

## Heat Transfer and Pressure Drop Characteristics of Turbulent Flow In a Tube Fitted With Conical Ring and Twisted Tape Inserts

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Received on: 17/8/2010

Accepted on: 5/1/2011

### Abstract

The present work shows the results obtained from experimental investigations of the augmentation of turbulent flow heat transfer in a horizontal tube fitted with combined conical-ring turbulators and a twisted-tape swirl generator. The air is the working fluid for Reynolds number range of 5000-23000 under constant wall heat flux thermal boundary condition. In this study, two enhancement heat transfer devices are used. One is the conical-ring used as a turbulator and placed in the tested tube with constant diameter ratio ( $d/D=0.538$ ) and the other is the twisted-tape swirl generator placed at the core of the conical-rings. Three twisted-tapes of different twist ratios,  $Y=2, 3$ , and  $6$ , are introduced in each run. The experimental data obtained are compared with those obtained from the plain tube and from the literature to ensure the validation of experimental results. Correlations for Nusselt number, friction factor, and enhancement efficiency are developed. It is observed that the heat transfer process enhances by using combined conical-ring and twisted-tape inserts or each one alone when compared to plain tube at the same mass flow rate, and this enhancement increases as twist ratio decreases for the case of combined insertion.

**Keywords:** heat transfer, conical ring, twisted tape.

### خصائص انتقال الحرارة وفرق الضغط لجريان مضطرب بانبوب مزود لحلقة مخروطية وشريط ملتو كادوات حشر

#### الخلاصة

يبين العمل الحالي النتائج التي تم الحصول عليها من التحريات العملية لتحسين انتقال الحرارة للجريان المضطرب داخل انبوب افقي مجهز بمضطربات حلقة مخروطية ومولدات دوامة عبارة عن شريط ملتو. مائع الاختبار هو الهواء لرقم  $Re$  يتراوح بين  $5000 - 23000$  تحت فيض حراري منتظم حدي حراري. يتم تحسين عملية انتقال الحرارة في تلك الدراسة باستخدام وسيلتين. الاولى عبارة عن حلقة ذات شكل قطع مخروطي مقطوع تستخدم كعامل مضطرب وتوضع داخل انبوب الاختبار بنسبة قطر ثابتة ( $d/D=0.538$ ) والوسيلة الاخرى عبارة عن شريط ملتو مولد للدوامات يوضع في قلب الحلقات المخروطية. استخدمت ثلاثة نسب التواء للأشرطة الملتوية هي  $2, 3$  و  $6$  في كل اختبار. قورنت البيانات العملية بتلك التي تم الحصول عليها من الانبوب الخال من هذه الوسائل ومن الادبيات السابقة للتأكد من صحة النتائج العملية استنبطت معادلات تجريبية لرقم  $Nu$ , عامل الاحتكاك وكفاءة التحسين. لوحظ ان عملية انتقال الحرارة تتحسن باستخدام الحلقات المخروطية والشريط الملتو معا" او كل على انفراد مقارنة" بحالة الانبوب البسيط عند نفس معدل التدفق الكتلي, وان ذلك التحسين يزداد بنقصان نسبة الالتواء لحالة الحشر المركب.

## Introduction

The heat transfer in heated pipe is a classical problem that is still an active area of research for different thermal application such as flat plate solar collectors, nuclear reactors and heat exchangers. Heat transfer enhancement technology has been developed and widely applied to heat exchanger applications to accommodate high heat fluxes. So, there has been a great attempt to reduce the sizes and cost of the heat exchanger, and energy consumption. The most significant variable in reducing these characteristics is the overall heat transfer coefficient and pressure drop, which relatively leads to less capital cost and to another advantage of a reduction in the temperature driving force, which increases the second law efficiency and decreases entropy generation.

Industrial heat transfer equipments generally are operated in turbulent swirl flow conditions where their performance interms of energy transfer rate is high, compared with laminar flow by virtue of the high degree of turbulence in turbulence/swirl flow. Also in turbulent flow, a high intensity of turbulence will enhance the rapid mixing of fluid properties, and the mixing can help to increase the effective area of heat transfer, leading to higher heat transfer rates [6]. The swirl flow devices can be classified into two kinds: the first is the continues swirl flow in which the swirling motion persists over the whole length of tube; so the heat transfer coefficient and pressure drop keep constant with the axial distance such as twisted-tape inserting and wire coil inserting. The other kind of swirl flow is the decaying swirl flow in which the swirl is generated at the entrance of the tube and decays along the flow path leads to

decreasing the heat transfer coefficient and pressure drop with the axial distance for example the radial guide vane swirl generator, the tangential flow injection device and the snail swirl generator.

Many researchers investigated heat transfer enhancement concept in which swirl was introduced in the flow. The problem of fully developed, laminar and turbulent, uniform property flow in a tube containing a twisted-tape had been formulated by **Date** [1] numerically interms of partial differential equations of momentum and heat transfer. Results show that the using of augmented turbulent viscosities made the Nusselt number predictions more realistic. **Svashanmugam and Sundaram** [2] studied experimentally the improvement in performance of a double pipe heat exchanger fitted with twisted tape as a turbulence promoter with various twist ratios. A maximum percentage gain of energy transfer rate was obtained for the twisted tape of smallest twist ratio. **Promvonge and Eiamsa** [3] submitted study into heat transfer and turbulent flow friction in a circular tube fitted with conical-nozzle turbulators. **Eiamsa et al** [4] submitted an experimental investigations of heat transfer and pressure drop characteristics of turbulent flow through circular tube fitted with various regularly-spaced twisted tape. Results show that the lowest value of spacing twisted tape gives the heat transfer lower than full length twisted tape around 5-15% while it can be decreased the pressure drop around 90%. **Promvonge** [5] used several conical rings as turbulators to increase convection heat transfer in a uniform heat flux tube. Three different diameter ratios of the ring to tube diameter were introduced in this

