Convective Heat Transfer in a Circular Tube with Short-length Twisted Tape Insert

A graduation project

is submitted to the Air Conditioning and Refrigeration Techniques Engineering Department in partial fulfillment of the requirements for the degree of Bachelor of Science in Air Conditioning and Refrigeration Techniques engineering.

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Certification

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Dedication

To Who Made of Me What I am… To My Family, the Cause of My Success.

Students

Acknowledgements

First of all, praise be to God Who offered me patience, power, and faith in a way that words cannot express. People who helped me with their advice, patience and support, I would like to express my deepest thanks for them.

I wish to express my sincere thanks and deep gratitude to my supervisor, **Dr. Azher M. Abed** for his valuable guidance, continuous encouragement and interest during this work.

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Abstract:

This study investigate the heat transfer improvement in a single tube heat exchanger with and without twisted tape insert with twist ratio (y/w = 3). The performance of the heat exchanger studied on the oblique and horizontal tube. The laboratory work included fabricating a circular tube heat exchanger which made from carbon steel material of length (60 cm) and inner and outer diameter of (42.6 and 48.6) mm, respectively Distilled water under turbulent flow condition (Re =5733 - 13276) flows through insulated tube by using fiber glass at the outer surface to reduce the thermal losses. The inserted tape was made from aluminum strip of thickness (0.5 cm) and full length inserted in the test section. The results showed that the employ of inserted tape yield a considerable increase in the coefficient of heat transfer about (16-27)% more than smooth tube. A comparison of thermal performance for plain and twisted tape insert tubes was implemented and gave good agreements between the experimental and numerical results with a maximum deviation of 10%.

Chapter On

Introduction

1.1 Introduction

Thermal systems are one of the important things which is used for engineering applications. Therefore, there are several methods developed to improve heat exchanger and reach to the optimum performance in these systems. Different augmentation techniques that uses surface improvements have been used to enhance the rate of heat transfer in conventional heat exchangers[1].

This improvement resulting from the use of enhanced surfaces which minimize the boundary layer development and increase the turbulence degree which leads to the creation of swirling and secondary flows. Swirl flow devices are one of the passive techniques used in heat transfer enhancement. Swirl flow devices cause swirl flow or secondary flow in the fluid. This effect can be caused by different types of devices such as tube insert, duct geometry alterations as well as altered tube flow arrangements[1].

Tubes with twisted tape insert have been widely used as the continuous swirl flow devices for augmentation the heat transfer rate in heat exchanger tubes and applied in many engineering applications Insertion of twisted tape in a tube provides a simple passive technique for enhancing the convective heat transfer by producing swirl into the bulk flow and by disrupting the boundary layer at the tube surface. It has been explained that such tapes induce turbulence and super- imposed vortex motion (swirl flow) causing a thinner boundary layer and consequently resulting in higher heat transfer coefficients However, the increase in friction is seemed to be the penalty of the technique.Thus, tube with twisted tape insert is frequently used in heat exchanger systems because of it low cost, less maintenance and compact[2].

Heat exchangers are well-known for being the backbone of process industry. The optimization of heat exchangers can lead to energy savings, which is currently the most vital research area. There are two methods of heat transfer enhancement known as active and passive methods . The passive method is commonly used to improve heat transfer because it does not require any additional energy source. Passive methods include inserts (Twisted tape, longitudinal strip, turbulators), additives in fluids, fins and so on. Many methods are used in the past to improve its performance like using

foam filled DPHE , using porous substrates, time constant method, using nanofluid and core insert, outward helically corrugated tape [3].

In order to increase the efficiency of a double pipe heat exchanger, various types of inserts were used. Varun et al. [10] reviewed on twisted tape inserts for augmentation of heat transfer. In this article optimum shape and size of twisted tape was suggested on the basis of heat transfer performance. It was observed that square and rectangular cut have better heat transfer enhancement relative to other modified twisted tape compared. It was concluded that twisted tape performance for DPHE depend upon the twist ratio, depth of cut, spacing between cuts, depth, width and pitch ratio[3].

It was observed that modified twisted tape have high thermal performance factor. So twisted tape attracted the attention of most of the investigator. Different modification has been done in twisted tape to enhance thermal performance factor for double pipe heat exchanger (DPHE)[3].

The text of the current research is heat transfer by convection in a circular tube with a short-length twisted bar, as the convective heat transfer between the hot surfaces and the cooler ambient barrier is governed by Newton's law of refrigeration, which states that "the rate of convective heat transfer is directly proportional to the temperature difference between the heated surface and the damper. The perimeter is also directly proportional to the area of contact between them, and it is expressed by the following formula:

$$\boldsymbol{Q_{CONV} = hA(T_S - T_F)}$$

Convection heat is transmitted in one of the following ways:

1. Increasing the temperature difference between the surface and the barrier.
2. Increasing the convective heat transfer coefficient by enhancing the flow rate of the inhibitor on the surface.
3. Increase the contact area.

Most of the time in industrial applications, the temperature difference cannot be controlled, and increasing the heat transfer coefficient by load requires the installation of an additional pump or fan at an additional price, and the best solution is to increase the surface area of the contact by adding extended surfaces.

Chapter two

Experimental Procedure:

During experiments, inlet water, was drawn through the clam section to achieve a fully developed flow prior to being heated by an adjustable electrical heater wrapping along the test section, water flow rate was varied corresponding to Reynolds number (Re) between 200 and 2400. The electrical output power was controlled via variac transformer by keeping the current below 3 amps to achieve a constant heat flux condition along the entire test section. Pressure drop across the test section was measured using a pressure gauge filled by the manometric fluid having low specific gravity (SG.) of 0.816 to ensure reasonably accurate measurement of the low pressure drop encountered at low Reynolds numbers. In the present test, all data were taken at steady state.

The schematic diagram of the experimental heat exchanger system is depicted in Fig. (3-1). The experimental system consisted of a 0.5Hp pump, an orifice meter to measure the volume flow rate, inclined or U-tube manometer to measure the pressure drop across the test section, data logger to record the temperature of the inlet and outlet test section and also measured the tube wall temperature of the test tube, and the heat transfer test section. The schematic view of the heat transfer test section and the short-length twisted tape is depicted Fig. (3-2). The short-length twisted tapes were made of Aluminum strips They were fabricated by twisting a straight strip, about its longitudinal axis at constant twist ratio (y/w) of 4.0, while being held under tension. In the test run, the tapes were placed in the tube. The tube was heated by continually winding flexible electrical wire to provide a uniform heat flux boundary condition. The outer surface of the test tube was well insulated to minimize convective heat loss to surroundings, and necessary precautions were taken to prevent leakages from the system. In the apparatus setting above, the inlet bulk air induced by a 0.5 Hp pump was directed through the heat transfer test section and passed to an orifice meter. The calibrated thermocouples were used to measure the temperatures of the fluid at the inlet/outlet of the test section. In the experiments, it was necessary to record the data of temperature, volumetric flow rate and

pressure drop of the water at steady state conditions in which the inlet air temperature were maintained at 25 °C.

