Experimental Analysis of Twisted Tube Heat Exchanger

[1]Rakesh Prajapati, [2]Saurabh Modi, [3]Ronak Soni
[1 to 3]Assistant Professor
[1]Mechanical Department,
[1]Parul Institute of Technology, Parul University, Vadodara, India

Abstract : Now days, All new device applications in oil refinement, chemical, petro-chemical, and power generation area unit accommodated through the utilization of typical shell and tube sort heat exchangers. The elemental basis transfer necessities. However, there are a unit limitations related to the technology that embrace inefficient usage of shell facet pressure drop, dead or low flow zones round the baffles wherever fouling and corrosion will occur, and flow evoked tube vibration, which may ultimately lead to breakdown. We would like to presents a recent innovation and development of a brand new technology, referred to as Twisted Tube technology, which has been able to overcome the limitations of the traditional technology. This analysis compares the development, performance, and economic science of Twisted Tube exchangers against typical styles for copper materials as well as reactive metals for this datum is shell and tube technology is a price effective, tried resolution for a good type of heat.

***IndexTerms*–Spiral Heat Exchanger, Fouling factor, Reynolds number, Nusselt number, Prandlt number, LMTD, Effectiveness, NTU**

I. Introduction

The heat exchanger is the best application of heat transfer and Heat exchangers are devices application to transfer heat between dual or more fluid streams at different temperatures. A greater number of production facilities in industries use processes in which heat is transferred between different Medias. The basic principle of heat transfer is two fluids at different temperatures are placed in contact with a conductive wall, heat transfer begin from hot fluid to the cold fluid until both fluid reach the equal temperature level. The driving force for heat transfer is the temperature difference levels between the hot & cold fluids, the greater the difference the higher the rate at which the heat will flow between hot and cold fluid. With complex processing sequences, designer must optimize the temperature levels at each stage to enhance the total rate of heat flow.A **second factor** controlling the heat transfer is the area provided of the conductive wall for heat flow. The greater area the larger the amount of heat that should flow in a given duration with given temperature difference. The designer has to be minimizing this area to provide cost effective solutions to his client with skill. The amount of area can be minimized and configured to reduce the ascendancy volume and overall cost.The **third** and most crucial factor controlling the heat transfer is the rate at which the heat flows into and out from individual of the fluids. A high resistance is heat flow in either fluid will produce a slow overall rate of transfer. The level of resistance to heat flow determine from many different factors including the thermal characteristics of the fluids but can be affected by the designer in a very positive way through the generation of turbulence within the fluids to resist the creation of a thermally resistant static "boundary layer" of fluid in contact with the heat transfer surface.The **fourth factor** also under the control of the designer is the flow of heat through the conductive barrier between the fluids. The material selection has to be compatible with the fluids of the process, it will not corrode or contaminate food product, it must have an adaptable level of mechanical strength to prevail against working temperature and pressure it must have a lower resistance to heat flow so that it should not become the overriding factor in the heat transfer process. The basic examples of heat exchangers are boiler, condensers, radiators, electronics cooling, etc.

II. Classification of Heat Exchanger

Nature of Heat Exchanger Process:

Direct contact heat exchanger: It is done by complete physical mixing of heat and mass transfer. Examples, water cooling towers and condensers in steam power plants. In a direct contact heat exchanger, the exchange of heat takes place by direct mixing of hot and cold fluids. And the transfer of heat takes place simultaneously. Example: cooling towers, jet condensers, direct contact feed heaters. The above figure shows a direct contact heat exchanger in which steam mixes with cold water, it gives its latent heat to the cold water and gets condensed. The hot water and the non condensable gas leave the container as shown in the figure.

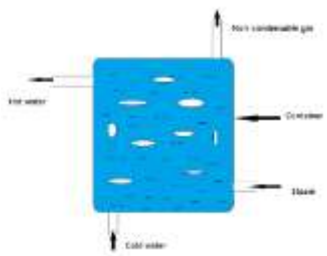

Fig.1 Direct Contact Heat Exchanger

Regenerator is a type of heat exchanger where heat from the hot fluid is stored in a thermal storage medium before it is transferred to the cold fluid. To achieve the hot fluid is brought into contact with the heat storage medium, and then the fluid is displaced with the cold fluid, which takes in the heat.

