UNIVERSITY OF TURKISH AERONAUTICAL ASSOCIATION
INSTITUTE OF SCIENCE AND TECHNOLOGY

PARAMETRIC STUDY OF HEAT TRANSFER AND PRESSURE DROP CHARACTERISTICS IN A TUBE USING DIFFERENT TYPES OF INSERTS

Master Thesis

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The heat exchanger performance can be enhanced by a number of enhancement techniques. The study reported the employee of different inserted tapes fitted in a single tube to enhance the mixing of fluid, that leads to higher rate of convective heat transfer in comparison with that of the typical inserted tape. The tube has been fabricated from copper material of (23 mm) inner diameter, (1000mm) length. Experimental investigation of twisted tape has been performed in this thesis for horizontal circular insulated tube under constant heat flux condition in turbulent flow region to study its effect on the heat transfer improvement and pressure drop. Heat transfer and friction factor experiments were implemented with four variant geometries of twisted tapes made from copper materials by changing their twist ratios and the perforated shapes along the tape center. The results show that, the use of twisted tapes leads to increase the heat transfer enhancement and friction losses more than using distilled water in smooth tube. Also, the heat transfer and friction losses increase with the increases in the Reynolds number and with a decrease in the twist ratio. The perforated circular shape shows better performance on heat transfer than perforated square shape. Where, the best performance of twisted tapes on heat transfer enhancement is for the perforated circular clockwise-counter clockwise twisted tape over the other twisted tapes used. The obtained maximum Nusselt number ratio ($\overline{Nu}_{\text{twisted tape}} / \overline{Nu}_{\text{plain tube}}$) was (5.3) which occurred in perforated circular clockwise-counter clockwise twisted tape with twist ratio (TR = 4) at Reynolds number (4140), and the maximum factor of thermal performance was (4.2) at Reynolds number (4140). Empirical correlations are developed to predict the average Nusselt number and factor of friction of the fluids flow inside twisted tapes inserted

tubes. These correlations can predict the experimental data with a maximum error of (±7%).

Keywords: Heat transfer augmentation, Nusselt number, Twisted tape.

OZET

FARKLI TÜRDE UÇLAR KULLANAN BİR TÜPTE ISI AKTARIMI VE BASINÇ DÜŞMESİ ÖZELLİKLERİNİN PARAMETRİK İNCELEMESİ

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Isı değiştiricinin performansı bir takım takviye teknikleri ile ciddi ölçüde geliştirilebilirdir. Mevcut çalışma sıvı karışımını geliştirmek için tek tüplü ısı değiştiriciye uyan değişik biçimli ve kıvrımlı şeritlerin kullanılmasının düz kıvrımlı şeritlere oranla daha yüksek ısı transfer oranı sağladığını öne sürmektedir. Tüp (23mm) iç çap ve (1000 mm) uzunluğuna sahip olup, bakır malzemeden üretilmiştir. Bu tezde, kıvrımlı şeridin deneysel araştırılması ısı transferini arttırma ve sürtünme faktörü üzerindeki etkilerini incelemek için sabit ısı akışı altında ve karışık akıntı bölgesinde yatay, dairesel ve yalıtımlı tüp ile yapılmıştır. Isı transferi ve sürtünme faktörü deneyleri bakır malzemeden yapılmış şeritlerin bükülme oranları ve şerit merkezi oyunca oluklu şekilleri değiştirilerek elde edilen dört farklı geometriye sahip kıvrımlı şeritler üzerinde yapılmıştır. Deneysel sonuçlar kıvrımlı şeritlerin kullanılmasının düz tüpte damıtılmış su kullanılmasına oranla büyük ölçüde ısı transferi artışının ve sürtünme azalışının ortaya çıktığını göstermiştir. Ayrıca ısı transferi ve sürtünme kaybının Reynolds sayısında artış ve bükülme oranında azalış ile arttığı gözlemlenmiştir. Oluklu dairesel şekil ısı transferi açısından oluklu kare şekle göre daha iyi performans göstermektedir. Burada, ısı transferi artışında kıvrımlı şeritlerin en iyi performansı kullanılan diğer kıvrımlı şeritlere göre oluklu, dairesel, saat yönünde ve saat yönünün tersinde kıvrımlı şeritler ile elde edilmiştir. Elde edilen maksimum Nusselt sayı oranı ($Nu_{\text{kıvrımlı şerit}}/Nu_{\text{düz tüp}}$) (5.3) olup, bu oran (4140) Reynolds sayısında bükülme oranı (TR=4), Reynolds sayısında termal performans faktörü (4.2) olan, damıtılmış su kullanan, oluklu, dairesel, saat yönünde-saat yönünün tersinde kıvrımlı olan şerit ile elde edilmiştir. Kıvrımlı şerit sokulmuş tüplerin içerisindeki sıvı akışının sürtünme faktörü ve Nusselt sayısının tahmin edilebilmesi için

deneysel korelasyonlar geliştirilmiştir. Bu korelasyonlar deneysel veriyi (% ±7) hata payı ile tahmin etmektedir.

Anahtar Kelimeler: Isı transfer artışı, Nusselt sayısı, Kıvrımlı şerit

LIST OF ABBREVIATIONS

CWT	:	Constant wall temperature
UHF	:	Uniform heat flux
A	:	Surface area of the tube (m^2)
$\mathbf{Cp_w}$	:	Specific heat of water at constant pressure (kJ/kg.K)
DR	:	Depth ratio
D	:	Diameter (m)
f	:	Friction factor
h	:	Heat transfer coefficient ($W/m^2.K$)
I	:	Current (Amp.)
$\mathbf{K_w}$	:	Thermal conductivity of water (W/m.K)
L	:	Length (m)
$\dot{m}$	:	Mass flow rate (kg/s)
Nu	:	Nusselt number
$\Delta\mathbf{p}$	:	Pressure ($N/m^{2)}$
$\mathbf{Pr_w}$	:	Prandtl number for water
Q	:	Total heat power (W)
$\dot{\mathbf{q}}_\mathbf{w}$	:	Heat flux ($W/m^{2)}$
$\mathbf{Re_w}$	:	Reynolds number of water
T	:	Temperature (°C)
TR	:	Twist ratio
t	:	Tape thickness (m)
V	:	Voltage (volt)
W	:	Tape width (m)
WR	:	Width ratio
x,y,z	:	Cartesian coordinate (m)
μ_w	:	Dynamic viscosity of water (kg/m.s)
ν_w	:	Kinematic viscosity of water (m^2/s)
η	:	Thermal performance factor
ρ_w	:	Density of the water (kg/m^3)

CHAPTER ONE

INTRODUCTION

1.1 Background

The thermal equipment industry is expensive part of the world economy, because of the size and of cost the thermal equipment. The study of convective heat transfer enhanced surfaces is taking attention for a wide range of industrial and transport applications; it is also applied in other technological fields like microelectronic and biotechnology. In most cases enhanced surfaces are employed as the basic element of heat exchangers [1].

Recently, the reason behind increasing the efforts to fabricate more effective heat exchange tools is the high cost of the materials and energy. The heat exchangers design need into an accurate analysis for heat transfer ratio and drop of pressure and thus, it is considered a complicated operation. The main challenge that face the heat exchanger design is to manufacture compact equipment and accomplish a high rate of convective heat transfer by employing least amount of pumping power [2].

At present, compact heat exchanger design is based primarily on utilization of known standard calculation procedures, while new successful designs can only emerge by building and testing promising solutions. Jacobi and Shah in a recent article (1998) wrote: “the conceive-build-and-test approach has reached the point of providing only incremental improvements” [1].

The method for enhancing the performance of heat transfer denoted to the intensification of heat transfer. The techniques of augmentation lead to increase the transfer of heat by lowering the thermal resistance in the heat exchanger. Also, through the usage of these methods, resulted in increasing the heat transfer efficiency and the drop of pressure. Therefore, it must be noticed that the transfer rate of heat and drop of pressure must be accomplished when we want to design a heat exchanger by the use of these methods [3].

Currently, there are a large number of the researchers who work at the field of the thermal engineering seek to improve the heat transfer methods between the surrounding fluid the and surfaces. Because of that, Bergles, Adrian [4, 5] have classified the process of improving the heat transfer as dynamic, passive and hybrid techniques.

The techniques of passive does not require any power input, except for pumping in order to accelerate the fluid and include finned surfaces usage, roughened surfaces, vortex flow generators and among some others. In contrast, the dynamic methods need additional power in order to influence in the process of optimal heat transfer [6]. In addition, a hybrid method includes the using of two or more from each of active and passive technique.

Consequently, the techniques of passive are the suitable choice and they have seen larger applications. Swirl flow generation by using twisted-tape inserts with full-length is found to be extremely effective, of the many enhancement techniques that can be used [4, 7].

Swirling flows are flows combining rectilinear motion and rotation around the flow axis. The average motion is characterized by spiral streamlines, increasing the path traveled by the fluid compared to a flow without rotation. The most common swirl generation systems are angled vanes, eccentric fluid injection, rotating pipes and inserted tape [8].

The large amount of thermal enhancement can be gotten especially in laminar flows. Moreover, there are another techniques which through it can be produced swirl flows such as convoluted or curved ducts, and the injection of tangential fluid. Their heat transfer enhancement potential and thermal hydraulic characteristics have been outlined by Nandakumar and Masliyah [9], Webb [10] and Bergles [4].

1.2 Heat Transfer Enhancement

It is known that the transport of energy is augmented if the fluid is mixed fine. This way consider the primary attitude in the improvement methods which produce vortex flows [1].

The heat exchanger industry seeks to improve the amount of heat transfer and reduce the power of pumping so as to improve the heat exchangers thermal performance. The suitable design of the heat exchanger must contain the least entropy generation or the least losses of energy in a system including a heat exchanger. If the design is efficient, the losses can be decreased but it is impossible to be stopped [11].

Conversely, for the proper rate of heat transfer, the heat exchanger size can be reduced. In fact, the surface geometry treatment is suitable for the heat exchangers design where an perfect employ of space is required [12]. The external and internal flows effectiveness of heat exchanger can be easily improved by the augmentation of heat transfer. They rise the fluid mixing by the increase of flow vorticity, or by lowering the boundary layers evolution near to the walls of heat transfer. The rate of heat transfer can be increased by somewhat different method which is the use of longitudinal vortex generator. These swirl devices tempt longitudinal stream wise vortices in the field of flow. Because of the different of pressure between the front and the back surface the flow, the vortices are developed beside the vortex generator edge. Since the vortices rotation axes are aligned to the main flow direction, the vortices are called longitudinal vortices. However, the additional pressure loss due to the use of longitudinal vortex generators is very modest, because the form drag for these slender bodies is low. Further, this increase in pumping power will be insignificant when the complete ducting of the equipment is considered. The use of augmented surfaces can satisfy one of the four possible basic design objectives ad shown in the following [13]:

1. Reduced the surface area of heat transfer for fixed pressure drop and heat duty. This objective allows a smaller heat-exchanger size and a reduced capital cost, particularly when the material reduction is accompanied by reduced manufacturing cost.

2. Increase of the heat transfer rate, represented by the product among the whole coefficient of heat transfer and the surface area of heat transfer. That increase can be exploited in two ways: it can be used to get augmented heat duty for the fluid temperature of fixed entering or to decrease the logarithmic difference of mean temperature for a fixed heat transfer rate.

3. Reduction of the inflating power for a heat duty and fixed size. By not taking full advantage of the possible enhancement in the coefficient of heat transfer, it is possible to obtain an advantage in both the first cost and the pumping cost.

4. Reduction in fouling of heat exchangers. Due to the elimination of the plain circular tube and the perpendicular plate baffle, and due to the increase in turbulence produced by the enhancement device, the extent of fouling will be appreciably less than that in a conventional shell and tube heat exchangers (STHE).

1.3 Applications of Heat Transfer Enhancement

Performance of heat exchanger can be increased by employing the heat transfer enhancement applications, which leads to minimize the heat exchanger size and operating cost. Augmentation of heat transfer has important meanings for environmental problems and conservation of energy.

The heat transfer enhancement technology (HTET) is advanced and extensively applied to applications of heat exchanger the over last decade starting from the conversion, using and thermal energy recovery in different domestic, industrial and commercial applications. The famous examples of these applications including generation of steam and power condensation, plants of cogeneration, sensible cooling and heating in chemical thermal processing, the agricultural and pharmaceutical staffs and heating of fluid in industrial and waste heat recovery etc. if the performance of the heat exchanger has been increased, this may lead to more economic design of the heat exchangers that help to save the cost, material and energy associated to heat exchange process.

The cooling and heating enhancement in an industrial process help to save in energy, lengthen the equipment working life and reduce process time. Some processes are affected by the enhanced heat transfer action. A number of serious works has been conducted in order to get an understanding of the heat transfer augmentation for their practical application. Thus the high thermal processes presence has created more request to invent new technologies for enhancing heat transfer [14].

1.4 Heat Transfer Augmentation Techniques

Heat transfer augmentative techniques most of the effort to date in the heat transfer field research has been directed toward understanding the process under normal conditions. More recently, requirements for more efficient heat transfer systems have led to increased interest and investigation of techniques which augment or intensify heat transfer [15].

Laboratory investigation of these techniques has reached the point where many of them may begin to be seriously considered for application to heat-exchange equipment on a commercial basis . Techniques which have been found to improve heat transfer generally fall into one of the following categories:

1. Surface promoters.
2. Displaced promoters.
3. Vortex flows.
4. Vibration of surface.
5. Vibration of fluid.
6. Electrostatic fields.

These techniques increase transfer of heat, but usually with penlty of added pumping power, externally applied power to the system, and/or increased cost and weight. One of the problems is to establish generally-applicable selection criteria for the use of augmentative techniques. This appears to be a near impossible task because of the large number of factors which enter into the decision problem. Many of these are economic factors , such as initial cost, maintenance cost, development cost, etc., and other considerations such as reliability and safety also enter the picture. The techniques of heat transfer augmentation can be classified into three kinds: Active, Passive and Hybrid methods.

1.4.1 Active Heat Transfer Augmentation Methods

Since these ways require input of external power in order to make the required flow modification and enhancement in the heat transfer of rate, they are considered difficult from the view point of the design. The electro-hydrodynamic enhanced boiling has been concerned by the industrial and academic researchers according to its great potential for the practical applications. Pool boiling improvements of heat transfer factors up to 900% have been conveyed for refrigerants.

While dynamic methods are active in decrease the superheat of wall and/or increase the precarious heat flux, the practical applications may be controlled, mainly due to the essential in order to develop consistent, low-cost transducers or power deliveries and related improved heat exchangers. The effective methods have not presented much potential as compared with the passive methods as they are more complicated in order to fit out the exterior power input in most case studies [4].

1.4.2 Passive Heat Transfer Augmentation Methods

Passive methods do not require external power input and the amount of desired power to augment heat transfer is reserved from the obtainable power in the system that eventually causes a pressure loss. The important principle of these methods is to enhance the transfer of convective heat which can be accomplished by the following ways [5]:

1) Minimizing the thickness of thermal boundary layer.
2) Duplicating the interruption in the flow of fluid.
3) Increasing the gradient of velocity at the surface of heat transfer.

Generally, they use surface modifications by incorporating additional devices or inserts to the flow channel. The passive techniques depend on the same standard. The employ of this method causes vortex in the fluids bulk and disturbs the real layer of the boundary and thus to rise the active surface, time of residence and accordingly the factor of heat transfer in the present system. The passive method includes a problem of cost of manufacturing as in finned tube, or cost of adding material as in twisted tape and can be classified as [4]:

1.4.2.1 Treated Surfaces

This type of technique involves metallic or nonmetallic coating on the surface. They can be employed to improve the transfer of heat in single stage conviction or boiling and condensation. Non wetting coating for example Teflon can be used in order to stimulate dropwise condensation.

1.4.2.2 Extended Surfaces

It is a mutual way to heat transfer enhancement by employing prolonged walls. A basic fin leading to rise the surface area of the heat exchangers, but a distinct shape extended surface may rise the area of heat exchanger and heat transfer factor due to the higher heat transfer coefficient for liquids. Typically, the extended surfaces that used for liquids is smaller in heights than that used for gases.

The increasing of fins height with liquids leads to decrease fin effectiveness and outcome in poor material use. There are many examples of the extended surfaces for liquids such as the internally and externally finned tube.

1.4.2.3 Displaced Insert Devices

These turbulators are inserted in the flow stream and work on changing the flow characteristics and highly enhance the transport of energy in the heated surface. They are used in single or two phase flow.

1.4.2.4 Swirl Flow Devices

These types of devices are placed in the forced flow and create rotary and secondary flow. Also, they are devices that inserted for coiled tubes, forced flow, twisted tape insert, inlet vortex generators, and axial core inserts with winding of screw type.

1.4.2.5 Rough Surfaces

Another passive technique for heat transfer enhancement in heat exchangers is an introduction of artificial roughness elements of different shapes. Two dimensional roughness of ridge-type or of groove-type in cross-flow direction, together with three dimensional uniform roughness find increasing practical use.

1.4.2.6 Surface Tension Devices

These devices involve of wicking or grooved surfaces in order to straight the liquid flow in condensing or boiling.

1.4.2.7 Additives for Liquids

These are gas bubbles and solid particle in flows with single-phase and liquid trace extracts for boiling systems which are used for heat transfer improvement. Extracts can be such suspensions for example a dilute polymer-water solution that decrease the friction of fluid. Suspensions in dilute polymer and surfactants solutions decrease the transfer of heat and the friction of fluid. Using coarse surfaces recuperates some of the heat transfer decrease. Nevertheless, the expense of this process is that the friction coefficient will be increased.

1.4.2.8 Additives for Gases

These are liquid droplets (water droplets added to air stream) or solid particles (glass, sand, zinc, graphite, aluminum oxide etc.), either dilute phase (gas-solid suspensions) or dense phase (fluidized beds) used for heat transfer enhancement. The improvement rate of gas solid suspension flowing in the tube is 3.5 times high [4].

1.4.3 Hybrid Heat Transfer Augmentation Methods

Remember that complex improvement comprises two (or more) of the methods that can be applied at the same time in order to create an improvement which is greater than the individual methods applied discretely as shown below in table (1-1). This technique has limited applications and hence involves complex design [4].

Table 1 . 1: Classification of augmentation techniques

Method	Approach	Description
Passive	Surface Interruptions	Slits or offset fins interrupt the boundary layer, restarting it , creating secondary flows, and /or generating flow unsteadiness .
	Surface Roughness	Accelerates transition from laminar flow to turbulent ; also increases turbulent flow heat transfer .
	Surface Protuberances	Ridges or 3-D shapes (cube , pyramid , etc.) generate secondary or unsteady flows .
Active	Forced Flow Unsteadiness	Surface vibration or sound waves thins or restarts boundary layer or induces secondary flows .
	Boundary Layer Injection	Enhancement primarily for multiphase flows .
	Boundary Layer Suction	Removal of boundary layer restarts boundary layer downstream.

1.5 Main Categories of Twisted Tape

Inserted tapes are the metallic strips twisted with proper methods and chosen shape which are introduced in the pipe [16]. This review will analyze the following types of inserted tapes:

A) Full length twisted tape: Its length equal to the test section length, as shown in figure (1-1)

B) Varying length twisted tape: These have length about (3/4, 1/2, 1/4) of the test section length.

C) Regularly spaced twisted tapes: These tapes have short length with diverse spaced by connecting with each other, as shown below in figure (1-2).

D) Insert with attached baffles: The twisted tape can be attached with each other by baffles at some intervals in order to realize more improvement, as shown in figure (1-3).

E) Tapes with holes and slotted tapes: Holes and slots with appropriate dimensions that can be fabricated in the inserted tape to produce high turbulence, as shown below in figure (1-4).

F) Tapes with different surface modifications: Some isolating material is supplied to tapes that can avoid the fine influence. In many cases, dimpled surfaced materials are employed for tape manufacturing, this tape consists V-cut twisted tape, Horizontal wing cut- twisted tape.

Figure 1. 1: Full length twisted tape

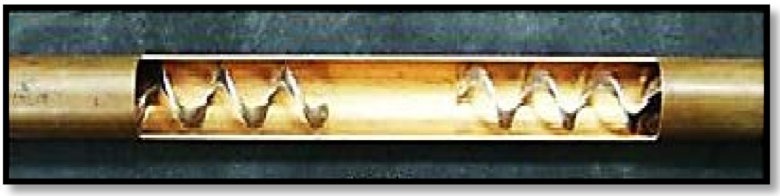


Figure 1. 2: Regularly spaced twisted tape

Figure 1. 3: Tape with attached baffles

Figure 1. 4: Holes twisted tape

1.6 Vortex Flow Generators

They consist a number of geometric activities or tube insets for forced convective flow which generate secondary flow or rotating twisted-tape insets and axial-core insets with an inlet vortex generators, screw-type winding … etc.

Transverse swirl producers make vortices where its axis is transverse to the direction of main flow while the longitudinal make vortices where its axis is parallel to the direction of main flow. In fact, the longitudinal swirl generators are more appropriate than the transverse swirl generators when the heat transfer enhancement with little friction is a significant demand [17].

To augment the rate of heat transfer, different type of inserted tapes is putted in the passages of flow and these inserts minimize the flow hydraulic diameter. Heat transfer enhancement in a round tube impasses, separating of the main and secondary flow. In addition, contraction in the flow passages rise the drop of pressure and primes to viscous influence. The secondary flow supplies a better contact of thermal between the surface and the fluid due to its creating for swirl which enhances the fluid temperature gradient that leads ultimately to rise the fluid heat transfer factor [18].

1.6.1 Internal Twisted Tape Insert Devices

Inserted tapes are made of metal twisted by using appropriate methods in order to produce the preferred dimension and shape, introduced in the flow.

The inserted tape are commonly employed in compact heat exchangers in order to enhance the rate of heat transfer with less pressure drop penalty [19, 20].

However, if the fit is too loose, the fin effect and some of the swirling action of the fluid is lost. Damage to the tubes may also be caused by being repeatedly hit by the tapes. If the fit is too tight, insertion of the tapes into the tubes may be difficult. Since they are very simple in construction, twisted tape inserts might appear to be cost-effective.

Inserted tape in a round tube provides a passive methods for improving the factor of heat transfer by presenting vortex into the bulk flow and decreasing the thermal boundary layer thickness at the wall of tube because of the continuous changes in the geometry of surface. Therefore, it can be said that such tapes convince turbulence which convinces a thinner boundary layer and thus, outcomes in a better factor of heat transfer because of the changes in the geometry of inserted tape [21]. Nevertheless, the inserts of twisted tape can be used to increase the drop of pressure inside the tube. In order to accomplish a better thermal performance and loss of friction, there are many researches have been numerically and experimentally implemented. The heat transfer improvement by the use of twisted tapes relies on the pitch and twist ratio of the strip in the tube.

The twist ratio can be defined as the ratio of pitch to the tube inside diameter $y = H / d$ where (H) represents the twist pitch length and (d) represents the tube inside diameter. Pitch is the space between two points which locates at the same plane, measured parallel to the inserted tape axis.

1.6.2 Geometrical Features of Twisted Tape

The twisted tape geometrical features, as shown in figure (1-5) are described by its 180º twist pitch H, the thickness δ , and the width w. In most uses, where fitted tapes are used, $w \cong d$, and the tape twist ratio is $y = (H/d)$. The inserted tape imposes a helical force on the bulk flow that reinforces the secondary circulation production [22].

The subsequent well-mixed helical vortex flow work on improves the convective transfer of heat significantly. In all cases, depending on what material it is made of and how strongly the tape turns at the wall of tube. Also, some tape-fin may influence too [23].

1.6.3 Terms used in twisted tape insert

- Pitch (H): It can be defined as the space that locates between two pints which are at the same plane and it is measured parallel to the inserted tape axis.

- Twist Ratio (y): It can be defined as the strip pitch rate to the intimate tube diameter $y = H/d_i$, where (H) represents the length of twist pitch and (d_i) represents the diameter of tube intimate.
- Width Ratio (WR): It is the ratio between width of the cut (w) to the width of the twisted tape (W).
- Depth Ratio (DR): It is the ratio between depth of the cut (de) to the width of the twisted tape, as shown in figure (1-6).

Figure (1-7) shows the single-phase flow modes induced by twisted tape and figure (1-8) shows the flow streams with and without twisted tape.

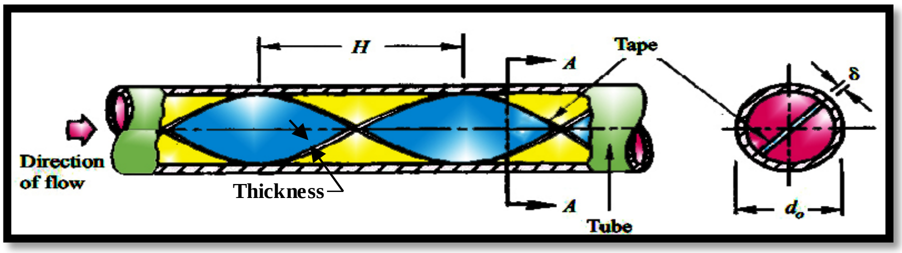


Figure 1. 5: Full-length twisted tape insert inside the tube

Figure 1. 6 : Dimensions of V-cut twisted tape

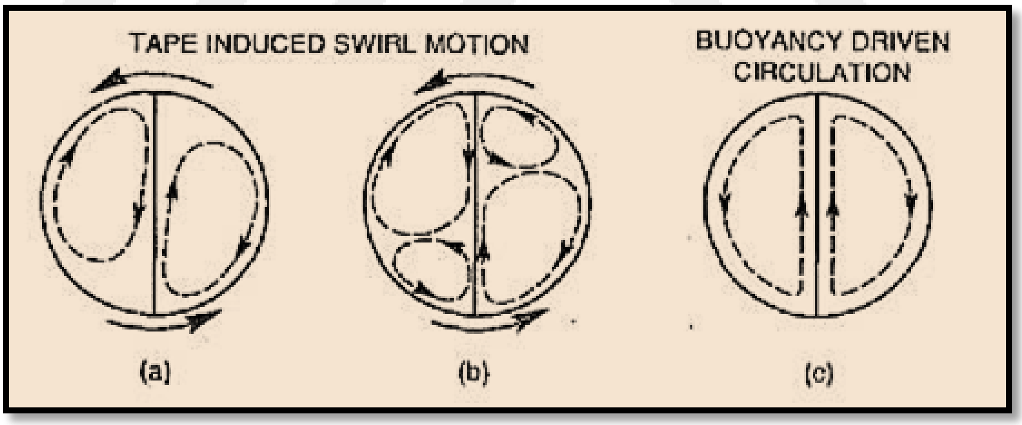


Figure 1 . 7: Single-phase flow modes induced by twisted tape [24]

(a) Low Re and/or high twisted ratio.
(b) High Re and/or low twisted ratio.
(c) Infinite twist ratio (flat tape).

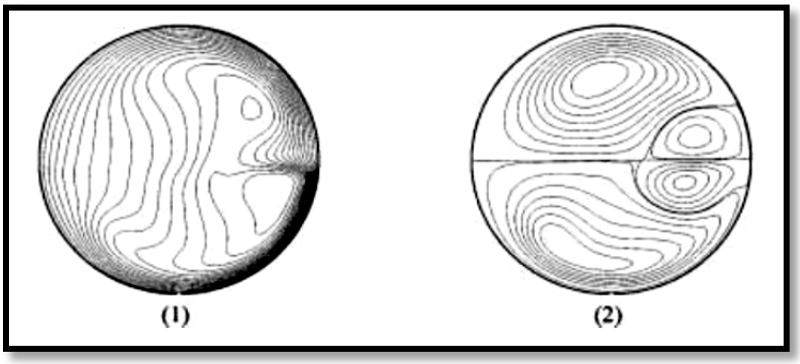


Figure 1. 8: (1) Flow in smooth tube, (2) Flow in tube with twisted tape insert [25]

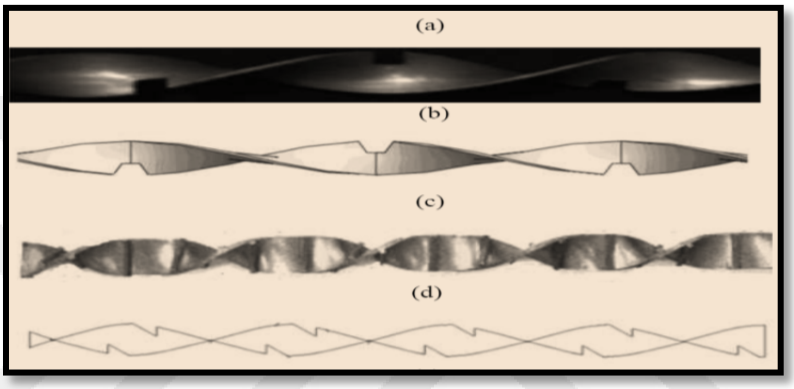


Figure 1. 9: Cuts shapes of twisted: a) Trapezoidal cut, b) Square cut, c) Serrated cut, d) Oblique-tooth cut, [26]

1.7 Literature Survey

We have included the literature survey in order to ensure that the development in the heat transfer promotion with the inclusion of inserted tape. In order to conduct a useful research review, it's necessary to mention the mechanisms and methods used to augment heat transfer by employing twisted tape at the turbulent flow.

In fact, the literature in heat transfer enhancement is increasing quicker. In any way, 50% in the literature of the heat transfer is focused now on the augmentation methods of heat transfer. There are many researches have been carried out in transfer of heat tools by using inserted tape both numerically and experimentally. Nevertheless, there is still need to verify the flow profiles and the associated transfer of heat in the complex geometries .The enhancement of heat transfer by twisted tapes is done in two regimes, as following:

1.7.1 Circular Tube with Inserts in Laminar Flow

The factor of the heat transfer is increased by the inserted tape with an increase in the drop of pressure. Many researchers have studied several and different types of inserted tape such as short length, full-length, reduced width and regularly spaced inserted tape.

The coefficients of heat transfer in laminar flow were generally low. So, for a given heat transfer rate, larger heat transfer whereas will have to be provided as compared with turbulent flow heat transfer situations. The process of using the inserted tapes in heat improvement processes can return early to the end of 19th century. Number of studies of various researchers has been carried out by using different inserts, and their findings are as follows.

(Manglik and Bergles, 1993) [27] correlated experimentally the drop of pressure and heat transfer for inserted tape of twist ratio (3, 4.5 and 6.0) for constant wall temperature circumstances employing working fluid of water and ethylene glycol ($3.5 < Pr < 6.5$), over laminar flow circumstance and described the enhancement mechanism. Through the dependence on the rate of flow and tape shape, the heat transfer improvement is because of the flow blockage and tube partitioning, the secondary fluid circulation and great flow path. They have suggested appropriate empirical correlations for the coefficient of friction and average Nusselt number comprising the vortex factor that define the interaction between convective inertia, viscous and centrifugal forces.

(Agarwal and Raja, 1996) [28] conducted experimental investigations of non-isothermal and isothermal average (Nu) and friction factor for identical surface temperature (UWT) heating and cooling of servotherm oil (Pr = 195-375) in a horizontal tube with (Re = 70 - 4000) and twist ratios (y = 2.41- 4.84). They found that factor of isothermal friction about (3.13 - 9.71) times the values of plain tube. It is observed that the Nusselt number was (2.28 -5.35) and (1.21- 3.7) times the smooth tube at constant flow rate pumping power correspondingly. They have suggested an predictedl correlation signifying the transfer of heat influence on the friction coefficient for the practical applications.

(Saha and Chakraborty, 1997) [29] found that water laminar flow (145 < Re <1480, 4.5 < Pr <5.5, tape ratio 1.92 < y <5.0) and features of pressure drop in a round tube fixed with frequently spaced, there is severe decrease in drop of pressure consistent decrease in heat transfer. Therefore, it is found that at fixed power of pumping, a large turn number may enhanced the thermal performance in comparison to a single turn on inserted tape.

(Al-Fahed et al., 1999) [30] implemented tests in order to compare the coefficients of heat transfer and factor of friction for a plain, micro fin, and inserted-tape -tubes. Throughout this study, it has been used three ratios of twist with two various widths.
The data of heat transfer which have gotten in a single shell-and-tube heat exchanger where steam is employed in the shell side to get a constant wall temperature and oil is the working fluid in the tube. The results demonstrate that at the lower twist ratio (y = 5.4) and high loss of pressure, a loose suitable twisted tape is a better choice for the heat exchanger because of it is ease to be installed and removed for the cleaning issues. For other twist rates tight fit provides improved thermal performance that the loose- fit inserted tapes.

(Klaczak, 2000) [31] investigate experimentally the thermal and hydraulic characteristics under laminar flow condition employing working fluid of water in a vertical copper pipe cooled by air with twisted tape inserts of various pitch value. The tests were implemented for Reynolds number range ($110 \leq Re \leq 1500$, $8.1 \leq Gz \leq 82.0$ and $1.62 \leq y \leq 5.29$). Result shows that the transfer of heat increases with increase in twisted tape pitch value.

(Liao and Xin, 2000) [32] have conveyed experimental results on the complex enhancement of heat transfer techniques and settled that the heat transfer improvement in a tube with 3-D internal prolonged surfaces by substituting incessant inserted tape with nearly segmented inserted tape insets results in a reduction in the coefficient of friction with a small Stanton number reduction. The Stanton number can be defined as the rate of heat transfer to the difference of enthalpy and it is a measure to the coefficient of heat transfer.

(Patil, 2000) [33] reported the pressure drop and thermal features of pseudo plastic type power law fluid in a horizontal round tube fitted with full length inserted tapes with variable width under persistent wall temperature circumstances. Patil discovered that the deliberations of improved transfer of heat and reserves in the power of pumping and the cost of tape material, decreased width twisted tape is better for improving vortex flow heat transfer. As well as, he perceived that 17-60 % decrease in friction coefficient and 5-24 % decrease in (Nu) for 15-50 % reduction in tape-width.

(Saha and Dutta, 2001) [34] reported experimental data on Nusselt number and pressure drop by employing inserted tape produced laminar swirl flow for a large range of Prandtl number ($205 < Pr < 518$) and perceived that, the twisted tape of short length is a good selection due to that in this case, on a constant pumping power basis, vortex generated by the inserted tape declines gradually downstream which rises the coefficient of heat transfer with least drop of pressure, as associated with the full length twisted tape.

(Lishan You, 2002) [35] implemented a numerical work in order to expect the rapidity and temperature supplies in laminar completely advanced flows by using tubes comprising twisted tape insets. The swirl flow was replicated by following the helically twisted flow path in the partitioned tube signified by a semi-circular cross-section geometry by the use of finite volume method. For the case study, we have considered the uniform heat flux (UHF) boundary conditions and the constant wall temperature (UWT). Numerical results based on the velocity and temperature fields variations with tape twist ratio and Reynolds number were offered. As well as, the distributions of temperature reflected the fluid Prandtl number effect. The inserted tape persuaded vortex flow field is considered by a single longitudinal vortex that pauses up into two counter-rotating helical vortices with (Re) increasing or declining the ratio of twist. Similarly, the substantiality is increased by both the coefficient of heat transfer and drop of pressure.

(Sarma et al., 2003) [36] assumed new technique which find the coefficients of heat transfer and factor of friction using inserted tape insets in a tube where the gradients of temperature and the wall shear are correctly modified throughout the factor correlation of friction which lead to the heat transfer enhancement from the surface of tube. The eddy diffusivity appearance of van Driest is adjusted to rejoin to the internal flows case in a tube with inserted tapes for a wide ranges of (Re) number conforming to the flow of laminar in tubes. The outcomes are equaled with some existing correlations in the previous work for the inserts of twisted tape.

(Suresh Kumar et al., 2003) [37] a practical work was implemented to identify the features of pressure drop in a great diameter annular test division for the hydro-dynamically advanced laminar flow under uniform heat flux circumstances. The results obtained are compared under two conditions: with inserted tape and without. The variations of coefficient of friction with (Re) for different twist ratios along the circumference of the test section were investigated. The thermal- hydraulic performance is better for twisted tape as compared to wire coil for the similar helix angle and twist rate.

(Bharatdwaj et al., 2009) [38] practically identified the heat transfer and the features of friction coefficient of working fluid (water) flow in a 75-start spirally grooved tube using inserted tape. It has been considered a laminar to wholly turbulent ranges of (Re). In terms of the flow direction, the grooves are clockwise. The heat transfer augmentation because of the spiral grooves is auxiliary enhanced by implanting inserted with three twist ratios (y = 10.15, 7.95 and 3.4), as compared to that of plain circular tube.

(Lin and Wang, 2009) [39] numerically investigated the transfer of heat in a round tube tailored with inserted tape. The conduction influence on the (Nu), the heat transfer improvement sensitivity to the circumstances of thermal boundary by the use of secondary flow, and its influence on the hydraulic boundary layer have been argued. The study data exposed that under fully developed flow, diverse tube surface thermal boundaries lead to diverse influences of conduction in the tape on features of heat transfer.

(Sheeba et al., 2009) [40] experimentally investigated the thermal characteristics of thermos syphon solar system using helical insert with different ratios of twist under condition of laminar flow. The study data stated that the improvement in coefficient of heat transfer in the collector of inserted tape is larger than the plain tube collector with the least twist rate and regularly declines with the rise of twist rate.

(Yadav, 2009) [41] investigated experimentally the drop of pressure features and transfer of heat in a U-bend double pipe heat exchanger using half-length twisted tape. Their results were compared with that of smooth tube. The data exposed that increasing the rate of heat transfer of the inserted tape is found to be highly effected by vortex motion. The coefficient of heat transfer is rising by 40% with half-length inserted tape insets in comparison with smooth tube and the performance of heat transfer of using half-length inserted tape is superior to smooth tube through the rate of equal masse.

(Eiamsa-ard et al., 2010) [42] studied the effect of peripherally-cut inserted tape on transfer of heat, loss of friction and thermal performance coefficient features in a circular tube. Nine diverse peripherally-cut twisted tapes with persistent twist rate (3.0) and diverse three tape depth rates (0.11, 0.22 and 0.33), everyone with three diverse tape width rates (0.11, 0.22 and 0.33), have been tested. From the result, it is revealed that number of Nusselt, loss of friction and thermal performance coefficient are revealed to be amplified with rate width and rate of depth.

Kapatkar et al., 2010) [43] reported a practical study on transfer of heat and drop of pressure of a plain tube together with packed length inserted tape under continuous heat flux circumstance. The aim of the experiment is to study the influence of the tape fine by the use of full length tape inserts of multiple materials including insulated tape, Stainless steel and Aluminum. The twist rate of the tape from 5.2 to 3.4. It is revealed that, in terms of the flow in smooth tubes, the full length of the twisted tape produce enhancement in average Nusselt number, for (Re) ranging from 200 to 2000, while the tapes of aluminum the greatest enhancement in Nusselt number range from 50 % to 100 %, for Stainless steel tapes, the greatest enhancement in Nusselt number range from 40 % to 94 % and for insulated tapes, the greatest augmentation in (Nu) range from 40 % to 67%.

(Jian Guo et al., 2011) [44] found a center cleared inserted tape for accomplishing good thermal- hydraulic performance. A comparative investigation between center cleared inserted tape and the short-width inserted tape was conducted numerically under laminar flow condition. The predicted data presented that for a round tubes fitted with center-cleared inserted tapes, the transfer of heat can be even improved in the circumstances with an appropriate central approval rates. The coefficient of thermal performance for the tube fitted with center-cleared inserted tape can be improved by 7-20 % in comparison with tube fitted with traditional inserted tape.

(Suhas and Vijay, 2011) [45] investigate experimentally thermal-hydraulic features in a concentric double pipe heat exchanger (square duct inner and circular tube outer) by the use of full length insertions with diverse twist rates (y = 2.66 and y = 3.55).

The data were taken for flow under laminar region (Re = 30-1100). The investigations have been implemented for identical wall temperature boundary circumstance by the use of working fluid (ethylene glycol). The results that been obtained showed that whenever the twist rates increase, the heat transfer improvements will be increased. Friction coefficients were found about (6-13) times the smooth duct values. While the average (Nu) for the inserted tape is greater than those for the smooth duct about 6.0 and 5.30 times for y = 2.66 and y = 3.55 correspondingly. The investigational data indicated that (Nu) are found about 5.44-7.49 and 2.46-4.88 times the smooth square duct depending on continuous flow ratio and continuous pumping power standards correspondingly, for y = 2.66.

(Wongcharee and Eiamsa-ard, 2011) [46] studied the thermal- hydraulic features of the around tubes prepared fitted with alternative clockwise and counterclockwise inserted-tapes (TA) for a wide range of (Re) extending from 830 - 1990 with three different twist rates (3, 4 and 5) which have been implanted separately under condition of constant heat flux, where the working fluid is water. The results which have been gotten from this study showed that friction coefficient and thermal performance coefficient related to TA were greater than those related to characteristic twisted tape. Between the examined tapes, the one that has the minimum twist rate of (3) was found to be the greatest effective for transfer of heat improvement.

(Zhang et al., 2012) [47] performed numerical simulation to study the thermal and fluid flow of multi-longitudinal vortices in a circular tube fitted with triple and inserted twisted tape inserts for (Re=300 to 1800). If we compare with the smooth tube, the tubes with triple and quadruple inserted tapes possessed higher heat transfer rates until 171% and 182% correspondingly, these accompanied with the increases of friction factors of around (4-7) times, respectively. The factors of thermal performance of the tube inserts varied between 1.64 and 2.46.

(Sami et al., 2013) [48] presented numerical study for reproduction of the swirling flow in a round tube with elliptic-cut and typical inserted tape. Influence of the twist ratio (2.93, 3.91, and 4.89) and cut depth (0.4, 0.8, and 1.4 cm) on the convective heat transfer improvement and factor of friction are investigated.
The simulation was carried out by the use of commercial CFD package (FLUENT-6) in the laminar flow system for the (Re) extending from 200 to 2100.
The result of this study revealed that the rate of transfer and friction coefficient in the classical twist tube is less than those induced by elliptic-cut twisted tape.

(Suvanjan et al., 2013) [49] reported the experimental friction coefficient and heat transfer data throughout an around duct having essential transverse ribs and tailored with center-cleared inserted-tape. As well as, they presented the empirical friction coefficient and average (Nu). The main results of this practical investigation is that the center-cleared inserted tapes in mixture with transverse ribs implement better than the individual improvement method performing alone throughout a circular duct until a definite expanse of center-clearance.

1.7.2 Circular Tube with Inserts in Turbulent Flow

Heat transfer improvement inserts under turbulent flow is active until definite Reynolds number. Increasing the blocks of Reynolds number will work on increasing the drop of pressure. The resistance of the dominant thermal through the turbulent flow is restricted to a tinny viscous sublayer close to the wall. The following steps clarifies the performance of tape twisted inserts under turbulent flow.

(Yamada et al., 1984) [50] studied experimentally the performance of transfer of heat a cross flow shell-and-tube compact heat exchanger for great temperature use where the heat transfer is augmented by employing surface radiation with tube sides and shell. The twisted cross-tapes are inserted in the tube side while plates are inserted into the shell side. The entire factors of heat transfer which have been measured about a greatest 80 percent that is greater than the heat transfer that registered without radiation. Whereas, the hot gas internal temperature close from 800°C and finally, the temperature of the clod gas is near to the temperature of room. It must be mentioned that the analytical results and the experimental results are agreed with each other and the procedure of the calculation is found to be adequately accurate and simple for the practical applications.

