PERFORMANCE IMPROVEMENT OF DOUBLE PIPE HEAT EXCHANGER BY USING TURBULATOR

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Abstract

The various techniques for achieving improved heat transfer are usually referred to as "heat transfer augmentation" or "heat transfer enhancement" and the heat exchanger provided with heat transfer enhancement techniques as "Augmented Heat Exchanger". The objective is to reduce as many of the factors as possible: Capital Cost, Power Cost, Maintenance Cost, Space and Weight, Consistent with safety and reliability. Present work describes the principal techniques of industrial importance for the augmentation of single phase heat transfer on the inside of tubes namely twisted tapes. So twisted tape should be used in heat exchanger when high heat transfer rate is required and pressure drop is of no significance.

Keywords: *Nusselt number, Prandalt number, heat exchanger, twisted tapes.*

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1. INTRODUCTION

Heat exchangers are important engineering devices in many process industries since the efficiency and economy of the process largely depends on the performance of the heat exchangers. High performance heat exchangers are therefore very much required. Improvement in performance may result in reduction in size of heat exchanger. Alternatively high performance heat exchangers of a fixed size can give an increased heat transfer rate; it might also give a decrease in temperature difference between the process fluids enabling efficient utilization of thermodynamic availability. The vital present day need to conserve energy and materials has resulted in huge growth in research and development to improve heat transfer equipment design.

2. HEAT TRANSFER ENHANCEMENT

Heat transfer enhancement is the practice of modifying a heat transfer surface to increase the heat transfer coefficient between the surface and a fluid. The effects of heat transfer enhancement are:

- To reduce the heat transfer surface area required for a given application and thus reduces the heat transfer surface required for a given application which in term reduces the heat exchanger size and cost.
- Increase the heat duty of the exchanger.
- Permit closer approach temperature.
- Reduce the fouling factors
- All of these can be visualized from the expression for heat duty for a heat exchanger.

2.1. AUGMENTED DOUBLE PIPE HEAT EXCHANGER

One of the passive methods that can be used in shell and tube heat exchanger is the swirl flow method. Swirl flow can be produced by inserting twisted tapes in the tubes of double pipe heat exchanger.

Twisted tape is a metallic tape twisted about its longitudinal axis and inserted in a tube. The twist ratio is defined as the ratio of half pitch of the tape to the tube inside diameter. Usually tape is easy to make and fit. The strips were snuggly fitted inside the tube.

According to reference (1) the following correlation for Nusselt number and friction factor may be used in case of turbulent flow:

N_u = F [0.023{1+ (π/2y) 2}0.4 R_e 0.8 P_r 0.4 +0.193 {2R_e (y)-1(d_e/d_i) β ΔT P_r} 1/3]

f= 0.046(y)-0.046 R_e-0.2

where,
β = Thermal expansion coefficient of fluid (1/k)
ΔT = T_w – T_b = (wall temp – bulk fluid temp) (k)

D_i = tube inside diameter
d_e = hydraulic diameter
y = twist ratio = $(P_h/2d_i)$
P_h = helical pitch (360^o twist)

F is fin factor which represents the ratio of total heat transfer to heat transferred by wall alone. If the tape is loose F = 1.0, but may be as high as 1.25 for a tight fit.

3. DESIGN CONSIDERATIONS

a. Boundary layer effect

In order to increase the heat transfer, the techniques must affect the boundary layer (laminar and buffer layers) by reducing its thickness, or increasing its surface area, or increasing the turbulence.

The boundary layer thickness may be reduced by fitting protuberances to the heat transfer surface. These interrupt the fluid flow so that a thick boundary layer cannot form. Alternatively a boundary layer thickness may be reduced by imparting a rotational motion to a fluid flowing inside a tube. The boundary layer surface area may be increased by extending the surface with fins, spines, coils and strips, etc. The turbulence may be increased on the internal and external surfaces by artificial roughening, or using special devices inside tubes known as turbulence promoters.

b. Relative thickness of the thermal and momentum boundary layers

The thickness of the momentum and thermal boundary layer are not necessarily the same. For many fluids the viscous boundary layer is thicker than the thermal boundary layer. Liquid metals are notable exceptions to this rule.

The relationship is generally expressed in a dimensionless group called the pandtl number, after the great German physicist Ludwig Prandtl.

The Prandtl Number P_r is defined simply as the ratio

$$P_r = \nu/\acute{\alpha}$$

where; ν = Kinematic viscosity of fluid
$\acute{\alpha}$ = Thermal diffusivity of fluid.

