

Experimental and numerical study of enhancement heat transfer in tube heat exchangers

Ahmed Q Salam

Abstract - Fully-developed turbulent flow for the inner side in "TWISTED TUBES" with different cross sections geometries are experimentally and numerically investigated in the case of constant wall temperature. Two different geometries are considered. The tubes are helically twisted along the axis perpendicular to their cross-section. The helical twist geometry is described by the 360° twist. The results are comparing with tube without twisting "SMOOTH TUBE" in order to point out the heat transfer enhancement and the friction factor increase in the turbulent flow field. Numerical simulations are performed for three - dimensional and fully - developed flow by using RNG k-ε turbulence model. Both experimental and numerical results show that the enhancement in the heat transfer decrease as the Reynolds number increase and the values of friction factor decrease as the Reynolds number increase.

1- INTRODUCTION

In recent years, energy and material saving considerations have promoted an expansion of efforts aimed at producing more efficient heat exchanger equipment through the augmentation of heat transfer. As the efforts to produce more efficient heat transfer equipment continue, an increasing number of augmented technique and surface are being produced commercially. These techniques of enhancement can be classified in two main categories, passive and active techniques, in addition to a hybrid technique which includes two or more from each of passive and active technique [1] and [2]. Over 85% of all new heat exchanger applications in oil refining, chemical, petro-chemical, and power generation are accommodated through the use of conventional shell and tube type heat exchangers. Conventional TEMA (Tubular Exchanger Manufacturers Association) type shell and tube type heat exchangers consist of a number of round tubes attached to a tube sheet inside a cylindrical vessel, with tube sizes, tube lengths, and shell diameters varying depending on the requirements of the application. The tube bundle normally contains a number of baffles to accomplish the dual objectives of providing a support structure for the tubes, and to direct the shell-side flow across the tubes rather than along the tubes. The resulting back and forth shell-side flow will yield a higher than expected pressure drop per unit of heat transfer because energy is used to reverse the flow rather than to enhance heat transfer.

Ahmed Qusai Salam is with mechanical Engineering Dept., University of Technology, Baghdad, Iraq.
E. mail: ahmedqussaysalam_1987@yahoo.com

Also, the energy consumed in reversing the flow will tend to force the shell - side fluid through baffle to-tube and baffle - to-shell clearances yielding lower cross flow and lower heat transfer coefficients. And dead spots, Finally, fluid flow around the baffles is non - uniform resulting in areas of low flow which are prone to fouling accumulation, corrosion, and poor heat transfer.

The thermal effectiveness of a shell and tube exchanger is normally calculated assuming perfect radial and no axial mixing of the shell side stream. In practice however, there is considerable axial mixing within a baffle compartment, and further, the stream is in cross-flow for part of the time rather than axial flow. These effects are further complicated by leakage of flow that occurs at the baffle-to tube and baffle-to-shell joints that does not take full part in the heat transfer in the bundle. The overall effect of these limitations is the actual thermal effectiveness will be lower than the theoretical value, and it will be lower than the values obtained for other types of heat exchangers that do not suffer from these limitations. Typically, thermal effectiveness of a conventional shell and tube type exchanger will be in the range of 60% to 80%.

1-1 Tube Thermal Applications:

Bergles [3] shows the cross-section of eight commercially available tubes with internal enhancement and two with external enhancement are shown in Figure (1 and 2). These figures illustrate a few of the many enhanced tubes which are available commercially. It should be noted that some tubes have enhancement on both side. In fire tube boiler where the flue gasses pass inside the tube and the boiling water takes place at the external surface, the augmentation should focus on the internal side where low heat transfer coefficient exists due to the lower thermal conductivity of the gas. Also the heat is transferred by convection in most of the tubes rather by radiation as it is in the combustion chamber and in the first pass of the boiler. In shell and tube heat exchanger the heat transfer coefficient on the shell side (flow over the tubes) is 100-200% greater than the convective heat transfer coefficient inside the tube for the same Reynolds number. This means that the enhancement heat transfer at the internal tube surface is necessary to reduce the overall resistance of heat transfer.

The limitation to enhanced tube used in shell and tube exchanger or boilers suggests several important restrictions. First, the maximum outside diameter of any

tube is less than that of the tube-hole in the tube-sheets. This constraint is a consequence of the common procedure in the fabrication and repair of the shell- and tube heat exchanger of inserting or pulling the tube through the tube-holes in the tube-sheets. Enhanced tubes typically have plain-end section so that they can be fastened securely to the tube-sheets by rolling or welding. Secondly the enhancement technique should allow an access alone to the tube length during the maintenance operation especially, when the mechanical cleaning is necessary. These restrictions make the method of surface roughness or extend surface is limited and the insert turbulent promoter such as the helical wire or twisted tap is the first candidate to enhance the heat transfer in these equipment.

