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Effect of Half Length Twisted-Tape Turbulators on Heat Transfer and Pressure Drop Characteristics inside a Double Pipe U-Bend Heat Exchanger

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Abstract

Influences of the half length twisted tape insertion on heat transfer and pressure drop characteristics in a U-bend double pipe heat exchanger have been studied experimentally. In the experiments, the swirling flow was introduced by using half-length twisted tape placed inside the inner test tube of the heat exchanger. The results obtained from the heat exchangers with twisted tape insert are compared with those without twisted tape i.e. Plain heat exchanger. The experimental results revealed that the increase in heat transfer rate of the twisted-tape inserts is found to be strongly influenced by tape-induced swirl or vortex motion. The heat transfer coefficient is found to increase by 40% with half-length twisted tape inserts when compared with plain heat exchanger. It was found that on the basis of equal mass flow rate, the heat transfer performance of half-length twisted tape is better than plain heat exchanger, and on the basis of unit pressure drop the heat transfer performance of smooth tube is better than half-length twisted tape. It is also observed that the thermal performance of Plain heat exchanger is better than half length twisted tape by 1.3-1.5 times.

Keywords: Heat transfer coefficients; heat transfer enhancement; heat exchanger; turbulences promoters; Twist tape.

Nomenclature

A	Area of heat transfer m^2
di	Inside tube diameter, m
d_0	Outside tube diameter, m
d_e	Equivalent Diameter $\{(D_i^2-d_o^2)/ d_0 \}$, m
Di	Inside diameter of annulus pipe, m
D_0	Outside diameter of annulus pipe, m
Y	Twist ratio =Half Pitch/tube diameter {P/di}
L	Length of tube
Δp	Pressure drop, kg/cm^2
M	Volume rate of flow, LPM
M	Mass flow rate, kg/sec
Q	Rate of heat transfer, W
C_p	Specific heat of fluid, KJ/kg-K
Th_i, Th_0	Inlet and outlet temperature of hot fluid, ^oC
Tc_i, Tc_0	Inlet and outlet temperature of cold fluid, ^oC
h_h	Inside heat transfer coefficient for oil, W/m2-k
h_c	Water side heat transfer coefficient for oil, W/m2-k
θ	Log mean temperature difference, K
Re	Reynolds No.
Pr	Prandt1 No.
N_u	Nusselt No.
K	Thermal conductivity of fluid, W/m-k
f	Friction factor
U	Overall heat transfer coefficient, W/m^2-k

Greek Symbols

α	Thermal diffusivity, m^2/sec
μ	Dynamic viscosity of fluid, N-sec/m^2
ρ	Density, kg/m^3
γ	Kinematic viscosity of fluid, m^2/sec

Subscripts

h	For hot fluid (oil)
c	For cold fluid (water)
i	Inlet
o	Outlet

1. Introduction

Heat exchanger is the apparatus providing heat transfer between two or more fluids, and they can be classified according to the mode of flow of fluid or their construction methods. Heat exchangers with the convective heat transfer of fluid inside the tubes are frequently used in many engineering application. Enhancement of heat transfer intensity in all types of thermo technical apparatus is of great significance for industry. Beside the savings of primary energy, it also leads to a reduction in size and weight. Up to the present, several heat transfer enhancement techniques have been developed. Twisted-tape is one of the most important members of enhancement techniques, which employed extensively in heat exchangers.

A detailed survey of various techniques to augment convective heat transfer is given by Bergles [1].Twisted tape techniques have been used to augment heat transfer in

