Enhancement of Heat Transfer in Heat Exchanger using Punched and V-cut Twisted Tape Inserts

#1 Imran Quazi, Prof. #2 V.R. Mohite, #3 Prof. J Bali

1 imranquazi1987@gmail.com
2 vrmohite@gmail.com
3 bali.jagadeesh@gmail.com

#1 DPCOE-Mechanical Department, SPP University
Pune, India
#2 BSIOTR
Wagholi, Pune, India

ABSTRACT

The heat transfer enhancement is crucial factor in design of heat exchangers. Rise in heat transfer rate can be achieved using various inserts in the tubes of heat exchangers. In the present research work, the outcomes observed after conducting experiments including the amplification of turbulent flow heat transfer in a horizontal circular tube with the help of punched and v-cut twisted tape inserts in which air is used as the working fluid, are represented. The procedure was conducted for plain tube with and without punched and v-cut twisted tape insert keeping wall heat flux constant and mass flow rates are varied. The punched and v-cut twisted tapes are made of aluminium and copper, but with three different twist ratios as 9, 10 & 11. The range of Reynolds number used is between 5000 to 10500. Both heat transfer coefficient and pressure drop are calculated and the outcomes are compared with those of plain tube. It was found that the rise of heat transfer with punched and v-cut twisted tape inserts as compared to plain tube varied from 18% to 52% for various inserts. This rise was observed due to extensive turbulence obtained as a result of the geometry of the inserts used.

Keywords — punched and v-cut twisted tape insert, heat transfer enhancement, twist ratio.

I. INTRODUCTION

Number of techniques (both passive and active) are investigated for increasing the heat transfer rates inside circular tubes and variety of inserts have been used, especially if it is a case of turbulent flow. Some of these means are applied to increase thermal performance of heat transfer devices such as treated surfaces, coarse surfaces, swirling flow devices, coiled tubes, and surface tension devices [1]. Twisted tape swirl tabulator is one of the widely used passive technique for heat transfer augmentation as they present advantages of stable performance, simple structure and ease of installation [2]. Sarma et al. [3] presented generalized correlations to predict friction factor and convective heat transfer coefficient for a tube equipped with twisted tapes over a wide range of Reynolds number and the Prandtl number. Optimized design of recurrently set apart short-length twisted tapes for a circular tube for turbulent heat transfer was conducted by Wang et al. [4] by using computational fluid dynamics (CFDs) modeling. Eiamsa-ard et al. [5] projected experimental study of convective heat transfer inside a circular tube hving short-length twisted tapes inserted under uniform heat flux. Akhavan-Behabadi et al. [6] conducted similar experiments to analyze effects of twisted tapes on heat transfer enhancement and pressure drop in evaporators placed horizontally. The working fluid used was R-134a. Heat transfer and friction factor of CuO/water nanofluid and
water were experimentally checked in circular tube fitted with modified twisted tapes as another substitute [7–9]. Eiamsa-arde et al. [10] carried research works on heat transfer and friction factor features in a double pipe heat exchanger equipped with twisted tape component. They executed their analysis for both continuously placed twisted tape and twisted tape placed with various free spaces inside a circular tube. The heat transfer increase and pressure drop through condensation of HFC-134a in a horizontal tube fitted with twisted tapes were experimentally studied [11]. Jaisankar et al. [12] experimentally examined the heat transfer, friction factor and thermal behaviour caused by twisted tape for solar water heater.