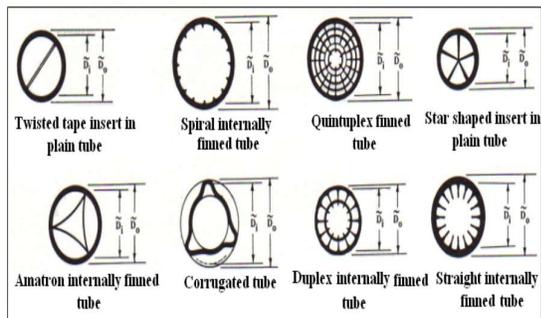


Fig.1, commercially available tubes with internal enhancement

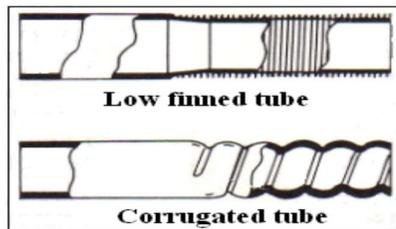


Fig. 2 commercially available tubes with external enhancement

The surplus of the pressure drop induced by these promoters (twisted tape or the helical wire) is due to the drag force induced by their obstacle shapes and the friction force due to the additional surface. This means that less number of obstacles in the flow stream and the small surface area are required to create the swirl flow, the less pressure drop and lower cost of material used. After we looked typical problems with shell and tube heat exchangers then we solved them with a revolutionary tube shape and bundle construction. In the present work the swirl flow is created by the tube having a shape similar to the screw shape. It is twisted tube. " TWISTED TUBE " technology has to be more efficient, reliable, and trouble - free than any other exchanger.

1-2 Simple topics on " TWISTED TUBE ":

1-2-1 - More efficient heat transfer:

" TWISTED TUBE" heat exchanger expect a higher heat transfer coefficient than any other type of tubular heat exchanger . Here's why:

a – Complex swirl flow on the shell side induces the maximum turbulence to improve heat transfer.

b – Powerful tube side turbulence is achieved even at high viscosities and \ or low velocities.

c – Uniform flow distribution gives more effective length and surface area than shell and tube exchangers.

1-2-2 - Baffle – free tube support:

The "TWISTED TUBE" design avoids the need for baffles. The unique helix shaped tube is arranged in a triangular pattern. Each tube is firmly and frequently supported by adjacent tubes , as shown in figure 3 , yet fluid swirls freely along its length. This support system eliminates tube vibration, which is common problem in some heat exchanger services. The twisted arrangement for baffle – free support with gaps aligned between the tubes also expect for easier cleaning on the shell side . The " TWISTED TUBE " heat exchangers are round at each end, allowing for convectional tube –to-tube sheet joints to be used.

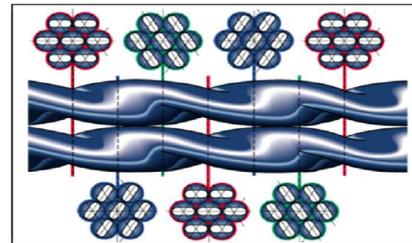


Fig.3, bundle construction of the "TWISTED TUBE" heat exchanger

1-2-3 - "TWISTED TUBE" cleaning efficiency:

1-2-3-1 Shell side:

a – Cleaning lanes allow complete mechanical cleaning by hydro blasting .

b – Chemical cleaning - in – place (CIP) is more effective in "TWISTED TUBE" heat exchangers than convectional S & T due to uniform flow distribution.

1-2-3-2 Tube side:

a – Tube side effectively cleaned by hydro blasting.

b – Chemical cleaning - in –place (CIP) is more effective in" TWISTED TUBE" heat exchangers than convectional S & T due to swirl flow.

c – No special tools required.

1-2-4 Improving flow:

1-2-4-1-Improving tube flow:

a – "TWISTED TUBE" technology expect the highest heat transfer coefficient possible in tubular heat exchanger.

b – Swirl flow in tubes creates turbulence to improve heat transfer as shown in the figure 4.

c – Turbulent flow achieved even at low velocities and \ or high viscosities.