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double pipe heat exchanger. Kumar et al. [2] found that coiled type of swirl generator causes the increase of 72% in heat transfer, but frictional power also increases 90%. For water flowing through a vertical stainless steel tube, pitch to diameter ratio of 1.0 to 5.5 was used. Hong et al. [3] found that in the case of high Prandtl Number fluid in laminar flow, the heat transfer rate increases considerably for a moderate increase in pressure drop. Saha et al. [4] have considered twisted tape element connected by thin circular rods. They observed that the pressure drop associated with the full-length twisted tape could be reduced without impairing the heat transfer augmentation rates in certain situation. Smith Eiamsa-ard et al. [5] conducted experiments on a concentric tube heat exchanger. Hot air passed through inner tube while the cold water was flown through the annulus. A maximum percentage gain of 165% in heat transfer rate was obtained by using the helical insert in comparison with the plain tube.Saha et al. [6] investigated the use of turbulences promoters with short length Twist tape, and regularly spaced Twist tape element. They achieved better thermodynamics performance with short length Twist tape, and regularly spaced Twist tape element instead of full-length Twist tape while working with twisted tape. Saha et al. [7] have considered different types of strips. The strips are of longitudinal rectangular, square and crossed cross-section, full length and short length, as well as regularly spaced types. They observed that short length strips perform better than full-length strips. Friction factor reduces by 8-58% and Nusselt no reduces by 2- 40% for short length strips. For regularly spaced strips elements, Friction factor increases by 1-35% and Nusselt no increase by 15-75%.Smith et al. [8] compared three different heat transfer enhancement methods namely: Microfins, Twist tape, and High fins to that of smooth tube. They observed that the heat transfer coefficients are increased by approximately 46%, 87% and 113% on average compared to those of smooth tube, using respectively Twist tape, High fins, and Micro fins. They also observed that on average the pressure drop of Micro fins tube is 38% higher than those of smooth tube. High fins tube increases the pressure drop by 81% in comparison to smooth tube, and Twist tape increases the pressure drop by 148%. Heat transfer and pressure drop characteristics of laminar water flow through a circular tube with longitudinal inserts were experimentally studied by Hsieh et al. [9]. Testing was performed on tubes with square and rectangular as well as crossed longitudinal strip inserts with Aspect ratio AR=1 and 4. Solanki et al. [10] conducted experimental and theoretical studies of laminar forced convection in tubes with polygon inner cores. Ray et al. [11] investigated experimentally correlations of heat transfer and flow frictions in a square duct with twisted-tape insert.Sivashanmugam et al. [12] investigated experimentally the heat transfer and friction factor characteristics of circular tube fitted with Right-Left helical screw inserts of equal and unequal length of different twist ratio.Naphon [13] investigated heat transfer and the pressure drop characteristics in a horizontal double micro-fin tube with twisted tape inserts. An overview of hundred research works of enhancement heat transfer rate by using twisted tape was reported by Dewan et al. [14]. Anil Singh Yadav [15] investigated heat transfer and the pressure drop characteristics in a double pipe heat exchanger with full length twisted tape inserts.

The objective of this paper is to study heat transfer and pressure drop characteristics of the u bend double pipe heat exchanger with and without half length twisted tape insert. The effects of various relevant parameters on heat transfer characteristics and pressure drop are also investigated. New data gathered during this work for heat transfer and pressure drop characteristics for the u bend double pipe heat exchanger with half length twisted taped insert are proposed for practical applications.

2. Experimental Setup

Figures 1 and 2 show a schematic view of the experimental setup and a concentric double pipe heat exchanger fitted with a half length twisted tape. The set-up consists of:

1. An oil tank with heater of 0.64 m^3 capacity placed on floor.
2. An overhead water tank 0.5 m^3 capacity located at an elevation of 2.75 meters.
3. Double pipe u bend heat exchanger.
4. Measuring devices like Rotameter, temperature indicator, and pressure gauge.
5. Twisted tape.
6. Gear pump.