(Algifri and Bharadwaj, 1985) [51] presented an analytical work of the convective heat transfer features in rotting flow produced by diminutive inserted tapes which placed at the entry of the test sector. The analytical expressions were obtained from a series solution of the swirl equation extracted from the equations of Navier-Stokes with the help of magnitude analysis order. It is discovered that the approach is in well arrangement with the available experimental data, with a higher deviation of 15%. An improvement in the convective heat transfer higher than 80 % is obtained.

(Gupte and Date, 1989) [52] semi-empirically assessed the heat transfer and friction coefficient for inserted tape produced vortex flow in an annulus. The results which have been gotten for radius rates of 0.41 and 0.61 and twist rates of ∞, 5.302, 5.038, and 2.659. The rise in the heat transfer rates and drop of pressure are compare to the reported data for twisted tape produced vortex flow in tubes.as well as, at the same rates of heat transfer, the requirements of the pumping power are compare constructively with those for empty annuli. The analytical expectations that depend on superposition standard of pressure drop and analogy among momentum transfer and heat transfer have produced exceptional expectations for $y = \infty$ and 5.302 but somewhat poor agreement at $y = 2.659$.

(Saha et al., 1990) [53] reported the experimental study results for transfer of heat and the features of pressure loss through a round tube having commonly spaced elements of twisted tape that associated by tinny rods of oracular.

Reynolds number, space rate Prandtl number, rate of twist and rod to tube diameter ratio are controlled the features. As well as, they reported (Nu) and correlations of factor of friction. It is revealed that depending on both the constant heat duty and constant power of pumping, commonly the full-length inserted tapes perform better than the spaced inserted tape due to that the vortex breaks down in-between the spacing of a frequently inserted tape.

(Hijikata et al., 1994) [54] practically investigated the flow and transfer of heat features for a round pipe with a pair of twisted tapes to enhance the transfer of heat by convection and radiation. In terms of the three dimensional velocity field dimension, it is examined that the secondary flow is tempted by the existed promoter, the performance of heat transfer is improved around three times as compare with that of plain pipe, and an obvious increase of the rate of coefficient of local heat transfer was showed near the secondary flow impinging region. The turbulator performance was evaluated based on the total surface area of heat transfer at equal value of mass flow rate, pressure loss, and heat load. Moreover, they discovered that the ratio of heat transfer can be improved about 50 percent by the radiation between the twisted tape and the pipe surface.

(Rao and Sastri, 1995) [55] carried out experimental investigation for the factor friction for rotating tube fitted with inserted tape for heat transfer enhancement. The data which have been gotten are compared with the presented data of non-rotary tube fitted with inserted tape. It is discovered that the improvement in the heat transfer offset through the factor of friction because of the rotation with respect to the plain tube over stationary circumstances. They proposed an empirical correlation for gotten results.

(Naumov and Semashko, 1999) [56] employed a mathematical model for an adiabatic cross-section in order to investigate the pipes pulsed asymmetric heating through an exterior quasi-stationary heat flux, $q_m \approx 10$ MW/m^2. The heat is removed by using a coolant without phase transitions from the parts of wetted pipe cross-section perimeter in the wall boiling regime. The mathematical formulas are compared with that for a collector cooper pipe fitted with twisted tape and heated by a pulse of heat flux from the directed ion beam, characteristic for the prescribed mode of operation under plasma heating by injection in T-15 tokamak.

(Sivashanmugam and Sunduram, 1999) [57] studied experimentally the performance improvement in double pipe heat exchanger fitted with inserted tape having twist rates of 15.649, 8.54, 5.882, 4.95, and 4.149 over continuous heat flux by the use of water as working fluid. The greatest percentage increase of 44.7% in energy transfer ratio was gotten for the inserted tape of twist rate 4.149. For the whole twist rates, the gain declines with the (Re) and becomes continuous for (Re) larger than 3,000. The lesser the twist rate is, the greater the increase in energy for a particular (Re).

(Kumar and Prasad, 2000) [58] studied practically the thermal performance of heater fitted with inserted tape having ratio of twist pitch to diameter of tube ranging from 3-12 in a solar water system at different mass flow rates. They found that the transfer of heat in the inserted tape was increased by 18-70 %, while the drop of pressure raised by 87-132 %, in comparison with plain collectors. The effect of geometry of inserted tape, (Re) and solar radiation intensity on the thermal performance of the solar system has been reported. It has been observed that the solar collectors fitted with inserted tape introduced better enhancement in the lower range of flow (Re =12,000), beyond which the increase in the factor of thermal performance is monotonous.

(Hong et al., 2007) [59] studied the drop of pressure and complex heat transfer features of a converging-diverging tube with consistently spaced twisted-tapes (CD-T tube). Vortex has been produced by consistently spaced inserted tape components that differ in twist rate and angle of rotation. Also, the rate of space has a significant influence on the features. For contrast, they have carried out experiments in a round plain tube and a (CD-T) tube without inserted tapes. The exposed results clarified that the twisted tape with twist rate $y = 4.72$ and rotation angle $\theta = 180^0$ has the optimal performance between the four kinds of inserted tapes existed in this study.

(Chang et al., 2007) [60] studied experimentally the local factor of heat transfer and the factor of friction for a circular tube fixed with a broken twisted tape having twist rate 1, 1.5, 2 and 2.5 is performed in the (Re) range of 1000 – 40,000. This type of inserted tape is not available in previous investigations. The result show that the local (Nu) and mean Fanning coefficient of friction rises as the twist ratio declines. The coefficients of heat transfer, mean fanning friction coefficients and index of thermal performance in the tube are augmented, respectively to 1.28-2.4, 2-4.7 and 0.99-1.8 times more than of tube fitted with typical twisted tape.

(Chang et al., 2008) [61] examined experimentally the transfer of heat in a swinging tube fitted with a serrated strip inset over sea circumstances. This swirl tube swings near two orthogonal axes over single and composite rolling and pitching fluctuations. Synergistic influence of composite rolling and pitching oscillations with either harmonic or non-harmonic rhythms enhanced the thermal performance of heat transfer. Influence of buoyancy in the swinging tube increase the local Nu, and then decrease with swinging force relative strength increases. An empirical correlations is derived that allows the interactive and individual effects of single and compound swinging force influences with interactions of buoyancy on the (Nu) to be quantified.

(Smith Eiamsa-ard et al., 2009) [62] offered an experimental investigation on the average thermal performance features, drop of pressure and Nusselt number in a corpulent tube fitted with short-length inserted tape over continuous heat flux circumstances. The working fluid is air and the measured data are taken at various (Re) values. The full-length strip is inserted at a unique twist ratio of y/w = 4.0 whereas the short-length strips fixed at the sector of the entry test at multiple tape rates of 0.29, 0.43, 0.57 and 1.0 (full-length strip). The result shows that the short-length strips of LR = 0.29, 0.43 and 0.57 offer minor transfer of heat and pressure values as compared with the full-length strip around 14 %, 9.5 % and 6.7 %; and 21 %, 15.3 % and 10.5 %, correspondingly.

(Thianpong et al., 2009) [63] investigated experimentally the friction and composite thermal performances in a dimpled tube fixed with inserted tape by employing working fluid of air. The twist rates and pitch influence on the average factor of heat transfer and the loss of pressure are identified in a round tube for the (Re) ranging from 12,000 to 44,000. The experiments are accomplished by the use of two dimpled tubes with different pitch rates of dimpled surfaces (PR=0.7 and 1.0), in addition to three types of inserted tapes with three various twist rates (y/w =3, 5, and 7). Furthermore, they carried out the experiment by the use of smooth tube and dimpled tube acting unique for the purpose of comparison. The results that been gotten showed that both the factor of friction and the coefficient of heat transfer in the dimpled tube fixed with the tape of twisted are greater than those in the dimple tube acting only and smooth tube. As well as, it is revealed that the coefficient of heat transfer and factor of friction in the composite device rise where the pitch ratio (PR) and twist ratio (y/w) decline. In addition, an empirical correlation depending on the experimental data of the existed work is adequately precise for forecast the (Nu) and the behavior of friction coefficient (f).

(Behabadi et al., 2010) [64] studied experimentally the factors of heat transfer and drop of pressure through condensation of HFC-134a in a circular tube at the occurrence of different inserted tapes. The assessment field is a 1.04 m long double-tube counter-flow heat exchanger. The flows of refrigerant in the internal copper and the flows of cooling water in annulus. The assessments are accomplished for a plain tube and with four inserted tapes having twist ratio of 6, 9, 12 and 15. The study revealed that the inserted tape with twist rate offers the greatest improvement in the coefficient of heat transfer and the greatest drop of pressure in comparison to the smooth tube on a nominal area basis. At this case, the transfer of heat improvement and the drop of pressure are raised by 40 and 240% in contrast with to the plain tube. It is clarified that the inserted tape with the twist rate of 9 has the best performance improving the transfer of heat with the least drop of pressure. As well as, the empirical correlations have been improved in order to forecast the smooth tube and swirl flow pressure drop. Forecast data are compared to experimental results and it is revealed that the correlations are dependable to estimate the drop of pressure.

(Eiamsa-ard et al., 2010) [65] made experimentally a relative work of enhancement of heat transfer and loss of pressure by employing of three types of inserted tapes, full-length dual and regularly spaced dual twisted tapes as vortex creators, in a horizontal circular tube under constant heat flux conditions and Reynolds number ranging from 4000 to 19,000. The experiments are implemented by the use of full-length dual twisted tapes and single inserts with various twist rates (y/w =3.0, 4.0 and 5.0). As well as, they used regularly-spaced double inserted tapes with three various space rates (S/D = 0.75, 1.5 and 2.25). The influence of main factors on the drop of pressure and heat transfer have been conferred and the consequences are compared with results that been gotten from the smooth tube.

(Kalyani et al., 2010) [66] implemented experimental study for flow augmentation of heat transfer in a circular tube having variable width inserted tape using air as the employed fluid. To decrease the extreme drop of pressure related with the complete width inserted tape with the least consistent decline in the coefficient of heat transfer, they introduced different width twisted tapes with the range from 10 mm to 22 mm. The experiments of the study have implemented for smooth tube with/without inserted tape at continuous wall heat flux and various ratios of mass flow. The inserted tapes use are of different various twist rates (3, 4 and 5) and everyone with five various widths (26-full width, 22, 18, 14 and 10 mm) correspondingly. The (Re) ranging from 6000 to 13500. The coefficient of heat transfer and the drop of pressure have been computed and the optioned data are compared with the data of the plain tube. It is revealed that the thermal improvement using inserted tape as compare with the smooth tube differ from 36 to 48 % for full width (26mm) and 33 to 39 % for reduced width (22 mm) inserts.

(Seemawute and Eiamsa-ard, 2010) [67] reported experimentally the influence of peripherally-cut inserted tape with alternative axis (PT-A) on the heat transfer and fluid flow improvement characteristic in an identical heat flux circular tube. Experiments were implemented by using working fluid of water, and the Reynolds number ranging from 5000 to 20,000. Evidently, the rates of convective heat transfer in the tube fitted with the PT-A, Peripherally-cut twisted tape (PT) and typical twisted tape (TT) are correspondingly enhanced until 184 %, 102 % and 57 % of that in the smooth tube. They also observed that the PT-A, PT and TT presents the greatest thermal performances at continuous pumping power of 1.25, 1.11 and 1.02, correspondingly.

(Hata and masuzaki, 2011) [68] investigated the heat transfer and drop of pressure by using inserted tape. The effect of y and (Re) depending on swirl speed, Re_{sw}, on the inserted tape-induced vortex flow heat transfer was examined in particulars and the extensively and exactly expectable correlation was imitative depending on the experimental results. The empirical correlation are developed for a various twist rates ranges (y = 2.39 – 4.45), mass speeds (G = 4022–15140 kg/m^2 s) and Reynolds numbers depending on swirl velocity ($Re_{sw} = 2.88\times10^4$ to 1.22×10^5) within −10 to +30% difference.

(Murugesan et al., 2011) [69] investigated the influence of V-cut inserted tape on convective transfer of heat, factor of friction and factor of heat transfer performance features in a round tube having three twist rates (2.0, 4.4 and 6.0) and three different mixtures of depth and width rates (DR = 0.34 and WR = 0.43, DR = 0.34 and WR = 0.34, DR = 0.43 and WR = 0.34). The data which have been gotten at this study revealed that the average (Nu) and the average factor of friction in the tube fitted with V-cut inserted tape (VTT) rise with lessening twist rates, width rates (WR) and rising depth rates (DR).

(Yangjun, 2011) [70] presented the formation enhancing of commonly spaced short-length inserted tape in a round tube using working fluid of air by the use of computational fluid dynamics (CFD) technique. The formation factors comprise the free space rate, twist rate and angle of rotation. The results indicated that the greater rotated angle produces a greater value of heat transfer and a larger resistance of flow. While the smaller twist rate results in better thermal performance using twist ratio differ from 2.5 to 8.0 excluding for a large angle of rotation and a large (Re).

(Halit and Veysel, 2012) [71] investigated experimentally the flow friction and the thermal performance of vortex generator inserted tube. The twisted tapes have been introduced individually from the wall of tube. The twist rates influences (2, 2.5, 3, 3.5 and 4) and clearance rates (0.0178 and 0.0357) have been conferred in the (Re) range from 5132 to 24,989. Constant heat flux is smeared to the wall of tube. The working fluid that is used at this study is the air. The use of twisted tapes provides significant rise on transfer of heat and drop of pressure if they are compared with smooth tube. When the clearance rate increases, the (Nu) increases. As well as, The (Nu) increases when the twist rate is increased. The maximum heat transfer improvement is accomplished as 1.756 for clearance rates = 0.0178 and twist rate = 2 at Reynolds number of 5183. Therefore, the result of the study offered that the best operating system of the whole twisted inserts is noticed at low Reynolds number which lead to more compressed heat exchanger.

(Burse et al., 2014) [72] implemented experimental investigation to understand influence of inserted tape of various twist ratios on thermal characteristics in a round tube. The experiment emphases on convective heat transfer experimental investigation and skin friction factor features of a round pipe having inserts and the working fluid is the air. Nusselt number and factor of friction got experimentally were confirmed against those got from theoretic correlations. The inserted tapes used having three various twist rates for instance 1.78, 2.32 and 2.77. Initially experiment was carried out for plain tube at continuous wall heat flux and diverse mass flow ratios of air. Secondly experiment was implemented with insertions in a tube for the same working conditions as that of plain tube. The experimental results gotten are compared with smooth tube data.

(Gulia and Parinam, 2014) [73] carried out geometric investigation in a round tube and found thermal performance for the tube fitted with inserted tape implanted for y/w = 4. The inserted tapes with length rate LR=0.29 , 0.43 , 0.57 and 1, heat transfer rate rises 15 %, 18.8 %, 22.6 % and 31% more than smooth tube (fluid-Air). The greatest heat transfer performance for using LR= 0.29, 0.43, 0.57 and 1 is found to be 1.23, 1.3, 1.32 and 1.37.

(Kulkarni, 2015) [74] investigated experimentally the heat improvement in a round horizontal tube having clockwise and counter clockwise-corrugated inserted tape. The working fluid that is employed is the air (Re = 4000 - 10000) with different rate of mass flow and constant heat flux. The c-cc corrugated twisted tape are of same pitch and twist ratio but three various angle of rotation in clockwise and counter clockwise route as 30°, 60°, 90° respectively. The conclusion made by him as describe here. The heat transfer improvement with clockwise and anticlockwise-crenelated twisted tape insets if compared with the smooth tube ranging from 12 to 46% for 90 º angle of rotation and 10 to 39% for the rotation 60ºangle.

(Kannan et al., 2016) [75] studied experimentally the thermal characteristics and factor of friction for a horizontal tube fitted with perforated inserted tape for Reynolds number from 21,000 to 50,000. The twisted tape specifications, (mild steel Twisted tape, width =45 mm, TR = 2, 3, 4) has been used for the test. The result shows that (Nu) varies inversely to the twist ratio. The thermal factor is more than 1.44 while using the pierced twisted tape with lower twist ratio and wire coil.

(Kumar, 2016) [76] did the numerical analysis of heat transfer in horizontal tubes having five types of different inserts inserted into the horizontal rod having fluid of different values of Reynolds numbers ranging from (6000-14000). The overall conclusions which he found during the study that increase in pumping work associated with a rise in heat transfer factor in comparison with that of smooth tube.

(Orhan and Veysel, 2016) [77] implemented experimental work on thermal improvement of a tube having coiled-wire installed with a gap from the tube surface. The wire insert had an equilateral triangular sections with a continuous side length of e =6 mm and they were coiled with various pitch-to-diameter ratios: P/D=1, P/D=2, P/D=3. The inserts were installed with 1 and 2 mm farewell from the internal tube surface. Thus, the transfer of heat improvement because of the laminar boundary layer trouble could be studied. All cases were conducted for (Re = 3429-26,663). The result revealed that the greatest heat transfer performance was perceived around 1.82 for the P/D=1, s=1 type at a Reynolds number of 3429. In conclusion, the laminar boundary layer disturbance can be efficiently improved by the use of these kinds of coiled-wire inserts.

1.8 Scope of Present Work

Permitting with the literature, it is clear that the employ of inserted tape efficiently enhanced the rate of heat transfer in comparison to the individual use of distilled water. The attractive characteristics of a twisted tape, as mentioned, have motivated the present research to study the influence of their types and twist ratios on the enhancement rate. The present work presents experimental study of the effects of four twisted tape types [typical twisted tape, clockwise-counterclockwise twisted tape, perforated circular clockwise-counterclockwise twisted tape (circular holes with two diameters 7mm and 4mm) and perforated square clockwise-counterclockwise twisted tape (square holes with two dimensions 7mm and 4mm)] on heat transfer enhancement and flow of fluid. In the experimental part a test rig is to be built to study the thermal enhancement and drop of pressure for a various range of Reynolds number ($3684 \leq Re \leq 10814$) with uniform heat flux (4459-18599 W/m^2), using modern measurement equipment and devices that can be calibrated accurately. Finally, empirical correlations have been developed to calculate the average Nusselt number and coefficient of friction depending on the experimental results.

CHAPTER TWO

THEORETICAL ANALYSIS

2.1 Introduction

This chapter is separated into three parts. First is discusses the physical model of system and the basic design of plain tube heat exchanger. The second is experimental equations for the heat transfer in horizontal tube of the working fluid (distilled water). The third part is a standard correlation to verify the result obtained.

2.2 Physical Model of System

The system geometry in the present work consists of a horizontal circular tube. The tube is insulated well to prevent the thermal losses to the external environment. The material of the tube is copper. The wall of the tube is heated under constant heat flux. Working fluid (DI-water) is passing through the tube with different flow rates. Four type of inserted tape is used in the experimental tests to study the improvement in the convective transfer of heat and pressure losses for the heat exchanger.

2.2.1 Physical Model of Plain Tube

Considered the case of pure water flowing inside a round tube with length (L) and inside diameter of 1 m and 23 mm correspondingly. To ensure the flow is hydrodynamic fully developed, entrance length should be added to the test section length, for turbulent flow the entrance length is according to the equation [78]:

$$\frac{L_e}{D} = 4.4 \times (\mathrm{Re})^{\frac{1}{6}} \tag{2-1}$$

Table 2. 1: Entrance length at different Reynolds numbers.

Re	3684	4140	5781	7360	9210	10814
L_e (m)	0.397	0.405	0.428	0.466	0.463	0.475

And from table (2-1), it is noticed that the maximum length does not exceed (0.475m), so (2.3 m) has been selected as entrance length for all Reynolds number ranges. The test section surface is subjected to a constant heat flux in the range (4459-18599 W/m^2), as shown in figure (2-1).

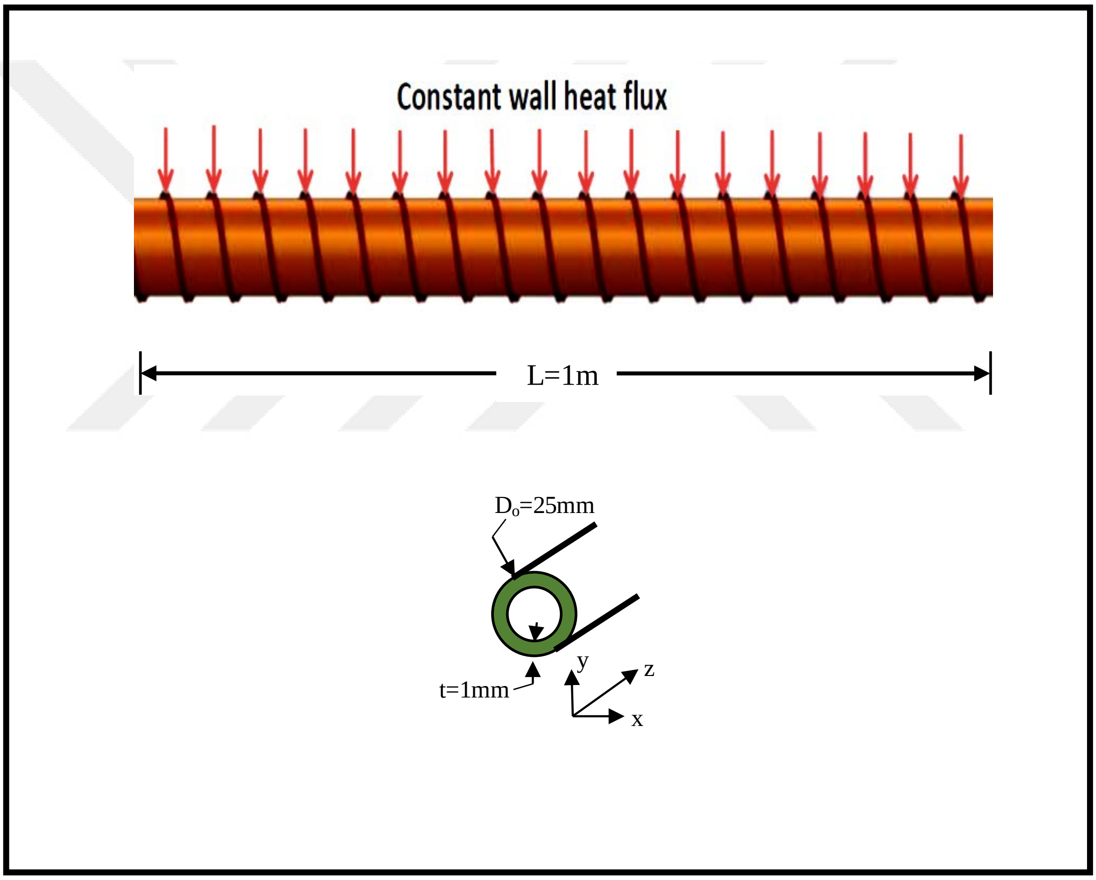


Figure 2 . 1: Physical geometry of plain tube

2.2.2 Physical Model of Twisted Tape

A schematic diagram of inserted tape through a horizontal tube is shown in figure (2-2). The inserted tape is geometrically defined by the tape thickness (δ) and its twist ratio.

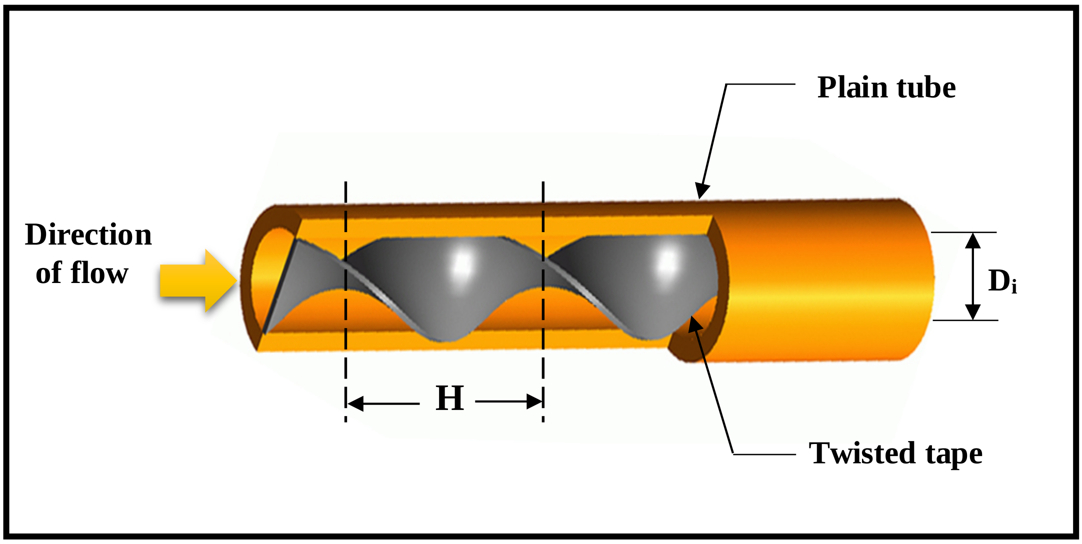


Figure 2 . 2: Physical Model of Twisted Tape

The twist ratio (y) is defined as the axial length (pitch) (H) for a 180° turn of the tape divided by the width (w) of the tape as shown below:

$$TR\ (y) = \frac{H}{W} \qquad (2\text{-}2)$$

The twisted tape is assumed in full contact with the internal surface of the tube, so the tape width is taken the same as the diameter of the tube (W=23 mm), and the tape length and thickness are (L=1m, δ =1mm). The type of twisted tape used is (typical twisted tape, clockwise-counterclockwise twisted tape, perforated circular clockwise-counterclockwise twisted tape and perforated square clockwise-counterclockwise twisted tape) with twist ratios (TR =4, 6) are used in each type of twisted tape.

The clockwise-counter clockwise twisted tape changes its direction of rotation every two pitch distance (i.e., the tape rotate 360° in the clockwise direction along the distance "2H", and then it rotates in the counter clockwise direction 360o for the next distance "2H", this sequence will continue until the test section length reached.

2.2.3 Basic Design of Plain Tube Heat Exchanger

The following assumptions are used in the model:

1. Steady state and turbulent flow condition.
2. The fluid properties remain the same and buoyancy effect is assumed to be negligible.
3. Neglect changing in the kinetic and potential energies.
4. Constant specific heat with little loss in accuracy.
5. The axial conduction along the tube is usually insignificant.
6. The heat exchanger outer wall is assumed to be insulated perfectly and no heat generation within the heat exchanger.
7. No change in flow phase inside heat exchanger.

2.3 Experimental Calculations

During the experiment calculation, the performance of the present heat exchanger regarding with thermal aspects is studied for horizontal circular tube with constant thermal flux on the outer wall. This will be achieved by changing the flow of water side, afterward adoption of twisted tapes to observe the effect of adding it on the heat exchanger thermal performance. In this work, temperatures of water has been measured. Also surface temperatures for tube are recorded in various positions.

2.3.1 Heat Transfer Coefficient inside Tube

Electrical power applied at the external surface of the tube, can be achieved by the following equation:

$$Q_{heater} = I \cdot V \tag{2-3}$$

Where:

I : The electric current is delivered through heater coils (A).

V : The electric voltage is delivered through heater coils (V).

The amount of the heat transferred from the heating wire (heater coils) to the distilled water is given by:

$$Q_{fluid} = \dot{m}_w \, Cp_w \, (T_o - T_i) \tag{2-4}$$

Where:

$\dot{m}_w$: Mass flow rate of the water (kg/s).

Cp_w : Specific heat of the water (kJ/kg.°C).

T_i , T_o: Water and outlet temperature at the tube section respectively (°C), as shown in figure (2-3).

The balance of heat between the water (Q_{fluid}) and heat input (Q_{heater}) was found to be within 3.0% for all runs. That is:

$$\left|\frac{Q_{heater} - Q_{fluid}}{Q_{heater}}\right| \prec 3\% \qquad (2\text{-}5)$$

It is important to know the water heat transfer coefficient before any calculations. The local coefficient of heat transfer is expressed by:

$$h(z) = \frac{\dot{q}_w}{(\Delta T)_z} \qquad (2\text{-}6)$$

Where: $(\Delta T)_z$ is the temperature difference between the inner wall of the tube T_{si} (z) and the water temperature T(z) at distance Z from the entrance of the tube.

To calculate T_{si} (z), the conduction equation in the cylinder is used:

$$\dot{q}_w = \frac{Q_{fluid}}{\pi D_i L} = \frac{2k\left[T_{so}(z) - T_{si}(z)\right]}{D_i \times \ln\left(\frac{r_o}{r_i}\right)} \qquad (2\text{-}7)$$

Where:

$\dot{q}_w$: Rate of heat transfer per unit area (W/m^2).

D_i : The internal tube diameter (m).

L : The tube test length (m).

Also, from the balance of energy in the tube test, the mean temperature of water can be expressed by [79]:

$$T_o(z + \Delta z) = T_{in}(z) + \frac{\dot{q}_w \ \pi D}{\dot{m}_w \ Cp_w} z \qquad (2\text{-}8)$$

Where: $T_{in}(z)$, $\dot{m}_w$ and Cp_w are the water temperature at the inlet of test section, the mass flow rate and the heat capacity of the water respectively.

Thus, the coefficient of local heat transfer becomes:

$$h(z) = \frac{\dot{q}_w}{T_{si}(z) - T(z)} \tag{2-9}$$

Where:

$\dot{q}_w$: Rate of heat transfer per unit area (W/m^2).

T(z) = [Tin(z) +To (z+Δz)] / 2 : is the water bulk temperature.

The local Nusselt Number has been found from the following expression:

$$Nu(z) = \frac{h(z)\,D}{k_w} \tag{2-10}$$

The average value of Nusselt number in the thermal region can be calculated from the integral:

$$\overline{Nu} = \frac{1}{L}\int_0^L Nu(z)\,dz \tag{2-11}$$

Simpson rule 1/3 is used to calculate the above integration.

The Prandtl and Reynolds number are:

$$Pr_w = \frac{\mu_w\, Cp_w}{k_w} \tag{2-12}$$

Where:

μ_w : Water dynamic viscosity (kg/m.s).

k_w : Water thermal conductivity (W/m.K).

$$Re_w = \frac{4\dot{m}}{\pi d_i \mu} \tag{2-13}$$

Where:

$\dot{m}_w$: Water mass flow rate (kg/s).

μ_w : Water dynamic viscosity (kg/m.s).

d_i: Internal tube diameter (m).

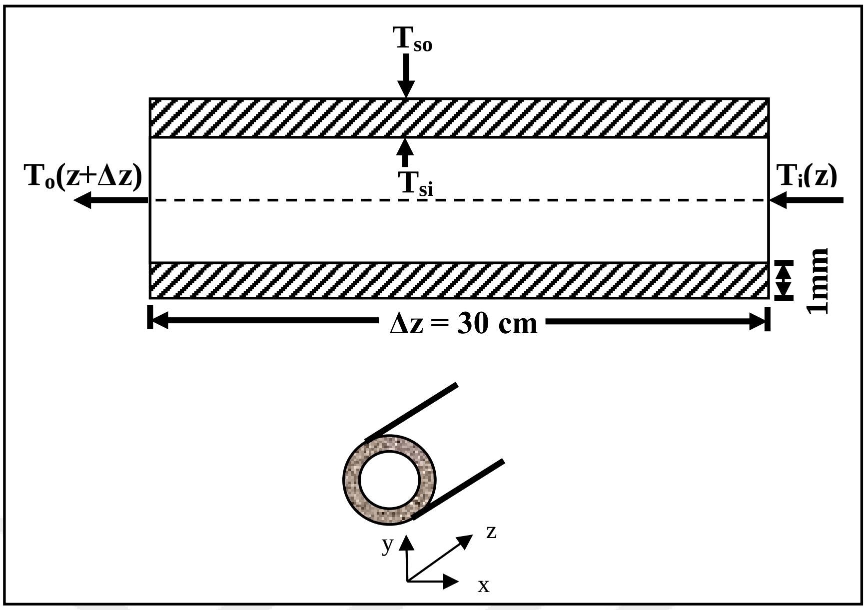


Figure 2 . 3: Heat Transfer Coefficient inside Tube

2.3.2 Friction Factor of the water

Based on the practically calculated drop of pressure, Darcy factor of friction can be calculated using the expression [78]:

$$f = \frac{2\,\Delta P\, d_i}{L\,\rho\, V^2} \tag{2-14}$$

$$V = \frac{\dot{m}}{\rho\, A_c} \tag{2-15}$$

$$A_c = \frac{\pi}{4} d_i^2 \tag{2-16}$$

Where:

Δp: is the drop of pressure across the test tube measured by the manometers.

V: Mean velocity (m/s).

ρ : Density of water (kg/m^3).

$$\Delta P = \rho_w\, g\, \ h \tag{2-17}$$

2.3.3 Thermal Performance Factor (η) Evaluation

Index of heat transfer improvement or factor of thermal performance is one of the key parameters necessary to define the transfer of heat augmentation performance. The equation predicted by Eiamsa-ard et al. [80, 81] for constant pumping power is given by:

$$\eta = \frac{Nu_{\text{twisted tape}} / Nu_{\text{plain tube}}}{\left(f_{\text{wisted tape}} / f_{\text{plain tube}}\right)^{1/3}} \tag{2-18}$$

2.4 Standard Correlation Used

To verify the test facility and the data obtained, transfer of heat and drop of pressure for a smooth tube of (1m) long as a test section were investigated using empirical correlation of Dittus–Boelter [79] and Blasius equation [78] respectively.

$$Nu = 0.023\,\text{Re}^{0.8}\ \text{Pr}^{0.4} \tag{2-19}$$

$$f = 0.316\ \text{Re}^{-0.25} \tag{2-20}$$

All properties in equations (2-19) and (2-20) are evaluated at the bulk temperature.

Table 2 . 2: Physical properties of water (SI units).

Temperature T (°C)	Specific Weight γ (kN/m³)	Density[a] ρ (kg/m³)	Dynamic Viscosity[b] μ ($\times 10^{-3}$ kg/m·s)	Kinematic Viscosity ν ($\times 10^{-6}$ m²/s)	Surface Tension[c] σ (N/m)	Modulus of Elasticity[a] E ($\times 10^{9}$ N/m²)	Vapor Pressure P_v (kN/m²)
0	9.805	999.8	1.781	1.785	0.0765	1.98	0.61
5	9.807	1000.0	1.518	1.519	0.0749	2.05	0.87
10	9.804	999.7	1.307	1.306	0.0742	2.10	1.23
15	9.798	999.1	1.139	1.139	0.0735	2.15	1.70
20	9.789	998.2	1.002	1.003	0.0728	2.17	2.34
25	9.777	997.0	0.890	0.893	0.0720	2.22	3.17
30	9.764	995.7	0.798	0.800	0.0712	2.25	4.24
40	9.730	992.2	0.653	0.658	0.0696	2.28	7.38
50	9.689	988.0	0.547	0.553	0.0679	2.29	12.33
60	9.642	983.2	0.466	0.474	0.0662	2.28	19.92
70	9.589	977.8	0.404	0.413	0.0644	2.25	31.16
80	9.530	971.8	0.354	0.364	0.0626	2.20	47.34
90	9.466	965.3	0.315	0.326	0.0608	2.14	70.10
100	9.399	958.4	0.282	0.294	0.0589	2.07	101.33

CHAPTER THREE

EXPERIMENTAL APPARATUS AND PROCEDURE

3.1 Introduction

This chapter describes the building and designing the experimental apparatus. It provides a detailed description of the equipment used in the laboratory experiments and experimental study of thermal enhancement and flow of water in a circular tube having twisted tape insert at different Reynolds number. It can be divided into two categories:

The first concerns with experimental test rig appliances and arrangement by establishing circular tube heat exchanger test rig with models including smooth and plain tube with internal twisted tape. The second category is concerned with the measurement procedure for two types of (plain tube without inserts, plain tube with inserts). In this work the circular tube performance is investigated by changing the water mass flow rates, twist ratios and types of inserts. Constant heat flux, steady state, constant inlet temperature conditions are considered in all experiments.

3.2 Test Rig and Experimental Apparatus

The test rig consisted of the following main parts and measuring apparatus, figure (3-1) and plate (3-1) show the schematic diagram and photo of the experimental apparatus:

1. Tube section. **2.** Spiral heat exchanger.

3. Water tank. **4.** Pump.

5. Globe valves. **6.** Variac transformer.

7. U-tube manometer. **8.** Thermocouples.

9. Rotameter. **10.** Temperature recorder.

11. Voltmeter and Ammeter used to measure the power of the heater.

Many of these parts are designed and fabricated for this investigation and intensive care was taken to forbid leakage of fluid between the connected sections and re-fixing. Multi-layer insulator is used to insulate the hot cycle to prevent the heat dissipation. These parts have been fixed on the frame structure which is designed to make both easy operation and maintenance through the experiments.

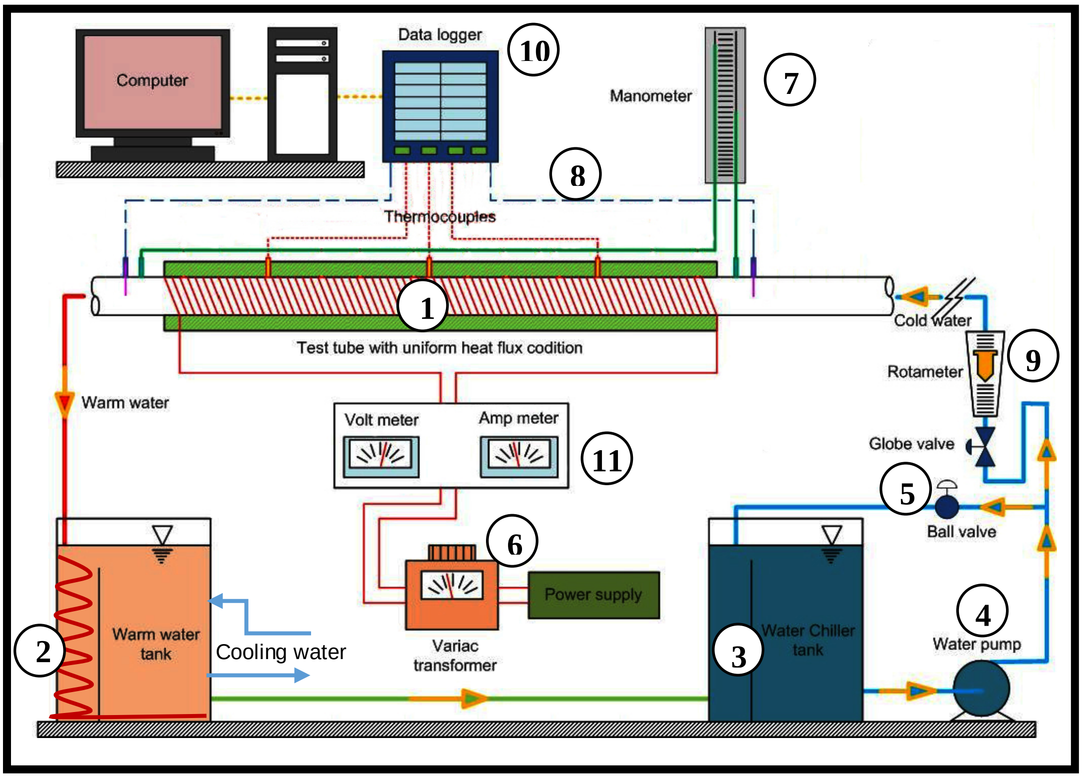
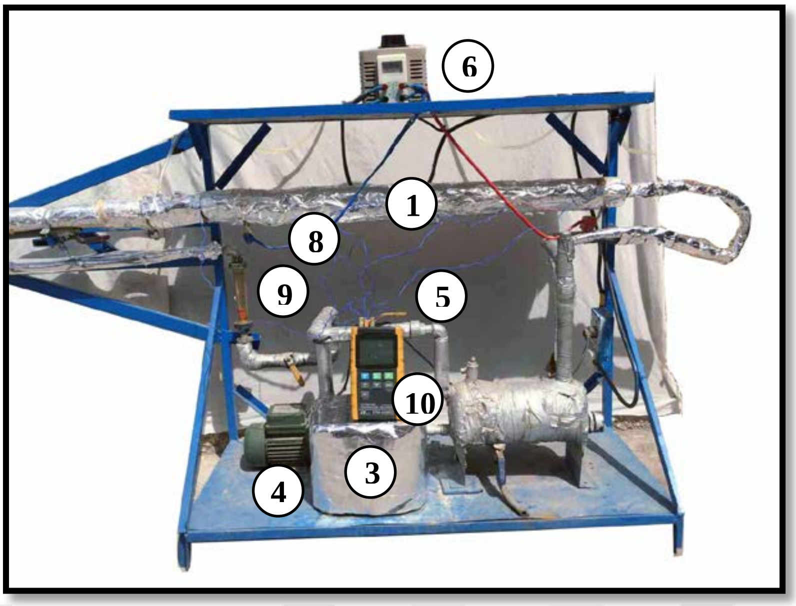
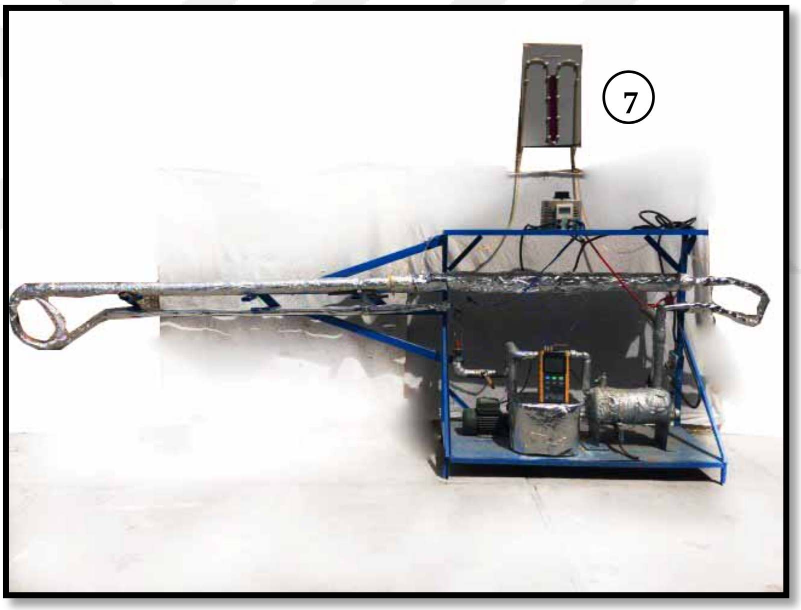


Figure 3. 1: Schematic diagram of experimental test rig



(a)



(b)

Plate (3-1): Photograph view of experimental setup.

3.2.1 The Test Section

It is a copper tube of (2.3 cm) diameter, the tube has a hydrodynamic fully developed length of (2.3 m) and a (1 m) test section length. The tube outer wall is heated electrically by a coil made from tungsten material connected to an AC power supply to produce heat flux, it is (3 m) long and (3.5 mm) width with (1210 W) heating power, winded over an electric insulator on the tube to generate the uniform heat flux, as shown in plate (3-2). Aluminum foil is used between the heater and the insulation. Sectional pipe insulation of rock wool type with (2.5 cm) internal diameter and (4.5 cm) outer diameter thermal insulation operating in temperature up to (230ºC) nominal density is (64 kg/m^3) used to insulate the test section as shown in figure (3-2).

(a)

(b)

Plate (3-2): Photograph view of for the test section with insulation.