The kinematic viscosity is indicative of the rate at which momentum diffuses through a fluid because of molecular motion. The thermal diffusivity is indicative of the rate of diffusion of heat in the fluid. The ratio of these quantities is therefore a measure of the relative magnitudes of diffusion of momentum and heat in the fluid. These diffusion rates are precisely the quantities that determine how the thick boundary layer will be for a given flow: large diffusivity means that viscous or temperature effects are expressed further out in the flow.

The Prandtl number is therefore correction between the velocity field and the temperature field. Now

$$\nu = \mu/\rho \quad \text{and} \quad \acute{\alpha} = k/(\rho\, Cp)$$

Then $$P_r = \nu/\acute{\alpha} = Cp\ \mu / k$$

The common values of $0.7 < P_r < 1$ for many gases is striking. Also many liquids, apparently dissimilar have a P_r in range 2 to 4.

c. Heat transfer coefficient

The rate of heat transferred q can be calculated very simply from the equation

$Q = h\, A\, (Tw - T\, \infty)$

Where, q = rate of heat6 transferred
h = convective heat transfer coefficient
A = contact area for heat transfer between fluid and wall
T_w = wall temperature
T_∞ = Fluid free stream temperature.

It is both convenient and conventional to express the heat transfer in terms of yet another dimensionless group, the Nusselt number Nu, after Wilhelm Nusselt, another noted German engineer and scientist active in developing the science of heat transfer in the 1930s. The Nusselt number is composed of three elements:

$$Nu_x = h.x/k$$

For Laminar flow in tube

$$Nu_d = hd_o/k$$

For turbulent flow in smooth tube

$$Nu = 0.023\, Re^{0.8}\, P_r^{\ n}$$

d. Pressure drop

Pressure drop inside circular tube is given by fanning equation

$$\Delta p = \frac{4fLV^2}{2gd_i}$$

4. EXPERIMENTAL PROCEDURE

It is essential to have calibrated instrument in order to get the good experimental results. The heat exchanger special designed for this work was calibrated and was tested for heat balance. It was found that approximate 2 to 5 % error was found in heat load of hot fluid and cold fluid side of heat exchanger.

The experiments were carried out in a experimental facility shown in Fig.1.
The set-up consisted of:

1. An oil tank with heater of $0.64m^3$ capacity placed on floor.
2. An overhead water tank ($0.5m^3$ capacity located at an elevation of 2.75 meters).
3. A double pipe u bend heat exchanger.
4. Measuring devices like Rotameter, temperature indicator, pressure gauge.

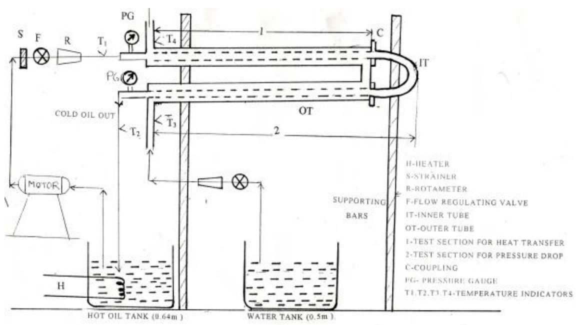


Fig.1 Experimental Setup For Double Pipe Heat Exchanger

First the plain tube double pipe heat exchanger (i.e. without tabulator) 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 is at room temperature. First the oil flow rate was fixed to 2 lit/min. a prescribed heat input was given to the oil in tank and steady state was allow. Once the steady state was reached the flow rate of hot and cold fluid, temperature reading at inlet and outlet section of hot and cold fluid and bourdon pressure tube readings were taken. The series of readings up to steady state are shown in table. The flow rate of cold water was kept constant and above procedure was repeated for different flow rate of hot fluid viz. 4,8,12,18,24,30 (lit/min) one after other.

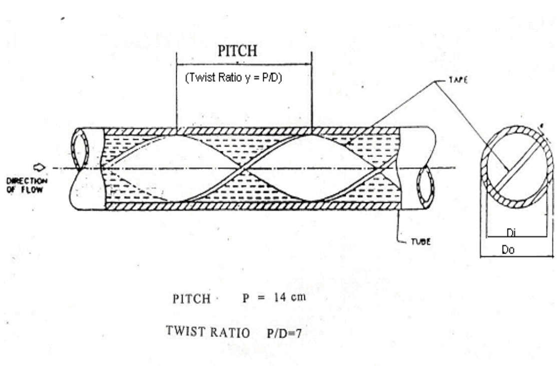


Fig.2 Twisted tape inside a circular tube

After completing the test with plain heat exchanger (i.e. without turbulator), the u bend double pipe heat exchanger was removed from loop. Then twisted tape-1 of pitch = 15 cm was inserted into the both straight legs (4m 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. The same experimental procedure was repeated with two sets of twisted tape inserts. Throughout the experimental work the hot transformer oil was circulated inside tube and cold oil through annulus in counter flow arrangement.