The practical aspect begins with preparing the device, then filling the first tank with water at room temperature, then operating the device to start heating the water to a certain temperature. Also, through the pump, we raise the water to the isolated test tube. We measure the flow rate as well as the pressure of the water leaving the test tube, as well as the temperatures are measured With the presence and absence of the twisted tape, and in this way we can know the amount of heat transferred through the liquid.

2.1Literature review:

In recent years, there has been considerable effort in the development process of heat transfer augmentation techniques to increase the performance of heat exchanger and to enhance inside tubes convective heat transfer coefficient.

Yadav [8] studied experimentally the convective heat transfer and friction factor in a U-bend double pipe heat exchanger equipped with half-length inserted tape to generate swirl flow. The result revealed that the use of inserted tape leads to increase the heat transfer coefficient about 40% more than that of plain tube. However, the plain tube thermal performance was found to be better than half-length twisted tape by (1.3-1.5) times.

Naga et al. [9] examined the utilization of reduced width twisted tape inserts in a horizontal circular tube in order to enhance the heat transfer rate. The tube has an inside diameter of 27.5mm and air was employed as working fluid. The experiments were conducted using twisted tapes with three different twist ratios (3, 4 and 5). Each of these tapes has five different widths (26-full width, 22, 18, 14 and 10 mm). The range for the Reynolds number was from 6000 to 13500. They observed that as y/w ratio decreases, the heat transfer increases.

Bodius et al. [11] carried out an experimental study to measure heat transfer coefficient of water on tube side for turbulent flow. The tube is circular and is fitted with stainless steel twisted tape insert of 5.3 twist ratio. Nichrome wire covered with fiber glass was wrapped around the test section in order to retain a uniform heat flux condition. The temperature of the tube outer surface in the test section was measured at five different spots. T-type thermocouples were used to measure the temperature with a thermometer placed in a mixing chapter at the outlet section. The study was conducted over a range of 9500-20000 of Reynolds numbers. Heat flux was varied from 9 to 18 kW/m2 for smooth tube and 15 to 31 kW/m2 for tube with twist tape insert. It was noticed that over the same Reynolds number, twist tape insert caused an improvement in Nusselt number by 2.9 to 4 times when compared to the smooth tube.

Results were compared with Dittus and Boelter correlation and the error was -13% to 18%.

Naresh et al. [12] analyzed the heat transfer performance of helical strip insert with regular space cut the passages generated a turbulent flow in a circular pipe. The range of the Reynolds number used in the experiments was from 5000-30000. Three different helical strips were used with helix angles of 30◦, 45◦ and 60◦. The experiments showed that with the insertion of helical strips, the heat transfer rate is improved as a result of the turbulent flow generated in the circular pipe. “The local heat transfer coefficients were found to be increasing to very high values along the downstream of the helical strip, and then decreasing with the distance. Number of helical channels and the helix angle did not have a big effect on the heat transfer. Overall, the helical tape led to a maximum 20% improvement in the heat transfer rate depending on Reynolds number With the increase of Reynolds number, the efficiency of heat transfer enhancement decreased.

Sami et al. [13] modeled, simulated and analyzed the effect of Parabolic-Cut Twisted tape (PCT) inserts fitted in a circular tube on the heat transfer rate using Computational Fluid Dynamics (CFD) modeling. A commercial CFD package (FLUENT-6.3.26) was used to carry out the simulation. The modeled circular tube is a constant heat-fluxed tube having a laminar flow. Three different twist tapes were considered in the simulation with twist ratio (y=2.93, 3.91and 4.89) and cut depth (w =0.5, 1 and 1.5 cm). It was discovered that the Nusselt number and the friction factor in the tube fitted with PCT increase with the decrease of twist ratios (y) and cut depth (w). The CFD predicted results matched with the literature correlations for plain tube for validation; with the discrepancy of less than ±8%for Nusselt number and ±6.5 % for friction factor.

Selvam et al. [14] conducted an experiment for studying the flow and thermal characteristics of tube induced with various type of twisted tape (Twisted tape with pins and twisted tape with pins bonded). Three different twist ratios have been used during the study (3.33, 4.29, and 5.71). The experimental results revealed that the smaller twist ratio leads to higher heat transfer values about 23.86% more than smooth tube and for twist ratios 4.29 and 5.71 the improvement was 19.9% and 14.4%, respectively. They also observed that the friction factor for twist ratio 3.33 is higher than those of twist ratios 4.29, and 5.71 because of the less contact surface area of the turbulator. Empirical correlations have been developed with maximum deviation of ±7.28, and ±7.16% for Nusselt number and friction factor, respectively.

Osama [15] investigated experimentally and numerically the most advantageous design parameters of helical coiled tube. A helical tube (15 mm diameter) heat exchanger with the effects of insertion a coil wire of twisted tape into the helical tube heat exchanger were examined. Dean number is in the range of 700 to 2000. Coiled wire with different insertion of 15, 20 and 30 mm is used firstly, and then experiments were conducted at a constant pitch of 15 mm with different sections circular of square thirdly, at constant insert pitch of 15 mm with different square wire thickness (a=1 and 2 mm). Nanoparticles of Al2O3 and TiO2 (d=80 nm,30 nm) respectively, dispersed in distilled water. The volume concentrations were in the range of (0.08, 0.1, 0.2 and 0.3%) in order to simulate the flow of heat transfer in the helical coil tube with nanofluids. A commercial program ANSYS Fluent 14.5 is used. The maximum heat transfer enhancement resulting from using coil wire and nanofluid was exceeds over 120% as compared to smooth tube and pure water. An empirical correlation for Nusselt number were developed with maximum deviation of (±20 %).

Suresh et al. [16] investigated and compared the thermal performance of Al2O3/water and CuO/water nanofluids when a helical screw tape inserted in a straight circular duct. Three different helical screw tape were used with twist ratio of (y=1.78, 2.44 and 3). The experiment was carried out with a 0.1% volume concentration for both Al2O3/water and CuO/water nanofluids. The maximum increase in Nusselt number as a result of the use of water, Al2O3/water and CuO/water with inserts compared to plain tube are 156.24%, 166.84%, 179.82% respectively at y= 1.78.

