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A Review on Tube and Shell Heat Exchanger with Elliptical Twisted Geometry

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***Abstract:** In various engineering applications such as heat exchanger, air conditioning, chemical reactors and refrigeration systems, heat transfer enhancement techniques are popularly used. This paper reviews a multi-objective optimization of a shell and tube type heat exchanger with elliptical twisted tube geometry, which has been compared with cylindrical tube geometry. A series of experimental runs will have been constructed with respect to constant mass flow rate 0.25kg/s of Cold water and varying mass flow rate of Hot water from 0.05, 0.15, 0.25, 0.35, 0.45kg/s to study the effect of geometrical and flow parameters on heat transfer using RNG k-ε turbulence model. All properties such as pressure drop, Surface heat transfer coefficient, Reynolds number will be computed in further studies.*

***Keyword:** Twisted tube, Elliptical tube, CFD, Heat Exchanger, Twisted elliptical pipe, Comparative study, RNG, k-ε model*

I. INTRODUCTION

Global warming and pollutant emission caused by the consumption of oil, gas and coal. More efficient engines with less waste heat are being developed by engineers. As a general device, the heat exchanger plays an important role in the heat recovery process as in air conditioning, refrigeration systems, and thermal power plants. System efficiency as well as its cost are influenced by the performance of the heat exchanger. Therefore, it is important to enhance the heat transfer performance to achieve sustainable energy development.

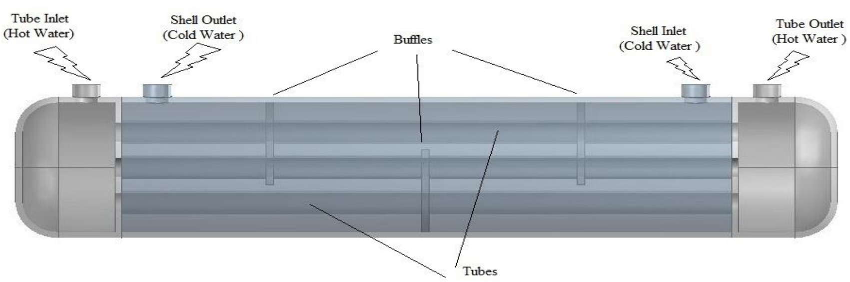

Fig.1 Tube and Shell type Heat Exchanger (Counter Flow)

Heat transfer methods can be classified into three broad categories: active, passive and compound methods. The active method involves some external power input for the enhancement of heat transfer. On the contrary, the passive method can enhance the heat transfer by using modified surfaces or geometries such as rough surfaces, extended surfaces, tube inserts, etc. As the name implies, the compound method combines the passive method with the active method.

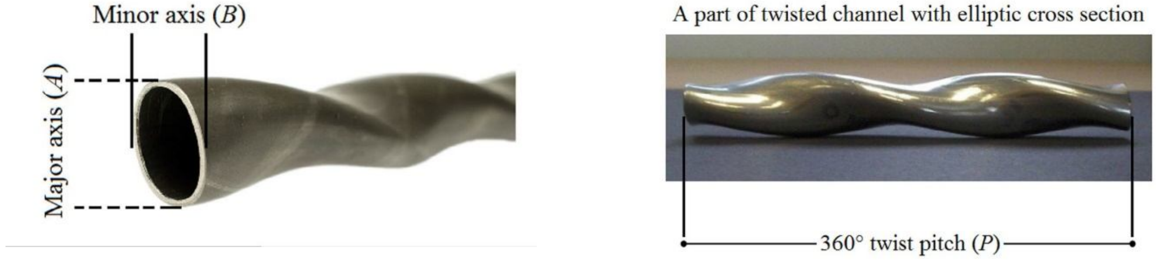

Fig.2 Cross-section of Elliptical tube- (a) Major and Minor Axis, (b) Twist Pitch (36.66mm)

Heat transfer enhancement inside ducts is very important for improving the thermal performance of heat exchange devices. It is

International Journal for Research in Applied Science & Engineering Technology (IJRASET)

usually obtained by creating some rotational motions in the flowing fluid. These swirl flows are usually initiated by the fluid turbulators. Twisting non- circular ducts, i.e. twisted channels, is the other technique to produce the swirl flows. The twisted channels are usually made into the elliptical cross section with a superimposed twist, as depicted in Fig. 1. Usually, the twisted channels can increase the heat transfer coefficient of the tube side and decrease the pressure drop of the shell side. Consider- able experimental and numerical reports about the shell side of the twisted channel heat exchangers can be found in literature. However, in terms of the tube side, studies are very scarce.

