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NUMERICAL INVESTIGATION FOR HEAT TRANSFER ENHANCEMENT IN A TRIANGULAR TWISTED TUBE

Khudheyer S. Mushatet and Haiyder M. Hmood
Department College of Engineering University of Thi-Qar, Nassiriya, Iraq
E-Mail: khudheyersalim@gmail.com

ABSTRACT

This paper is focused on turbulent heat transfer, enhancement in twisted triangular tube heat exchangers. Triangular twisted tube with (1m) length and (0.03m) diameter was tested for different twist ratios (5, 10 and 20) and Reynolds numbers (Re=5000 to 25000) consequently. The governing equations used in the analysis of the flow field and heat transfer are the momentum, continuity and energy equations. The numerical solution for turbulent flow field and heat transfer is governed by the techniques of a finite volume method with ANSYS Fluent 17.1. The computational data indicated that the twisted tube showed increasing in heat transfer as compared with a plain tube. The twisted ratio was the main affecting parameter for heat transfer improvement and friction losses. The numerical results are compared with published results and an acceptable agreement is achieved.

Keywords: heat exchanger, twisted tube, CFD.

1. INTRODUCTION

Heat, exchangers is a widely used in several industrial and engineering applications, for this, increasing the efficiency of this equipment needs to systemize some designing parameters. Heat transfer c0efficient and pressure drop play vital role in the enhancement of heat exchangers performance [1]. Consequently, improvement of heat exchanger is done by controlling of conditions boundary, flow geometry and the area of heat transfer. For examples increasing the surface roughness and the heat transfer tend to increase the coefficient of heat transfer. Also, any inserts and designing create a swirl flow like taps inserting and tube twisting will lead to increase the turbulence which in turn to increase heat transfer [2]. Many efforts were done in this field to progress the heat efficiency of heat exchanger. Li Zhang *et al*. [3] made an experiment on the condensation heat transfer features of steam flow through the elliptical horizontal twisted tube and circular smooth tube at different temperature of condensation. It was found that the circular twisted tube has a large condensation coefficient of heat transfer than the other tested sections. Four different types of construct twisted tube heat exchangers were studied by Zhang *et al.* [4]. It was found that the geometry volume and Reynolds number had a considerable effect on heat transfer and friction losses in both the tube side and the shell side of twisted tube. The longitudinal vortex created by the wall of the twisted tube enhanced the interaction between the gradient of temperature and vector of velocity that caused increase in the heat transfer efficiency. Wang *et al*. [5] advanced a practical analysis of shell and tube heat exchanger to overcome some of limitations by using twist tubes technology to improve the performance characteristics of conventional types of heat exchanger. The important results showed that the twisted tube heat exchanger type provides a higher performance than conventional types. Sriharsha *et al.* [6] presented experimentally study to compare between the oval type tubes and circular type tube in a cross-flow method. The results showed that the oval tube gives higher heat transfer than the circular type tube. Wongcharee *et al.* [7], introducing experiential for studying the heat transport amelioration and friction factor in a triangle twisted tube. The studied torsion ratio was 5, 10 and 20. Xiang-hui Tan *et al*. [8] offered an experiential and numerical study to evaluate the thermal performance of twisted oval tube heat exchanger. A comparison with smooth tube was done in addition to study the effect of geometric parameters on the performance of twisted tube. They found that with using twisted oval tube, the heat transfer was improved, but with increasing the pressure drop. The coefficient of heat transfer and the friction factor increased with increasing the axis ratio (a/b) at (a) inner major axis of the twisted oval tube and (b) inner minor axis of the twisted oval tube. Salam [9] studied experimentally and numerically the turbulent flow in twisted tube with different cross section geometries and constant wall temperature. He concluded that the heat transfer was increased and the friction factor increased with using twisted tube compared with a smooth tube. Thianpong *et al*. [10] presented an experimental study to investigate the compound heat transfer and the behaviors of friction by using a typical twisted tube with three twisted ration (3, 5 and 7) and Reynolds number ranges from (12000 to 44000). They found that there is an increasing in friction factor and the heat transfer in this combined system decreased with decreasing the ration of twisted (y). Vivek *et al*. [11] discussed the overall heat transfer coefficient in a twisted elliptical tube. They noted from the experiential and numerical results that there is an increases in the heat transfer rate and the flow resistance with decreasing the aspect ratio and pitch length. Yang and Zhi-Xin [12] presented an numerical study for an elliptical twisted tube with laminar flow to investigate the heat transfer rate. They concluded that there is an growth for laminar flow in convection heat transfer with using elliptic twisted tube compare with the smooth. Khudheyer and Farah [13] studied heat transfer augmentation in a heated circular tube by using the combined twin twisted tape and nano fluid. The obtained results showed that a counter clockwise twisting tape offered better thermal