Tape width, twist ratio, space ratio, rod-diameter and phase angle effects on heat transfer and pressure drop were analysed experimentally in a circular tube fitted with regularly spaced twisted tape elements [13]. Naphon [14] also conducted experiments by using conventional twisted tape inserts in horizontal double pipe. Ferroni et al.[15] conducted particular experiments in circular tube fitted with physically separated, multiple, short-length twisted tapes. Convective laminar heat transfer augmentation in twisted tape inserted tube was conferred experimentally by Sarma et al.[16]. In some of the studies, researchers concentrated the thermal effects of twisted tape inserts in modified tube instead of smooth tube, for example; Thianpong et al. [17] examined heat transfer enhancement in a dimpled tube with a twisted tape swirl generator inserted inside. They also suggested the empirical correlations based on the experimental data obtained from their study for predicting the Nusselt number and friction factor for Reynolds number variation from 12,000 to 44,000. Bharawaj et al. [18] performed experiments by using conventional type of twisted tapes to define pressure drop and heat transfer characteristics of water in a spirally grooved tube 75-start. Some researchers[19] modified the conventional twisted tape geometries, for example; Murugesan et al. [20] conducted experiments with V-cut twisted tapes to understand change in heat transfer and pressure drop for a circular tube.

II. EXPERIMENTAL SETUP

The punched and v-cut twisted copper tape used in this experimental study is shown in Fig 2. A 0.24 hp blower is used to forcefully send the air through the test tube. Uniform heat flux is applied to external surface of the test section with the help of heating coil by electrical winding and its output power is controlled by a variac transformer to supply constant heat flux along the whole section of the test tube. The outer surface of the test tube is well insulated with glass wool to reduce the convective heat loss to the surroundings. The external surface temperatures of the test tube wall are measured by 6 K-type thermocouples. These were placed on the outer wall of the test section. Also, the inlet and outlet temperatures of the bulk air are measured by 2 K-type thermocouples at specific points. For measurement of pressure drop an inclined manometer was used across the test tube to have accurate readings. After passing through the test tube, air then enters the orifice meter, which is used for finding the volumetric flow rates. For this purpose a separate U-tube manometer is placed across orifice meter. The volumetric flow rate of air supplied from the blower is controlled by controlling the position of the control valve. The experiments were conducted by varying the flow rate in terms of Reynolds numbers from 5000 to 10500 of the bulk air. The test tube is heated from the external surface during the experiments, and the information of temperature, volumetric flow rate, pressure drop of the bulk air and electrical output are noted once the system reaches to the steady state condition. The Nusselt number, Reynolds number, friction factor, heat transfer enhancement are calculated based on the average outer wall temperatures and the inlet and outlet temperatures of air.

III. DATA COLLECTION & STUDY

The data reduction of the obtained results is summarized in following ways:

A. Heat Transfer Calculations

\[
\text{Avg. Surface Temp., } T_s = \frac{(T_2 + T_3 + T_4 + T_5 + T_6 + T_7)}{6} \quad (1)
\]

\[
\text{Avg. Temp. of air, } T_a = \frac{(T_1 + T_8)}{2} \quad (2)
\]

\[
\text{Air head, } h_a = \frac{h_w \cdot \rho_w}{\rho_a} \quad (3)
\]

where,

\[
\rho_w = \text{Density of water} = 1000 \text{ kg/m}^3
\]

\[
\text{Air volume flow rate, } Q_a = C_d \cdot A_o \sqrt{(2 \cdot g \cdot h_{in})} \quad (4)
\]

where,

\[
A_o = \text{cross sectional area of orifice.}
\]

\[
\text{Mass flow rate, } m = Q_a \cdot \rho_w \quad (5)
\]

\[
\text{Velocity of air, } V = \frac{Q_a}{A} \quad (6)
\]

where,
The number is the Nusselt number of an enhanced tube with be. Re= Reynolds agreement with the results obtained from Webb[19].

