

Experimental Study of Heat Transfer Enhancement in Tube in Tube Heat Exchanger Using Corrugated tube and Twisted Tape

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Abstract - The research work summarized in this presents an experimental investigation of various aspects of Heat transfer, pressure drop and effectiveness in a double pipe heat exchanger made of corrugated inner tubes. The inner tube was corrugated by means of a special technique. New various arrangements are made having corrugation dia. 7.5mm and 5mm and longitudinal pitch 20mm and 30mm of concave corrugated tube were investigated, the twisted tape with longitudinal Pitch 50mm is introduced in inner tube to analyze enhancement in heat transfer. Counter flow heat exchanger setup is manufactured. Heat transfer coefficient was determined using experimental calculations and are compared with simulation using Ansys Fluent 18.1. Hot water (inner tube) and cold water (outer tube) inlet temperatures were maintained at around 600 C and 280 C respectively. Experiments were performed over the Reynolds number range of 300-1500, based on the hydraulic diameter of the annular space between the two tubes. Hot water Reynolds number was kept constant at around 700. Results indicated that the inner tube corrugations and arrangement type of corrugated tubes have significant effect on thermal and frictional characteristics. Maximum effectiveness was obtained for heat exchanger made of concave corrugated outer tube and convex corrugated inner tube with dia. 7.5 mm and 20 mm longitudinal pitch. Introduction of Twisted tape results in the rise in pressure drop with very less temperature gradient. Flow is visualized using Ansys Fluent 18.1.

Keywords: Tube in Tube Heat Exchanger, Corrugated tube, Nusselt Number, Eulers Number, Convection

1. INTRODUCTION

Heat exchangers are used in various industrial or domestic applications such as in thermal power plants, in transport, heating as well as in air cooling systems, electronic equipment and space applications. In all these applications, improvements in the efficiency of heat exchangers can lead to substantial cost, space and materials savings. Therefore, considerable research work has been done in the past to seek effective ways to increase the efficiency of heat exchangers. The referred investigations include the selection of working fluids with high thermal conductivity, selection of their flow arrangement and high effective heat transfer surfaces made from high-conductivity materials. For single-phase and two-phase heat transfer, effective heat transfer enhancement techniques have been observed. However, in the present work only the single-phase forced convection enhancement techniques have been considered. Over long

range of technical papers and reports have been published in various bibliographic reports, reviews, monographs and edited texts with yearly growth tendencies. The heat transfer enhancement methods reported are systemized in many forms but primarily they may be grouped as passive and active enhancement methods. Over the years corrugated tubes have been used as elements for effective heat transfer in various applications where space and weight are important constraints such as in the power plant industry for cooling of gas turbine blades, in the electronics industry for cooling of electronic components and recently also in the hot water boilers of central heating systems. Thus the thermal and fluid dynamic characteristics of flow through the corrugated tube with twisted tape have been the interest of numerous investigators.

2. Literature Survey

Wei Wang et al [1] (2017) studied the turbulent heat transfer flow in novel outward helically corrugated tubes. Outward helically corrugated tube to obtain the equilibrium performance for heat transfer, pressure drop, and energy benefit. The Reynolds number (Re) has the most effect on Nusselt number (Nu), and its sensitivity coefficient is five times higher than that for corrugation pitch-to-diameter ratio (pl/D) and corrugation height-to-diameter ratio (Hl/D). The sensitivity analysis showed that Re had the most effect on Nu, and its sensitivity coefficient was five times higher than that of pl/D and Hl/D. The sensitivity degrees of Re and Hl/D are similar to PEC, the PEC presented upgrade firstly then descending latter tendency with the variation of pl/D and reached the peak value at pl/D=1.5. If the heat transfer performance is dominant, higher Re and Hl/D with a lower pl/D are better for heat exchangers designed. An equilibrium interval was obtained on the Pareto front at which the optimization values of Nu, fRe, and PEC correspond to 87.55–108.81, 1595.88–2502.69, and 1.08–1.04, respectively. The Reynolds number (Re) has the most effect on Nusselt number (Nu).