Maddah et al. [17] studied experimentally the influences of inducing inserted tape and using nanofluid in double pipe heat exchanger. The internal pipe with inner diameter of 8mm and thickness 4 mm. The hot water flows in the inner tube while the cold water was used in shell side. The inserted tape is 120 cm long and 5 mm wide and have a thickness of 1mm. Aluminum sheet was used to make the twisted tapes. The nanofluid was prepared from Titanium dioxide with a volume concentration of 0.01% and 30nm diameter. It was observed that using twisted tapes in addition to nanofluid will lead to 10 to 25 percent increase in heat transfer coefficient. Furthermore, increasing mass flow rate and operating temperature will increase the coefficient as well. Moreover, the experiment showed that friction factor and pressure drop is higher with the use of twisted tapes.

2.2The main idea of the project :

The present work is an experimental mixed heat transfer considering the new design f the circular shape. The fluid used in the present work is water that circulates within closed thermal systems.

2.3: The importance of the project :

1. Gain distinct practical skills at the level of technical education.
2. Creating a nucleus of technicians to develop the industry and production of air conditioning, refrigeration and heating devices at the national level with a quality not less than the quality of industry and production in the Middle East.
3. Enabling graduates to understand the theoretical foundations of the processes of obtaining artificial cold and to identify the different fields of its use.
4. Recognize the different forms of energy and the mechanisms of its transformation.
5. Introducing the graduate to the nature of work in the relevant centers, institutions and companies.
6. Providing the graduate with skills in communication, teamwork, dialogue, construction, negotiation and production.

Project Objectives :

1. Indicate the search variables
2. Know the components of the device
3. Knowing how heat is transferred by convection in a round tube with a short-length twisted bar.

Chapter three

The device and its components:

Apparatus: The pilot rig consists of a short-length torsion-bar circular chute, an insulated electric heater as well as a pressure gauge, 5-thermocouples, a flow rate meter, and a temperature at specific locations. 2-water tanks and a pump were also installed to supply the water with the required driving force through the systems. The heat flow was provided by the power supply.

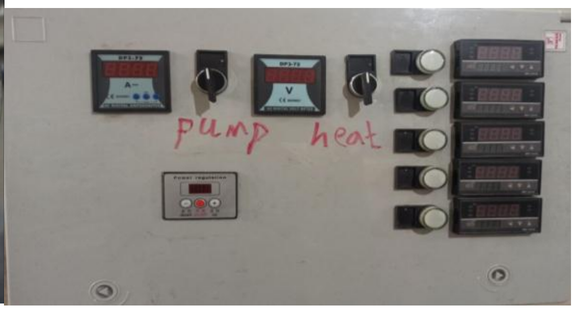
Figure (3-1): The device used for studying.

1- An electric pump: An electric water pump is used to draw water from the tank and push it into the test tube to obtain a constant flow

Type	Head (m)	Q (Lpm)	Rotation (rpm)	Power (hp)	Electric force (V)	Electrical current(A)
TAAM70	50	55	2850	0.5	220-240	4

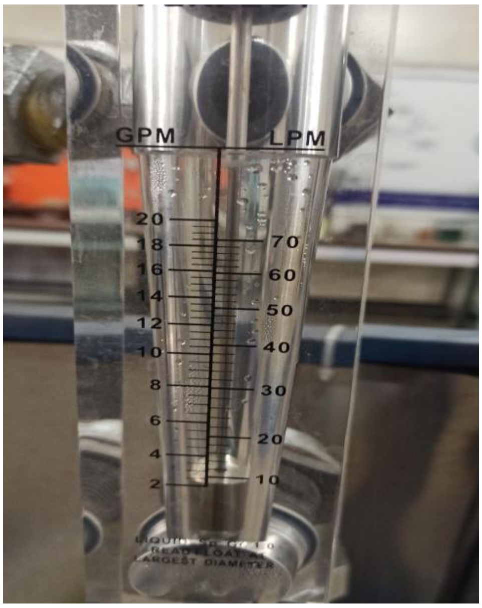
 

2. Pressure Gage: Pressure Gage / Detects the pressure rate in the system with specification (0-6 bar) and (0-87pci)



3. **An electric board** In the electric board there is a switch to operate the water pump, a switch to operate the electric heater, a voltmeter and an ampere meter (digital screens and signal lamps) for the work of thermocouples and a controller to increase and reduce voltages

4. **<u>flow meter (Ratometer)</u>** or flow meter To measure the volumetric flow rate of a fluid in units of liters/minute and Gram/ minute (10-70 lpm) (2-20 gpm)

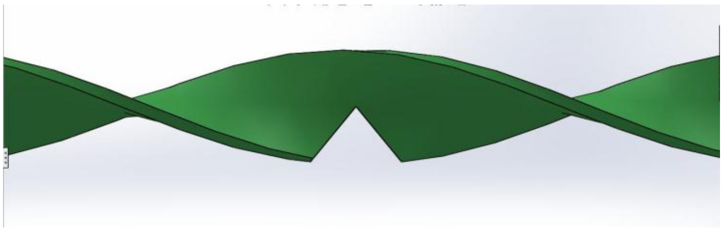


5. Test tube

Test tube: It is one of the most important parts of this device. There are thermocouples on the surface of the tube - and inside the tube there is a short-length twisted stripe a short-length twisted stripe The benefit of the twisted tape to obtain turbulent flow and study the effects of the twisted geometry The Friction Factor, Nusselt Number, and . are included heat boosting factor

Test tube dimensions

Outside diameter	Inside diameter	Length	material
48.6 mm	42.6mm	60 cm	galvanized iron



1	The total length of the fin	40cm
2	The length of the fin yet twisting	35cm
3	twisted length	12 cm
4	Number of twisted	3
5	fin thickness	$\frac{1}{2}\ cm$
6	Torsion angle	60° -180- 360
7	The distance between one center and another of the twisted	There is no space because it is sequential
8	Material	steel

