NUMERICAL STUDY OF A SHELL-AND-TUBE HEAT EXCHANGER FOR HEATING RICH MONOETHANOLAMINE USING HOT FLUE GASES. PART II: TUBE-SIDE HEAT TRANSFER ENHANCEMENT

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Abstract. The heat transfer and pressure drop characteristics of the tube-side of a shell-and-tube heat exchanger fitted with full-length twisted tape inserts were numerically investigated. There were analyzed four different twist ratios (y = 3, 4, 5 and 6) of twisted tape. The flue gas was used as the tube-side working fluid. Uniform wall temperature conditions were applied to the surface of the tube. The numerical results obtained for the case with twisted tape ratio y = 3 were compared with those from the studies in literature for validation of numerical method. It was found that the average heat transfer coefficient and pressure drop, respectively, for tubes with twisted tape inserts is 2.1-2.6 and 4.1-5.6 times higher than that of the plain tube. The effect of twist ratio on the heat transfer and pressure drop becomes more visible as the flue gas mass flow rate increases.

1. Introduction

Shell-and tube heat exchangers are widely used apparatuses in several industries such as chemical, petroleum referring and power generation. They are characterized by high heat transfer coefficient, compactness, relatively low cost, ease of construction and maintenance.

The heat transfer in the tube-side of a shell-and-tube heat exchanger can be enhanced by means of several types of swirl generators (e.g. helically coiled wires, helical tapes, twisted tapes) [1]. Twisted tape inserts are the most frequently used turbulators in tubular heat exchangers [2]. Twisted tapes have been also considered in fired-tube boilers [2,3,4]. Twisted tapes provide a simple passive technique for enhancing the heat transfer by producing swirl into the bulk flow and by disrupting the boundary layer at the tube surface. The effect of the twisted tape width on the heat transfer and pressure drop, under laminar flow conditions, was studied experimentally by Chakroun and Al-Fahred [5]. The presence of the twisted tape caused the heat transfer to increase by a factor of 1.5 to 3 compared to the plane tube, depending on the flow conditions and the twisted tape geometry. The effect of the tape width on the heat transfer was only visible in the high range of Reynolds numbers. The friction factor for the twisted tape was 3 to 7 times larger than that of the plain tube for $Re \approx 500$ and 2300, respectively. The decrease in the twisted tape width did not substantially change the friction factor. Base on their experimental results, the authors recommend to use loose-fit twisted tapes instead of tight fit ones for low Reynolds numbers.

In a more recent work [8], the effect of the clearance between the edge of tape and tube wall on the heat transfer and friction factor in the turbulent flow regime has been numerically investigated. They have shown that the heat transfer rates and friction factor decrease with increasing the clearance between the edge of tape and tube wall. There has been also found that the tight-fit twisted tape inserts give the best thermal performance characteristics as compared to loose-fit twisted tap inserts.

Manglik and Bergles [9] analyzed numerically the heat transfer and friction factor, in fully developed laminar flows, in circular-segment ducts with a straight tape insert at two uniform wall temperature conditions.

Rahimi et al. [10] used the RNG k-ε turbulence model to predict the heat transfer enhancement, friction factor and performance ratio in a tube with various twisted tape inserts (perforated, notched and jagged tapes). The phenomenon of flow (tangential velocity and turbulent intensity) through twisted tape is also described.

Thermal-hydraulic characteristics of air flow inside a tube fitted with different tube inserts (longitudinal strip and twisted tape) were numerically and experimentally investigated by Chiu and Jang [11]. They found that, in the range of Reynolds number from 7000 to 42000, the heat transfer coefficient and pressure drop of the tubes with twisted tapes ($y \approx 5.9$, 3.6 and 2.4) were 13-61% and 150-370%, respectively, higher than those of plane tube.

