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An Experimental Study of Heat Transfer and Friction Factor Characteristics in a Circular Tube fitted with a Helical Tape

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Abstract

This paper provides heat transfer and friction factor data for single-phase flow in a double concentric tube heat exchanger fitted with a helical tape insert. In the double concentric tube heat exchanger, hot air was passed through the inner tube while the cold water was flowed through the annulus. The influences of the helical insert on heat transfer rate and friction factor were studied for counter flow, and Nusselt numbers and friction factor obtained were compared with previous data (Dittus 1930, Petukhov 1970, Moody 1944) for axial flows in the plain tube. The flow considered is in a low Reynolds number range between 2300 and 8800. A maximum percentage gain of 165% in heat transfer rate is obtained for using the helical insert in comparison with the plain tube.

Keywords: enhancement heat transfer, swirl flow devices, a helical tape insert

1. Introduction

In the past decade, heat transfer enhancement technology has been developed and widely applied to heat exchanger applications; for example, refrigeration, automotive, process industry, solar water heater, etc. The aim of augmentative heat transfer is to accommodate high heat fluxes (or heat transfer coefficient). Up to the present there has been a great attempt to reduce the sizes and cost of the heat exchanger, and energy consumption. The most significant variable in reducing the size and the cost of the heat exchanger, which generally leads to less capital cost and another advantage, is reduction of the temperature driving force, which increases the second law efficiency and decreases the entropy generation. Thus, this captivates the interests of the number of researchers.

The great attempt on utilizing different methods is 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 two groups: One is the passive method; it is the way without being stimulated by the external power such as surface coatings, rough surfaces, extended surfaces, the swirl flow devices, the convoluted (twisted) tube, additives for liquid and gases. The other is the active method. This way requires the extra external power sources, for example mechanical aids, surface-fluid vibration, the injection and the suction of the fluid, the jet impingement, and the electrostatic fields.

The swirl flow devices can be classified into two kinds: the first is the continuous swirl flow and the other is the decaying swirl flow. For the continuous swirl flow, the swirling motion persists over the whole length of the tube for example twisted tape inserts [1,2], coiled wires inserted along the whole tube [3] and helical grooves in the inner surface of tube generate, while in the decaying swirl flow, the swirl is generated at the entrance of the tube and decays along the flow path for example the radial guide vane swirl generator and the tangential flow injection device [4,5,6,7,8]. For the decaying swirl flow, the heat transfer coefficient and pressure drop decrease with the axial distance, while for the continuous swirl flow, the heat transfer coefficient and pressure drop keep constant. In this reports, the experiments were set to study the effect of swirling flow or rotation flow on the improvement in performance of a double pipe heat exchanger, equipped with a helical tape placed the whole length of the inner tube which is assumed as a vortex/swirl generators or turbulence promoter to agitate the fluid as shown in figure 1.

2. Calculation of Heat Transfer and Friction Factor

For fluid flows in a concentric tube heat exchanger, the heat transfer rate of the hot fluid (air) for the inner tube can be expressed as:

$$Q = mC_p(T_o - T_i) \tag{1}$$

while the heat transfer of the cold fluid (water) for the outer tube is

$$Q = hA(\widetilde{T}_w - T_b) \tag{2}$$

whereas,

$$T_b = (T_o + T_i)/2 \tag{3}$$

and

$$\widetilde{T}_w = \sum T_w / 15 \tag{4}$$

Where T_w is the local wall temperature and evaluated at the outer wall surface of the inner tube. It must be measured at the depth from the outer surface of 0.5 mm. The average wall temperatures are calculated from 15 points, lined between the inlet and exit of the inner tube. The average heat transfer coefficient and the mean Nusselt number are estimated as follows:

$$h = mC_p(T_o - T_i)/A(\widetilde{T}_w - T_b) \tag{5}$$

$$Nu_m = hD_h / k \tag{6}$$

The Reynolds number is given by

$$Re = VD_h / v \tag{7}$$

Friction factor can be written as follows:

$$f = \frac{\Delta P}{\left(\frac{L}{D}\right)\left(\rho \frac{V^2}{2}\right)} \tag{8}$$