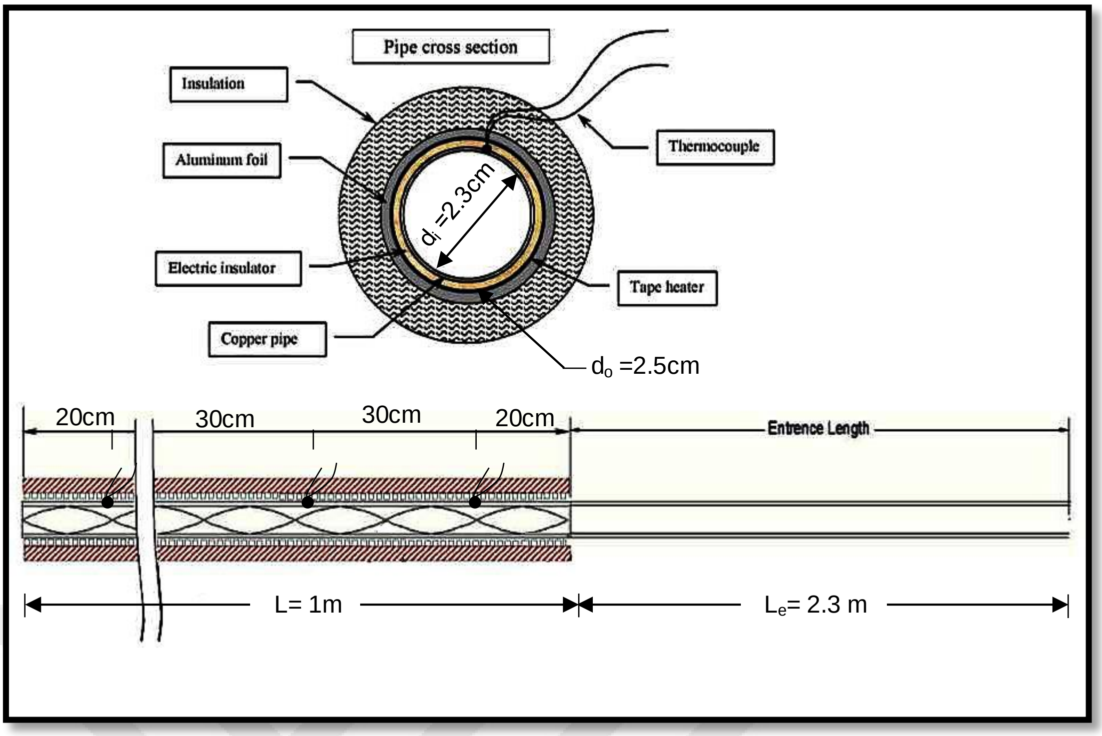
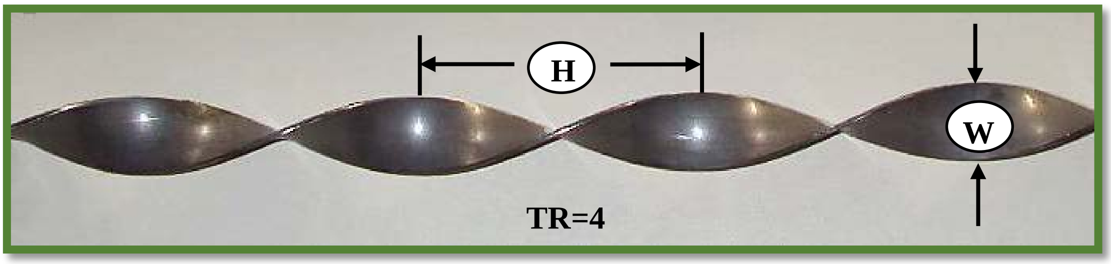


Figure 3. 2: Schematic diagram for test section with dimensions.

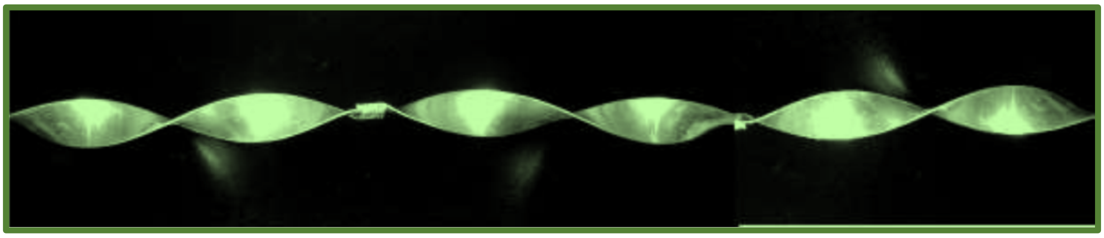
3.2.2 Twisted Tape

The inserted tapes were manufactured from a metal strip of finite length twisted with twist ratios (TR=4, 6) and inserted along the test section. The twisted tape was fabricated from a copper strip of length (1 m), width (23 mm) and thickness (1 mm), both ends of every tape were clamped by metallic clamps, so that the two ends could be fixed. The proper twist pitch (H) was obtained by using a lathe machine. One end was kept fixed on the tool post of the lathe, while the other end was given a slow rotational motion by rotating the chuck side. The tape thickness of (1mm) is chosen to prevent the difficulty of thin tape twisting which turned easily during the twisting operation, and to avoid any friction in the system that might be caused by the thicker tape.

During the operation time, a mild pressure on the side of tool post was applied to avoid the twisted tape distortion. Four configurations of the twisted tape were fabricated, as shown below in plate (3-3).



(a)



(b)

(c)

(d)

Plate (3-3): Twisted tape configuration.

a) Typical twisted tape.

b) Clockwise-counterclockwise twisted tape.

c) Perforated circular clockwise- counter clockwise twisted tape.

d) Perforated square clockwise- counter clockwise twisted tape.

3.2.3 The Spiral Heat Exchanger

Spiral heat exchanger is fabricated from a copper tube coil, (1500 cm) long and (2.3 cm) diameter and employed to cool the hot fluid to make sure that overheating does not occur. The coil is placed in a stainless steel tank of (40 cm) diameter and (100 cm) height filled with re-circulating water.

3.2.4 The Working Fluid Tank

Working fluid (water) tank is a stainless steel tank which fabricated with dimensions (28 cm) diameter and (40 cm) height, used to collect the accumulated cold fluid coming from the heat exchanger and feeds the pump with the required amount of working fluid.

3.2.5 Centrifugal Water Pump

The pump is another essential apparatus to the smooth running of the flow loop. The type of the pump [Marquis - 0.37 kW], China industry. The pump works with a constant voltage (220 V) and (2.5A) and [Q_{max} = 30 L/m, H_{max} = 30 m]. The pump is able to operate with a fluid temperature up to 70 ^{o}C, see table (3-1).

3.2.6 Heater Circuit

The electrical circuit of heating element consists of Variac (2000W) to adjust the heater input power as required, while a digital Wattmeter (type Dm – 9020) was used to measure the power consumed by the heater, and a digital clamp meter and voltmeter was used to measure the heater voltage and current, see plate (3-4) and table (3-1).

(a)

(b)

(c)

Plate (3-4): Wire heater with bead element setup.

3.2.7 Network Pipes

Polyvinyl chloride (PVC) pipes with (12 mm) diameter are employed to connect all the test rig parts because of both its flexibility and ease of refixing after repair.

3.2.8 Set of Valves

Three ball valves are employed in the cycle of water in order to control the flow rates of water (one for main pumping line , one for bypass and one for maintenance and calibration case).

3.3 Measurement Devises

3.3.1 Thermocouples Distribution and Fixation

Five thermocouples (k–types, TP – 01) were used to measure the temperatures of water inside the tube and the surface temperature of the test tube. Three thermocouples are fixed along the tube with a space distance of (200 mm) for the first one and (300 mm) between other thermocouples. The other two thermocouples are fixed at the exit and entrance of the test tube to find the inlet and outlet temperatures of water by the same procedure. All thermocouples were linked to digital reader data logger to record the thermocouples readings. The thermocouples brought specified from the manufacture to have an uncertainty of 0.65 % from (-50 ~ 400^{o}C). The thermocouple use chromel alloy made of (nickel, chromium) and alumel alloy made of (nickel, aluminum).

3.3.2 Digital Data Logger

Twelve channel portable temperature recorder model (BTM-4208SD) has been used to measure the temperatures and for data saving along the time into SD memory card. Various thermocouples can be used with this device as (J/K/T/E/R/S). There are two options of auto or manual data logger and wide range of sampling time (1 to 3600) seconds. Plate (3-5) shows the temperature recorder from Lutron Electronic Company.

Plate (3-5): Digital data logger.

3.3.3 Flow Meter

In this study, one available (LZS -15) flow meter as shown in plate (3-6) was employed to measure the rate of flow circulation through water side. The range of flow meter are (100 - 1000 liter \ hr). Most of these devices depend on gravity principle, so they are linked in a vertical direction. The flow meter was connected and positioned just after the circulating pump with accuracy of ± 5 % of full scale flow meter and temperature limits (0-60 ºC), see table (3-1).

Plate (3-6): Water flow meter.

3.3.4 U-tube Manometer

U-tube manometer is a tool used for measuring the drop of pressure across the test tube. The U-tube manometer is connected to the pressure tap at the inlet and outlet of the test tube. The liquid used in manometer is water, the density of it is (1000 kg/m^3).

Table 3. 1: Technical specifications of the instruments.

<table>
<tr><th>Sl.</th><th>Name of</th><th colspan="5">Specifications</th></tr>
<tr><td rowspan="2">1</td><td rowspan="2">Water pump</td><td colspan="2">Phase</td><td>RPM</td><td>HP</td><td>Volts</td></tr>
<tr><td colspan="2">Single</td><td>2850</td><td>0.5</td><td>220</td></tr>
<tr><td rowspan="2">2</td><td rowspan="2">Hot Water Rotameter</td><td>Body</td><td>Float</td><td>flow</td><td>Mfg. Name</td><td>Model No.</td></tr>
<tr><td>Crystal</td><td>SS , 316</td><td>100-1000</td><td>Flowtech</td><td>LZS-15</td></tr>
<tr><td rowspan="2">3</td><td rowspan="2">Ammeter</td><td colspan="2">Phase</td><td colspan="2">Make</td><td>Current rating</td></tr>
<tr><td colspan="2">Single</td><td colspan="2">TAIFA</td><td>1-5000/5 Amps</td></tr>
<tr><td rowspan="2">4</td><td rowspan="2">Voltmeter</td><td colspan="2">Phase</td><td colspan="2">Make</td><td>Voltage rating</td></tr>
<tr><td colspan="2">Single</td><td colspan="2">TAIFA</td><td>0-600 Volts</td></tr>
<tr><td rowspan="2">5</td><td rowspan="2">Digital Data logger</td><td>Range (˚C)</td><td>Number of channel</td><td>Make</td><td colspan="2">Model No.</td></tr>
<tr><td>-10 ~ 1000</td><td>12</td><td>Lutron Electronic</td><td colspan="2">BTM–4208SD</td></tr>
<tr><td rowspan="2">8</td><td rowspan="2">Variac</td><td colspan="2">Power</td><td>Make</td><td colspan="2">Model No.</td></tr>
<tr><td colspan="2">2000W</td><td>China</td><td colspan="2">NATP–6P</td></tr>
</table>

3.4 Calibrations

3.4.1 Calibration of the Thermocouple

All thermocouples were calibrated against the melting point of ice and boiling point of distilled water and mid-point between them, the readings were taken with thermometer and the thermocouples through the digital reader data logger. The maximum error was $\pm 1^oC$. A sample of calibration results is shown in figure (3-3).

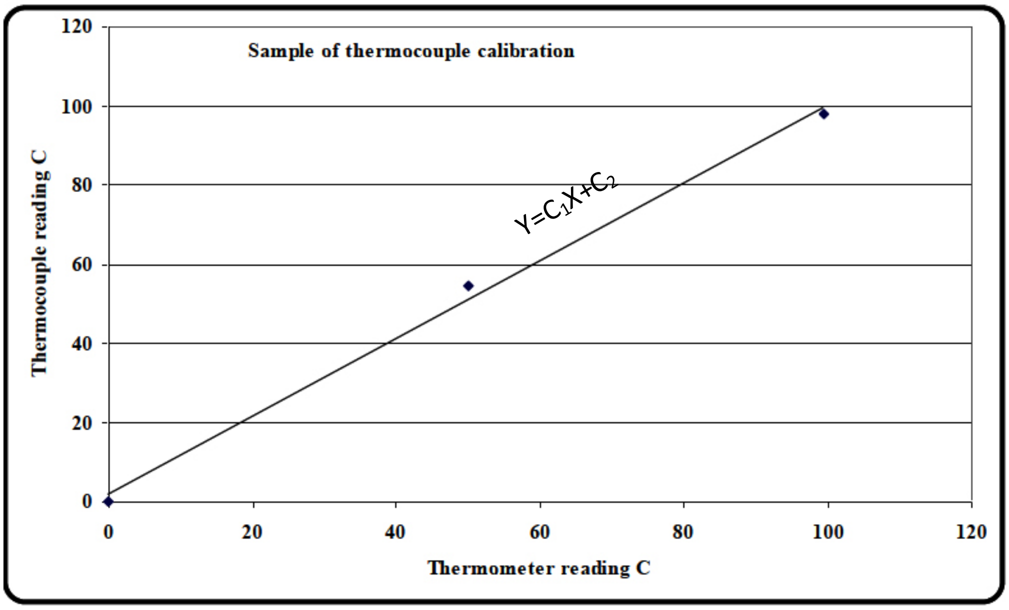


Figure 3. 3: Sketch of thermocouples' calibration.

3.4.2 Calibration of the Flow Meter

The flow meter was calibrated by employing a scaled tank by volume and stop watch to calculate the time required to fill a specified volume of the tank. The results showed that the flow meter has (±0.25 l/min) error, as shown below in figure (3-4).

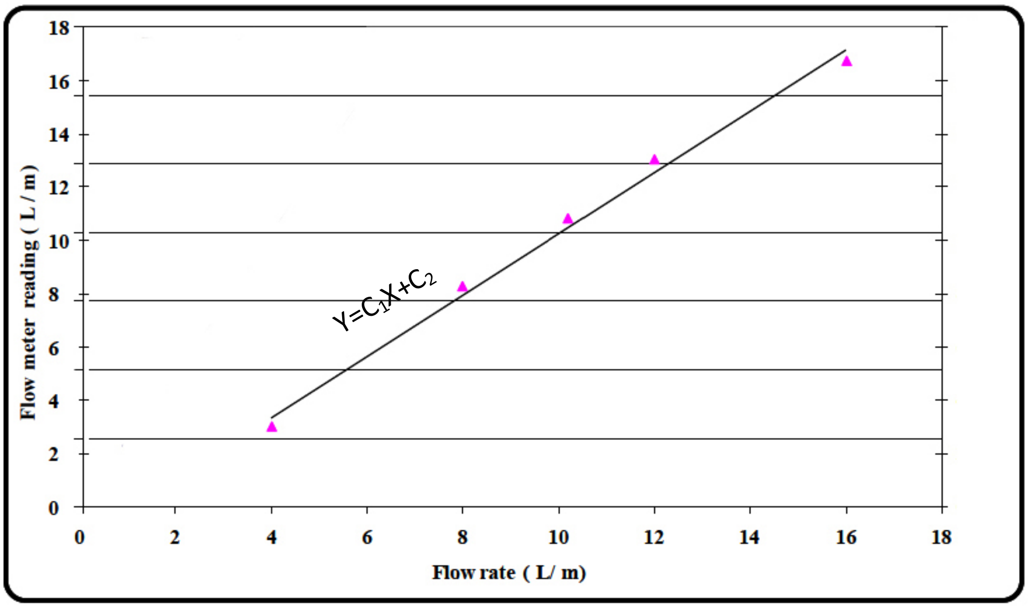


Figure 3. 4: Flow meter calibration

3.5 Distilled Water Tests

The initial test was done with distilled water, so as to validate the test rig performance. After completion of construction and calibration of the flow loop, initial tests were run for the smooth tube and a tube with various types of the inserts used for the whole range of Reynolds number.

3.6 Experimental Procedure

The following procedure has been applied for the experiment after checking water side as shown in figure (3-5). Then switching on the electric devices on test rig with water rotating in a cycle until achieving steady state. All the tests were implemented under fully developed turbulent flow with Reynolds number range (3684-10814) and uniform heat flux range (4459-18599 W/m^2). The number of the total tests were 192 and the steps of experiments procedures are:

1- The required type of twisted tape was inserted.

2- The control valve that is located before the pump is opened to enter the distilled water to the loop for the first flow rates.

3- The pump is operated to circulate the distilled water in the test rig.

4- The flow rate is adjusted by using control valve and the balance valve to get the desired flow rate.

5- The electrical heater is operated and adjusted to get the desired heat flux by the regulating device.

6- The thermocouple readings is observed at the exit and entrance of the test tube until a steady state is obtained which is reached after one hour for the first run and 20 min for the subsequent runs.

7- After the steady state is reached, the following readings are recorded:

a) The distilled water temperature at inlet and outlet of the testing tube.

b) The testing pipe surface temperature at the four points.

c) The drop of pressure through the test tube by means of manometer.

d) Power consumed by the heater by means of voltmeter and ampermeter.

e) The flow rate.

8- The procedure from steps (1-6) is repeated for the other flow rates under flow condition.

9- Replace the twisted tape with the next one and repeat the steps from (2).

10- Starting from step (1), the same steps were repeated for another twist ratios.

11- After finishing the readings for each of the experiments, the test rig will be cleaned using the draining valve.

12- The rig is left to be cool down before the next experiment.

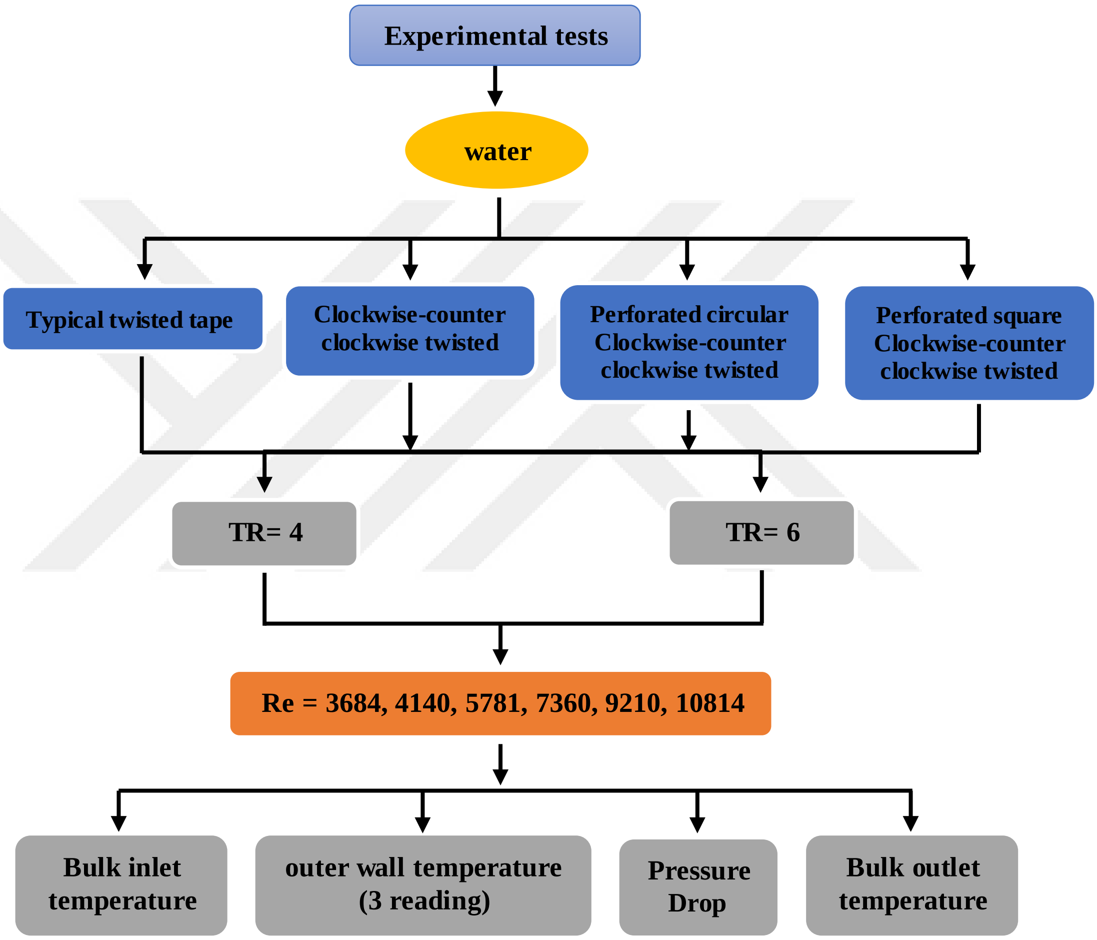


Figure 3 . 5: Block diagram for experimental procedure

3.7 Uncertainties Analysis

To calculate the error in the obtained results, Kline and McClintock method [82] was used:

Let the result (R) be a function of n independent variables: Y_1, Y_2…Yn

$$R = R\,(Y_1,\ Y_2 \ldots\ldots Y_n) \tag{3-1}$$

For small variations in the variables, this relation can be expressed in linear form as:

$$\delta R = R_{Y_1}\ \delta Y_1 + R_{Y_2}\ \delta Y_2 + \ldots\ldots\ldots + R_{Y_n}\ \delta Yn \tag{3-2}$$

Hence, the uncertainty interval (e) in the result can be given as:

$$\left(\frac{e_R}{R}\right)^2 = \left[\left(R_{Y_1}\frac{e_{Y_1}}{Y_1}\right)^2 + \left(R_{Y_2}\frac{e_{Y_2}}{Y_2}\right)^2 + \ldots\ldots + \left(R_{Y_n}\frac{e_{Y_n}}{Y_n}\right)^2\right] \tag{3-3}$$

Where: $R_{Y_n} = \dfrac{\partial R}{\partial Y_n}$ (3-4)

The errors of the used measuring apparatus are given in table (3-2).

Table 3. 2: Measuring apparatus errors.

Measuring Apparatus	Error
Power of the heater	± 0.5 Watt
Thermocouple wire	± 1 ^{o}C
Flow meter	± 0.25 l/min
Digital reader of the thermocouple	± 0.01 ^{o}C
Mercury thermometer	± 0.15 ^{o}C

The local Nusselt number equation can be expressed as follows:

$$Nu = \frac{h\,D}{k} \tag{3-5}$$

$$Nu = \frac{Q\,D}{A_s\,\Delta T_s\,k} \tag{3-6}$$

$$Nu = \frac{\dot{m}\,Cp\,\Delta T_b\,D}{\pi\,D\,L\,\Delta T_s\,k} \tag{3-7}$$

The experimental error in the local Nusselt number calculation can be expressed in the following manner:

$$\frac{\partial Nu}{\partial \dot{m}} = \frac{Cp\,\Delta T_b}{\pi\,L\,\Delta T_s\,k} \tag{3-8}$$

$$\frac{\partial Nu}{\partial \Delta T_b} = \frac{\dot{m}\,Cp}{\pi\,L\,\Delta T_s\,k} \tag{3-9}$$

$$\frac{\partial Nu}{\partial T_s} = \frac{\dot{m}\,Cp\,\Delta T_b}{\pi\,Lk\,\Delta T_s^2} \tag{3-10}$$

then,

$$e_{Nu} = \left[(Nu_{\dot{m}}\;e_{\dot{m}})^2 + (Nu_{\Delta T_b}\;e_{\Delta T_b})^2 + (Nu_{\Delta T_s}\;e_{\Delta T_s})^2\right]^{1/2} \tag{3-11}$$

or

$$\left(\frac{e_{Nu}}{Nu}\right)^2 = \left[\left(\frac{\partial Nu}{\partial \dot{m}}\,\frac{e_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{\partial Nu}{\partial \Delta T_b}\,\frac{e_{\Delta T_b}}{\Delta T_b}\right)^2 + \left(\frac{\partial Nu}{\partial T_s}\,\frac{e_{\Delta T_s}}{\Delta T_s}\right)^2\right] \tag{3-12}$$

The above formula gives the error in Nusselt number as shown in below in table (3-3):

Table 3. 3: Errors in the average Nusselt number.

Volume flow rate (LPH)	**Error percentage (%) for $\overline{Nu}$**
150	2.5
450	3.3
600	3.2

CHAPTER FOUR

RESULTS AND DISCUSSION

4.1 Introduction

In this chapter, experimental and theoretical data are reported and discussed, which include the tests of water in a horizontal smooth tube (tube without twisted tape) and a tube with inserted tape using two types of twisted taped tapes with twist ratios (TR = 4, 6). All the above tests were implemented under constant heat flux and turbulent flow condition. Transfer of heat and drop of pressure for smooth tube were investigated, and experimental data were collected in order to calculate the level of enhancement of heat transfer and to validate the present test rig data with well-established and available data in the literature.

4.2 Accuracy and Reliability of Results

The present experimental results are compared with the well-known correlation for the fully developed turbulent flow under constant heat flux condition in tubes to ensure the accuracy of the experimental work results.

Figures (4-1) to (4-4) show the Nusselt number verses Reynolds number for fully developed turbulent flow inside a plain tube using distilled water as a working fluid for the present test and the well-known empirical correlation of Dittus –Boelter equation [79]. The present heat transfer results of test facility are in good agreement with the above equation with maximum deviation of (7.1 %).

Figures (4-5) to (4-8) show the agreement of friction factor with published Blasius equation [78], with maximum deviation of (6%).

4.3 Experimental Results

4.3.1Effect of Twisted Tape on the Thermal and Hydrodynamic Characteristics of the Flow

4.3.1.1 Thermal Effect of Twisted Tape

The variation in Nusselt number with Reynolds number for the plain tube in comparison with the four types of inserted tape and with twist ratios (TR = 4, 6) is shown in figures (4-9) to (4-24), for different heat fluxes. The effect of the twisted tape types for the twist ratios (TR = 4) is shown in figures (4-25) to (4-28).

In general, the Nusselt number increases as the Reynolds number increases for all the ranges of Reynolds number and types of inserted tape.

Nusselt number is greater when twist ratio is low for all types of twisted tape used, as shown in figures (4-9) to (4-24). Also, the Nusselt number increases when converting the twisted tape from typical twisted tape to clockwise-counter clockwise type and to perforated circular clockwise-counter clockwise twisted tape type, for twist ratio (TR=4), as shown in figures (4-25) and (4-28).

For the tube having inserted tape, the velocity of water in x and y directions are not zero, therefore, a rotational or secondary flow is created with a desired effect which promote mixing in the plane normal to the direction of bulk flow. This mixing works to keep a high gradient of temperature near the tube surface and in turn increases the improvement in heat transfer. The augmentation in the heat transfer (Nusselt number) relative to the smooth tube is explained using the Nusselt number ratio ($Nu_{\text{twisted tape}}/Nu_{\text{plain tube}}$), as shown in figures (4-29) to (4-44).

Figures (4-29) to (4-32) represent the relation between the Nusselt number ratio with Reynolds number for the typical inserted tape and twist ratios (TR=4, 6). It is clear that the Nusselt number ratio increases as twist ratio decreases, because of the increase in swirl flow as the ratio of twist decreases. The maximum enhancement in the heat transfer was for ratio (3.8) which occurred with twist ratio (TR = 4) at Reynolds number (4140) and heat flux ($\boldsymbol{q}''$=18599 W/m^2).

In figures (4-33) to (4-36) for clockwise-counter clockwise twisted tape, the maximum heat transfer enhancement was for ratio (4.4) which occurred with twist ratio (TR = 4) at Reynolds number (4140) and heat flux ($\boldsymbol{q}''$=18599 W/m^2).

Figures from (4-37) to (4-40) for perforated square clockwise-counter clockwise inserted tape, the maximum improvement in the heat transfer was for ratio (4.7) which occurred with twist ratio (TR = 4) at Reynolds number (4140) and heat flux ($\boldsymbol{q}''$=18599 W/m^2) with (L=4mm).

In figures (4-41) to (4-44) for perforated circular clockwise-counter clockwise inserted tape, the maximum augmentation in the heat transfer was for ratio (5.3) which occurred with twist ratio (TR = 4) at Reynolds number (4140) and heat flux ($\boldsymbol{q}''$=18599 W/m^2) with (D=4mm).

4.3.1.2 Hydrodynamic Effect of Twisted Tape

The variation in friction coefficient with Reynolds number for the smooth tube in comparison with the four inserted tape types, when the twist ratios are (TR = 4, 6), is shown in figures (4-45) to (4-60), at different heat fluxes.

Generally, the friction factor decreases as the Reynolds number and twist ratio increase for the whole range of Reynolds number and types of twisted tape. As the ratio of twist decreases, the swirling flow (secondary flow) increases, this leads to the high friction factor, the highest friction factor occurs with the smallest twist ratio.

The friction coefficient increases when converting the twisted tape from the typical twisted tape to clockwise-counter clockwise twisted tape and from perforated square clockwise-counter clockwise twisted tape to perforated circular clockwise-counter clockwise twisted tape for all the twist ratios (TR = 4, 6) as shown in figures (4-49) to (4-60), this increases the disturbance in the laminar sub layer of the boundary layer because of the perforated circular and square presence in the twisted tape.

The increase in the pressure drop (factor of friction) relative to the smooth tube is explained using the friction factor ratio ($f_{\text{twisted tape}}$ / $f_{\text{plain tube}}$), as shown in figures (4-61) to (4-76).

Figures (4-61) to (4-64) represent the relation between friction coefficient ratio and Reynolds number for the typical inserted tape for twist ratios (TR = 4, 6). From this figure, it is clear that the friction coefficient ratio decreases as Reynolds number and twist ratio increase. The maximum increase in the pressure drop (friction factor) was for friction factor ratio (1.57) which is found with twist ratio (TR = 4) at Reynolds number (4842), and the minimum improve in the friction coefficient was for friction factor ratio (1.27) which occurred with twist ratio (TR = 6) at Reynolds number (7209).

For clockwise-counter clockwise twisted tape, figures (4-65) to (4-68), the maximum increase in the friction coefficient was for ratio (1.92) which found with twist ratio (TR = 4) at Reynolds number (10814), and the minimum increase in the friction coefficient was for ratio (1.55) which occurred with twist ratio (TR = 6) at Reynolds number (7209).

In figures (4-69) to (4-72), for perforated square clockwise-counter clockwise inserted tape, the maximum increase in the friction factor was for ratio (2.16) which found with twist ratio (TR = 4) at Reynolds number (10814) and (L=4mm). The minimum increase in the friction factor was for ratio (1.75) which occurred with twist ratio (TR = 4) at Reynolds number (7209) and (L=7mm).

For perforated circular clockwise-counter clockwise twisted tape, figures (4-73) to (4-76), the maximum increase in the friction factor was for ratio (2.28) which found with twist ratio (TR = 4) at Reynolds number (10814) and (D=4mm). The minimum increase in the friction factor was for ratio (1.85) which occurred with twist ratio (TR = 4) at Reynolds number (4040) and (D =7mm).

As a result, typical twisted tape has the minimum value of friction factor ratio, but the perforated circular clockwise-counter clockwise inserted tape has the maximum value of friction factor ratio.

4.3.1.3 Thermal Performance Factor Evaluation

The heat transfer performance index (ŋ) can be calculated from equation (3-18), which is defined as the ratio of the improved Nusselt number by enhancement techniques (swirl generators) to original Nusselt number for plain tube ratio to the ratio of friction coefficient with enhancement techniques to the plain tube friction factor at the same pumping power. This coefficient is one of the key parameters necessary to define the heat transfer augmentation performance. In general, the factor of thermal performance above unity indicates that the heat transfer augmentation influence due to (turbulator device) is more dominant than the influence of rising friction and vice versa.

Figures (4-77) to (4-92) indicate the thermal performance variation at various Reynolds number flow. It is noticed that, the thermal performance factor results have the highest level for twisted tapes inserts than smooth tube. Also, it can be seen from figures that the coefficient of thermal performance values increases as Reynolds number increase due to considerable significant for the enhancement in heat transfer rate produced by the twisted tape.

In figure (4-77) to (4-80), the maximum factor of thermal performance for typical twisted tape was (3.32) for twist ratio (TR = 4) at Reynolds number (4140). Figures (4-81) to (4-84) depict that the maximum factor of thermal performance for clockwise-counter clockwise inserted tape was (3.74) for twist ratio (TR = 4) at Reynolds number (7209). The maximum augmenated in the heat transfer occurs with twist ratio (TR = 4) because at this ratio the increase in Nusselt number overcomes the increase in friction coefficient.

In figure (4-85) to (4-88), the maximum factor thermal performance for perforated square clockwise-counter clockwise twisted tape was (3.86) for twist ratio (TR = 4) at Reynolds number (4140) and (L=4mm).

Figures (4-89) to (4-92) show that the maximum thermal performance coefficient for perforated circular clockwise-counter clockwise inserted tape was (4.2) for twist ratio (TR = 4) at Reynolds number (4140) and (D=4mm).

As a result, typical twisted tape has the minimum value of thermal performance factor, but the perforated circular clockwise-counter clockwise twisted tape has the maximum value of thermal performance coefficient.

4.4 Correlations

Experimental data of average Nusselt number and factor of friction have been empirically correlated through using the DGA (Dissolved Gas Analysis)program in terms of algebraic expression of the form ($Y = ax_1^{N1}x_2^{N2}$). The obtained data are related with Re, Pr and TR (y) with distilled water through the following correlations:

1- Correlations for Tube with Inserted Twisted Tape:

For four types of twisted tapes, the equation is:

$$\overline{Nu} = C\, Re^m\, Pr^n\, y^f \qquad (4\text{-}1)$$

The fitted values of average Nusselt number are represented by the equations from (4-2) to (4-5) for typical, clockwise-counter clockwise, perforated square clockwise-counter clockwise and perforated circular clockwise-counter clockwise twisted tapes, respectively. The fitted values are compared with experimental data as shown in figures (4-93) to (4-96). The predicted values are agreeing with the experimental result within ± 7%, ± 3%, ± 4.5% and ± 4% for typical, clockwise-counter clockwise, perforated square clockwise-counter clockwise and perforated circular clockwise-counter clockwise twisted tapes, respectively.

$$\overline{Nu} = 0.0028\ Re^{1.08}\ Pr^{0.47}\ y^{-1.9} \quad \text{(Typical insert)} \qquad (4\text{-}2)$$

$$\overline{Nu} = 1.71\ Re^{0.16}\ Pr^{1.69}\ y^{-1.3} \quad \text{(Clockwise-counter clockwise)} \qquad (4\text{-}3)$$

$$\overline{Nu} = 0.0017\ Re^{1.84}\ Pr^{0.17}\ y^{-1.2} \quad \text{(Perforated square)} \qquad (4\text{-}4)$$

$$\overline{Nu} = 0.0012\ Re^{1.58}\ Pr^{0.60}\ y^{-0.1} \quad \text{(Perforated circular)} \qquad (4\text{-}5)$$

2- Correlations for Friction Factor:

The friction factor is correlated generally by:

$$f = C\ Re^{m}\ y^{f} \qquad (4\text{-}6)$$

The fitted values of friction factor are represented by the equations (4-7) to (4-10) for typical, clockwise-counter clockwise, perforated square clockwise-counter clockwise and perforated circular clockwise-counter clockwise twisted tapes, respectively. Figures (4-97) to (4-100) give the representation of these correlations. The fitted data are agreeing with the experimental results within ± 6%, ± 5.7%, ± 4.2% and ± 2.9% for typical, clockwise-counter clockwise, perforated square clockwise-counter clockwise and perforated circular clockwise-counter clockwise twisted tapes, respectively.

$$f = 0.823\ Re^{-0.0093}\ y^{-0.87} \quad \text{(Typical insert)} \qquad (4\text{-}7)$$

$$f = 0.850\ Re^{-0.0061}\ y^{-0.89} \quad \text{(Clockwise-counter clockwise)} \qquad (4\text{-}8)$$

$$f = 0.868\ Re^{-0.0046}\ y^{-0.90} \quad \text{(Perforated square)} \qquad (4\text{-}9)$$

$$f = 0.875\ Re^{-0.0041}\ y^{-0.91} \quad \text{(Perforated circular)} \qquad (4\text{-}10)$$

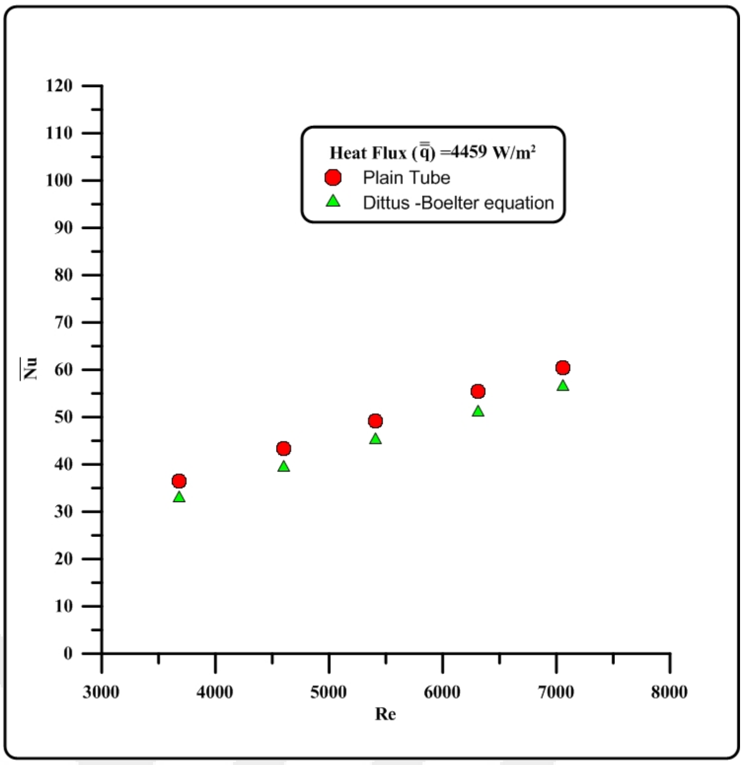


Figure 4. 1: Comparison between experimental work and Dittus –Boelter equation at $\boldsymbol{q}$"=4459 W/m2

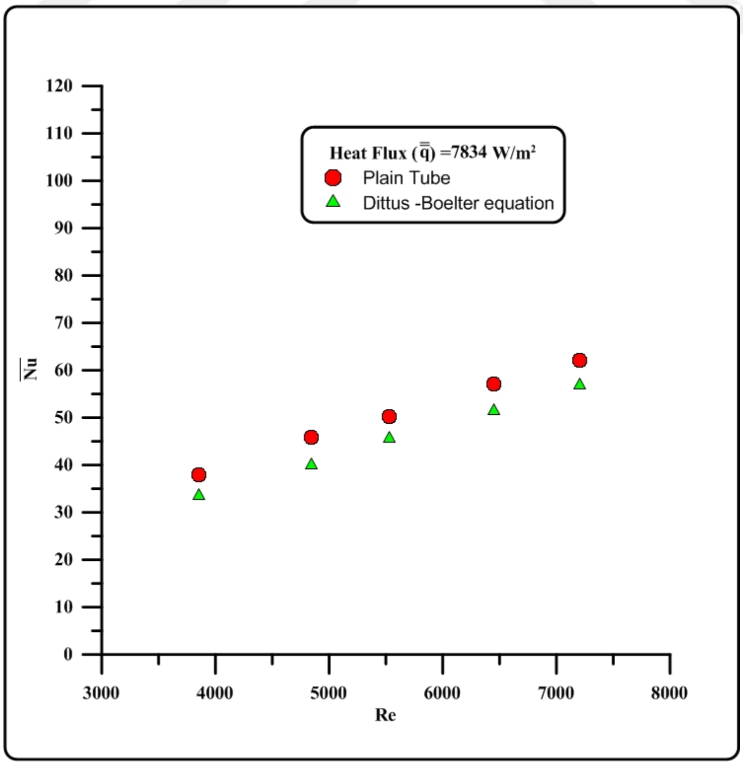


Figure 4. 2: Comparison between experimental work and Dittus –Boelter equation at $\boldsymbol{q}$"=7834 W/m2

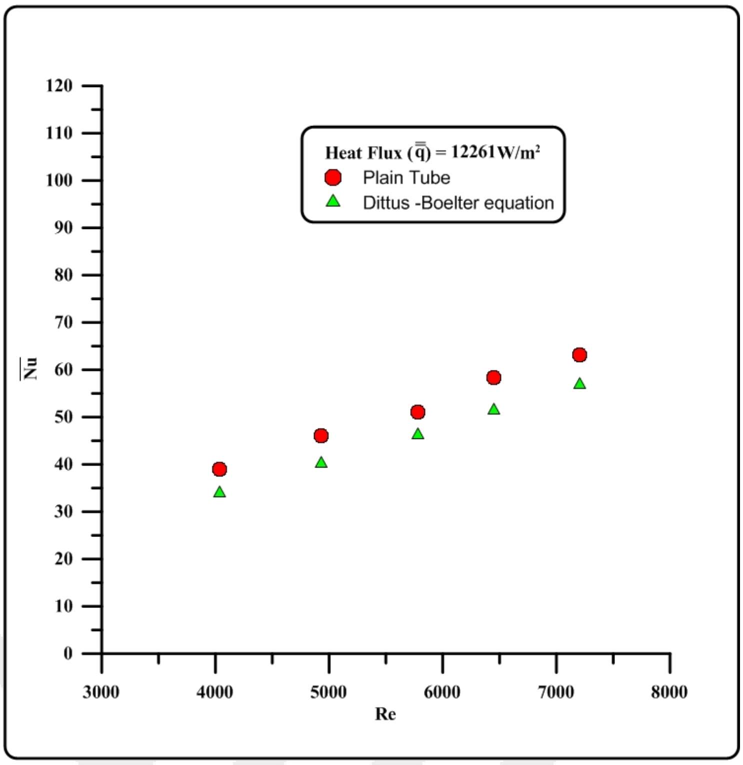


Figure 4. 3: Comparison between experimental work and Dittus –Boelter equation at ***q***"=12261 W/m2

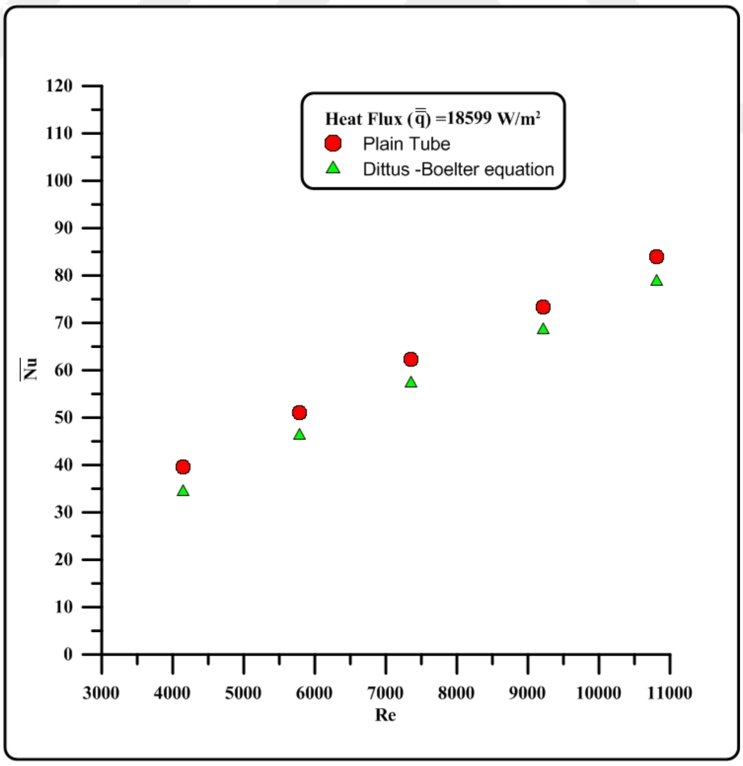


Figure 4. 4: Comparison between experimental work and Dittus –Boelter equation at ***q***"=18599 W/m2