study, and for each ratio, the rings were placed with three different arrangements (converging conical ring CR, diverging conical ring DR, and converging-diverging conical ring CDR). An augmentation of up to 197%, 333%, and 237% in Nusselt number was obtained in the turbulent flow for the CR, DR, and CDR arrays; respectively. Results show that the enhancement efficiency decreases as Reynolds number increases. Heat transfer, friction factor and enhancement efficiency characteristics in a circular tube fitted with conical-ring turbulators and a twisted-tape swirl generator had been investigated experimentally by **(Promovong and Eiamsa-ard [6]. Promovong [7]** investigated experimentally influences of insertion of wire coils in conjunction with twisted tapes on heat transfer and turbulent flow friction characteristics in a uniform heat flux, circular tube using air as the test fluid. The results indicated that the presence of wire coils together with twisted tapes leads to a double increase in heat transfer over the use of wire coil/ twisted tape alone especially at smaller twist and coil pitch ratios under the same conditions. Results show that the heat transfer rates from using both nozzle-turbulators, in general, were found to be higher than that from the plain tube at range of 236 to 344%. **Kumar and Prasad [8]** developed and tested the modified solar water heater having twisted tape (artificial roughness) inserted inside the tubes along the plain one. Experimental results show that in the range of parameter investigated, thermal enhancement factor varied between 1.18 to 2.7 and the maximum value of collector efficiency increased by about 30% compared to that of plain ones at same operational condition.

**Murugesan et al [9]** studied experimentally the heat transfer and friction factor in a circular tube fitted with full length twisted tape with trapezoidal-cut. The results show that there was a significant increase in heat transfer coefficient and friction factor for tape with trapezoidal cut. **Eiamsa-ard et al [10]** presented thermohydraulic investigation of turbulent flow through a round tube equipped with twisted tapes consisting of center wings (WT) and alternate-axes (T-A). All twisted tapes used were twisted at constant twist length. The wings were generated along the center line of the tape with three different angles of attack ( $43^\circ$ ,  $53^\circ$ , and  $74^\circ$ ). Results show that the heat transfer rate increased with increasing angle of attack.

In this study present the technology of conical ring turbulators and twisted tape swirl generator inserted inside a circular tube which provides a simple technique for enhancing the convective heat transfer by introducing swirl flow and by disrupting the boundary layer at the tube surface due to repeated changes in the surface geometry.

#### **Experimental Setup**

A schematic layout of the open test loop is shown in Fig.(1). The loop consisted of 6.5 KW blower, orifice meter to measure the flow rate, and the heat transfer test section. The inside of outside diameters of the aluminum test tube having a length 1200mm and 3.7mm thickness, are 51.9mm and 59.3mm ; respectively. The tube was heated by continually winding flexible electrical wires providing a uniform heat flux boundary condition. The electrical output power was controlled by a variac transformer to obtain a constant heat flux along the entire length of the test section. The outer

surface of the test tube was well insulated to minimize convective heat loss to surroundings. A selector switch and digital thermometer were used to get multi-channel temperature measurements. Eighteen thermocouples were tapped on the local wall of the tube. One thermocouple was placed at the inlet of the test section and two thermocouples were placed at outlet of test section to measure the temperature of the bulk air at inlet and outlet of test section, respectively. The mean local wall temperature was determined by means of calculations based on the reading of chrome alumel thermocouples.

Fig.(2a&b) shows the conical-ring turbulators and twisted tape used in the present experiments. The conical-nozzle turbulator was made of aluminum with 51.9 mm (1.0 D) in length (l) and its small end diameter (d) was 28 mm (0.538D), with a 1.5mm thickness and 180mm as a pitch length (s) . The twisted-tapes made of steel strip with twist ratios of  $Y=y/w=2,3$  and 6, were used in the present study. The twist ratio is defined herein as the ratio of the pitch or one twist length (w, 180°) to the tape width (y).

**Data Reduction**

In the present work, air is used as the tested fluid and flowed through a uniform heat flux and insulated tube. The net heat input to the fluid was determined from the electrical energy input to the system as follows:

$$Q_1 = V \cdot I - Q_{\text{losses}} \quad \dots\dots\dots(1)$$

Where  $Q_{\text{losses}}$  is the heat losses by conduction through the lagging.

The enthalpy rise  $Q_2$  of the fluid was calculated from the equation

$$Q_2 = \dot{m} C_p (T_{bo} - T_{bi}) \quad \dots\dots\dots(2)$$

The set of data taken in a run were accepted only if the difference between the net heat input  $Q_1$  and the enthalpy rise  $Q_2$  was less than 5%. In such a case the actual heat input to the test section was taken as the average of  $Q_1$  and  $Q_2$ .

$$Q'' = \frac{Q_1 + Q_2}{2} \quad \dots\dots(3)$$

The bulk temperature of the fluid at any axial position along the tube axis was computed by assuming a linear temperature variation along the length since the experiment results show that the temperature distribution along the tube is linear. The steady state of the heat transfer rate is established after 2-3 hours. In this state, all thermocouple readings were noted.

