Numerical and Experimental Investigation of Heat Transfer Enhancement in Double Pipe Heat Exchanger

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Abstract. This study included two parts, one of them experimental work and the other the Artificial Neural Networks (ANN) application. For the first part, a heat exchanger of double pipe with counter flow of water were used. This heat exchanger included a semicircular disc baffles attached on the outer surface of the inner tube fitted with twisted tape. The second part used to determine the transfer functions effect and training algorithms on the experimental work. The heat exchanger outer tube material was made from iron of (0.054 m and 0.05 m) outer and inner diameters respectively also inner tube material was copper of (0.022 m and 0.02 m) outer and inner diameters respectively. The dimensions of baffles were with (22 mm, 11 mm and 1 mm) outer radius, inner radius, and depth respectively while baffles space was 150 mm. the types of twisted tape for the inner tube were plain twisted tape (PTT), twisted tape with circular ring CRT and wire nail-twisted tape (WN-TT). The ratio of twist was \( y = 5 \). The results show that the errors in all cases for Nusselt number and factor of friction not exceed 5%.

1. Introduction
Heat exchanger is equipment which transfers the heat energy between two fluids that are at different temperature while keeping the mixing with each other. Heat exchangers are widely used in engineering applications such as refrigeration and air conditioning systems, thermal power plants, automobiles, chemical and textile processing industries, etc.. El-Shamy [1] used two tubes of 0.3 (Di/Do) ratio forming an annular canal and then examined the baffle spacing, and Reynolds number of flow effects on the canal thermal attitude.

Sarmad [2] investigated experimentally the turbulent flow and heat transfer behavior in a double pipe counter water flow heat exchanger with inserted semi circular disc baffles. Ahmet [3] studied the transfer functions and training algorithms effect and attitude (with artificially formed network usage) on practically found Nusselt and entropy generation numbers and irreversibility distribution ratios for nine different tubes which were determined as the target data. These nine different tubes have many baffle inserted to them with different width to diameter ratio, different parameter, different angles and different baffle spacing. The input data for the artificially formed network were taken from previous author’s studies. Reynold number, tube length to baffle spacing ratio, baffle orientation angle and pitch to diameter ratio were considered as input variables of ANNs. For reaching the target outcome the TRAINBR training function was the best way.

Tubes with twisted tape insert have been studied experimentally and numerically by Karami et al [4] by using special exchanger with butterfly inserts, the ability of ANN(artificial neural network ) to expect heat transfer was studied. Taken into consideration the effects of the inserts inclined angle and Reynolds number (Re) variation on the heat transfer. The experimental data is used as a template for the training one. The Levenberg-Marquardt back propagation algorithm is used in ANN training. Using MATLAB functions, the proposed ANN is developed. Trained data error of 0.109% and test
data error of 0.509% were obtained for having the best ANN structure. The research show that predicted values were very close to experimental values. A Rahimi et al. [5] using tube with twisted tape inserts, the heat transfer feature was numerically and experimentally investigated. Four types of inserts including the classic, perforated, notched and jagged twisted tape inserts were employed in the experiments. The results indicated that the heat transfer and performance of the jagged insert are higher than other ones. Shabanian et al. [6] numerically and experimentally investigate the tube inserts and Reynolds number effect on the heat transfer. The result showed that with different tube insert (butterfly, jagged and classic twisted tape inserts ) usage, the heat transfer from air cooler were increased with best result when butterfly insert with 90° angle was used. While Reynolds number was inversely related to the thermal performance due to the more significant role of inserts in increasing the turbulence intensity at lower velocities. Murugesan et al. [7] study how heat transfer, thermal performance factor and friction factor be affected when use V-cut twist tape insert, and then see the relation between these factors with three ratios [twist (Y), width (WR) and depth (DR)]. friction factor was inversely related to ( width and twist) ratios while it was positively related to depth ratio. P. Murugesan et al. [8] studied experimentally the friction factor and heat transfer of circular tube fitted with plain twisted tapes (PTT) with twist ratios $Y = 2.0, 4.4$ and $6.0$. The goal of this paper is to study the heat transfer behavior in a horizontal double pipe heat exchanger fitted with plain twisted tape (PTT), twisted tape with circular ring CRT and wire nail-twisted tape (WN-TT) insert. Semicircular disc-baffles which are attached on the opposite distances from the outer surface of the length of the inner tube used in the present study. using water as work fluid. The experimental results obtained for the tube.