This factor is introduced by corrugated twisted tape insert to that of a smooth tube, at a ratio of th
given below:

where,

\( h = \text{heat transfer coefficient.} \)

\( T_s = \text{surface temperature} \)

The Reynolds number for the fluid is defined by,

\[ Re = \frac{VD}{\nu} \quad (9) \]

where,

\( V= \text{velocity of the fluid.} \)

\( \nu = \text{Kinematic viscosity of the fluid.} \)

For internal flow conditions, if Reynolds number (Re) is greater than 4000 then the flow is said to be turbulent. After the flow is decided i.e. laminar or turbulent then the Nusselt number can be calculated. The theoretical Nusselt number is calculated below without considering friction which is theoretical Nusselt number and then calculated by considering friction which is experimental Nusselt number.

\[ Nu_{th} = 0.023 \times (Re)^{0.8} \times (Pr)^{0.4} \quad (10) \]

This equation is called Dittus-Boelter equation.

\[ f_s = (1.82 \log_{10} Re^{1.64})^2 \quad (11) \]

Above equation is used to find friction factor and is called as Petukhov equation for smooth surface.

where,

\( f_s = \text{Friction factor for smooth tube.} \)

\( \text{Re} = \text{Reynolds number.} \)

The actual pressure drop & friction factor is calculated with the help of tappings provided on both the ends of test pipe connected to U-tube manometer and the friction factor is calculated from the formula given below:

\[ f = \frac{P}{(L/D)^2} \left( \frac{\rho}{\nu} \right)^{1/2} \quad (12) \]

where,

\( P= \text{pressure difference at both ends of test pipe.} \)

\( L= \text{length of test pipe.} \)

\( D= \text{Inner diameter of pipe.} \)

The experimental Nusselt number are calculated as given below:

\[ Nu = \frac{hD}{k} \quad (13) \]

\( Nu= \text{Nusselt number} \)

\( h = \text{heat transfer coefficient} \)

\( k = \text{thermal conductivity of fluid} \)

\( D = \text{diameter of test section} \)

The overall enhancement efficiency is expressed as the ratio of the Nusselt number of an enhanced tube with corrugated twisted tape insert to that of a smooth tube, at a constant pumping power. This factor is introduced by Webb[19].

\[ \text{PEC}= \eta = \left( \frac{Nu_{with}}{Nu_{w/o}} \right) / \left( f_{with}/f_{w/o} \right)^{1/3} \quad (14) \]

A. Heat Transfer and Overall Augmentation Characteristics

The variation of Nusselt number with Reynolds number for various punched and v-cut inserts is shown in Figure 6. Uppermost Nusselt number was obtained for tape with twist ratio of11. The Nusselt number for punched and v-cut inserts ranges from 33% to 56% compared to plain tube. This is due to heavy-duty turbulence intensity caused by punches & cuts on inserts leading to quick mixing of the flow causing heat transfer augmentation. The variations of friction factor with Reynolds number for corrugated tape inserts are presented in Figure 7. It is observed that the friction factor gradually reduced with rise in Reynolds number. It is observed to be maximum, for insert having twist ratio of 11. It is clear from Figures 6, 7 and 8 that when a punched and v-cut twisted tape is inserted into a plain tube, there is a significant growth in Nusselt number due to secondary flow, with greater augmentation being recognized at lower Reynolds numbers and higher twist ratio. This enhancement is largely due to the centrifugal forces resulting from the spiral motion of the fluid and partly because of the tape acting as fin. It is detected that the rise in twist ratio results in increment of Nusselt number as well as growth in pressure drop. From Figure 6, the percentage rise in Nusselt numbers for corrugated twisted tapes compared to plain tube are about 30-89%, 33-100% and 58-146% respectively for

\[ q = m \times C_p \times (T_i - T_f) \]

\[ h = \frac{Q}{A(T_e-T_h)} \quad (7) \]

\[ A= \text{cross sectional area of pipe.} \]

Heat carried out, \( q = m \times C_p \times (T_i - T_f) \)

where,

\( h = \text{heat transfer coefficient.} \)

\( T_e = \text{surface temperature} \)

where,

\( V= \text{velocity of the fluid.} \)

\( \nu = \text{Kinematic viscosity of the fluid.} \)
tape with twist ratio 9, 10 & 11 respectively.