II. LITERATURE REVIEW

A. Cheng et al. [1] employed the low Reynolds k-e model to investigate heat transfer and flow characteristics of water flow inside twisted oval tube for Reynolds number in the range of 50–2000. Three dimensional numerical study is conducted to study the effects of the geometric parameters on the performance of twisted oval tube for a uniform wall temperature case. The flattening of 1.2, 1.4, 1.63, 1.8 and 2.0, and the twisted pitch ratio of 0.17, 0.25, 0.33 and 0.5. Local distributions of Nusselt number and friction factor are presented by him. He applied the filed synergy principle to reveal heat transfer enhancement mechanism. The results show that the heat transfer performance of twisted oval tube has been enhanced while having an increasing of pressure drop. One of the key findings of his study is that laminar to turbulent flow transition point was identified and located at Re = 500. The fluid is in laminar states with Reynolds number range of 50–250, while the fluid is in turbulent flow when the Reynolds comes to 500–2000. He also found that the twisted oval tube performs well compared with the smooth tube due to the effect of secondary flow.

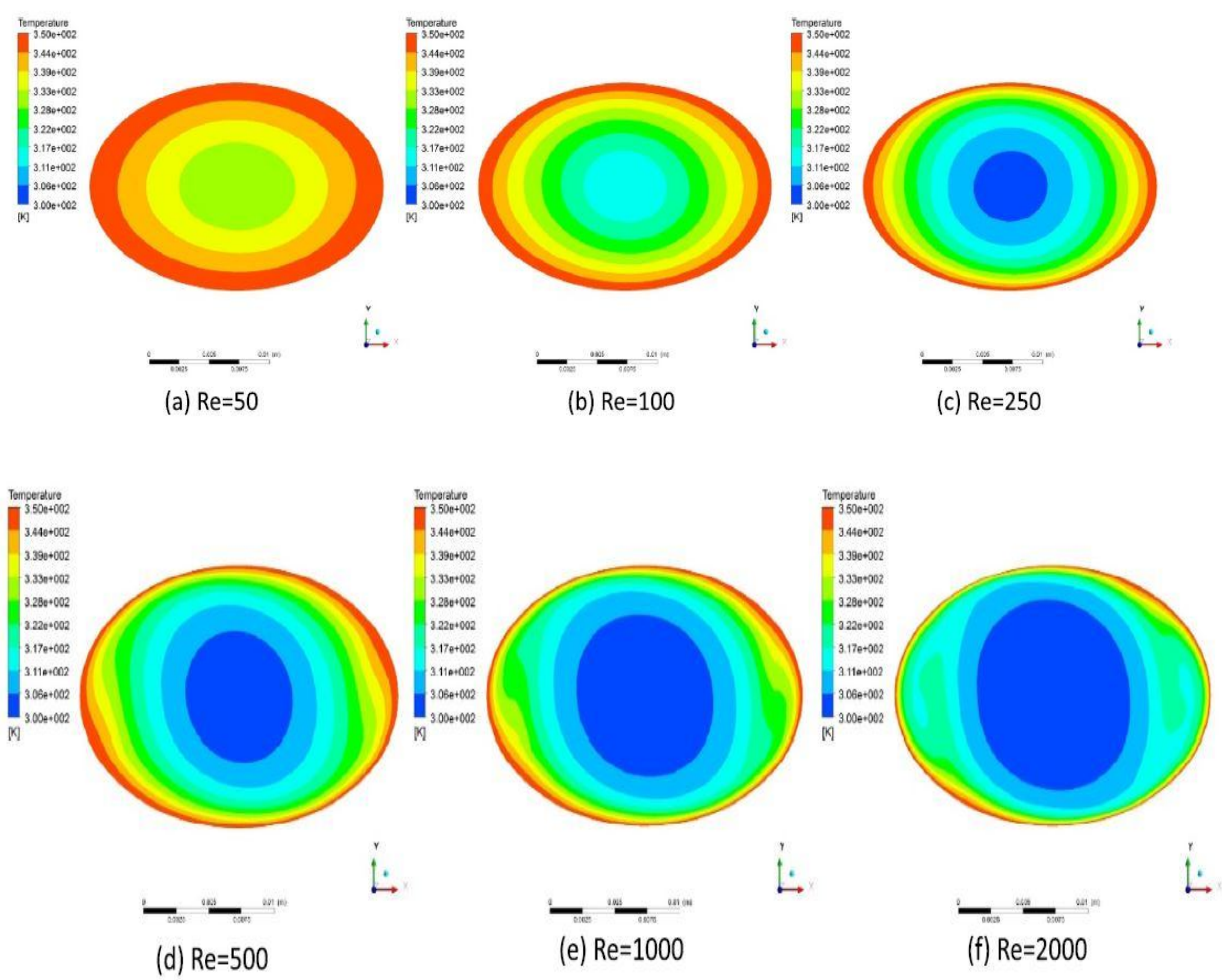

Fig.3 Local temperature distributions on the middle cross-section of Case 3 with different Reynolds numbers at z = 0.3 m.