Hamed Sadighi Dizaji et al [2] (2016) Investigated distinct arrangements of concave and convex type of corrugated tubes Exergy loss due to simultaneous employing of corrugated tubes as the inner and outer tube (shell) of a shell and tube heat exchanger are also investigated. Corrugations causes increment of both exergy loss and NTU. If both tube and shell are corrugated, the exergy loss and NTU increase about 17% - 81% and 34% - 60% respectively. Maximum exergy loss was observed for heat exchanger made of convex

corrugated tube and concave corrugated shell. It is observed that the corrugations increase both NTU and dimensionless exergy loss in all arrangements. The general reasons of NTU and exergy loss enhancement due to corrugations can be explained by two mechanisms as follows. Corrugations create secondary flows through the heat exchanger. These secondary flows decrease the thermal boundary layer formed on the outer (inner) surface area of the inner tube. Corrugated tubes have more surface area in comparison with the smooth tubes in the same length. To this reason, the heat exchanger made of corrugated tubes have further amount of the NTU compared to the heat exchanger made of smooth tubes.

Gómez M.A. et al [3] (2018) Investigated The Boussinesq approximation to solve the fluid movements that occur by natural convection, a fitting method is used to calculate the heat transfer of the corrugated surface in a smooth coil. The model is applied to a tank with 150 L of water, which is heated by a heat exchanger in the form of an outer jacket. The heat accumulated in the tank is exchanged with the drinking water by means of a corrugated coil located inside the tank. The model is tested by simulating several steady-state continuous consumption experiments and by conducting a transient test of heating and subsequent cooling. For the comparison of results, the data used include the flow rates through the water jacket and the inner coil, the inlet and outlet temperatures of these flows and various temperatures recorded by thermocouples located inside the tank at different heights. Comparison of Experimentation and Numerical investigation shows 33.33% good Agreement. The corrugation provided to the tube increases heat transfer rate about 10%. However Because of this corrugation pressure drop occurs to be in the range of corrugated regime.

Hamed Sadighi Dizaji et al [4] (2015) have experimentally investigated pressure drop, heat transfer and effectiveness of double pipe heat exchanger made of corrugated outer and inner tubes. Heat transfer coefficient was determined using Wilson plots. Hot water (inner tube) and cold water (outer tube) inlet temperatures were maintained at around 40°C and 80°C respectively. Experimental results indicated that the outer tube corrugations and arrangement type of corrugated tubes have significant effect on thermal and frictional characteristics. The effectiveness increases about 5-19% while if in addition to the inner tube, outer tube is corrugated too, the effectiveness increases about 28-55%. Maximum effectiveness was obtained for heat exchanger made of concave corrugated outer tube and convex corrugated inner tube. Arrangement type of corrugated tubes (concave and convex) had significant effect on thermal and frictional characteristics.

J.I. Córcoles-Tendero et al [5] (2018) have experimentally analyzed the effect of spirally corrugation in a simple tube on the heat transferred and friction factor using numerical simulations. The behaviour of both smooth and spirally corrugated tubes considering turbulent flow at four Reynolds numbers (15-40×10³) and two Prandtl numbers (2.9 and

4.3). For the smooth and corrugated tubes, stainless steel tubes with an inner diameter of 18 mm, a length of 6m and a wall thickness of 1mm were used. The corrugated tube has a corrugation depth of 0.43mm and a helical pitch of 15.86 mm. The results for the corrugated tube indicate that the numerical model predicts an average Nusselt number within a maximum relative error of 17% compared with the experimental data, and the differences in the Fanning factor are lower than 9%. For the smooth and corrugated tubes, stainless steel tubes with an inner diameter of 18mm and a wall thickness of 1mm were used. The highest Fanning friction values were obtained for the corrugated tube, with differences observed between the corrugated and smooth tubes. These differences were approximately 86% (Reynolds number 15×10³) and 100% (Reynolds number 40 ×10³) higher for the corrugated tube than for the smooth one.

Fabio Bozzoli et al [6] (2016) investigated the effect of wall corrugation on local convective heat transfer in coiled tubes. They both were characterized by eight coils with a helix diameter and a pitch of 310 mm and 200 mm, respectively, thus yielding a coiled pipes length L of approximately 10 m. The corrugated-wall tube considered in the present investigation is included in the general category usually known as spirally enhanced tubes. The heat flux provided to the fluid was selected to make the buoyancy forces negligible in comparison to inertial ones for the investigated fluid velocity ranges. More Pressure drop occurs at corrugated surface compare to smooth pipe but heat transfer rate increases as surface area increases in case of corrugations.