The heat transfer coefficient by convection from the test section can be written as:

$$h = \frac{Q''}{A(T_{\text{ave}} - T_b)} \quad \dots\dots\dots(4)$$

where

$$T_{\text{ave}} = \frac{\sum_{i=1}^{18} T_{w_i}}{18} \quad \dots\dots(5)$$

and

$$T_b = \frac{T_{bi} + T_{bo}}{2} \quad \dots\dots\dots(6)$$

$T_{w_i}$  is the local surface temperature at the outer wall of the inner tube. The average surface temperature  $T_{\text{ave}}$  is calculated from 18 points of  $T_w$  lined between the inlet and the exit of the test tube.

The average Nusselt number, Nu is estimated as follows:

$$Nu = \frac{h \cdot D}{\kappa} \quad \dots\dots\dots(7)$$

The Reynolds number is given by:

$$Re = \frac{U \cdot D}{\nu} \quad \dots\dots\dots(8)$$

The friction factor ( $f$ ) can be written as:

$$f = \frac{\Delta P}{(L/D)(\rho U^2 / 2)} \quad \dots\dots (9)$$

Where  $U$  is the mean air velocity in the tube. All of the thermophysical properties of the air are determined at the overall bulk temperature from Eq.(6).

### Results and Discussion

The experimental results on heat transfer characteristics in a circular tube with combined effects of conical-ring and twisted tape are presented. The results of the plain tube are compared with the past correlations for the fully developed turbulent flow in circular tubes as shown in Fig.(3) which shows the variation of average Nusselt number versus Reynolds number with  $\pm 6.5\%$ - $7.5\%$  in comparison with past correlations of Promvonge and Eiamsa-ard [5] and Dittus and Boelter, 1934 [11]. As can be seen from this figure that the present work agrees well with the available correlations.

The temperature distribution along the tube axes for different twisted-tape ratios is shown in Fig.(4). The figure show that the values of temperature increase along the tube length and decrease as twist ratio decreases. The cases of empty (plain) tube and the tube fitted with conical-ring alone give the higher level of temperature distribution, respectively .Fig.(5) shows the variation of average Nusselt number versus Reynolds number for various twist ratios . It is noticed that the value of the average Nusselt number increases as Reynolds number increases for the same conditions and as twist ratio decreases for the case

of combined twisted-tape and conical-ring. In addition to these facts, the heat transfer process in a tube fitted with combined conical ring and twisted tape is better than that in a tube fitted with conical ring alone. The physics behavior of heat transfer process and fluid flow can be explained as follows:

The use of conical-ring inserts leads to considerably higher heat transfer rates than the plain tube due to the effects of reverse flow and boundary layer disruption which help to enhance the convection heat transfer and momentum process. The reverse flow increases the effective axial Reynolds number leads to improve the convection currents, since the same throughput must be accommodated by a reduced cross-sectional area of flow. As a result, more severe mean velocity and temperature gradients will be existed, which produce higher fluxes of heat and momentum due to larger effective driving potential for each. Using of twisted tape together with conical ring causes spiral flow along the tube length, so, the chaotic mixing between the core and the wall regions will be better, thus enhancing the convective process. Considering the lower of twist ratio offer higher heat transfer rate than the higher twist ratio because intensity of turbulence and flow length obtained from lower twist ratio are higher than those at higher twist ratio.

The variation of friction factor with Reynolds number is shown in Fig.(6). It shows that the mean fanning friction factor decreases with increase in Reynolds number. The friction factor with combined effects of conical-ring and twisted tape is higher than that in the tube fitted with conical-ring alone and the empty tube, and the smaller twist

ratio leads to higher friction factor due to increase in swirl flow with decrease in twist ratio, leading to higher tangential contact between secondary flow and the wall surface of the tube.

With the help of the experimental data in the test tube, the following empirical Nusselt number and friction factor relationships are derived for the case of combined conical-ring and twisted-tape case, as follows:

$$Nu = 1.121 Re^{0.231} Pr^{0.3} (d/D)^{-1.12} Y^{-0.021} \dots\dots(10)$$

$$f = 20.23 Re^{-0.13} (d/D)^{-2.65} Y^{-0.13} \dots\dots(11)$$

Figs(7 & 8) show comparisons between the present experimental data and the predictions obtained from the present correlations; within  $\pm 9\%$  in comparison with experimental data for the average Nusselt number and within  $\pm 8.5\%$  for the friction factor.

**Performance Evaluation**

As preliminary design guidance for selection of a technique, the heat transfer enhancement efficiency can be evaluated based on the power consumption per unit mass of fluid. The enhancement efficiency ( $\xi$ ) is defined as the ratio of the heat transfer coefficient for the tube fitted with combined conical-ring and twisted tape ( $h_c$ ) or conical ring alone to that for the plain tube ( $h_p$ ) at a constant pumping power (pp) as follows [12]:

$$\xi = \frac{h_c}{h_p} \Big|_{PP} \dots\dots\dots (12)$$

As can be seen in Fig.(9), the variation of performance ratio decreases with increasing of Reynolds number, and the enhancement efficiency increases with reduction of the diameter ratio

and twisted tape, especially at lower Reynolds number. This can be expressed that the conical-ring turbulators are not feasible in terms of energy saving at higher Reynolds number values. It is worth noting that the enhancement efficiency ( $\xi$ ) of the case of conical-ring with twisted tape is found to be the highest compared to that of the case of conical ring alone. The empirical equations of the enhancement efficiency are deduced as shown below:

For tube fitted with conical-ring alone

$$\xi = 9.34 Re^{-0.121} (d/D)^{-0.114} \dots\dots\dots(13)$$

For tube fitted with combined conical-ring and twisted tape

$$\xi = 12.23 Re^{0.132} (d/D)^{-0.112} Y^{-0.011} \dots\dots\dots(14)$$

A plot of the thermal enhancement efficiency and twist tape ratio has been shown in Fig.(10). A straight line relation between the thermal enhancement efficiency and twist tape ratio has been shown in this figure which could further be represented in the form of straight line equation correlating the thermal enhancement efficiency and twist tape ratio written as:

$$\xi = 0.768 + 0.192Y \dots\dots\dots (15)$$

where  $2 \leq Y \leq 6$ , and 0.768 is the value of intercepts on ordinate and 0.192 is the value of slop of the straight line. This equation is very useful in the applications that include uniformly heated tube fitted with combined conical-ring and twisted tape with turbulent air flow in the range of Reynolds number from 5000 to 23000.

