ENHANCEMENT OF HEAT TRANSFER RATE IN SHELL AND TUBE HEAT EXCHANGER WITH HELICAL TAPES

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ABSTRACT

This paper provides heat transfer and friction factor data for single -phase flow in a shell and tube heat exchanger fitted with a helical tape insert. In the double concentric tube heat exchanger, hot air was passed through the inner tube while the cold water was flowed through the annulus. The influences of the helical insert on heat transfer rate and friction factor were studied for counter flow, and Nusselt numbers and friction factor obtained were compared with previous data (Dittus 1930, Petukhov 1970, Moody 1944) for axial flows in the plain tube. The flow was studied under laminar region. A maximum percentage gain of 165% in heat transfer rate is obtained for using the helical insert in comparison with the plain tube. It is due to the swirl flow motion provided by helical tapes.

Keywords: Enhancement heat transfer, Swirl flow Devices, a Helical tape insert

1.INTRODUCTION

In the past decade, heat transfer enhancement technology has been developed and widely applied to heat exchanger applications; for example, refrigeration, automotive, process industry, solar water heater, etc. The aim of augmentative heat transfer is to accommodate high heat fluxes (or heat transfer coefficient). Up to the present there has been a great attempt to reduce the sizes and cost of the heat exchanger, and energy consumption. The most significant variable in reducing the size and the cost of the heat exchanger, which generally leads to less capital cost and another advantage, is reduction of the temperature driving force, which increases the second law efficiency and decreases the entropy generation. Thus, this captivates the interests of the number of researchers. The great attempt on utilizing different methods is to increase the heat transfer rate through the compulsory force convection. Meanwhile, it is found that this way can reduce the sizes of the heat exchanger device and save up the energy. In general, enhancing the heat transfer can be divided into two groups: One is the passive method; it is the way without being stimulated by the external power such as surface coatings, rough surfaces, extended surfaces, the swirl flow devices, the convoluted (twisted) tube, additives for liquid and gases. The other is the active method. This way requires the extra external power sources, for example mechanical aids, surface-fluid vibration, the injection and the suction of the fluid, the jet impingement, and the electrostatic fields.

The swirl flow devices can be classified into two kinds: the first is the continuous swirl flow and the other is the decaying swirl flow. For the continuous swirl flow, the swirling motion persists over the whole length of the tube for example twisted tape inserts [1, 2], coiled wires inserted along the whole tube [3] and helical grooves in the inner surface of tube generate, while in the decaying swirl flow, the swirl is generated at the entrance of the tube and decays along the flow path for example the radial guide vane swirl generator and the tangential flow injection device [4,5,6,7,8]. For the decaying swirl flow, the heat transfer coefficient and pressure drop decrease with the axial distance, while for the continuous swirl flow, the heat transfer coefficient and pressure drop keep constant. In this reports, the experiments were set to study the effect of swirling flow or rotation flow on the improvement in performance

It is well known that energy transport is considerably improved if the flow is stirred and mixed well. This has been the underlying principle in the development of enhancement techniques that generate swirl flows. Among the techniques that promote secondary flows, twisted-tape inserts are perhaps the most convenient and effective .They are relatively easy to fabricate and fit in the tubes of shell-and-tube or tube-fin type heat exchangers. A typical usage in the multi-tube bundle of shell-and-tube heat exchanger. The geometrical features of a conical tape, as depicted in Fig, are described by its 180º cone pitch *H*, the diameter *d*. In most usage, the helical twisting nature of the tape, besides providing the fluid a longer flow path or a greater residence time, imposes a helical force on the bulk flow that promotes the generation of secondary circulation. The consequent well-mixed helical swirl flow significantly enhances the convective heat transfer[9,10,11).

2.RESEARCH METHODOLOGY

The research methodology involves the prediction of flow and heat transfer behaviors, the available flow procedures for swirling flows and boundary layer are employed to solve the heat transfer calculations

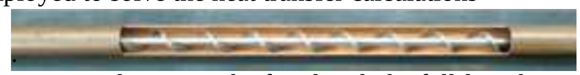


FIGURE 1: The inner tube fitted with the full-length helical tape.

The major assumptions are:

- The flow through the twisted tape is Turbulent and incompressible;
- The flow is in steady state;
- The convection is forced;
- The properties of the fluid are temperature independent;
- Newton's convective law is applicable.

