A Numerical Simulation Study to Improve Heat Transfer Rate in a Double Pipe Heat Exchanger using Different ways.

Zomorrood Ahmed Salman¹,⁵ Zena Khalefa Kadhim¹⁶, Kamil Abdulhussein Khalaf³,⁷, Hassanein Ali Kamil⁴,⁸.

¹,²,³,⁴ Mechanical engineering Department, College of Engineer, Wasit University, Iraq

⁵Zomorods527@uowasit.edu.iq, ⁶drzena@uowasit.edu.iq, ⁷kamil@uowasit.edu.iq, ⁸hassanien.a_alwaili@yahoo.com.

Abstract. In this work the thermo hydraulic performance of double pipe heat exchanger made of stainless-steel with inner and outer diameters are (11.43 and 16.83cm) respectively, 0.305cm thickness and 150cm length, studied numerically by using Solid Works 2016 package. It is used to the purpose of preheat heavy fuel oil flow in the outer pipe by hot air flow inside the inner one. Helical tape with different pitches (11, 14, and 17) cm the inner side of the inner tube and helical fin with different fin spaces (10, 20, and 30) cm over the inner tube are used as an enhancement heat transfer device. The study was conducted with specific identifiers, Reynold’s number (Re) values are (31668, 47361, 63008, 78589) for air side (inner tube), oil inlet temperature is (313) K and flow with rate of oil (0.1 and 0.06 kg/s). The results were first verified by using both the inner helical tape and outer helical fin separately and comparing the results of them with the plain tube, and then combining each of the helical tape and helical fin together and indicating the improvement in heat transfer rate, the result show that the maximum heat transfer is (4559.726 W) obtained by merged the helical tape with smaller pitch (11 cm) and helical fin with low fin pitch (10 cm) at oil flow rate (0.1 kg/s) as compared with (2052.385W) for plain tube, the maximum enhancement percentage in heat transfer rate and overall heat transfer coefficient were is (122.167 % and 142.941%) at the same conditions. The maximum enhancement in Nusselt number (Nu) and convection heat transfer coefficient were (224.572 % and 129.523 W/m²·K) respectively were achieved by using helical tape with pitch (11 cm). At higher Reynolds number, the higher-pressure drop for air side is (7437.8 pa) obtained when using minimum pitch for helical tape and it is (1086.26 pa) for oil side by using helical fin with pitch (10 cm).

1. Introduction
Heat exchangers, as the name suggests, it transfers heat between two materials. It is a device for transferring heat between two or more liquids that are treated. Heat exchangers have large-scale industrial and domestic applications and in the heat recovery process industry. It is an important component in thermal power system, refrigeration system and other cooling systems. In all of these systems, heat is transferred from one liquid to another. The counter flow heat exchanger is the most preferred since it granted high heat transfer rate for a given surface area in many thermal equipment.
designs based on, the steady state in the premium variable calculation. [1] Enhancement techniques are used to decrease the thermal resistance for heat exchanger to get higher heat transfer coefficient, therefore, this will reduce the size, these techniques are classified as active and passive techniques [2]. The active technique required external power, whereas the passive techniques required fluid additives, special surface geometries or swirl/vortex flow devices like different types of fin [3]. [4] Carried out study to improve heat transfer rate. For double pipe heat exchange depends on two method. The first is by adding helical fin on the out surface of the inner pipe with long of 1 m and pitch fin 17 mm, it wills results in increase the area of heat transferred and reduce the hydraulic diameter of the channels flow, this process was performed for stationary the inner tube. The second method to improve heat transfer rate is the inner tube rotate that will improve the turbulence and mixing of fluid molecules. The results of experimental work show the Nusselt number for helical fins is highest by 4 times from smooth tube for stationary condition. Nusselt number for 50rpm and 100rpm are (36% and 64%) respectively higher than that inner tube which is station. The Nusselt numbers at 50 rpm is less than 100 rpm. A combined multiple shell-pass shell-and-tube heat exchanger with continuous helical baffles in outer shell pass has been invented to get the best performance of heat transfer, its compared with the conventional shell-and-tube heat exchanger with segmental fins by means of (CFD) studied by [5]. The theoretical results revealed, at the same mass flow rate and overall heat transfer rate, the average overall pressure drop of the first type of heat exchanger is lower than the conventional heat exchanger by 13%. On the other hand for the same overall pressure drop, the overall heat transfer and mass flow rate is nearly 5.6% and 6.6% higher than the conventional heat exchanger. The thermo hydraulic performance of a applicable design of double pipe heat exchanger (air and water) as working fluid with helical fins .The spacing fin in the range of 0.05–0.2 m on the gas side examined theoretically by [6]. The results reveals that the consumption of heat transfer rate at same pump power is higher with helical fins than that with longitudinal. The heat transfer surface areas for helical fins with pitches of (0.2, 0.1, 0.067 and 0.05 ) m are 3%, 7%, 15%, and 24% respectively, the heat transfer coefficient also increases when fins spacing decrease, on average, about 14%, 42%, 63%, and 83% for same pitches arrangement above ,which is higher than the longitudinal fin configuration.