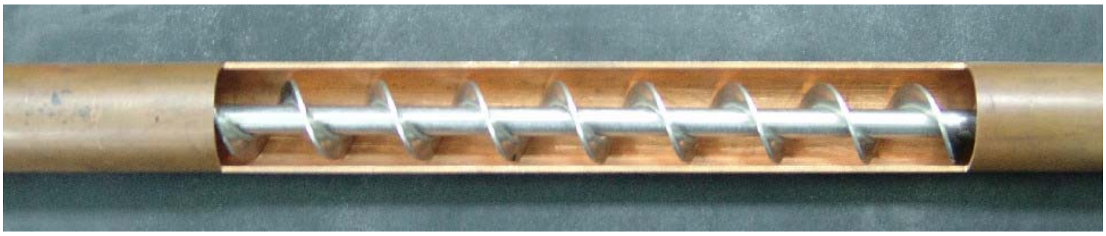

Figure 1. The inner tube fitted with the full-length helical tape.

Nomenclature

Q	heat transfer rate (W)	Re	Reynolds number
m	mass flow rate (kg/s)	Nu	Nusselt number
C_p	specific heat capacity (kJ/kgK)	V	average axial velocity (m/s)
T	temperature (K)	v	kinematic viscosity (m^2/s)
h	heat transfer coefficient (W/m^2K)		
k	thermal conductivity (W/mK)	*Subscripts*	
A	heat transfer area (m^2)	i	inlet
D_h	hydraulic diameter (m)	o	outlet
D	tube diameter (m)	w	local wall
L	inner tube length (m)	c	cold fluid
P	pressure (Pa)	h	hot fluid

f	friction factor	b	mean

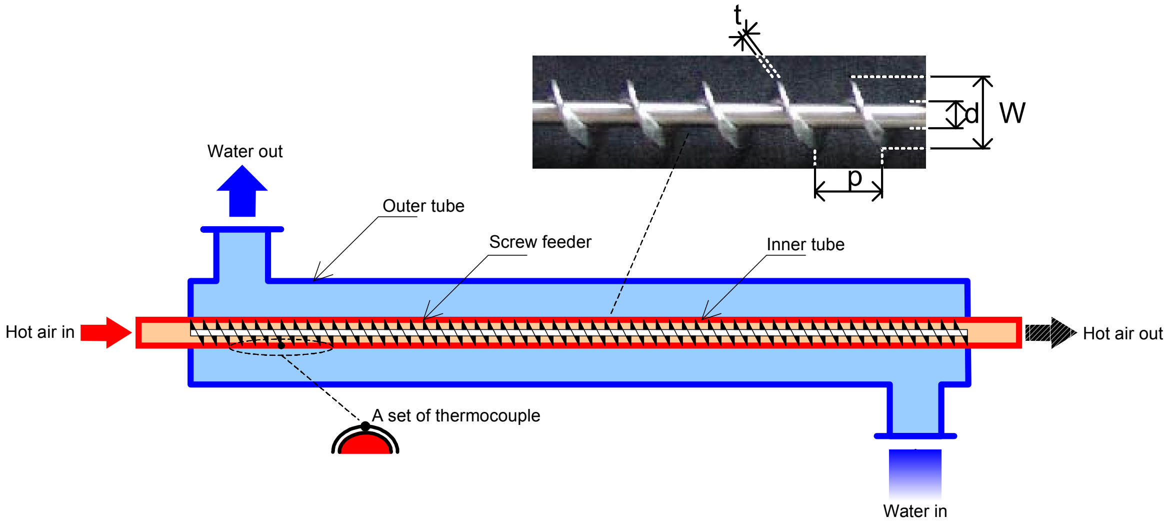

Figure 2. A double concentric tube heat exchanger fitted with a helical and definition of geometric parameters of the helical tape.