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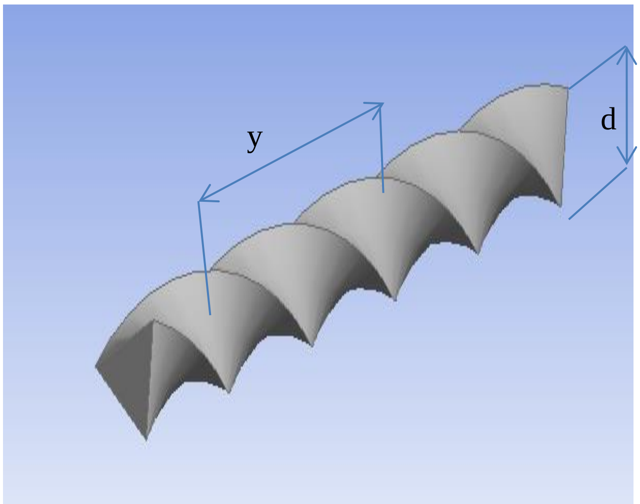
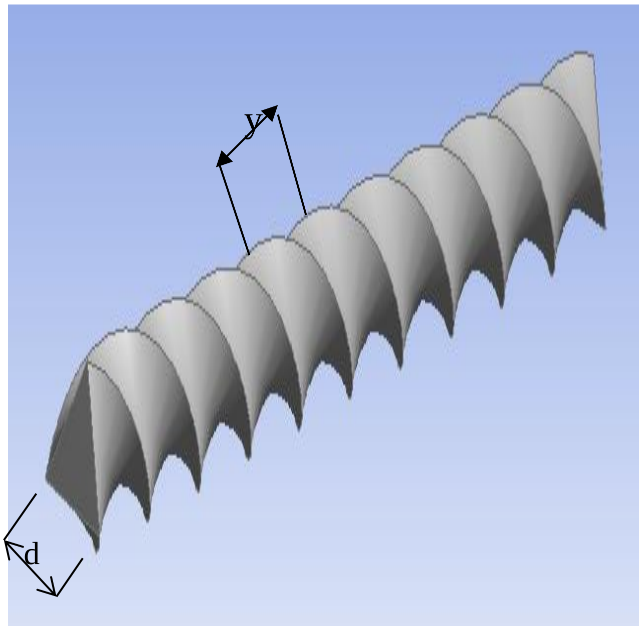
performance than clockwise twisting. Khudheyer and Hayder [14] investigated intensification of heat transfer in a circular and square twisted tubes. They showed there is an intensification in heat transfer when using a twisted tube as compared with a plane tube.

2. PHYSICAL MODEL

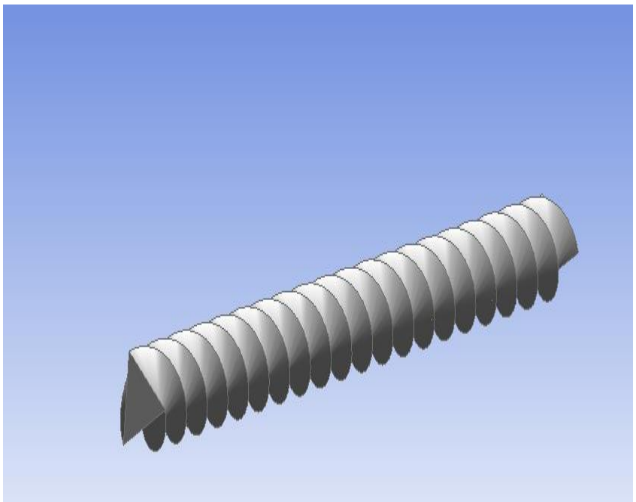
The physical model consists of a twisted triangular tube as shown in Figure-1. Three values of twist ratio (Tr=20, 10 and 15) were examined at different values of Reynolds number (5000 to 25000).The length and hydraulic diameter 0f the tube are (L= 1m, D_h= 0.03m).



a. Twisted ratio =20



b. Twisted ratio =10



c. Twisted ration=5 D

Figure-1. Schematic picture for the material problem.