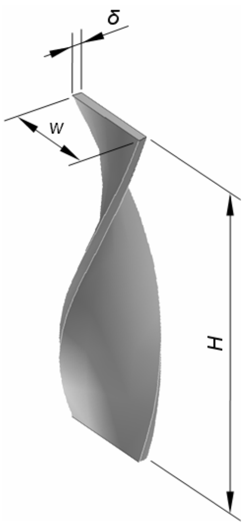
In the first part of this work [12], two shell-and-tube heat exchangers (with and without baffles), having the same main geometrical characteristics, have been numerically analyzed. Numerical results have clearly indicated that the proposed heat exchanger (8 segmental baffles, baffle cut of 25%, and larger the inlet and outlet nozzles on the shell side) is more efficient from both points of view, higher heat transfer and lower pressure drop.

The present numerical work has been conducted in order to study the heat transfer and pressure drop characteristics in the tube-side of a shell-and-tube heat exchanger with/without twisted tape inserts. The numerical simulations have been performed using the commercial CFD software package FLUENT6.2, and the pre-processor GAMBIT2.2 is used to create and generate the mesh.

2. Simulation procedure

2.1. Computational model

The computational domain of the models has been meshed with the unstructured Tet/Hybrid grids, which are generated by GAMBIT2.2. Computations have been carried out using FLUENT6.2, a commercially available software. Segregated solver and standard laminar model were employed and energy equation has been included. The SIMPLE algorithm with second order up winding for momentum and energy is used.



Twisted tape characteristics

Thickness, δ (mm)	1
Tape width, w (mm)	20
Twist pitch, H (mm)	60, 80, 100, 120
Twist ratio, y (-)	3, 4, 5, 6

Fig. 1 – Geometrical characteristics of twisted tape.

The geometrical characteristics of the twisted tape insert are shown in Figure 1. In the figure, there are also given the values of the parameters used in the present numerical investigation.

2.2. Governing equations and boundary conditions

The governing equations for continuity, momentum and energy in the computational domain can be expressed as follows.

Continuity:

(1) $$\frac{\partial}{\partial x_i}(\rho u_i) = 0$$

Momentum:

(2) $$\frac{\partial}{\partial x_i}(\rho u_i u_k) = \frac{\partial}{\partial x_i}\left(\mu \frac{\partial u_k}{\partial u_i}\right) - \frac{\partial p}{\partial x_k}$$

Energy:

(3) $$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_i}\left(\frac{k}{c_p}\frac{\partial T}{\partial x_i}\right)$$

Created models in GAMBIT were exported to FLUENT in which the following boundary conditions were set up: (i) the working fluid is the flue gas; (ii) the fluid properties are temperature dependent [13]; (iii) inlet conditions were defined as velocity inlets, which correspond to the mass flow rates of 1-5 kg/h (u_{in} =

constant, uniform) and constant temperature (T_{in} = 523 K); and, (iv) the temperature of 423 K was set for the tube wall.

3. Data reduction

The heat transfer coefficient h is calculated from:

$$h = Q / A\Delta T \quad (1)$$

where: Q is the heat transfer rate; A is the heat transfer aria; and, ΔT is the log-mean temperature difference, which are defined by Equations (2), (3) and (4), respectively.

$$Q = Mc_p\left(T_{in} - T_{out}\right) \quad (2)$$

$$A = \pi d_i l_t \quad (3)$$

$$\Delta T = \frac{T_{in} - T_{out}}{ln\frac{T_{in} - T_w}{T_{out} - T_w}} \quad (4)$$

where: M is the flue gas mass flow rate; c_p is the specific heat of the flue gas; d_i and l_t are, respectively, the inside diameter and length of the tube; T_{in} and T_{out} are the flue gas temperatures at the inlet and outlet, respectively; and, T_w is the temperature of the tube wall.

