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Experimental study on the heat transfer enhancement by Dean Vortices in spiral tubes

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Abstract
Spiral tubes have been reported to be able to improve the heat transfer performance of the fluid flowing through, which is traditionally believed to be caused by the helix structure. In this literature, however, the mechanism of the heat transfer enhancement of spiral tubes was studied experimentally in terms of the control factors of Dean vortices such as Reynolds number, fluid viscosity and curvature ratio. Meanwhile, the torsion of the spiral tube was also taken into consideration. Experiments were carried out under constant wall temperature condition with one straight copper tube and 7 kinds of spiral copper tubes. The results showed that the Nusselt number of the flow in spiral tubes increases with Reynolds number, viscosity and curvature ratio, but changes anomalously with the torsion. And it also clearly revealed that the pressure drop for spiral tubes increases with Reynolds number, curvature ratio and viscosity, while decreases with the torsion. In addition, the heat transfer performance of spiral tubes is apparently better than that of the straight tube. Based on the experiment results, the effect of Dean number on Nusselt number and pressure drop of the spiral tubes was determined, and the correlation equations of heat transfer enhancement by Dean vortices were obtained through multiple regression.

Keywords: Spiral tube; Dean Vortices; Heat transfer enhancement; Nusselt number.

1. Introduction

Spiral tubes have been introduced as an important heat transfer enhancement component of heat exchangers and are widely used in various industrial applications, such as the air conditioning systems, heat recovery processes, chemical reactors. Traditionally, the mechanism of heat transfer enhancement by spiral tubes was attributed to their large heat transfer areas, the compact structure, and, above all, their special geometry. Therefore, studies were carried out in terms of the structure of the spiral tubes to determine the mechanism of the heat transfer enhancement. For example, Heo [1] described the heat transfer augmentation phenomenon of spiral coils under different tube diameter, length, height, pitch, radius, and number of turns. And the natural convection heat transfer from spiral coiled tubes was studied at different inlet temperatures, bath temperatures, coil dimensions, and fluid flow rate by Prabhanjan [2]. There are still other similar works, such as the studies by Yi [3], Ali [4, 5], E. Martínez [6] and so on.
However, the heat transfer characteristics in spiral tubes have rarely been investigated in terms of the Dean vortices, though recent studies have proved that the heat transfer enhancement for the spiral tube

strongly depends on the motion of Dean vortices [7]. The Dean vortices, caused by the centrifugal force, is a kind of secondary flow first found by Dean in 1920s. One unique of the Dean vortices is the counter-rotational-vortex structure [8]. Perturbation solutions were obtained by Dean under the curved tube with small curvature ratio and circular cross-section, and the dimensionless Dean number which could be used to weigh the strength of the vortices was presented [9],

$$De = \mathrm{Re}\cdot\left(r/R_c\right)^{0.5} = 2r\cdot V\cdot\upsilon^{-1}\cdot\left(r/R_c\right)^{0.5} \tag{1}$$

where De denotes the Dean number, Re means the Reynolds number, r and R_c indicate the radius and curvature radius of the tube(m), respectively. And the symbols V and υ are the main flow velocity(m/s) and viscosity of the fluid in the tube, respectively.

Theoretically, the flow pattern of Dean vortices can promote the mixing effect of the fluid in the tube, thus augment of the heat and mass transfer of the fluid. Many researchers have studied, theoretically and experimentally, the Dean vortices for a long time. However, most of the studies were focused on the flow characteristics and mass transfer augmentation. For example, R.Moll and his colleagues studied the mass transfer characteristics experimentally and numerically by measuring the velocity profiles and the local wall shear stress induced by Dean vortices [10,11]. And H. Fellouah presented the studies on Dean instability for Newtonian fluids in laminar secondary flow in 180° curved channels experimentally and numerically [12], and he applied the numerical and experimental methods to analyze the Dean instability in power-law and Bingham fluids in a 180° curved channels of rectangular cross section to understand the effect of rheological fluid behavior on the Dean instability [13]. Except the results stated above, there are still many authors studying on the flow characteristics [14, 15] and mass transfer augmentation characteristics [16-19] of Dean vortices. But few published studies on the heat transfer enhancement by Dean vortices were found, except the works by Ligrani [20]and Tilak [21].

Critical literature review reveals that no explicit correlation between Nusselt number and Dean number is presented to better describe the heat transfer augmentation characteristics in spiral tubes in the present studies available. Also, a lack of consideration exists on the experiments in the literatures: no experiment considered all the controlling factors of Dean vortices. Therefore, it is of necessity to carry out experiments considering the Reynolds number, viscosity, curvature ratio, and torsion of spiral tubes, to better study the heat transfer augmentation by the Dean vortices. Finally the relationship between the Nusselt number and Dean number, and the relationship between pressure drop and Dean number can be determined. The determination of these relationship is an important step in the design of heat exchangers.

2. Experimental work

2.1 Experimental setup

A schematic diagram of the experimental setup is shown in Figure 1. The apparatus needed in the experiments are given in Table 1, and the physical dimensions of test tubes used in the experiment are listed in Table 2.

