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Turbulent Heat Transfer in a Horizontal Helically Coiled Tube

Bofeng Bai, Liejin Guo, Ziping Feng, and Xuejun Chen
State Key Laboratory of Multiphase Flow in Power Engineering, Xi'an Jiaotong University, China

An experiment has been conducted in detail to study the turbulent heat transfer in horizontal helically coiled tubes over a wide range of experimental parameters. We found that the enhancement of heat transfer in the coils results from the effects of turbulent and secondary flows. With Reynolds number increasing to a high level, the contribution of the secondary flow becomes less to enhance heat transfer, and the average heat transfer coefficient of the coil is closer to that in straight tubes under the same conditions. The local heat transfer coefficients are not evenly distributed along both the tube axis and the periphery on the cross section. The local heat transfer coefficients on the outside are three or four times those on the inside, which is half of the average heat transfer. A correlation is proposed to describe the distribution of the heat transfer coefficients at a cross section. The average cross-section heat transfer coefficients are distributed along the tube axis. The average value at the outlet section should not be taken as the average heat transfer coefficient. © 1999 Scripta Technica, Heat Trans Asian Res, 28(5): 395–403, 1999

Key words: Turbulent heat transfer, horizontal helically coiled tube, secondary flow, local heat transfer

1. Introduction

Because of their compact structure, high heat transfer coefficients, and ease of manufacture and arrangement, helically coiled tubes are used extensively in heat recovery systems, compact heat exchangers, storage tank heating systems, nuclear reactors, chemical plants, and other equipment [1, 2]. They can also be used in navigation and other modern technologies where efficient heat transfer and space limitations are of great importance.

Many investigators have studied turbulent flow and heat transfer in coils. Mori and Nakayama [3, 4] worked theoretically and experimentally on them under fully developed conditions and studied the distribution of the secondary flows. A correlation (1) covering a wide range of Reynolds number was obtained but they did not have sufficient experimental evidence to back it up:

$$\mathrm{NuPr}^{-0.4} = \frac{1}{41}\mathrm{Re}^{5/6}\left(\frac{d}{D}\right)^{1/12}\left\{1 + \frac{0.061}{[\mathrm{Re}(d/D)^{2.5}]^{1/6}}\right\} \tag{1}$$

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Seban and McLaughlin [5] conducted an experiment with two coils using water as the working fluid. *D*/*d* values of the coils were 17 and 104, where *D* and *d* are the coil and tube diameters. The working fluid was heated with electricity. The Reynolds number range varied from 6000 to 65,600. In the process of the experimental data reduction, they assumed that the thermal properties were constant, the tubes were taken as straight, and the peripheral heat conduction was negligible. They considered that 10% error might have resulted from the above assumptions. Experimental data show that distributions of the turbulent heat transfer coefficients were uniform longitudinally but not uniform peripherally. A correlation for the average heat transfer coefficient of coils was presented as follows:

$$\mathrm{Nu} = 0.023\ \mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}\ [\mathrm{Re}\ (d/D)^2]^{0.05} \qquad (2)$$

Here, the average temperature of the liquid film was defined as the characteristic temperature. The results show that this correlation is more suitable for the small coils than for larger ones.

Rogers and Mayhew [6] studied the effect of thermal properties on turbulent heat transfer. Three coils of *D*/*d* 10.8, 13.3, and 20.1 were used. The Reynolds number range was from 10,000 to 100,000. The following equations were proposed for the average turbulent heat transfer coefficient:

$$\mathrm{Nu}_\mathrm{f} = 0.021\ \mathrm{Re}_\mathrm{f}^{0.85}\mathrm{Pr}_\mathrm{f}^{0.4}\ (d/D)^{0.1} \qquad (3)$$

$$\mathrm{Nu}_\mathrm{b} = 0.023\ \mathrm{Re}_\mathrm{b}^{0.85}\mathrm{Pr}_\mathrm{b}^{0.4}\ (d/D)^{0.1} \qquad (4)$$

Here, the subscripts f and b denote liquid film and bulk temperature, respectively.