The performance of a double pipe heat exchanger, inner and outer diameters were (114.3 and 168.3 mm) made of stainless-steel with 3.05 mm thickness and 1500 mm with full length tight-fit twisted tape studied numerically by [7]. The results showed that At Reynolds number variation between 25148 and 64883 for air side (the inner tube), the flow rates of heavy fuel oil is 0.1 kg/s and four different twisted tape ratio 1.5, 2.0, 2.5 and 3.0, the maximum enhancement in in heat transfer rate, Nusselt number and convection heat transfer coefficient were 26.6%, 106.1% and 91.7% respectively and the results approved experimentally [8]. The effect of helical fin over the inner tube on the performance of the double tube heat exchanger and compare it with the plain tube ,at constant hot fluid inlet temperature 80°C and the flow rate of hot fluid change from 0.01 kg/s and 0.05 kg/s, the experimental analysis was done by [9]. The results evaluate when impact the helical fin over the inner pipe the heat transfer rate and heat exchanger efficiency is found to be 38.46% and 35% respectively higher than smooth inner pipe being used for each mass flow rates, similarly, over-all heat transfer coefficient is also increased when helical fin are equipped inside inner pipe.

2. Numerical modeling
To foretell the heat transfer performance of double pipe heat exchanger SOLIDWORKS PREMIUM 2016 software programs package was used. Figure 1 show the helical tape inside the inner tube with central rod, figure 1b illustrate the helical fin over the inner pipe (oil side). Heavy fuel oil properties change with the changing of temperature. During the simulation after considering the degreasing of density (by increasing the oil temperature) with the effect of gravity and considering the heat exchanger was putting in vertical way. The results shows that the oil flow streams passing from the inlet to the outlet without proper mixing .its show in the upper part of the shell the hot oil is accumulated at the upper part of heat exchanger because its density is lower than the cold oil. This case will cause many disadvantages like oil cracking [Pub31_Heavy Fuel Oil], reach the auto ignition
temperature and reduce the thermal efficiency by forming deposits on inner pipe wall. In order to avoid these disadvantages by providing a good mixing and increase the turbulence energy for the oil in the shell side, more than one inlet had been used as shown in figure 1c by providing the heat exchanger with (360°) inlet manifold. The best case was by using four inlet points (90° between them) because it’s give better mixing and a relatively higher heat transfer rate.

![Figure 1. 3-D model of heat exchanger parts](image)

### 3. Governing Equations

In order to solve the system of equations for the above design, Navier-Stokes equations were used, which are equations for the conservation of mass, momentum and energy of fluid flows. The program used turbulent kinetic energy and its dissipation rate transport equations called k-mode. The program uses a single system of equations to show the laminar and turbulent flow, and is also suitable for flow with high turbulence [10][11].

#### Continuity equation

\[
\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = 0
\]  

(1)

#### Momentum equation

\[
\frac{\partial \rho u_i}{\partial t} + \frac{\partial (\rho u_j u_i)}{\partial x_j} + \frac{\partial p}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \tau_{ij} + \tau^R_{ij} \right) + S_i \quad i = 1, 2, 3
\]  

(2)

#### Energy equation:

\[
\frac{\partial \rho H}{\partial t} + \frac{\partial (\rho u_j u_i)}{\partial x_j} = \frac{\partial}{\partial x_i} \left( u_j \left( \tau_{ij} + \tau^R_{ij} \right) + q_i \right) + \frac{\partial p}{\partial x_i} - \tau^R_{ij} \frac{\partial u_i}{\partial x_j} + \rho \varepsilon + S_i u_i + Q_H
\]  