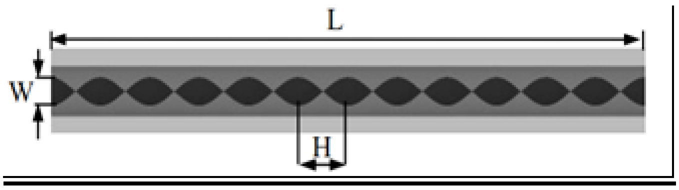
6. Accessory parts

TWO tanks

Three valves

was heated by continually winding flexible electrical wire to provide a constant heat flux boundary condition. The electrical output power was controlled by a variac transformer to obtain a constant heat flux along the entire length of the test section and by keeping the current less than 3 amps. The outer surface of the test tube was well insulated to minimize convective heat loss to surroundings, and necessary precautions were taken to

prevent leakages from the system. The inner and outer temperatures of the bulk air were measured with a multichannel temperature measurement unit in conjunction with the K-type thermocouples. Five K-type thermocouples were tapped on the local wall of the tube. The mean local wall temperature was determined by means of calculations based on the reading of K-type thermocouples



Chapter four

Forced convection theory

4.1 Forced Convection Heat Transfer

Convection is the mechanism of heat transfer through a fluid in the presence of bulk fl uid motion. Convection is classified as natural (or free) and forced convection depend ing on how the fluid motion is initiated. In natural convection, any fluid motion is cau sed by natural means such as the buoyancy effect, i.e. the rise of warmer fluid and fall the cooler fluid. Whereas in forced convection, the fluid is forced to flow over a surfa ce or in a tube by external means such as a pump or fan[4].

4.2:Methods of improving heat transfer can be divided into two main groups

1. Passive Methods: In which heat transfer is stimulated without the need for external power, for example, surfaces rough, elongated surfaces, (Surface wip) or Coiled Extension tubes.
2. Active Methods: The heat transfer is stimulated by an external power, and this leads to additional energy consumption, so it is used only in applications where energy losses are not important.

Fluid flow within channels has had great applications in the engineering industry such as refrigeration systems, air conditioning systems, heat exchangers, solar collectors, nuclear power plant reactor cooling, electronic derivatives cooling, etc[5]

4.3. Basics and types of heat transfer :

4.3.1. Heat Transfer by Natural Convection :

It is a type of flow, the movement of a fluid like water or a gas like air, in which the movement of fluids is not generated by any external source (such as pump, fan, suction device, etc.) but by some parts of the fluid heavier than others. In most cases, this leads to natural circulation, that is, the ability of the fluid in the system to circulate continuously, with gravity and possible changes in thermal energy. The driving force of a normal load is gravity.

4.3.2. Forced Convection of Heat :

In natural convection, any fluid motion occurs by natural means such as the buoyancy effect, that is, the rise of the warmer fluid and the fall of the cooler fluid. Whereas in forced convection, the fluid is forced to flow over a surface or into a tube by external means such as a pump or fan.

4.3.3. Mixing Convection Heat Transfer:

This flow occurs when forced and natural (free) convection mechanisms contribute significantly and simultaneously to heat transfer. The relative contribution of each mechanism depends on the flow regime (laminar or turbulent) and the magnitude of the temperature driving force for heat transfer.

4.4: Mechanism of Forced Convection

Convection heat transfer is complicated since it involves fluid motion as well as heat c onduction. The fluid motion enhances heat transfer (the higher the velocity the higher the heat transfer rate). The rate of convection heat transfer is expressed by Newton's l aw of cooling[4]:

$$\dot{q}_{conv} = h(T_s - T_\infty) \qquad (W/m^2)$$
$$\dot{Q}_{conv} = hA(T_s - T_\infty) \qquad (W)$$

The convective heat transfer coefficient h strongly depends on the fluid properties and roughness of the solid surface, and the type of the fluid flow (laminar or turbulent)[4]

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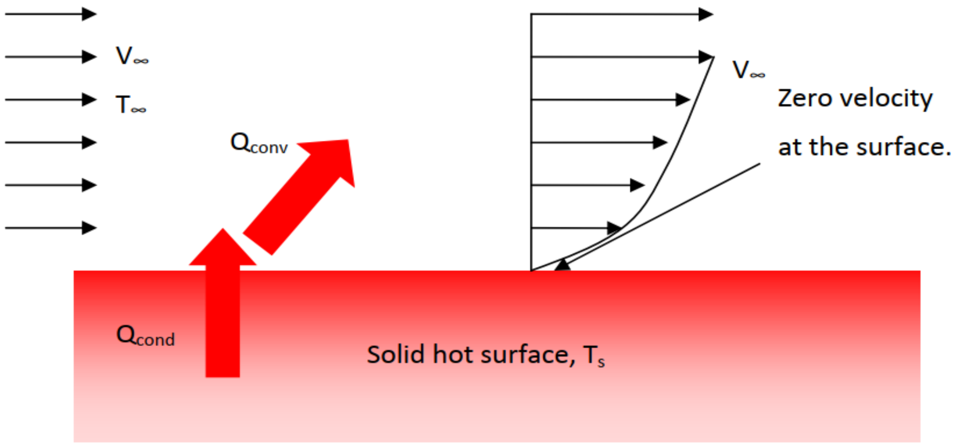


Figure (2.1): Forced Convection.

It is assumed that the velocity of the fluid is zero at the wall, this assumption is called no
slip condition. As a result, the heat transfer from the solid surface to the fluid layer adj acent to the surface is by pure conduction, since the fluid is motionless[4]. Thus,

$$\left.\begin{aligned} \dot{q}_{conv} = \dot{q}_{cond} = -k_{fluid} \left.\frac{\partial T}{\partial y}\right|_{y=0} \\ \dot{q}_{conv} = h(T_s - T_\infty) \end{aligned}\right\} \rightarrow h = \frac{-k_{fluid} \left.\frac{\partial T}{\partial y}\right|_{y=0}}{T_s - T_\infty} \qquad (W/m^2.K)$$

Theoretical Analysis

Assumptions of single tube Heat Exchanger Steady state condition and the fluid mass flow rate and its properties consider constant. The changes are negligible in kinetic and potential energies. The water specific heat can be treated as constant. The heat transfer process by conduction has been neglected along the tube because it is low. The tube outer surface is insulated and the heat generation is negligible. There is no change in phase of flow inside the heat exchanger. Data Reduction steps[6]:

Heat Transfer

In this work, the water used as the test fluid is flowed through a uniform heat-flux and insulated tube. The steady state of the heat transfer rate is assumed to be equal to the heat loss in the test section which can be expressed as Rate[6]:

$Q_{water} = Q_{conv}$

in which

$Q_{water} = \dot{m}\, Cp_{,water} (T_o - T_i) = V.\ I$

$(electric\ power) = I * V$

The heat supplied by electrical winding in the test tube is found to be 3 to 8% higher than the heat absorbed by the fluid for thermal equilibrium test. Thus, only the heat transfer absorbed by the fluid is taken for internal convective heat transfer coefficient calculation. The convection heat transfer from the test section can be written by

$Q_{conv} = h\, A\, (T_s - T_b)$

Where

$T_b = (T_o + T_i) / 2$

And

$T_s = (T_1+T_2+T_3+T_4+T_5) / 5$

where for a constant heat flux, the average surface temperature Ts can be calculated from 5 points of the local surface temperatures, lined equally apart between the inlet and the exit of the test tube. The average heat transfer coefficient, h is estimated as follows

$h = \dot{m}\ C\ p,_{water}\ (T_o - T_i) / A_S\ (T_s - T_b)$

On the other hand, the calculation of a local heat transfer coefficient is based on a specific local wall temperature. Nusselt number can be calculated using the following equation;

$Nu = h\ D / k$

where D is an inner diameter of the test tube and k is a thermal conductivity of the fluid (air).