2. Experimental Set-Up

Schematic diagram of the experimental set-up is shown in “figure 1” and the photograph of test rig in “figure 2” respectively. The experimental apparatus is formed by two concentric tubes in which water (cold in the inner, hot in the outer tube) flows through them with different direction. The inner tube is a Copper tube with dimension of (di = 20 mm, do=22 mm) while the outer tube is with iron pipe and (di = 50 mm, do=54 mm) dimensions. The length of the heat exchanger, pressure tape are 1500 mm, 1600 mm respectively. To minimize heat loss with the surroundings ,outer tube is insulated with glass wool (Insulation thickness = 10 mm). Hot water inlet and cold water outlet are equipped with four thermocouples (2 in each) type K to measure the temperature. Water flow are enhanced with two centrifugal pumps usage. To measure flow rate, Two calibrated flow meters having flow ranges of 0-18 LPM are put at the water inlet also to each flow meter inlet control and bypass valves are added. Two tanks of capacity 30L are used for water storage, with 3 KW electrical heater coupled to the hot water tank. Digital manometer is used to measure the pressure difference between inlet and outlet of test section.

![Figure 1. Experimental apparatus diagram](image-url)
In the present study at inner tube’s outer surface, semicircular disc-baffles with dimensions of 11 mm inner radius, and 1 mm depth are connect to them as shown in “figure 3”. The space between baffles is 150 mm.

“Figure 4” shows the types of twisted tapes (plain twisted tape PTT, twisted tape with circular ring CRT and wire nail-twisted tape (WN-TT)) which are made from copper with dimensions of [w (width) = 20 mm, t (thickness) = of 1 mm and l (length) = 1500 mm] and constant twist ratio, y = 5. The twist ratio (y) is defined by ratio of twist one length or pitch length (p = 100 mm) to diameter.

The circular rig is made from copper wire with inner diameter = 17 mm and outer diameter = 20 mm as shown in “figure 4c”. Nail (dw = 2 mm, Lw = 15 mm, dhw = 4 mm) twisted tape was obtained by punching small holes and carefully inserting nails on plain twisted tapes as shown in “figure 4d”.

Figure 2. Device image

Figure 3. The inner tube with semicircular disc-baffles

(a)

(d) (c) (b) (e)
3. Procedure
At the beginning of the experiment, the tank of water is filled with cold water and maintained its temperature up to 60 °C by using water heater. Then, this hot water is flowed through the flow meter by opening the flow control valve with help of hot water pump and through the annulus of heat exchanger. The cold water from the cold water tank is allowed to enter the inner tube through flow meter by flow control valve and cold water pump. Adjust the flow rate of hot water at 12 LPM and made it constant throughout the experiment. The flow rate of cold water is adjusted at 4 LPM. When the steady state is attained, the temperatures of the inlet and outlet of the hot and the cold waters are recorded and the pressure drop across the test tube is measured for plain tube without inserts. There after repeat same procedure for inner tube with semicircular disc baffles without and with (plain, circular ring, nail) twisted tapes having constant twist ratio (y = 5) for various flow rates of cold water like 4, 6, 8, 10, 12 LPM.

4. Data Reduction
The data reduction of the measured result is summarized in the following procedures:

Rate Heat transfer to the hot water, \( Q_h \), can be calculated from:

\[
Q_h = \dot{m}_h \cdot C_p \cdot (T_h - T_o) \tag{1}
\]

The rate heat transfer from the cold water, \( Q_c \), can be calculated from:

\[
Q_c = \dot{m}_c \cdot C_p \cdot (T_o - T_c) \tag{2}
\]

The average rate heat transfer, \( Q_{ave} \), of the hot and cold water can be determined from:

\[
Q_{ave} = \frac{Q_h + Q_c}{2} \tag{3}
\]