Huai-Zhi Han et al [7] (2015) studied a fully developing three-dimensional heat transfer and flow model, a multi-objective optimization aims to fulfill the geometric design for double-tube heat exchangers with inner corrugated tube with RSM. Dimensionless corrugation pitch (p/D), dimensionless corrugation height (H/D), dimensionless corrugation radius (r/D) and Reynolds number (Re) are considered as four design parameters. Considering the process parameters, the characteristic numbers involving heat transfer characteristic, resistance characteristic and overall heat transfer performance calculated by CFD, and are served as objective functions to the RSM (Nuc , f_c , Nuc/Nus , f_c/f_s and h). The results of optimal designs are a set of multiple optimum solutions, called 'Pareto optimal solutions'. It reveals the identical tendency of Nuc/Nus and f_c/f_s reflecting the conflict between them that means augmenting the heat transfer performance with various design parameters in the optimal situation inevitably sacrificed the increase of flow resistance. According to the Pareto optimal curves, the optimum designing parameters of double pipe heat exchanger with inner corrugated tube under the constraints of $Nuc/Nus_{1.2}$ are found to be $P/D = 0.82$, $H/D = 0.22$, $r/D = 0.23$, $Re = 26,263$, corresponding to the maximum value of $h = 1.12$.

Xuemei Su et al [8] (2015) the effects of the heat flux and mass flow flux on the circumferential averages of the HTCs at $z = 140$ mm and $z = 270$ mm. It can be seen that the

circumferential averages of the HTC's increase with the increase of the heat flux or mass flow flux. The average HTC's increase obviously with the heat flux and slightly with the mass flow flux in the corrugated tube. The variation of the average HTC's with vapor quality should be analyzed by considering the corresponding flow patterns and heat transfer mechanisms in the corrugated tube. In horizontal two-phase flows with asymmetric distributions of vapor and liquid in the tube, the variation of the local HTC's is not only influenced by the vapor quality, but also by the flow regimes and the corresponding heat transfer mechanisms.

2.1 Literature Review for Twisted Tape Heat Exchanger

P. Samruaisin et al [9] (2017) Investigated with Experiments using all twisted tapes with constant twist ratio of 2.5 under constant heat flux condition. The results show that RS-QTTs in all arrangements yield lower heat transfer rate and cause lower friction loss than full-length quadruple twisted tapes (QTTs) due to the weaker swirling intensity caused by the decaying of swirling flow. Heat transfer rates given by the RS-QTTs with s/y of 0.5, 1.0, 1.5 and 2.0 are lower than those given by the QTTs by around 6.2%, 8.13%, 10.6% and 12.8%, respectively. For RS-QTTs, heat transfer rate, friction loss and TEF increase with decreasing s/y . Over the range reported, the RS-QTTs in cross-arrangement with $s/y = 0.5$ gives heat transfer rate up to 6.6% over than that of the QTTs and the maximum TEF of 1.27. Moreover, the flow and thermal behaviors were also numerically studied for a better understanding on heat transfer mechanism.

Amar Raj Singh Suri et al [10] (2017) concluded that the determination encompassed the geometrical parameters namely, wing depth ratio (W/W_d) of 0.042–0.167, perforation width ratio (a/W) of 0.250, twist ratio (T/W) of 2.5, and number of twisted tapes (N) of 4.0. The effect of multiple square perforated twisted tape with square wing has been investigated for the range of Reynolds number (Re) varied from 5000 to 27,000. The maximum enhancement in $Nurs$ and frs is observed to be 6.96 and 8.34 times of that of the plain circular tube, respectively. Correlations of $Nurs$, frs and η_p are established in term of Re and geometrical parameters of wings twisted tape which can be used to predict the values of $Nurs$, frs and η_p with considerably good accuracy.

Kwok Yoong Lim et al [11] (2017) The performance evaluation is based on two different conditions variable and constant pumping power. An analysis on the dominance between free convection and swirl-flow-induced convection is performed through the buoyancy force and centrifugal force balance. This provides a sound explanation of the distinct heat transfer characteristics for different ranges of Reynolds number.