In the above-mentioned research, all of the coils were vertically oriented, and the range of Reynolds number was not large. Although Mori and Nakayama considered that their correlation obtained on the basis of a simplified model might be suitable for a wide range of Reynolds number, there were insufficient experimental data to back this up. In fact, the distribution of the heat transfer at every cross section is not uniform, although little work has been done in this regard.

In the present paper, experiments were conducted to obtain a correlation for the average heat transfer over a wider range of horizontal helical coils and to obtain a deeper understanding of the local heat transfer characteristics in both the peripheral and longitudinal directions.

2. Experimental Apparatus and Procedure

The experimental apparatus is a closed-cycle loop and schematically illustrated in Fig. 1. The loop is made of 1Cr18Ni9Ti stainless steel. It consists of a centrifugal pump, a pressurized nitrogen tank to control the system pressure, a series of orifices to measure water mass flow rate, a preheater to control the inlet fluid temperature, a test section, a water-cooled condenser, and a water tank. The loop is thermally insulated by wrapping with fiberglass. The resistance of the tube walls of both the test section and the preheater was used to heat the working fluid with alternating current.

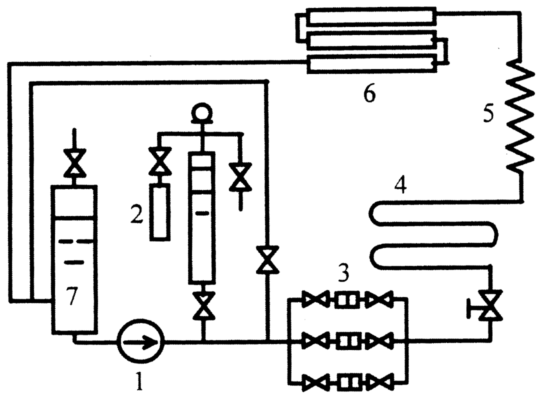

Fig. 1. Schematic representation of the experimental system. 1, centrifugal pump; 2, pressurized nitrogen tank; 3, orifice; 4, preheater; 5, test section; 6, condenser; 7, water tank.

The test section (Fig. 2) is made of stainless steel of 1Cr18Ni9Ti. The total length of the test section is 6448 mm. The inner diameter is 11 mm and the thick mass of the tube wall is 2 mm. The helix angle, coil diameter, and the pitch are 4.27°, 256 mm, and 60 mm, respectively.

In the experiments, deionized water is used as the working fluid. The mass flowrate of the working fluid is measured using three calibrated orifice meters of different ranges. Two manometers are employed to measure the pressure at the inlet and outlet. The differential pressure of the test section is measured using the DP1151 capacitance-type differential pressure transducer. Three ϕ 3 mm NiCr–NiSi armored thermocouples are installed inside the tube to measure the fluid temperature at the inlet, outlet, and the center section of the test section. Eight (first, second, and third turn) ϕ 0.3

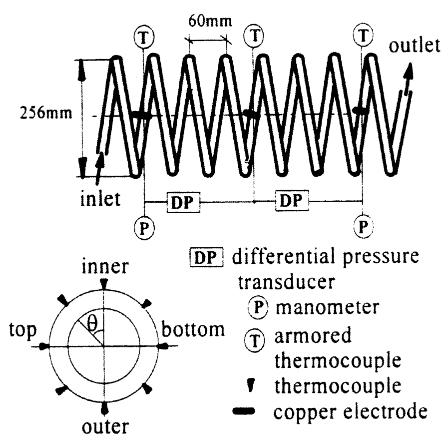

Fig. 2. Coil tube configuration and the arrangement of probes, transducers, and thermocouples.

mm NiCr–NiSi thermocouples are installed on one cross section every quarter turn. These thermocouples are attached to the pipe wall and electrically insulated so that the effect of heating current on the thermocouple readings is reduced.

All of the signals of the mass flowrate, pressure, temperature of the tube wall and the fluid, and the input heating powers of the preheater and the test section are monitored and stored in a computer via six isolated measurement pods.

Experiments were completed under the following conditions: pressure ranged between 0.5 and 3.0 MPa, mass flow rate between 200 and 2500 kg/s, heat flux on the inner wall surface between 230 and 450 kW/m^2.