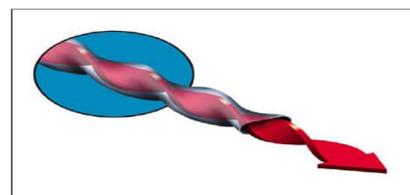


Fig.4 Show the tube side flow in "TWISTED TUBE" heat exchanger

1-2-4-2 Uniform Shell side Flow:

- a – Heat transfer coefficient is consistently high.
- b – Complex interrupted swirl flow on shell side maximizes turbulence while minimizing pressure drop as shown in the figure 5.
- c – Flow distribution and velocity are homogeneous.

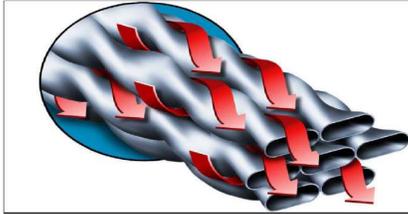


Fig.5 Show the shell side flow in "TWISTED TUBE" Heat exchanger

1-2-5 Improving efficiency:

1-2-5-1 – Increased heat transfer coefficient:

- a- Swirl flow creates turbulence resulting in higher tube side coefficient.
- b- Uniform fluid distribution combined with interrupted swirl flow results in optimized shell side coefficient.

1-2-5-2 Lower pressure drop

- a- The longitudinal swirl flow of " TWISTED TUBE " technology reduces the high pressure drop associated with segmental baffles.
- b- " TWISTED TUBE " heat exchangers are usually shorter in length and have fewer number of passes, for a lower pressure drop on the tube side.

1-2-5-3 - No Vibration

- a- Baffle – free design directs shell side fluid to true longitudinal flow.
- b- Each tube using " TWISTED TUBE " technology is extensively supported at multiple contact points along its entire length.
- c- Tube fretting and failure due to vibration is eliminated.

1-2-5-4 - Reduced fouling

- a- Baffle – free design eliminates dead spots where fouling can occur.
- b- Velocity is constant and uniform.
- c- Constant flow distribution controls tube wall temperature.

Figure 6 show the comparison between conventional shell & tube and "TWISTED TUBE" heat exchanger.

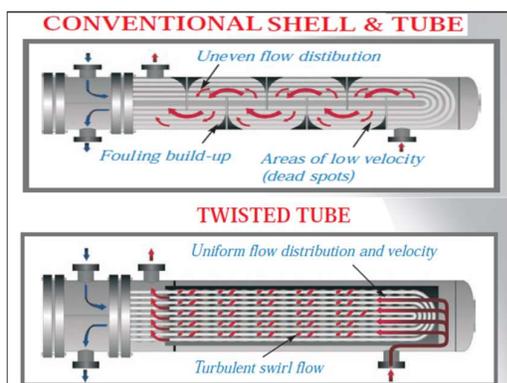


Fig. 6 Show the comparison between conventional shell & tube and " TWISTED TUBE " heat exchanger

2-Experimental Study:

This system consists of two main parts; air flow system and steam generation system as shown in Fig. 7.

2-1- Air flow system:

Centrifugal fan with impeller diameter (500mm) are used to supply the air to the test section. The fan is drive by an electric motor of (3 hp)(3000 r.p.m). The air supply is controlled by a slide vane located at the inlet section. The inlet section diameter is (120mm) and the outlet section diameter is (88.9mm). The outlet section is joint to the connected tube I as shown in the figure 8. Connected tube I is a galvanized steel of (500mm) length ,(88.9mm) outer diameter and (83mm) inner diameter, jointed at beginning to air blower with flange of (128mm)outer diameter and (88.9mm)inner diameter and four bolts of (10mm) diameter .This tube connected to the orifice meter by the reducing tube-joint I from 3.5inch to 1.5inch see figure 8. The orifice meter is joint to the connected tube II by the reducing tube-joint II from inch 1.5inch to 0.75inch as shown in the figure 9. Connected tube II is the Galvanized steel of (250mm) length, (23mm) outer diameter and (20mm) inner diameter, jointed to the orifice meter by the reducing tube-joint II from 1.5inch to 0.75inch as shown in the figure 9. This tube connected to Inlet air tube by union joint.

Inlet air tube is a carbon steel of (1060mm) length, (21mm) outer diameter, (18mm) inner diameter joined at beginning to the connected tube II by union joint and to the testing tube from the other side. Three types of tubes of carbon steel of (1600mm) length are used to examine in the experimental work. As shown in the figure 10. "SMOOTH TUBE" of (18mm) inner diameter, "TWISTED TUBE"NO.1 (16.06789034mm) hydraulic diameter and "TWISTED TUBE" NO.2 (13.00966033mm) hydraulic diameter . Each tube is welding on the both sides with two bushes (70mm) outer diameter. Each bush is made from iron shaft (70mm) outer diameter; the iron shaft is cutting to six parts by the electric hack saw, and drilling in the centre part to make a hole across the length. This hole allow to joint inlet air tube and outlet air tube from both sides with two bushes by two flanges and bolts .Outlet air tube is a carbon steel (of (360mm) length, (21mm) outer diameter, (18mm) inner diameter joined at beginning to the test section by flange and bolts and from the other side to the atmosphere.