3. FEATURES OF HELICAL TAPES: Helical tape tube inserts causes swirl flow or secondary flow in the fluid. Among the swirl flow devices, helical- tape inserts had been very popular owing to their better thermal hydraulic performance in single phase, boiling and condensation forced convection, as well as design and application issues.

FIGURE 2: Shows a typical configuration of helical tape which is used.

Helical tape inserts increases the heat transfer coefficients with relatively s increase in the pressure drop. It can increase the heat duties of the existing shell and tube heat exchangers. This also leads to increase in the pressure drop and in some cases causes' significant secondary flow. Secondary flow creates swirl and the mixing of the fluid elements and hence enhances the temperature gradient, which ultimately leads to a high heat transfer coefficient.

4.EXPERIMENTAL SETUP OF SHELL AND TUBE HEAT EXCHANGER

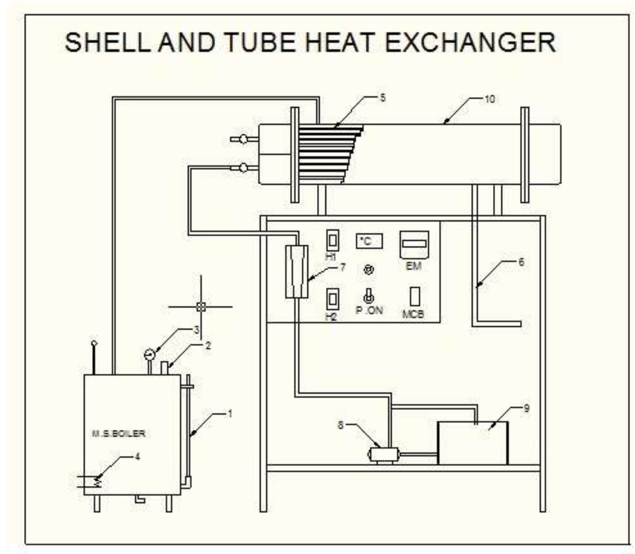


FIGURE 3: Schematic diagram of shell and tube heat exchanger

5.CONSTRUCTION OF SHELL AND TUBE HEAT EXCHANGER

The principal components of an STHE are:

- Shell: Shell diameter should be selected in such a way to give a close fit of the tube bundle. In this setup the shell diameter is usually taken as 0.2m and is made of Stainless steel.
- Tubes : Tube OD of ¾ and 1" are very common to design a compact heat exchanger. With increase in number of tubes, the heat transfer coefficient is increased. Stainless steel is commonly used tube materials.
- Tube pitch: Tube pitch is the shortest centre to centre distance between the adjacent tubes. The tubes is generally triangular patterns (pitch).
- Tube passes: The number of passes is chosen to get the required tube side fluid velocity to obtain greater heat transfer co-efficient and in this setup 2 passes is chosen.
- Rotameter: In this setup two rotameter are used. One is for measuring mass flow rate on shell side and another for tube side. The readings will appear in digital form.
- Pumps: Two pumps are used of half HP pump.
- Heater: heater is provided on one side of tank, to heat water. The hot water is supplied to shell or tube depending upon requirement.

Table No 1:HEAT EXCHANGER DATA AT THE SHELL SIDE

Sl. No	Quantity	Symbol	Value
1	Shell side fluid		Water
2	Shell side Mass flow rate(kg/sec)	Mt	0.060
3	Shell ID(m)	Ds	0.2
4	Shell length(m)	Ls	0.800
5	Tube pitch(m)	Pt	0.03
6	No. of passes	N	1
7	Baffle spacing(m)	Lb	0.2
8	MeanBulk temp.(^{o}C)	ΔT	
9	No. of baffles	N	4

Table no 2 HEAT EXCHANGER DATA AT THE TUBE SIDE

Sl. No	Quantity	Symbol	Value
1	Tube side fluid		Water
2	Tube side Mass flow rate (Kg/sec)	Mt	0.060
3	Tube OD (m)	Do	0.019
4	Tube ID(m)	Di	0.016
5	Tube thickness(m)	Tp	0.0162
6	Number of Tubes	N	18
7	Tube length(m)	Lt	0.825

Table no 3 :FLUID PROPERTIES OF WATER

Sl. no	Property	Unit	Cold water shell side	Hot water tube side
1	Specific Heat (Cp)	KJ/kg. K	4.187	4.187
2	Thermal conductivity (k)	W/m. K	0.00098	0.00098
3	Density (ϱ)	kg/m^3	1000	1000
4	Viscosity (Ω)	kg/m. s	0.00088	0.00086