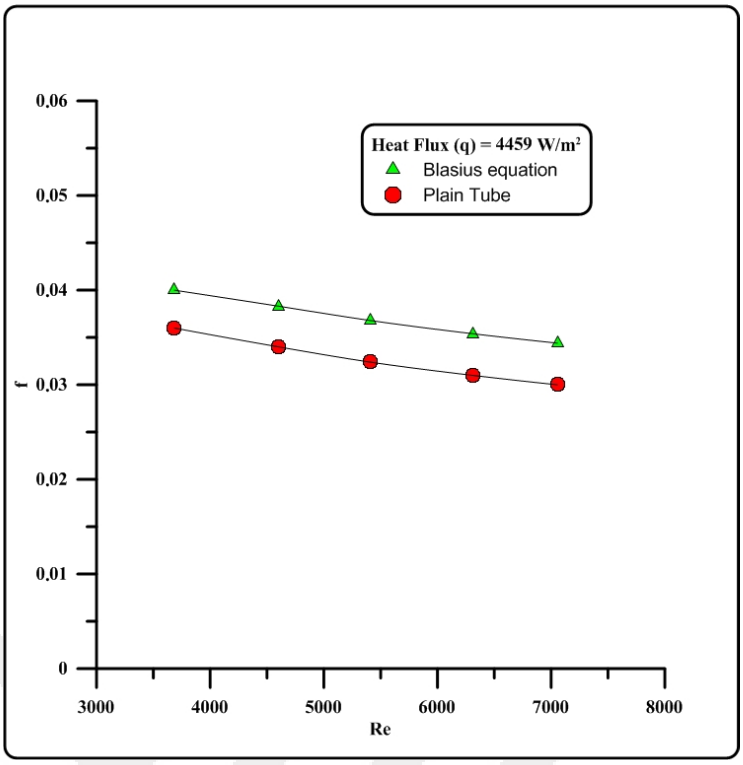


Figure 4. 5: Comparison between experimental work and Blasius equation at ***q***"=4459 W/m2

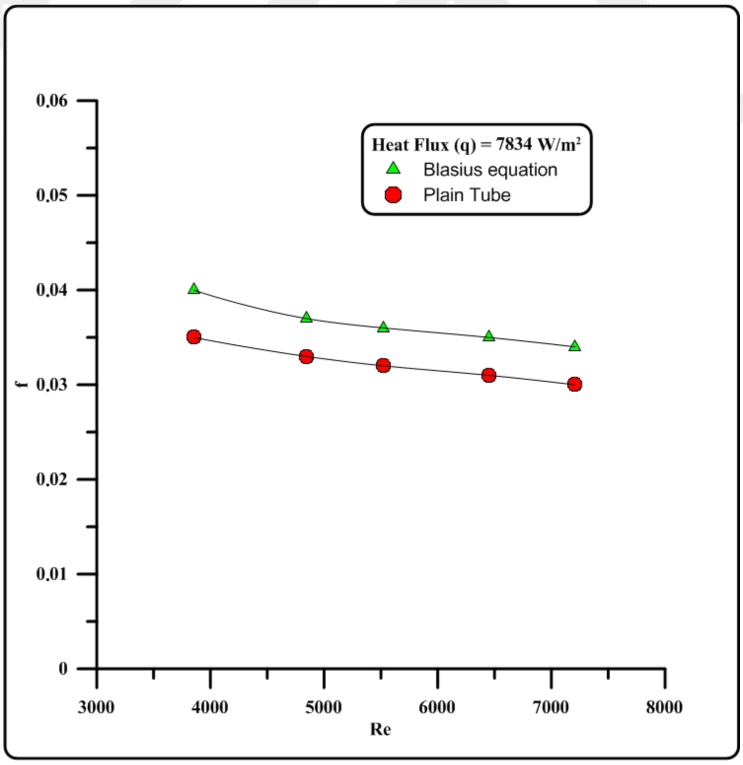


Figure 4. 6: Comparison between experimental work and Blasius equation at ***q***"=7834 W/m2

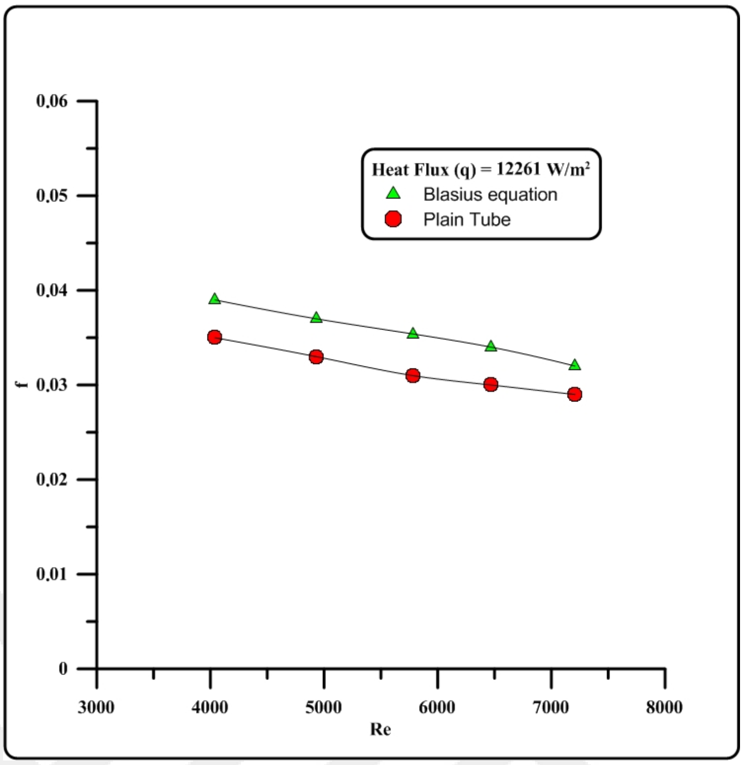


Figure 4. 7: Comparison between experimental work and Blasius equation at ***q***"=12261 W/m2

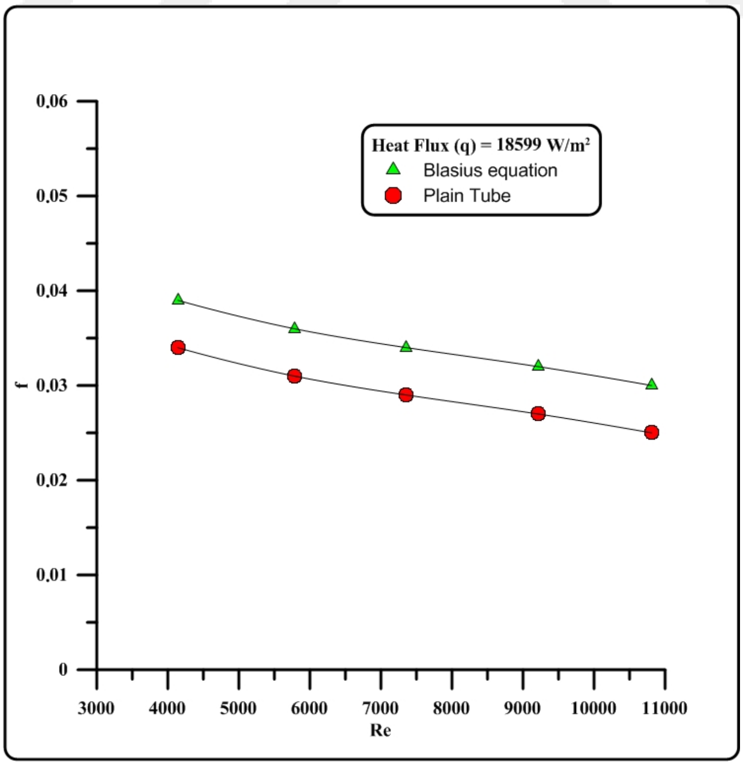


Figure 4. 8: Comparison between experimental work and Blasius equation at ***q***"=18599 W/m2

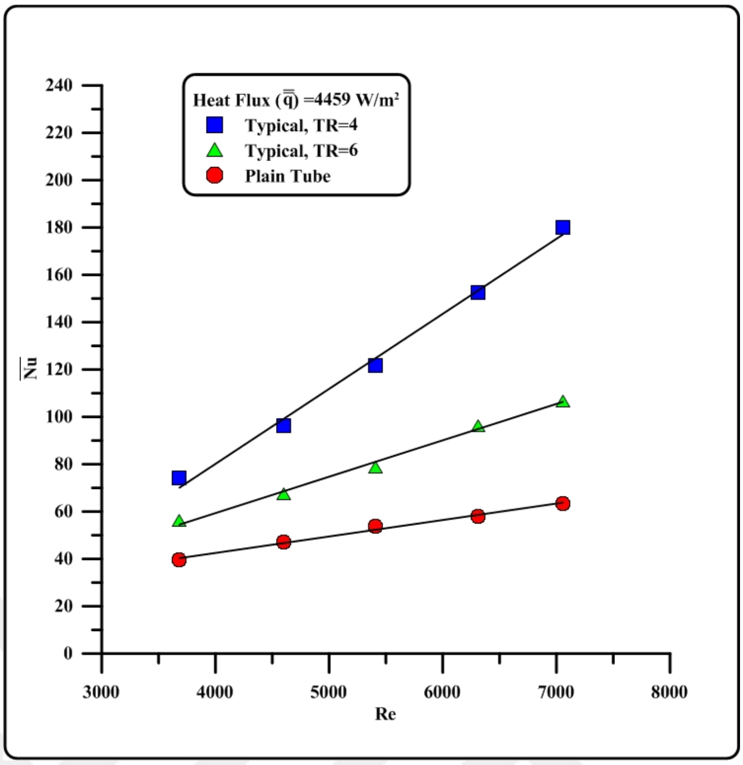


Figure 4. 9: The effect of Reynolds number and twist ratio for typical twisted tape on Nusselt number at q''=4459 W/m2

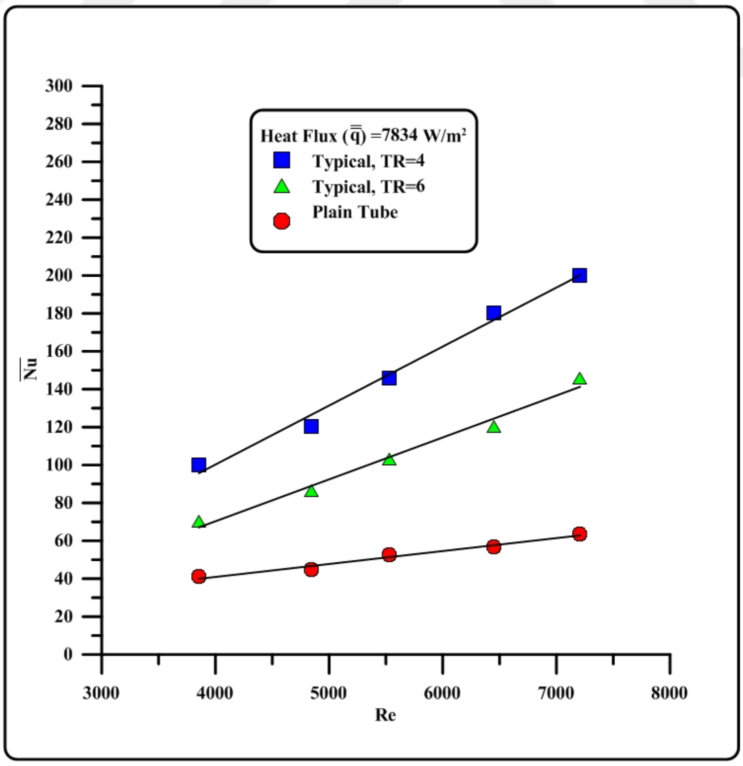


Figure 4. 10: The effect of Reynolds number and twist ratio for typical twisted tape on Nusselt number at q''=7834 W/m2

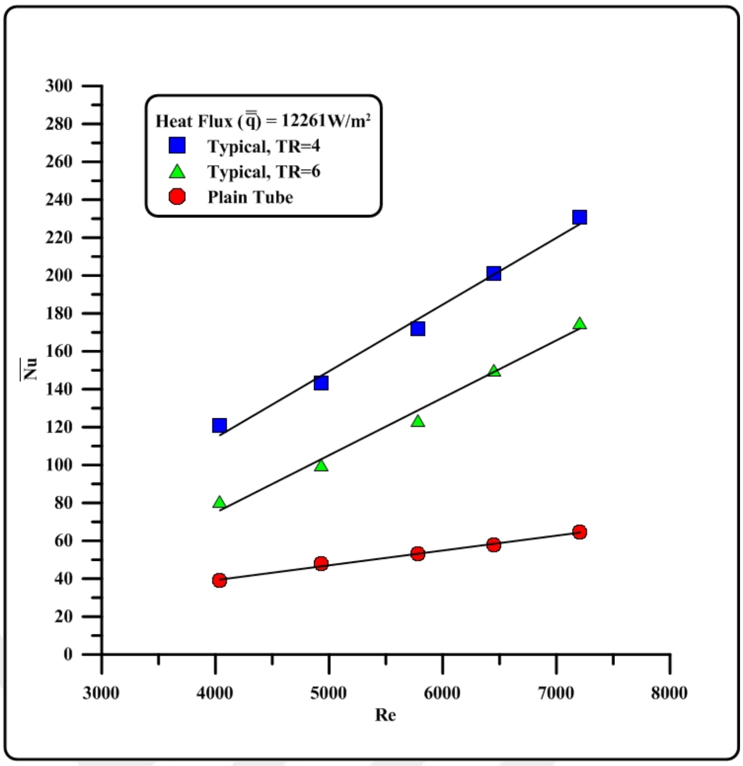


Figure 4. 11: The effect of Reynolds number and twist ratio for typical twisted tape on Nusselt number at ***q***"=12261 W/m2

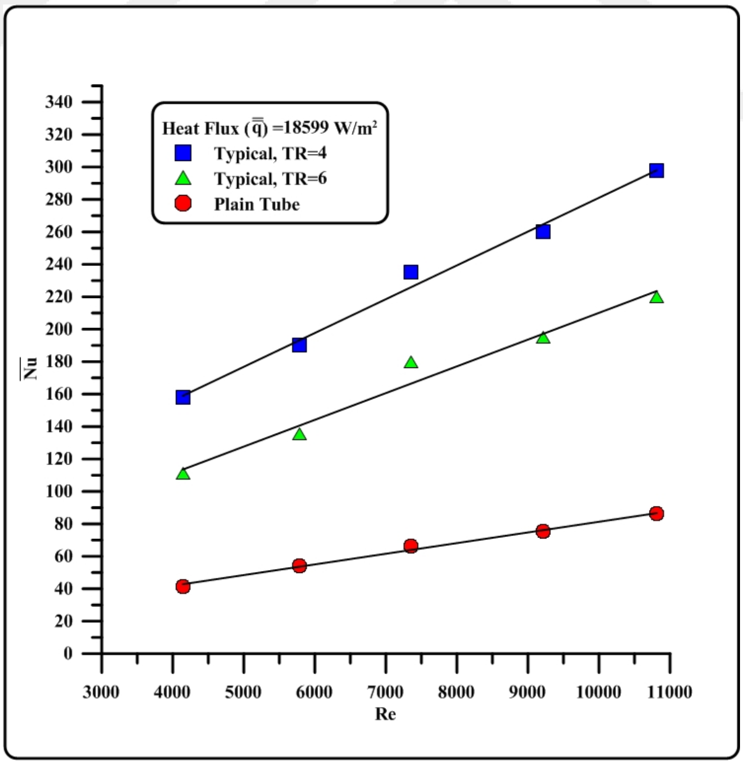


Figure 4. 12: The effect of Reynolds number and twist ratio for typical twisted tape on Nusselt number at ***q***"=18599 W/m2

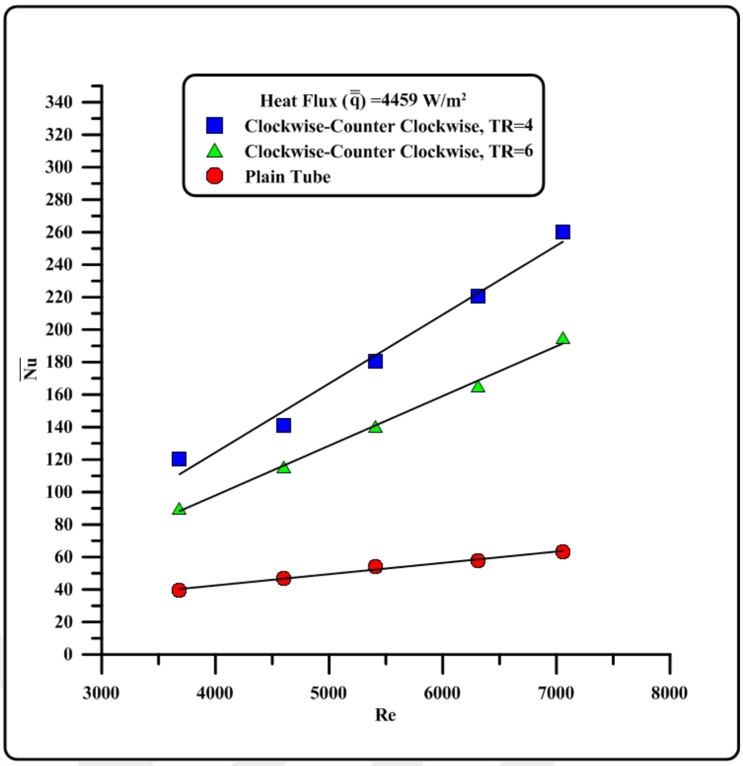


Figure 4. 13: The effect of Reynolds number and twist ratio for clockwise-counter clockwise twisted tape on Nusselt at ***q***"=4459 W/m2

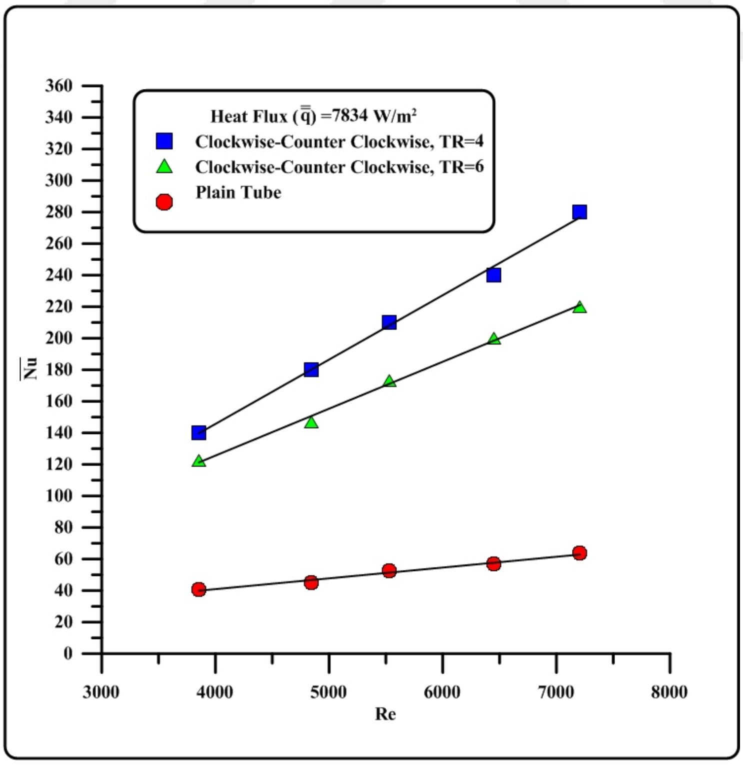


Figure 4. 14: The effect of Reynolds number and twist ratio for clockwise-counter clockwise twisted tape on Nusselt number at ***q***"=7834 W/m2

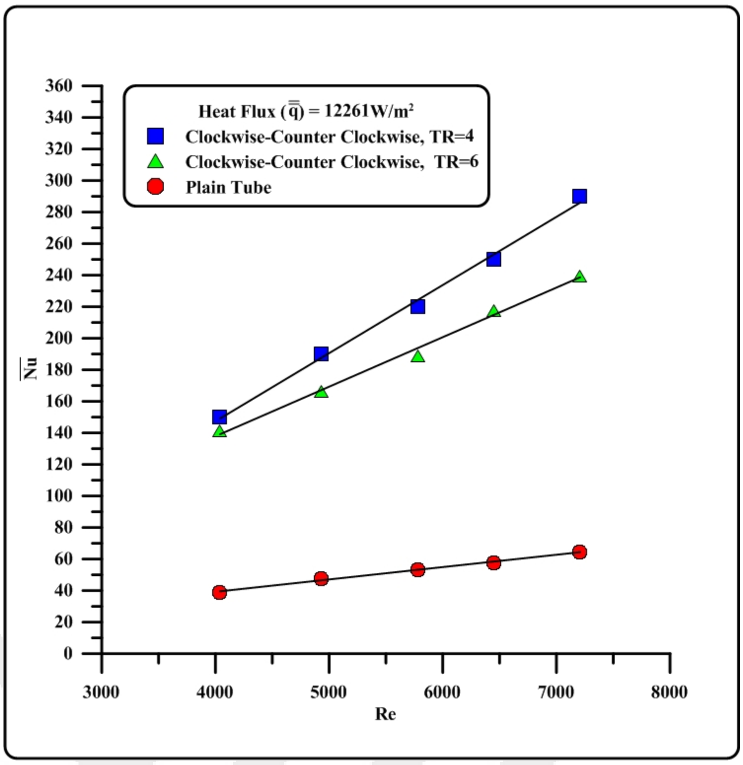


Figure 4. 15: The effect of Reynolds number and twist ratio for clockwise-counter clockwise twisted tape on Nusselt number at $\boldsymbol{q}$"=12261 W/m2

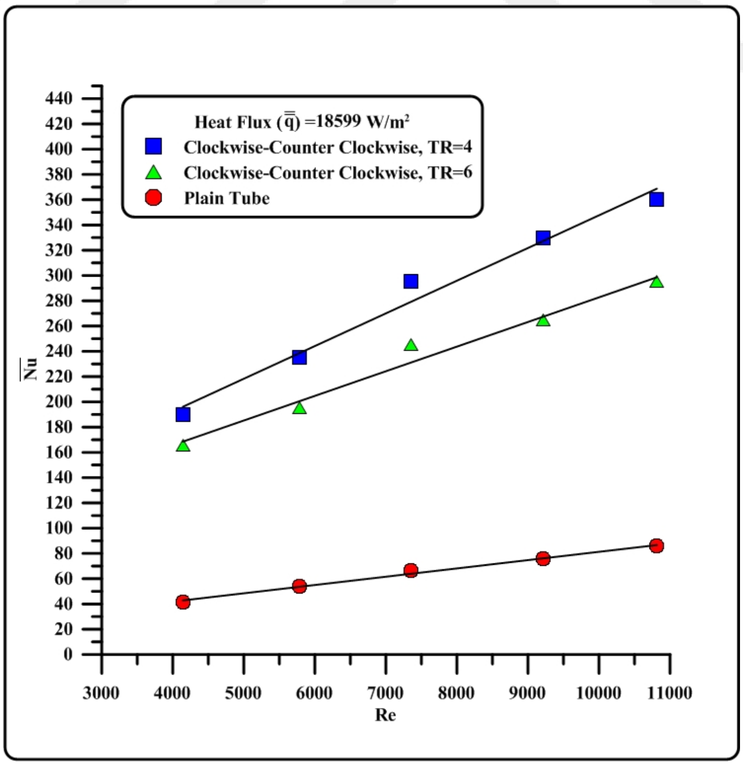


Figure 4. 16: The effect of Reynolds number and twist ratio for clockwise-counter clockwise twisted tape on Nusselt number at $\boldsymbol{q}$"=18599 W/m2

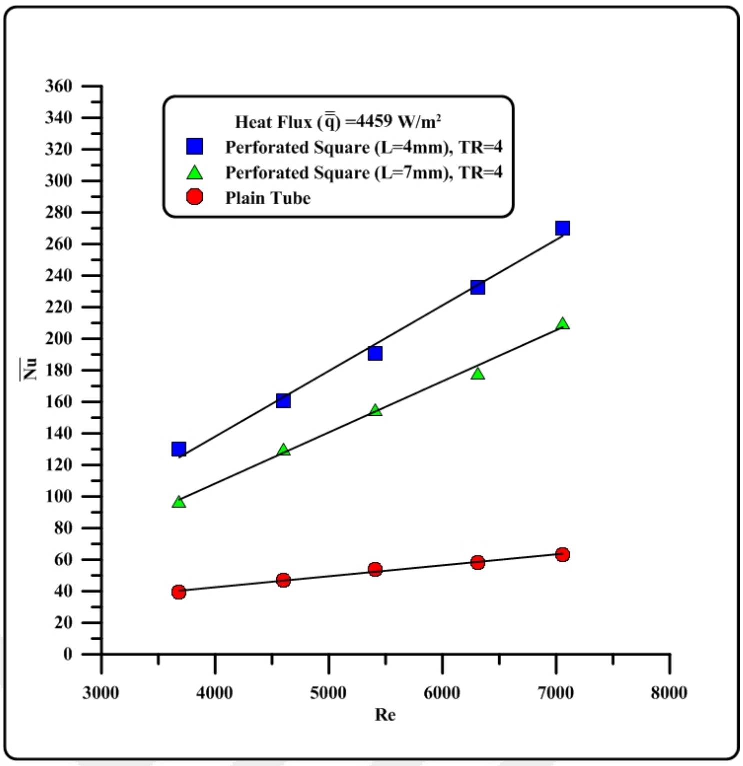


Figure 4. 17: The effect of Reynolds number and twist ratio for perforated square clockwise-counter clockwise twisted tape on Nusselt number at ***q***"=4459 W/m2

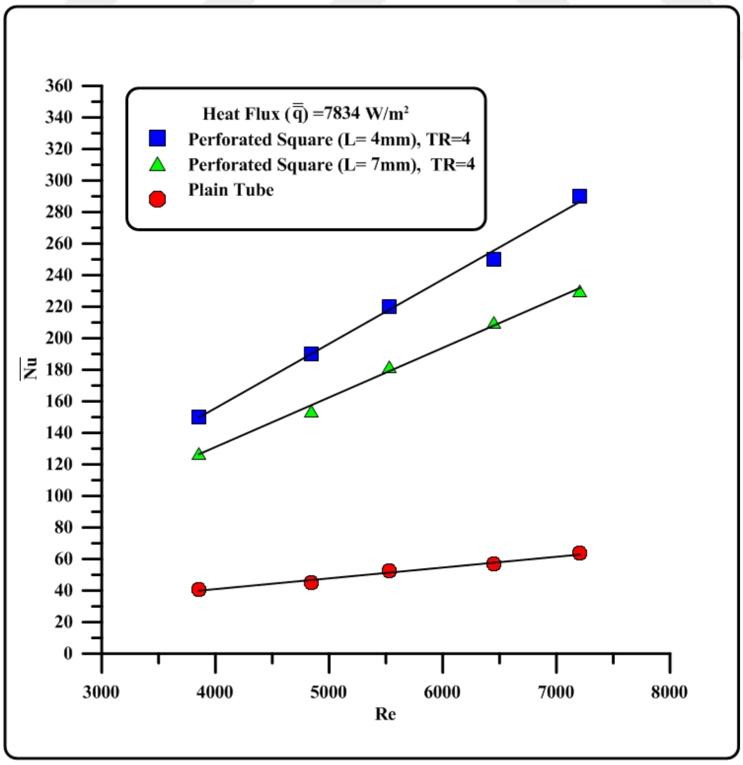


Figure 4. 18: The effect of Reynolds number and twist ratio for perforated square clockwise-counter clockwise twisted tape on Nusselt number at ***q***"=7834 W/m2.

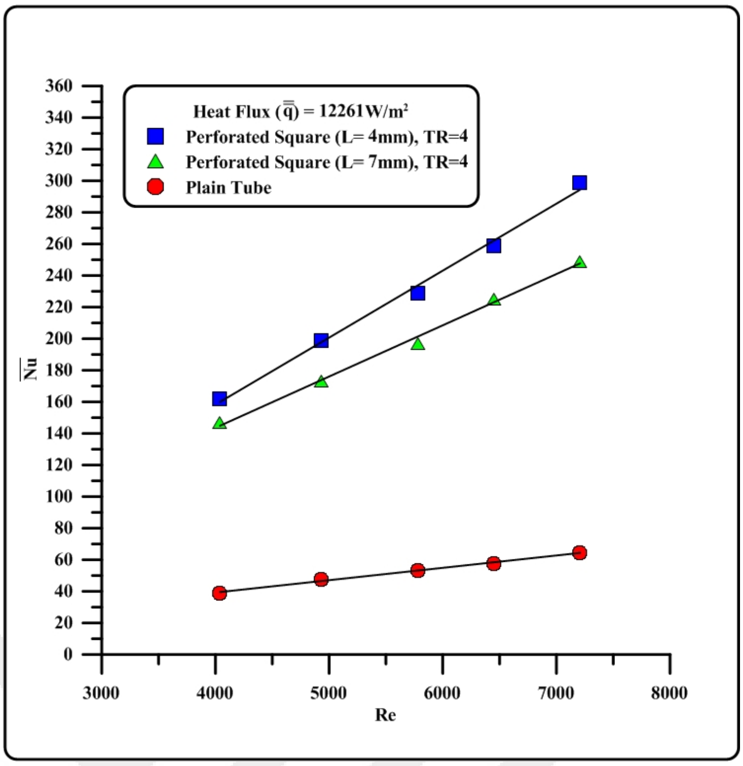


Figure 4. 19: The effect of Reynolds number and twist ratio for perforated square clockwise-counter clockwise twisted tape on Nusselt number at q"=12261 W/m2.

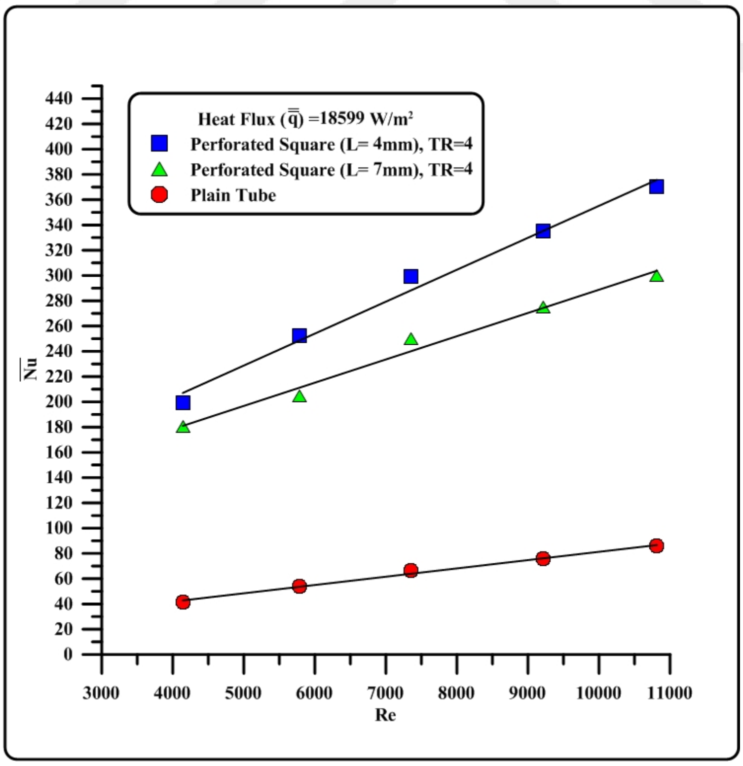


Figure 4. 20: The effect of Reynolds number and twist ratio for perforated square clockwise-counter clockwise twisted tape on Nusselt number at q"=18599 W/m2.

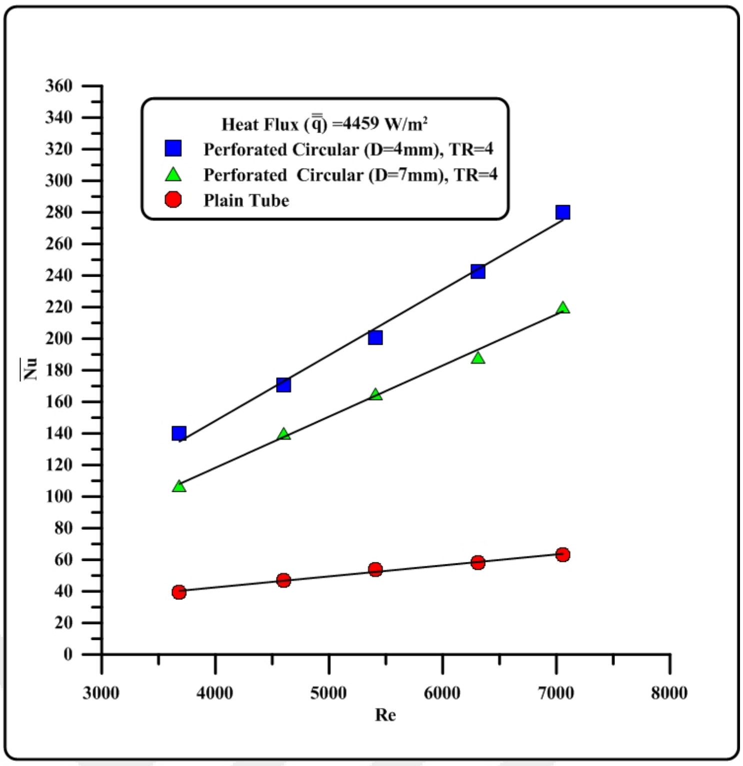


Figure 4. 21: The effect of Reynolds number and twist ratio for perforated circular clockwise-counter clockwise twisted tape on Nusselt number at q''=4459 W/m2.

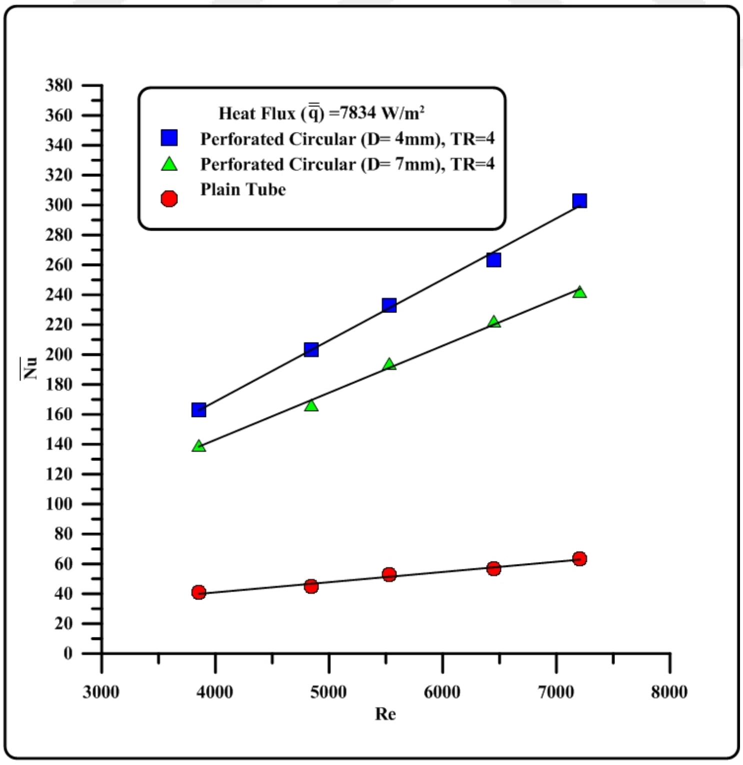


Figure 4. 22: The effect of Reynolds number and twist ratio for perforated circular clockwise-counter clockwise twisted tape on Nusselt number at q''=7834 W/m2.

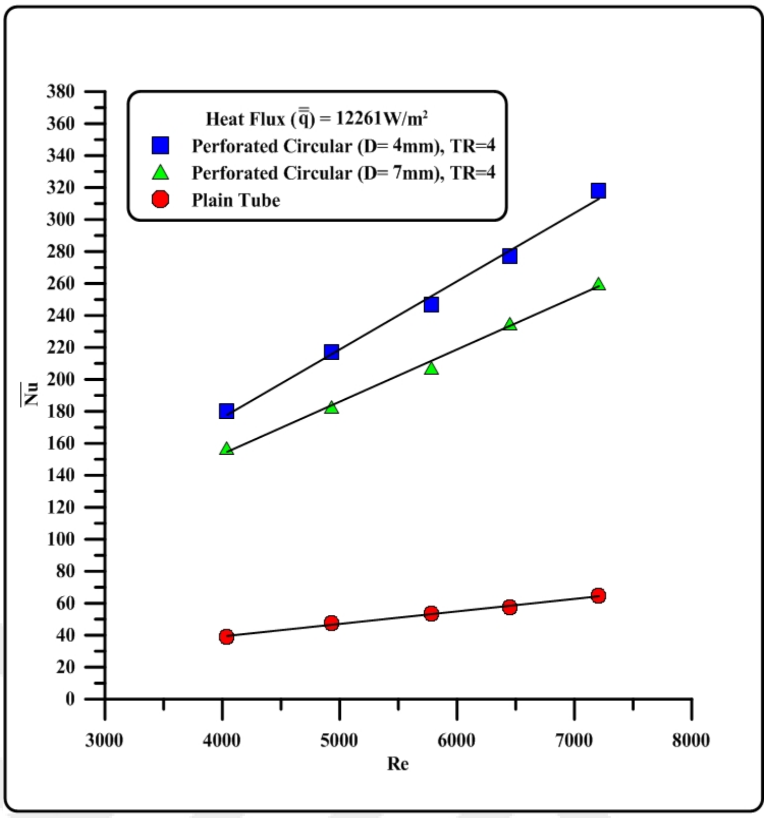


Figure 4. 23: The effect of Reynolds number and twist ratio for perforated circular clockwise-counter clockwise twisted tape on Nusselt number at q"=12261 W/m2.

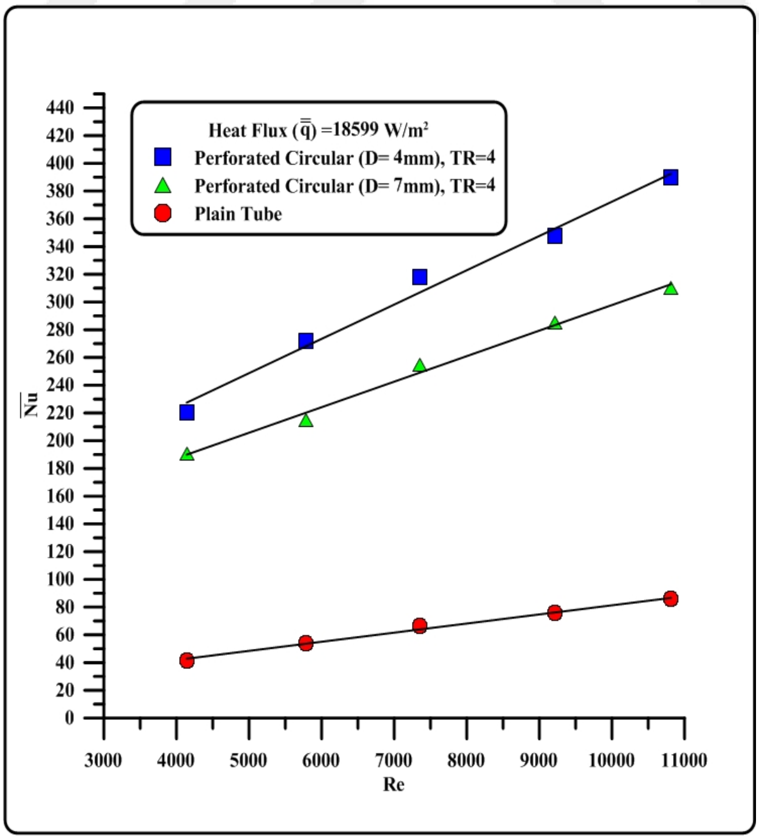


Figure 4. 24: The effect of Reynolds number and twist ratio for perforated circular clockwise-counter clockwise twisted tape on Nusselt number at q"=18599 W/m2.

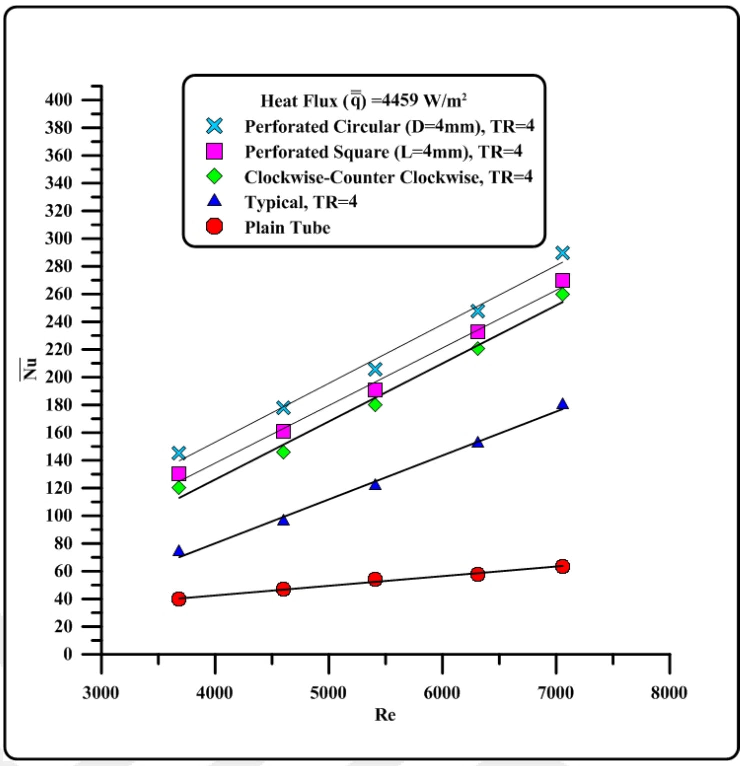


Figure 4. 25: The effect of Reynolds number and type of twisted tape for twist ratio (TR=4) on Nusselt number at ***q***"=4459 W/m2.