**Conclusions**

1. Conical-ring with or without twisted tapes could be inserted inside the flow tube

for enhancing heat transfer rate, however, pressure drop has been found to increase.

2. Correlations for the Nusselt number, friction factor and enhancement efficiency based on the present experimental data are introduced for practical use.
3. Heat transfer coefficient and friction factor increases with the decrease in twist ratio compared with plain tube.
4. Combined conical-ring and twisted tape inserted inside tube gives higher heat transfer rates than that tube fitted with conical-ring alone.
5. In all the cases , the heat transfer rates increase at the expense of high friction losses.

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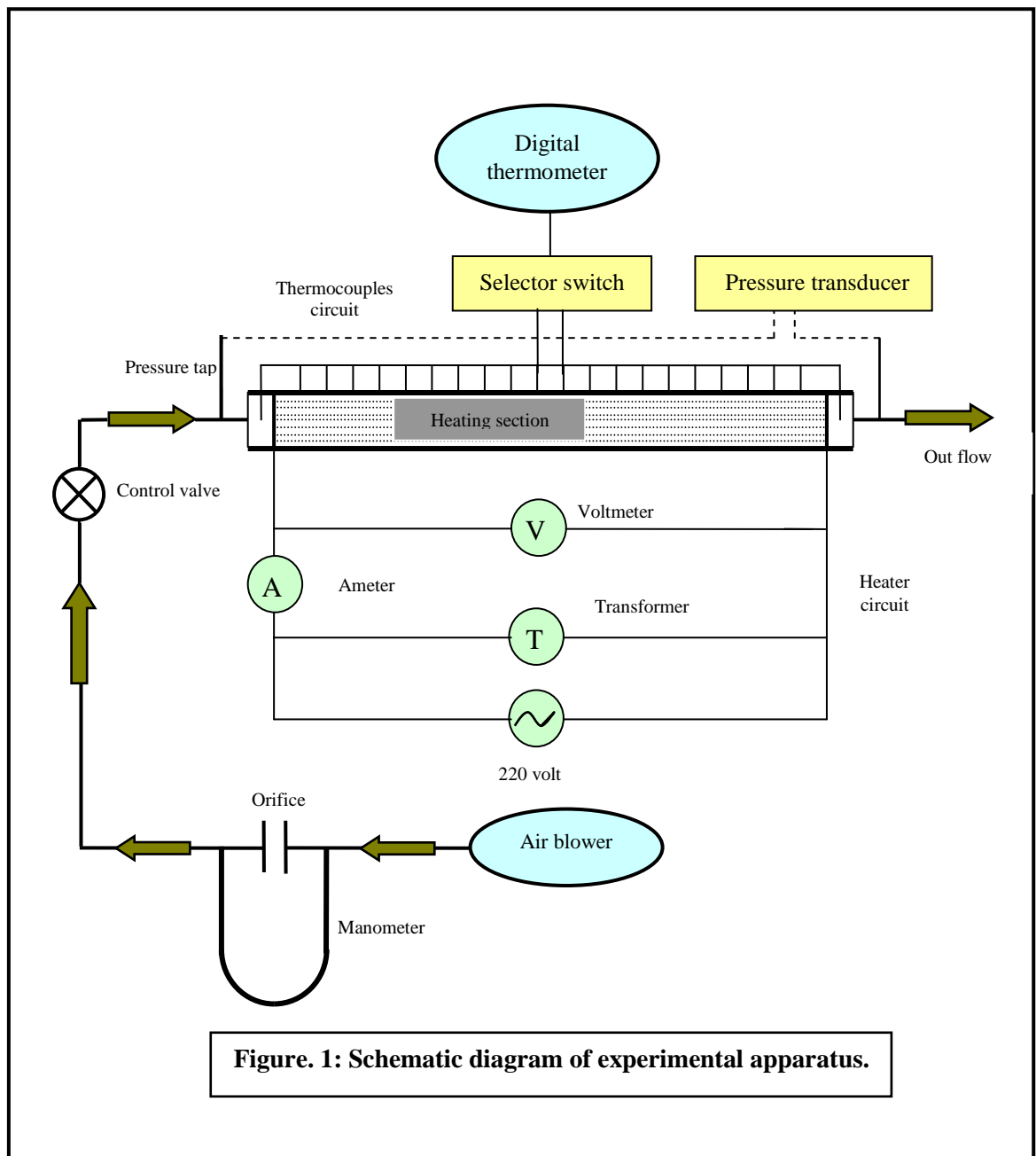
#### Nomenclature