6.CALCULATIONS PROCEDURE

Shell and tube heat exchanger is designed by trial and error calculations. The procedure for calculating the shell-side heat-transfer coefficient and pressure drop for a single shell pass exchanger is given below The main steps of design following the Kern method are summarized as follow

6.1CALCULATION OF SHELL SIDE HEAT TRANSFER COEFFICIENT

$A_s = \{(P_t - D_o)*D*L_b\} / P_t$

$= \{(0.03-0.01924)*0.2*.2\}/(0.03)$

$= 0.01435\ m^2$

$G_s = ms/ A_s$

$= 0.0257/0.01435$

$= 1.79\ Kg/m^2\ sec$

$D_e = [4*\{(P_t^2*\sqrt{3})/4\} - \{(\pi * D_o^2)/8\}] / [(\pi * D_o)/2]$

$=[4\{(0.03^2*\sqrt{3}/4\}\{(\pi*0.01924^2)/8]/[(\pi*0.0924)/2]$

$= 0.03236m$

$R_{es} = (G_s * D_e) / \mu_s$

$= (1.79/0.03236)/0.00088$

$= 65.82$

$P_{rs} = (C_{ps}*\mu_s) / K_s$

$= (4.187*0.00088)/0.00098$

$= 3.76$

$h_s = 0.36*(K_s / D_e)* (R_e\text{^}0.55)*(P_r\text{^}0.33)*\{(\mu_s/ \mu_w)\text{^}0.14\}$

$=0.35*(0.000616/0.03236)*(65.82^{0.55})*(3.76^{\ 0.33})*1$

$= 0.16882\ W/m^{2o}K$

CALCULATION OF SHELL SIDE PRESSURE DROP

$N_b = \{L_s/ (L_b + t_b)\} - 1$

$= \{0.800/(0.2+0.00162)\}-1$

$Nb+1= 3.96$

$f = \exp \{0.576 - (0.19*Ln\ R_{es})\}$

$= \exp \{0.567-0(0.19*Ln*65.82)\}$

$= 0.8029$

$\Delta P_s = [f* G_s^2* D_s*(N_b+1)] / [2* \rho_s* D_e*\{(\mu_s/ \mu_w)\text{^}0.14\}]$

$= [0.8029*(1.79^2)*0.2*3.96] / [2*1000*0.03236*1]$

$= 0.03148\ Pa$

6.2CALCULATION OF TUBE SIDE HEAT TRANSFER COEFFICIENT

$A_t = \{(\pi * D_i^2) / 4\}*(N_t / 2)$

$= \{(\pi*0.016^2)/4\}*(18/2)$

$= 0.00180\ m^2$

$G_t= m_t/ A_t$

$= 0.0201/0.00180$

$= 11.166\ Kg/m\ sec$

$U_t = G_t/ \rho_t$

$= 11.166/1000$

$= 0.011166m/sec$

$R_{es} = (G_t * D_i) / \mu_t$

$= (11.166*0.016)/0.00086$

$=203.018$

$P_{rt} = (C_{pt}*\mu_t) / K_t$

$= (4.187*0.00088)/0.00098$

$= 3.674$

$f = \{(1.58*\text{Ln } R_{et}) - 3.28\} \wedge (-2)$

$= \{(1.58*\text{Ln } 223.090-3.28\}\wedge(-2)$

$= 0.03822$

$N_{ut} = \{(f/2)*(Ret-1000)* P_{rt}\} / \{1+(12.7*\sqrt{(f/2)}*(P_{rt}\wedge(2/3))-1)\}$

$=\{(0.03822/2)*(203.018$

$1000)*3.674]/\{1+(12.7*\sqrt{(0.038220/2)}*(3.674^{2/3}- 1)\}$

$=16.356$

$h_i = (N_{ut} * K_t) / D_i$

$= (-16.356*0.00098)/0.016$

$= -1.0018\ W/m^{2\ 0}K$

CALCULATION OF TUBE SIDE PRESSURE DROP

$\Delta P_t = [\{(4* f* L_t* n_p) / D_i\} + (4* n_p)]$

$=[\{(4*0.03822*0.825*2)/0.016\}+(4*2)]*[(10$

0 $0*0.011166^2)/2]$

$= 1.473Pa$

7.RESULTS

TUBE SIDE						
Sl. no	Ret	Prt	hi	dPt	Thi	Tho
1	207.75	3.67	-1.102	1.456	40.7	31.9
2	267.7	3.67	-0.922	2.213	41.9	31.9
3	311.1	3.67	-0.82	2.839	42.9	31.9
4	352.45	3.67	-0.736	3.501	42.7	31.9
5	381.39	3.67	-0.684	4.002	43.2	31.9