2-2- Steam Generation System:

This system is used to obtain constant wall temperature, the system is connected to air flow system with shell which receives steam and surrounds test tube. To generation steam, we used water tube boiler, consists of closed cylinder of (1700mm) length, and (800mm) diameter made from iron plate of (10mm) thickness. This cylinder consisted of helical tube as path of water. To heating, Burner system is located above the cylinder and flame is heating the water into helical tube and generation steam. To adjust pressure value and amount of steam

required continuously, pressure switch is in series connected to control electric circuit of Burner system and pump water (which delivers water to the boiler). Electric circuit of Burner system and pump water set to cut-off current when the pressure reaches to the maximum limit (3.5bar). Also there is a safety valve in emergency and set to open at higher limit (4bar). The water is delivered to water pump through (0.75 inch) diameter tube. The steam is delivered to steam shell through (0.5 inch) diameter tube, and there is a manual valve for adjusting steam amount. The test section includes steam shell which made from galvanized tube of (1600mm) length (88.9mm) outer diameter and (83mm) inner diameter. This shell fixed in two flange {(110mm) of length and (140mm) outer diameter} on two sides by welding. These flanges are made from disks of iron (120mm) length, (150mm) outer diameters and designed to contain corner in the inner diameter of the flanges by (lathe machine). This corner has a shape of hollow disk with (95mm) outer diameter equal to the inner diameter of the flanges and (70mm) inner diameter equal to the outer diameter of the bushes (with -0.4mm clearance for easy setup) as mentioned in air flow system, these bushes is putting into the inner diameter of the corner. To avoid steam of high pressure leaking from terminals of steam shell, two rings of thermal rope were used between inner diameter for flanges of steam shell and bushes of air flow system. The thermal rope was pressed and tightened with two metal rings and with the corners from the other side. These rings are made from disks of iron (150mm) outer diameter, and made hollow and supplied with flanges by (lathe machine). These rings with (95mm) outer diameter equal to the inner diameter of the flanges of steam shell (with -0.4mm clearance for easy setup) and (70mm) inner diameter equal to the outer diameter of the bushes (with -0.4mm clearance for easy setup). The flanges of the rings with (140mm) outer diameter equal to the outer diameter of the flanges of steam shell. The flanges of the rings were fixed with the flanges of steam shell by bolts. The heat is exchanged between air in test tube and steam in annulus of steam shell. Four tubes (0.5 inch) diameter are branched from steam shell, the first to measure steam pressure with a Borden gauge connect to it, the second to steam inlet, the third to relieve air which may stay in steam shell, and the last to reject steam condensate as water drop downward as shown in the figure 11. A hole of (10mm) diameter was made to put thermocouple to measure temperature of steam. To calculate the Air flow rate, orifice meter was made according to British standard NO. 1042-1981 with two orifice plates. One is the small area ratio (0.1) for low flow rates and the other is the large area ratio (0.2) for high flow rates, also two pressure tapings were made to measure pressure difference across the orifice plate, the pressure tapings are connected to water manometer. Temperatures are measured by using two portable digital Thermometers, one is produced by Twintex (TH-621) company and the other is Lutron (YK-2005AH) company used of thermocouple, see Figure 11. Thermocouples used during the experiment were Chromel - Alumel (Cr -Al) (K- Type, range: 0°C to 800°C). One thermocouple (type K pin-wire TP-01) and two thermocouples (type K wire TP-01) were used for temperature measurements; two types of pressure

instrument were used. Borden gauge produced by (MONEKS) with range (0-4 bar) was used to measure steam pressure in steam shell and two water manometers of range (1000 mm H₂O) to measure pressure difference across orifice plate, and pressure drop through testing tubes.

3 - Calculations:

3-1- Volume discharge, mass flow rate, and mean velocity^[4]:

3-1-1-Volume discharge:

$$Q = C.A_o \sqrt{\frac{2(p_2 - p_1)}{\rho}} \quad (1)$$

where C is discharge coefficient.