(3)

Where; \( H = h + \frac{u^2}{2} \)

mass distributed external force is;

\[
S_i = S_i^{\text{porous}} + S_i^{\text{gravity}} + S_i^{\text{rotation}} \quad \text{And,} \quad S_i^{\text{gravity}} = -\rho g_i
\]

For Newtonian fluids the viscous shear stress tensor is;

\[
\tau_{ij} = \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k} \right)
\]  

(4)

Following Boussinesq assumption the Reynolds-stress tensor is:
Turbulent eddy viscosity:
\[ \tau_{ij}^R = \mu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k} \right) - \frac{2}{3} \rho k S_{ij} \]  \hspace{1cm} (5)

Turbulent viscosity factor:
\[ \mu_t = f_\mu \frac{C_{\mu} \rho k^2}{\varepsilon} \]  \hspace{1cm} (6)

Turbulent viscosity factor:
\[ f_\mu = \left( 1 - \exp\left( -0.0165 R_y \right) \right)^2 \left( 1 + \frac{20.5}{R_T} \right) \]  \hspace{1cm} (7)

The following constants are defined empirically; in flow simulation these typical values are used
\[ C_\mu = 0.09, C_{e1} = 1.44, C_{e2} = 1.92, \sigma_e = 1.3, \sigma_k = 1 \]

4. Preformation of Boundary Conditions
To rating the performance of the present heat exchanger, many boundary conditions need to be clear:
1. Inlet boundary conditions: air velocity inlet was (10, 15, 20 and 25 m/s), oil mass flow rate is (0.1 and 0.06 kg/s), and the temperature inlet of the heavy fuel oil at outer tube is (313 K), while the inlet air temperature at inner tube is (473 K).
2. Boundary conditions of outlet pressure: The outlet field is defined as the pressure outlet for both sides [12].
3. Boundary conditions of wall: in the inner tube wall there is no slip boundary condition.

5. Mesh independent study
In order to check the effect of mesh size on the results, six mesh settings with total number of cells (150,826), (303,612), (450,198), (986,642), (1,568,864) and (2,473,988) were initially considered, the results of heat transfer rate were compared with the fine mesh settings and the intermediate mesh settings with (450,198) cells for plain tube was chosen for all simulation cases to reduce the time required for finishing the simulation with same results with less than (0.5% deviation) by comparing it with the finest mesh. Noticing that at the same mesh settings the mesh cells were, (450,198), (491,482), (512,074), (580,542) for plain tube, helical fin, helical tape and compound cases respectively.
The way that used to grid cells was finite volume method which is parallel, rectangular lines.

5. Validation
The results of the Nusselt number that obtained from numerical simulation using the Sold Work program 2016 were compared with the Nusselt number obtained from existing correlation predictions as in figure 2. The difference between the correlation and the simulation results of Nusselt number for air side are less than 10%. The three correlations were (Dittus Boelter's, Gnielinski's and Petukhov's) [14], [15] in terms of Nusselt number (Nu), Reynolds number (Re) and Prandtl number (Pr):

**Dittus-Boelte correlation**
\[ Nu = 0.023 \Re^{0.8} \Pr^{0.3} \]  \hspace{1cm} (8)

**Gnielinski's correlation**
\[ Nu = \left( \frac{1}{\Re \left( \Re - 1000 \right) \Pr} \right) \left( 1 + 12.7 \left( \Pr^{0.667} - 1 \right) \frac{\Re}{\Pr^{0.5}} \right) \]  \hspace{1cm} (9)

**Petukhov's correlation**
\[ Nu = \left( \frac{1}{\Re} \Pr \right) \left( 1 + 12.7 \left( \Pr^{0.667} - 1 \right) \frac{\Re}{\Pr^{0.5}} \right) \]  \hspace{1cm} (10)

Where \( f = \left( 0.79 \ln(\Re) - 1.64 \right)^{-2} \); \( C = 1.07 + \frac{900}{\Re} - \frac{0.63}{\left( 1 + 10 \Pr \right)} \)
6. Effect of helical tape, helical fin, and the effect of emerging on a temperature distribution

It is important to analyze the temperature profile inside the inner tube to know how the heat distribution and mode of transmission will be in the case of plain tube, adding helical tape inside the inner tube, and equipped helical fin over the inner pipe and the compound of both helical tape and helical fin. The flow through the tube can be classified into two regions, the core region and the fluid area near the wall. Using helical tape will provide good mixing for liquid temperature between the core area and the region near the wall. This mixing will thinning the boundary layer near the wall and enhance the heat transferred from the liquid to the wall of the tube. Figure 3 show that at (200°C) air inlet temperature, (25 m/s) air inlet velocity, (0.1 Kg/s) oil flow rate and (11 cm) helical tape pitch provide high mixing and maximum heat transfer rate as compared with other helical tape pitches at give low mixing and plain tube where there is no mixing between core and the area near the wall in case of plain pipe.