SHELL SIDE							
Sl.no	Res	Prs	ho	dPs	Tci	Tco	Uc
1	66.15	3.76	0.16	0.03	31.9	34	0.15
2	77.48	3.76	0.18	0.04	32.0	33.9	0.16
3	92.41	3.76	0.20	0.05	32.1	33.8	0.19
4	104.3	3.76	0.21	0.07	32.1	33.8	0.20
5	116.0	3.76	0.23	0.08	32.1	33.8	0.22

8.GRAPHS

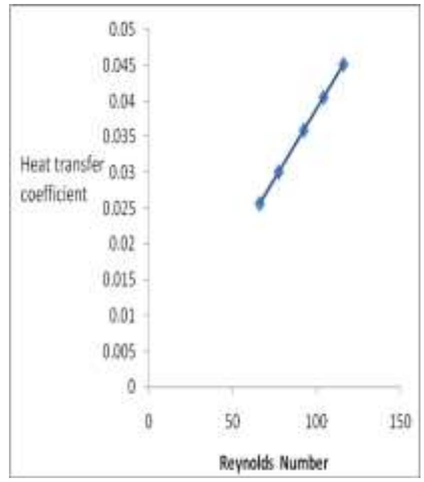
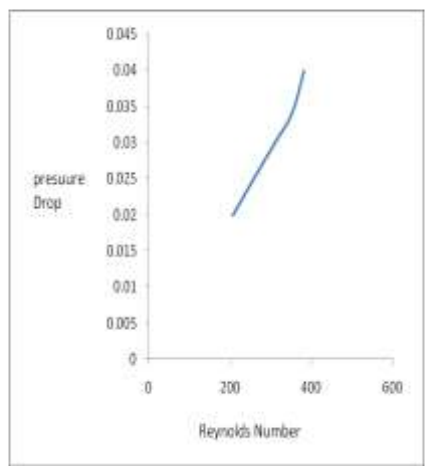
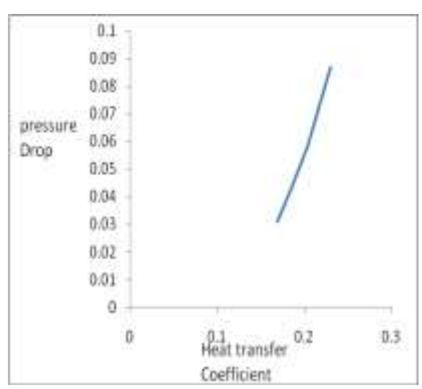


Fig:4 overall heat transfer

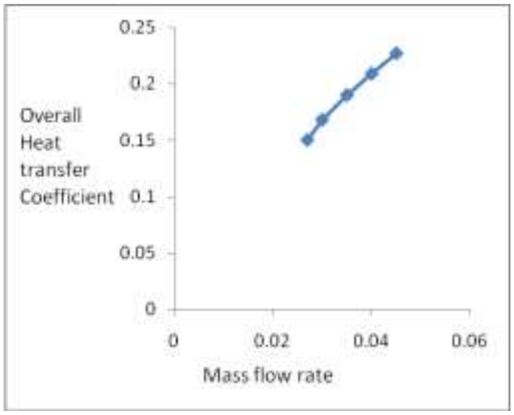
co efficient v/s Reynolds number



FigFig.5 pressure drop v/s Reynolds number(shell side)



FigFig.6 Pressure drop V/S heat transfer coefficient



FigFig .7 overall heat transfer coefficient V/S Mass flow rate

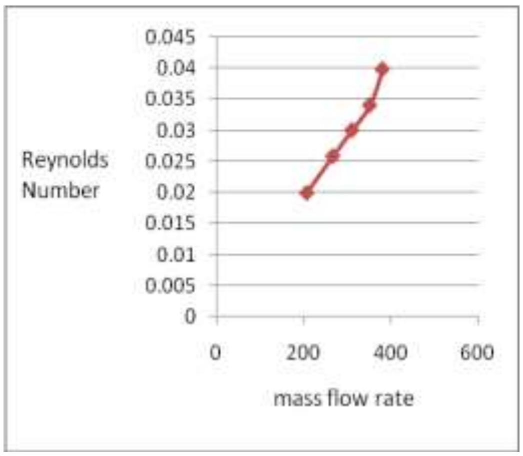


Fig:8Reynolds number (tube side) vs mass flow rate

9.CONCLUSION

The heat transfer enhancement increases due to increased turbulence of water. It is due to the swirl flow motion provided by helical tapes.The friction factor increases with the decrease of helical ratio again due to swirl flow exerted by the helical tape.