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B. Li Zhang et al. [2] conducted an experimental research to investigate the heat transfer characteristics of steam condensation on horizontal twisted elliptical tubes (TETs) with different structural parameters. Experiments were carried out at the steam saturation temperature of around 100.5°C with the wall subcooling from around 2°C to 14°C. Experimental results indicate that the condensation heat transfer coefficients for all the tubes reduce with the increase of wall subcooling, while the enhancement factor of each TET is almost constant. Not all the tested TETs have better condensation heat transfer performance than the smooth circular tube, the average enhancement factors provided by the five TETs range from 0.87 to 1.34. Condensation heat transfer coefficients increase with the rise of the tube ellipticities.

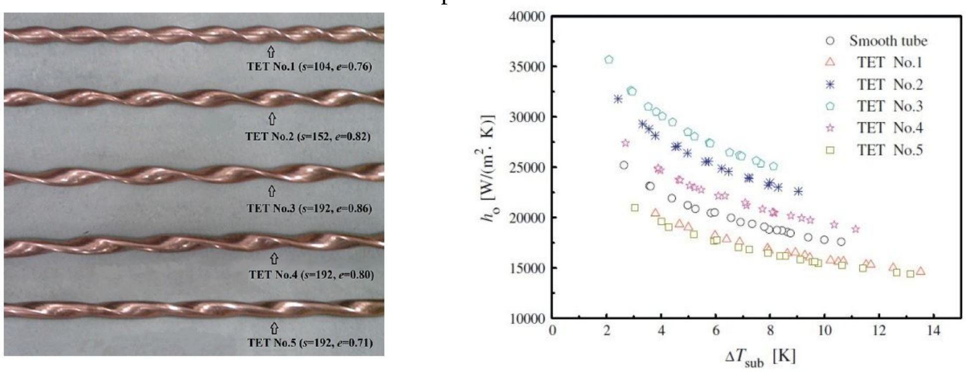

Fig.4(a) Photograph of tested TETs, (b) Condensation heat transfer coefficients of steam on TETs and smooth circular tube as a function of the wall sub-cooling.

C. R. Deepak kumar et al. [3] conducted a three-dimensional numerical study on finned-tube heat exchangers with multiple-rows of tubes. The effect of different combinations of circular and elliptical tubes on air-side flow and heat transfer characteristics are studied with various inlet air velocities in the range, 0.5 to 2.5 m/s. The results are presented in the form of Nusselt number, friction factor and synergy angle. The results show that at a low inlet velocity, elliptical tube followed by circular tube is a better alternative for heat exchangers with circular tube alone.