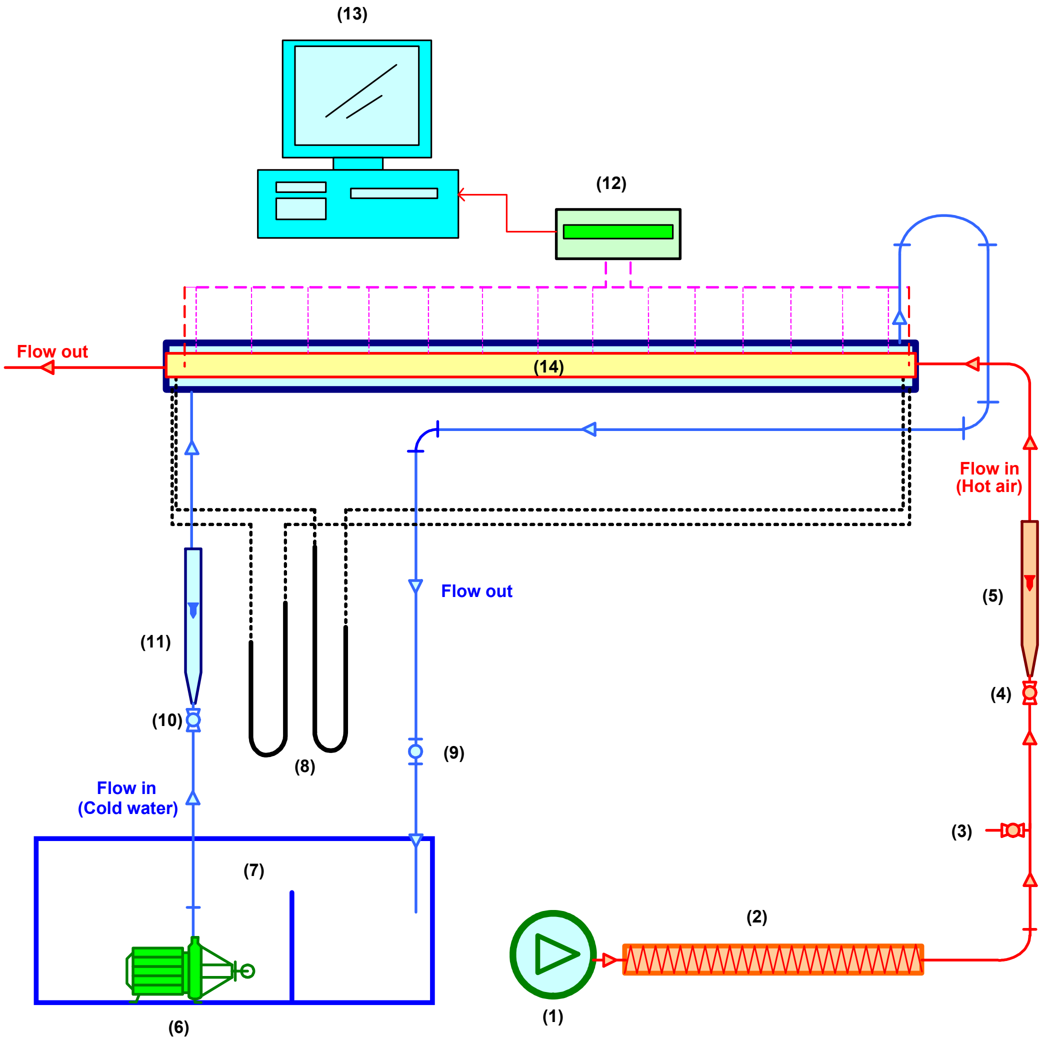

Figure 3. Schematic diagrams of experimental apparatus: (1) 10 hp blower, (2) an electrical heater with thermostat, (3) ball valve, (4) globe valve, (5) rotameter for hot air, (6) ½ hp water pump, (7)

chiller, (8) U tube manometer, (9) ball valve, (10) globe valve, (11) rotameter for cold water, (12) data logger, (13) PC computer, and (14) a double concentric tube heat exchanger.

3. Experimental Apparatus and Procedure

The arrangement of the details of test section and the experimental system of a concentric tube heat exchanger was set up are depicted in figures 2 and 3. The double pipe heat exchanger consisted of two concentric tubes; the inner tube for hot air flow and the outer tube for water flow. The diameters of the inner and outer tubes were 19 and 40 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 was wrapped with insulation to minimize heat loss. In figure 2 is presented the helical tape insert used in this test. In the present experiments, the geometric conditions of the helical insert were kept constant. The helical insert was made of stainless steel and has the geometric dimensions of W=17 mm (0.95D), d=5 mm (0.26D), P=18 mm (0.95D), and t =1 mm (0.05D), respectively.

Hot air from a 10 hp blower was directed through the inner tube, while cold water was pump through the annulus. Liquid and air flow meters were used to measure both Liquid and air flow rates. The volumetric flow rates of the hot air and cold water were controlled by globe valves, situated before the inlet ports. The inlet air was heated by an adjustable electrical heater. Both the inlet and outlet temperature of the hot air and the cold water were measured by multi-channel with iron-Constantan thermocouple (type K). It was necessary to measure the temperature at 15 stations altogether at the outer surface of the inner tube for finding out the average Nusselt number. All fifteen evaluated temperature probes were connected to the data logger sets. The entrance and the exit of the inner-outer tubes were provided with pressure tapings for measuring the pressure drop by connecting to the U-tube manometer.