$Q_h = h_i A_s\ (T_s - T_m)$

$A_s = \pi d_i\ L$

The convection heat transfer coefficient, in general, varies along the flow direction. T he mean or average convection heat transfer coefficient for a surface is determined by (properly) averaging the local heat transfer coefficient over the entire surface.

<u>Flow across Cylinders</u>

The characteristic length for a circular tube or sphere is the external diameter, D, and t heReynolds number is defined:

$$\mathrm{Re} = \frac{\rho V_\infty D}{\mu}$$

The critical Re for the flow across tubes is $2x10^5$. The approaching fluid to the cylin der will branch out and encircle the body, forming a boundary layer.

It is important to clarify some basic definitions, terminologies and criteria that are often used in internal convective heat and mass transfer. These include:

1. Mean velocity, temperature, and concentration

2. Fully developed flow, temperature, and concentration profiles

3. Hydrodynamic, thermal, and concentration entrance lengths

The figure (2-2) shows the development of a velocity profile inside a duct or tube with uniform inlet velocity for laminar flow of an incompressible Newtonian fluid. The velocity profile at some distance away from the tube's inlet no longer changes along the flow direction, where it is referred to as the fully developed flow condition.

The fully developed condition is often met at some distance away from the inlet. However, there are also applications in which fully developed flow is never reached.

Momentum, thermal, and concentration boundary layers form on the inside surface of the tube. The thickness of the layers increases in a similar manner as boundary layer flow over a flat plate (which was presented in detail in external forced convection).

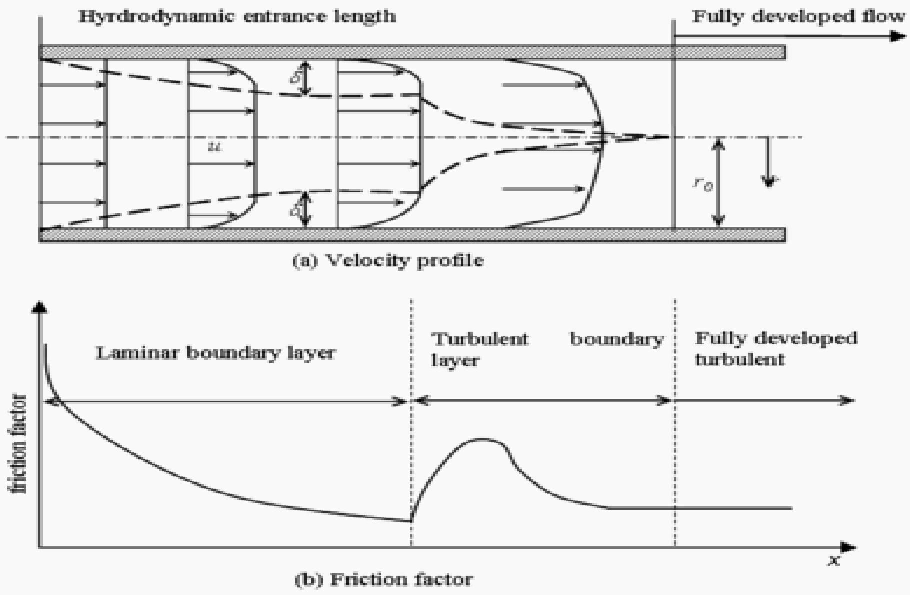
Part (a) of the figure shows how the momentum boundary layer builds up in a pipe along the flow direction. At some distance away from the inlet, the boundary layer fills the flow area. The flow downstream from this point is referred to as fully developed flow since the velocity slope does not change after this point. The distance downstream from the inlet to where the flow becomes fully developed is called the hydrodynamic entrance length. If the flow is laminar (Re < 2300 for flow inside circular tubes), the fully developed velocity is a parabolic shape. It should be noted that the fluid velocity outside the boundary layer increases with x, which is required to satisfy the conservation of mass (or continuity) equation.

The center line velocity finally reaches a value two times the inlet velocity, u_{in}, for fully developed, steady, incompressible, laminar flow inside tubes. It should be noted that the hydrodynamic entrance length for fully developed flow does not start from the point where the friction coefficient,

$$c_f = \frac{\tau_w}{\rho u_{in}^2/2} \tag{1}$$

The friction coefficient variation for laminar flow inside a circular tube with uniform inlet velocity is shown in part (b) of the figure.

The friction coefficient is highest at the entrance and then decreases smoothly to a constant value, corresponding to fully developed flow. Two factors cause the friction coefficient to be higher in the entrance region of tubes than in the fully developed region. The first factor is the larger velocity gradient at the entrance on the wall. The gradient decreases along the pipe and becomes constant before the velocity becomes fully developed. The second factor is the velocity outside the boundary layer, which must increase to satisfy the conservation of mass or continuity equation. Accelerating velocity in the core produces an additional drag force when its effect is considered in the friction coefficient.



Figure(2-2): Velocity profiles and friction factor in turbulent flow in a circular tube.

The turbulent velocity profile and friction coefficient variation for a circular pipe are shown in figure to the right. Even for a very high inlet velocity, the boundary layer will be laminar over a part of the entrance. This transition from laminar to turbulent is clearly shown by the sudden increase in momentum boundary layer thickness as shown in part (a) of the figure. The friction coefficient variation for turbulent flow in a pipe entrance is shown in part (b).