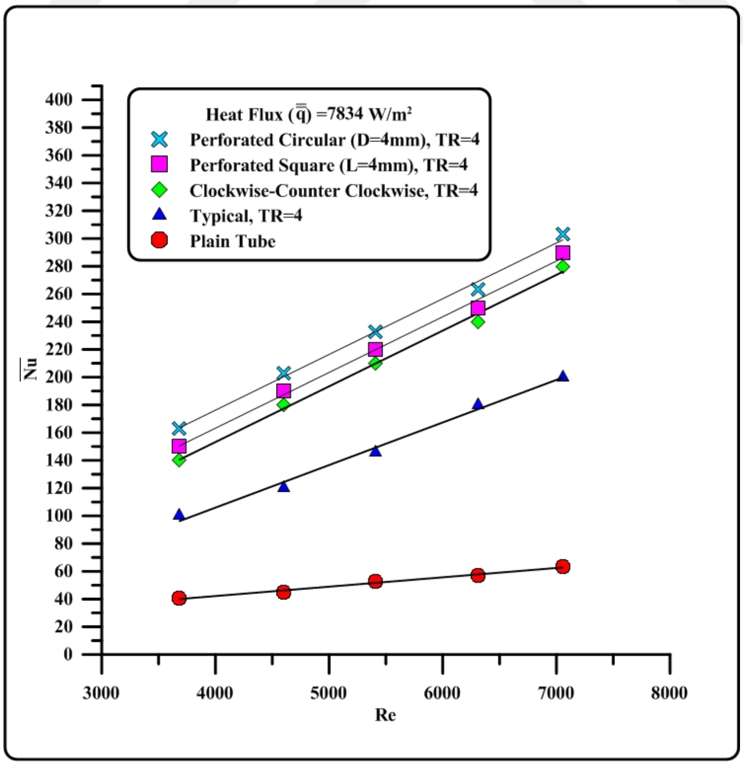


Figure 4. 26: The effect of Reynolds number and type of twisted tape for twist ratio (TR=4) on Nusselt number at ***q***"=7834 W/m2.

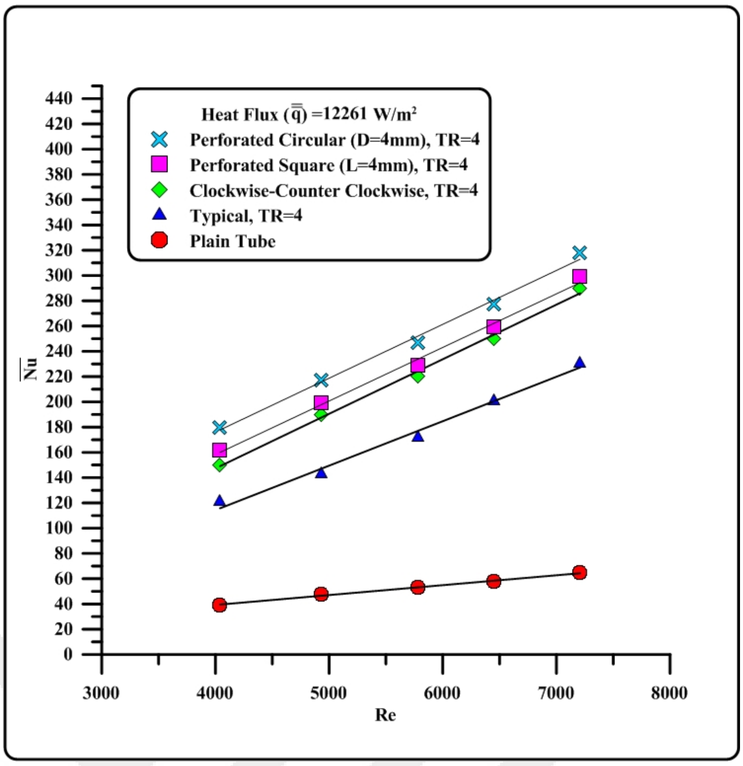


Figure 4. 27: The effect of Reynolds number and type of twisted tape for twist ratio (TR=4) on Nusselt number at q''=12261 W/m2.

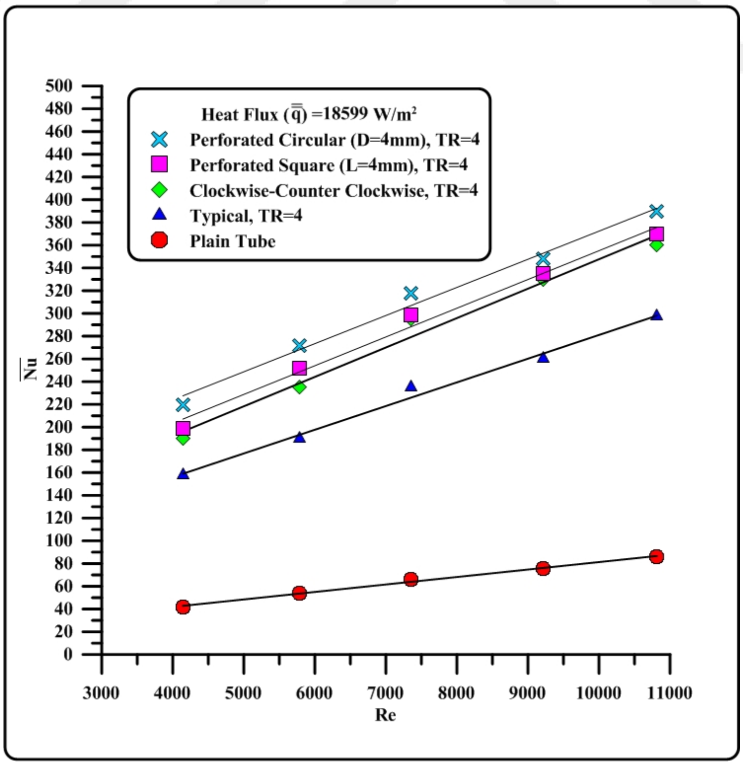


Figure 4. 28: The effect of Reynolds number and type of twisted tape for twist ratio (TR=4) on Nusselt number at q''=18599 W/m2.

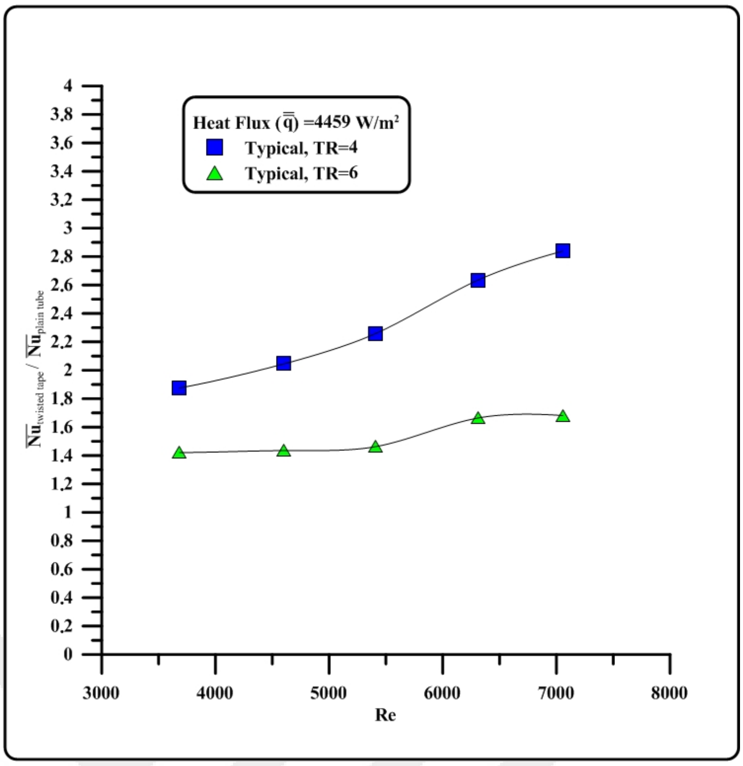


Figure 4. 29: The effect of Reynolds number and twist ratio of typical twisted tape on Nusselt number ratio for distilled water at $\boldsymbol{q}$"=4459 W/m2.

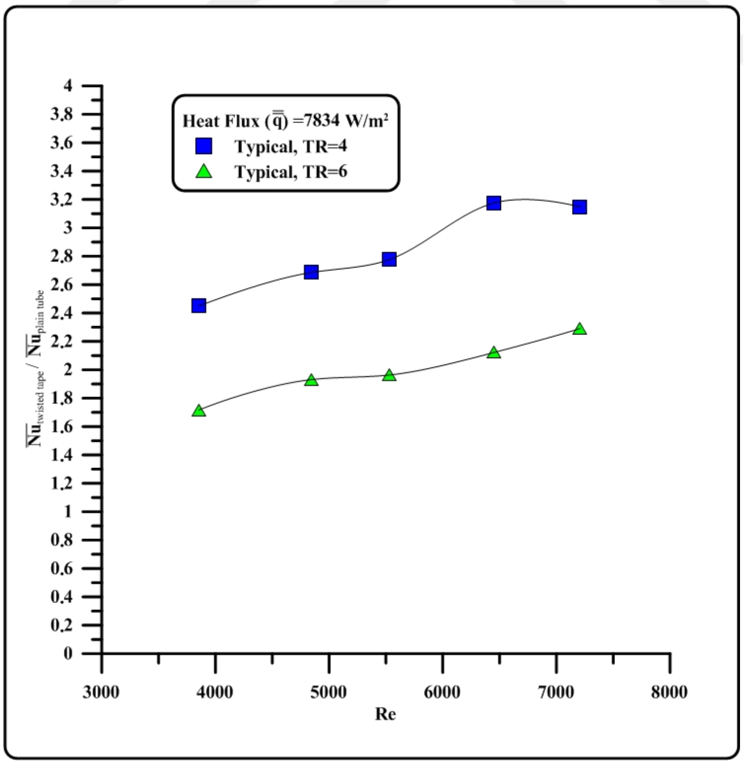


Figure 4. 30: The effect of Reynolds number and twist ratio of typical twisted tape on Nusselt number ratio for distilled water at $\boldsymbol{q}$"=7834 W/m2.

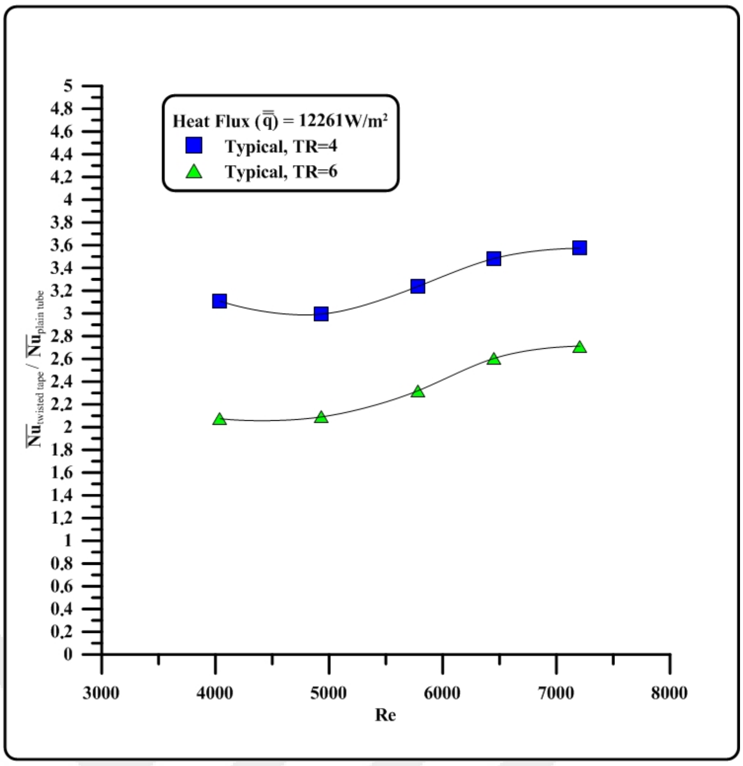


Figure 4. 31: The effect of Reynolds number and twist ratio of typical twisted tape on Nusselt number ratio for distilled water at q''=12261 W/m2.

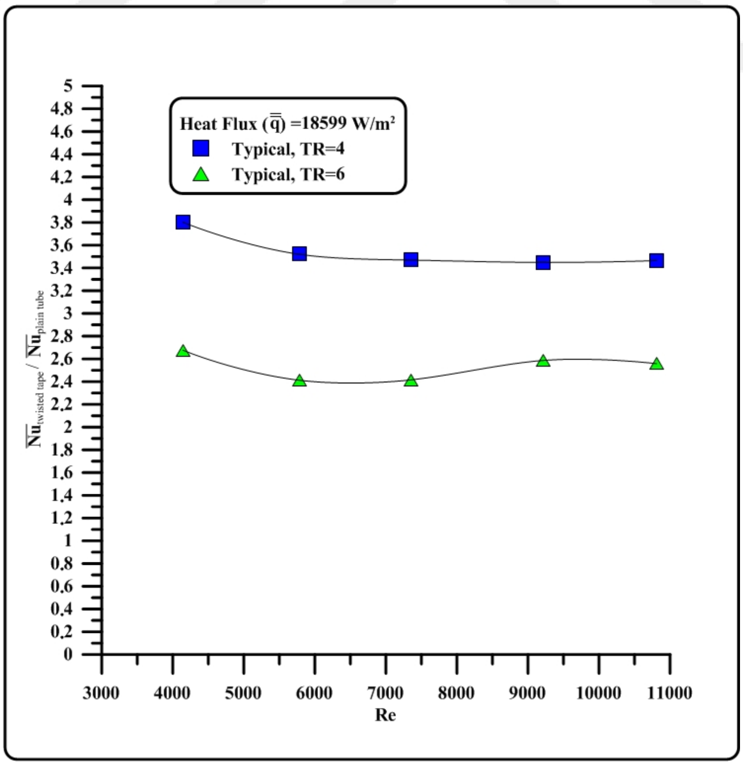


Figure 4. 32: The effect of Reynolds number and twist ratio of typical twisted tape on Nusselt number ratio for distilled water at q''=18599 W/m2.

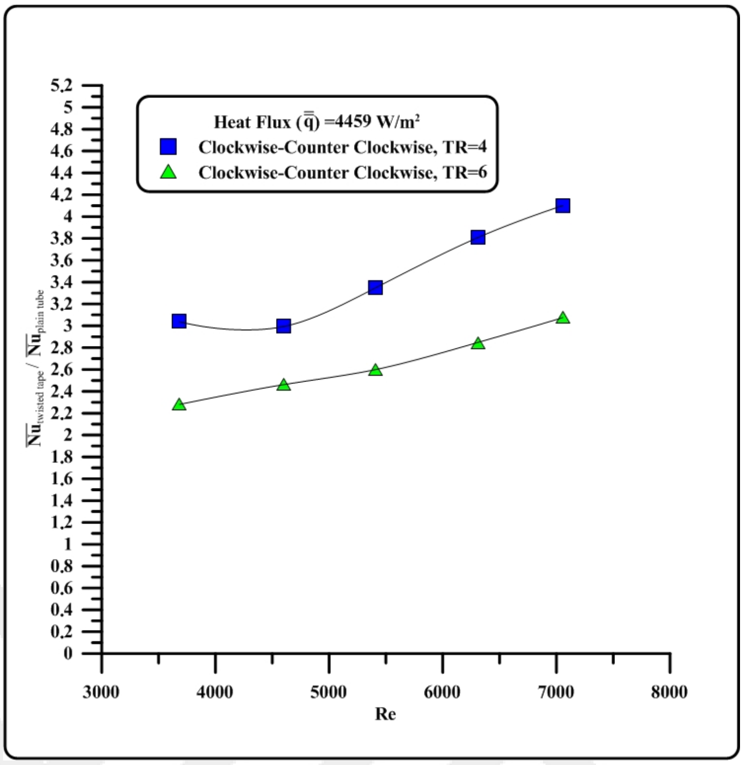


Figure 4. 33: The effect of Reynolds number and twist ratio of clockwise-counter clockwise tape on Nusselt number ratio for distilled water at ***q***"=4459 W/m2.

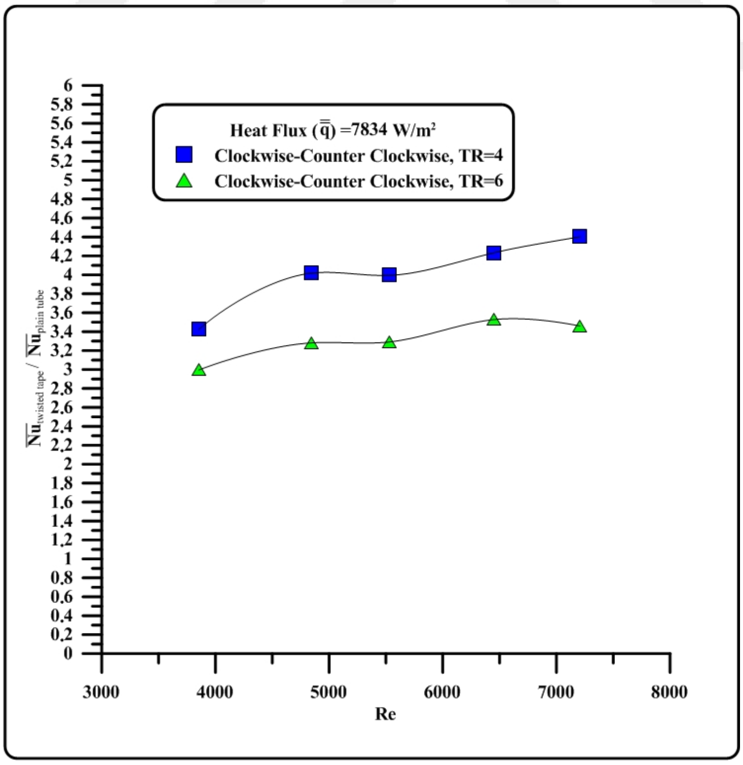


Figure 4. 34: The effect of Reynolds number and twist ratio of clockwise-counter clockwise tape on Nusselt number ratio for distilled water at ***q***"=7834 W/m2.

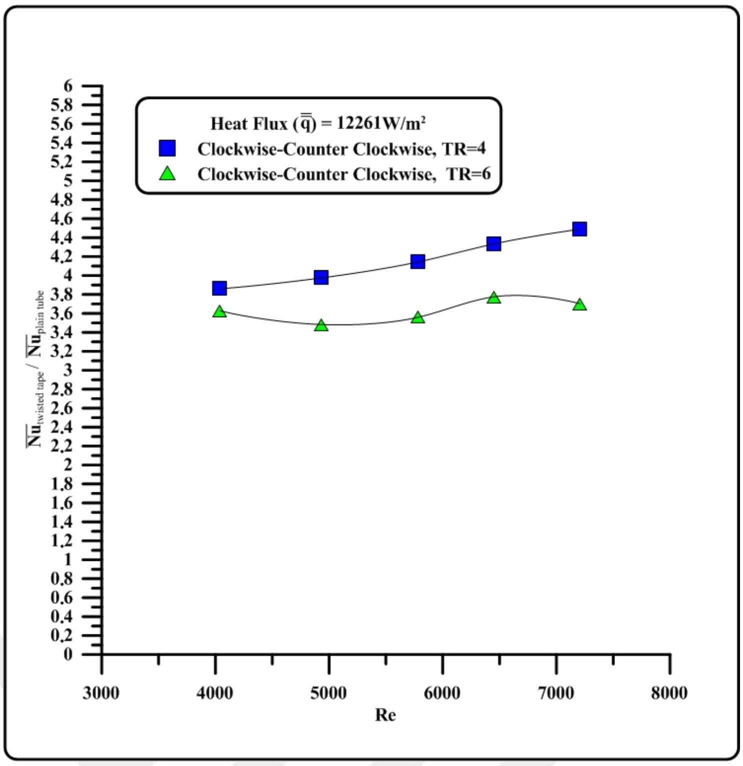


Figure 4. 35: The effect of Reynolds number and twist ratio of clockwise-counter clockwise tape on Nusselt number ratio for distilled water at q''=12261 W/m2.

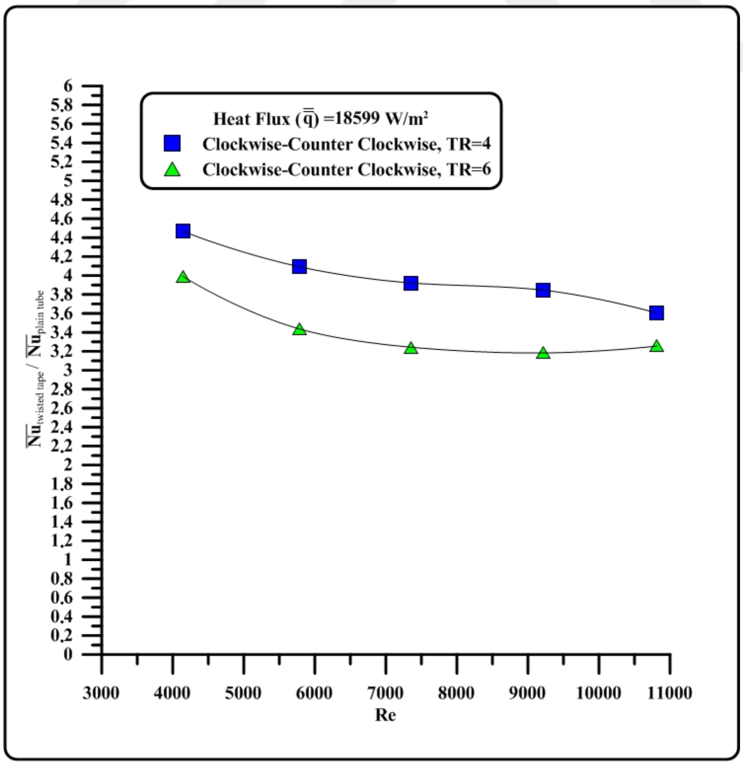


Figure 4. 36: The effect of Reynolds number and twist ratio of clockwise-counter clockwise tape on Nusselt number ratio for distilled water at q''=18599 W/m2.

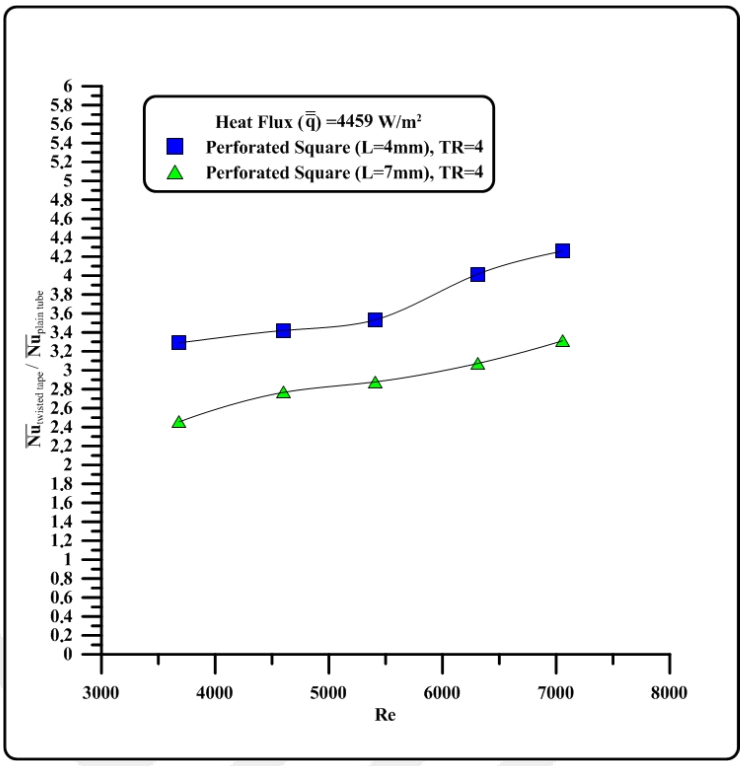
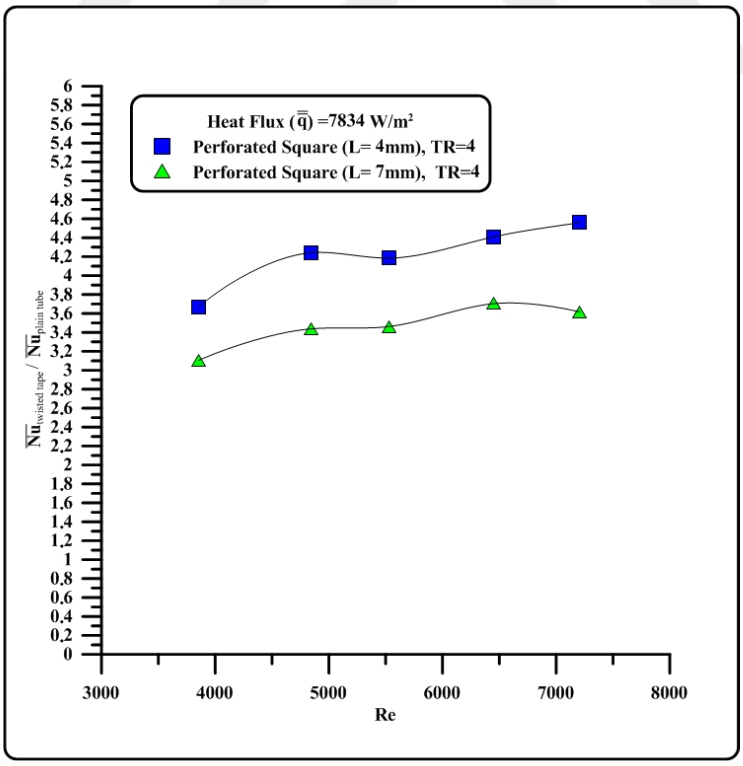


Figure 4. 37: The effect of Reynolds number and twist ratio of perforated square clockwise-counter clockwise tape on Nusselt number ratio for distilled water at ***q***"=4459 W/m2.



- **Figure 4. 38:** The effect of Reynolds number and twist ratio of perforated square clockwise-counter clockwise tape on Nusselt number ratio for distilled water at ***q***"=7834 W/m2.

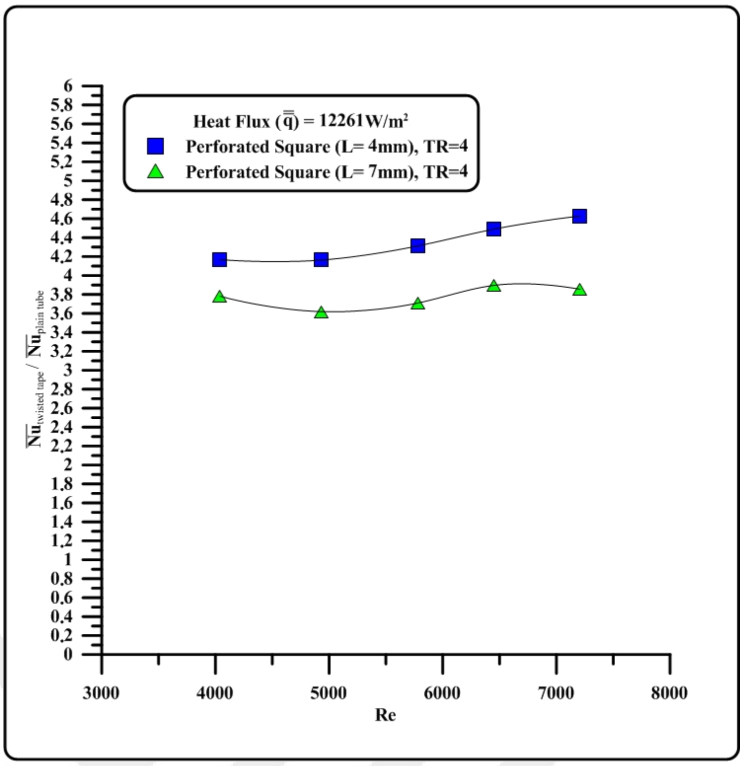


Figure 4. 39: The effect of Reynolds number and twist ratio of perforated square clockwise-counter clockwise tape on Nusselt number ratio for distilled water at q"=12261 W/m2.

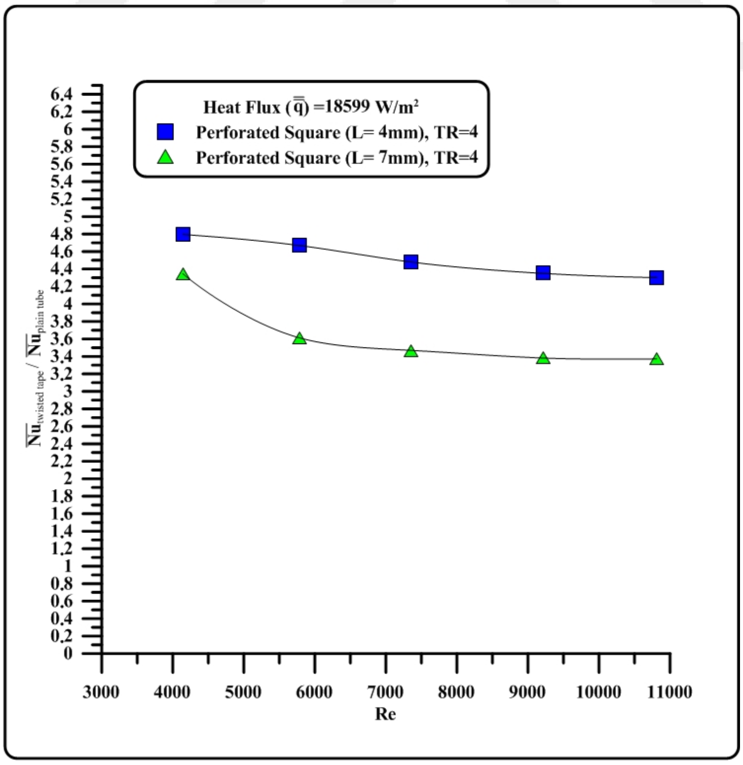


Figure 4. 40: The effect of Reynolds number and twist ratio of perforated square clockwise-counter clockwise tape on Nusselt number ratio for distilled water at q"=18599 W/m2.

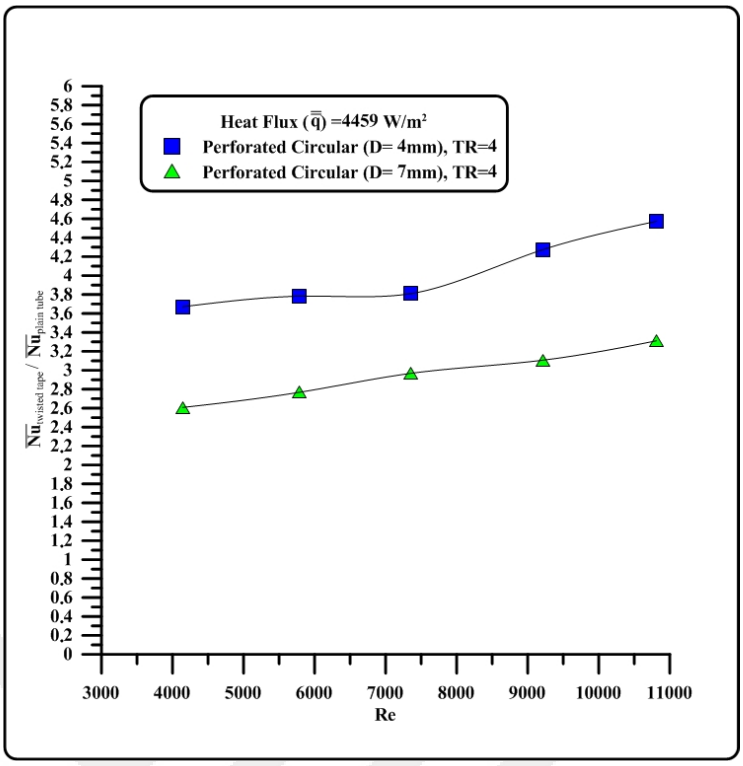


Figure 4. 41: The effect of Reynolds number and twist ratio of perforated circular clockwise-counter clockwise tape on Nusselt number ratio for distilled water at ***q***"=4459 W/m2.

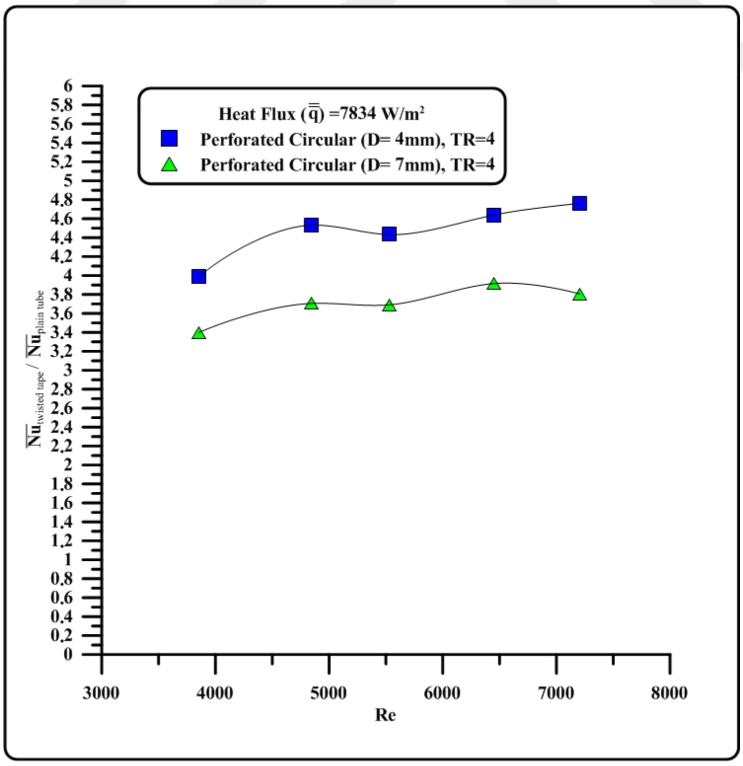


Figure 4. 42: The effect of Reynolds number and twist ratio of perforated circular clockwise-counter clockwise tape on Nusselt number ratio for distilled water at ***q***"=7834 W/m2.

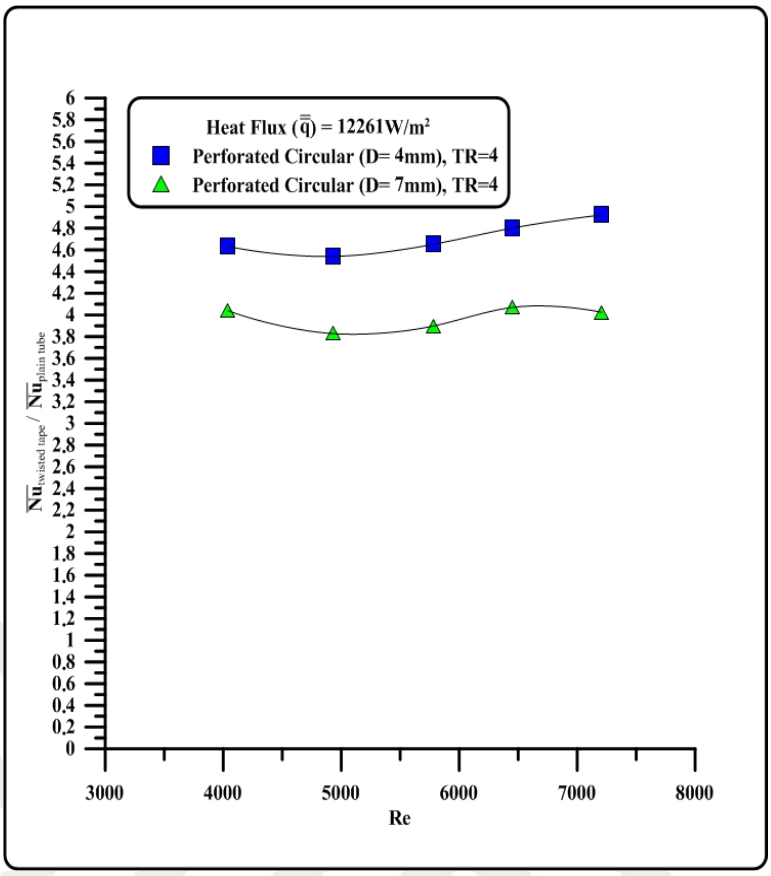


Figure 4. 43: The effect of Reynolds number and twist ratio of perforated circular clockwise-counter clockwise tape on Nusselt number ratio for distilled water at q''=12261 W/m2.

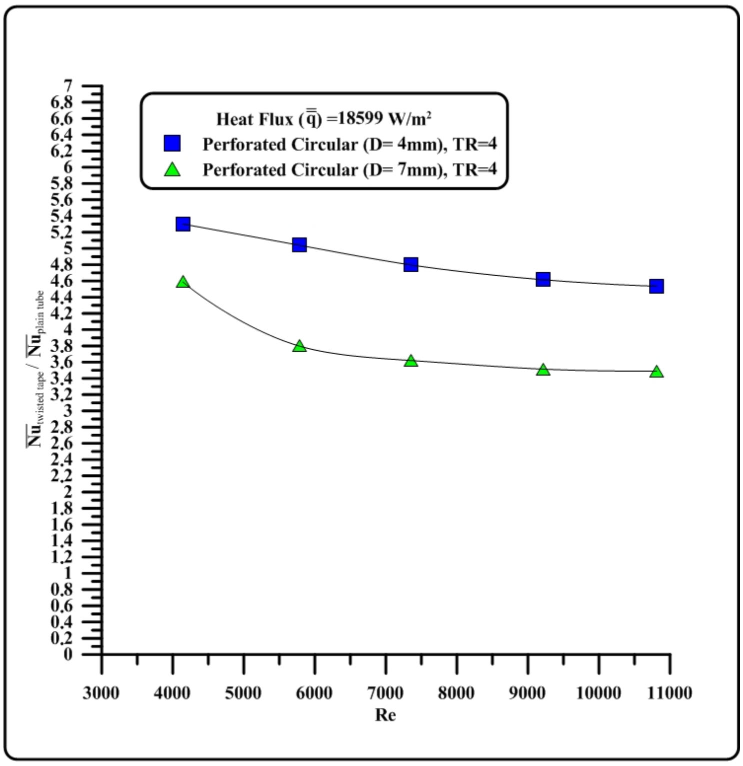


Figure 4. 44: The effect of Reynolds number and twist ratio of perforated circular clockwise-counter clockwise tape on Nusselt number ratio for distilled water at q''=18599 W/m2.

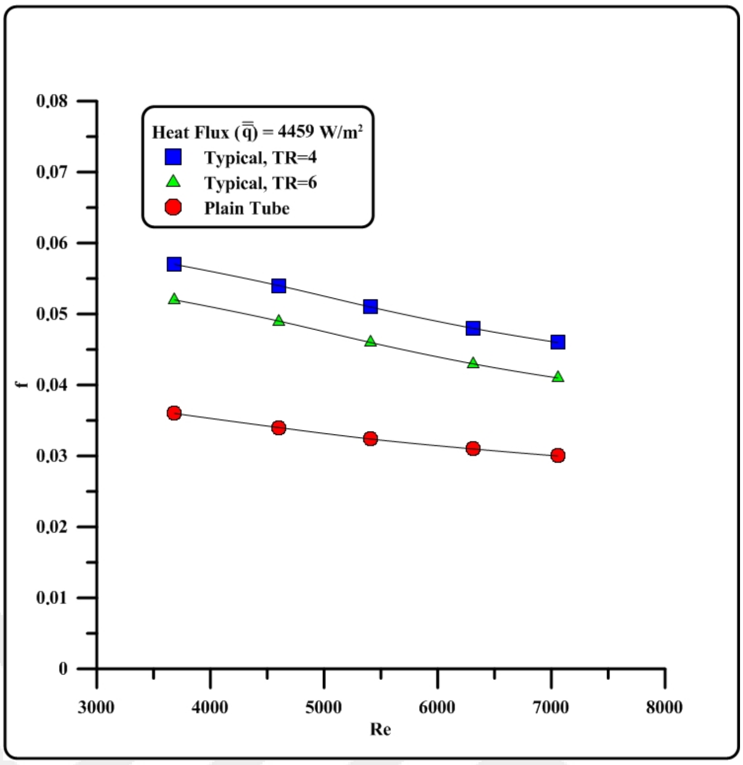


Figure 4. 45: The effect of Reynolds number and twist ratio for typical twisted tape on friction factor for distilled water at q"=4459 W/m2.

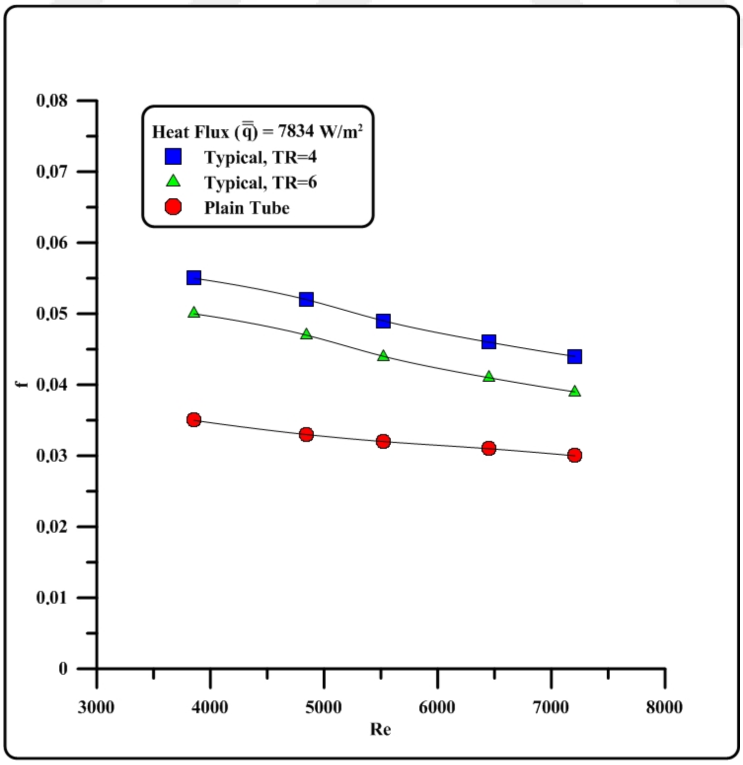


Figure 4. 46: The effect of Reynolds number and twist ratio for typical twisted tape on friction factor for distilled water at q"=7834 W/m2.

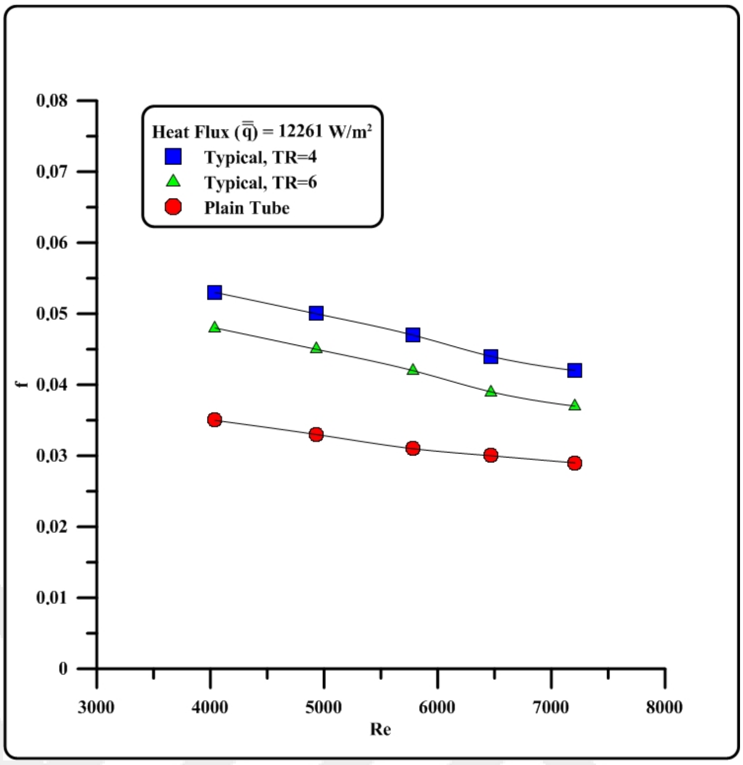


Figure 4. 47: The effect of Reynolds number and twist ratio for typical twisted tape on friction factor for distilled water at ***q***"=12261 W/m2.