3. MATHEMATICAL MODEL

The considered presumption was made as follows [15]:

a) Steady state.

b) A constant inlet velocity.

c) The fluid is incompressible with constant thermo physical properties.

d) The body forces are neglected

The governing considered equations [16] are as follows:

$$\frac{\partial}{\partial x}\rho u+\frac{\partial}{\partial y}\rho v+\frac{\partial}{\partial z}\rho w=0 \qquad (1)$$

$$\rho\left(\frac{\partial u^2}{\partial x}+\frac{\partial uv}{\partial y}+\frac{\partial uw}{\partial z}\right)=-\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}\left(2\mu_{eff}\frac{\partial u}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial u}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial u}{\partial z}\right)+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial x}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial x}\right)\quad\dots \qquad (2)$$

$$\rho\left(\frac{\partial vu}{\partial x}+\frac{\partial v^2}{\partial y}+\frac{\partial vw}{\partial z}\right)=-\frac{\partial p}{\partial y}+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial v}{\partial x}\right)+\frac{\partial}{\partial y}\left(2\mu_{eff}\frac{\partial v}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial v}{\partial z}\right)+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial y}\right)+\frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial y}\right)\quad\dots \qquad (3)$$

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$$\frac{\partial}{\partial z}\left(2\mu_{eff}\frac{\partial w}{\partial z}\right)+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial z}\right)$$
$$+\frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial z}\right) \quad (4)$$

$$\frac{\partial uT}{\partial x}+\frac{\partial vT}{\partial y}+\frac{\partial wT}{\partial z}=\frac{\partial}{\partial x}\left(\Gamma_{eff}\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(\Gamma_{eff}\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(\Gamma_{eff}\frac{\partial T}{\partial z}\right) \quad (5)$$

$$\mu_{eff}=\mu+\mu_t \quad (6)$$

$$\Gamma_{eff}=\Gamma+\Gamma_t$$

$$Nu=\frac{h.Dh}{K} \quad (7)$$

$$\Delta Tm=\frac{\Delta T_1-\Delta T_2}{ln\frac{\Delta T_1}{\Delta T_2}} \quad (8)$$

$$\bar{h}=\frac{\dot{m}\ Cp\ (To-Ti)}{As\ \Delta Tm} \quad (9)$$

3.1 Turbulence Model

The k-Ɛ turbulence model is used to treat the turbulence:

$$\rho\left(\frac{\partial}{\partial x}(\boldsymbol{k}u)+\frac{\partial}{\partial y}(\boldsymbol{k}v)+\frac{\partial}{\partial z}(\boldsymbol{k}w)\right)=\frac{\partial}{\partial x}\left(\frac{\mu_t}{\sigma_k}\frac{\partial \boldsymbol{k}}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\mu_t}{\sigma_k}\frac{\partial \boldsymbol{k}}{\partial y}\right)+\frac{\partial}{\partial z}\left(\frac{\mu_t}{\sigma_k}\frac{\partial \boldsymbol{k}}{\partial z}\right)+G-\rho\varepsilon \quad (10)$$

$$\rho\left(\frac{\partial}{\partial x}(\varepsilon u)+\frac{\partial}{\partial y}(\varepsilon v)+\frac{\partial}{\partial z}(\varepsilon w)\right)=\frac{\partial}{\partial x}\left(\frac{\mu_t}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\mu_t}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial y}\right)+\frac{\partial}{\partial z}\left(\frac{\mu_t}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial z}\right)+C_{1\varepsilon}\rho\frac{\varepsilon}{k}G-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \quad (11)$$