For each test run, it was necessary to record the data of temperature, volumetric flow rate and pressure drop of the hot air and the cold water at steady state. The inlet air temperature was maintained at 80^{o}C and the cold water was at 27^{o}C during the experiment. The Reynolds number of the heated air was from 2300 to 8800. The various characteristics of the flow and the Nussels number, the Reynolds numbers were based on the average surface wall temperature.

4. Results and Discussion

The heat transfer and friction factor data for the plain tube were collected first. Such data served as a check for the validity of experimental apparatus and measurement techniques over the whole range of flow conditions. The plain tube data are shown to agree well with Petukhov correlation (1970) [9] which the data of Dittus and Boelter (1930) [10] correlations gave higher value than present work of plain tube because of correlation of those correlations were used at high turbulent flow which $Re > 1x10^5$.

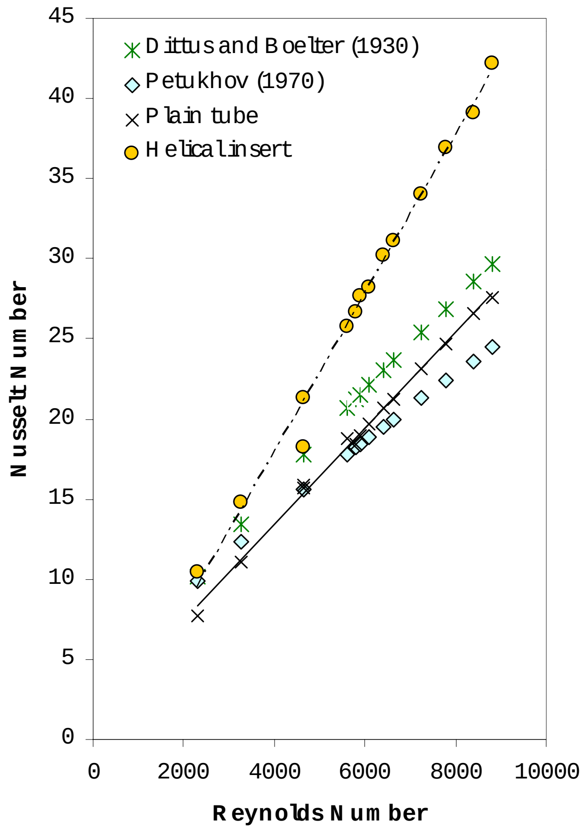

Figure 4. Relationship between Nusselt number and Reynolds number.

The results revealed the heat transfer rate and friction factor in a concentric tube heat exchanger fitted with a swirl generator or a helical insert as depicted in figures 4 and 5. The experiments were arranged in a counter flow system.

The experimental heat transfer rates for the hot air obtained in this study are shown in figure 4. It can be seen that, in case of the helical tape insert gives higher values of heat transfer rate than those for the plain tube. In the case of the swirling flow or the helical tape insert, the means Nusselt numbers increased around 150% in comparison with the plain tube. It is defined that a helical insert caused swirl flow or secondary air 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 swirl and pressure. Therefore, heat could be transferred easily through the flow. Moreover, swirl would cause flow to be turbulent which leaded to even better convection heat transfer. From the figure, it is depicted that the effect of the helical insert decreased at low Reynolds numbers due to the weak swirl and 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 gave the highest heat transfer at about 165% which compared with plain tube.

