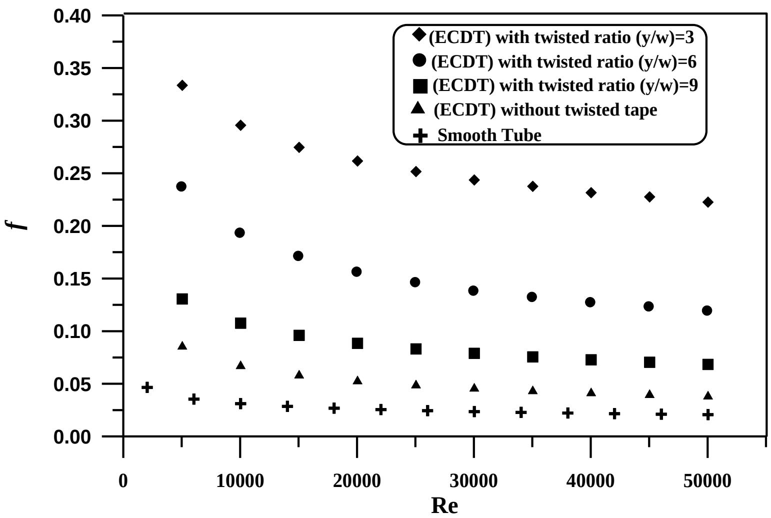
Effect of combined of (ECDT) and twisted tape

Experimental results of the Nusselt number (Nu) and friction factor (*f*) characteristics in (ECDT) combined with a twisted tape (y/w=3, 6, 9) are presented in **(Figure 8)**, and **(Figure 9)**, respectively. The Nusselt number and friction factor of the plain tube and (ECDT) acting alone are also plotted for comparison. The data show that the Nusselt number increases with increasing Reynolds number for the conventional turbulent tube flow. Depending on

Reynolds number (Re), the Nusselt number of the (ECDT) with the twisted tape insert is 15 to 45% higher than that in the (ECDT) acting alone and 52 to 280% higher than that in the plain tube **(Figure 8)**. It is noted that the increasing Nusselt number in the (ECDT) combined with a twisted tape is caused by the generating of pressure gradient along the radial direction, and this leads to redeveloping of boundary layer. The higher increase of the Nusselt number in this style of both turbulence and swirl flows is a result of the higher reduction of boundary layer thickness and increase of resultant velocity. **(Figure 9)** shows the effect of a (ECDT) combined with a twisted tape on pressure loss, which indicates the friction in a heat exchanger. The mean increase of friction factor in the combined devices is up to 2.12 times of that in the (ECDT) acting alone and 6.8 times of that in the smooth tube.



(Figure: 8) Effect of twist ratio on Nusselt Number for (a) y/w=3, (b) y/w=6, and y/w=9



(Figure: 9) Effect of twist ratio on friction factor for (a) y/w=3, (b) y/w=6, and y/w=9