The overall heat transfer coefficient, \( U \), is determined from the following equation:

\[
U = \frac{Q_{ave}}{A \cdot \Delta LMTD} \tag{4}
\]

Where, \( A = \pi d_i L \)

The coefficient of heat transfer for the tube side is than determined using:

\[
\frac{1}{h_o} + \frac{1}{h_i} = \frac{1}{U} \tag{5}
\]

Where the coefficient of heat transfer for annulus side \( (h_o) \) is estimated using the correlation of Dittus-Boelter.

\[
Nu_o = 0.023 \cdot Re^{0.8} \cdot Pr^{0.3} \tag{6}
\]

Where, \( D_h = D_i - d_o \)

The Reynolds number is based on the different flow rate at the inlet of the concentric tubes heat exchanger.

\[
Re = \frac{\rho \cdot v \cdot d_i}{\mu} \tag{7}
\]

Thus the experimental value of Nusselt number is

\[
Nu_{exp} = \frac{h_i \cdot d_i}{k} \tag{8}
\]

Theoretical value of Nusselt number without semicircular disk baffles (plain tube) can be determined by using the determined by using the correlation of Dittus-Boelter.
\[ \text{Nu}_{\text{theo.}} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} \]  \hspace{1cm} (9)

Experimental friction factor can be written as:
\[ f_{\text{exp.}} = \frac{\Delta \rho \cdot d_{i}}{\rho \cdot L \cdot v^{2}} \] \hspace{1cm} (10)

\[ v = \frac{m}{\rho \cdot A_{i}} \] \hspace{1cm} (11)

Theoretical friction factor for smooth tube is calculated from the correlation of Blasius \[ (9) \]
\[ f_{\text{theo.}} = \frac{0.046}{\text{Re}^{0.2}} \] \hspace{1cm} (12)

All of thermo physical properties of the water are determined at the overall bulk temperature.

\[ \text{TEF} = \frac{\frac{\text{Nu}}{\text{Nu}_{0}}}{\frac{1}{\text{f}_{1}/\text{f}_{0}}} \] \hspace{1cm} (13)

This parameter is called the Thermal Performance factor which means the comparison is made based on constant pumping power

5. Model of Heat Exchangers Using Artificial Neural Network:
Artificial Neural Networks (ANNs) have been successfully used in many engineering applications to simulate nonlinear complex system without requiring any input and output knowledge such as dynamic control, system identification and performance prediction of thermal systems in heat transfer applications. ANN have been widely used for thermal analysis of heat exchangers during the last two decades. The applications of ANN for thermal analysis of heat exchangers are reviewed in detail [1].

5.1 Back Propagation (ANN) Method:
Three-layer BP network used as shown in “figure 5”. \( (V_{j,i}) \) represents the weights between the input layer vectors and hidden layer vectors, and \( (W_{k,j}) \) represents the weights between the hidden layer vectors and output layer vectors. The input layer has four neurons, including temperature of inlet hot water \( (T_{hi}) \), Renolds number \( (\text{Re}) \), circulating cold water inlet mass flow rate \( (m_{ci}) \), temperature of inlet cold water \( (T_{ci}) \). Output layer has four neurons, including circulating hot water temperature \( (T_{ho}) \), outer cold water temperature \( (T_{co}) \), Nuselt number \( (\text{Nu}) \) and friction factor \( (f) \). The hidden layer has two layers at \( (33,35) \) neurons.

The network part is implemented under the Matlab environment, and the activation function is chosen as the tangent sigmoid function in the hidden layer and the pure line function in the output layer.

5.2 Determination Of The Number Of Nodes In The Hidden Layer
The performance of an ANN is affected by the characteristic of the network, such as the number of hidden layer and the number of nodes in hidden layer. But up to now, there is not a definite method to choose the optimal number of hidden layer and the number of nodes in hidden layer. General speaking, the (ANN) model which has one hidden layer can meet with simulative requirements Hecht , [11]. So the model of one hidden layer is used in his study. Three formulas which are used to determine the node number in hidden layer are given by equations(14),(16) Xin, [12],

\[ \sum_{i=1}^{n} C_{n1}^{i} \geq K \] \hspace{1cm} (14)