3. Data Reduction

In order to calculate the heat transfer coefficients, the temperature and heat flux values on the interior surface of the tube and the bulk temperature of the working fluid are required. The two-dimensional inverse heat conduction problem is solved with the least-squares method based on the following assumptions [7]:

1. The interior heat source is evenly distributed.

2. The longitudinal bulk temperature distribution is linear.

3. The thickness of the tube wall is considered to be constant.

4. The longitudinal heat conduction is small and negligible.

The fluid bulk temperature at every cross section was obtained using the heat equilibrium method and the state equation of water, $T_b = f$ (pressure, enthalpy), under assumption 2. The mean of those of the last five cross sections was taken as the average heat transfer coefficient of the coil.

4. Experimental Results and Discussion

4.1 Average turbulent heat transfer coefficient

Figure 3 shows the curve of $\mathrm{NuPr}^{-0.4}(\mu_b/\mu_w)^{-0.11}$ versus Reynolds number. The average heat transfer coefficient for developed turbulent flow in a horizontal helically coiled tube can be calculated with the following correlation within the parameter range of this test:

$$\mathrm{Nu} = 0.328\ \mathrm{Re}^{0.58}\mathrm{Pr}^{0.4}(\mu_b/\mu_w)^{0.11} \quad \text{for } 4.5 \times 10^4 < \mathrm{Re} < 19 \times 10^4 \tag{5}$$

Here, the characteristic temperature is defined as the temperature of the bulk fluid. Nu, Pr, Re, and μ are the Nusselt number, Prandtl number, Reynolds number, and viscosity, respectively; b and w refer to the temperature of the bulk fluid and the tube wall, respectively. The maximum error between Eq. (5) and the test data are ±6%. The Dean number $[\mathrm{Dn} = \mathrm{Re}\sqrt{(d/D)}]$ is often used to describe the secondary flows. The heat transfer may be enhanced with increasing Dn while the friction also

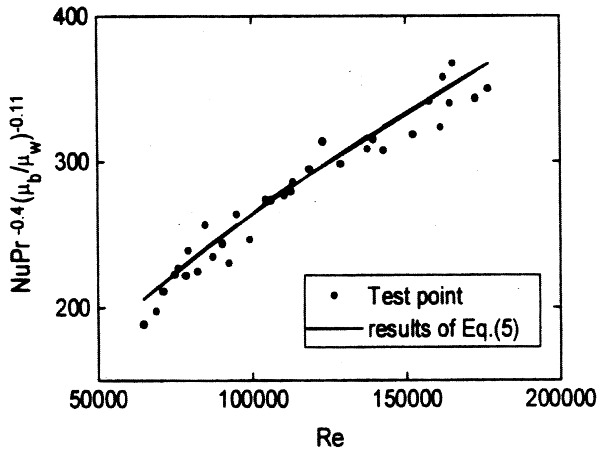

Fig. 3. Test results on the average turbulent heat transfer.

increases [8]. Equation (5) is not expressed with the Dean number here because only one coil was used in the experiment.

The average heat transfer coefficient for developed turbulent flow in a horizontal helically coiled tube is compared with Dittus-Boelter's Eq. (6) for a straight tube [9] to show the differences of enhancement of heat transfer between coils and straight tube:

$$\mathrm{Nu} = 0.023\ \mathrm{Re}^{0.8}\ \mathrm{Pr}^{0.4} \tag{6}$$

This correlation can be used under the following conditions: $0.7 < \mathrm{Pr} < 120$, $10^4 < \mathrm{Re} < 1.2 \times 10^5$, $L/d > 60$. The characteristic temperature is defined as the temperature of bulk fluid.

Figure 4 shows the comparison among Eqs. (1), (4), (5), and (6). These curves are obtained under the same conditions. It is seen that for most Reynolds numbers the experimental values are smaller than those calculated with Eqs. (1) and (4), but greater than those calculated with Eq. (6). If the Reynolds number increases greatly, the difference between the test data and the values of Eqs. (1) and (4) becomes greater, but the difference between the test data and the values of Eq. (6) becomes less. In this experiment, the heat transfer of the coil is close to that of a straight tube under the same condition when the Reynolds number increases up to 180,000. The results show that the enhancement of heat transfer of coils tends to be less with the Reynolds numbers increasing to a high level.