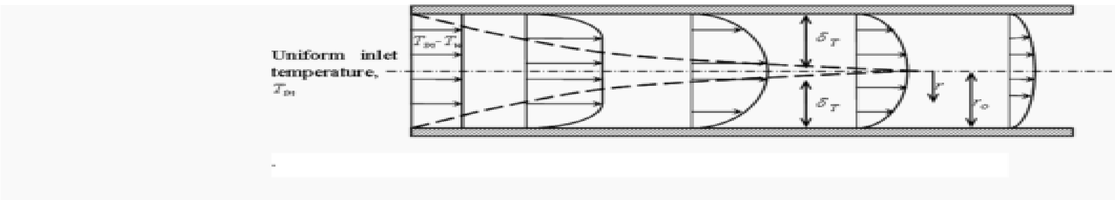
The hydrodynamic entry length required for fully developed flow should be obtained by a complete solution of the flow and thermal field in the entrance region. A rule of thumb to judge whether or not the flow is fully developed for circular pipes is[7]:

$$\frac{L_H}{D} \geq 0.05\mathrm{Re} \text{ for laminar flow} \tag{2}$$

$$\frac{L_H}{D} \geq 0.625\mathrm{Re}^{0.25} \text{ for turbulent flow} \tag{3}$$

where L_H is the hydrodynamic length and the Reynolds number is defined by

$$\mathrm{Re} = \frac{u_m D}{\nu}$$



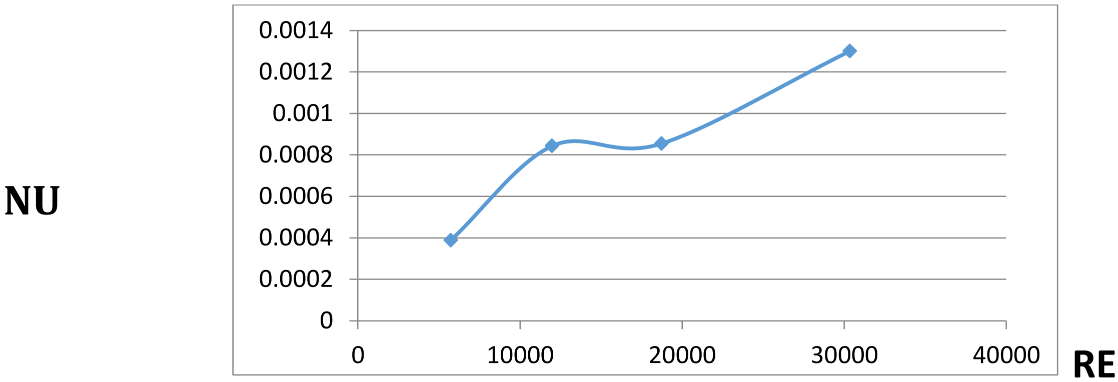
Chapter 5

Result and discussion

Table (5-1):Values of Reynolds numbers and nusselt numbers of water at power =30 watt at different values of flow without twisted tape.

Nu	Re	T5	T4	T3	T2	T1	Q(l/min)
0.000388	5733.945	30	30	31	25	23	10
0.000842	11976.05	31	31	32	26	25	20
0.000854	18738.29	30	30	31	28	26	30
0.0013	30344.41	31	31	32	28	26	40

Figure (5-1):schematic diagram of Reynolds numbers and Nusselt numbers of water at power=30 watt and different values of flow without twisted tape.

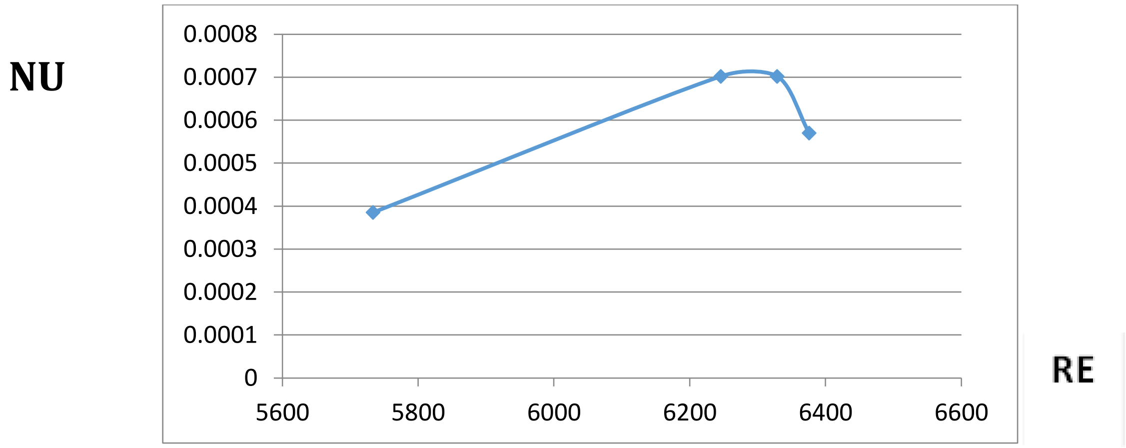


Increasing the water flow by 10, we notice an improvement in the heat transfer. Note that the reading was without the use of the twisted tape with the stability of power 30

Table (5-2):Values of Reynolds numbers and Nusselt numbers of water at Q=10 l/min at different values of power without twisted tape.

Nu	Re	T5	T4	T3	T2	T1	P(watt)
0.000385	5733.945	30	30	31	25	23	30
0.000702	6246.096	33	32	34	27	27	35
0.000702	6329.114	34	32	35	28	27	40
0.00057	6375.925	36	35	37	28	27	45

Figure (5-2):schematic diagram of Reynolds numbers and Nusselt numbers of water at Q=10 l/min and different values of power without twisted tape.

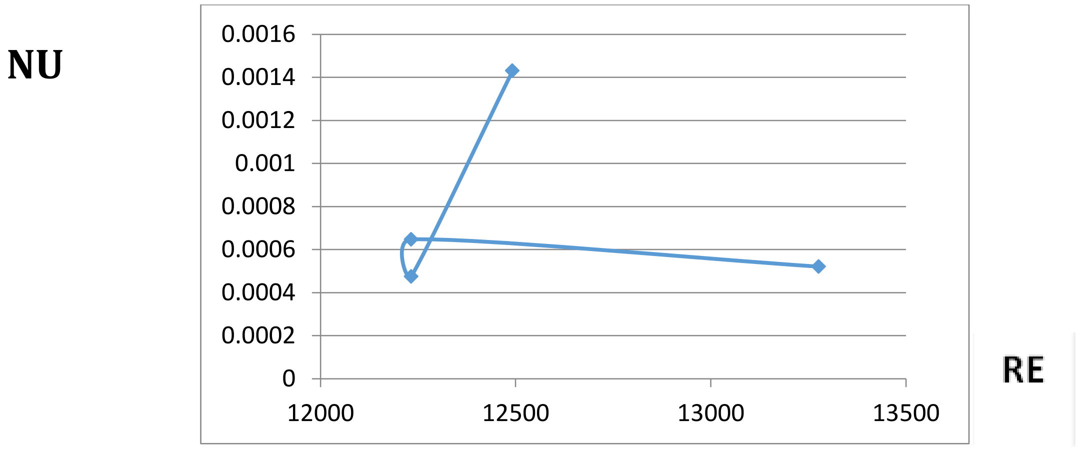


The increase in the capacity by 5 between one reading and another, we notice an improvement in the heat transfer, knowing that the reading without the twisted tape with the stability of the flow is 10

Table (5-3):Values of Reynolds numbers and Nusselt numbers of water at Q=20 l/min at different values of power without twisted tape.