![Figure 2. validation of numerical results with correlation predictions.](image)

![Figure 3. Contours of Temperature for heat exchanger (a) smooth pipe (b) helical tape with pitches (11, 14, 17 cm)](image)

Helical fin increase mixing between the cooled oil particles and increase the heat transfer area. By decreasing helical fin pitches the mixing and heat transfer area will increase which lead to increase heat transfer rate as shown in figure 4.
Figure 5 shows the gradient in the temperature distribution in case of using both helical tapes inserted and helical fin, the higher mixing in both air and oil side, and maximum heat transfer rate is obtained by using helical tape with (p) (11 cm) and helical fin with (10 cm) pitch.

7. Velocity vectors and velocity contours
The using of helical tape inside the inner tube of the double pipe heat exchanger works on increasing swirling flow and the tangential velocity near the wall of the inner tube more than plain tube. Figures 6 and 7 show the velocity vectors for cross sections from the heat exchanger for four cases plain tube, helical tape inserted, the dots in plane tube reveals that, the tangential velocity in the inner pipe is almost non-existent. In case of helical tape inserted it generates high tangential velocity and it causes additional movement to flow near the inner wall more than and along all the flow axis, the maximum increasing in tangential velocity is by using helical tape at smaller (p =11 cm).

The effect of equipped helical fin over the inner tube had no much influence in increase velocity vector as compared helical tape as in figure 8.
Figure 9 illustrate the tangential velocity for air side in case of plain tube and three helical tape pitches. Almost there is no tangential velocity in case of plain tube and it increase by using helical tape especially at lower helical tape pitch (11cm).

**Figure 6.** Velocity vectors for plain tube

**Figure 7.** Velocity vectors for helical tape

**Figure 8.** Velocity vectors for (a) plain tube, (b) helical tape (c) helical fin, (d) compound.
8. Results and discussion

8.1. Helical tape inside the inner tube:

By using helical tape with central rod, heat transfer rate enhances by creating swirls flow and increase flow path length that lead to giving an additional time to transfer to the heat along the vertical heat exchanger. It is also increase turbulence and reduce thermal boundary layer thickness near the wall by increasing tangential velocity, the thermal resistance, and enhance heat transfer rate. The following diagrams are a comparison of the heat transfer rates between a heat exchanger with plain tube and with helicoidal.

8.2. The enhancement of heat transfer rate, overall heat transfer coefficient and Nusselt number for helical tape inserted inside the inner tube.

The following diagrams show a comparison of the heat transfer rates (Q) between a heat exchanger with plain tube and with helical tape inserted inside the inner tube. Figures (10.a) and (10.b) show the (Q) versus with four Re (31668, 47361, 63008 and 78589) for three different helical tape pitches (11, 14 and 17 cm) compared with plane tube. From Figure 5.a,b at Re (78589) and higher oil flow rate (0.1 kg/s), air inlet temperature (200°C), the maximum (Q) is (2968.852 W) at smaller (p) (11 cm) as compared with (2052.385 W) for plain tube at the same boundary.

Under the same condition above but oil flow rate (0.06 kg/s) the maximum (Q) is (2749.476 W) for (p) (11cm) and (1898.279 W) for smooth tube as in figure 5a, b.
The amount percentage of heat transferred enhancement represented by table 1 where the maximum percentage enhancement is (44.84) for helical tape with minimum helical pitch (11 cm).

Figure 11 illustrate the effect of Reynolds number on the overall heat transfer coefficient for plain tube and three helical tape pitches. From this figure the overall heat transfer coefficient increase with Reynolds number increases generally for plain tube and helical tape, however, it decreases with increasing the distance of helical tape pitches. In addition to that the overall heat transfer coefficient for helical tape with three pitches is higher than that plain tube, the maximum over all heat transfer rate (40.526) concluded at minimum (p =11 cm) and higher values of (Re) (78589), oil flow rate (0.1 kg/s) and inlet hot air (200 °C).