5. RESULTS AND DISCUSSION

Measurements were conducted for a different range of mass flow rates for smooth tube and for different types of turbulators alternatively from 4 LPM to 30 LPM. The comparison in heat transfer coefficient and related pressure loss are of obvious importance.

In this work a comparison is made between smooth tube and different augmentation techniques using turbulators of twisted tape-I and twisted tape –II.

a. Effect of Mass Flow Rate on Heat Transfer:

Graph-I shows that average heat transfer Coefficient inside tube increases with increase in the flow rate of fluid in each cases. For the flow rate between 2 LPM to 30 LPM the average heat transfer coefficient increases from 42 W/m-k to 175 W/m-k for smooth tube, 139 W/m-k to 277 W/m-k for twisted tape-I, 155 W/m-k to 298 W/m-k for twisted tape-II. This is an expected result because with increase in flow rate of fluid flowing through tube, the flow velocity increases, which in term increases the Reynolds's number for the fluid. And Nusselt number is directly related to Reynolds no. as given by Dittos- Boeltier equation for turbulent flow through tubes

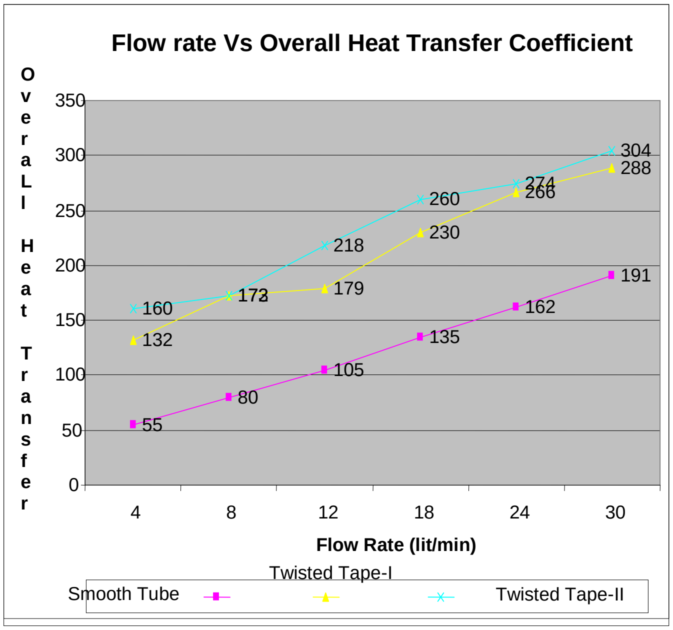
$$N_u = 0.023\ Re^{1/8}\ P_r^{1/4}$$

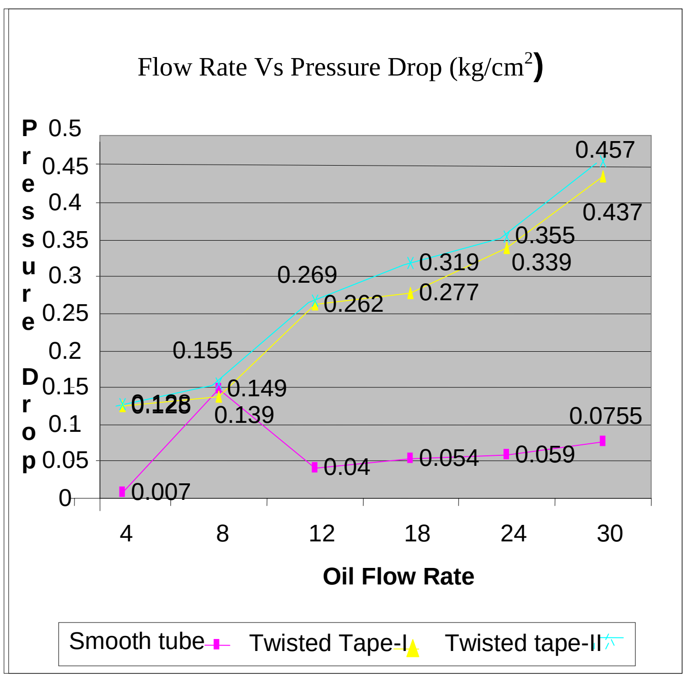
On comparing different curves of Fig1 it has been observed that heat transfer coefficient of turbulator tubes are higher than that of the smooth tube for the same mass fluxes. For example at 30 LPM, heat transfer coefficient of smooth tube is 175 w/m k, for turbulator is 277 w/m-k and for turbulator II is 298 w/m-k. This shows that heat transfer capacity for turbulator I & II is higher than smooth tube at the same mass fluxes. The heat transfer coefficient are increased by approximately 61%, 78 % on an average compared to those of smooth tubes.