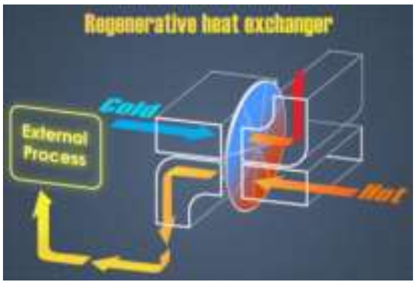

Fig.2Regenerator Heat Exchanger

Recuperator is a special purpose counter-flow energy recovery heat exchanger positioned within the supply and exhaust air streams of an air handling system, or in the exhaust gases of an industrial process, in order to recover the waste heat.

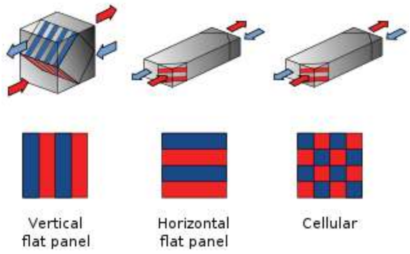

Fig.3 Recuperator Heat Exchanger

Relative direction of motion of fluids:

According to flow of fluids, the heat exchangers are classified into three categories:

Parallel flow heat exchangers:

In parallel flow heat exchangers, both the tube side fluid & shell side fluid flow in same direction. In this case, the two fluids enter the heat exchanger from the same end with a higher temperature difference.

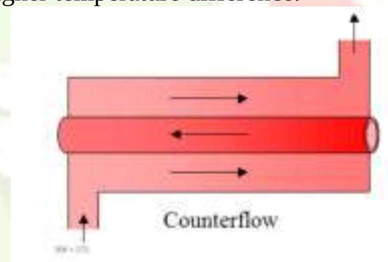

Fig.4 Parallel flow heat exchangers

Counter flow heat exchangers:

In counter flow heat exchangers, two fluids flow in opposite directions. Each of the fluids enters the heat exchanger from opposite end. Because the cold fluid exist the counter flow heat exchanger at the end where hot fluid enters the heat exchanger, the cooler fluid will approach the inlet temperature of hot fluid.

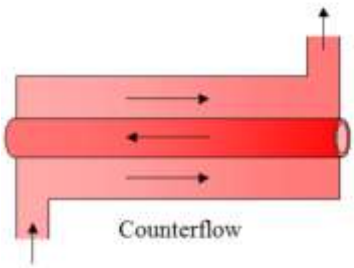

Fig.5 Counter flow heat exchangers

Cross flow heat exchangers:

Heat transfer is usually better when a flow moves across tubes than along their length. Hence, cross-flow is often the preferred flow direction, and tends to be better than parallel flow or counter flow configurations. Cross flow configurations are shown in the three figures below. The first figure shows cross flow over a finned array of tubes. The second and third figures show cross flow over a finned array of tubes, with square fins and circular fins as shown in fig.5. The purpose of fins is to increase heat transfer between the hot and cold mediums.

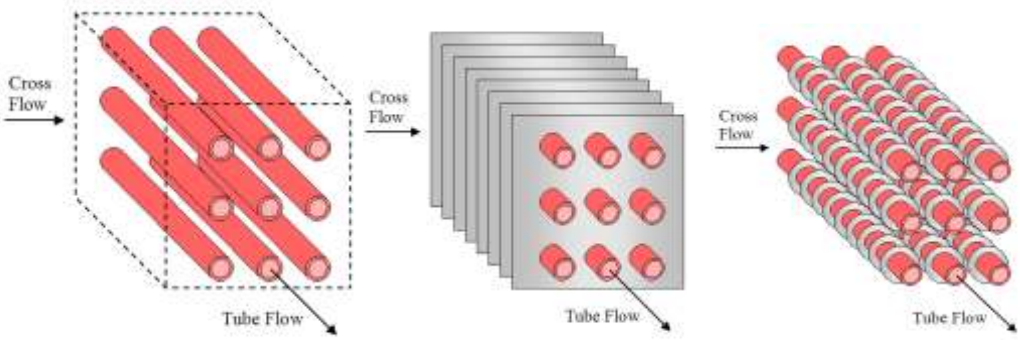

Fig.5 Cross flow heat exchangers

Twisted Tube Heat Exchanger:

Heat transfer rate improvement is one among the quick growing areas of heat transfer technology. In fact techniques are on the market for the advance of various modes of heat transfer. Second and third generation improvement technology is already in use in process industries. In Heat exchanger technology twist type Heat exchangers, extended surface heat exchanger have greater benefits with conventional heaters.