The oil tank is placed on the floor; and is provided with heating coil of variable input. The tank dimensions are 0.8 m x 0.8 m x 1 m. The tank is provided with PVC tube of 1.85 m long and 5cm diameter, which is connected to 1.5HP motor. The motor outlet is connected to the inlet of heat exchanger through pipe to circulate hot oil in ckt. This pipe is connected to inner tube of heat exchanger through flange coupling. This pipe is provided with different measuring devices like rotameter, temperature indicator, and pressure gauge. An overhead tank is a Sintex tank of 0.5 m^3 located at a height of 2.75 m from the floor. The flow rate of water is kept constant at the rate of 15 Lit/min. Test section is double pipe heat exchanger of u bend type as shown in Figure 1. The Heat Exchanger consists of 2 m lengths in each arm and 0.465 m length of u-bend section. The heat exchanger is made up of stainless steel tubes. The inner diameter of inner tube is 2.11cm, and outer diameter of inner tube is 2.5 cm. Inner diameter of annulus pipe is 5 cm. The two straight legs of inner tube are connected to U-bend section with the help of flange coupling. The test section was heavily insulated by asbestos rope insulation. Rotameter is used to measure the flow rate of oil in the inner tube. Rotameter is connected at inlet to inner pipe of heat exchanger. The range of rotameter is 0-50 Lit/minute. Two Burdon pressure gauges are used at inlet section and another at outlet section of hot oil. The range of Pressure gauge is 0-5 kg/cm^2. The difference in reading of inlet and outlet pressure gauge gives the pressure drop in heat exchanger. Four Digital Thermometers are used at inlet and outlet section of each hot and cold fluid.

In all experiments twisted tapes were made out of 0.8 mm thick stainless steel strip. The width of which was 1 mm less than the inside diameter of test section.

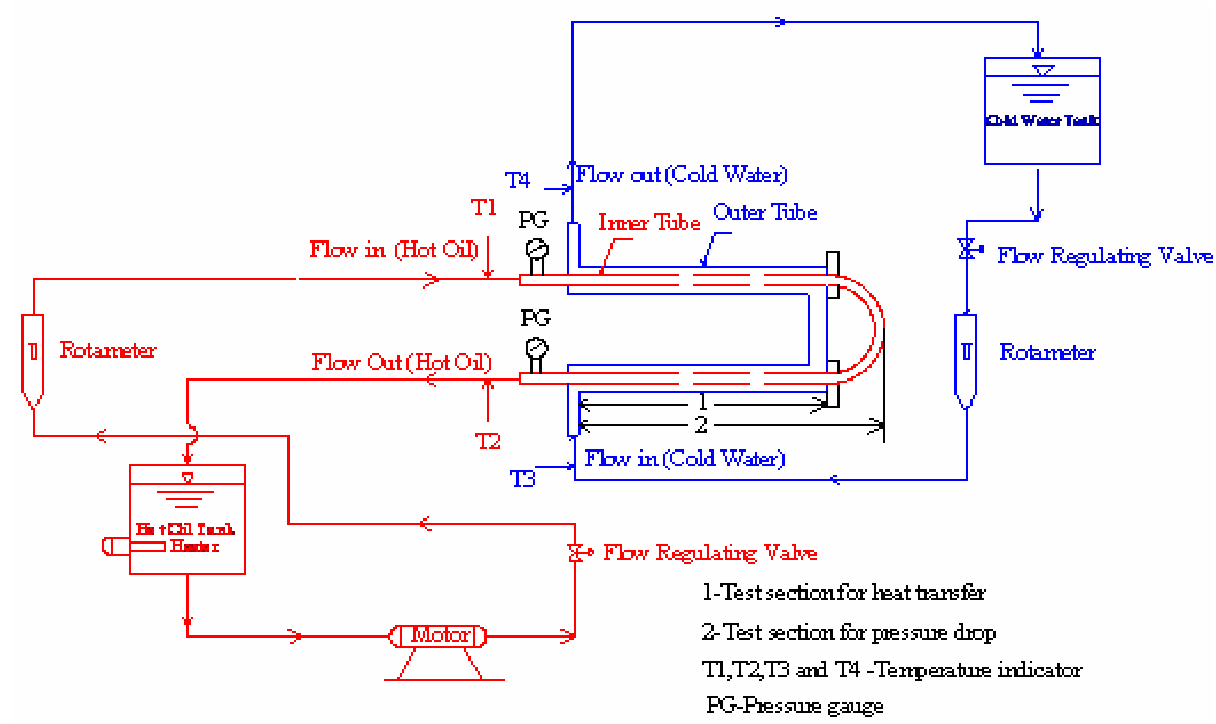

Figure 1: Schematic Diagrams of Experimental Setup.