Where G is referred to the generation term and is given

$$G=\mu_t\left[2\left(\frac{\partial u}{\partial x}\right)^2+2\left(\frac{\partial v}{\partial y}\right)^2+2\left(\frac{\partial w}{\partial z}\right)^2+\left(\frac{\partial v}{\partial y}\frac{\partial u}{\partial x}\right)^2+\left(\frac{\partial v}{\partial z}\frac{\partial w}{\partial x}\right)^2+\left(\frac{\partial v}{\partial z}\frac{\partial w}{\partial y}\right)^2\right] \quad (12)$$

Also,

$$\boldsymbol{k}=\frac{1}{2}\left(\overline{u'^2}+\overline{v'^2}+\overline{w'^2}\right) \quad (13)$$

$$\varepsilon=\overline{e'_{iJ}.e'_{iJ}} \quad (14)$$

$$\mu_t=\rho c_\mu\frac{\boldsymbol{k}^2}{\varepsilon} \quad (15)$$

The values of constants in the mentioned turbulence model are:

Table-1. Empirical constants in the (*k*-*ε*) model.

C_μ	$C_{1\varepsilon}$	$C_{2\varepsilon}$	σ_k	σ_ε
0.09	1.44	1.92	1.00	1.30

3.2 Boundary Conditions

At inlet section:
the velocity is uniform ranging from 2.64m/s to 12.36m/s
A constant temperature (T = 25 ˚C) is imposed.
At outlet:
Zero relative pressure at outlet is considered.
At wall:
there is no slip (u=v=w=0) with constant temperature equal to 120 ˚C.
$k=0$ and $\epsilon=0$

4. NUMERICAL ANALYSIS

The above governing equations of the present model are solved numerically. The CFD mercantile Fluent 17.1 code is used to resolve the three-dimensional Navier-Stokes equations through means of the formula of limited volume. From CFD modeling the values of air flow characteristics can be gotten at a large number of points in a twisted tube. These points are in general linked together in the form of numerical web. 1 $*10^{-6}$ is used as the value for convergence standard for momentum and energy equations. To can the goodness of sophisticated CFD model with high precise solution the grid independent test was carry.

To value the quality of advanced CFD type with tall exact arrangement the lattice autonomous test was conducted. The arrangement is network free, in the event that the work is refined (i.e. the cells are made littler in estimate subsequently bigger in number) and the comes about of arrangement were unaltered.

Table-2: shows that the s0lution is grid independent and there is no or minimum effect on Nusselt number for all selected meshes, therefore and for more accuracy the mesh 4 is used for all calculations.

Table-2. The mesh independence test.

Meshes	No. of element	Nusselt number
Mesh 1	685495	63.98
Mesh 2	695895	64.58
Mesh 3	703525	65.09
Mesh 4	704364	65.1

5. RESULTS AND DISCUSSIONS

The obtained Numerical results are demonstrated the Nusselt number, friction factor variation and overall performance for different values of twisted ratios and Reynolds.

To check the validity of the studied numerical model, a comparison was made with the experimental model displayed in [17]. Figure-2 appears the comparison between the simulation results of present model with those

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of experimental results of [17]. From this figure it can be seen that, that the deviation between the results is about (9%). This deviation is due to the difference in the value of twisted ratio.

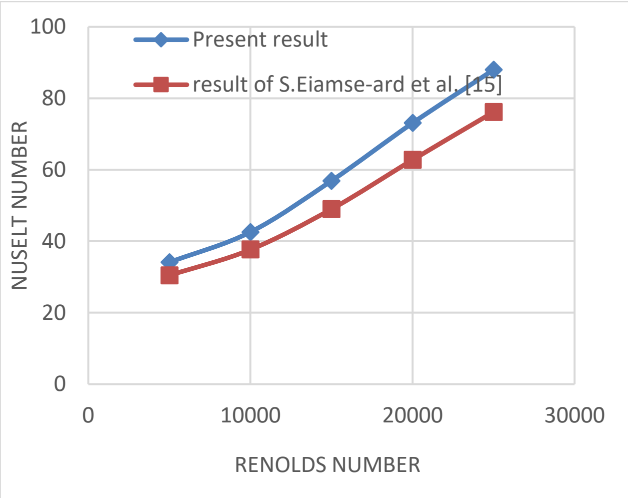

Figure-2. The comparison between the present model and published results of S. Eiamse-ard *et al.* [17] for Tr=5.