3-1-2-Mass flow rate of air:

$$\dot{m} = Q \cdot \rho \quad (2)$$

3-1-3- Mean velocity:

$$u_m = \frac{\dot{m}}{A_p \cdot \rho} \quad (3)$$

3-2 – Heat transfer calculation:

3-2-1 –Heat transfer coefficient^[5]:

$$\bar{h} = \frac{\dot{m} \cdot c_p \cdot (T_{ao} - T_{ai})}{A_s \cdot (LMTD)} \quad (4)$$

Where (LMTD) represents logarithmic mean temperature difference, it is calculated as follow:

$$LMTD = \frac{(T_s - T_{ao}) - (T_s - T_{ai})}{\ln((T_s - T_{ao}) / (T_s - T_{ai}))} \quad (5)$$

3-2-2- Nusselt number (NU)^[6]:

$$Nu_D = \frac{\bar{h} D_h}{k_f} \quad (6)$$

Then, we have calculated Nusselt number for smooth tube from for comparison:

$$Nu_D = 0.023 Re_D^{4/5} Pr^n \quad (7)$$

Dittus –Boelter equation

where (n=0.4) for heating, and (n=0.3) for cooling.

3-2-3- friction factor (f)[7] :

$$\frac{\Delta P}{\gamma} = f \frac{L}{D} \frac{\bar{v}^2}{2g} \quad (8)$$

Darcy- Weisbach equation

Then, we have calculated friction factor for flow in "SMOOTH TUBE":

$$f = \frac{0.316}{Re^{0.25}} \tag{9}$$

Blasins equation (for $Re \leq 10^5$)

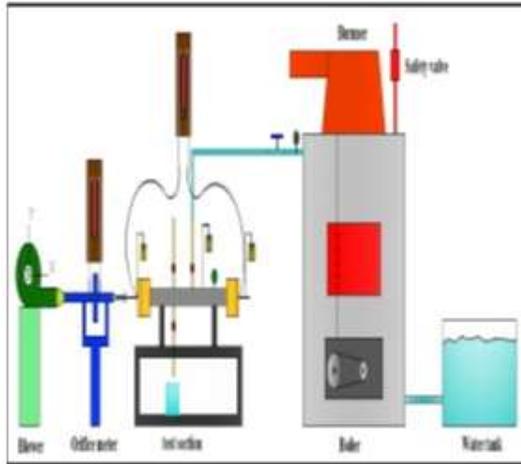


Fig.7, the schematic diagram of the testing apparatus



Fig.8, the photo of air blower system

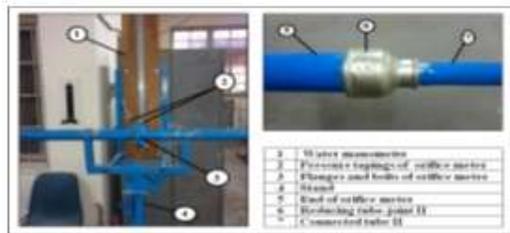


Fig. 9, the photo of orifice meter system



Fig.10, the testing tubes



Fig.11 Show the photo of the test section

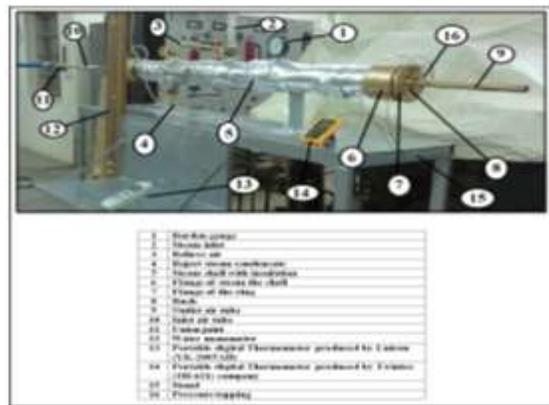


Fig. 13, the photo of the testing apparatus

4-Numerical Simulation:

FLUENT commercial CFD codes are used and all the cases studied. Her, two cases are chosen to simulate the "TWISTED TUBES" where are examining in the experimental study. All cases simulated are 3-D, steady, segregated and air is the working fluid. The equations solved for each case are continuity equation, momentum equation, energy equation and turbulence model. As a result of the curvature of spiral channels, the "TWISTED TUBE" generates rotation of the flow within the channel and the core flow. The core flow is basically in a stable, solid body rotation. In the region between the core flow and the spiral channel flow, interchange of the angular momentum occurs, resulting in the decrease of the angular momentum in the channels. This is the primary cause of instability.

The twisting of the tube causes an appreciable secondary flow in the region between the core flow and spiral channel flow while in the centre part of the cross – section (core flow) , this secondary flow is trivial. This instability enhances the turbulent exchange in the region between the core flow and the spiral channel flow, resulting in improved heat transfer.