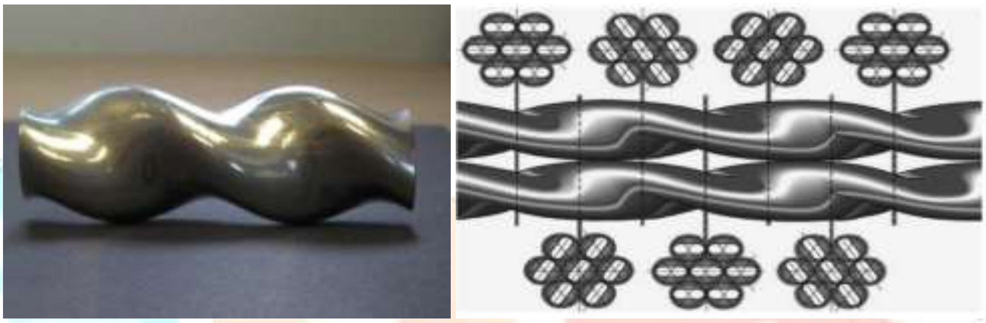

Fig.6 Twisted Tube Heat Exchanger

III. LITERATURE REVIEW

Ganesh Patil, Prof. (Dr.) C.H.Bhosale, Prof. N.N.Shinde, Prof.M.M.Wagh (2015) developed the phase change material based heat exchanger using organic and inorganic material. [1]

Fig.7 Experimental Setup

Pitambar Gadhve and Shambhu Kumar (2012) expounded use such a surface dimple surface to enhance forced convection heat transfer. Heat transfer improvement is based on principle of scrubbing action of cooling fluid inside the dimple which serves turbulent mixing in flow and enhance heat transfer. An experimental set up has been designed and fabricated to study reaction of dimpled surface on heat transfer in rectangular duct. Results compared with flat surface tube and found heat transfer enhancement over the later one.[2]

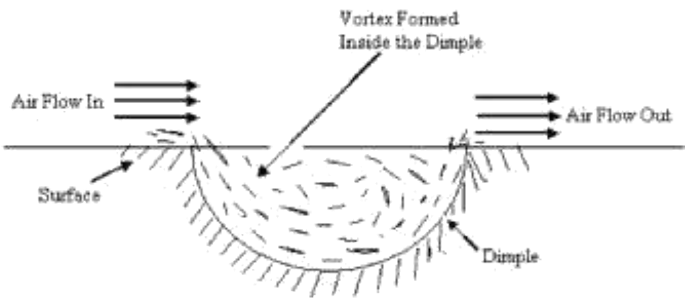

Fig.8 Vortex heat transfer enhancement Mechanism

J.P. HARTNETT & W.J. MINKOWYCZ They were found the average in tube heat transfer co-efficient in spiral coil heat exchanger. The test section is spiral coil heat exchanger which has six layer of concentric spiral coil tube. They achieve the experiment result of tube heat transfer coefficient in spiral coil heat exchanger under dehumidifying conditions. They give the experimental equation and compare with the current correlation and obtain new correlation.[3]

Jamshid Khorshidi, Salman Heidari (2016) disused about the design parameters and fabrication of spiral heat exchanger.[4]

Fig.9 (A) Spiral plate heat exchanger (B) Hot and cold channels (C) Completed heat exchanger

In 1985, **Kanchan Chowdhury, Helmut Linkmeyer, M.KhalilBassiouny**, Holger Martin have present the analytical studies on the temperature distribution in spiral heat exchangers:Straight forward design formulae for efficiency and mean temperature difference. The temperature distribution in SPHE has been calculated numerically to obtain the efficiency and LMTD correction factor F as a function number of transfer units NTU, the number of turns n, and the heat capacity ratio C. It has been found that the LMTD correction factors, when plotted against the number of transfer units per turn NTU/n, fall approximately on a single curve, that curve for balanced counter current operation (C= -1) can be very closely representation of our numerical results it was concluded that a simpler physical model exist to represent the overall behaviour of a SPHE equally well. In fact, a counter current cascade of n concurrent heat exchangers does results exactly in above mentioned formulae for LMTD correction factor. From the model the F-factors for other heat capacity rate ratio C ($-1 < C \leq 0$) can be calculated. [5]