The enhancement of Nusselt number is much higher than that of enhancement in friction factor for the same ratio that justifies the usage of helical tape. The performance of conventional shell and tube heat exchanger can be improved by the use of helical tape.

NOMENCLATURE

A_s = Area of the shell side cross flow section
A_t = Area of the tube side cross flow section
P_t= Tube pitch (m).
D_o = Tube outside diameter (m).
D_i = Tube inside diameter (m).
G_s =Shell side mass velocity (kg/ m^2-s).
G_t = Tube side mass velocity (kg/ m^2-s).
U_s = Shell side linear velocity (m/s).
U_t= Tube side linear velocity (m/s).
m_s= Mass flow rate of the fluid on shell side
m_t= Mass flow rate of the fluid on tube side
ρ_s = Shell side fluid density (kg/m^3).
ρ_t = Tube side fluid density (kg/m^3).
D_e= Shell side equivalent diameter (m)
R_{es}= Shell side Reynolds number.
R_{et} = Tube side Reynolds number.
P_{rs} = Shell side Prandtl number.
P_{rt} = Tube side Prandtl number
μ_s = Shell side fluid Viscosity (N-s/ m^2).
μ_t= Tube side fluid viscosity (N-s/ m^2).
μ_w= Viscosity a wall temperature (N-s/ m^2).
C_{ps}= Shell side fluid heat capacity (kJ/kg'K).
C_{pt}= Tube side fluid heat capacity (kJ/kg'K).
K_s = Shell side fluid thermal conductivity
K_t = Tube side fluid thermal conductivity
h_s = Shell side heat transfer coefficient
h_i = Tube side heat transfer coefficient .
f = Friction factor.
ΔP_s= Shell side pressure drop (Pa).
ΔP_t =Tube side pressure drop (Pa).
n_p = Number of tube passes.

10.REFERENCES

1. R.M. Manglik and A.E. Bergles, Heat Transfer and Pressure Drop Correlations for Twisted-Tape Inserts in isothermal Tubes: Part II-Transition and Turbulent Flows, in: Enhanced Heat Transfer, M.B.Pateand M.K.Jensen, eds., ASMEHTD, 202, 99-106 (1992)
2. R.L. Webb, Principles of Enhanced HeatTransfer, Wiley, New York, (1994)
3. R. Sethumadhavan and M.R. Rao, Turbulent Flow Heat Transfer and Fluid Friction in Helical-Wire-Coil-Inserted Tubes, Int.J.Heat Mass Transfer, 26, 12, 1833-1845 (1983)
4. M. Yilmaz, O. Comakli and S. Yapici, Enhancement of heat transfer by turbulent decaying swirl flow, EnergyConvers. Mgmt., 40, 1365-1376 (1999)
5. V.X. Tung, V.K. Dhir, F. Chang, A.R. Karagozian and F. Zhou, Enhancement of Forced Convection Heat Transfer in Tubes Using Staged Tangential Flow Injection, Annual Report, June 1987-Sept. 1988, GRI report No.GRI-89/020 (1989)
6. V.K. Dhir, F. Chang and J. Yu, Enhancement of Single Phase Forced Convection Heat Transfer in Tubes Using Staged Tangential Flow Injection, Final Report, June 1987-Dec.1989, GRI reportNo. GRI-90/0134 (1990)

7. V.K. Dhir and F. Chang, Heat Transfer Enhancement Using Tangential Injection,
8. ASHRAE Transactions, 98, BA-92-4-1(1992) .
9. D. Son, and V.K. Dhir, Enhancement of Heat Transfer in an Annulas Using Tangential Flow Injection, in: Heat Transfer in Turbulent Flows, ASME HTD-Vol. 246, (1993)
10. F.W.Dittus and L.M.K. Boelter, University of California at Berkley, Publications on Engineering, 2: 443 (1930).
11. B. S.Petukhov, Irvine in T. F. and Hartnett J. P., Eds., 1970. Advances in Heat Transfer. Vol. 6. Academic Press. New York. L. F. Moody, Trans. 1944. ASME, 66. 671.