Case 1: ("TWISTED TUBE") NO.1: RE=24000

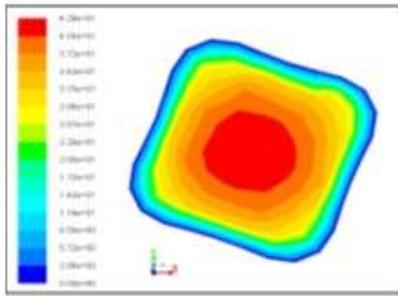


Fig.14, the contours of axial velocity profile at outlet

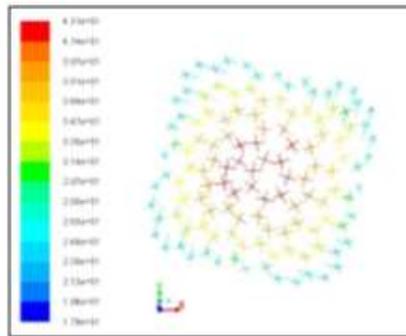


Fig.15, the vectors of axial velocity profile at outlet coloured by axial velocity

Case 2: ("TWISTED TUBE") NO.2: RE=24000

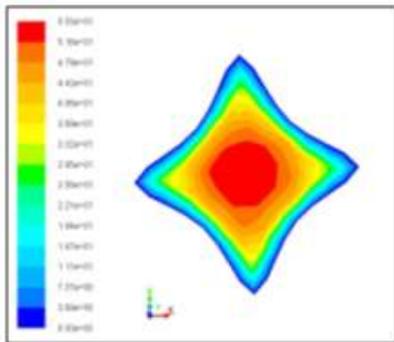


Fig.16 Show the contours of axial velocity profile at outlet

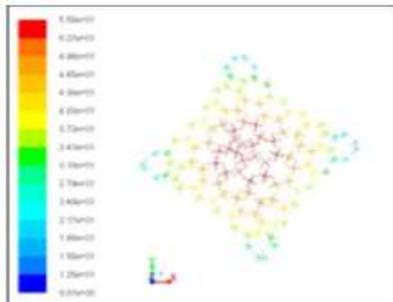


Fig.17 Show the vectors of axial velocity profile at outlet coloured by axial velocity

5- Experimental results:

The experimental investigation imposes a study on three types of tubes one is the "SMOOTH TUBE" and two types of "TWISTED TUBES". Nusselt number and friction factor are used to specify the heat transfer and pressure drop respectively. For the range of Reynolds number from 10000 to 24000 experimentally study can be discussed her. For "SMOOTH TUBE" the Nusselt number increase as the Reynolds number increase for two cases experimental and theoretical as shown in figure 18. The maximum error between the experimental and theoretical Nusselt number is 3.1 %. The friction factor decrease as the Reynolds number increase for two cases experimental and theoretical as shown in figure 19. The maximum error between the experimental and theoretical friction factor is 2%. Figure 20 and 22 show the variation in Nusselt number with Reynolds number for "TWISTED TUBE" NO.1 and NO.2 respectively. The Nusselt number increase as the Reynolds number increase .When the Reynolds number increase from 10000 to 24000 ,the enhancement in the heat transfer (Nusselt number) relative to the " SMOOTH TUBE" decrease from 1.2 to 1.015 for " TWISTED TUBE " NO.1 and from 1.36 to 1.06955 for " TWISTED TUBE " NO.2 . Figure 21 and 23 show the variation of friction factor with Reynolds number for "TWISTED TUBE" NO.1and 2 respectively from these figures we see that the values of friction factor decrease as the Reynolds number increase.

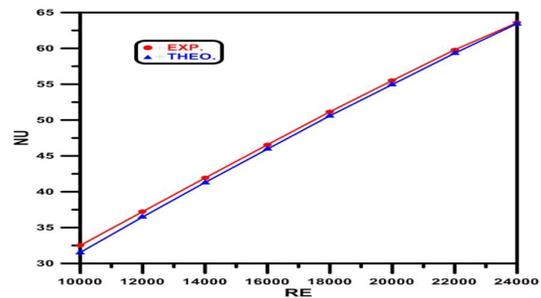


Fig.18 Show the relationship between NU & RE for "SMOOTH TUBE" Experimental and theoretical results

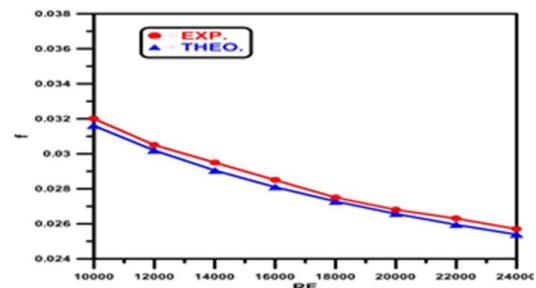


Fig.19 Show the relationship between f & RE for "SMOOTH TUBE" Experimental and theoretical results