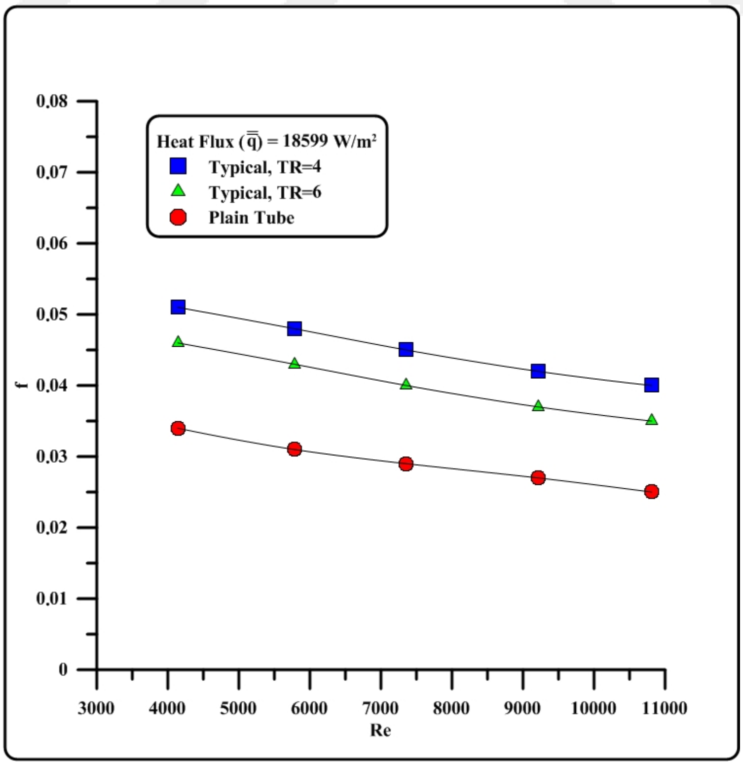


Figure 4. 48: The effect of Reynolds number and twist ratio for typical twisted tape on friction factor for distilled water at ***q***"=18599 W/m2.

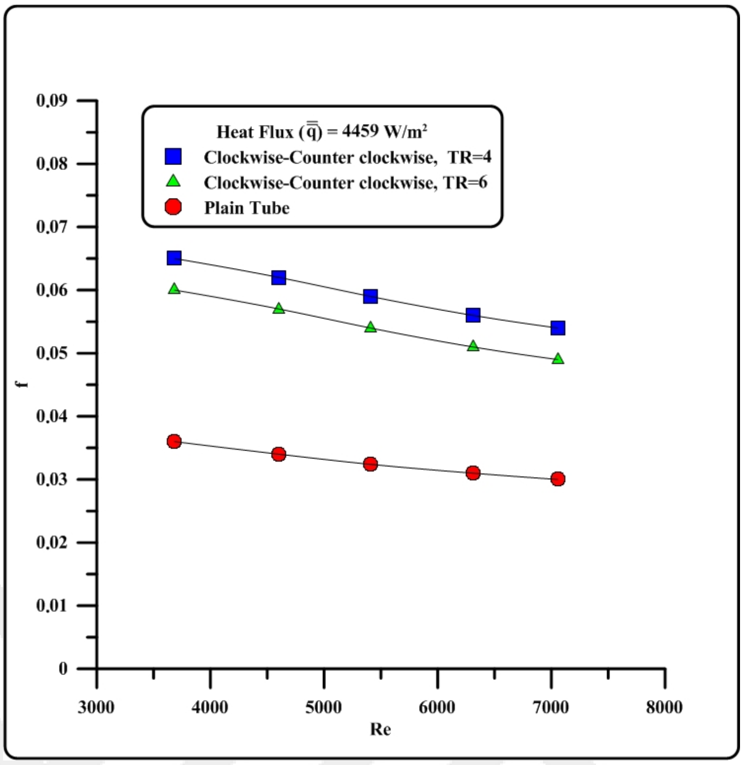


Figure 4. 49: The effect of Reynolds number and twist ratio for clockwise-counter clockwise tape on friction factor for distilled water at q"=4459 W/m2.

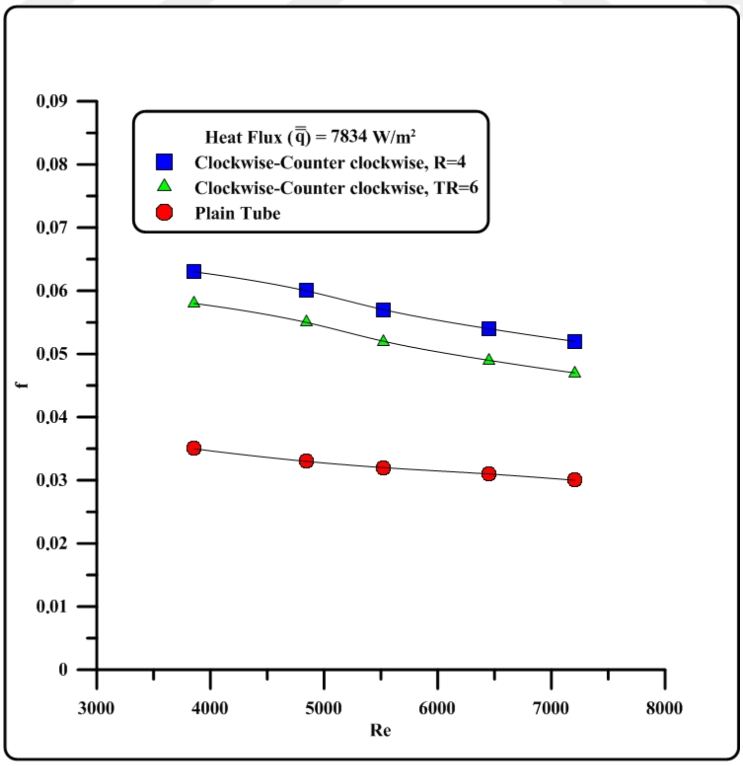


Figure 4. 50: The effect of Reynolds number and twist ratio for clockwise-counter clockwise tape on friction factor for distilled water at q"=7834 W/m2.

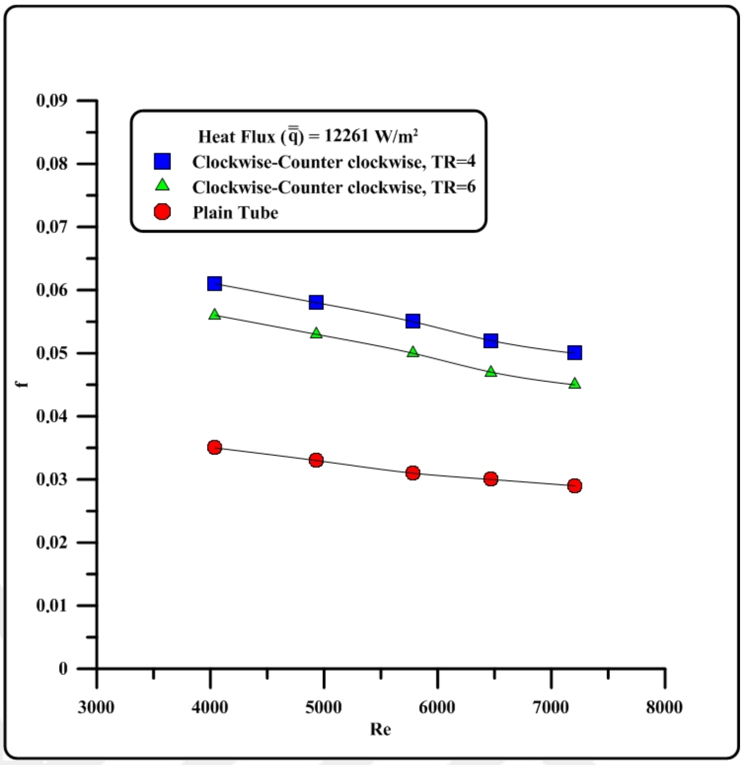


Figure 4. 51: The effect of Reynolds number and twist ratio for clockwise-counter clockwise tape on friction factor for distilled water at q''=12261 W/m2.

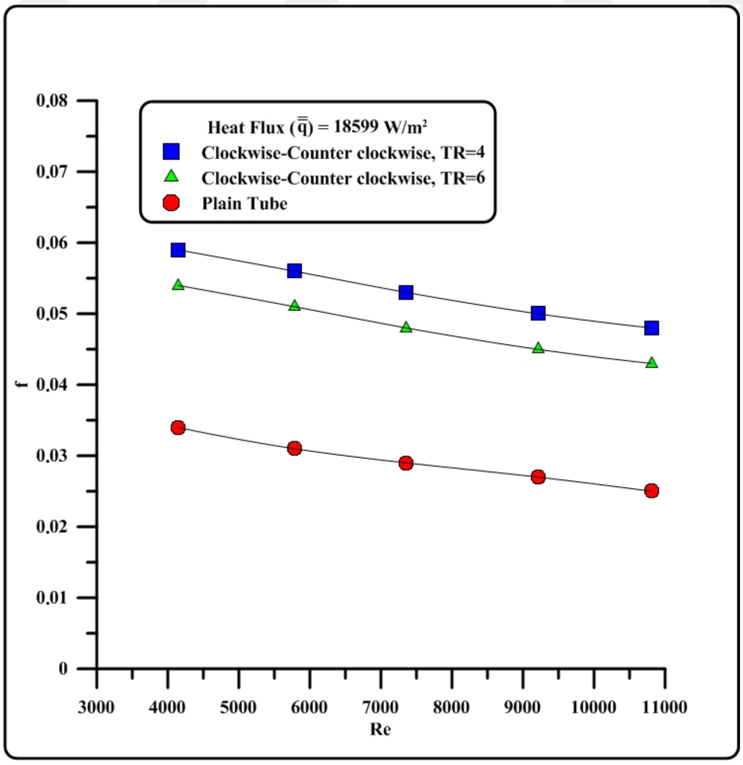


Figure 4. 52: The effect of Reynolds number and twist ratio for clockwise-counter clockwise tape on friction factor for distilled water at q''=18599 W/m2.

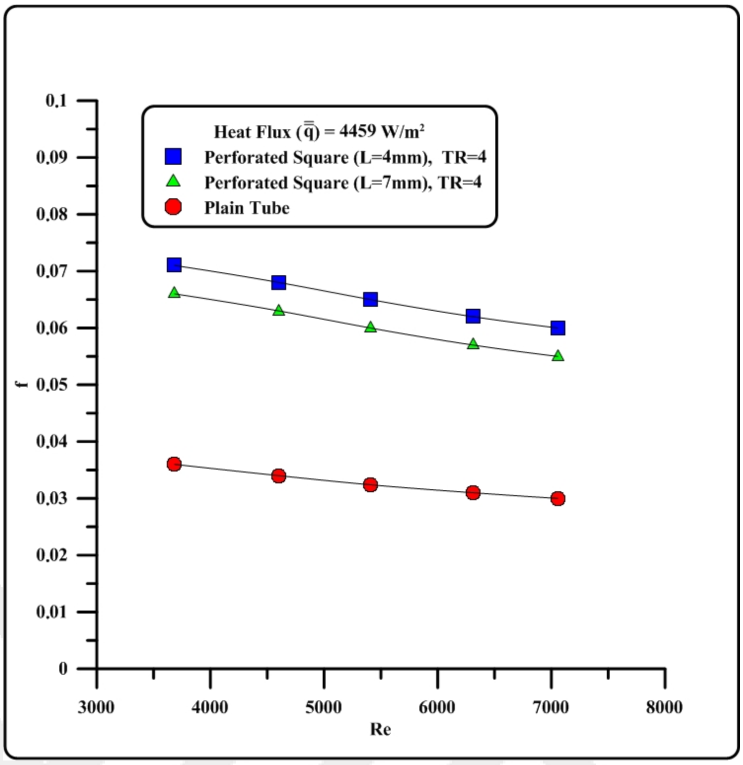


Figure 4. 53: The effect of Reynolds number and twist ratio for perforated square clockwise-counter clockwise tape on friction factor for distilled water at q''=4459 W/m2.

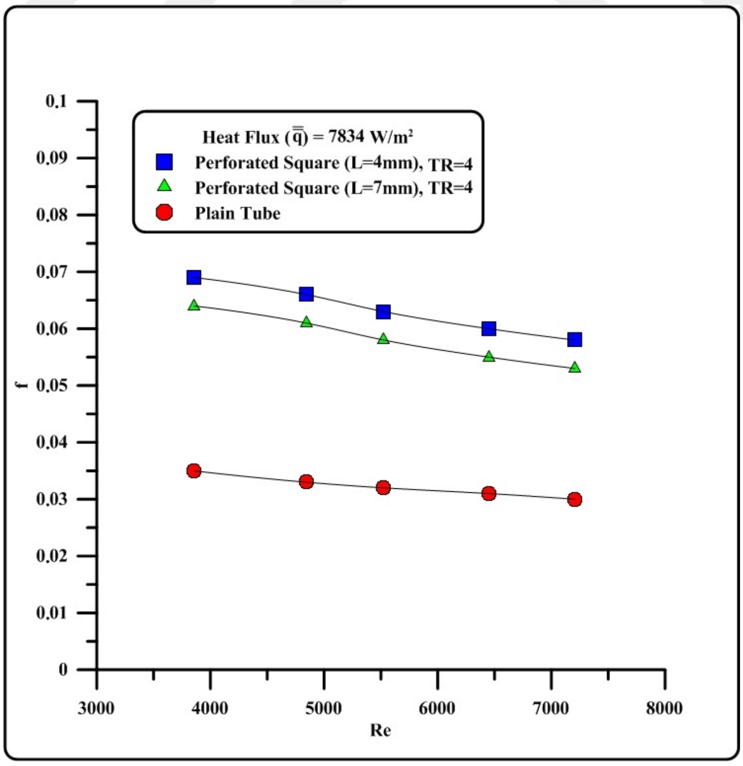


Figure 4. 54: The effect of Reynolds number and twist ratio for perforated square clockwise-counter clockwise tape on friction factor for distilled water at q''=7834 W/m2.

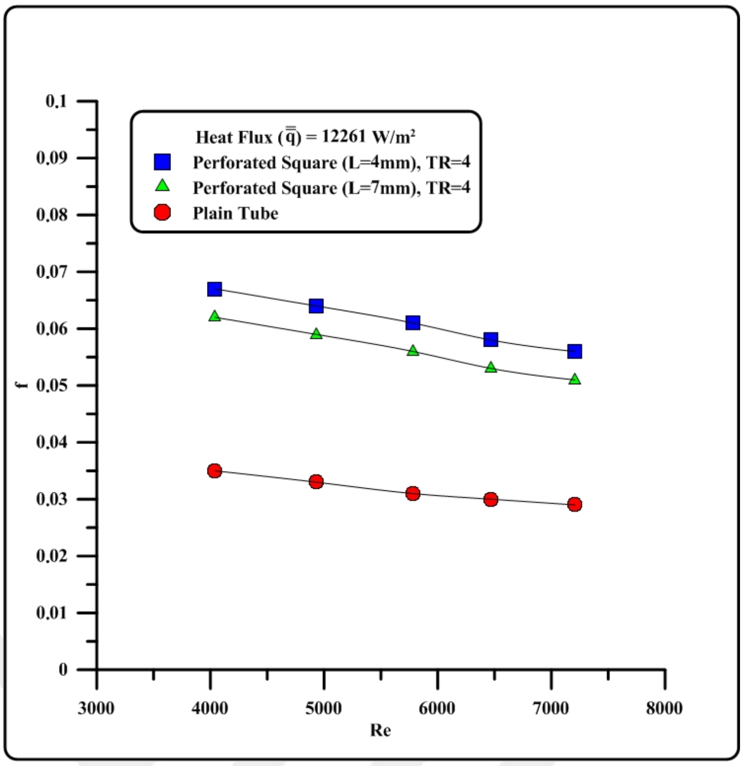


Figure 4. 55: The effect of Reynolds number and twist ratio for perforated square clockwise-counter clockwise tape on friction factor for distilled water at ***q***"=12261 W/m2.

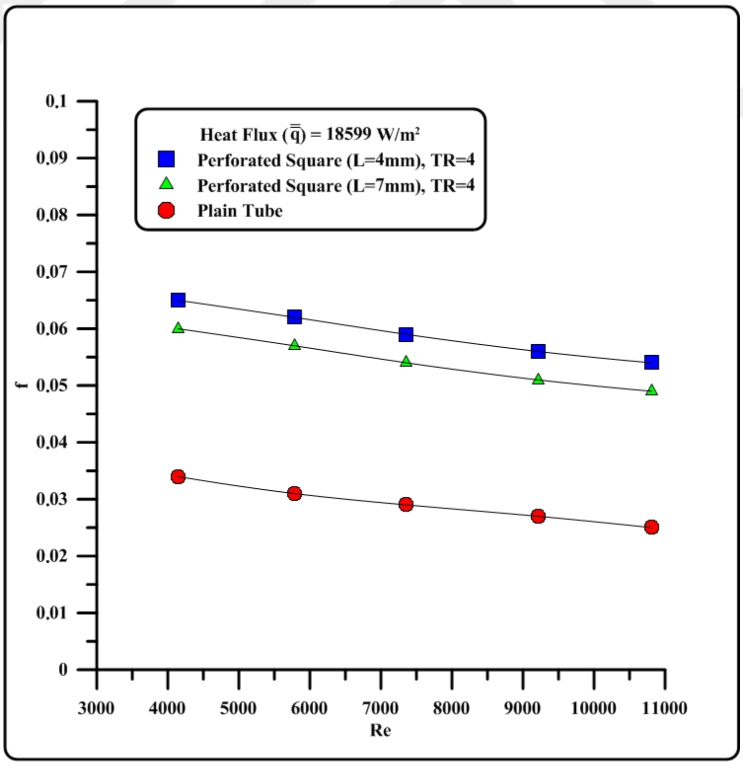


Figure 4. 56: The effect of Reynolds number and twist ratio for perforated square clockwise-counter clockwise tape on friction factor for distilled water at ***q***"=18599 W/m2.

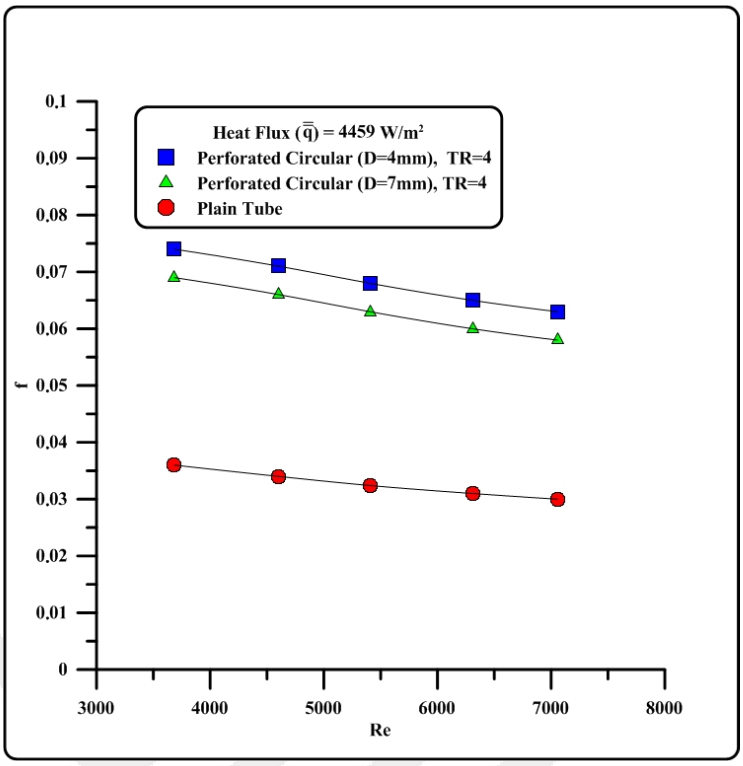


Figure 4. 57: The effect of Reynolds number and twist ratio for perforated circular clockwise-counter clockwise tape on friction factor for distilled water at q"=4459 W/m2.

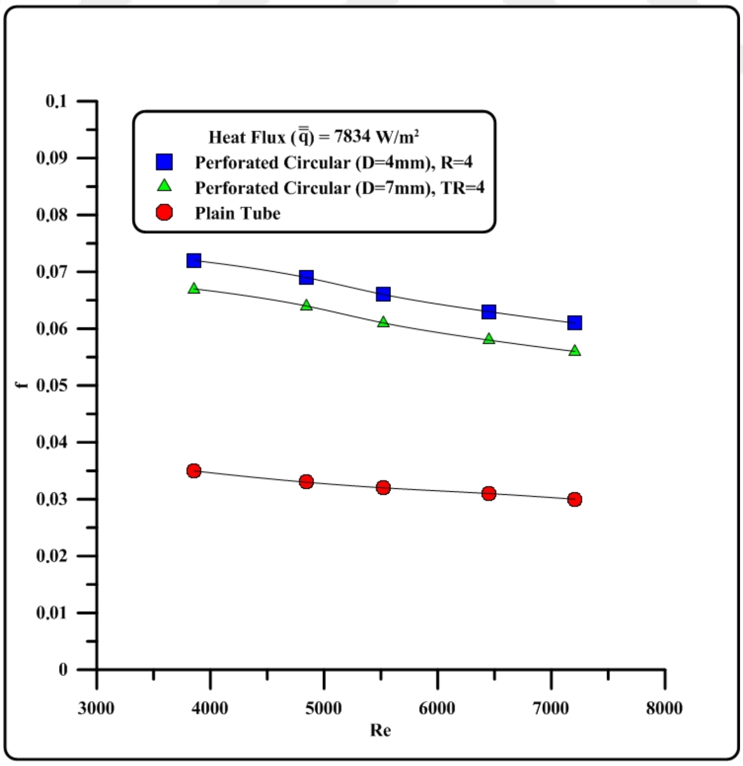


Figure 4. 58: The effect of Reynolds number and twist ratio for perforated circular clockwise-counter clockwise tape on friction factor for distilled water at q"=7834 W/m2.

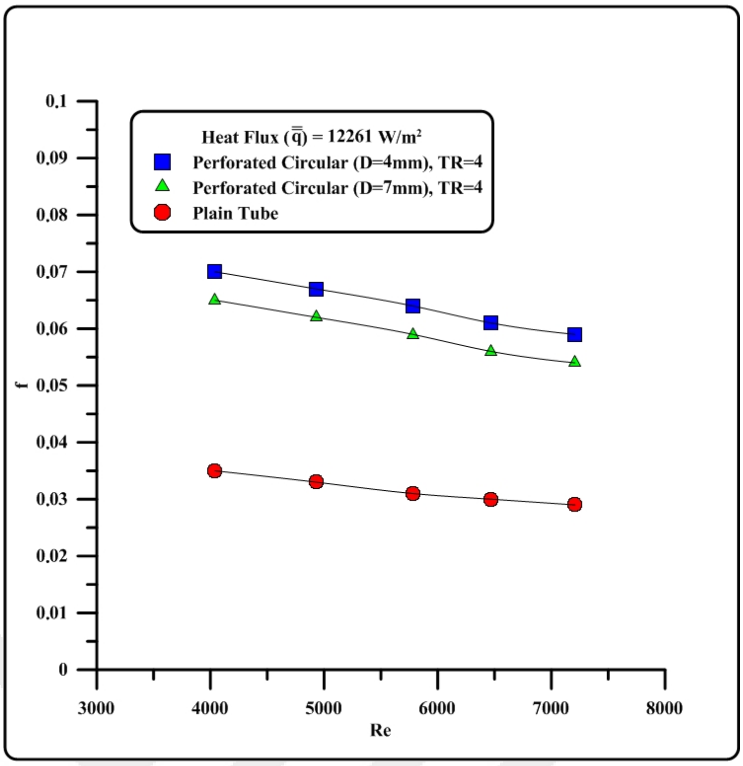


Figure 4. 59: The effect of Reynolds number and twist ratio for perforated circular clockwise-counter clockwise tape on friction factor for distilled water at q"=12261 W/m2.

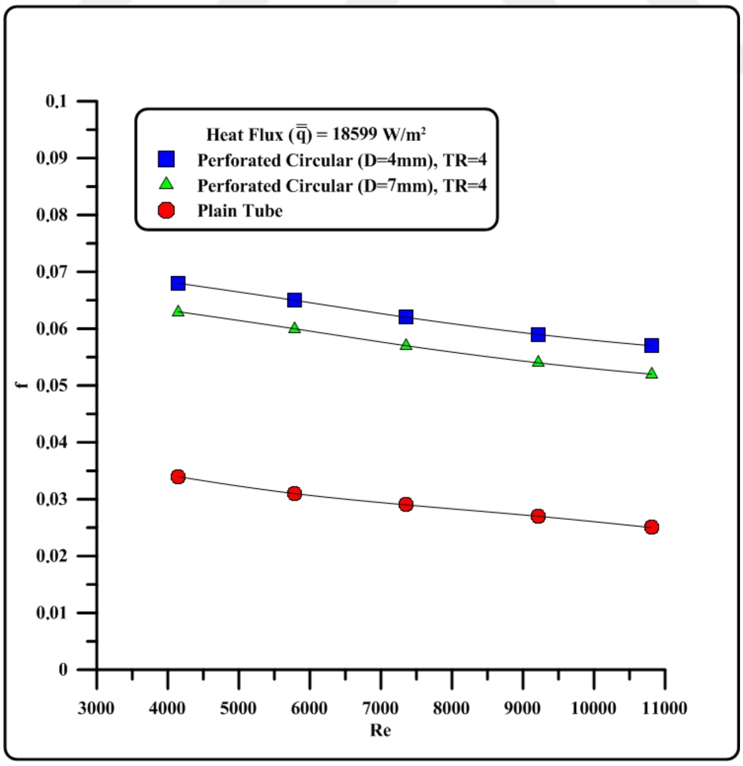


Figure 4. 60: The effect of Reynolds number and twist ratio for perforated circular clockwise-counter clockwise tape on friction factor for distilled water at q"=18599 W/m2.

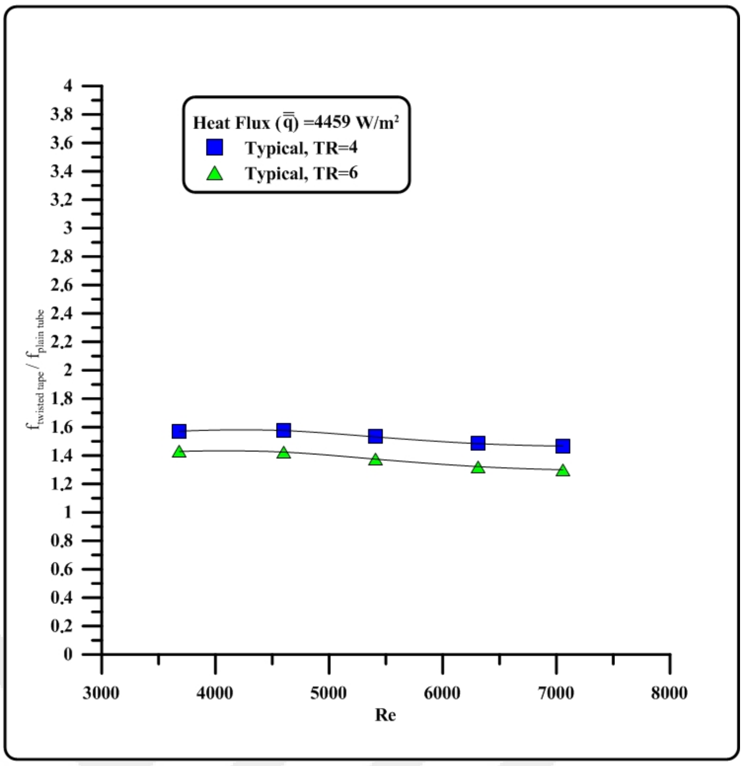


Figure 4. 61: The effect of Reynolds number and twist ratio of typical twisted tape on friction factor ratio for distilled water at ***q***"=4459 W/m2.

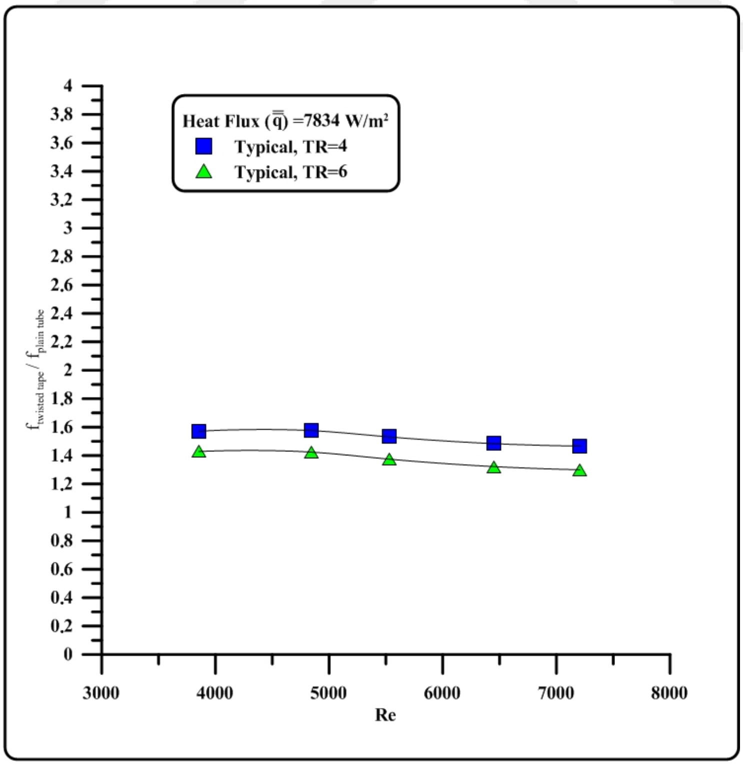


Figure 4. 62: The effect of Reynolds number and twist ratio of typical twisted tape on friction factor ratio for distilled water at ***q***"=7834 W/m2.

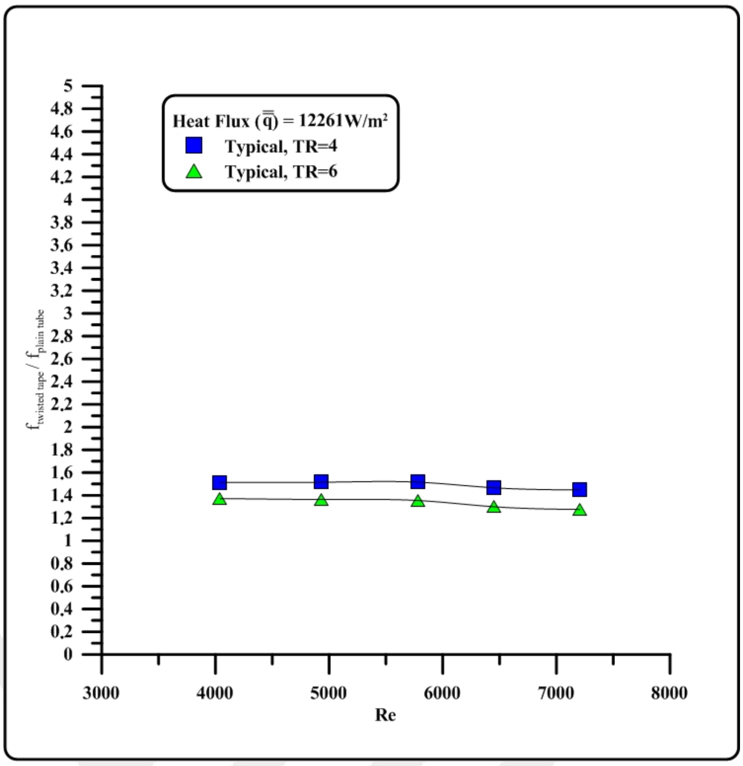


Figure 4. 63: The effect of Reynolds number and twist ratio of typical twisted tape on friction factor ratio for distilled water at q"=12261 W/m2.

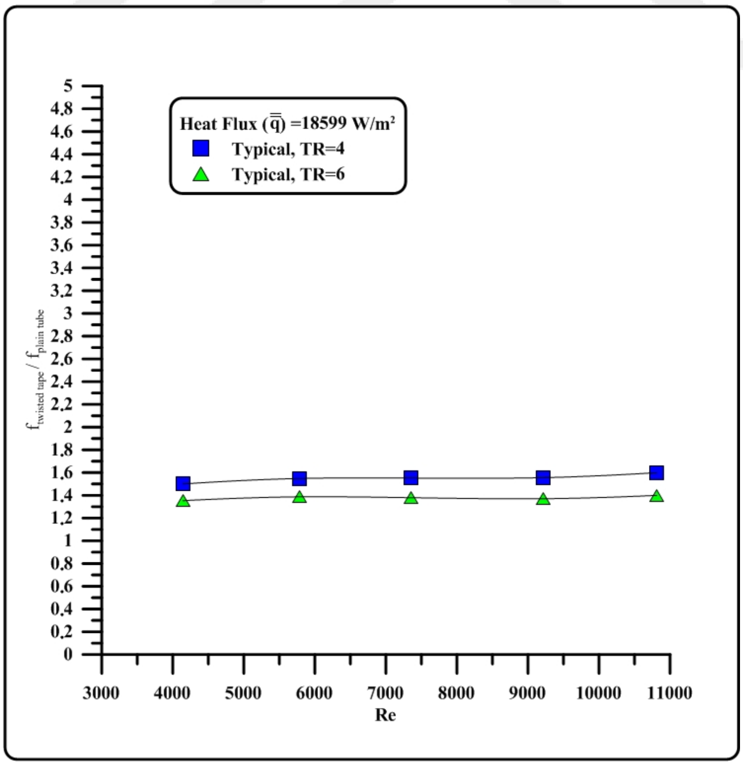


Figure 4. 64: The effect of Reynolds number and twist ratio of typical twisted tape on friction factor ratio for distilled water at q"=18599 W/m2.

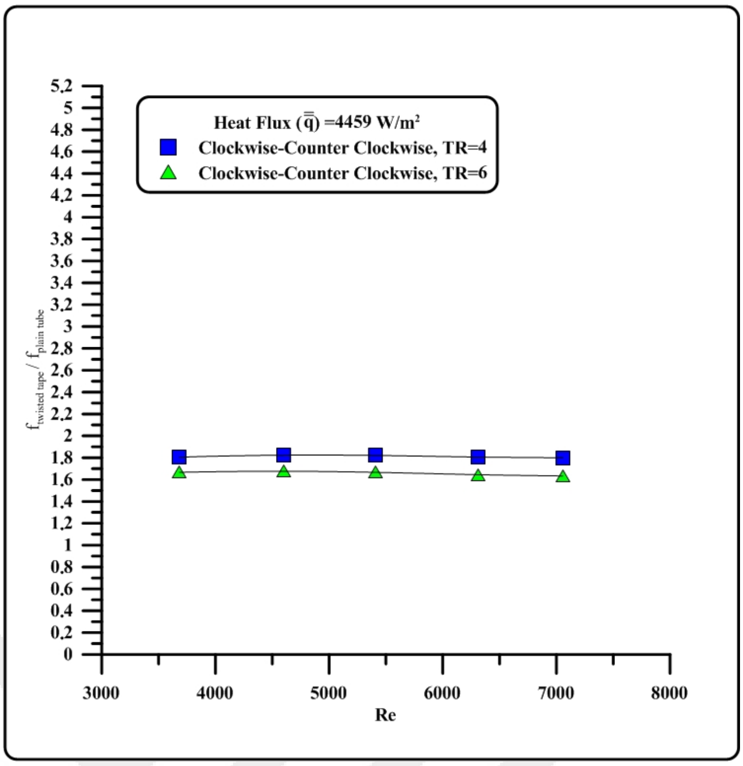


Figure 4. 65: The effect of Reynolds number and twist ratio of clockwise-counter clockwise twisted tape on friction factor ratio for distilled water at q''=4459 W/m2.

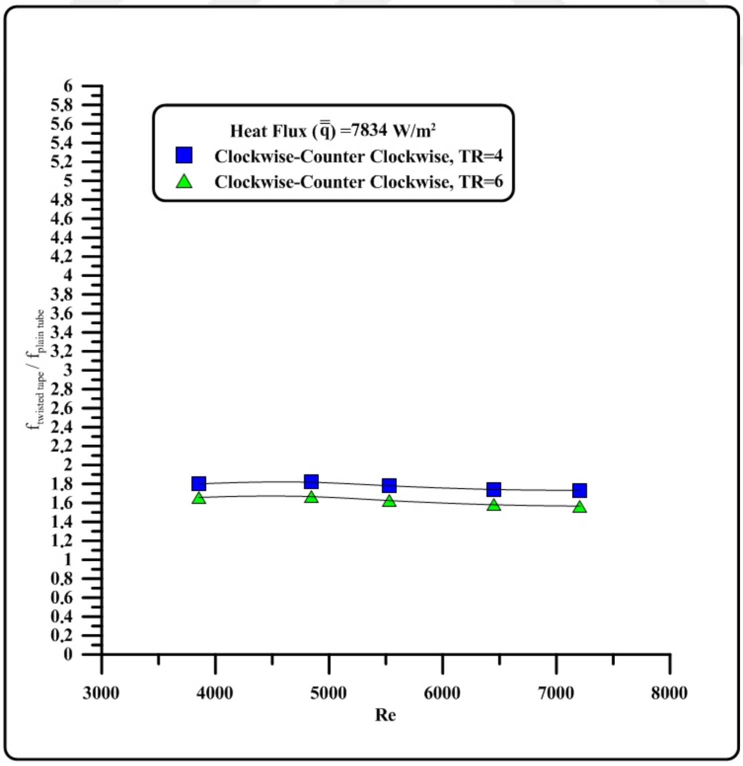


Figure 4. 66: The effect of Reynolds number and twist ratio of clockwise-counter clockwise twisted tape on friction factor ratio for distilled water at q''=7834 W/m2.

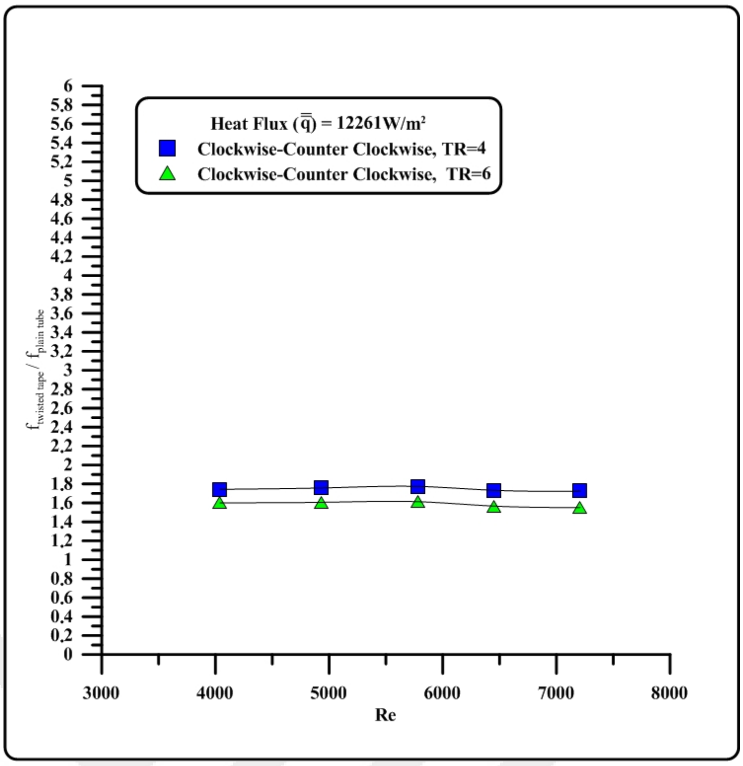


Figure 4. 67: The effect of Reynolds number and twist ratio of clockwise-counter clockwise twisted tape on friction factor ratio for distilled water at ***q***"=12261 W/m2.

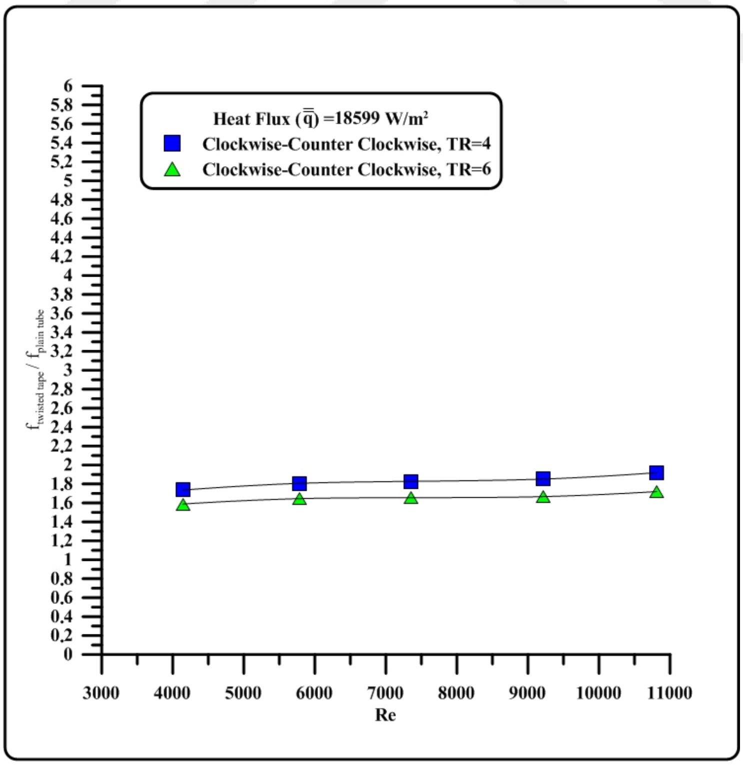


Figure 4. 68: The effect of Reynolds number and twist ratio of clockwise-counter clockwise twisted tape on friction factor ratio for distilled water at ***q***"=18599 W/m2.

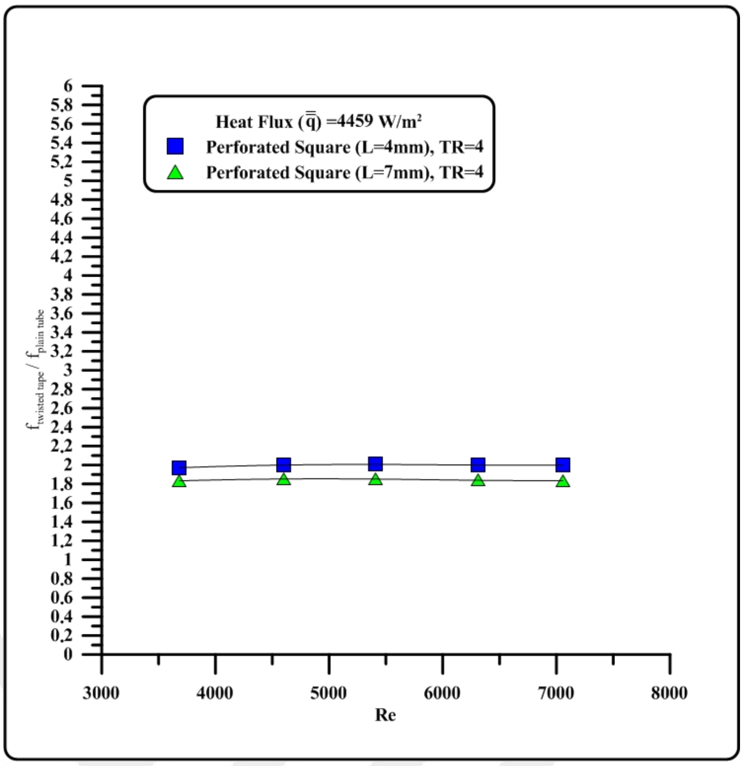


Figure 4. 69: The effect of Reynolds number and twist ratio of perforated square clockwise-counter clockwise twisted tape on friction factor ratio for distilled water at q"=4459 W/m2.