This comprehensive analysis of performance evaluation provides interesting insights into the role played by the twisted tape insert in the contribution of energy transport in laminar forced convection. the effectiveness ratio are similar to those of the Nusselt number ratio. the effectiveness as a

performance indicator, the thermal performance enhances with increasing twist ratio at low Reynolds number ($Re=600$) while the trend is reversed at higher Reynolds number.

3. Experimental Analysis

Considering heat transfer enhancement, apart from the utilization of various surface enhancement elements, efforts have also made to select a proper counter flow arrangement of the heat exchanger in order to obtain the maximum advantage for a given heat exchanger configuration. It may be noted that in a counter-flow arrangement of a heat exchanger, the outlets of fluid streams lie at opposite ends and this enables the outlet temperature of cold fluid to rise above the outlet temperature of the hot fluid. This is not possible in a parallel flow arrangement, and hence as far as the heat transfer rate is concerned, the counter-flow heat exchangers are superior to the parallel type [4]. Heat exchanger with counter flow arrangement characterized by a better temperature distribution compared with the parallel-flow heat exchanger. The optimal flow arrangement were employed during the experimental investigation of the corrugated tube heat exchanger described in the present work.

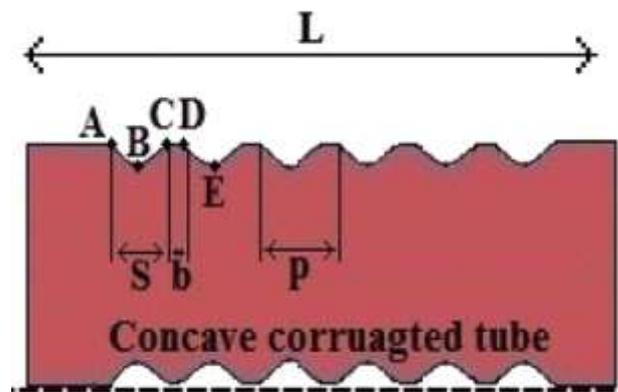


Fig.1:Schematic illustration of corrugated tube

First for some approximate theoretical considerations of the heat transfer from a Smooth tube are considered. LMTD is given

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{\ln \frac{\Delta T_2}{\Delta T_1}} \quad (1)$$

Where ΔT_2 and ΔT_1 are temperature gradient and inlet an outlet of counter flow heat exchanger.