A heat transfer area,  $m^2$   
 $C_p$  specific heat of air, J/kg.K  
 $d$  small end diameter of conical ring, m  
 $D$  inner diameter of test tube, m  
 $f$  friction factor  
 $h$  heat transfer coefficient,  $W/m^2.K$   
 $I$  current, A  
 $\kappa$  thermal conductivity of air,  $W/m.K$   
 $L$  length of test tube, m  
 $l$  conical ring length, m  
 $\dot{m}$  mass flow rate, kg/s  
 $Nu$  Nusselt number  
 $\Delta P$  pressure drop, Pa  
 $Pr$  Prandtl number  
 $Q''$  heat flow rate, W  
 $Re$  Reynolds number  
 $s$  ring pitch, m  
 $t$  thickness of test tube, m  
 $T$  temperature,  $^{\circ}C$   
 $U$  mean axial velocity, m/s  
 $y$  twist pitch, m  
 $Y$  twist ratio  
 $V$  voltage, volt  
 $w$  width of tape, m  
 $\xi$  thermal enhancement efficiency  
 $\nu$  kinematics viscosity,  $m^2/s$   
 $\rho$  density,  $kg/m^3$   
 $\mu$  dynamic viscosity,  $Ns/m^2$

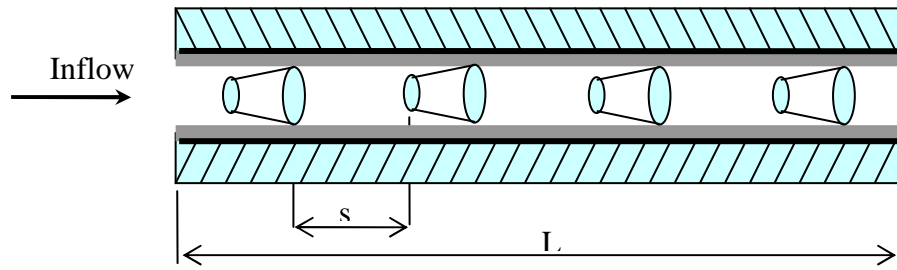
#### Subscripts

a air  
ave average  
b bulk  
i inlet  
m mean  
o out  
p plain tube  
pp pumping power  
w wall

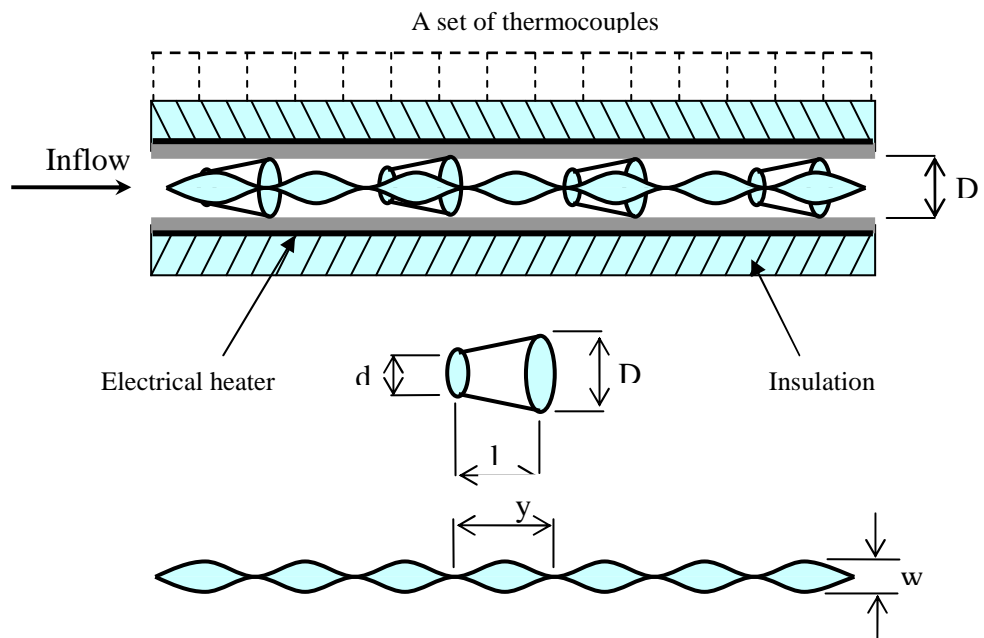




**Figure. 1: Schematic diagram of experimental apparatus.**



(a) Conical ring arrangement



(b) Geometry of conical ring and twisted tape arrangement

Figure.2: Test tube fitted with conical-ring and twisted tape insert.