b. *Effect of Mass Flow Rate on Pressure Drop*

The pressure drops for different types of enhancement methods are shown in graph 2. With increase in flow rate from

2 LPM to 30 LPM pressure drop increase from 0.00514 to 0.0747 kg/cm2 for smooth tube, from 0.12 kg/cm^2 to0.45 kg/cm^2 for tape-I and from 0.128 to 0.57 kg/cm^2 for tape-II.

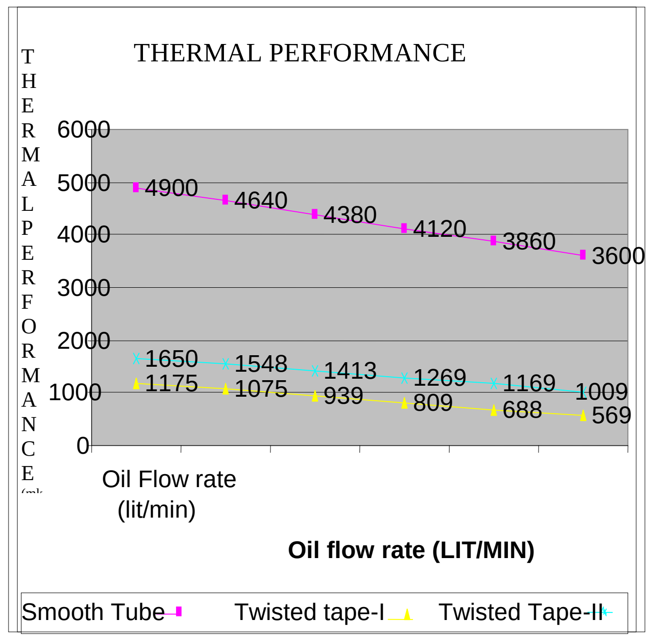




c. *Effect of Mass Flow Rate on Thermal Performance*

Performance of heat exchanger for equal pressure drop becomes important when different heat transfer enhancement techniques are to be compared. In this work performance of heat exchanger for unit pressure drop is done according to reference [5]. Thermal performance of heat exchanger is ratio of heat transfer coefficient to pressure drop.

Thermal performance for each case is also shown in graph 3. It has been observed that thermal performance for smooth tube is better than twisted tape by 1.3 times at low flow rate to 1.54 times at higher flow rate. This is because at higher flow rate pressure drop increases fastly in twisted tape. Thermal performance decreases with use of turbulators because increase in pressure drop is more than increase in heat transfer coefficient. But this increase in pressure drop does not influence pumping cost by significant amount as pumping cost is proportional to the square root of pressure drop.



$$\text{Thermal performance} = \frac{h_i}{\Delta P} \,(\text{m k-1 s-1})$$

So twisted tape should be used in heat exchanger when high heat transfer rate is required and pressure drop is of no significance. The above results are in good agreement with the similar results shown with twisted tape by F. J. Smith and J. P. Meyer in ref [7].

CONCLUSION

After investigating different heat transfer augmentation techniques it has found that:

As compared to conventional heat exchanger the augmented has shown a significant improvement in heat transfer coefficient by 61 % for twisted tape I and 78% for twisted tape II.

When only heat transfer capacity of heat exchanger is criteria regardless of pressure drop or pumping power the twisted tape is more superior as compared with smooth tube (1.6 to 1.8 times).

On equal pressure drop and equal pumping power basis the smooth tube is better to twisted tape (1.3 to 1.7 times).
With increase in floe rate of oil keeping flow rate of water constant the thermal performance decreases slightly for each other.

Twisted tape of lower twist ratio (p/d =3.5) gives higher heat transfer coefficient (by 1.39 times) than higher twist ratio of p/d = 7.

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