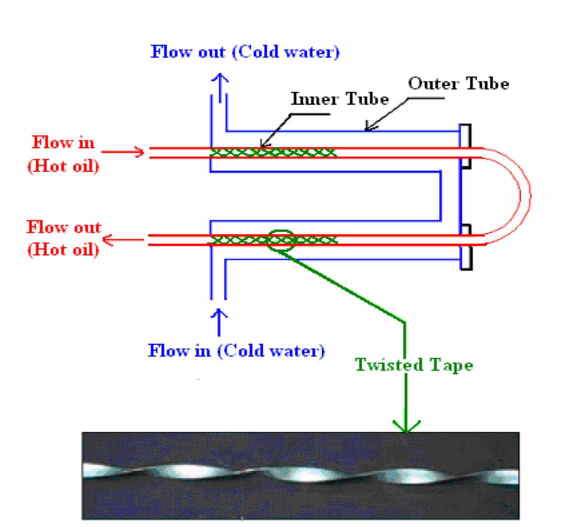

Figure 2: The inner tube fitted with half length twisted tape.

The strip were at first pushed into a tube, and then one end of the strip was tightened in a vice keeping tube in perpendicular position, and other end was twisted by tong. Twisted tapes were manufactured in the Amsler torsion-testing machine to the desired twist ratio; and were later inserted in the test section.

3. Experimental Procedure and Data Reduction

First the plain tube double pipe heat exchanger (i.e. without turbulator) was tested. At the beginning of series of tests, the hot oil was circulated through inner tube and cooling water through annulus tube in counter-flow configuration. The air was bled at various locations. The flow rate of water was fixed to 15 Lit./min. The cooling water coming in heat exchanger was at room temperature. First the oil flow rate was fixed to 2 Lit/min. A prescribed heat input was given to the oil in oil tank in sufficient state. Usually 1/2 hour was required for the attainment of steady state for a run. Once the steady state was reached the flow rates of hot and cold fluid, temperature reading at inlet and outlet section of hot and cold fluid and burden pressure gauge readings were taken. The flow rate of cold water was kept constant, and above procedure was repeated for different flow rates of hot fluid.

After completing the test with plain heat exchanger (i.e. without turbulator), the u bend double pipe heat exchanger was removed from loop. Then half-length twisted tape was inserted into the both straight legs (2m each) of the u-tube. The tape was inserted from one side and pulled from other end by thread or thin wire. Then the heat exchanger was connected in loop; and took various readings. Transformer oil was circulated inside tube and cold water through annulus in counter flow arrangement.

The range of values of various parameters considered in the present investigations is given in table 1.Heat input was determined from the enthalpy rise of the fluid. A linear variation in the bulk temperature was assumed over the test length. The tube wall inside temperature was calculated by one-dimensional conduction equation. The average wall temperature and the bulk mean temperature were combined with heat flux to give the Nusselt No. all the fluid properties were evaluated at the mean film temperature. Pressure drop data was obtained under isothermal condition, and the fanning fraction factor was calculated.

Table 1: Range/values of parameters.

Parameters	Range/values
The flow rate of oil (M_H) (Lit/min)	4, 8, 12,18,24,30
The flow rate of water (M_C)	15 Lit/min (Constant)
ID of inner tube (di)	0.0211 m
OD of inner tube(do)	0.025 m
ID of outer tube(Di)	0.05 m
The water temperature at inlet temperature)	25°C (ambient
Twist ratio for half length twisted tape	7
Thickness of Twisted tape	0.8mm
Length of Twisted tape	2m (2-piece)
Heat exchanger area(A_0)	0.628m^2
Test length of heat exchanger:	
For heat transfer	8m
For pressure drop	8.46m

4. Data Reduction Equations

The data reduction of the measured results is summarized in the following procedures:

For fluid flows in a concentric tube heat exchanger, the heat transfer rate of the hot fluid (oil) in the inner tube can be expressed as:

$$q_h = m_h c_h (T_{hi} - T_{ho})$$

Where m_h is the mass flow rate of hot oil, c_h is the specific heat of hot oil, T_{hi} and T_{ho} are the inlet and outlet hot oil temperatures, respectively.Heat transfer coefficient on hot oil side (h_h) can be calculated from

$$h_h = Nu_h \left(k_h / d_i \right)$$

Where Nu_h is the Nusselt number for hot oil side and is given by:

1. For turbulent flow

$$Nu_h = 0.023 \operatorname{Re}_h^{0.8} \operatorname{Pr}_h^{0.4}$$

2. For laminar flow

$$Nu_h = 1.86 \left(\frac{d_i}{L} \operatorname{Pr}_h \operatorname{Re}_h \right)^{0.33} \left(\frac{\mu}{\mu_w} \right)^{0.14}$$

While the heat transfer rate of the cold fluid (water) in the annulus is

$$q_c = m_c c_c (T_{co} - T_{ci})$$

Where m_c is the mass flow rate of cold fluid, c_c is the specific heat of cold fluid, T_{ci} and T_{co} are the inlet and outlet cold fluid temperatures, respectively.Heat transfer coefficient on cold fluid (water) side (h_c) can be calculated from

$$h_c = Nu_c \left(k_c / d_e \right)$$

Where Nu_c is the Nusselt number for cold fluid (water) side and is given by

1. For turbulent flow

$$Nu_c = 0.023 \operatorname{Re}_c^{0.8} \operatorname{Pr}_c^{0.4}$$

2. For laminar flow

$$Nu_c = 1.86 \left(\frac{d_e}{L} \operatorname{Pr}_c \operatorname{Re}_c \right)^{0.33} \left(\frac{\mu}{\mu_w} \right)^{0.14}$$

The overall heat transfer coefficient based on Outer surface (U_c) can be determined from

$$U_O = \frac{1}{\frac{1}{h_c} + \left(\frac{d_o}{k} \right) \ln \left(\frac{d_o}{d_i} \right) + \left(\frac{d_o}{d_i} \right) \left(\frac{1}{h_h} \right)}$$

Friction factor, *f*, can be calculated from

$$f = \frac{\Delta p}{\left(\rho u^2 / 2 \right) \left(L / D_i \right)}$$

Where ΔP is the pressure drop across the test section, ρ is the density of oil, d_i is the inner diameter of tube, u is the velocity of oil, and L is the length of tube.

5. Result and Discussion

After having studied heat transfer and pressure characteristic, it becomes necessary to combine these to evaluate the performance of half-length tapes. For this purpose, their performance was studied, for each heat flux separately, for equal mass flow rates and unit pressure drop.

5.1. Equal Mass Flow Rate Basis

Figure 3 shows the performance evaluation for the half-length tapes on equal mass flow rates basis. This is a simple criterion for performance evaluation.

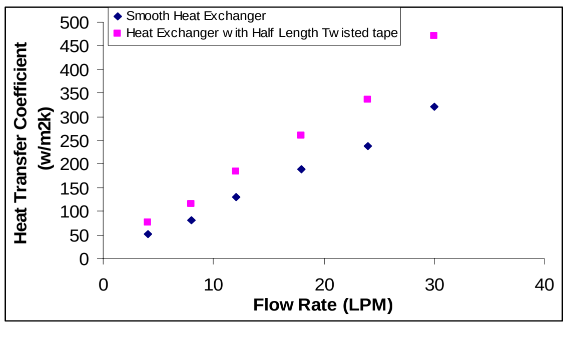

Figure 3: Flow rate vs. heat transfer coefficient.

Figure 3 shows that the average heat transfer coefficient inside tube increases with increase in the flow rate of fluid in each case. On comparing the different curves, it has been observed that heat transfer performance of half-length twisted tape is maximum followed by smooth tube. The heat transfer coefficient is increased by approximately 40% on average compared to those of smooth tubes using half-length twisted tape.