Where \( K \) is the sample number, if \( (i > n1, C_{in}=0) \)
\[ n_1 = \sqrt{n + m + c} \]  \hspace{1cm} (15)

where \( c \) is a constant which belongs to Ding, [13].

\[ n_1 = \log_2 n \]  \hspace{1cm} (16)

When there is only one hidden layer in this (ANN) model, according to Xie, [14], the calculated formula of node number in hidden layer is defined by,

\[ n_1 = \sqrt{mn} \]  \hspace{1cm} (17)

Another empirical formula is put forward in Yao, [15]. It can be used to determine the node number in hidden layer, and the expression is given by

\[ n_1 = \sqrt{0.43mn + 0.12m^2 + 2.54n + 0.77m + 0.86} \]  \hspace{1cm} (18)

In the above five formulas (equations. (14) to (18)), \( n_1 \) is the node number in hidden layer, \( m \) is the node number in output layer, \( n \) is the node number in input layer.

5.3 Linear Transformation

The linear transformation for neural network suitable for the experimental data are shown in “figure 6”, where:

\[
\begin{align*}
A - A_{\text{min}} & = \frac{x - x_{\text{min}}}{x_{\text{max}} - x_{\text{min}}} \times A - A_{\text{min}} \\
x & = x_{\text{min}} + (x_{\text{max}} - x_{\text{min}}) \times A - A_{\text{min}} \\
A & = A_{\text{min}} + (A_{\text{max}} - A_{\text{min}}) \times \frac{x - x_{\text{min}}}{x_{\text{max}} - x_{\text{min}}} 
\end{align*}
\]  \hspace{1cm} (19, 20, 21)

Since \( x_{\text{min}} = -1 \) and \( x_{\text{max}} = 1 \) Then;

\[
x = 2^{[(A - A_{\text{min}})/(A_{\text{max}} - A_{\text{min}})]} - 1 \\
A = A_{\text{min}} + [(A_{\text{max}} - A_{\text{min}}) \times (x + 1)/2]
\]  \hspace{1cm} (22, 23)

For example from experimental data:

\[ T_{\text{b}}(\text{max}) = A_{\text{max}} = 60 \quad , \quad T_{\text{c}}(\text{min}) = A_{\text{min}} = 15 \quad , \quad A = 23.2 \]

![Artificial neural network (ANN) architecture](image.png)
6. Results and discussion
The experimental and numerical investigations of heat transfer, friction factor and thermal performance behaviors in a tube with semicircular disc baffles fitted with the PTT, CRT, and WN-TT are described. At first, the plain tube is tested and validated. “Figure 7” and “figure 8” show the validations of the present experimental results and with those obtained with well-known correlations under similar condition. As shown, the present Nusselt numbers agree well with those obtained from Dittus-Boelter equation(9) within ±10.7% and the present friction factors are within ±16.3% of those calculated from Blasius equation(12).
Figure 7. Experimental & theoretical Nusselt number versus Reynolds number for plain tube

Figure 8. Experimental & theoretical friction factor versus Reynolds number for plain tube

“Figure 9” shows the predicted and measured Nusselt numbers variation with Reynolds number for plain tube and tube with semicircular duct without and with PTT, CRT and WN-TT. The results reveal that the Nusselt number can be predicted with a good precision and the absolute value of errors in all cases is less than 5%. This figure concludes that the Nusselt number increases with increase in Reynolds number. Therefore the rate of heat transfer is more with higher Reynolds number.

The experimental results also reveal that the nail twisted tape (WN-TT) results in a higher Nusselt number compared with other twisted tapes and plain tube. This can be attributed to the fact that the nails act as turbulator and gives intensive mixing of fluid that promotes the turbulence near the tube wall surface that break the boundary layer at the surface which enhance the heat transfer. In addition, the simultaneous use of the WN-TT and semicircular disc baffles considered in the present experimental lead to further heat transfer enhancement.
Figure 9. Variation of Nusselt number with Reynolds number

“Figure 10” shows the Nusselt number ratio, Nu/Nuo, which defined as enhanced Nusselt number to Nusselt number ratio of plain tube plotted against the value of Reynolds number. The results indicate that the increase of Re number in tubes fitted with various tube inserts, the Nu/Nuo ratios decrease. Besides that, as can be seen in this figure, the ratio of Nusselt number for the WN-TT insert is higher than other twisted tapes in the studied range of Reynolds numbers. For example, in the range of 2972.06 to 9854.65 Re numbers, the ratios of Nu/Nuo for tube with semicircular disc baffles, PTT, CRT and WN-TT are between 1.83-1.64, 2.5–2.08, 3.05–2.37 and 3.577–2.6.