In 2007, M. Picon-Nunez, L. Canizalez-Davalos, G. Martinez-Rodriguez and G.T. Polley have present a shortcut design approach for Spiral heat exchanger. The approach consist of an interactive process where physical dimension like plate width and external spiral diameter are given initial value; convergence is achieved until the calculated pressure drop and heat duty meet the required specification of the design problem. The results of the application of the approach are compared with case studies reported in the literature. A numerical study using computational fluid dynamics is performed to the performance of the geometry. The temperature profiles of the exchanger calculated analytically show the same tendency as those obtained numerically; thus, the method provides a good starting point of estimating the dimensions of spiral heat exchangers in single phase applications.[6]

IV. DESIGN OF TWISTED TUBE HEAT EXCHANGER

These devices consists of helically twisted, double radius oval tubes, welded by their round ends to tube sheets. The purely longitudinal shell-side flow in twisted tube bundles is rarely mentioned in the theory of heat exchanger design, despite its ability to provide considerable surface density, low-pressure drop, and high heat transfer coefficient. The bundle enhances heat transfer, tube-side and shell-side. In the tube-side flow shown in Fig 2, fluid becomes swirled and is affected by wall turbulence and by the different fluid layer velocities. In the course of the shell –side flow over tubes shown in Fig 1, circulation and mixing is generated. A secondary circulation generated by the centrifugal force of swirling also affects the in-bundle flow. Inside and outside helically shaped tubes double to triple the overall heat transfer rate with about doubled hydraulic resistance. A simple use of such tubes instead of plane, rounded tubes are effective to reduce size of the heat exchanger while keeping a similar heat exchanger duty.

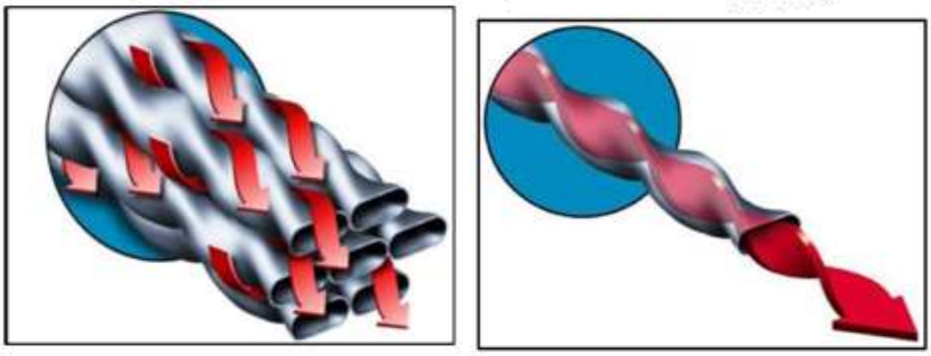

Fig.10 Flow inside Twisted tube Bundle

The twisted tube is a shell-and-tube heat exchanger. It consists of a bundle of tubes fitted inside a cylindrical outer shell. The design differs from a traditional shell-and-tube exchanger by having oval tubes twisted along the longitudinal axis. The number of twists per unit length can vary from design to design. The twist pitch s, is the tube length between each 360˚ twist. Each twisted tube is manufactured with round ends making it possible to fit them into the tube sheets by conventional methods.The tubes are normally manufactured in a one-step operation ensuring a constant wall thickness and that the material yield point is not exceeded. The mechanical properties of the used material are therefore retained. Twisted tubes can be manufactured from a full range of materials including carbon steels, stainless steels, titanium, copper and nickel alloys. Arrangement of the tubes can either be done in a triangular or a rectangular pitch and the tube cross-section can be varied.

Design Methodology

The following assumptions are made in the development of the model are: the effects of the change of velocity at the entry and exit of the exchanger are neglected; pure counter-flow provision is assumed; steady state; the heat transfer coefficient is constant throughout the length of the exchanger; losses to atmosphere are negligible; the bolts separating the plates are considered as not to affect the fluid movement.The basic concept of the methodology is the development of a thermo-hydraulic model that is relates the heat transfer coefficient with the pressure drop through the geometry of the heat exchanger and the physical properties. A set of equations are solved simultaneously to find the exchanger geometry such as diameter spiral shape and width of plate. The initial information is the operating data and also the stream physical properties. Next a plate width (H), space between two plates and plate thickness are fixed.The selection of the right expression for heat transfer and pressure drop depends on the values of the Reynolds and the critical Reynolds number (Eqs.1 and 2)[9]

$$Re = \frac{10000\,(\frac{F}{1000})}{H\mu} \ldots\ldots\ldots\ldots\ldots (1)$$

And

$$Rec = 20000\,(\frac{De}{Dh})^{0.32} \ldots\ldots\ldots\ldots (2)$$

Where F is the mass flow rate, H is the plate width, μ is the viscosity, De is the equivalent diameter and Dh is the hydraulic diameter.