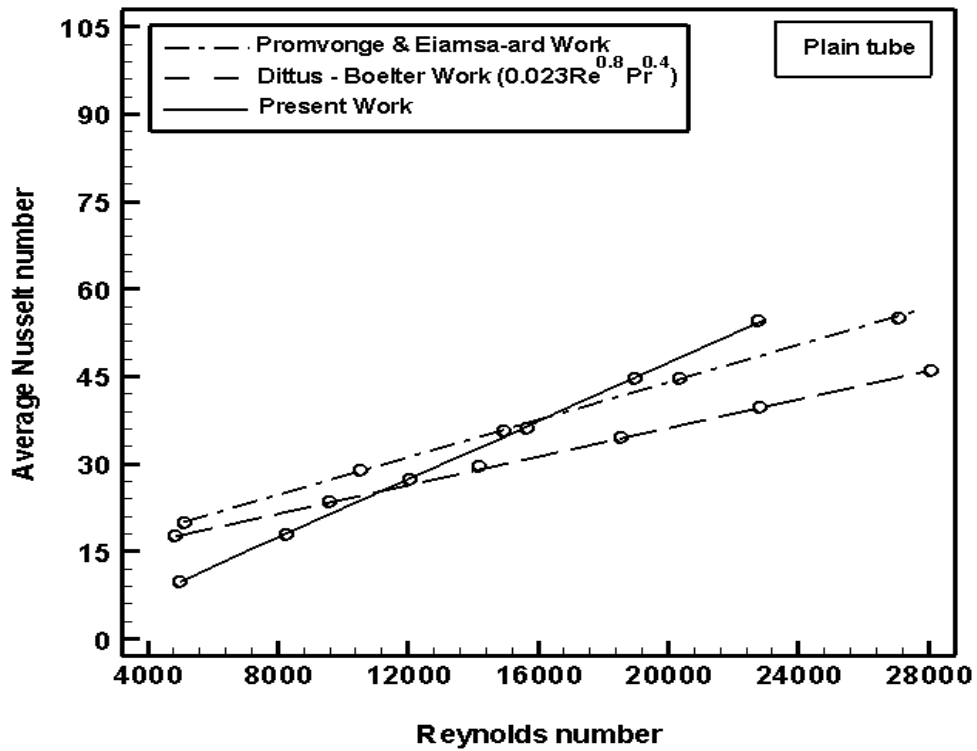


Figure (3): Verification of Nusselt number of plain tube.

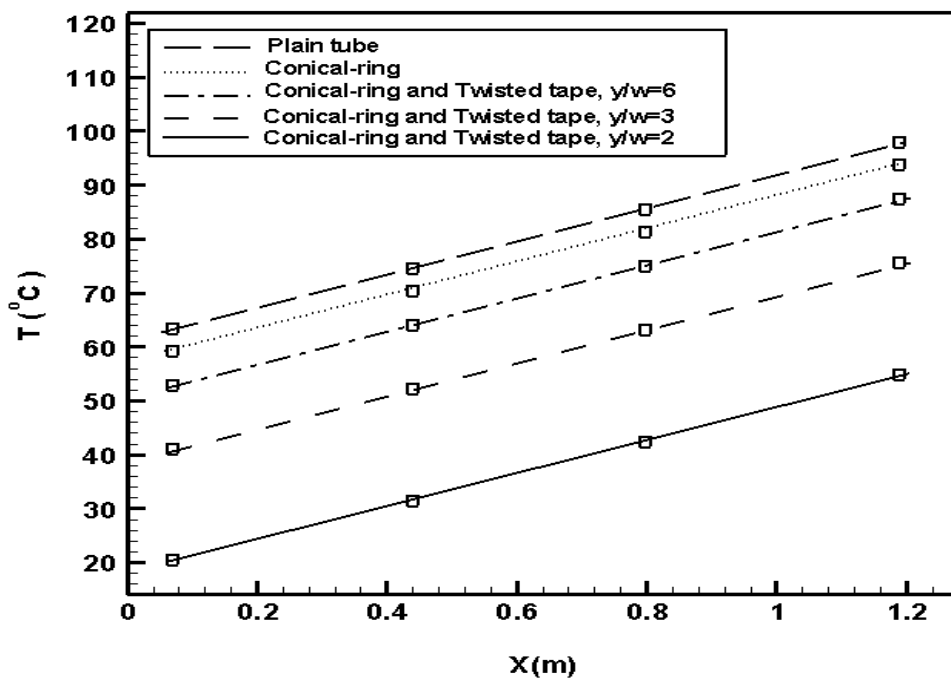


Figure (4): Temperature distribution along the tube axes for various twisted-tape ratios.

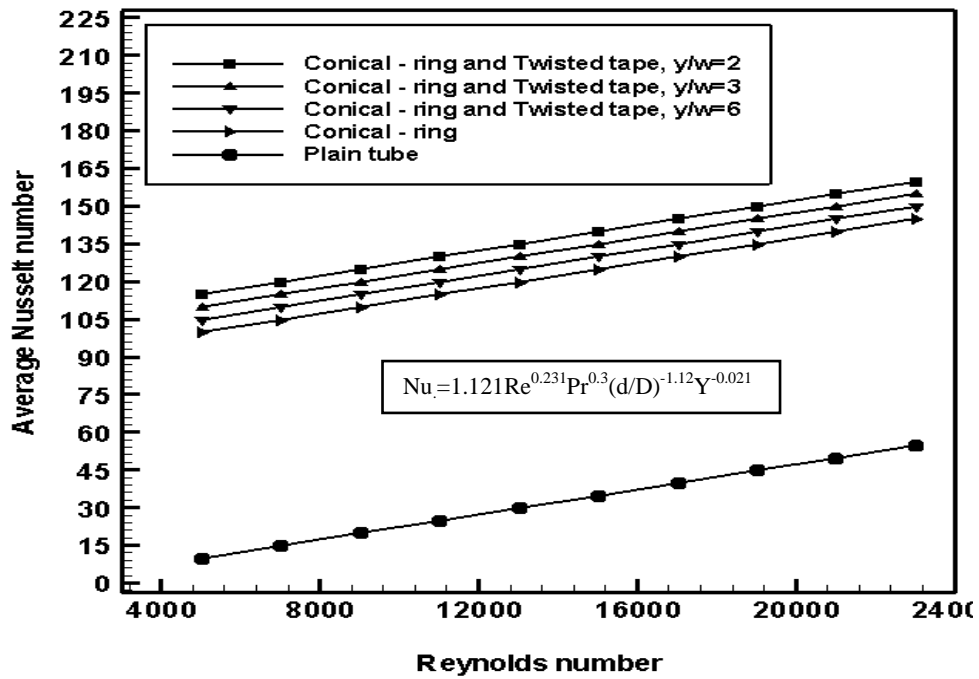


Figure (5): Nusselt number versus Reynolds number with various twisted-tape ratios.