### Table 1. the percentage of heat transferred enhancement for three helical tape pitches

| T air inlet | Air inlet velocity (m/s) | P 11  | P 14  | P 17  |
|-------------|--------------------------|-------|-------|-------|
|             | Oil flow rate (0.06 Kg/s)|       |       |       |
| 200°C       | 25           | 44.84 | 27.70 | 24.11 |
|             | 20           | 47.27 | 31.19 | 26.10 |
|             | 15           | 42.74 | 31.79 | 27.18 |
|             | 10           | 42.58 | 34.31 | 29.15 |
|             | Oil flow rate (0.1 Kg/s)|       |       |       |
| 200°C       | 25           | 44.65 | 38.40 | 34.16 |
|             | 20           | 43.85 | 36.85 | 33.35 |
|             | 15           | 44.55 | 37.73 | 32.61 |
|             | 10           | 44.38 | 37.88 | 34.12 |
8.3. The effect of Reynolds number on Nusselt number (Nu).

Figure 12.a and b show the (Nu) variation at four values of (Re) (31668, 47361, 63008 and 78589) for three different (p) (11, 14 and 17 cm). At (Re) (78589), (p = 11 cm) and (200°C) air inlet temperature. The maximum (Nu) was (453.995 and 447.047) at oil flow rate (0.1 and 0.06 kg/s) respectively, as compared with (Nu) (202.16 and 200.57) for plain tube at the same variables mentioned above. At low (Re) the flow tend to be dominated by laminar flow, with increasing (Re) of inlet hot air, the flow velocity, flow turbulence, and turbulence kinetic energy increase, also convective heat transfer coefficient increase since a velocity is a function of it. As well as convection heat transfer coefficient increased, the heat transfer rate, and (Nu) increase.

At inlet air temperature (200°C) the maximum enhancement percentage for (Nu) by using helical tape was (224.5 %) at the lower (p) (11 cm) and (Re) (78589) as in table 2.
Table 2. enhancement percentage of Nusselt number for three helical tape pitch

| T air inlet | Nu enhancement percentage % | Air inlet velocity (m/s) | P 11 | P 14 | P 17 |
|-------------|----------------------------|--------------------------|------|------|------|
| Oil flow rate (0.06 Kg/s) |
| 200°C       | 25                          | 122.8                    | 90.4 | 72.9 |
|             | 20                          | 116.7                    | 87.5 | 69.6 |
|             | 15                          | 112.2                    | 80.3 | 68.0 |
|             | 10                          | 107.3                    | 84.5 | 70.3 |
| Oil flow rate (0.1 Kg/s) |
| 200°C       | 25                          | 224.5                    | 95.8 | 78.8 |
|             | 20                          | 216.9                    | 90.9 | 75.9 |
|             | 15                          | 214.5                    | 89.7 | 74.2 |
|             | 10                          | 210.9                    | 88.1 | 72.7 |

Figures 13.a and 13.b illustrates, at inlet air temperature (200°C), (Re) (78589) and mass flow rate (0.1 and 0.06 kg/s) the convection heat transfer is (135.947, and 134.28 W/m² K) respectively.

Fig. (13.a) Reynolds number vs. convection heat transfer coefficient for plain tube and three helical tape pitches at (0.1 kg/s and 200 °C).

Fig. (13.b) Reynolds number vs. convection heat transfer coefficient for plain tube and three helical tape pitches at (0.06 kg/s and 200 °C).

The maximum enhancement in convection heat transfer coefficient is (129.52 %) at minimum pitch (11 cm) as illustrated in table 3.
Table 3. Enhancement percentage of convection heat transfer coefficient for three helical tape pitches.

| T air inlet | Air inlet velocity (m/s) | P 11 | P 14 | P 17 |
|-------------|--------------------------|------|------|------|
|             | Oil flow rate (0.06 Kg/s) |      |      |      |
| 200°C       | 25                       | 127.78 | 94.88 | 76.75 |
|             | 20                       | 121.50 | 91.74 | 73.93 |
|             | 15                       | 118.32 | 88.56 | 72.91 |
|             | 10                       | 114.13 | 90.60 | 76.54 |
|             | Oil flow rate (0.1 Kg/s)  |      |      |      |
| 200°C       | 25                       | 129.52 | 100.16 | 82.77 |
|             | 20                       | 122.51 | 95.80 | 80.39 |
|             | 15                       | 120.73 | 95.23 | 79.27 |
|             | 10                       | 118.95 | 94.64 | 80.031 |