The length of the exchanger L is calculated by using the allowable pressure drop from corresponding equation (3, 4 or 5).

For single phase, Re > Rec

$$\Delta P = 0.001\ \frac{L}{S}\left[\frac{F}{d_s H}\right]^2 \left[\frac{1.3\,\mu^{\frac{1}{3}}}{(d_s + 0.125)}(H/F)^{\frac{1}{3}} + 1.5 + \frac{16}{L}\right] \ldots\ldots\ldots\ldots\ldots (3)$$

Without phase change, 100 < Re < Rec

$$\Delta P = 0.001\ \frac{L}{S}\left[\frac{F}{d_s H}\right] \left[\frac{1.035\,\mu^{\frac{1}{3}}}{(d_s + 0.125)}(\mu_f/\mu_b)^{0.17}(\frac{H}{F})^{1/2} + 1.5 + \frac{16}{L}\right] \ldots\ldots\ldots\ldots\ldots (4)$$

Without phase change, Re < 100

$$\Delta P = \frac{Ls_\mu}{3385\,(ds)^{2.75}}(\mu_f/\mu_b)^{0.17}(H/F) \ldots\ldots\ldots\ldots\ldots (5)$$

Where P is the fluid pressure drop, s is the specific gravity, ds and is the plate spacing.Once the plate length is known, the real pressure drop is calculated according to equations (3, 4 or 5) as applicable. This allows for the calculation of the velocity of the fluid using equation (6).

$$Vf = \frac{F}{Ac} \ldots\ldots\ldots\ldots\ldots (6)$$

Where Ac is the free flow area.The Prandlt number on the hot and cold side can be obtained from equation (7).

$$\Pr = \frac{Cp\,\mu}{K} \ldots\ldots\ldots\ldots\ldots (7)$$

Where, Cp is the heat capacity and k is the thermal conductivity of the fluid.

The correlations for the calculation of heat transfer coefficients as reported by Minton (1970) are given by equations (8 and 9):

Liquid no phase change, Re < Rec

$$h = 1.86Cp\,V_f Re^{2/3} Pr^{2/3}(L/D_e)^{1/3}(\mu_f/\mu_b)^{-0.14} \ldots\ldots\ldots\ldots\ldots (8)$$

Where, h is the heat transfer coefficient, μf is the fluid viscosity at wall temperature conditions and μb is the fluid bulk viscosity. Knowing these values, the overall heat transfer coefficient can be calculated from equation 9:

$$U = \frac{1}{\frac{1}{h_1} + \frac{x\,0.0833}{K_W A_p} + \frac{1}{h_2}} \ldots\ldots\ldots\ldots\ldots (9)$$

Where, U is the overall heat transfer coefficient, kw is the thermal conductivity of the material of construction and Ap is the surface area of the rolled plate.The logarithmic mean temperature difference is obtained from equation 10:

$$\Delta T_{LM} = \frac{(T_{c1} - T_{f2}) - (T_{c2} - T_{f1})}{In\frac{(T_{c1} - T_{f2})}{(T_{c2} - T_{f1})}} \ldots\ldots\ldots\ldots\ldots (10)$$

Where, Tc_1 and Tc_2 are the hot stream inlet and outlet temperature and T_{f1}& T_{f2} are the cold stream inlet and outlet temperature respectively. The heat transfer area is calculated from equation 11:

$$A_T = \frac{q}{U\Delta T_{LM}} \ldots\ldots\ldots\ldots\ldots (11)$$

Where, q is the heat load. Finally, the plate width and the spiral diameter are calculated from equations (12) and (13):

$$H_{calc} = \frac{A_T}{2L} \ldots\ldots\ldots\ldots\ldots (12)$$

$$D_{s\,calc} = [15.36L(ds_f + ds_c + 2p) + C^2]^{1/2} \ldots\ldots\ldots\ldots\ldots (13)$$

If the value of these variables is different from the initial guess, the process starts again using the calculated values in place of the initial ones until convergence is achieved.