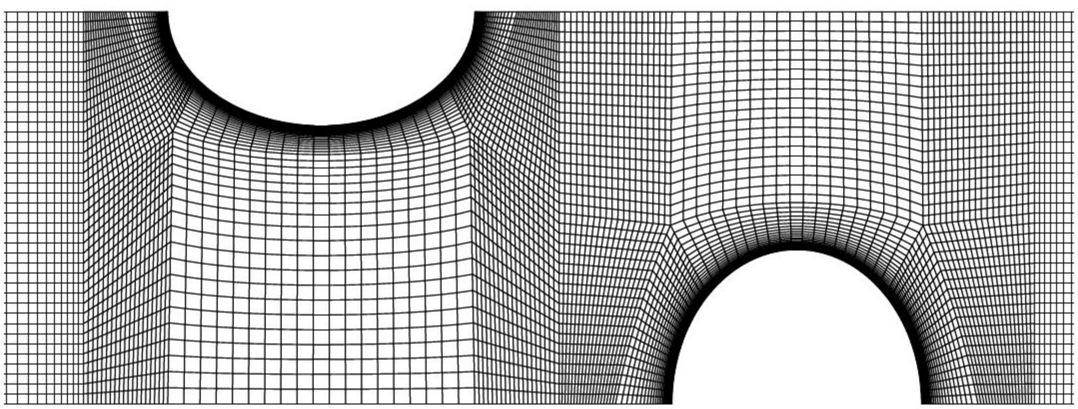

Fig.5 Schematic representation of grid

D. S. Eiamsa et al. [4] performed the effects of twisted tube combined with twisted tape on heat transfer and studied the twisted tubes fitted with tape in belly-to-neck and belly-to-belly forms and the tape in belly-to-neck form at w/D = 0.34 possesses the highest thermal performance. The numerical results of a twisted tube without tape and a circular plain tube are also given for comparison. The results are reported in terms of velocity field, temperature field, turbulent kinetic energy, local Nusselt number distribution, average Nusselt number, pressure loss and thermal performance factor. It is found that heat transfer and friction factor increase with tape width ratio. At a given tape width, the systems in bellytoneck arrangement are more efficient for heat transfer enhancement than the ones in belly-to-belly arrangement. The three-start spirally twisted tubes with twisted tapes in

International Journal for Research in Applied Science & Engineering Technology (IJRASET)

belly-to-neck arrangement at w/D = 0.1, 0.25 and 0.34 give higher Nusselt numbers than the twisted tube without tape up to 1.2%, 21% and 36%, respectively. The twisted tubes with triple-channel twisted tape in belly-to-belly arrangement provide higher Nusselt numbers than the twisted tube without tape up to 1.23%, 6.7%, 10% and 17%, respectively. The superior heat transfer of the combined devices in belly-to-neck arrangement (especially at large w/D) is attributed to the stronger interaction between the swirling flows induced by the tubes and those induced by the tapes. Moreover, the systems in belly-to-neck arrangement cause lower friction loss than the ones in belly-tobelly arrangement. Thus, the systems in belly-to-neck arrangement yield higher thermal performance factors. Among the studied cases, the twisted tube combined with triple-channel twisted tape in belly-to-neck arrangement at w/D = 0.34 possesses the maximum thermal performance of 1.32 at Reynolds number of 5000.

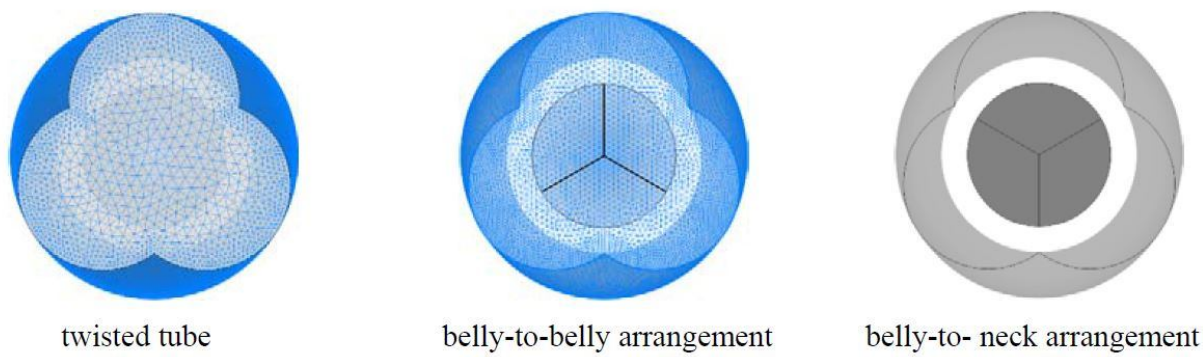

Fig.6 Configurations and grid arrangement of three-start spirally twisted tube combined with triple-channel twisted tape.

E. Anucha Saylroy et al. [5] did a numerical study of enhancing heat transfer and thermal performance of in a circular tube inserted by square cut twisted-tapes in comparison with that inserted by classical twisted tapes. The study was carried out for a turbulent periodic flow in a three-dimensional constant heat flux-wall tube. The influences of square cut twisted-tape geometries including perforated width to tape width ratios (WR = w/W = 0.5, 0.6, 0.7, 0.8 and 0.9) and perforated length to tape width ratios (LR = L/W= 0.7, 0.8 and 0.9) were determined. The main findings are that the heat transfer and pressure loss increase with decreasing perforated width to tape width ratio (WR) and perforated length to tape width ratio (LR) while thermal enhancement factor (TEF) increases as WR increases. The highest thermal enhancement factor (TEF) of 1.37 is achieved by using square-cut twisted tapes at the largest perforated width to tape width ratio (WR = 0.9) and the smallest perforated length to tape width ratio (LR = 0.7) at Reynolds number (Re) of 7000. The highest thermal enhancement factor (TEF) is around 1.32 times over than that given by the classical twisted tape.

III.METHODOLOGY

A. Model Geometry

Using Design Modeler (Ansys 17.0), two Geometries will be made. A Tube and Shell type Heat Exchanger with the Tube Geometry of Circular and Elliptical Twisted type. Total length of the tube supposed to be used in this geometry was 220mm. 3 Buffles used with the pitch of 54.875mm. Total number of 5 tubes will be used in this analysis.