8.4 The effect of helical tape on pressure drop for air side

The pressure drop for inner tube (hot air) increase when using a helical tape as an enhancement device. The decreasing in helical tape pitches and increasing Reynolds number, pressure drop is become higher, the maximum pressure drop is (7437.8 pa) at pitch (11 cm), Reynolds number (78589), oil mass flow rate (0.1 kg/s) and air inlet temperature (200°C) which is a big value as compared with (42.62 pa) pressure drop for plain tube as shown in figure 13c.

8.5 Helical fin over inner tube

The helical fins over the inner tube results into the increase in heat transfer area and enhance heat transfer rate due to impact by the multiple vortices induced in the flow region. The helical fins over the inner tube increases the turbulence and reduce the hydraulic diameter. The hydraulic performance of double pipe heat exchanger examined numerically by equipped helical fin with three different pitches (10, 20 and 30 cm) on the outer surface of the inner tube (heavy oil side) and compared the result according to plain tube. Increase air inlet temperature, also increasing mass flow rate of oil increase oil velocity which lead to increase turbulent and make a huge enhancement in heat in heat transfer

The surface area of heat transfer increases with decreasing helical fin spacing and that lead to increase heat transfer rate. At high oil flow rate (0.1 kg/s), helical fin pitch (10 cm), inlet air temperature (200°C) and Re (78589), the maximum (Q) is (2928.78 W) as compared with (2052.385 W) for plain tube as figure 14 showed;
The effect of decreasing oil mass flow rate is shown in figure 15 at (0.06 kg/s) oil mass flow rate and same boundary condition above, the maximum (Q) is (2690.715 W) as compared with maximum Q (1898.279 W) for plain tube.

Nusselt number and convention heat transfer coefficient have no change when using helical fin on the oil side, because Nusselt number is a function to h, and the last is a function for the air inlet velocity which not effected by editing the helical fin.

The amount of heat transferred enhancement percentage is represented by table 4. The maximum percentage of heat transfer rate enhancement is (42.70%) for smaller helical fin pitch (10 cm), high air inlet velocity (25 m/s) and oil flow rate (0.1 Kg/s).

8.6. Pressure drop for oil side by using helical fin
An important factor in the design of heat exchangers is the pressure drop. To maintain the required flow, the designer must pay attention to the designer of the pressure drop. Figure 16 shows the difference in the pressure drop on the oil side with the return values for the oil flow rate (0.1 and 0.06 kg / s), figure 16 shows the variation in the pressure drop on the oil side with the two values of oil flow rate (0.1 and 0.06 kg /s). The maximum pressure drop is on helical fin with fin (10 cm) which is (1086.26 pa) at oil flow rate (0.1Kg/s) as compared with (892.81 pa) for plain tube. The main contribution of high pressure drop comes from the maximum velocities in the entrance region owing to the sudden change in the flow pattern.

Table (4) the amount of heat transferred enhancement percentage for three helical fin pitches.

| T air inlet | Air inlet velocity (m/s) | Fin 10 | Fin 20 | Fin 30 |
|-------------|--------------------------|--------|--------|--------|
| 200°C       | 25                       | 41.74  | 37.24  | 34.82  |
8.7. Merging both helical tape inside the inner tube and helical fin over it:
After a theoretical study of each one of the helical tape inserted in the inner tube and helical fins on the outer surface of the inner tube separately, and showing their importance in improving the performance of the double pipe heat exchanger, the idea of combining them was reached to get the best hydraulic performance of the heat exchanger. In general both of them increase the turbulence of the flow in air and oil side. The simulation prove that the helical fin with pitch (10 cm) obtain high heat transfer area and helical tape with (11 cm) to get high tangential velocity near the inner tube wall which thinning the boundary layer and then increase the heat transfer rate. Fig (17) shows the heat transfer variant with four Reynolds number (31668, 47361, 63008 and 78589). At (0.1 kg/s) oil flow rate, (200°C) air inlet temperature and high Re (78589), the maximum heat transfer rate is (4559.726, 2968.852, 2928.78 and 2052.385) W, for compound, helical tape with pitch (11 cm), helical fin with pitch (10 cm) and for plain tube respectively.