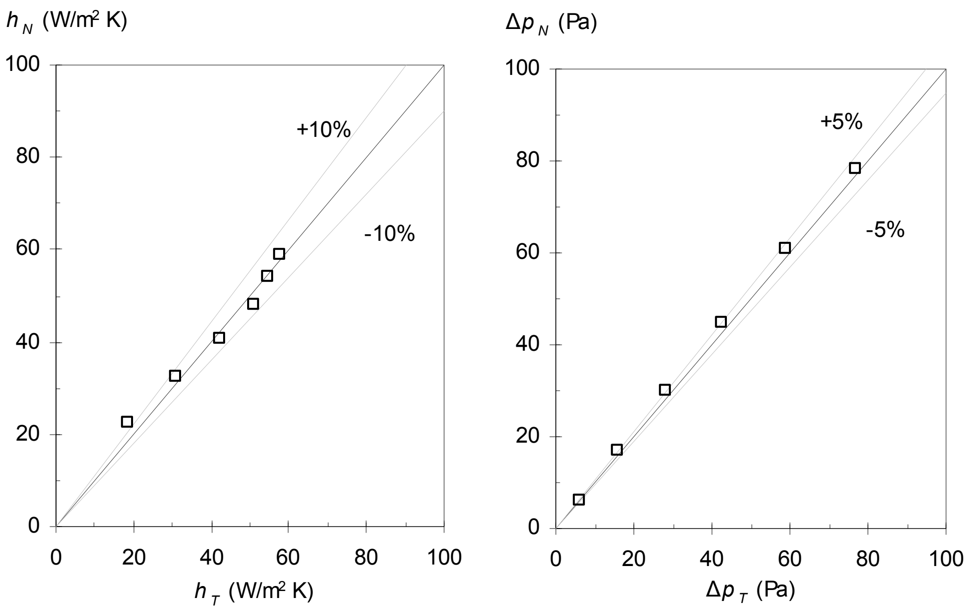
The pressure drop Δp is calculated as a difference between the average values of pressure at the inlet and outlet cross-sections:

$$\Delta p = p_{in} - p_{out} \quad (5)$$

4. Results

4.1. Numerical model validation

The present numerical results on heat transfer and pressure drop are validated with Manglik and Bergles correlations [6,7]. As can be seen, there is a good agreement between the numerical results and theoretical one (±10% for heat transfer and ±5% for pressure drop).

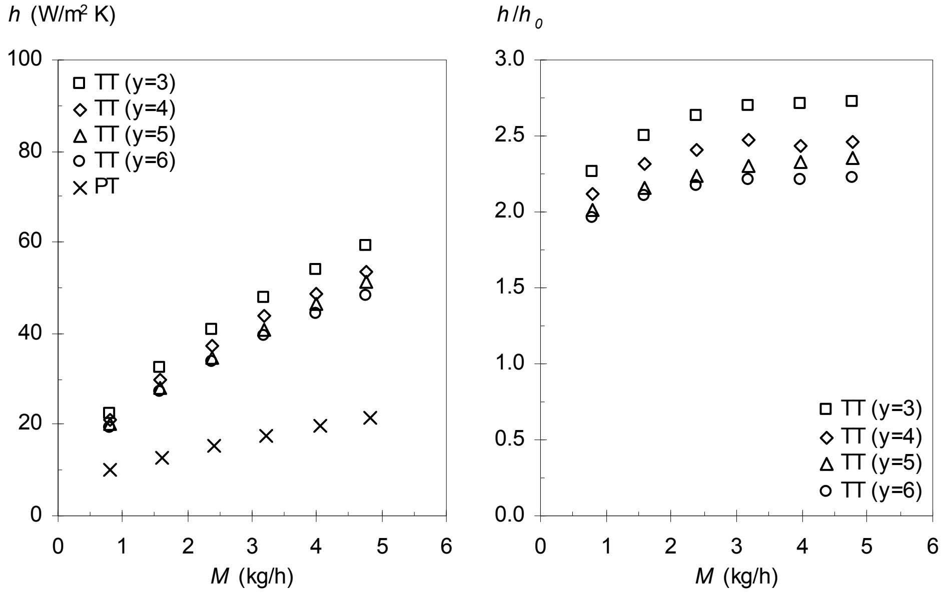


(a) **(b)**

Fig. 2 – Numerical versus predicted values of (a) heat transfer coefficient and (b) pressure drop for the case with a twisted tape insert of y = 3.

4.2. Heat transfer

The variation of heat transfer coefficient h with flue gas mass flow rate M for four different twist ratios (y = 3, 4, 5 and 6) of the twisted tape is shown in Figure 3a.