Figure-3 illustrates the variation of Nusselt number with Reynold number for the different values of twist ratio. It can be seen that, the Nusselt number is increased with decreasing twist ratio for specified Reynolds number because of the decreasing of the twist ratio increases the mixing in the flow so the amount of heat transfer increases and therefore the Nusselt number increases, where the Nusselt number is increased with increasing Reynolds number for all examined twisted ratios due to increase the flow rate. Also, this figure shows that the twist ratio Tr=5 has the maximum Nusselt Number.

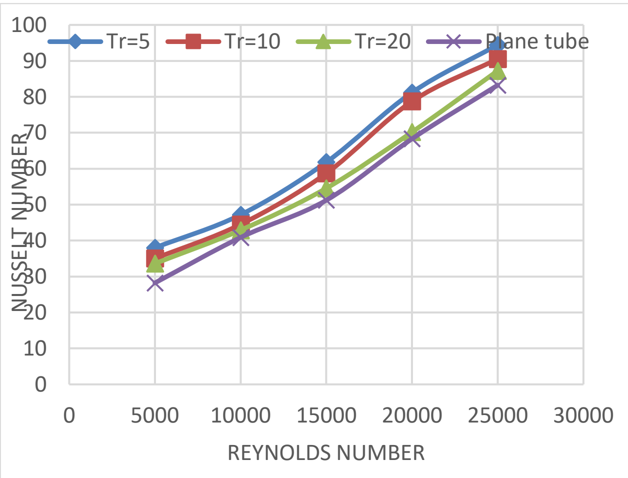

Figure-3. Effect of twisted ratio on average Nusselt number for multiple values of Reynolds.

The trend of friction factor versus Reynolds number for different values of twisted ratio is displayed in Figure-4. The friction number is decreased with increasing Reynolds number and the maximum value of friction number achieved at Tr = 5 because of the number of twist for this ratio is larger than the other ratio in the same length. For this reason the stream against more obstacles therefore the friction increasing.

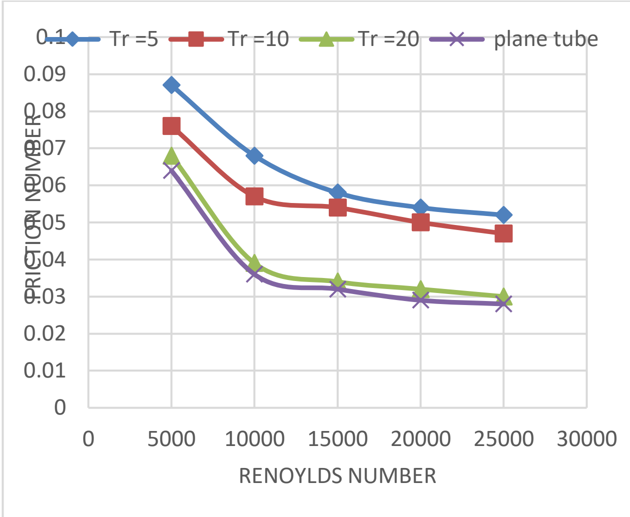

Figure-4. Effect of twist ratio on average friction for different values of Reynolds number.

Figure-5 depicts the overall performance versus Reynolds number. It is shown that the performance index decreases with increasing of Reynolds number. The higher performance is higher at Re =5000 for the studied twist ratio values. The twist ratio Tr =5 shows the larger performance values due to increase the turbulence a combined with the increase in heat transfer.

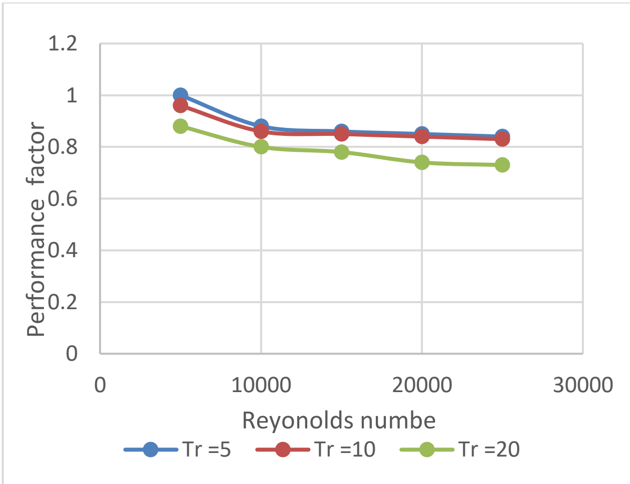

Figure-5. Effect of twist ratio on average overall performance for different values of Reynolds number.

