HEAT TRANSFER AND PRESSURE DROP CHARACTERISTICS IN A DOUBLE-PIPE HEAT EXCHANGER FITTED WITH A TURBULATOR

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ABSTRACT
The purpose of this study is to investigate heat transfer and pressure drop characteristics in a double pipe heat exchanger fitted with a helical-rod insert. Augmented heat transfer was above the plain tube values. The greatest improvement of heat transfer was found from helical-rod inserts where Nusselt numbers ranged from 150% to 160% comparison with the plain tube values at corresponding Reynolds numbers. It is shown that the pressure drop for the tube with the helical-rod insert is 6 to 9 times those of the plain tube for the range of Reynolds numbers tested.

Keywords: heat transfer, helical-rod, turbulence flow, double-pipe heat exchanger

1. INTRODUCTION
Heat exchanger is a device facilitating heat transfer between two or more fluids. It is extensively used in several industries, such as thermal power plants, chemical processing plants, air condition equipment, refrigerators, radiator for space vehicles as well as automobiles etc.[1-2]. Heat exchanger has been classified in many different ways. The design of heat exchanger is complicated, requiring a consideration of different modes of heat exchanger, pressure drop, sizing, long term performance estimation as well as economic aspect. Most of the designs are based on compactness of the unit. The compactness may be defined as the ratio of the heat transfer surface area on one side of the heat exchanger to the volume. Generally, a medium heat exchanger with a surface area density on any side greater then 700 m²/m³ is referred to as a compact heat exchanger regardless of its structural design. As a heat exchanger gets older, the resistance to heat exchanger rate increases due to fouling or scaling. This particularly true in heat exchangers used in marine as well as chemical industries. Also in some industries there is a need to increase the heat transfer rate in the existing heat exchanger. Therefore to maintain the desired heat transfer in existing heat exchangers several methods have been invented in the recent years.

Heat exchangers have several industrial and engineering applications. Techniques for enhancing heat transfer are relevant to several engineering applications. In recent years, the high cost of energy and material has resulted in an increased effort aimed at producing more efficient heat exchange equipment. The heat transfer rate can be improved by introducing a disturbance in the fluid flow (breaking the viscous and thermal boundary layers), but in the process pumping power may increase significantly and ultimately the pumping cost becomes high. Therefore, to achieve a desired heat transfer rate in an existing heat exchanger at an economic pumping power, several techniques have been proposed in recent years.

The great attempt on utilizing different methods is made to increase the heat transfer rate through the compulsory force convection. Meanwhile, it is found that this way can reduce the sizes of the heat exchanger device and save up the energy. In general, enhancing the heat transfer can be divided into 2 groups: One is the passive method, it is the way without being stimulated by the external power such as; surface coatings (treated surface) may be employed to enhance convection by promoting turbulence at the surface (special coatings are even used to promote condensation or boiling), rough surfaces are used to amplify mixing in the boundary layer near the surface this integral roughness is created by reforming and machining the surface, extended surfaces are commonly employed in heat exchangers to enhance heat transfer by increasing surface area, for example, wavy, lanced-offset, and perforated fins, the swirl flow devices for example, a twisted-tape [3-5], a wire-coil inserts, tangential injection devices [6] impart a tangential velocity component to the fluid that increases the heat-transfer coefficient, the convoluted (twisted) tube manifest both extended and roughened surface methods of heat-transfer augmentation by promoting turbulence [7] and additives for liquid and gases. The other is the active method. This way requires the extra external power sources, for example; mechanical aids involve mechanically stirring the fluid or rotating the heat exchanger, surface-fluid vibration may be employed by either vibrating the fluid or heat exchanger surface, the injection and the suction of the fluid, the jet impingement is the situation of the fluid flow direction normal to a surface from nozzle injection or orifice, and
the electrostatic fields are used to mix the fluid in order to increase the heat transfer coefficient.

In this experimental investigation, a helical-rod was used as turbulator for enhancing heat transfer rate in the double pipe heat exchangers. All of the experiments were carried out at the same inlet condition with the Reynolds number of the inner tube, Re=17000-44000.

Figure 1: Experimental apparatus.

Figure 2: Helical-rod inserted in the inner tube heat exchanger.
2. THEORETICAL ANALYSIS

For fluid flows in a heat exchanger the heat transfer rate can be expressed as:

\[ Q_{\text{tot}} = \dot{m} C_{p,\text{a}} (T_o - T_i) \]  

(1)

The convection heat transfer from the test section can be written by:

\[ Q_{\text{conv}} = h A (\bar{T}_w - T_b) \]  

(2)

whereas,

\[ T_b = (T_o + T_i)/2 \]  

(3)

and

\[ \bar{T}_w = \frac{\sum T_w}{15} \]  

(4)

Where \( T_w \) presents the local wall temperature and evaluated at the outer wall surface of the inner tube. The averaged wall temperatures are calculated from 15 points, lined between the inlet and the exit of the test pipe. The averaged heat transfer coefficient, \( h \) and the mean Nusselt number, \( Nu \) are estimated as follows:

\[ h = \frac{\dot{m} C_{p,\text{a}} (T_o - T_i)}{A (\bar{T}_w - T_b)} \]  

(5)

which

\[ Nu = \frac{h D}{k} \]  

(6)

The Reynolds number is given by

\[ Re = \frac{UD}{\nu} \]  

(7)

Friction factor, \( f \) can be written as:

\[ f = \frac{2}{D} \left( \frac{U^2}{\rho} \right) \]  

(8)

Where \( U \) is mean velocity of the tube. All of thermo-physical properties of the air are determined at the overall bulk air temperature from Eq. (3).