V. Calculation

Table 1: Design Parameters

Sr. No.	Parameter	Hot Stream	Cold Stream	Unit
1	Flow rate (m)	0.766	0.766	Kg/s
2	Inlet Temp.	353	298	K
3	Inlet Temp.	323	318	K
4	High Capacity (Cp)	4505	4178	J/(kg*k)
5	Thermal Conductivity (k)	0.348	0.322	W/(m*k)
6	Density	998.2	982.7	kg/m3
7	Relative Density	0.9982	0.9827	
8	Pressure Drop	0.0689	0.0689	Bar
9	Viscosity	0.000134	0.000467	kg/(m*s)
10	Plate thickness (t)	0.025	0.025	m
11	Internal Diameter (Ds)	3.04	3.04	m
12	Outer Diameter (Ds)	0.36287	0.36287	m
13	Length (L)	3	3	m
14	Width (H)	0.0762	0.0762	m
15	Free flow area	0.0036	0.0006	m^2
16	Hydrulic Dia	0.011881188	0.001980198	m
17	Reynolds number (Re)	19292.1531	5260.7402	
18	Critical Reynolds number (Rec)	6696.761221	3774.47379	
19	Mean fluid velocity	0.252124372	1.262122753	m/s
20	Prandlt number	1.72468243	6.059397516	
21	Heat transfer coefficient (h)	5.859589706	74.4947887	W/m^2K
22	Fluid Pressure drop	2171.043745	66703.38818	N/m^2
23	Thermal conductivity (k_m)	20	20	W/m.K
24	Overall heat transfer coeff. (U)	5.427874726		W/m^2K

From the data (As per data of research paper):

- Free Flow area Ac = b*H
  = 0.006*0.6
  = 0.0036 for hot stream
  = 0.0036 for cold stream
- Hydraulic Diameter: Dh = 2bH / b+H
  = 2*0.006*0.6 / 0.006+0.6
  = 0.0119 for hot stream
  = 0.0119 for cold stream
- Reynolds no: Re = (Dh / μ Ac) M
  = 8063.87119 for hot steam
  = 5421.96050 for cold steam
- Critical Reynolds Number

$$Re = \frac{10000\ (\frac{F}{1000})}{H\mu}$$

=6696.76 for hot stream
=3774.47 for cold stream

Here first condition is satisfied Re>Rec so,

- Heat Transfer coefficient

$$h = \left(1 + 3.54\frac{De}{Dh}\right) 0.023CpVfRe^{-0.2}Pr^{-2/3}$$

=2171.043 for hot stream

=66703.388 for cold stream

- Overall Heat Transfer Coefficient:

$$U = \frac{1}{\frac{1}{h} + \frac{T}{K} + \frac{1}{hc}}$$

= 5.4278

Table 2: Design Parameters

Sr. No.	**Parameter**	**Hot Stream**	**Cold Stream**	**Unit**
1	Inlet Temp.	333.4	291.5	K
3	Inlet Temp.	317.7	308.4	K
4	High Capacity (Cp)	4190	4181	J/(kg*k)
5	Thermal Conductivity (k)	0.74	0.68	W/(m*k)
6	Density	998.2	982.7	kg/m^3
7	Relative Density	0.9982	0.9827	
8	Fluid Velocity	0.293	0.293	m/s
9	Mass folw rate	0.665	0.665	Kg/s
10	Viscosity	0.0007978	0.0004658	kg/(m*s)
11	Plate thickness (t)	0.0015	0.0015	m
12	Internal Diameter (ds)	0.559	0.559	m
13	Outer Diameter (Ds)	0.609	0.609	m
14	Plate Spacing(b)	0.05	0.05	m
15	Length (L)	3.048	3.048	m
16	Width (H)	0.07	0.07	m
17	Free flow area	0.0035	0.0035	m^2
18	Hydraulic Dia (Dh)	0.011881188	0.001980198	m
19	Reynolds number (Re)	13682.6809	23435.0426	-
20	Critical Reynolds number (Rec)	9441.110	9703.849	-
21	Prandlt number	2.810	5.203	-
22	Heat transfer coefficient (h)	121.75	124.71	W/m^2.k
23	Fluid Pressure drop	1024.05	1046.03	N/m^2
24	Thermal conductivity of material (km)	43	43	W/m.k
25	Overall heat transfer coeff. (U)	1.4925	-	W/m^2.k

From the data (As per data of research paper):