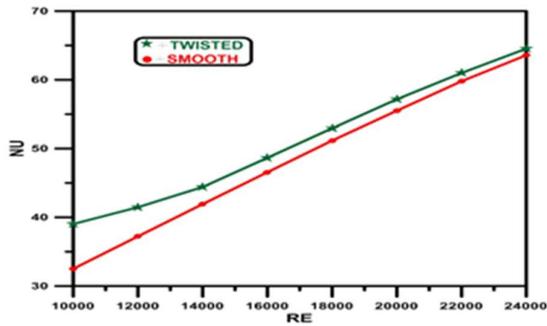


Fig.20 Show the relationship between NU & RE for "TWISTED TUBE"NO.1 "SMOOTH TUBE"

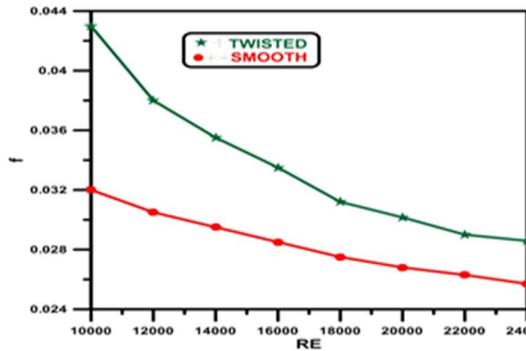


Fig.21 Show the relationship between f & RE for "TWISTED TUBE"NO.1 & "SMOOTH TUBE"

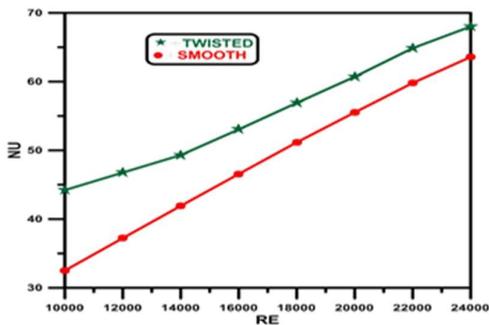


Fig.22 Show the relationship between NU & RE for "TWISTED TUBE"NO.2 & "SMOOTH TUBE"

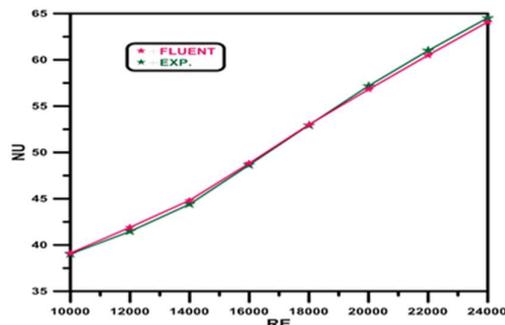


Fig.23 Show the relationship between f & RE for "TWISTED TUBE" NO.2 & "SMOOTH TUBE"

6-Validation between experimental results and numerical (FLUENT) results:

The values of many parameters in FLUENT post processing can be evaluated. The most important parameters in present study that can be evaluated is the Nusselt number and friction factor for each case and the experimental results are plotted to compare them. From the comparison we see that the maximum error between the experimental and numerical Nusselt number is 1.054 % for "TWISTED TUBE" NO.1 as shown in the figure 24 and 1.405 % for "TWISTED TUBE" NO.2. Also we see that the maximum error between the experimental and numerical friction factor is 1.316 % for " TWISTED TUBE " NO.1 as shown in the figure 25 and 3.08 % for " TWISTED TUBE " NO.2 .

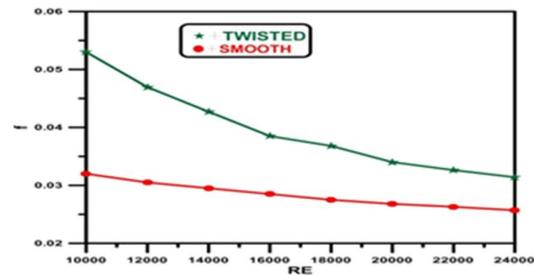


Fig.24 Show the relationship between NU & RE for "TWISTED TUBE"NO.1. Experimental results & numerical results obtained from FLUENT

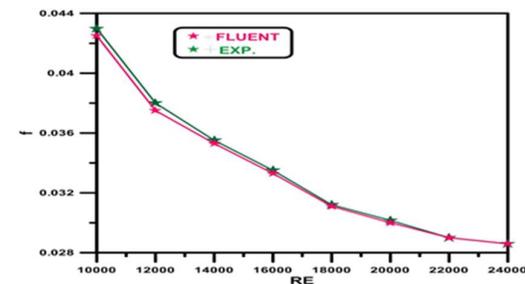


Fig.25 Show the relationship between f & RE for "TWISTED TUBE"NO.1. Experimental results & numerical results obtained from FLUENT

7-Developing of empirical equations:

Empirical equations for different cases studied in experimental work can be developed to calculate Nusselt number and friction factor by using the regression method to evaluate the constants of equations assumed.