The rate of heat transfer is given by

For Cold side

$$Q_1 = mc \times C_{pc} \times (T_2 - T_1) \quad (2)$$

For Hot Side

$$Q_2 = mh \times C_{ph} \times (T_3 - T_4) \tag{3}$$

$$Q_{avg} = (Q_1 + Q_2) / 2 \tag{4}$$

$$\text{Heat transfer Area, } A_i = \pi \times d_i \times l \tag{5}$$

$$U_i = \frac{Q}{A_i \times LMTD} \tag{6}$$

$$Re = \frac{\rho V D}{\mu} \tag{7}$$

$$Nu = 0.23 \times Re^{0.8} \times Pr^{1/3} \tag{8}$$

$$Nu = \frac{h \times D}{K_f} \tag{9}$$

$$f = \frac{\Delta P}{0.5 \left(\frac{L}{D_h} \right) \rho V^2} \tag{10}$$



Fig.3 Experimental Setup

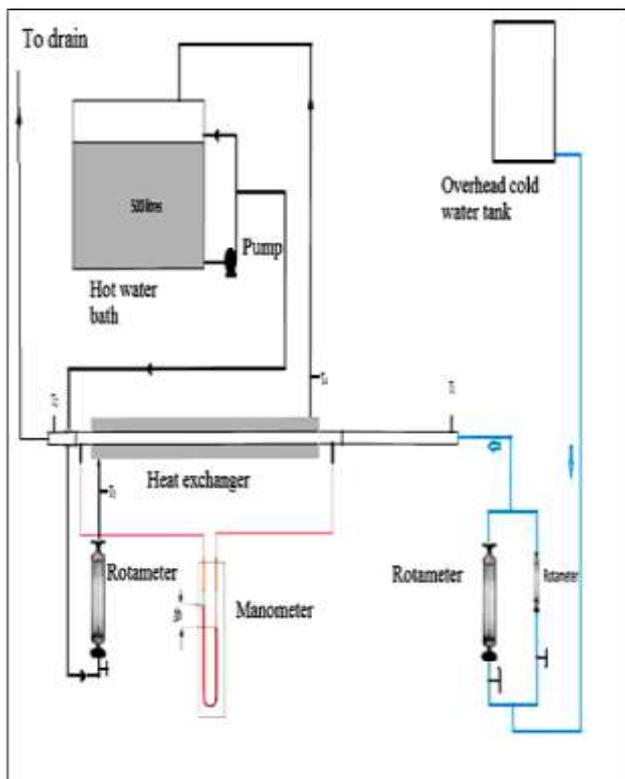


Fig.2: Schematic of the heat exchanger setup

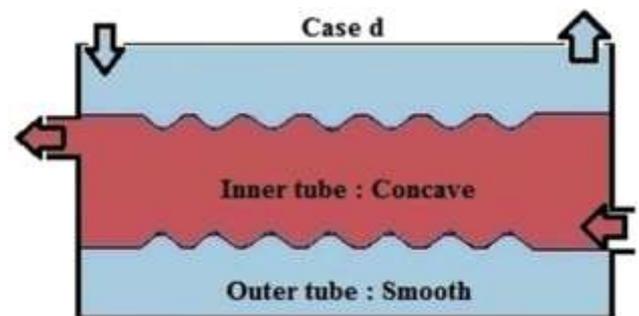


Fig.4: Basic Counter flow arrangement

Analogous to the tube banks, it can be expected that the pin arrangement will have a strong influence on the performance characteristics of flat-tube and corrugated tube heat exchanger. Keeping this in consideration, numerical computations were performed for both in-line and staggered pin arrangements, as depicted in Fig.5.

4. Experimental parameters

Experimentation is carried out on following configurations. Variable dimensions of Corrugation Diameter and Longitudinal Pitch. Following configurations are tested to understand effect of geometric parameters.

Experimentation is done by counter flow heat exchanger with Inner tube corrugated and smooth outer tube. Various configurations of inner tube corrugations are given below Table. This test initially carried out on all these 4 tubes without twisted tape and then with Twisted Tape. Twisted tape is fitted in the inner tube with the help of wings provided over tape.

Table-1 Specifications of Samples

Sr. No.	Specimen	Corrugation Dia.	L	Longitudinal Pitch
1	Case-I	5mm	340mm	20mm
2	Case-II	5mm	340mm	30mm
3	Case-III	7.5mm	340mm	20mm
4	Case-IV	7.5mm	340mm	30mm



Fig.5: Actual Corrugated Inner Tube

5. Result and Discussions

The quantitative heat transfer and pressure drop characteristics of Corrugated Tube heat exchanger, are presented in terms of significant non-dimensional numbers in Fig.8 and Fig.9.

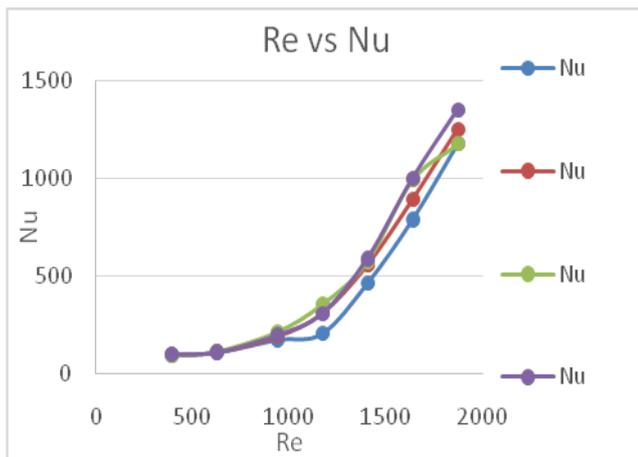


Fig.6: Nu of Corrugated tube heat exchanger.