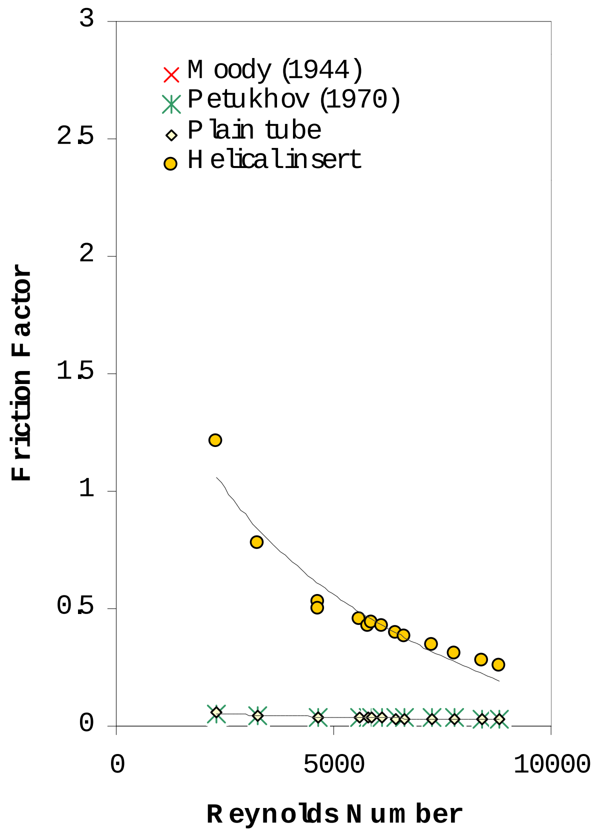

Figure 5. Relationship between friction factor and Reynolds number.

Friction factor in the inner tube is shown in figure 5 as function of Reynolds number which the friction factor of axial flow (plain tube) was also plotted for comparison [10-11]. It can be seen that the friction factor was in the similar trend for both the plain flow and the helical insert and the friction factor of the helical insert was higher than that in the plain tube because of the swirling flow or secondary flow and the dissipation of dynamic pressure of the fluid at high viscosity loss near the tube wall. Moreover, the pressure loss had high possibility to occur by the interaction of the pressure forces with inertial forces in the boundary layer. Therefore, the friction factor in the inner tube increased substantially with increasing Reynolds number. It was observed that the helical insert caused swirling flow into the tube and leaded to high friction factor above the plain tube.

5. Flow Visualization Results

Flow visualization experiments are performed in order to obtain an overall understanding flow characteristic of the inner tube fitted with a helical tape. In the experiments, the helical was inserted into the acrylic circular tube with 19 mm in diameter which the length and the diameter are equal to the length and the diameter of the inner tube of the concentric tube heat exchanger. In particular, when we take a picture of pathlines produced by the motion of small particles suspended in fluid, we can even estimate some quantities such as velocity and acceleration. A quantity of aluminium power with particles 2-7 μm in length to be used as tracer is suspended uniformly in the water of the tube. The photographic time interval available with this camera is between 1.3 to 600 s, in the case of our usual exposure time of 0.9s. The above studies have shown that using water as the working fluid in a model concentric tube heat exchanger introduces two basic advantages. First, by using water it is possible to greatly reduce the velocities involved in the experiment. Second, there are many easily photographed traced particles available with a density close to that of water.

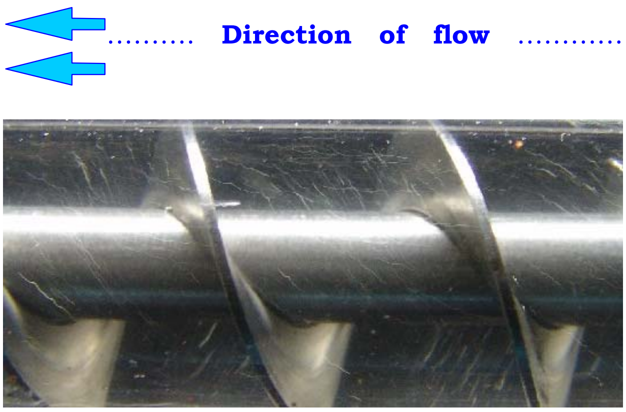

Figure 6. The swirling flow in a circular tube fitted with the helical.

Patterns of the flow through the helical in the tube were observed as shown in figure 6. The flow characteristics were photographed at the Reynolds number 4000. From the photograph, it can be observed that the swirling flow around the helical and along the whole tube length. This effect is related to the speed of rotation of the fluid and caused to the breakdown of the boundary layer in a short time then the heat transfer could be transfer easier.

6. Conclusions

An experimental study was performed on a helical insert in a circular tube using hot air as the test fluid. The influence of the helical tape insert on the heat transfer and friction factor characteristics has been demonstrated. The result shows that the increases in heat transfer and friction factor are strongly influence by swirling motion induced by the helical. As the Reynolds number increases, the swirling flow is stronger which in turn results in an increase in the heat transfer and friction factor while it decreases at low Reynolds number.

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