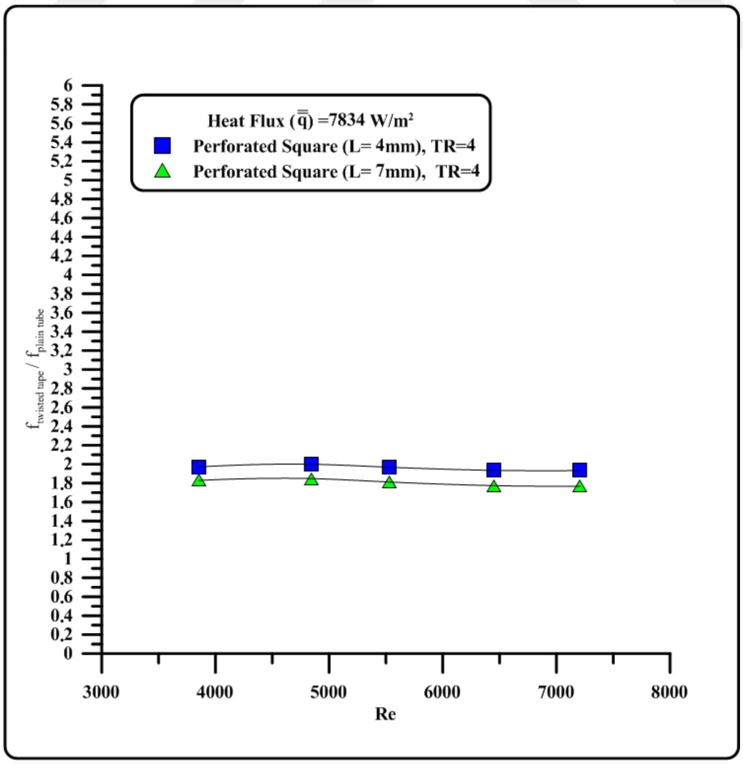


Figure 4. 70: The effect of Reynolds number and twist ratio of perforated square clockwise-counter clockwise twisted tape on friction factor ratio for distilled water at q"=7835 W/m2.

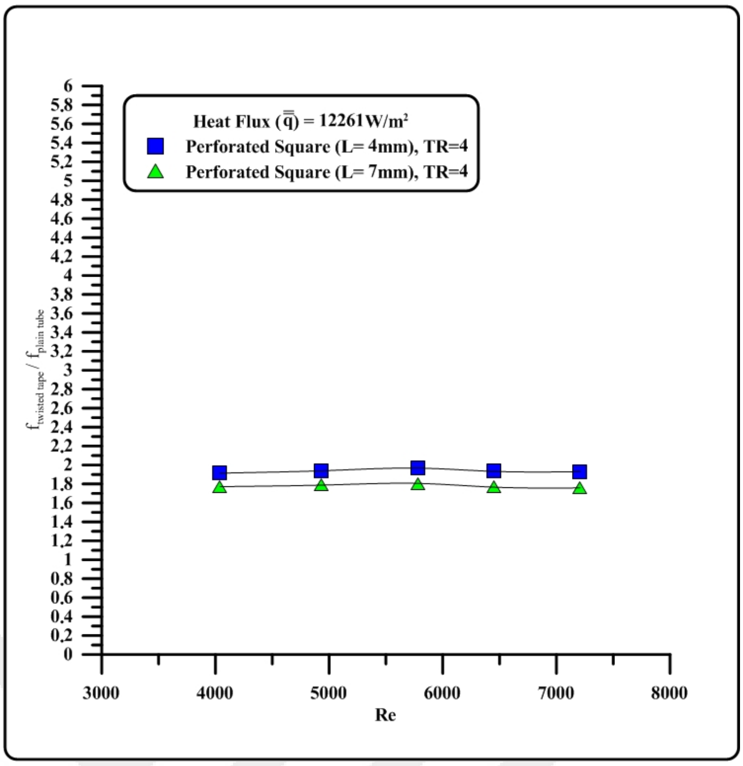


Figure 4. 71: The effect of Reynolds number and twist ratio of perforated square clockwise-counter clockwise twisted tape on friction factor ratio for distilled water at $\boldsymbol{q}$"=12261 W/m2.

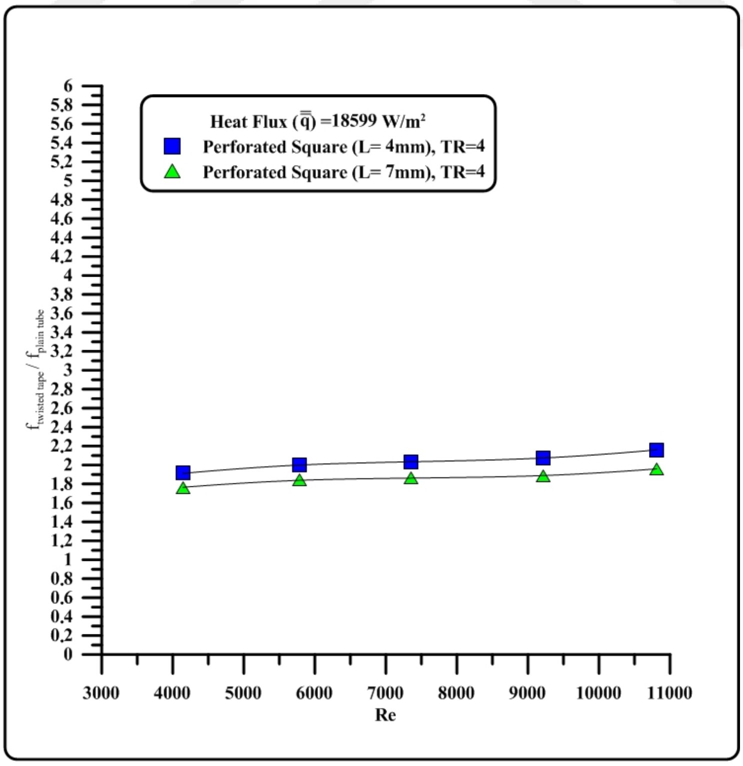


Figure 4. 72: The effect of Reynolds number and twist ratio of perforated square clockwise-counter clockwise twisted tape on friction factor ratio for distilled water at $\boldsymbol{q}$"=18599 W/m2.

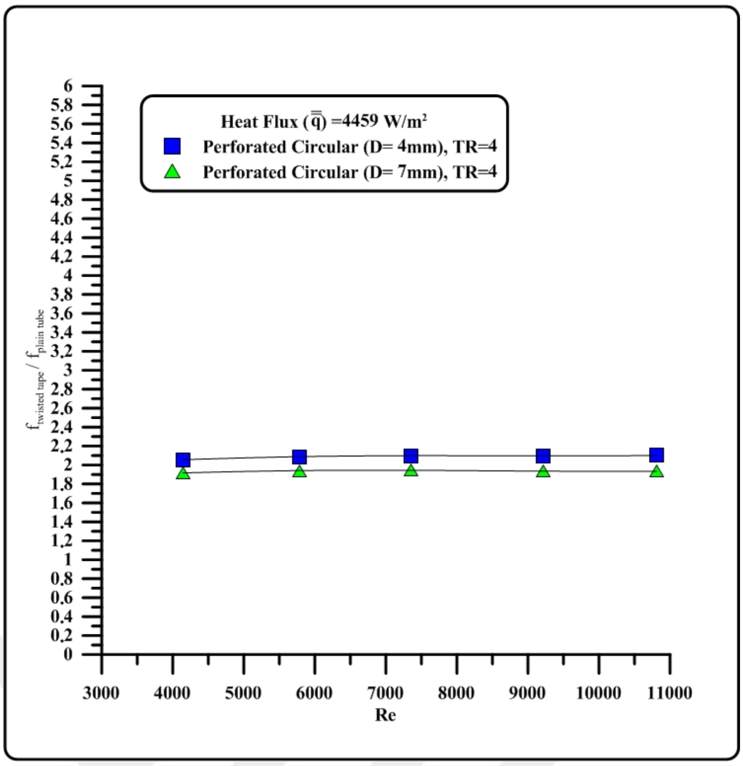


Figure 4. 73: The effect of Reynolds number and twist ratio of perforated circular clockwise-counter clockwise twisted tape on friction factor ratio for distilled water at $\boldsymbol{q}$"=4459 W/m2.

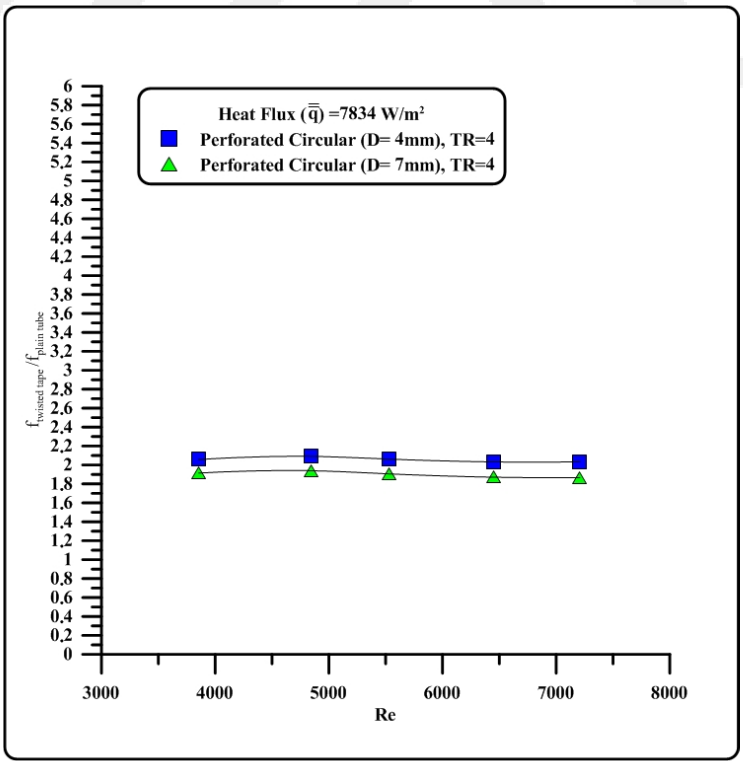


Figure 4. 74: The effect of Reynolds number and twist ratio of perforated circular clockwise-counter clockwise twisted tape on friction factor ratio for distilled water at $\boldsymbol{q}$"=7834 W/m2.

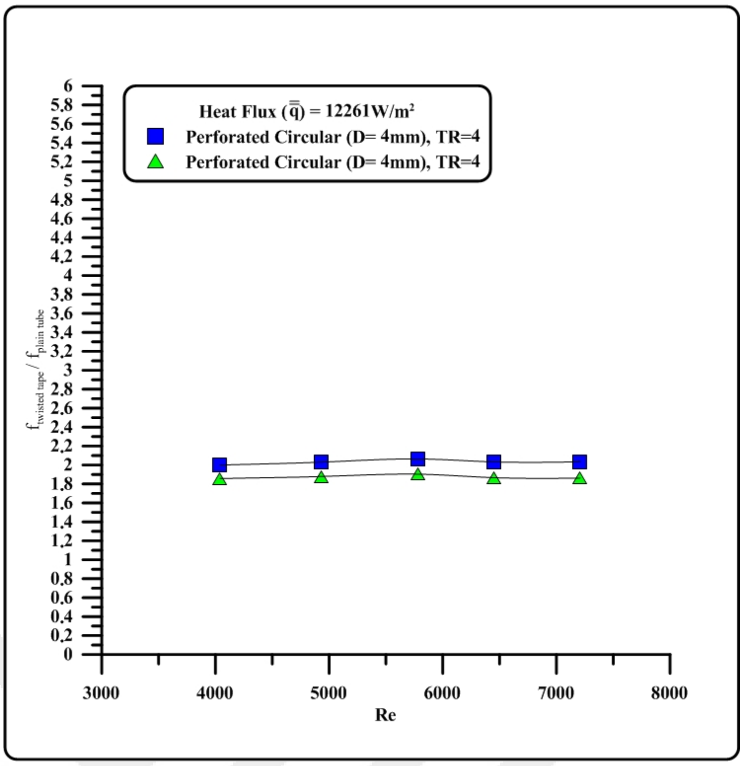


Figure 4. 75: The effect of Reynolds number and twist ratio of perforated circular clockwise-counter clockwise twisted tape on friction factor ratio for distilled water at q"=12261 W/m2.

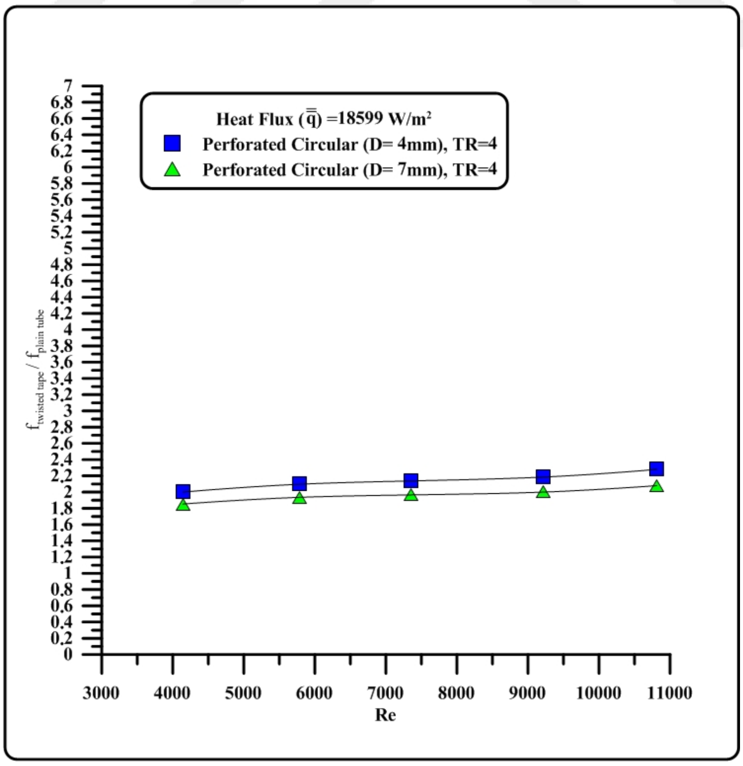


Figure 4. 76: The effect of Reynolds number and twist ratio of perforated circular clockwise-counter clockwise twisted tape on friction factor ratio for distilled water at q"=18599 W/m2.

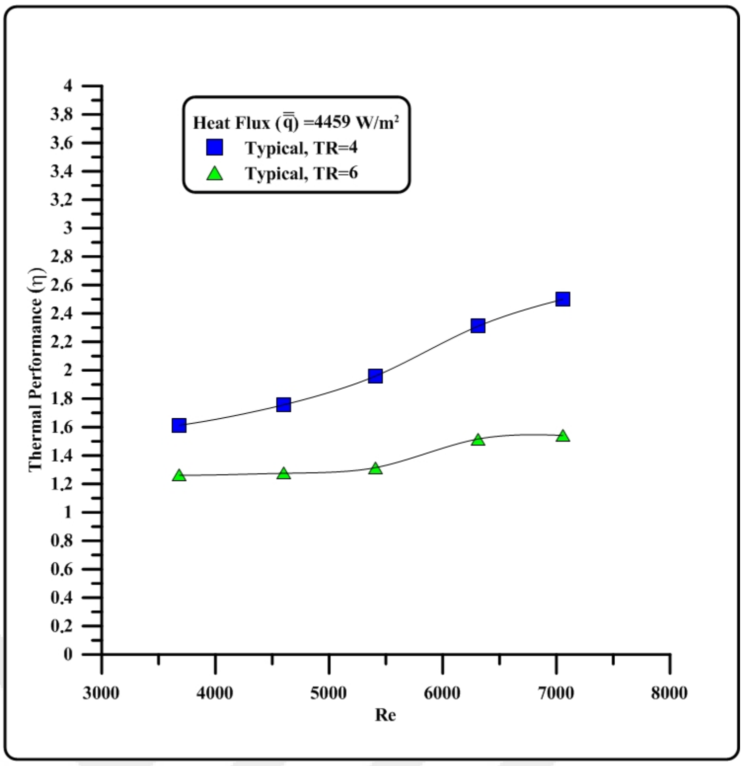


Figure 4. 77: The effect of Reynolds number and twist ratio of typical twisted tape on thermal performance factor (η) for distilled water at q''=4459 W/m2.

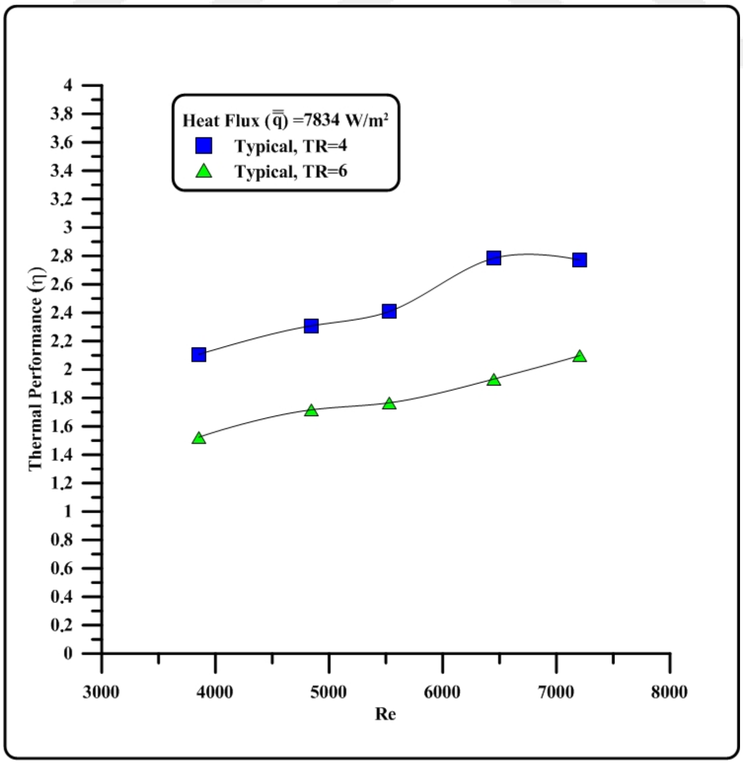


Figure 4. 78: The effect of Reynolds number and twist ratio of typical twisted tape on thermal performance factor (η) for distilled water at q''=7834 W/m2.

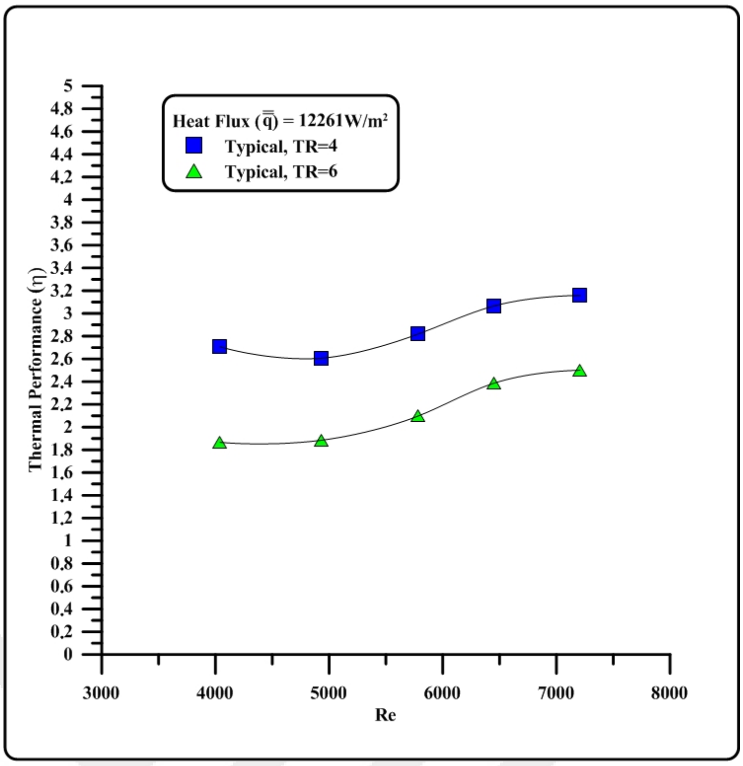


Figure 4. 79: The effect of Reynolds number and twist ratio of typical twisted tape on thermal performance factor (η) for distilled water at q''=12261 W/m2.

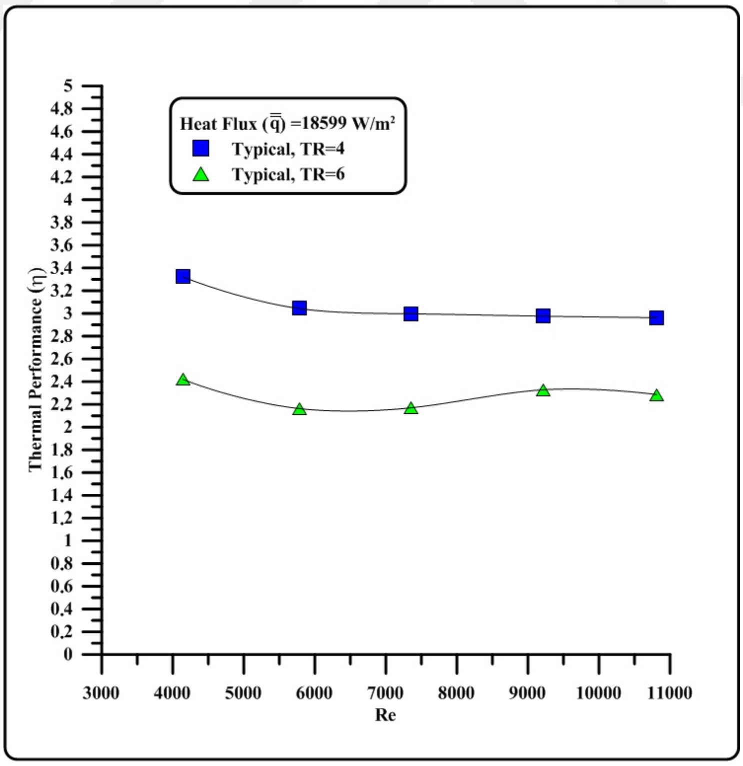


Figure 4. 80: The effect of Reynolds number and twist ratio of typical twisted tape on thermal performance factor (η) for distilled water at q''=18599 W/m2.

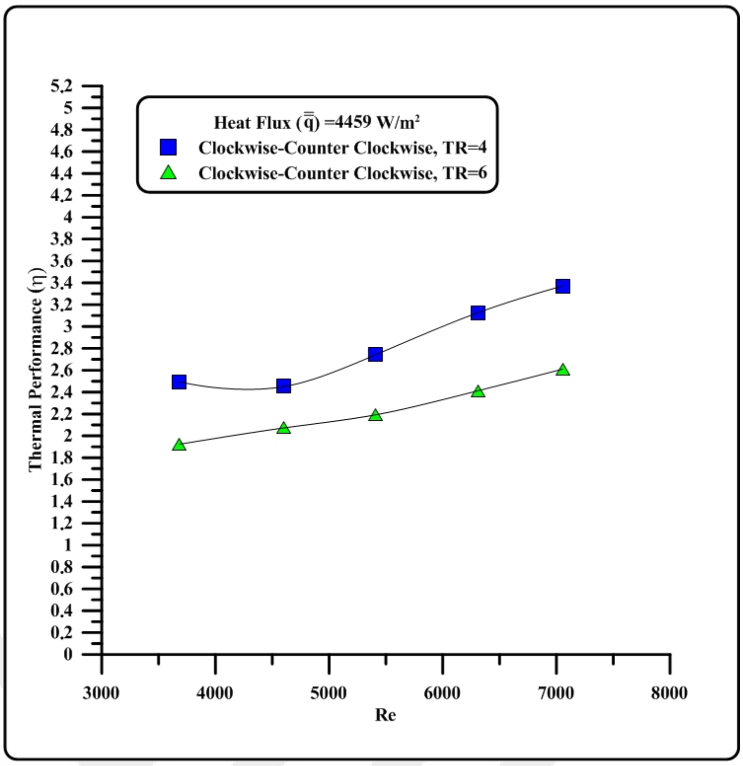


Figure 4. 81: The effect of Reynolds number and twist ratio of clockwise-counter clockwise twisted tape on thermal performance factor (η) for distilled water at q''=4459 W/m2.

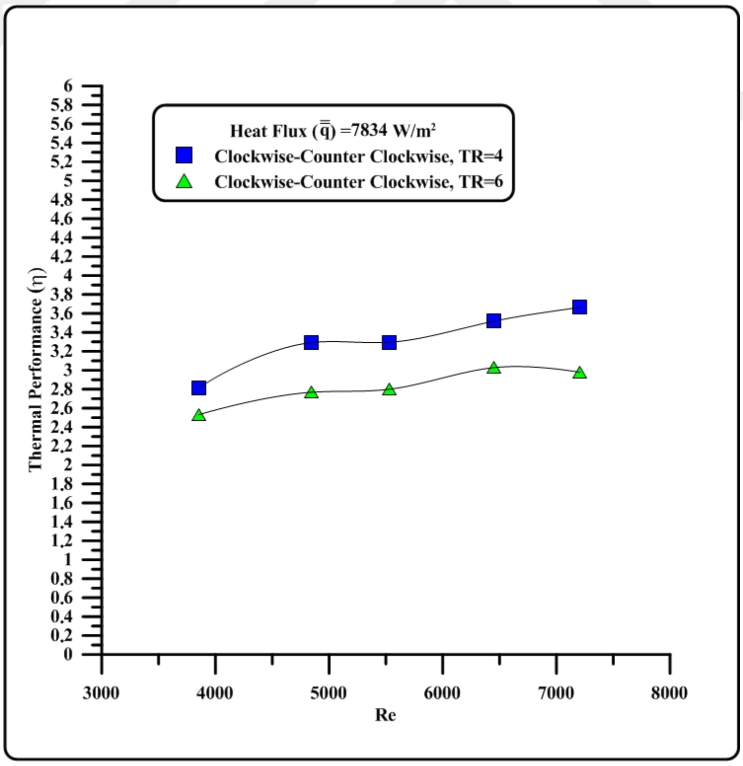


Figure 4. 82: The effect of Reynolds number and twist ratio of clockwise-counter clockwise twisted tape on thermal performance factor (η) for distilled water at q''=7834 W/m2.

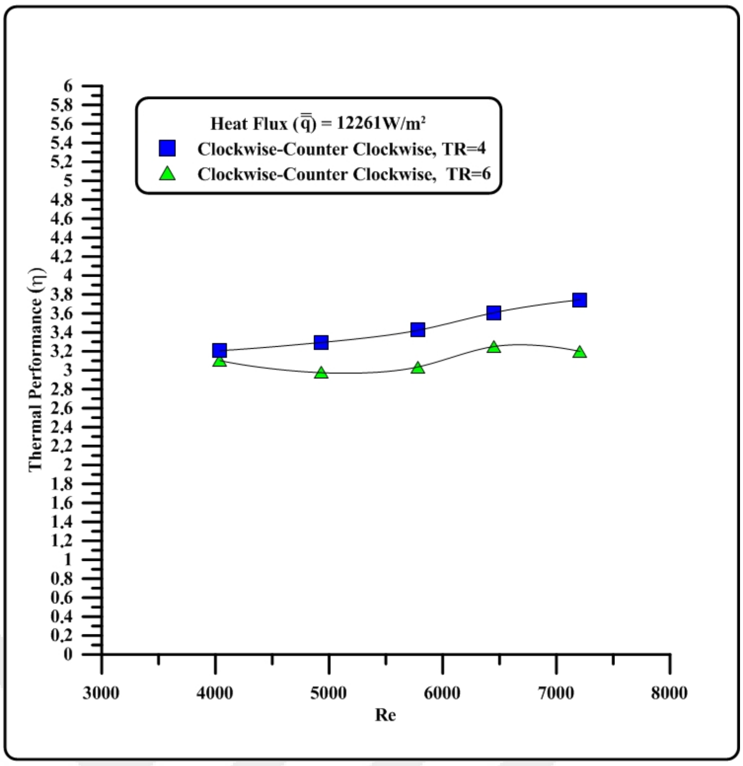


Figure 4. 83: The effect of Reynolds number and twist ratio of clockwise-counter clockwise twisted tape on thermal performance factor (η) for distilled water at q''=12261 W/m2.

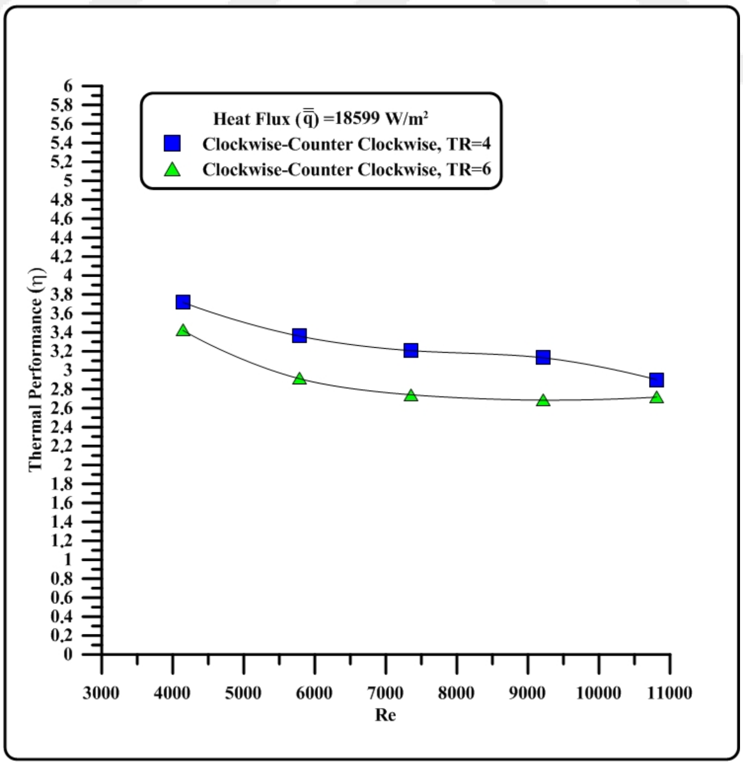


Figure 4. 84: The effect of Reynolds number and twist ratio of clockwise-counter clockwise twisted tape on thermal performance factor (η) for distilled water at q''=18599 W/m2.

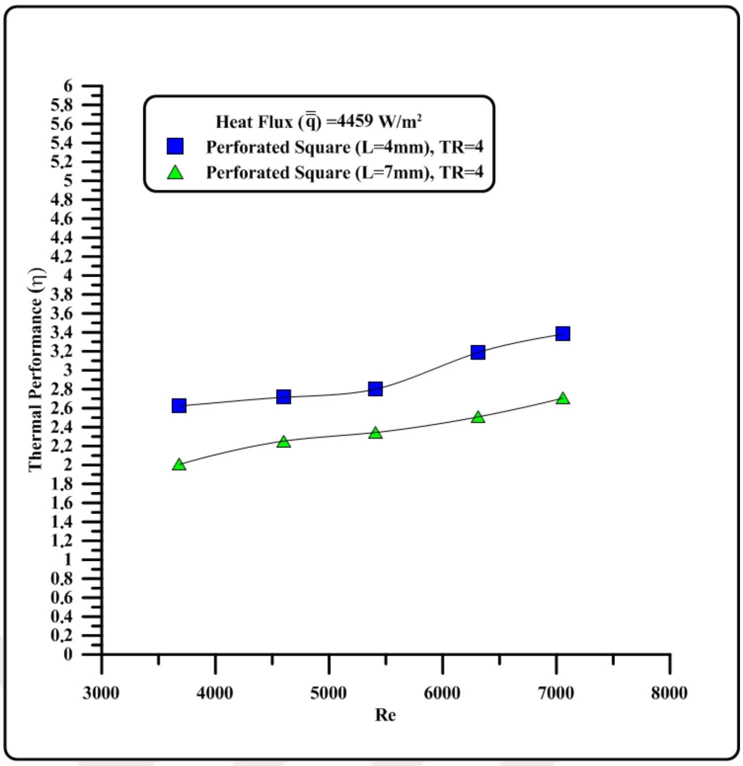


Figure 4. 85: The effect of Reynolds number and twist ratio of perforated square clockwise-counter clockwise twisted tape on thermal performance factor (η) for distilled water at ***q***"=4459 W/m2.

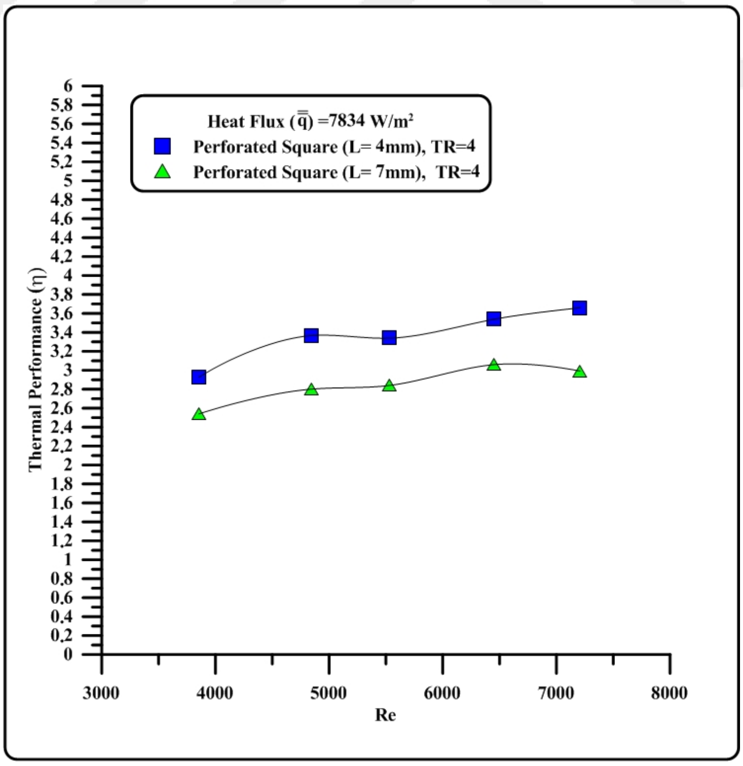


Figure 4. 86: The effect of Reynolds number and twist ratio of perforated square clockwise-counter clockwise twisted tape on thermal performance factor (η) for distilled water at ***q***"=7834 W/m2.

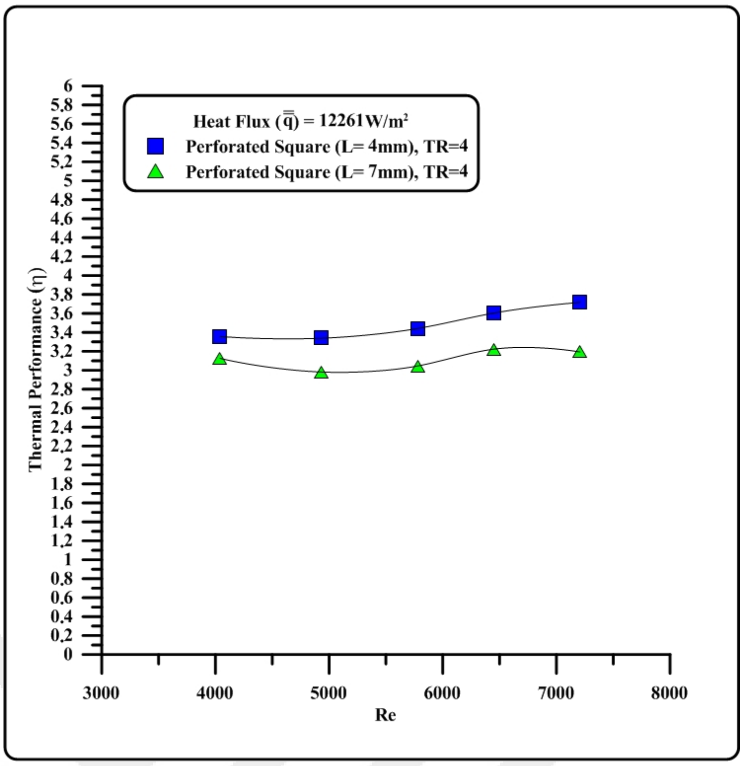


Figure 4. 87: The effect of Reynolds number and twist ratio of perforated square clockwise-counter clockwise twisted tape on thermal performance factor (η) for distilled water at q"=12261W/m2.

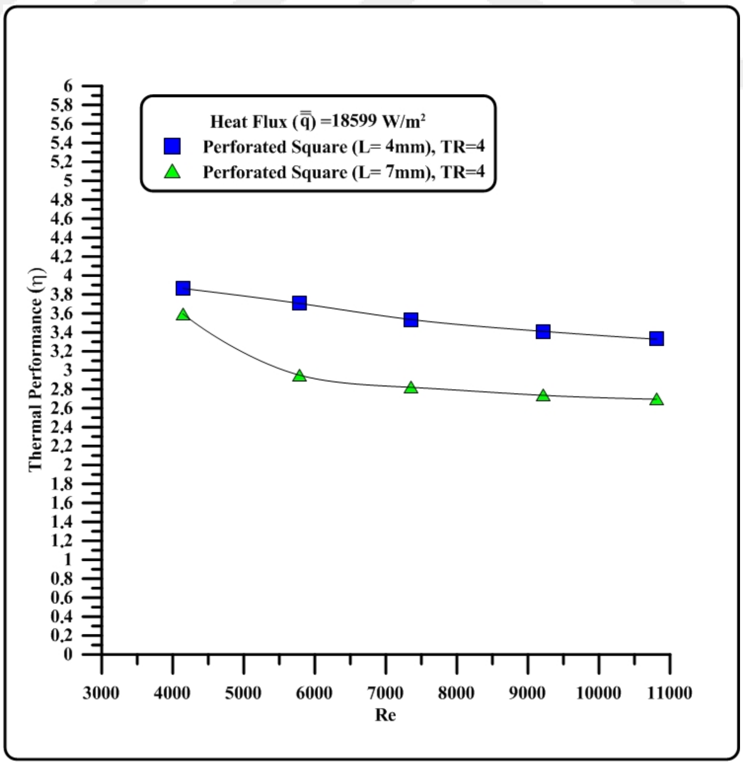


Figure 4. 88: The effect of Reynolds number and twist ratio of perforated square clockwise-counter clockwise twisted tape on thermal performance factor (η) for distilled water at q"=18599W/m2.

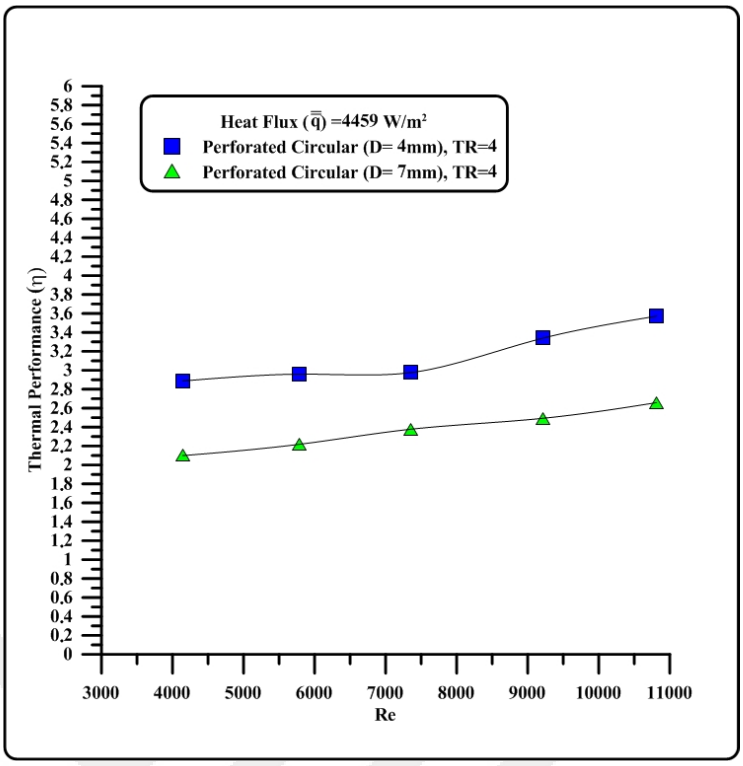


Figure 4. 89: The effect of Reynolds number and twist ratio of perforated circular clockwise-counter clockwise twisted tape on thermal performance factor (η) for distilled water at ***q***''=4459W/m2.

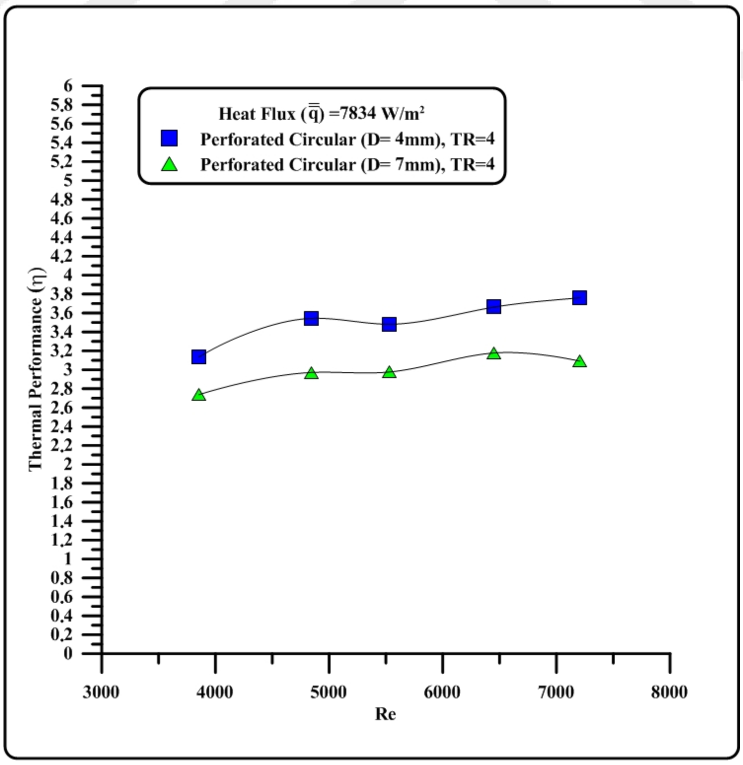


Figure 4. 90: The effect of Reynolds number and twist ratio of perforated circular clockwise-counter clockwise twisted tape on thermal performance factor (η) for distilled water at ***q***''=7834 W/m2.