Table-2 Nu vs. Re for different configurations of Corrugated tube

m (kg/s)	v (m/s)	Re	Nu			
			D=5mm P=20	D=5mm P=30	D=7.5mm P=20	D=7.5mm P=30
0.005	0.001224	389.5972199	89.95	95.58	89.65	100.5
0.008	0.001959	623.3555519	104.89	108.97	108.58	109.54
0.012	0.002938	935.0333278	169.99151	189.65	208.98	197.23
0.015	0.003673	1168.79166	201.58	308.95	350.58	308.52
0.018	0.004407	1402.549992	458.55	556.32	568.91	589.32
0.021	0.005142	1636.308324	784.58	890.5	988.21	998.7
0.024	0.005877	1870.066656	1172.3546	1250.97	1172.58	1350

Fig.8 clearly suggests that with an increase in corrugation diameter the Nu increases over the entire range of Re. The increase in Nu can be attributed to the increase in the wetted area, for smaller corrugation diameter. A similar behavior of Nu is observed also for variable Longitudinal pitch However, with a decrease in Re, Nu increases faster than for the case characterized with a reduction in Corrugation diameter. A contrasting behavior is observed in the context of variation of Nu with the Longitudinal Pitch. This because of the fact that Nu in the present work is calculated from a heat transfer coefficient that is evaluated with reference to the bare surface, which increases continuously with increase in Diameter.

Table-3 fvs. Re for different configurations of Corrugated tube

m (kg/s)	v (m/s)	Re	f			
			D=5mm P=20	D=5mm P=30	D=7.5mm P=20	D=7.5mm P=30
0.005	0.001224	389.5972199	0.043022553	0.042022553	0.035022553	0.025022553
0.008	0.001959	623.3555519	0.0088654	0.00798654	0.0075654	0.008654
0.012	0.002938	935.0333278	0.0219876	0.0209876	0.0219876	0.02519876
0.015	0.003673	1168.79166	0.009155466	0.009855466	0.009955466	0.009555466
0.018	0.004407	1402.549992	0.008775126	0.00798644	0.008375126	0.009377513
0.021	0.005142	1636.308324	0.0034258	0.004258	0.005258	0.006258
0.024	0.005877	1870.066656	0.003537819	0.00353782	0.002353782	0.003935378

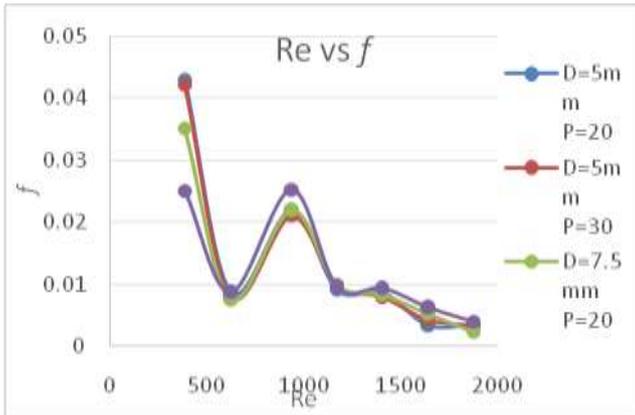


Fig.7:-f of corrugated inner tube heat exchanger

The f variations presented in Fig.7 confirm such propositions. Where Pressure drop ΔP is calculated by Manometers as shown in Fig.3 whereas mean fluid velocity of flow can be found out by flow meter recorded discharge.

However, Fig.7 show that the behavior of f in response to changes in Re is quite similar to the f behavior of the corresponding smooth tube. A completely different behavior can be observed regarding the variation of f with Re. whereas in the corresponding twisted tape arrangement f increases with a decrease in Re. The effect of the corrugation on the flow behavior can be observed when the velocity and pressure contours are represented in a region close to the contraction and expansion of the tube. At the downstream side of the corrugation, velocity is reduced, due to in this region the pressure slightly increases and prorogues an unfavorable pressure gradient that caused flow separation.

6. Validation of Result and Flow Visualization using ANSYS FLUENT 18.1

Ansys is the finite element analysis code widely used in computer aided engineering (CAE) field. ANSYS software help us to construct computer models of structure, machine, components or system, apply operating loads and other design criteria and study physical Response such as stress level temperature distribution, pressure.

Some assumptions are used such as (1) the flow through the tube is turbulent and incompressible and (2) steady state flow. In the present numerical solution, the time-independent incompressible Navier–Stokes and the turbulence (k- ω SST) equations were discretized using the finite volume technique. To evaluate the pressure field, the pressure–velocity coupling algorithm SIMPLE is selected. Also the solution convergence is met when the difference between normalized residual of the algebraic equation and the prescribed value is less than 10^{-3} for momentum, continuity and turbulence equations and 10^{-6} for energy equation.