In developed turbulent flow, the flow field can be divided into two parts: a laminar sublayer and a turbulent core [10, 11]. The sublayer is close to the wall where the radial gradient of fluid temperature is a maximum [12]. When the Reynolds number is increased, the sublayer becomes thin. Therefore, the radial gradient of the fluid temperature becomes greater and the heat transfer is enhanced greatly.

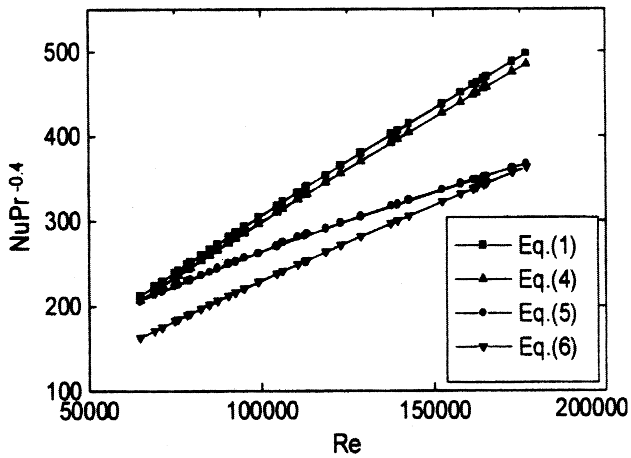

Fig. 4. Comparison among different equations.

On the other hand, the secondary flows carry the fluid flowing from the outer side to the inner side along the tube wall and then back through the core to the outer side. In this process, the heat and mass of the thermal boundary layer are transported to the core and mixed with the cool fluid in the core. The boundary layer is thicker; a greater amount of heat energy is transported by the secondary flow [13]. With Reynolds number increasing, the boundary layer becomes thinner, therefore the effect of secondary flows on the enhancement of heat transfer becomes small, and the enhancement of heat transfer mainly results from the turbulent flow. Some researchers failed to find this phenomenon since the Reynolds number in their tests did not reach a high level.

4.2 Peripheral local turbulent heat transfer coefficients

The local heat transfer coefficients are not evenly distributed around the peripheral section. The local turbulent heat transfer characteristics are examined in detail by means of the local relative heat transfer coefficient distribution versus the circumferential angle, θ, as shown in Fig. 5. Nu_L is the local Nusselt number, and Nu is the average on the cross section, which is calculated by Eq. (5). The local heat transfer coefficient has the largest value on the outer side which is 1.6 or 2.2 times that of the average heat transfer coefficient on the cross section. It is the smallest on the inner side which is half the average one, and it has a mean value on the top and bottom. With RePr increasing, the ratio of the local heat transfer coefficient at the outer side to the average one at the cross section tends to be greater, but the ratio of that on the inner side to the average remains constant. This phenomenon was also observed by Seban and McLaughlin [6]. The peripheral distribution is close to symmetric but is not symmetric in laminar flow because of natural convection and the tortuous path [14].

The local distribution of the heat transfer coefficients in the periphery can be estimated with the following correlation:

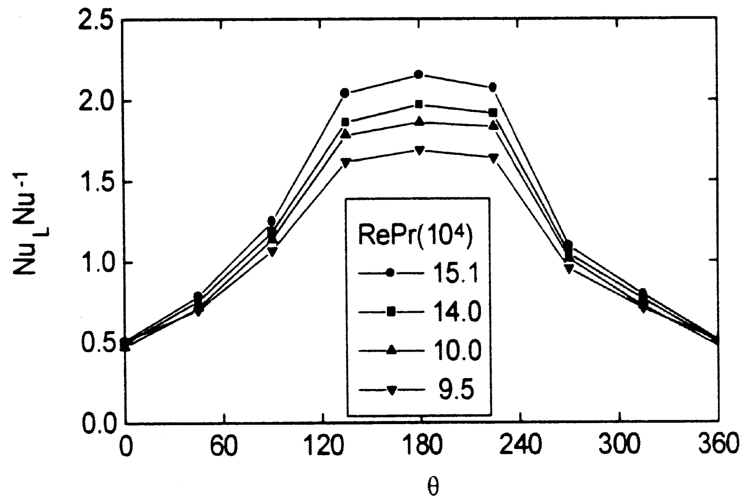

Fig. 5. Peripheral local turbulent heat transfer coefficient on the outlet cross section.