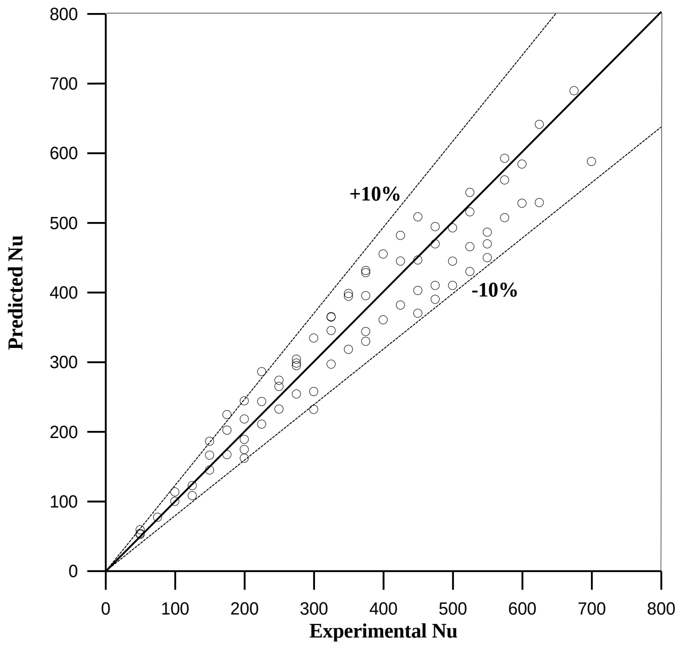
Effect of Twisted Ratio

(Figure 8) shows the effect of the twist ratios (y/w=3, 6 and 9) on the Nusselt number of a (ECDT) in combined with a twisted tape insert. The Nusselt number increases as the Reynolds number increases, and this trend is the most obvious for the smallest twist ratio (y/w=3). At the same Reynolds number, the Nusselt number increases with the decreasing twist ratio. The reasons are that, turbulence intensity and also residence time increase with the decreasing twist ratio. **(Figure 9)** shows the effect of twist ratio (y/w) on the friction factor in a (ECDT) combined with a twisted tape. It is obvious that the friction factor of the (ECDT) in conjunction with a twisted tape increases as twist ratio (y/w) decreases. Approximately, the friction factors for employing the (ECDT) combined with a twisted tape at the smallest twist ratio (y/w=3) are found to be 28% and 50% over those at the twist ratio, y/w=6 and 9, respectively. From the experimental results, the Nusselt number and friction factor in (ECDT) acting alone are function of Reynolds number (Re) only; while those in the (ECDT) fitted with a twisted tape are also function of twist ratio (y/w). The empirical correlations developed for a (ECDT) combined with a twisted tape are expressed as

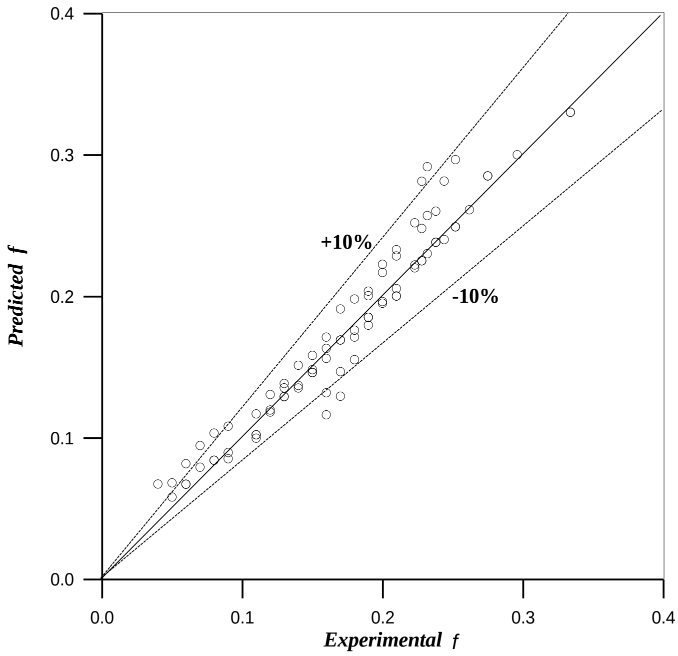
$$Nu = 0.054 \; Re^{.841} \; Pr^{0.4} \left(y/w \right)^{-0.165} \tag{13}$$

$$\boldsymbol{f} = 3.248 \; Re^{-0.2} \left(y/w \right)^{-0.435} \tag{14}$$

The fitted values of the Nusselt number and friction factor from **(Equation: 13)** and **((Equation: 14)** are compared with the experimental values and presented in **(Figure 10)** and **(Figure 11)**. The fitted values from the above equations are found agree well with the experimental data within ±10% for both Nusselt number and friction factor.



(Figure: 10) Comparison between experimental and predicted Nusselt Number



(Figure: 11) Comparison between experimental and predicted friction factor

Conclusions

An experimental study of fully developed turbulent flow in a (ECDT) combined with a twisted tape has been done. The influences of twist ratio on the heat transfer rate and friction factor characteristics have also been investigated. A (ECDT) in common with a twisted tape has significant effects on the heat transfer enhancement and friction factor. The heat transfer and friction factor are increase with decreasing the twist ratio (y/w). Depending on the twist ratio, the heat transfer rate and friction factor in the (ECDT) with twisted tape, are 15 to 45% and 6.8 times of those in the smooth tube respectively. The empirical correlations for the Nusselt number and the friction factor based on the present experimental data are also presented.

Nomenclature

A	surface area of test tube,	m^2
Cp	specific heat at constant pressure,	J/kg K
D	tube diameter,	m
f	friction factor	
h	convective heat transfer coefficient,	W/m^2 K
k	thermal conductivity,	W/m K
L	length of tube,	m
$\dot{m}$	mass flow rate,	kg/s
Nu	Nusselt number	
Pr	Prandtl number	
Q	heat transfer rate,	W
Re	Reynolds number	
T	temperature,	°C
u	mean velocity in tube,	m/s
U	overall heat transfer coefficient,	W/m^2 K
ρ	density,	kg/m^3
μ	dynamic viscosity,	kg/m s

Subscripts

c	cold
h	hot
i	inner
o	outer
w	water

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