Following are general steps involved

1. Model making using Design Modeller
2. Meshing using ICEM CFD
3. Setup of Solution
4. Result

Model Making using Design Modeller

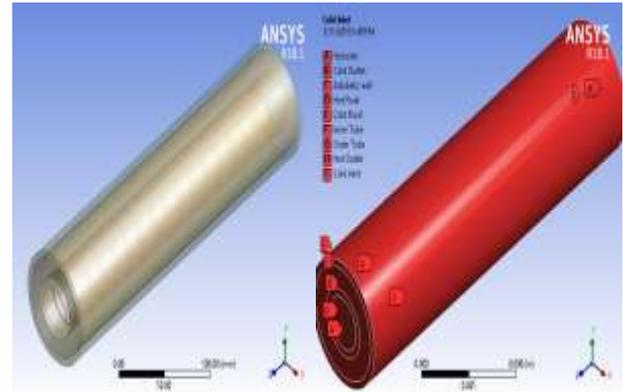


Fig.8:-Model of Tube in Tube counter flow heat exchanger

Meshing

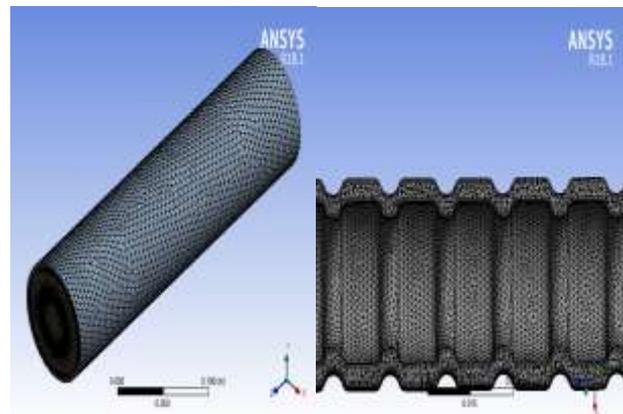


Fig.9:-Meshing Details of Heat Exchanger and Inner Tube

The Experimentation is setup in the Ansys 18.1 with variable mass flow inlet and with respect to that various Re value. The heat transfer is improved due to recirculation of Secondary flow.

6.1 FLOW PATTERN

Without Twisted Tape arrangement

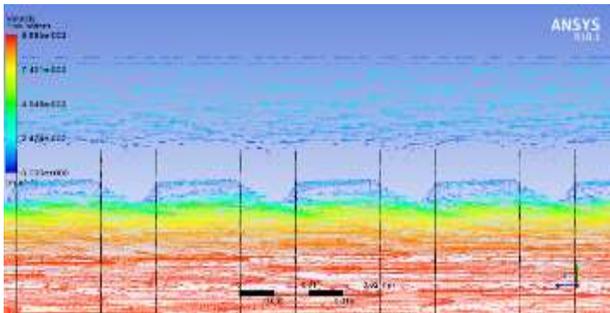


Fig.10:-Flow pattern in Inner tube

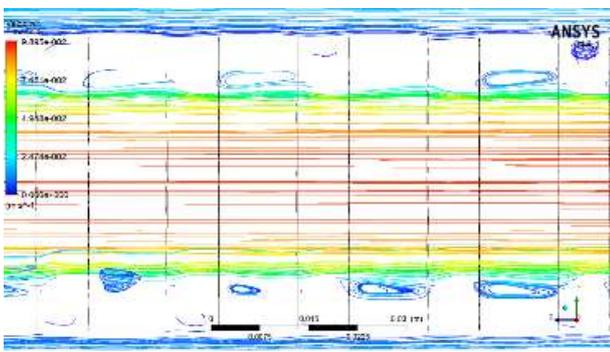


Fig.11:-Flow Streamlines in Inner tube

Fig. 10 and Fig.11 shows the flow through corrugated tube, as shown the secondary flow causes enhancement in heat transfer. Temperature gradient is shown in Fig. 12 and a significant pressure drop can be seen in Fig. 13