(a) **(b)**

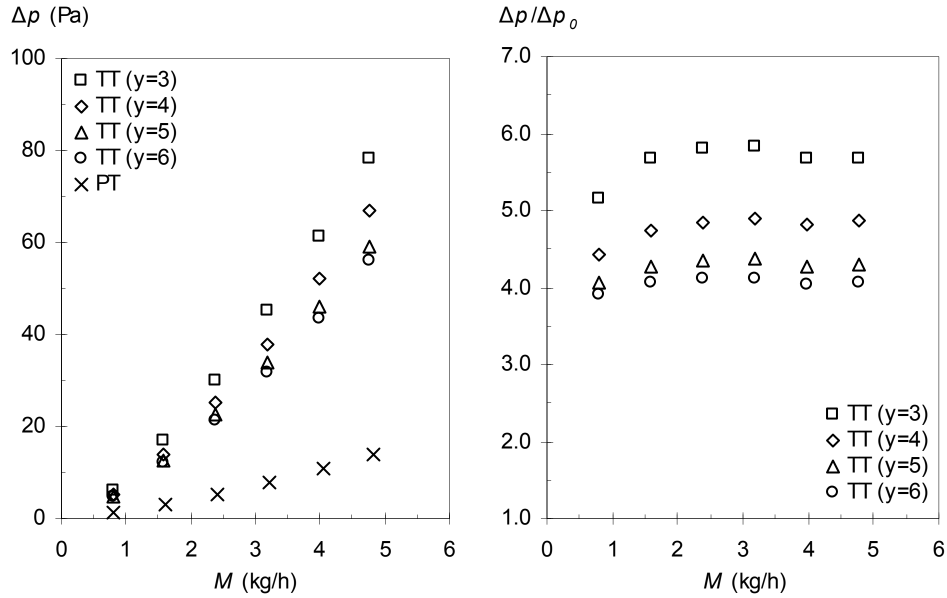
Fig. 3 – Variation of heat transfer with flue gas mass flow rate for tube with twisted tape at different twist ratios (a) h and (b) h/h_o.

It can be easily noted from this figure that the highest heat transfer is achieved for the twisted tape with the lowest twist ratio (y = 3). The heat transfer coefficient for the cases with y = 4, 5 and 6 is about 9.2%, 15.8% and 20.4%, respectively, lower than that of the tube with twisted tape ratio of y = 3. Compared to the plain tube, the heat transfer coefficient increases in the range 126-170%, 112-147%, 101-136% and 96-123% for twist ratios of y = 3, 4, 5, and 6, respectively, depending on the flow conditions. Figure 3b shows the variation of h/h_o with M. The h/h_o is the ratio of the mean heat transfer coefficient obtained from the tube with twisted tape insert to the heat transfer coefficient from plain tube. It is observed that the h/h_o tends to increase as the flue gas mass flow rate increases. For all twisted tape cases, the h/h_o is averagely 2.1-2.6 times those of the plain tube.

4.3. Pressure drop

The effect of using twisted tape inserts on the pressure drop is presented in Figure 4a and b. It is seen that the use of the smallest twist ratio yields the highest pressure drop. The pressure drop for y = 3 is found to be ~18%, 32% and 39% higher than those of the cases with larger twist ratios, y = 4, 5 and 6, respectively. It can be also observed that the pressure drop for y = 5 and 6 has approximately the same value while the pressure drop for y = 4 is seen to be the mean value between those for y = 3 and 6. The increase in pressure drop of the twisted tape tube, caused mainly by the swirl flow, is higher than that of the plain tube and is also much larger than that in heat transfer coefficient. Thus, the pressure drop increases in the range of 416-484%, 342-390%, 307-337% and 290-312% over the plain tube for tapes with y = 3, 4, 5 and 6, respectively.

Figure 4b presents the variation of pressure drop ratio $\Delta p/\Delta p_o$ with flue gas mass flow rate M.



(a) **(b)**

Fig. 4 – Variation of pressure drop with flue gas mass flow rate for tube with twisted tape at different twist ratios (a) Δp and (b) $\Delta p/\Delta p_o$.