3. EXPERIMENTAL APPARATUS

The arrangement of experimental system of a double-tube heat exchanger was studied and details of test section are depicted in Figures 1 and 2. The double-tube heat exchanger is consisted of two concentric tubes; the inner tube for the hot air flow and the outer tube for the water flow. The diameters of the inner and outer tubes were 40 and 65 mm respectively. The tubes were 2000 mm long and 1 mm thick. Copper and steel tubes were employed for the inner and outer tubes respectively. The outer tube surface is covered with insulation to prevent heat loss. A helical-rod was inserted into the inner tube for the whole length to create turbulence flow. The 19 mm outer diameter helical-rod was used with pitch length of \( l = 3 \) mm. Details of the helical-rod and test tube geometries are presented in Figures 2(a-b).

Hot air from a 7.5 kW blower flowed through the inner tube, while cold water from a water pump enters the annulus. The volumetric flow rates of the hot air and cold water were adjusted by the appropriate rotometer and two globe valves situated before the inlet ports. The heating of the inlet air was by an electrical heater and the energy input was adjusted with a variable-output voltage transfer. Both the inlet-outlet temperature of the hot air and the cold water were measured by multi-channel with iron-Constantan thermocouple (type K). It is necessary to measure the temperature at the outer surface of the inner tube altogether 15 points for find out the average Nusselt number and the whole evaluated temperatures are connected to the date logger sets. While the entry and the exit of the inner-outer tubes are set up with pressure tape for measuring the pressure drop by connecting to the two U-tubes manometer and both are filled with water.

In the experiments, the counter flow is adjusted by two ball valves to control the direction of the cold water. For each test run, it is necessary to record the data of the temperature, volumetric flow rate and pressure drop of the hot air and the cold water at steady state. During the experimentation, it is considered to maintain the inlet temperature of the hot air constantly at 70°C with Reynolds numbers in the range of 17000 to 44000 and the cold water was kept at 25°C. The various characteristics of the flow and the finding out of the Nussels number are taken to the consideration from the Reynolds numbers and the average surface wall temperature.

4. RESULTS AND DISCUSSION

The results revealed the heat transfer rate and pressure drop in a double-pipe heat exchanger with a helical-rod insert were presented in Figures 3 and 4. The experimental results of the heat transfer rates, obtained in this study, are shown in Figure 3. It can be seen that, the helical-rod inserts gives higher values of heat transfer rate than those for plain tube, and the means Nusselt numbers increased around 150% when compared with those found from the plain tube. It is defined that a helical-rod insert caused turbulence flow and pressure gradient being created along the radial direction. The boundary layer along the tube wall would be thinner with the increase of radial velocity and pressure. Therefore, heat could be transferred easily through the flow. Moreover, turbulator would cause flow to be turbulent which led to even better convection heat transfer. It is depicted that the effect of the helical-rod inserts decreased at low Reynolds numbers due to the low flow velocity. Thus, the increase in Nusselt number was low at smaller Reynolds number, while it became greater at the higher Reynolds numbers. From experimental results, the inner tube fitted with a helical-rod gave the maximum heat transfer rate at about 160% in comparison with plain tube.
Pressure drop in the inner tube are shown in Figure 4 as function of Reynolds number. The pressure drop of straight flow (plain tube) was also plotted for comparison. It can be seen that the pressure drop were in the similar trend for both the straight flow and the tube with a helical-rod which the pressure drop of a helical-rod was higher than that in the plain tube because of the turbulence flow and the dissipation of dynamic pressure of the fluid at high viscosity loss near the tube wall. Moreover, the pressure drop had high possibility to occur by the interaction of the pressure forces with inertial forces in the boundary layer. Therefore, the pressure drop in the inner tube increased substantially with increasing Reynolds number. It was observed that the helical-rod inserts caused turbulence flow into the
tube which leaded to high pressure drop of 6 times over the plain tube.

5. CONCLUSIONS
Experimental data on the intermediate range of Reynolds number, Nusselt number and pressure drop, have been presented for case of a double-pipe heat exchanger. From experimental results, it can be observed that the helical-rod which placed inside inner tube results in a pressure gradient being created in the radial direction, thus affecting the boundary layer development. The increased rate of heat transfer in such flows is a consequence of the reduced boundary layer thickness and increased resultant velocity. It is important to note that the increase in pressure drop with a helical-rod generated turbulence flow is high than the increase in Nusselt number (at the similar Reynolds number). The maximum increase in Nusselt number is 160% in comparison with the plain tube, while the corresponding pressure drop is less than 6 times over the plain tube pressure drop.

REFERENCES