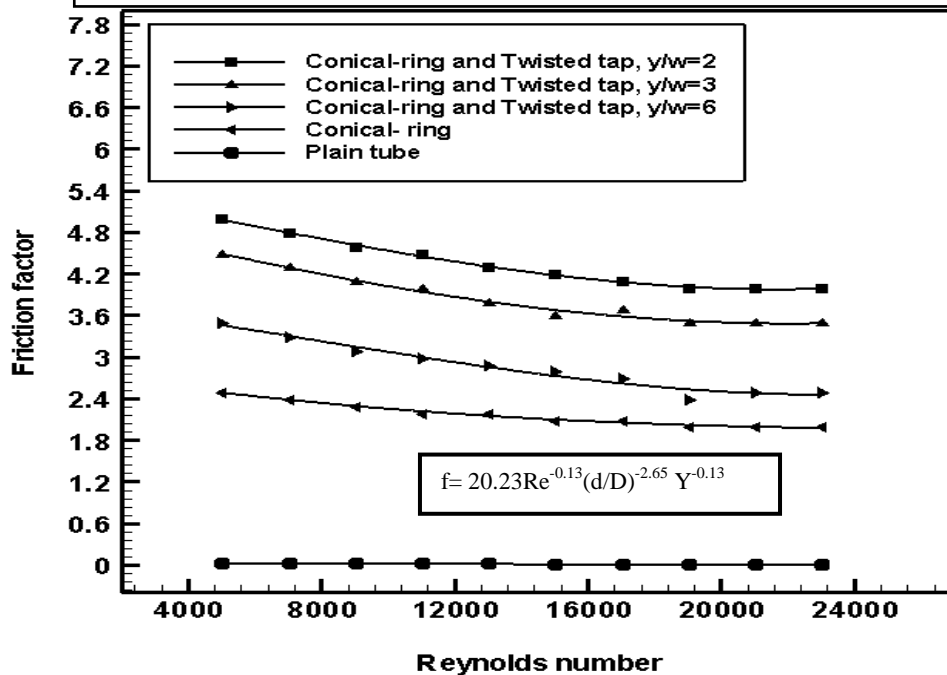


Figure (6): Friction factor versus Reynolds number with various twisted-tape ratios.

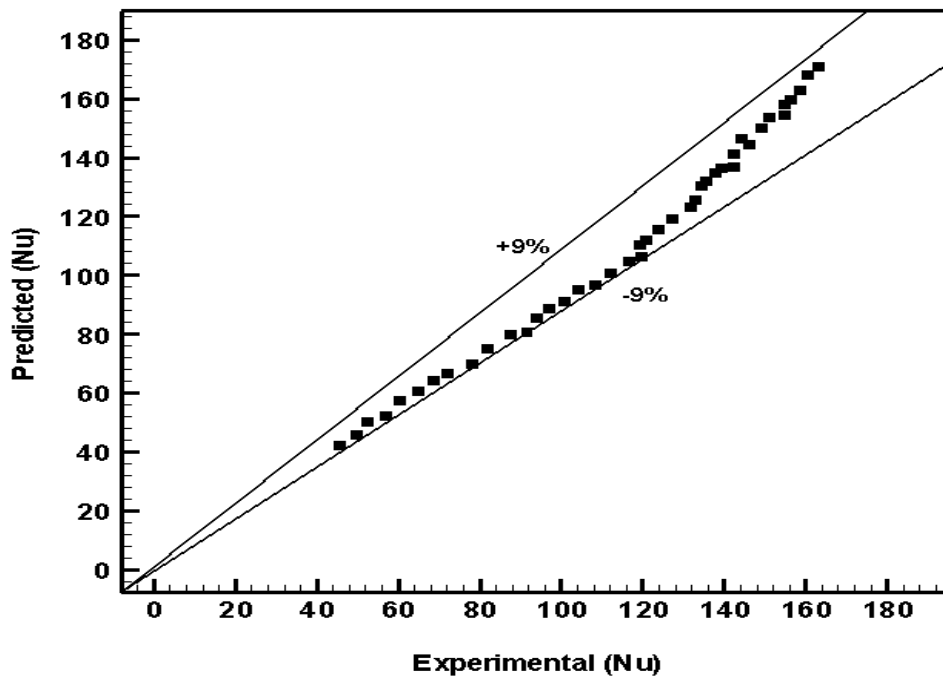


Figure (7): Predicted Nusselt number against experimental Nusselt number.