The increase in heat transfer coefficient from smooth tube to twisted tape can be well understood by boundary layer phenomenon. In smooth tubes, the flow is stream lined flow. Due to slip condition, the fluid in contact with tube (wetted perimeters) flow at very slow speed than inner core of tube. Due to this, boundary layer thickness is high, and heat transfer is retarded. The boundary layer thickness may be reduced by fitting turbulators to heat transfer surfaces. These twisted tape tabulators interrupt the fluid flow, so that a thick boundary layer cannot form.

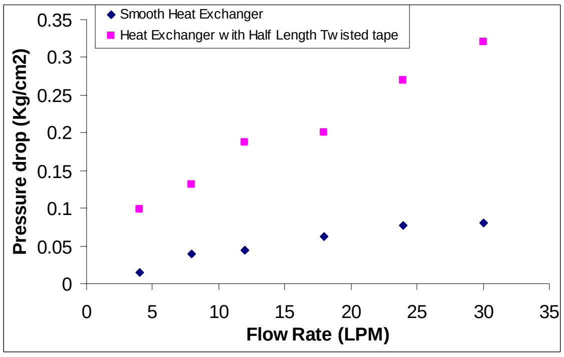

Figure 4: Pressure drop (ΔP) Vs. Flow rate.

5.2. Unit pressure drop basis

Unit pressure drop basis is an important criterion in heat exchange equipment design. An augmentative technique, which is effective from the heat transfer point of view, may fail in case it results in a pressure drop penalty greater than what the equipment can handle.

The increase in pressure drop is certainly a disadvantage resulting out of the use a turbulence promoter. The advantage gained in terms of increase of average heat transfer coefficient by using a turbulence promoter is partially offset by the increased pumping power requirements. In order to study the relative advantage of turbulence promoter vis-à-vis its disadvantage, the study of the parameter heat transfer coefficient per unit pressure drop appears to be appropriate. Figure 4 shows the plots of Pressure drop (ΔP) against Flow rate.

Figure 5 shows the plots of hi/ΔP against Flow rate. Thermal performance ratio of the heat exchanger is ratio of heat transfer coefficient to pressure drop. Thermal performance ratio = $h_i/\Delta P$ ($mk^{-1}s^{-1}$). On comparing the different curves of figure 5, it has been observed that Thermal performance ratio of smooth tube is maximum followed by half-length twisted tape. It has been observed that thermal performance of smooth tube is better than half length twisted tape by 1.0- 1.3 times. Thermal performance ratio decreases with use of turbulators because of increase in pressure drops more than increase in heat transfer coefficient.

6. Conclusion

From the present investigation on double pipe heat exchanger with and without twisted tapes inserts at different mass flow rate of oil, it was found that:

1. As compared to conventional heat exchanger, the augmented (with turbulator) heat exchanger has shown a significant improvement in heat transfer coefficient by 40% for half-length twisted tape.
2. On equal mass flow rate basis, the heat transfer performance of half-length twisted tape is maximum followed by smooth tube.
3. On unit pressure drop basis, the heat transfer performance of smooth tube is maximum followed by half-length twisted tape. It has been observed that thermal performance of smooth tube is better than half length twisted tape by 1.3-1.5 times.

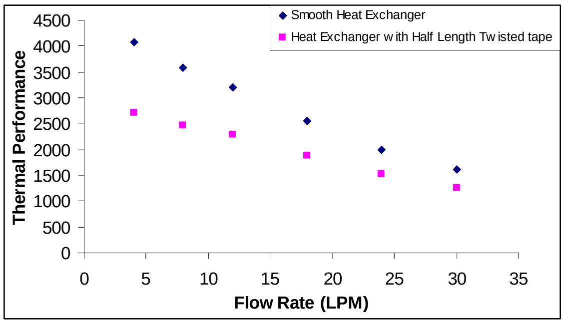

Figure 5: Flow rate Vs. Thermal Performance Ratio.

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