The heat transfer rate for compounding of both helical tape and fin increase for almost of oil flow rate values, but it gives lower increase in low flow rate (0.06 kg/s) under same condition above as show in figure 18 the maximum (Q) is (4029.454) as compared with (2749.476, 2690.715 and 1898.279 W) for helical tape with (p= 11 cm), helical fin with pitch (10 cm) and for plain inner pipe respectively.

| Oil flow rate (0.1 Kg/s) | 200°C |
|-------------------------|-------|
| 20                      | 40.09 | 35.60 | 33.22 |
| 15                      | 33.96 | 30.52 | 28.03 |
| 10                      | 29.21 | 27.09 | 24.89 |

| Oil flow rate (0.1 Kg/s) | 200°C |
|-------------------------|-------|
| 25                      | 42.70 | 40.36 | 37.83 |
| 20                      | 33.59 | 31.73 | 29.69 |
| 15                      | 28.14 | 26.57 | 24.94 |
| 10                      | 25.14 | 23.89 | 22.58 |

Figure 16. The effect of oil mass flow rate on pressure drop for oil side in case of plain tube and helical fin.
Nusselt number and conventional heat transfer coefficient have no change at compound case, because Nusselt number is a function to (h), and the last is a function for the air inlet velocity which not effected by emerge the helical fin with helical tape.

9. Conclusions
The following conclusions were drawn from the obtained results:

1. The helical tape inside the inner tube lead to increase the air flow turbulent which leads to increase the tangential velocity near the wall and thinning the boundary layer which increase the heat transfer rate. Decreasing the tape pitches increase the helical path length and consequently the turbulent, heat transfer rate, Nusselt number, convection heat transfer coefficient and pressure drop increase.

2. The helical fins over the inner tube results into increase in the heat transfer area and reduction in the hydraulic diameter of the flow channel. When fin pitch decreases the heat transfer area, oil mixing and oil pressure drop increase.

3. Increasing oil mass flow rate increase the heat transfer rate because of the mass flow rate is function for velocity. When the velocity increase and the flow become more turbulent and that make the oil keep pushing on near the inner hot wall of the heat exchanger.

4. For helical tape with pitch (11cm), (0.1kg) oil flow rate, (200°C) air inlet temperature and Reynolds number (78589), the maximum values of heat transfer rate, overall heat transfer coefficient Nusselt number and convection heat transfer coefficient were (2968.852 W, 40.526 W/m². K, 453.995 and 135.947 W/m². K) respectively.

5. For helical tape with pitch (11cm), (0.06kg) oil flow rate, (200°C) air inlet temperature and Reynolds number (78589), the maximum enhancement in heat transfer rate was (2749.476 W). At oil flow rate (0.1kg/s) and same conditions above the maximum enhancement in Nusselt number and convection heat transfer coefficient were (224.572 % and 129.523 % W/m². K).

6. For helical fin with pitch =10cm, (0.1kg) oil flow rate and (200°C) air inlet temperature the maximum values of heat transfer rate and overall heat transfer coefficient were (2928.78 W and 38.85 W/m². K) at Reynolds number (78589), the maximum enhancement percentage in heat transfer rate (42.70%) at the same conditions above.

7. By merging both helical tape with lower pitch (11 cm) inside the inner tube and helical fin with pitch (10 cm) over the inner pipe with high Reynolds number (78589) the maximum heat transfer rate is (4559.726 W) at oil flow rate (0.1 kg/s) and air inlet temperature (200°C) as compared with (2052.385W) for plain tube at the same boundary. The maximum enhancement percentage in heat transfer rate and overall heat transfer coefficient were is (122.167 % and 142.941%) at the same conditions.

8. By using helical tape with lower pitch (11 cm) in air side the pressure drop increase to (7437.8 pa) compared with (42.62 pa) for smooth pipe for high air inlet temperature (200°C) and high oil flow rate (0.1 Kg/s) and by using helical fin with lower fin pitch (10 cm) for oil side the pressure drop rise.
up to (1086.26 pa) compared with (892.81 pa) for smooth outer surface of inner tube at the same air temperature and oil flow rate above.

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