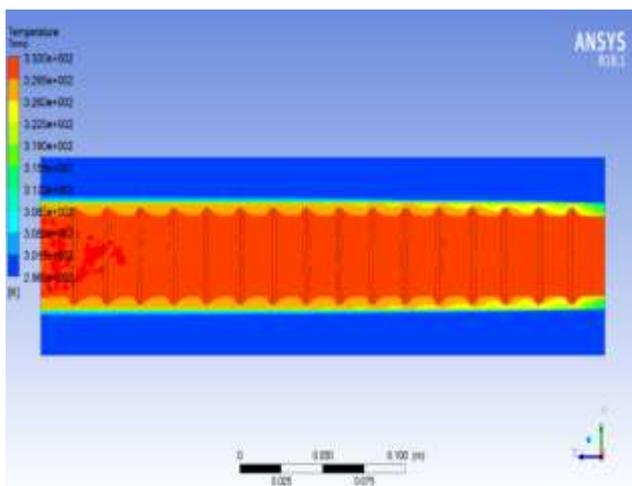


Fig.12:-Temperature contours

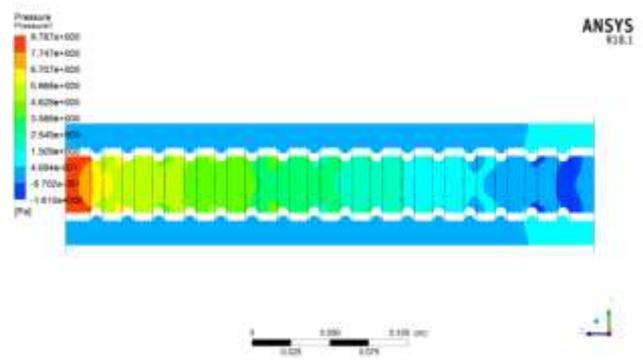


Fig.13:-Pressure contours

6.2 FLOW PATTERN With Twisted Tape

The twisted Tape arrangement heat transfer. It is clear (from Fig. 14 and Fig. 16) that using twisted tape in all twist ratios through the corrugated tube, leads to higher Nu number than empty corrugated tube. The theory behind this enhancement can be explained by respect to the fluid flow pattern through the tube. Fig. 14 shows the streamlines of corrugated tube equipped with a typical twisted tape. The swirl flow dominates the flow field due to the twisted tape and also corrugation pitch of tube.

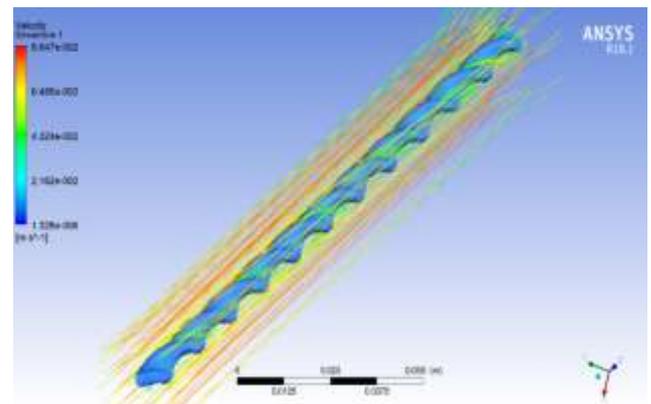


Fig.14:-Flow pattern on twisted tape

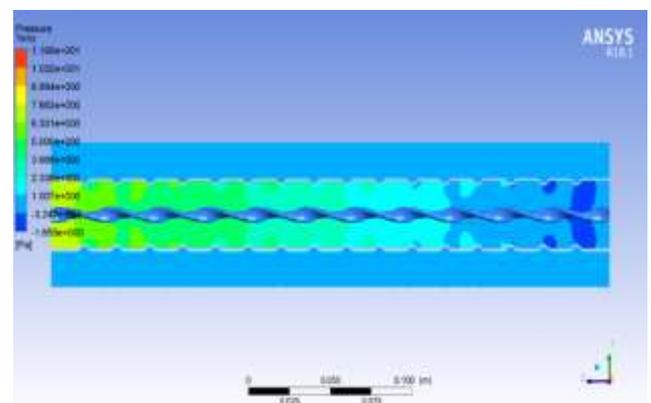


Fig.15:-Pressure drop across hot fluid domain

twisted tapes on fluid flow and heat transfer of heat exchanger tube , Case Studies in Thermal Engineering vol.no.10 (2017) pp-28-43

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