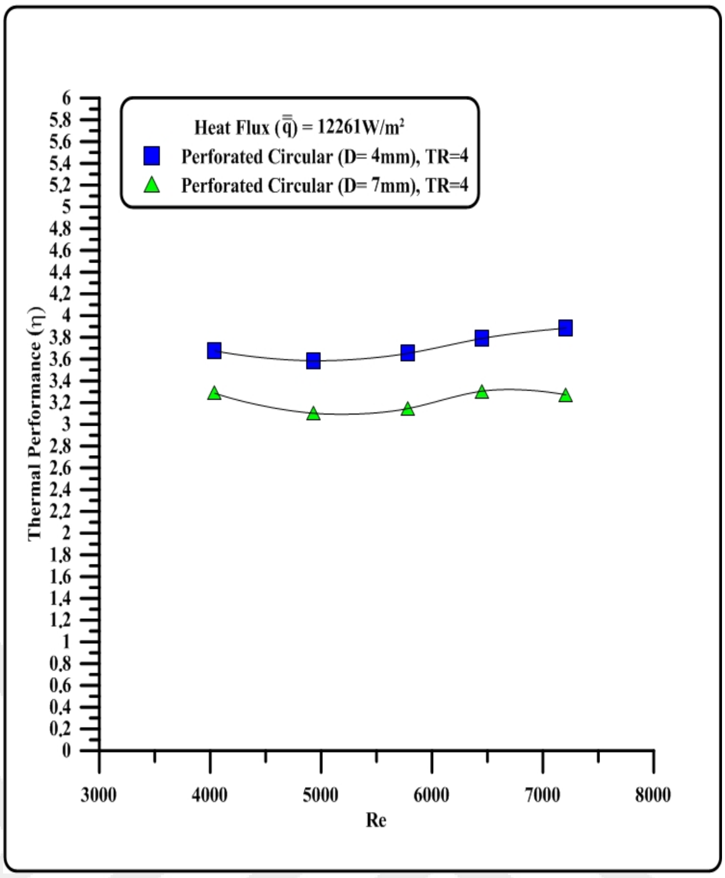


Figure 4. 91: The effect of Reynolds number and twist ratio of perforated circular clockwise-counter clockwise twisted tape on thermal performance factor (η) for distilled water at q''=12261 W/m2.

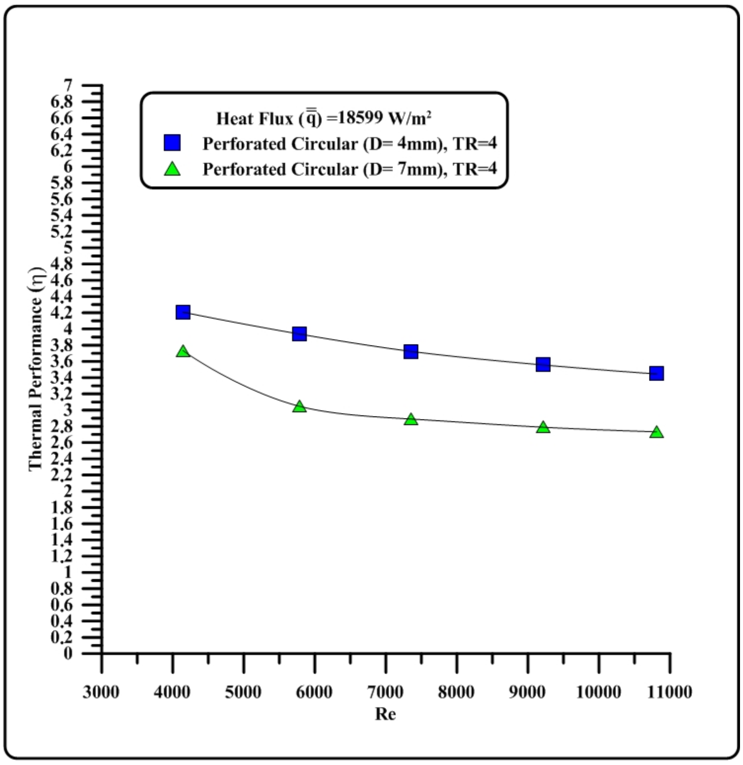


Figure 4. 92: The effect of Reynolds number and twist ratio of perforated circular clockwise-counter clockwise twisted tape on thermal performance factor (η) for distilled water at q''=18599 W/m2.

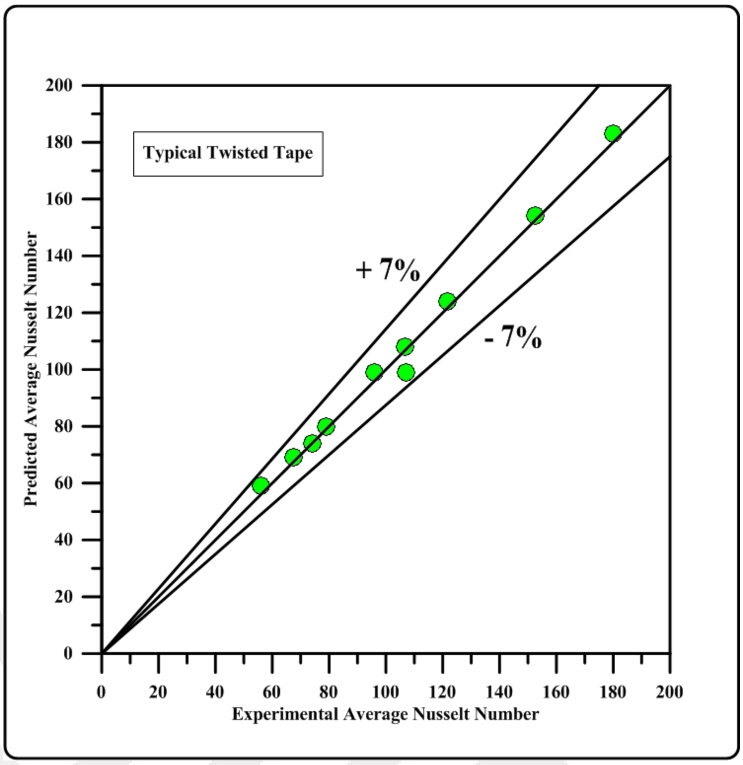


Figure 4. 93: Comparison of experimental with predicted average Nusselt number for distilled water in a tube with typical twisted tape.

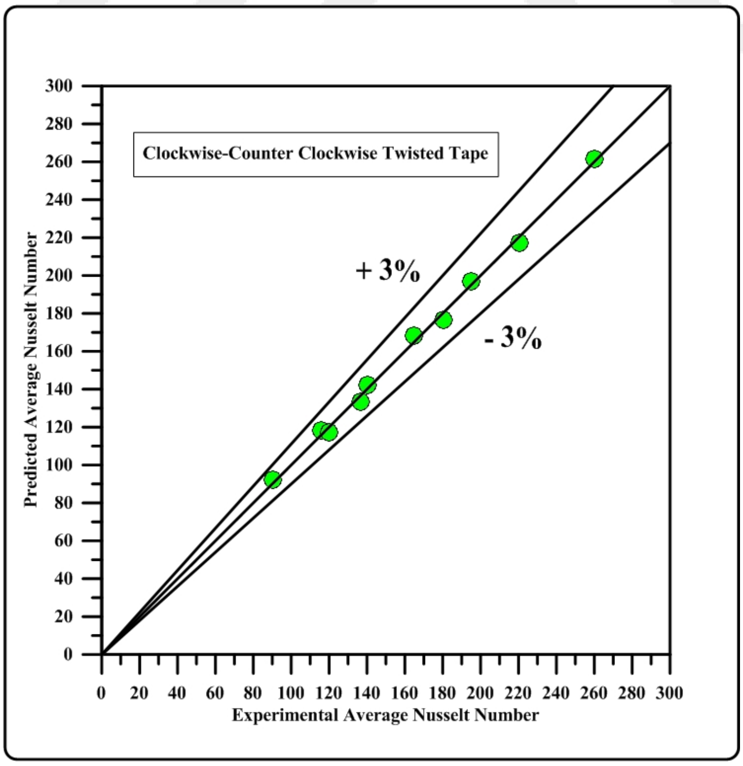


Figure 4. 94: Comparison of experimental with predicted average Nusselt average Nusselt number for distilled water in a tube with clockwise-counter clockwise twisted tape.

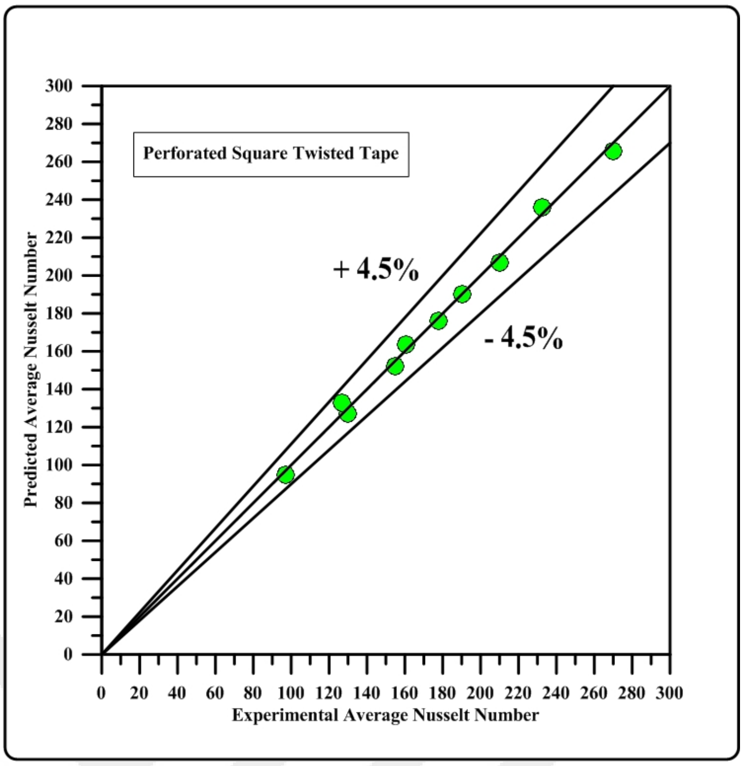


Figure 4. 95: Comparison of experimental with predicted average Nusselt average Nusselt number for distilled water in a tube with perforated square clockwise-counter clockwise twisted tape.

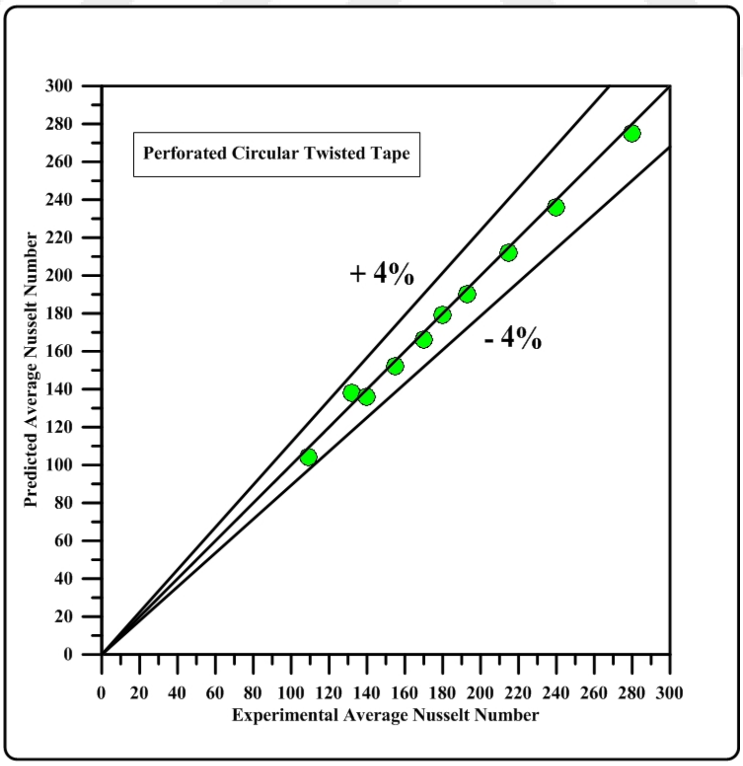


Figure 4. 96: Comparison of experimental with predicted average Nusselt average Nusselt number for distilled water in a tube with perforated circular clockwise-counter clockwise twisted tape.

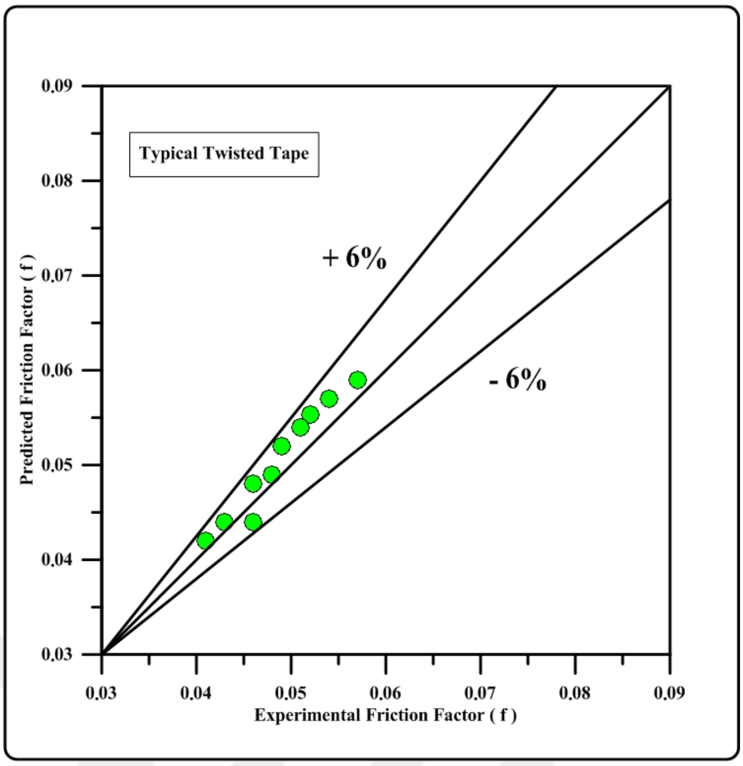


Figure 4. 97: Comparison of experimental with predicted friction factor for distilled water in a tube with typical twisted tape.

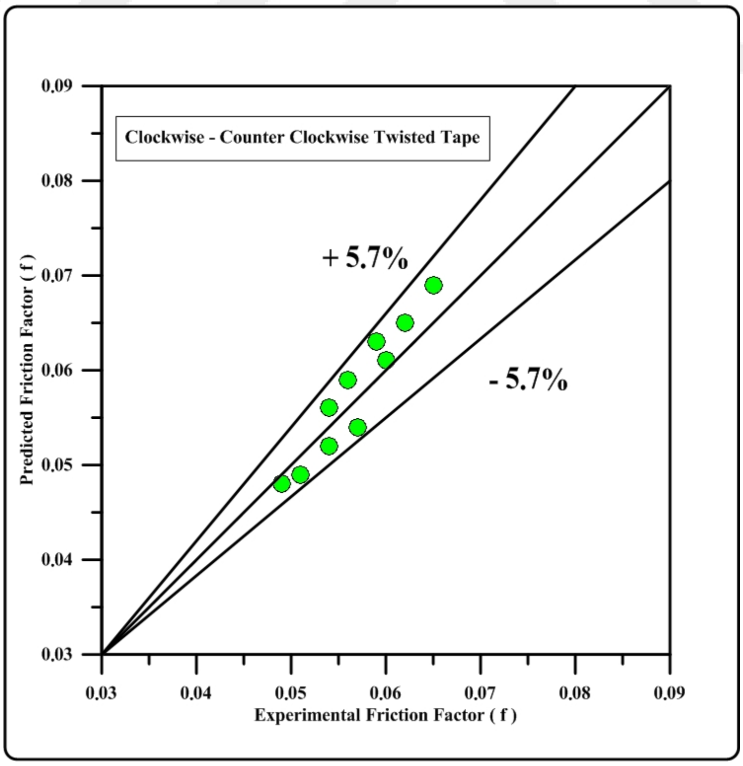


Figure 4. 98: Comparison of experimental with predicted friction factor for distilled water in a tube with clockwise-counter clockwise twisted tape.

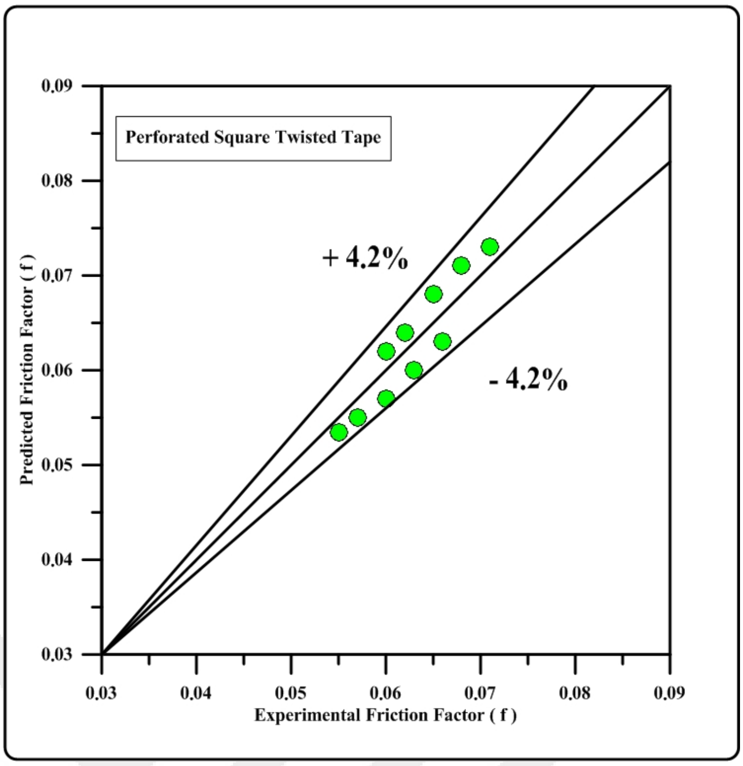


Figure 4. 99: Comparison of experimental with predicted friction factor for distilled water in a tube with perforated square clockwise-counter clockwise twisted tape.

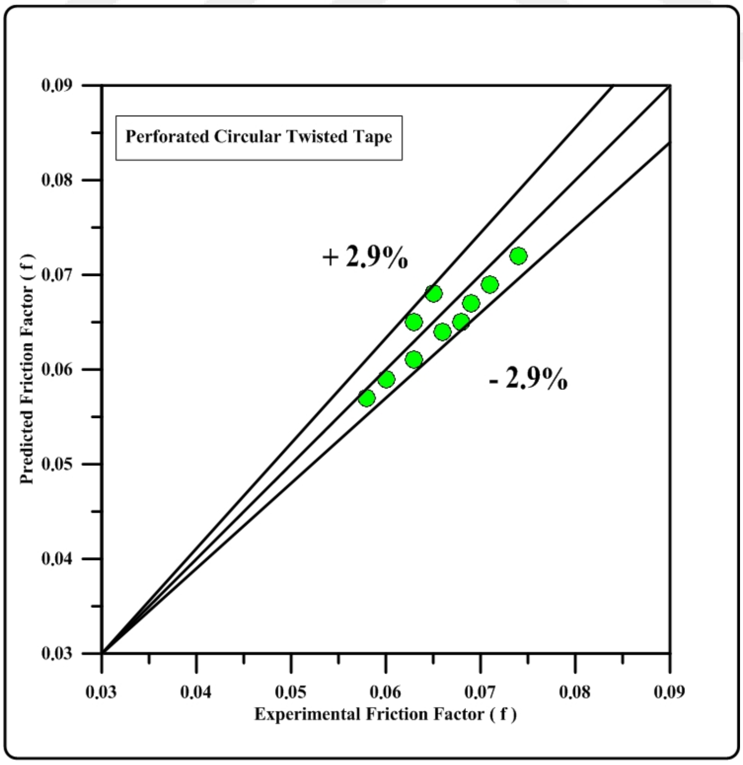


Figure 4. 100: Comparison of experimental with predicted friction factor for distilled water in a tube with perforated circular clockwise-counter clockwise twisted tape.

CHAPTER FIVE

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

This study represents experimental investigation on the performance of heat transfer through horizontal plain tube in turbulent flow region under constant heat flux by inserting various twisted tapes types with DI-water. The following remark could be concluded:

1. Generally observation, high heat transfer augmentation and drop of pressure occur with using the inserted tapes which depend on the twist ratio, type and the perforated shape for twisted tapes.

2. Decreasing the twist ratio of inserted tape leads to an increase in average Nusselt number and friction losses where $\overline{Nu}$ increases by 39% for Y=6 and by 44% for Y=4 at Re=4140 than for smooth tube case. While friction losses increase by 36% for Y=4 and by 27% for Y=6 than those for smooth tube case.

3. Perforated circular shape shows better performance in heat transfer enhancement and higher friction losses as compared to perforated square shape. Where $\overline{Nu}$ for perforated circular is higher by 6% , 8% and 15% than perforated square, clockwise-counter clockwise and typical inserted tape, respectively.

4. The average Nusselt number and factor of friction data for plain tube were validated with the standard correlations to verify the performance of the experimental set up for turbulent flow. The maximum experimental deviation obtained for both the Nusselt number and factor of friction is ±7% and ±6% respectively, with the standard correlations values.

5. In general, the variant inserted tapes used in the present study yield higher thermal performance than smooth tube. The reason for the enhancement is the small cut on the center region of the tape, which cause additional disturbance to the main swirl flow. This happened by developing secondary flow along with the main swirl flow.
6. The perforated circular clockwise-counter clockwise gives the enhanced thermal performance than that of typical insert only at lower twist ratios (y = 4). This behavior resulted due to the higher degree of turbulence mixing.

5.2 Suggestions for Future Research

Future work may include the following:

1. The effect of other twisted tapes designing such as broken twisted tape, edge fold twisted tape or other turbulators type such as coil wire insert, helical screw insert on heat transfer augmentation and friction losses through plain tube can be studied.
2. The inclination influence of the inserted tube on the flow and heat transfer process can be studied.
3. Change the type of material twisted tape such as (aluminum and stainless steel) and change the tape thickness.
4. Studying the heat transfer improvement by nanofluid with inserted tape.
5. Studying the influence of another design of inserted tape on the flow and heat transfer in a spiral fluted tube.
6. Studying the influence of using turbulators under constant wall temperature condition.
7. Theoretical and experimental investigation on the convective heat transfer and flow of Pico fluid in a straight twisted tube.
8. Investigation of the heat transfer and flow of distilled water in a round tube with a constant heat flux under turbulent flow by using a two phase flow approach.

REFERENCES

[1] Enrico Stalio " *Direct Numerical Simulation of Heat Transfer Enhancing Surfaces* " Universit`a degli Studi di Bologna - Mechanical Engineering Department . Ph.D. , thesis , Anno Accademico 2002/03.

[2] Rabia Abrahim Saikoshad " *CFD Simulation of Heat Transfer in Vertical Ribbed Tube* " University Tun Hussein Onn Malaysia - Mechanical Engineering Department . M.Sc. , thesis , MAY 2011 .

[3] Gaurav Johar " *Experimental Studies on Heat Transfer Augmen-tation Using Modified Reduced Width Twisted Tapes (RWTT) as Inserts for Tube Side Flow of Liquids*" National Institute of Technology, Rourkela -Mechanical Engineering Department. B.Sc., thesis, 2010.

[4] Bergles, A.E. Techniques to augment heat transfer. " *Hand book of Heat Transfer* " (W. M. Bohsenow, J.P. Hartnett, and Y.I. Cho, Eds.) , 3rd ed., ch. 11. McGraw-Hill , New York, 1998.

[5] Adrian B. and Allan K. D. " *Heat transfer enhancement* " In Heat Transfer Handbook , Chapter 14 , pp.1033 , -1101 , Wiley- 2003.

[6] Lishan You " *Computational Modeling of Laminar Swirl Flows and Heat Transfer in Circular Tubes with Twisted-Tape Inserts* " University of Cincinnati-Mechanical , Industrial and Nuclear Engineering Department . M.Sc. , thesis , 2002 .

[7] Smith berg, E., and Landis , F., " *Friction and Forced Convection Heat Transfer Characteristics in Tubes with Twisted Tape Swirl Generators* ", Journal of Heat Transfer 86 , pp.39-49, 1964.

[8] Radu Cazan " *Dynamics of Swirling Flows Induced by Twisted Tapes in Circular Pipes* " Georgia Institute of Technology- Mechanical Engineering Department . Ph.D. , thesis , May 2010 .

[9] Nandakumar, K., and Masliyah, J.H. " *Swirling Flow and Heat Transfer in Coiled and Twisted Pipes* ". In , Advances in Transport Processed, pp. 40-112 , Wiley Eastern , New Delhi, 1986.

[10] Webb, R.L. " *Principles of Enhanced Heat Transfer*" Wiley, New York, 1994.

[11] Bergles.A.E., " *The Implications and Challenges of Enhanced Heat Transfer for the Chemical Process Industries* " Institution of Chemical Engineers (IChemE) Vol.79 pp.437-444 , 2001.

[12] Munish Gupta " *Numerical Study of Heat Transfer Enhancement in a Plate-Fin Heat Exchanger Using Rectangular Winglet Type Vortex Generator* " National Institute of Technology - Mechanical Engineering Department . Ph.D., thesis, India, March, 2010.

[13] Webb, R.L." *Enhancement of Single-Phase Convective Heat Transfer*". Handbook of single-Phase Convective Heat Transfer, John Wiley and Sons, New York , 1987.

[14] Bhuiya M.M.K. and M.S.U.Chowdhury " *Heat Transfer Perfor-mance for Turbulent Flow Through a Tube Using Double Helical Tape Inserts* " , International Journal of Heat and Mass Transfer 39, pp.818-825, 2012.

[15] Warren Dean Snider " *Investigation of Heat Transfer Augmen-tation Through Use of Internally Finned Tubes* " Massachusetts Institute of Technology-Mechanical Engineering Department. M.Sc., thesis, May, 1968.

[16] Chintan Prajapati, Pragna Patel, Jatin Patel and Umang Patel, " *A Review of heat transfer enhancement using twisted tape*", International Journal of advanced engineering Research and Studies, Vol. 2, Issue 1, pp. 162-164, 2012.

[17] Sunny, G. and Loveleen , P., " *Review of Heat Transfer Enhancement Using Passive Techniques in Different Type of Twisted Tapes in Circular Tube*" , International Journal for Scientific Research & Development , Vol. 2, Issue 05, pp. 2321-0613, 2014.

[18] Suhas, V. Patil and P. V.Vijay Babu, " *Heat Transfer Augmentation in a Circular tube and Square duct Fitted with Swirl Flow Generators*" , International Journal of Chemical Engineering and Applications, Vol. 2 , No. 5, 2011.

[19] Elshafei EAM, Safwat Mohamed M, Mansour H, Sakr M. " *Experimental Study of Heat Transfer in Pulsating Turbulent Flow in a Pipe*", International Journal of Heat and Fluid Flow; 29:1029–38, 2008.

[20] Liebenberg L, Meyer JP. " *In-Tube Passive Heat Transfer Enhancement in The Process Industry* " . Applied Thermal Engin-eering; 27:2713, 2007.

[21] S. Liu, M.Sakr, "*A comprehensive Review on Passive Heat Transfer Enhancements in Pipe exchangers* ," Renewable and Sustainable Energy Reviews , Vol. 19 ,pp. 64–81, 2013.

[22] A. Dewan, P. Mahanta ,K Sumithraju, P. Suresh kumar , " *Review of Passive Heat Transfer Augmentation Techniques* " , Proc. Institution of Mechanical Engineers Vol. 218 Part A: Journal of Power and Energy, 2004.

[23] Manglik, R. M. and Bergles, A. E. " *Heat Transfer and Pressure Drop Correlations for Twisted Tape Insert in Isothermal Tubes* ", Part 1: Laminar Flows. Trans. ASME, J. Heat Transfer, 116, pp.881–889, 1993.

[24] Manglik .R .M and A. E. Bergles, "*Heat transfer and pressure drop correlations for twisted tape inserts in isothermal tubes*": part II transition and turbulent flows", Journal of heat transfer, Vol.115, Issue.4, (1993).

[25] Wang L., Sunden B., "*Performance comparison of some tube inserts*", International Communication Heat and Mass Transfer, Vol.29, pp. 45–56, (2002).

[26] Taiwo Oluwasesan, "*Numerical investigation of heat transfer and fluid flow in tubes induced with twisted tape inserts*", University of Glasgow, http://theses.gla.ac.uk/6237/, (2015).

[27] Manglik, R. K. and Bergles, A. E., " *Heat Transfer and Pressure Drop Correlations for Twisted-Tape Inserts in Isothermal Tubes* " Part I : laminar flows. Trans. ASME, J. Heat Transfer ,Vol.115, pp.881–889, 1993.

[28] S.K.Agarwal and M. Raja Rao, " *Heat Transfer Augmentation for The Flow of a Viscous Liquid in Circular Tubes Using Twisted Tape Inserts* " International. Journal of Heat Mass Transfer, 39, pp. 3547–3557, 1996.

[29] Saha, S. K. and Chakraborty, D.," *Heat Transfer and Pressure Drop Characteristics of Laminar Flow Through a Circular Tube Fitted with Regularly Spaced Twisted Tape Elements with Multiple Twists*",In Proceedings of 3rd ISHMT–ASME Heat and Mass Transfer Conference, India, pp. 313–318 (Tata McGraw-Hill , New Delhi), 1997.

[30] Al-Fahed, S., Chamra, L. M. and Chakroun, W. " *Pressure Drop and Heat Transfer Comparison for Both Micro-Fin Tube and Twisted-Tape Inserts in Laminar Flow* ".Experimental Thermal and Fluid Sci., Vol.18, pp.323–333, 1999.

[31] Klaczak A., " *Heat Transfer by Laminar Flow in a Vertical Pipe with Twisted-Tape Inserts* " , Heat and Mass Transfer,Vol. 36 pp. 195-199, Springer-Verlag, 2000.

[32] Liao, Q. and Xin, M. D., " *Augmentation of Convective Heat Transfer Inside Tubes with Three-Dimensional Internal Extended Surfaces and Twisted Tape Inserts* " , Chem. Eng. J., 78, pp. 95–105, 2000.

[33] Patil A.G.," *Laminar Flow Heat Transfer and Pressure Drop Characteristics of Power-Law Fluids Inside Tubes with Varying Width Twisted Tape Inserts*" , ASME J. Heat Transfer, 22(1), pp. 143–149, 2000.

[34] Saha, S. K. and Dutta, A. , " *Thermo-hydraulic Study of Laminar Swirl Flow Through a Circular Tube Fitted with Twisted Tapes*" , Trans. ASME, J. Heat Transfer, 123, pp. 417–421, 2001.

[35] Lishan You, "*Computational Modeling of Laminar Swirl Flows and Heat Transfer in Circular Tubes with Twisted-Tape Inserts*", Msc, thesis, University of Cincinnati, Department of Mechanical, Industrial and Nuclear Engineering, College of Engineering, 2002.

[36] Sarma P.K., Subramanyam, Kishore P.S., V. Dharma Rao and Kakac Sadik," *Laminar Convective Heat Transfer with Twisted Tape Inserts in a Tube* ", International Journal of Thermal Sciences, 42, pp. 821–828, 2003.

[37] Suresh Kumar, P., Mahanta, P. and Dewan, "*A Study of Laminar Flow in a Large Diameter Annulus with Twisted Tape Inserts*." In Proceedings of 2nd International Conference on Heat Transfer, Fluid Mechanics, and Thermodynamics, Victoria Falls, Zambia, paper KP3, 2003.

[38] Bharadwaj P., A.D. Khondge, A.W.Date, " *Heat Transfer & Pressure Drop in Spirally Gooved Tube with TwistedT Insert*." J. Heat Transfer, 52,5, pp. 1938-1944, 2009.

[39] Lin Zhi-min and Wang Bi "*Convective Heat Transfer Enhance-ment in a Circular Tube Using Twisted Tape*." J. Heat Transfer, 131, 8 , pp. 81-91, 2009.

[40] Sheeba K.N," *Experimental Studies on Heat Transfer and Friction Factor Characteristics of Forced Circulation Solar Water Heater System Fitted with Helical Twisted Tapes*", Communicated by Associated Editor Brian Norton, 2009.

[41] Yadav Anil," *Effect of Half Length Twisted Tape Turbulator on Heat Transfer & Pressure Drop Characteristics Inside a Double Pipe U-Bend Heat Exchanger*" Jordan journal of Mech. & industrial engg. 3, 1, pp.17-22, 2009.

[42] Eiamsa-ard S., Seemawute P., Wongcharee K., "*Influences of Peripherally Cut Twisted Tape Insert on Heat Transfer and Thermal Performance Characteristics in Laminar and Turbulent Tube Flows*", Experimental Thermal and Fluid Science, 34, , pp.711–719, 2010.

[43] Kapatkar V. N., Dr. Padalkar A. S. and Sanjay Kasbe," *Experimental Investigation on Heat Transfer Enhancement in Laminar Flow in Circular Tube Equipped with Different Inserts*", Proc. of. Int. Conf. On Advances in Mechanical Engineering, pp. 58-63, 2010.

[44] Jian Guo, Aiwu Fan, Xiaoyu Zhang, Wei Liu, " *A numerical Study on Heat Transfer and Friction Factor Characteristics of Laminar Flow in a Circular Tube Fitted with Center-Cleared Twisted Tape* ", International Journal of Thermal Sciences 50, pp. 1263-1270, 2011.

[45] Suhas V. Patil, P. V. Vijay Babu, " *Experimental Studies on Heat Transfer and Friction Factor Through Isothermal Square Duct with Twisted Tape Inserts* " , Proceeding of the International Conference on Advanced Science, Engineering and Information Technology Malaysia, pp. 77-82, 2011.

[46] Wongcharee K. and Eiamsa-ard S., " *Friction and Heat Transfer Characteristics of Laminar Swirl Flow Through The Round Tubes Inserted with Alternate Clockwise and Counter-Clockwise Twisted-Tapes*", International Communications in Heat and Mass Transfer , 38, pp.348–352, 2011.

[47] Zhang X,Liu Z, Liu W., " *Numerical Studies on Heat Transfer and Flow Characteristics for Laminar Flow in a Tube with Multiple Regularly Spaced Twisted Tapes* ", Int J Therm Sci;58, pp. 157–167, 2012.

[48] Sami D., Salman, A., Kadhum M., and Abu Bakar M., "*CFD Simulation of Heat Transfer and Friction Factor Augmentation in a Circular Tube Fitted with Elliptic-Cut Twisted Tape Inserts*",Mathematical Problems in Engineering, Article ID 163839, 7 pages, Volume 2013.

[49] Suvanjan B., Subhankar S., Sujoy K., " *Laminar Flow Heat Transfer Enhancement in a Circular Tube Having Integral Transverse Rib Roughness and Fitted with Centre-Cleared Twisted-Tape*", Experimental Thermal and Fluid Science, 44, pp.727–735, 2013.

[50] Yamada, Y., Akai, M. and Mori, Y., " *Shell and Tube Side Heat Transfer Augmentation by The Use of Wall Radiation in a Cross Flow Shell and Tube Heat Exchanger* ", Trans. ASME, J. Heat Transfer, 106, pp.735–742, 1984.

[51] Algifri, A. H. and Bharadawj, R. K., " *Prediction of Heat Transfer for Decaying Turbulent Swirl Flow in a Tube*", Int. J. Heat and Mass Transfer, 28, pp.1635–1643, 1985.

[52] Gupte, N. S. and Date, A. W.," *Friction and Heat Transfer Characteristics of Helical Turbulent Air Flow in Annuli* ", Trans. ASME, J. Heat Transfer, 111, pp. 337–344, 1989.

[53] Saha, S. K., Gaitonde, U. N. and Date, A. W. ," *Heat transfer and pressure drop characteristics of turbulent flow in circular tube fitted with regularly spaced twisted tape elements* ", Expl. Thermal and Fluid Sci., 3, pp. 632–640, 1990.

[54] Hijikata, K., Nagasaki, T. and Minami, K.," *Study of Heat Transfer Augmentation in a High-Temperature Field by a Radiation Promoted Generating a Secondary Flow*", Int. J Exp. Thermal and Fluid Sci., 7, pp. 31–42, 1994.

[55] Rao, M. M. and Sastri, V. M. K.," *Experimental Investigation for Fluid Flow and Heat Transfer in a Rotating Tube Twisted Tape Inserts*" , Int. J. Heat and Mass Transfer, 16, pp. 19–28, 1995.

[56] Naumov, V. K. and Semashko, N. N.," *Analytical Model of Estimating Thermo-physical and Strength Parameters of Cooled Pipe with The Twisted Tape Under Asymmetric Heating by a Pulse of External Heat Flux* ", Plasma Devices and Ops., 3, 267, 1994.

[57] Sivanshanmugam P. and Sunduram S.," *Improvement in Perform-ance of Heat Exchanger Fitted with Twisted Tape*", Journal of Energy Engg.,125, pp. 35–40, 1999.

[58] Kumar, A. and Prasad, B. N.," *Investigation of Twisted Tape Inserted Solar Water Heater Heat Transfer, Friction Factor and Thermal Performance Results* ", Renewable Energy, 19, pp. 379–398, 2000.

[59] Hong, Xianhe Deng, Kuo Huang, &Zhiwo Li, "*Compound Heat Transfer Enhancement of a Converging Diverging Tube with Evenly Spaced Twisted Tapes*" Chinese Journal of chemical Engg., 15,6,pp. 814-820, 2007.

[60] Chang S.W, Tsun Lirng Yang andJin Shuen Liou, " *Heat Transfer and Pressure Drop in Tube with Broken Twisted Tape Insert* " , Experimental Thermal and Fluid Science Vol. 32, pp. 489- 501, 2007.

[61] Chang S.W., Tong-Miin Liou, Jin Shuen Liou and Kun-Tse Chen, " *Turbulent Heat Transfer in a Tube Fitted with Serrated Twist Tape Under Rolling and Pitching Environments with Applica-tions to Shipping* " , Journal of Ocean Engineering, Vol. 35, pp. 1569-1577, 2008.

[62] Smith Eiamsa-ard, Chinaruk thianpong, Peterpics Eiasmsa-ard & Pongjet omvonge "*Convective Heat Transfer in a Circular Tube with Short Length Twisted Tape*" International Communications in Heat & Mass Transfer. 36,4, pp. 365-371, 2009.

[63] Thianpong Chinaruk, Petpices Eiamsa-ard, Khwanchit Wongcharee and Smith Eiamsa-ard," *Compound Heat Transfer Enhancement of a Dimpled Tube with a Twisted Tape Swirl Generator* ", International Communications in Heat and Mass Transfer Vol. 36, pp. 698–704, 2009.

[64] Behabadi M.A., Hejazi V., and Afshari A.," *Experimental Investigation of Twisted Tape Inserts Performance on Condensa-tion Heat Transfer Enhancement and Pressure Drop* ", International Communications in Heat and Mass Transfer, Vol. 37 pp. 1376–1387, 2010.

[65] Eiamsa-ard S., Thianpong C., Eiamsa-ard P., Promvonge P., " *Thermal Characteristics in a Heat Exchanger Tube Fitted with Dual Twisted Tape Elements in Tandem*", International Communications in Heat and Mass Transfer, 37, pp. 39–46, 2010.

[66] Kalyani Radha K., Naga Sarada S., Sita Rama Raju A.V., L. Shyam Sunder, " *Enhancement of Heat Transfer Using Varying Width Twisted Tape Inserts*, International Journal of Engineering, Science and Technology, Vol. 2, No. 6, pp. 107-118, 2010.

[67] Seemawute P., and Eiamsa-ard S., " *Thermohydraulics of Turbulent Flow Through a Round Tube by a Peripherally-Cut Twisted Tape with an Alternate Axis*", International Communica-tions in Heat and Mass Transfer, 37, pp. 652–659, 2010.

[68] Hataa K. and Masuzakib S., " *Twisted-Tape-Induced Swirl Flow Heat Transfer and Pressure Drop in a Short Circular Tube Under Velocities Controlled* " , Nuclear Engineering and Design, Vol. 241, pp. 4434 – 4444, NED-5980, 2011.

[69] Murugesan P., Mayilsamy K., Suresh S. and Srinivasan P.S., " *Heat Transfer and Pressure Drop Characteristics in a Circular Tube Fitted with and Without V-Cut Twisted Tape Insert* ", International Communications in Heat and Mass Transfer, 38, pp. 329–334, 2011.

[70] Yangjun W., "*Configuration Optimization of Regularly Spaced Short-Length Twisted Tape in a Circular Tube to Enhance Turbulent Heat Transfer Using CFD Modeling*", Applied Thermal Engineering, 31, pp. 1141-1149, 2011.

[71] Halit B. and Veysel O., " *Heat Transfer Enhancement in a Tube With Twisted Tape Inserts Placed Separately from The Tube Wall*", Experimental Thermal and Fluid Science, 41, pp. 51–58, 2012.

[72] Burse A. K., R. H. Yadav and N. K. Sane, " *Enhancement of Heat Transfer in a Horizontal Circular Pipe using Twisted Tape Inserts of Different Twist Ratios* " International Journal of Advances in Engineering and Management (IJAEM) , Vol. 1, Issue 5, pp. 45-52, 2014.

[73] Gulia S. and Parinam A.," *CFD Analysis for Heat Transfer Enhancement inside a Circular Tube using with different-2 Length Ratio Twisted Tapes* ", IJERGS,vol.2, Issue4, pp. 562-573, 2014.

[74] Kulkarni K.G., "*Experimental Investigation of Heat Transfer Enhancement by Using Clockwise and Counter -clockwise Corrugated Twisted Tape Inserts*", International Journal of innovations in Engineering Research and Technology [IJIERT] ISSN: 2394-3696 Volume 2, Issue 5, May-2015.

[75] Kannan E., Balasuthagar C. and S. Ponsankar, " *Experimental investigation on convective heat transfer characteristics in a circular tube fitted with perforated twisted tape and wire coil*", Journal of Chemical and Pharmaceutical Sciences, JCPS Volume 9 Issue 4, October - December 2016.

[76] Kumar V. Sudheer "*CFD Analysis on Heat Transfer Enhancement in Horizontal Tube by Forced Convection with Inserts*",International Journal & Magazine of Engineering, Technology, Management and Research Volume 3, issue no. 2, Feb 2016.

[77] Orhan K. and Veysel O.," *Experimental investigation on heat transfer enhancement of a tube with coiled-wire inserts installed with a separation from the tube wall*", International Communications in Heat and Mass Transfer, 08.024, 2016.

[78] Frank M. White, "*Fluid Mechanics*", Fourth edition, MacGraw-Hill books,2001.

[79] Holman J. P., "*Heat Transfer*", Tenth Edition, 2010 by The McGraw-Hill Companies, Inc.

[80] Eiamsa-ard S., Wongcharee K., Eiamsa-ard P., and Thianpong C., "*Heat transfer enhancement in a tube using delta-winglet twisted tape inserts*", Applied Thermal Engineering, 30, pp. 310–318, 2010.

[81] Eiamsa-ard S., Thianpong C., Eiamsa-ard P., "Turbulent heat transfer enhancement by counter/co-swirling flow in a tube fitted with twin twisted tapes", Experimental Thermal and Fluid Science, 34, pp. 53–62, 2010.

[82] Holman, J. P., "*Experimental Method for Engineers*", McGraw – Hill, Tokyo, Japan, 4^{th} Edition (1979).