Nu	Re	T5	T4	T3	T2	T1	Power
0.00143	12492.19	30	31	31	25	27	30
0.000475	12232.42	31	31	32	30	28	35
0.000647	12232.42	32	32	33	29	27	40
0.000521	13276.69	35	35	36	33	29	45

Figure (5-3):schematic diagram of Reynolds numbers and Nusselt numbers of water at Q=20 l/min and different values of power without twisted tape.

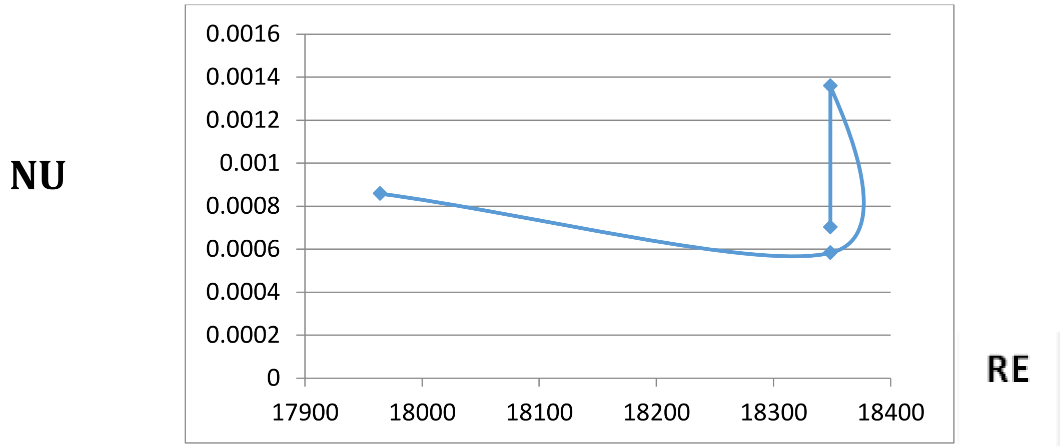


At the flux 20 and the power increased by 5 between one reading and another, we notice an increase in the Reynolds number higher than the previous experience

Table (5-4):Values of Reynolds numbers and Nusselt numbers of water at Q=30 l/min at different values of power without twisted tape.

Nu	Re	T5	T4	T3	T2	T1	power
0.000858	17964.07	30	30	31	28	26	30
0.000583	18348.62	31	32	32	30	28	35
0.001359	18348.62	33	32	33	29	26	40
0.000702	18348.62	31	32	32	29	27	45

Figure (5-4):schematic diagram of Reynolds numbers and Nusselt numbers of water at Q=30 l/min and different values of power without twisted tape.

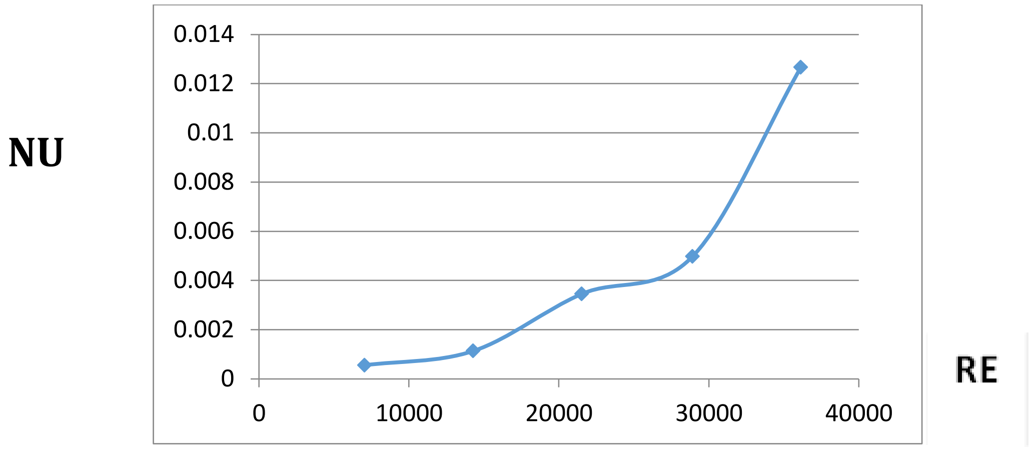


At the flow 30 and the power increased by 5 between one reading and another, we notice that there is an increase in the Reynolds number by a higher amount

Table (5-5):Values of Reynolds numbers and Nusselt numbers of water at Q=30 watt at different values of flow with twisted tape.

Nu	Re	T5	T4	T3	T2	T1	Q(l/min)
0.000553	7039.279	40	41	40	32	32	10
0.001133	14285.71	41	42	40	33	32	20
0.003452	21523.89	42	44	37	33	32	30
0.00497	28910.09	42	43	37	35	33	40
0.012656	36137.61	43	43	38	33	32	50

Figure (5-5):schematic diagram of Reynolds numbers and Nusselt numbers of water at power=30 watt and different values of flow with twisted tape.

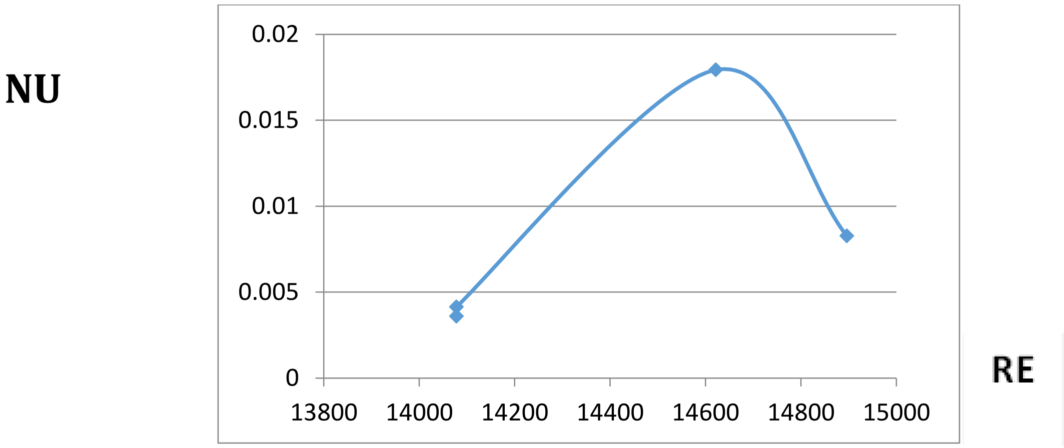


Increasing the water flow by 10, we notice an improvement in heat transfer. Note that the reading was with the use of the twisted tape with the stability of power 30

Table (5-6):Values of Reynolds numbers and Nusselt numbers of water at Q=10 l/min at different values of power with twisted tape.