Based on the numerical data there have been developed correlations for heat transfer and pressure drop (*Nu* and *f*).

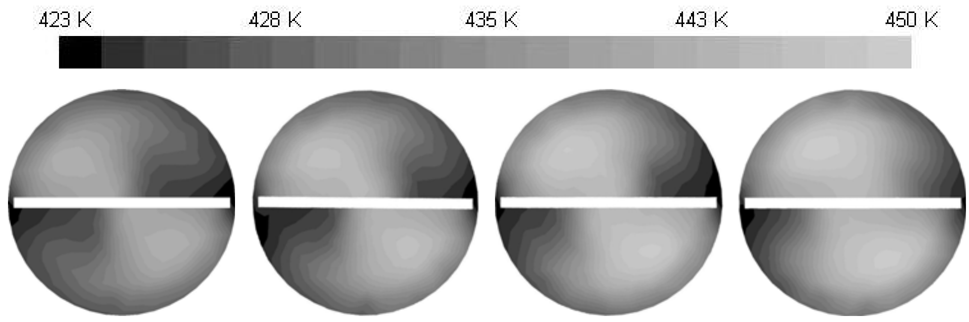
$$Nu = \frac{hd_i}{k} = 0.6029\, Re^{0.521}\; y^{-0.273} \tag{6}$$

$$f = \Delta p \frac{d_i}{l_t} \frac{2}{\rho u^2} = 52.6521\, Re^{-0.616}\; y^{-0.494} \tag{7}$$

These equations are valid within ±2% (h) and ±4% (Δp) error limit with numerical simulations for the following conditions: $500 < Re < 3300$, $Pr \approx 0.67$, $l_t/d_i \approx 32$ and $3 \leq y \leq 6$.

4.4. Temperature

Figure 5 shows the contour plots of temperature fields, at the outlet section of the twisted tape tube, for different twist ratios (y = 3, 4, 5 and 6). It is clearly seen that the twisted tape with the smallest twist ratio provides better temperature distribution than the twisted tapes with larger twist ratios.



(a) **(b)** **(c)** **(d)**

Fig. 5 – Temperature contour at the outlet exit from the tube with twisted tape inserts at different twist ratios ($M \approx 3.2$ kg/h): (a) $y = 3$; (b) $y = 4$; (c) $y = 5$ and (d) $y = 6$

5. Conclusions

The following conclusions can be drawn:

(i) Numerical simulation have clearly shown that the twisted tapes cause both higher heat transfer and much higher pressure drop in comparison with the plain tube. The presence of the twisted tape insert in the tube yields higher heat transfer rate h up to about 2.6, 2.4, 2.2 and 2.1 times of the plain tube, while the pressure drop Δp up to 5.6, 4.8, 4.3 and 4.1 times for using tapes with y = 3, 4, 5 and 6, respectively;
(ii) As the flue gas mass flow rate increases, the effect of twist ratio on the heat transfer becomes more visible. As expected, the twisted tape with twist ratio of y = 3 causes the highest pressure drop.

Nomenclature

A	heat transfer surface area (m^2)
c_p	specific heat (J/kg K)
d_i	tube inside diameter (m)
f	friction factor (dimensionless)
h	heat transfer coefficient (W/m^2 K)
H	twisted tape pitch (m)
k	thermal conductivity (W/m K)
l_t	tube length (m)
M	flue gas mass flow rate (kg/s)
Nu	Nusselt number (dimensionless)
p	pressure (Pa)
Δp	pressure drop (Pa)
Q	heat transfer rate (W)
Re	Reynolds number (dimensionless)
T	temperature (K)
ΔT	log-mean temperature difference (K)
u	velocity (m/s)
w	twisted tape width (m)
y	twisted tape ratio (dimensionless)

Greek symbols

ρ	density (kg/m^3)
δ	twisted tape thickness (m)
μ	dynamic viscosity (kg/s m)

Subscripts

in	inlet
out	outlet
w	tube wall
N	numerical
T	theoretical

Abbreviations

TT	twisted tape
PT	plain tube

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