For "TWISTED TUBE" NO.1:

$$NU = 0.1727996983 * Re^{0.5998799932} * Pr^{0.4}$$

accuracy = 0.984005

$$f = 3.095099908 * Re^{-0.4670897442}$$

Accuracy = 0.985768

For "TWISTED TUBE" NO.2:

$$Nu = 0.4583607913 * Re^{0.5081149535} * Pr^{0.4}$$

accuracy = 0.980829

$$f = 13.71919297 * Re^{-0.6045658432}$$

Accuracy = 0.994953

8- Conclusions:

1- More complex problems can be solved and design of complex geometries be made and very detailed information of its performance be obtained in economical way by skilful use of commercial computational CFD codes, and ability to optimize the results obtained before manufacturing the system part required.

2- The validity of the results numerically simulated greatly depends on suitability of commercial code to the application studied.

3- The use of "TWISTED TUBE" NO.2 produces maximum enhancement of heat transfer over maximum pressure losses. On the other hand minimum enhancing in heat transfer is obtained over minimum pressure losses occurs in "TWISTED TUBE" NO.1

4- The vortices induced between the core flow and spiral channel flow are more useful for enhancing the rate of heat transfer than those are induced only in core flow or in the spiral channel flow.

9- Nomenclature:

9-1 English symbols:

A_o = Orifice cross section area(m^2), A_p = Pipe cross section area (m^2)
 A_s = Pipe surface area(m^2), C = Discharge coefficient
 C_p = Specific heat at constant pressure($kJ/kg \cdot ^\circ C$), f = friction coefficient
 \bar{h} = Average heat transfer coefficient($W/m^2.K$), k_f = Fluid thermal conductivity($W/m.K$)
 k = Turbulent Kinetic Energy(m^2/s^2), L = Test section length(m), \dot{m} = Mass flow rate(kg/s)

Nu = Nusselt number, Pr = Prandtl number, P = pressure(pa), Q = Volumetric discharge(m^3/s)

Re = Reynolds number, T = Temperature($^\circ C$, K), \bar{u} = Mean velocity(m/s).

9-2 Greek symbols:

ρ = Density kg/m^3 ,

μ = Dynamic viscosity $kg/m.s$,

\mathcal{E} = Turbulent energy dissipation rate m^2/s^3

9-3 Sub-Script:

ai = Air inlet , ao = Air outlet , w = Wall

9-4 Abbreviations:

CFD	Computational Fluid Dynamics
FLUENT	Fluid and Heat Transfer Code.
LMTD	Logarithmic Mean Temperature Difference
RNG	Re Normalization-Group

10- Refrance:

- [1] Bergles, A. E., "Heat transfer enhancement – the encouragement and accommodation of high heat fluxes", *J. Heat Transfer* 119, 8-19. 1997.
- [2] W.Q. Tao, Y.L. He, Q.W. Wang, Z.G. Qu, and F.Q. Song "A unified analysis on enhancing single phase convective heat transfer with field synergy principle" *International Journal of Heat and Mass Transfer* 45 (2002) 4871–4879.
- [3] W. J. Marner, A. E. Bergles, J. M. Chenoweth "On The Presentation of Performance Data for Enhanced Tubes Used in Shell-and-Tube Heat Exchangers" *Transaction of ASME J. of Heat Transfer* May 1983 Vol. 105 p.p. 358-365.
- [4] Victor, L. Streeter, E. Benjamin Wylie "Fluid Mechanics" Textbook Seventh edition by McGraw-Hill, Kogakusha, LTD. 1984
- [5] j. p. Holman "Heat Transfer" Textbook, Fourth edition by McGraw-Hill, Kogakusha, LTD 1976.
- [6] Frank B. Incropera, and David B. Doot, "Principles of Heat Transfer" Mc.Graw Hill Co. (1986).
- [7] Robert W. Fox and Alan T. McDonald, "Introduction to fluid mechanics", Fifth edition by John Wiley & Sons, Inc. (1998).