Nu	Re	T5	T4	T3	T2	T1	Power
0.000362	**6905.124**	41	43	39	35	30	30
0.003432	**7586.102**	48	45	41	37	33	35
0.004784	**7865.345**	49	45	42	38	35	40
0.001595	**7865.345**	49	47	43	39	35	45

Figure (5-6):schematic diagram of Reynolds numbers and Nusselt numbers of water at Q=10 l/min and different values of power with twisted tape.

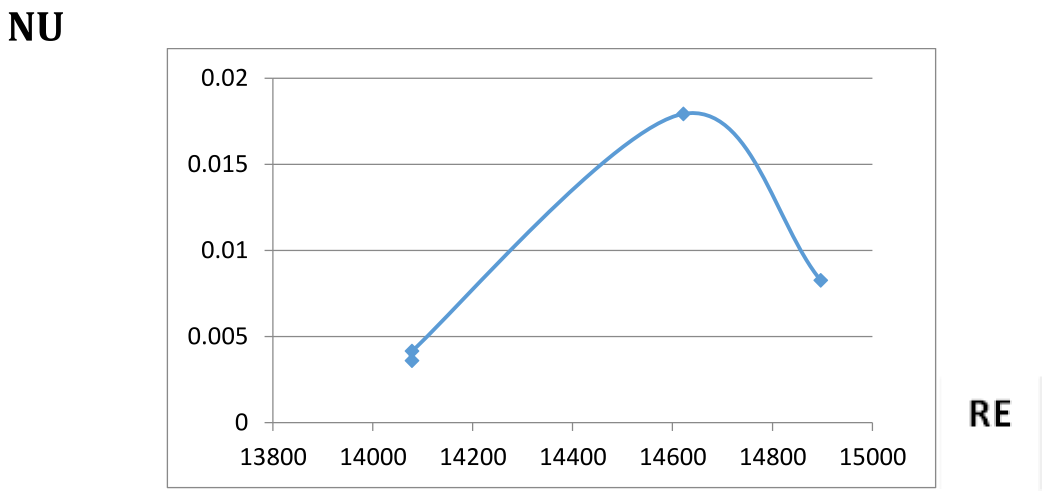


When the flow rate is stabilized and the power is increased by 5, we note that there is a small increase in the Reynolds number, knowing that the experiment with the presence of the twisted tape

Table (5-7):Values of Reynolds numbers and Nusselt numbers of water at Q=20 l/min at different values of power with twisted tape.

Nu	Re	T5	T4	T3	T2	T1	power
0.003593	14078.56	43	41	37	34	30	30
0.004146	14078.56	41	39	38	34	32	35
0.017925	14622.02	45	41	39	35	32	40
0.008262	14896.47	45	42	39	35	33	45

Figure (5-7):schematic diagram of Reynolds numbers and Nusselt numbers of water at Q=20 l/min watt and different values of power with twisted tape.

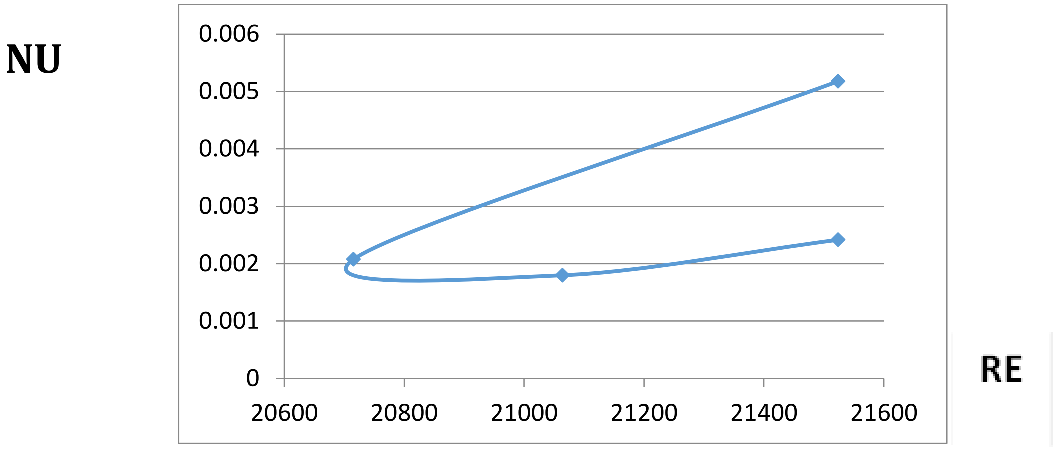


When the flow rate is stabilized by 20 and the power increase by 5, we notice that there is a very small increase in the Reynolds number, knowing that the experiment with the presence of the twisted tape

Table (5-8):Values of Reynolds numbers and Nusselt numbers of water at Q=30 l/min at different values of power with twisted tape.

Nu	Re	T5	T4	T3	T2	T1	power
0.005177	21523.89	42	43	37	33	32	30
0.002076	20715.37	40	40	36	34	30	35
0.001799	21064.46	42	42	38	34	29	40
0.002416	21523.89	44	44	40	33	30	45

Figure (5-8):schematic diagram of Reynolds numbers and Nusselt numbers of water at Q=30 l/min and different values of power with twisted tape.



When the flow rate is stabilized by 30 and the power is increased by 5, we notice that there is a high increase in the Reynolds number, knowing that the experiment with the presence of the twisted tape

Chapter SIX

Conclusions and recommendations of future work

Conclusions

From the present work the following conclusions can be listed:

1. Heat transfer increased with addition of twisted tape.
2. Heat transfer increased with increasing fluid flow
3. The flow of fluid becomes more turbulent with adding twisted tape.

Recommendation for future work:

1. Use another shape of twist.
2. Use twisted tapes as fine with different dimensions.
3. Use spring as turbulator.

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