RNG k–ε turbulence model will be adopted in the current study. The renormalization group (RNG) k–ε model of Yakhot and Orszag is adopted in the simulation because the model provides improved predictions of near-wall flows. The RNG k– ε model was derived by a statistical technique called renormalization method, which is widely used in industrial flow and heat transfer because of its economy and accuracy. The governing equations for continuity, momentum, energy, k and ε in the computational domain can be expressed as follows:

Continuity equation: $\frac{\partial \rho}{\partial x} + div(\rho u) = 0$

Momentum equations:

X-Momentum equation: $\frac{\partial(\rho u)}{\partial x} + div(\rho uu) = -\frac{\partial p}{\partial x} + div(\mu grad u) + S_{Mx}$

Y-Momentum equation: $\frac{\partial(\rho v)}{\partial y} + div(\rho v u) = -\frac{\partial p}{\partial y} + div(\mu grad v) + S_{My}$

Z-Momentum equation: $\frac{\partial(\rho w)}{\partial z} + div(\rho w u) = -\frac{\partial p}{\partial z} + div(\mu grad w) + S_{Mz}$

Energy equation: $\frac{\partial(\rho i)}{\partial t} + div(\rho i u) = -p div u + div(k grad T) + \Phi + S_i$

Turbulent kinetic energy: $\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho k u_j)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + \eta_t \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \rho\varepsilon$

Turbulent dissipation energy: $\frac{\partial(\rho\varepsilon)}{\partial t} + \frac{\partial(\rho\varepsilon u_j)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_1^* \frac{\varepsilon}{k}\eta_t \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - C_2\rho\frac{\varepsilon^2}{k}$

Where

$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}$, $C_1^* = C_1 - \frac{\eta(1-\eta/\eta_0)}{1+\beta\eta^3}$, $\eta = \left(2E_{ij} * E_{ij}\right)^{\frac{1}{2}}\frac{k}{\varepsilon}$, $E_{ij} = \frac{1}{2}\left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right]$

$C_\mu = 0.085$, $C_1 = 1.42$, $C_2 = 1.68$, $\beta = 0.012$, $\eta_0 = 4.38$

B. Domains definitions, meshes and boundary conditions

As mentioned before, two STHXs with different tube types will be modelled. In the further study, four domains will be defined, two fluid domain (water in the tube and shell side) and two solid domains (tubes bundle, shell). The computational domain will be meshed with unstructured tetrahedral grid Fig.4 and 5, which are generated using the ANSYS MESHING tool. The quality of the mesh for each STHX (elements quality, skewness) will be evaluated using the built-in Mesh Metrics in ANSYS MESHING.

The momentum boundary condition of no slip and no penetration will be set for all the solid walls. The thermal boundary condition of zero heat flux will be set for the shell wall and inlet and outlet nozzle walls, while the walls of tubes, baffles, and tube bundle, which also represent the solid-fluid interfaces between the two fluid domain and the solid domain, will have the thermal boundary condition of coupling heat transfer (two interfaces with coupled wall). The inlets for the shell and tube sides will be set as boundary conditions of mass flow inlet, the outlets are set as pressure-outlet. The outlets are assumed to have a pressure of zero so the inlet pressure is equal to the pressure drop on both shell and tube sides.

The ANSYS FLUENT 17.0 will be used to calculate the fluid flow and heat transfer in the computational domains. The governing equations will be iteratively solved by the finite-volume formulation with the SIMPLE algorithm. The second-order upwind scheme will be adopted for the momentum, energy, turbulence and its dissipation rate. The pressure term is treated with the standard scheme. Default under relaxation factors of the solver are used, which are 0.3, 0.7, 0.8, and 0.8 for the pressure, momentum, turbulent kinetic energy, and turbulent energy dissipation, respectively. The convergence criterion is that the normalized residuals are less than 1e-4 for the flow equations and 1e-8 for the energy equation.

IV. CONCLUSION

A series of experimental runs will be constructed with respect to constant mass flow rate 0.25kg/s of Cold water and varying mass flow rate of Hot water from 0.05, 0.15, 0.25, 0.35, 0.45kg/s to study the effect of geometrical and flow parameters on heat transfer. Following properties will be computed: -

A. Surface Heat Transfer Coefficient
B. Comparative Temperature difference
C. Comparative Pressure difference
D. Reynolds number
E. Nusselt number

V. ACKNOWLEDGMENT

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Volume 5 Issue VI, June 2017
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computed.

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