The temperature distribution contours along the twisted tube at the stations a = 0.25, b = 0.5, c = 0.75 and d = 1 for Tr = 20,10 and 5 and Reynold number of 25000 is shown in Figure-6. It can be seen that the fluidflow temperature enhanced due to increase the heat transfer

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rate. For Tr=5,The increase is more as compared with other twist ratio.

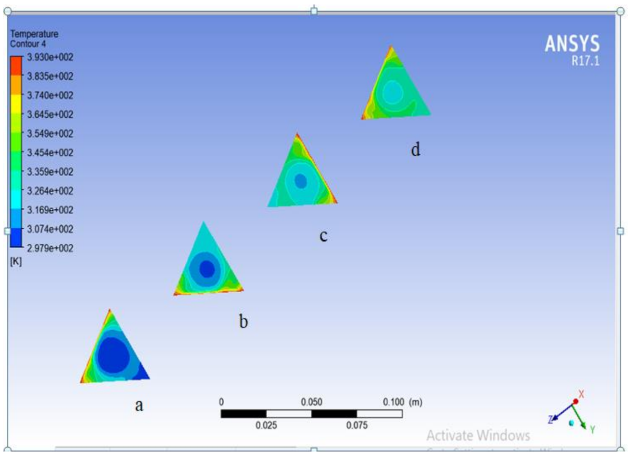
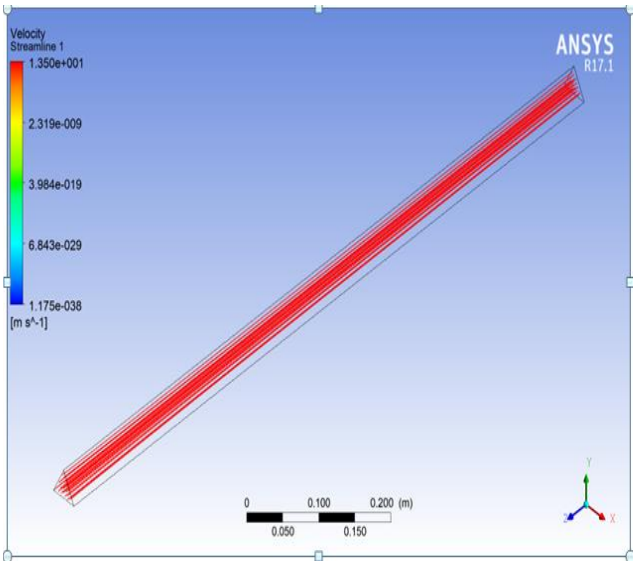
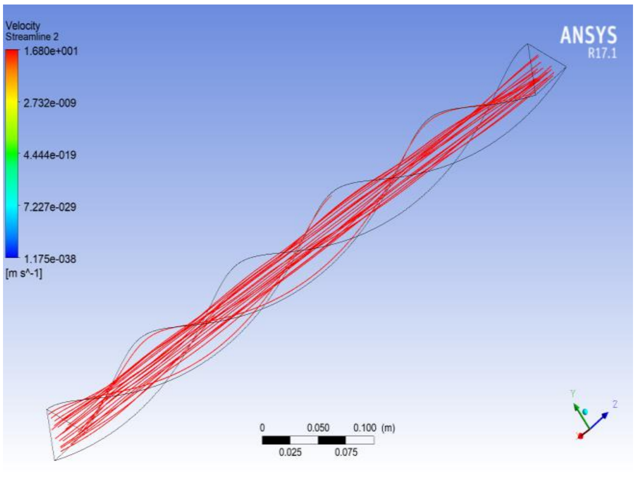


Figure-6. Temperature contours at Tr=5 for different stream twist statios (a=0.25, b =0.5, c= 0.75 and d =1).