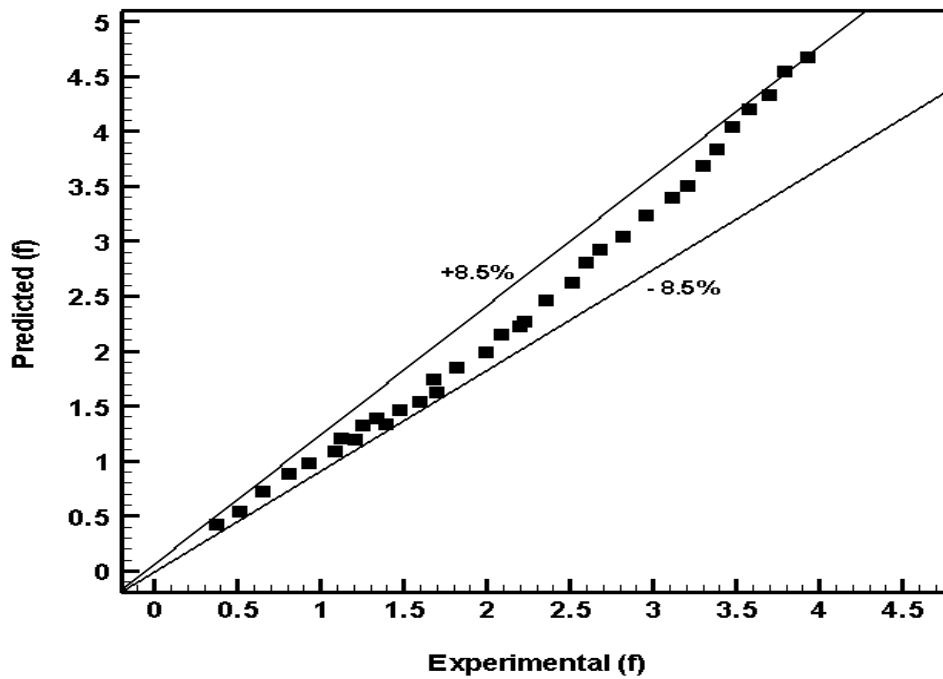


Figure (8): Predicted Nusselt number against experimental Nusselt number.

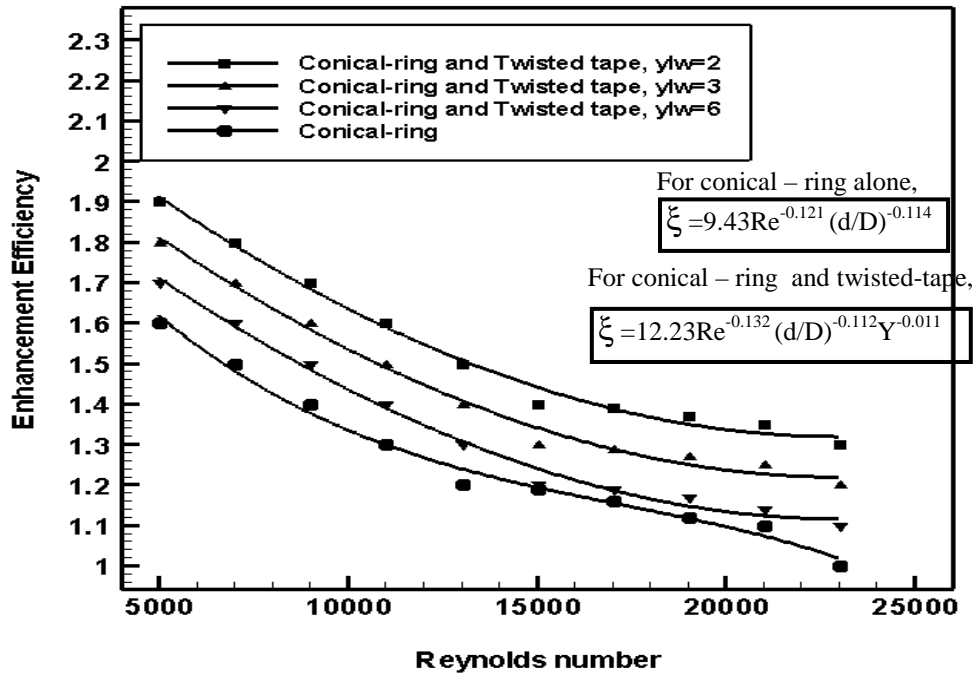


Figure (9): Enhancement efficiency versus Reynolds number with various twisted-tape ratios.

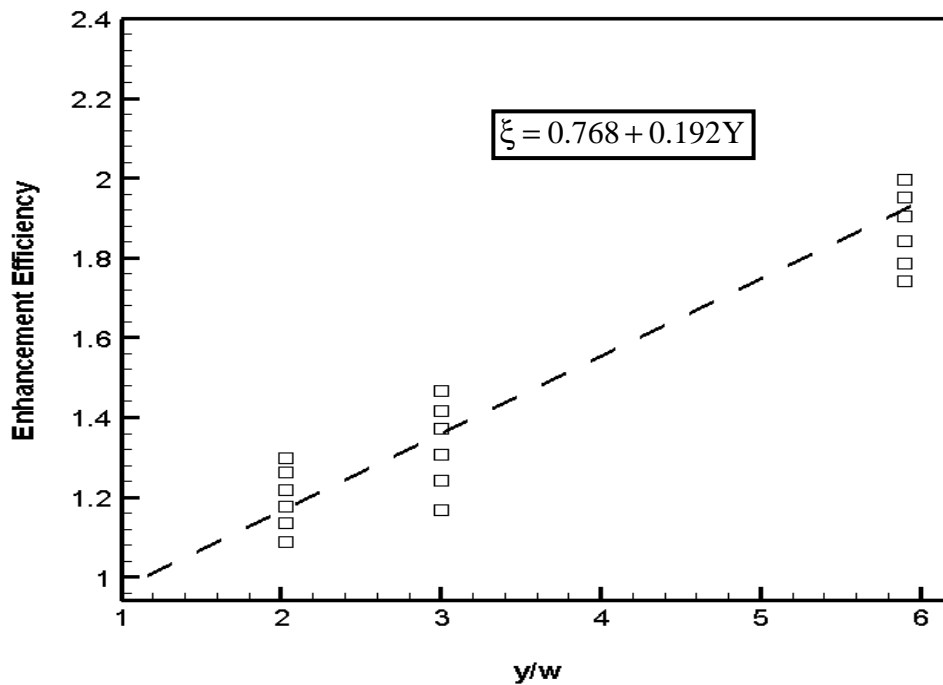


Figure (10): Enhancement efficiency against twisted tape ratios.