Figure 10. Variation of Nusselt number ratio with Reynolds number
“Figure 11” illustrate the predicted and measured results of observed friction factor of plain tube, tube with semicircular disc baffles and tube equipped with different twisted tape inserts. The results reveal that the friction factor can be predicted with a good precision and the absolute value of errors in all cases is less than 5%. It is cleared from this figure that the tube equipped with semicircular baffles fitted with tube inserts shows an increase in the factor of friction compared to the plain tube. The factor of friction tends to decrease with an increase in the number of Reynolds. The flow physics for the plain tube versus Reynolds number trend can be explained by Prandtl mixing length theory which argues that buffer layer and sub-layer thicknesses decrease as the Reynolds number number increases.

According to “figure 12” the WN-TT gives higher friction factor compared with other twisted tapes and plain tube. In addition, the simultaneous use of the WN-TT and semicircular disc baffles lead to increase of pressure drop and hence increase in pumping power. The reason for higher pressure drop was the dissipation of dynamic pressure of the fluid due to high viscosity loss near the tube wall.

“Figure 12” shows the friction factor ratio, $f/f_0$, plotted against the Reynolds number value. The results indicate that with the increase of Re number in tubes fitted with various tube inserts, the $f/f_0$ ratios decrease. Besides that, as can be seen in this figure, the friction factor ratio for the WN-TT insert is higher than other twisted tapes in the studied range of Reynolds numbers. In the range of 2972.06 to 9854.65 Re numbers, the ratios of $f/f_0$ for tube with semicircular disc baffles, PTT, CRT and WN-TT are between 2.6-3, 4.3-5, 5.5-6.2 and 6.4–6.9.
“Figure 13” show the Variation of Thermal enhancement factor Vs Reynolds number. It is noted from this figure at the same Re, factors of the thermal enhancement for WN-TT were to be greater than those for the PTT and CRT. The thermal enhancement factor for tests tends to decrease with increasing of Reynolds number. With the use of tube with semicircular disc baffles, PTT, CRT and WN-TT thermal enhancement factors are in a range between, 1.17 - 1.31, 1.29 - 1.5 and 1.35 - 1.7 and 1.49 - 1.88 respectively. The above data indicates that the use of WN-TT gave more efficient heat transfer enhancement than the application of PTT and CRT.

7. Conclusion
Heat transfer and friction factor were investigated numerically and experimentally, and factor of thermal enhancement of tube with semicircular disc baffles equipped with PTT, CRT and WN-TT in regimes of turbulent (2000 < Re < 10000) for constant twist ratios 5 are described in the present report. The conclusions were:
1- The results reveal that the Nusselt number and friction factor can be predicted with a good precision and the absolute value of errors in all cases is less than 5%.
2- The Nusselt number and factor of friction values for the tube with semicircular disc baffles are higher than that of plain tube

3- The Nusselt number and factor of friction values for the tube with WN-TT are higher than that of plain tube and also tube equipped with PTT and CRT.

4- Over the range of Reynolds number considered thermal enhancement factors in the tube with semicircular disc baffles, PTT, CRT and WN-TT are in a range between, 1.17 - 1.31, 1.29 -1.5 and 1.35- 1.7 and 1.49-1.88 respectively. The thermal enhancement factors for all the cases are more than unity indicates that the effect of heat transfer enhancement due to the enhancing tool is more dominant than the effect of rising friction factor and vice versa

5- The WN-TT give better enhancement of heat transfer than that PTT and CRT therefore WN-TT can be used in place of PTT and CRT to reduce the size of heat exchanger.

8. References

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