$$\frac{Nu_L}{Nu} = 0.22\left(\frac{RePr}{10^4}\right)^{0.45}(0.5 + 0.1\theta + 0.2\theta^2) \text{ for } 0 < \theta \le \pi \qquad (7)$$

Two reasons are responsible for this distribution. The axial velocity near the outer side is greater than that near the inner side of the wall surface, therefore the cooling effect of the turbulent flow on the wall near the outer side is greater than that on the inner side wall. In addition, the fluid carried by the secondary flow is heated during its circumferential transport. The fluid temperature is lowest at the outer side and greatest at the inner side, so the heat transfer between the fluid and the outside wall surface is greater than that on the inner side. The distribution of the fluid temperature was calculated by Mori and Nakayama [3, 4] and their results agreed with the present analyses.

4.3 Longitudinal local turbulent heat transfer coefficient

As the coil is horizontally oriented, the angle between the gravity and flowing direction changes constantly and periodically. Therefore, the velocity distribution on each cross section within one turn varies periodically along the tube axis. Figure 6 shows the distribution of the average heat transfer coefficient on the cross section in the form of Nu_L/Nu along the tube axis. Nu_L and Nu are the average value of the cross section and of the coil, respectively. Nu is calculated with Eq. (5). The horizontal coordinate, L, is the axial distance, which is equal to zero at the center section of the coil. In upward flow, the average heat transfer coefficient is the greatest which is 120% to 130% of the average of the coil. The distributions have a good periodical feature.

The phenomenon can be explained as follows: In the upward flow, the gravity is opposite to the fluid flow and it acts as a dragging force, so that the velocity profile becomes plane and the sublayer becomes thinner and heat transfer is enhanced by the turbulent flow. Therefore, it is not reasonable

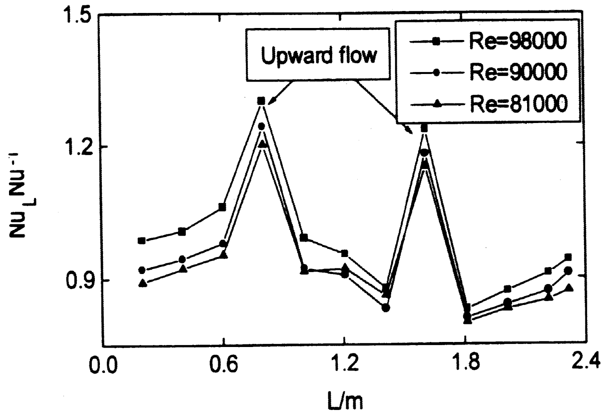

Fig. 6. Local turbulent heat transfer coefficient along the tube axis.

to take the average heat transfer coefficient on the outlet cross section as the average one of the horizontal coils.

5. Conclusions

1. The turbulent heat transfer has been experimentally investigated in the present study for a horizontal helically coiled tube over a wide range of parameters. The average heat transfer coefficient data of the coil can be correlated with Eq. (5) in the range of Reynolds number 4.5×10^4 to 19×10^4.

2. The average turbulent heat transfer mainly results from the turbulent and secondary flows. The boundary layer is thicker and the contribution of the secondary flow to the heat transfer enhancement in coils becomes greater. With Reynolds number increasing, the turbulent sublayer becomes thinner and the secondary flow provides a smaller contribution to the enhancement of heat transfer.

3. The distribution of heat transfer coefficient on the periphery of the cross section is approximately symmetric. The local heat transfer coefficient is the largest on the outer side, while it is the smallest on the inner side and a mean value on the top and bottom. A correlation to describe the peripheral local turbulent heat transfer coefficients on the cross section is proposed.

4. The longitudinal distribution of the average heat transfer coefficients on the cross sections has a good periodical feature. It is greatest in the upward flow. Therefore, it is not reasonable to take the average value at the outlet section as the average heat transfer coefficient of the whole coil as done previously.

Acknowledgments

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