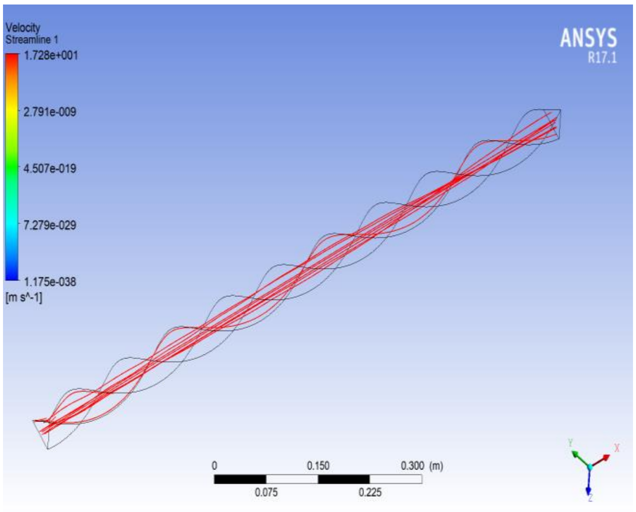
Figure-7 shows the stream line for the smooth triangle tube and twisted tube for Tr =20, 10 and 5. This Figure show that for the smooth tube the stream lines take the axial motion as shown in figure (a), while for the twisted tube the flow take the form of swirling motion and this form of motion increased with increasing twisted ratio as shown in b, c and d. The stream lines are seen to be closer each other at Tr =5 and this show a strong swirling as compared with the other twist ratios.



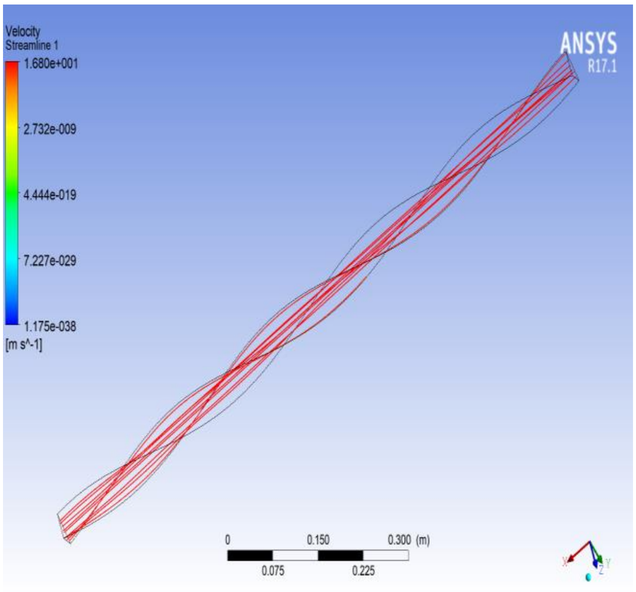
a- plane tube plain tubea plain tubea plain tubea plain tube



b-Tr=20



c. Tr =5



d. Tr =10

Figure-7. Streamlines of turbulent field at Tr =20, 10, 5 and Re =25000.

6. CONCLUSIONS

The following concluding remarks are obtained:

a) There is a capability using of twisted triangle Tube heat exchanger typedue to its high thermal performance.

b) The twisted ratio (Tr =5) showed the best heat transfer with high friction losses.

c) The overall performance index increases as the twisting ratio decreases.

d) The heat transfer enhancement for the considered twisted tube is larger by 150% for Tr=5.

NOMENCLATURE

Symbol	Description	Unit
Cp	specific heat	J\kg .K
D_h	Hydraulic diameter	m
h	heat transfer coefficient	W/m^2. K
Tr (y/d)	twisted ratio	
f	friction factor	
$\bar{h}$	Average heat transfer coefficient	w/m^2.k
$\dot{m}$	mass flow rate	kg/s
Nu	Nusselt number	
n	constants for fitting correlations	
P	Pressure	N/m^2
ΔP	Pressure drop	N/m^2
Pr	Prandtl number	
Q	heat transfer rate	W
Re	Reynolds number	
T	temperature	K
Nu	Mean Nusselt number	-

Greek Symbols

μ	dynamic viscosity	m^2/s
μ_t	Dynamic turbulent viscosity	m^2/s
ρ	density	kg/m^3
Ɛ	Turbulent energy dissipation rate	m^2/s^2

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