- Free Flow area Ac = b*H

  = 0.05*0.07

  = 0.0035 m^2

- Hydraulic Diameter: Dh = 2bH / b+H

  = 2*0.05*0.07 / 0.05+0.07

  = 0.05833m

- Reynolds no: Re = (D_h / μ A_c) M

  = 13682.6809 for hot steam

= 23435.0426 for cold steam

- Critical Reynolds Number

$$Re_c = 20000\,(\frac{De}{Dh})^{0.32}$$

=9441.11 for hot stream
=9703.849 for cold stream

Here first condition is satisfied Re>Rec so,

- Heat Transfer coefficient

$$h = \left(1 + 3.54\frac{De}{Dh}\right)0.023CpVfRe^{-0.2}Pr^{-2/3}$$

=121.75 W/m^2.K hot stream
=124.71 W/m^2.K for cold stream

- Pressure Drop:

$$\Delta P = 0.001\,\frac{L}{S}\left[\frac{F}{d_s H}\right]^2\,[\frac{1.3\,\mu^{\frac{1}{3}}}{(d_s + 0.125)}(H/F)^{\frac{1}{3}} + 1.5 + \frac{16}{L}]$$

=1024.05 N/m^2 for hot stream
=1046.03 N/m^2 for cold stream

- Overall Heat Transfer Coefficient:

$$U = \frac{1}{\frac{1}{h} + \frac{T}{K} + \frac{1}{hc}}$$

= 1.4925 W/m^{2}k

- Logarithm Mean Temperature Difference:

$$LMTD = \frac{[(Th1 - Tco) - (Tho - Tci)]}{Ln\frac{(Th1-Tco)}{Tho-Tci}}$$

= 25.44

- Heat Flow Rate:
  Q= U*A*LMTD
  = 37.962 W/m^2

- Effectiveness and NTU:
  Hot fluid heat Capacity rate Ch = mh*Cph = 2744.45 W/k
  Cold fluid heat Capacity rate Cc = mc*Cpc = 2738.55 W/k
  Minimum fluid flow rate (Ch OR Cc) = Cmin
  Maximum fluid flow rate (Ch OR Cc) = Cmax
  Qmax = Cmin (Thi – Tci)
  = 114471.39 W
  And,
  Q = Ch (Thi - Tho)
  = 42813.42 W
  Effectiveness = $\frac{Q}{Qmax}$
  = 0.374

$$NTU = \frac{1}{1\frac{Cmin}{Cmax}}ln\frac{1 - e\frac{Cmin}{Cmax}}{1 - e}$$

=0.666

VI. Modeling of Twisted Tube Heat Exchanger

Twisted Tube heat Exchanger parts modeling in solid works software.

Fig.11 (A) Single Twisted tube Model (B) Tube sheet 3D Model

Fig.12 (A) Twisted tube bundle 3D Model (B) Shell head 3D Model

Fig.13 (A) Shell 3D Model (B) Final Assembly for Twisted tube Heat Exchanger

VII. CFD ANALYSIS

Data from: International Journal of Engineering Research and General Science Volume 3, Issue 1, January-February, 2015 ISSN 2091-2730

Table 3: Temperature at different points

Sr. No	Mass Flow rate (Kg/sec)	Temperature at different points of tube (^{0}C)				
		T1 (At start 0 mm)	T2 (At start 250 mm)	T3 (At 500 mm)	T4 (At 750mm)	T5 (At 1000mm)
1	0.14	64	62	60	58.5	56.5
2	0.12	56	54	53	52	51
3	0.05	58	57	56	54	53

- CFD results for twisted tube:

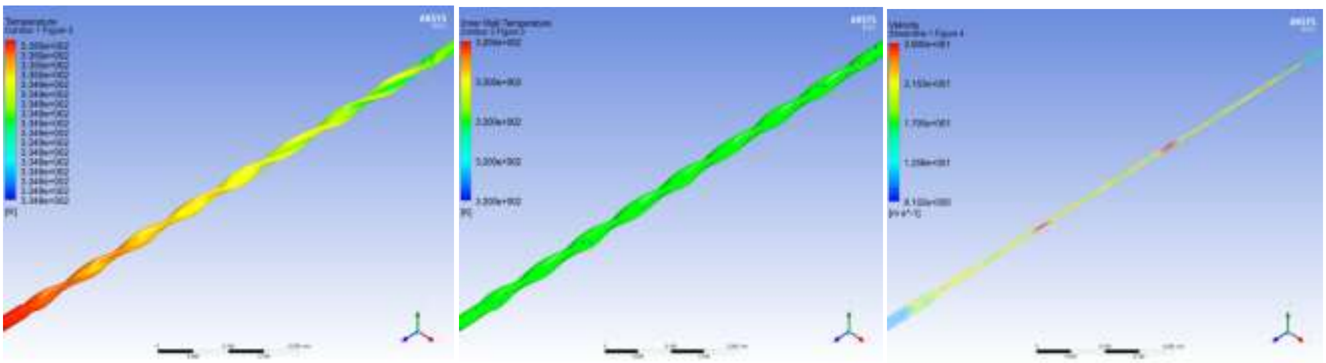

Fig.14 (A) Temperature Contour (B) Inner wall Temperature Contour (C) Velocity Stream Line

VIII. EXPERIMENTATION SET-UP

Material selection: ‘

A wide range of tube materials can be used including carbon and stainless steels, Cr-Mo alloys, duplex and super duplex alloys as well as titanium, zirconium and tantalum. Tube sizes may vary from ½ inch to 1 inch.While following the strategy for deciding candidate materials, the main and considerable factors are cost and reliability. It means using economical materials, resolving problems when they occur and choosing the most reliable material irrespective of the price. Main factors that influence material selection are:

- High heat transfer coefficient
- Low coefficient of thermal expansion and fit with the materials used in tube sheet, tube support and shell to resist thermal cycling.
- Good tensile and creep characteristics
- Good fatigue and corrosion fatigue and creep-fatigue behaviour
- High fatigue toughness and impact strength to prevent fast cracking

Corrosion resistance:

- Nominal corrosion rate to decrease the attack
- Resistance to corrosion from off normal composition caused by leak in upstream heat exchanger or failure in the chemistry control
- Withstand chemistry developed by mixing shell and tube fluids
- Material selection, such as copper, aluminum, carbon steel, stainless steel, nickel alloys, ceramic, polymer, and titanium.

Construction:

The concept of twisted tube heat exchanger is not as simple as it is sophisticated. A tube of length 1m and diameter 15mm is twisted by a twisting method as shown in Fig.15.

Fig.15 Temperature Contour

Apart from twisting there are various methods used in construction of twisted tube heat exchanger. These methods are as follows:

- Drilling
- Welding
- Rolling
- Surface finishing

Fig.16 (A) Drilling Process (B) Welding Process

Practical Experiment:

Fig.17 (A) Storage Tank (B) Submersible Pump (C) Rota meter (D) Thermocouple

Fig.18 (A) Heater (B) Pipes (C) Final Assembly

IX. CONCLUSION

This thesis investigates twisted tube type shell-and-tube heat exchangers in general. It is emphasized on the thermal-hydraulic characteristics, fouling and vibration properties. In order to gather information on performed research on this type of heat exchangers an extensive literature study has been carried out. It appears that research work performed on heat transfer enhancement by inducing swirl flow is mainly directed towards the application of twisted tape inserts in plain tubes. However, useful information and mathematical correlations for heat transfer and pressure losses were collected. The literature study forms the basis for the heat transfer correlations presented in this report. It is a challenging task due to the varieties of correlations for heating, cooling, gas and liquid applications. By extracting correlations from multiple sources it was possible to present reliable formulas for heat transfer in twisted tubes. (Kern’s Method) The conventional alternative has a somewhat lower shell side pressure loss than twisted tube, but both units are within the design specification. The conventional heat exchanger has a very poor hydraulic performance. This may be a result of a design error on the specific unit that is considered. Twisted tube has an advantage with respect to fouling and vibration issues. Fouling is low due to the flow pattern in this type of unit. The helix alternative is also expected to have small risk of fouling. The conventional concept is more exposed to fouling due to flow pattern and the resulting uneven velocity distributions.

Percentage = (over all heat transfer coefficient of twisted tube heat exchanger - over all heat transfer co-efficient of shell and tube heat exchanger) *100/ overall heat transfer coefficient of twisted tube heat exchanger

$$= \frac{705.38-391.10}{705.38}$$

= 44.55%

Twisted tube Heat exchanger= 1.4455*Shell and tube heat exchanger. From calculations and above study we can conclude that the twisted tube heat exchanger is perfect replacement of conventional shell and tube heat exchanger. The twisted tube heat exchanger used fewer tubes, less shell diameter and gives more heat transfer for the same mass flow of fluids.

VII. REFERENCES

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