The overall enhancement ratio is useful to assess the quality of heat transfer enhancement obtained over plain tube with constant pumping power. It is found to be more than unity for all the punched twisted tape inserts under consideration. Distinctions of overall enhancement ratio $\eta$ against Reynolds number for different tapes are shown in figure 8. It is observed that overall enhancement tended to decrease gradually with the rise of Reynolds number for all twist ratios. The highest value of overall enhancement is 1.32 for punched and v-cut twisted tape insert having twist ratio 11. It is seen in figure 8 that, for tapes of twist ratios 9, 10 & 11 curves are of decreasing order for a given twist ratio in the range of Re from 5000 to 10000.

The study states an experimental investigation of the potential of punched and v-cut twisted tape inserts to improve the rate of heat transfer in a horizontal circular tube having inside diameter 26 mm using air as working fluid. The Reynolds number varied from 5000 to 10500. The effects of factors such as modified twist ratio, Reynolds number on the heat transfer and overall enhancement ratio are considered.

The following deductions can be drawn.

- The enhancement of heat transfer with punched and v-cut twisted tape inserts as compared to plain tube varied from 3 to 16% for twist ratio 9 and from 15 to 32% for twist ratio of 11. This enhancement is chiefly due to the centrifugal forces resulting from the spiral motion of the fluid. Growth in twist ratio causes increase in Nusselt numbers as well as friction factors.

- The maximum friction factor rise was about 52% for twist ratio 9 and 66% for twist ratio 11 for punched and v-cut twisted tape inserts compared to plain tube.

- The overall enhancement for the tubes with punched and v-cut twisted tape inserts is 1.32 for twist ratio 9 and 1.16 for twist ratio 11 for punched and v-cut twisted tape insert.

Thus the improved performance can be achieved using punched and v-cut twisted tapes as compared to plain tube. Thus, from the considerations of higher heat transfer and savings in pumping power punched and v-cut tape inserts are seen to be promising for improving turbulent flow heat transfer in a horizontal circular tube.

Future work can include:

- Change tape material from Aluminium and Copper to brass.
- Compound enhancement techniques maybe used i.e., the tape inserts can be united with coil wire inserts for better augmentation.

**NOMENCLATURE**

$H/D$ twist ratio

$f_t$ friction factor (theoretical) for plain tube

$w$ width of corrugated tape insert, (mm)

$f$ friction factor (experimental) for plain tube
\( f_c \) friction factor obtained using tape inserts
\( H \) pitch, (mm)

\( h \) experimental convective heat transfer coefficient, (W/m²K)
\( h_a \) equivalent height of air column, (m)
\( h_{air} \) manometer level difference, (m)
\( Q_{air} \) air discharge through test section, (m³/sec)
\( D \) Inner diameter of test section, (m)
\( k \) thermal conductivity, (W/mK)
\( A \) test section inner tube area, (\( \pi/4 \) D²) (m²)
\( Cp \) specific heat of air, (J/kg K)
\( A_o \) area of orifice, (m²)
\( L \) length of test section, (m)
\( m \) mass flow rate of air, (Kg/sec)
\( Nu \) Nusselt number (experimental) for plain tube
\( N_u_{th} \) Nusselt number for plain tube (theoretical)
\( Pr \) Prandtl number, \( p \) pitch, (m)
\( P \) pressure drop across the test section, (Pa)
\( Q \) total heat transferred to air (W)
\( Re \) Reynolds number, (\( \rho \) V D/\( \mu \))
\( T_1 - T_6 \) air temperature at inlet and outlet, (°K)
\( T_s \) avg Surface temperature of the working fluid, (°K)
\( T_b \) bulk temperature, (°K)
\( V \) air velocity through test section, (m/sec)

Greek symbols
\( \nu \) Kinematic viscosity of air, (m²/sec)
\( \mu \) dynamic viscosity, (kg/m s)
\( \eta \) Over all enhancement
\( \rho_w \) density of water, (kg/m³)
\( \rho_a \) density of air (kg/m³)

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