The inner diameter of the tubes is 19mm each, including the straight tube. And all the tubes have a smooth and plain inner wall surface, therefore the surface roughness has negligible influences. It will not have significant influences on heat transfer phenomenon. The tubes in the experiments were connected to the circulation pipe with straight joints. And the length of the straight part of the circulation pipe connected to the inlet of the test tubes was at least 950mm (10 times of the tube diameter), to minimize to influences of entrance effects. What's more, the circulating pipe was covered by thermal isolating materials to minimize the heat loss. Digital thermometers and pressure transducers were installed at the inlet and outlet of the tubes to acquire pressure and temperature data. For the spiral tubes, the outlet located in the water tank due to the dimensions of the tank. As a result, the thermometer and pressure transducer should be covered with thermal insulation material to ensure the accuracy of the data acquisition. For the purpose of diminishing the influence of radiation, both the surfaces of the water tank and water container were covered by heat insulation material. In order to achieve a constant wall temperature condition, a electric heater with a power of 2500W was employed in the experiments. The electric heater was controlled by a thermal regulator with a precision of 0.1 °C, thus the water in the water tank could be stabilized at a constant temperature approximately. The flow rate in the test tubes was controlled by the pump and the ball valves. When the pump was running, a certain flow rate, which indicated a certain inlet velocity of the tubes, could be obtained by controlling the ball valves in the circulation pipes. The exact flow rate could be determined by measuring the volume of the fluid flowing out of the drain point, with the help of the graduate and timer. Finally, 3 kinds of fluid/solution with different viscosities were introduced to test the influences of viscosity on the heat transfer characteristics of test tubes. Apparatus of the experiments are listed in Table 1.

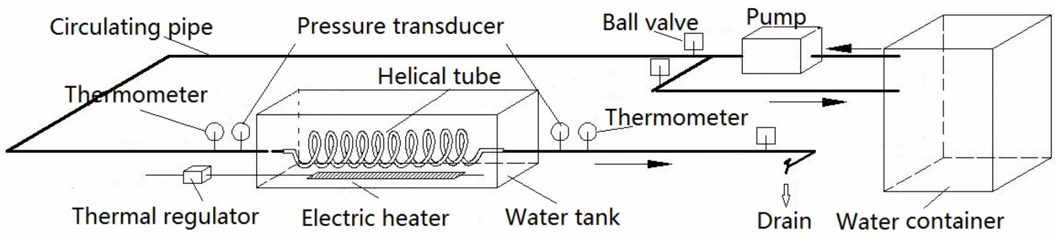

Figure 1. Scheme of the experimental setup

Table 1. Apparatus needed in experiments

Name	Parameter	Name	Parameter
Water container	1000×1000×500mm	Copper tube	L=750mm;r=19mm
Water tank	800×200×200mm	Electric heater	2500W
Pump	1.5m3/h	Thermal regulator	Precision: 0.1°C
Ball valve	DN20	graduate	Max scale: 1000ml
Pressure transducer	Precision: 1Pa	Timer	—
Thermometer	0-100°C, Precision: 0.1°C	Heat insulation material	—
90°elbow	DN20	Kerosene	$0.5m^3$
Straight tee	DN20	glycol solution	Concentration: 35%, $0.5m^3$
Straight joint	DN20	glycol solution	Concentration: 90%, $0.5m^3$

Table 2. Physical dimensions of spiral tubes used in experiment

NO.		r (mm)	R_c (mm)	r/R_c	Torsion τ	Pitch S (mm)
Straight tube	#1	19	+∞	0	0	0
Spiral tube	#2	19	38	0.25	4	38
	#3	19	38	0.25	5.9	57
	#4	19	38	0.25	7.6	76
	#5	19	38	0.25	9	95
	#6	19	57	0.167	4	86
	#7	19	76	0.125	4	162
	#8	19	95	0.1	4	189

Note: The torsion is applied here to evaluate the pitch of a spiral tube according to the correlation $\tau=(S/2\pi)/[R_c^2+(S/2\pi)^2]$, where S and R_c mean the pitch and curvature of a spiral tube, respectively [22].

As for the test tubes used in the experiment, 7 different kinds of spiral tubes and 1 straight tube were included. The parameters of these tubes are listed in Table 2. The straight tube has a length of 750mm, a inner diameter of 19mm, and wall thickness of 0.5mm. And all the spiral tubes are constructed with such a straight tube. The spiral tubes differ from each other in the curvatures and pitches(turns). However, the dimensionless parameter is preferred during the research, therefore the curvature and pitch are replaced by curvature ratio and torsion, respectively. As a result, there are 4 kinds of curvature ratios varying from 0.1 to 0.25 and 4 kinds of torsions changing from 4 to 9. Among them, the curvature ratio is closely relevant to the motion of the Dean vortices. And the torsion, another important parameter of spiral tube, varies with the curvature and pitch of spiral tubes.

2.2 Experimental procedure

The fluids were pumped to the spiral tubes using a centrifugal pump at an initial inlet temperature of 10 °C and different flow rate(that means different velocities or different Reynolds numbers). During the experiment, the fluid was taken from the water container which is made of plexiglass covered by heat insulation material(to minimize the influence of radiation heat loss), and was heated by the hot water in the water tank while passing through spiral tube, then flew out of the system as shown in Figure 1. The inlet and outlet temperatures of the spiral tubes were measured by digital thermometers calibrated to meet limits of error ±0.1°C. Similarly, the inlet and outlet pressure was obtained by the pressure transducers installed at the inlet and outlet of the test tube. In addition, the data acquisition started after the system had run for about 15 minutes, because it had been presumed that it would take about 15 minutes to reach a steady condition for a certain flow rate [23].

The exact procedure is detailed as follows: (1) Water was chosen as the fluid in the experiment with a initial temperature of 10 °C, and was pumped to the test tubes at different flow rate under which the Reynolds number ranges from 760 to 19000. The pressure and temperature data at the inlet and outlet of tubes can be recorded when the pressure and temperature values measured stop fluctuating greatly. The flow rate in the experiment and the corresponding Reynolds numbers are listed in Table 3. (2) Water was chosen again as the test fluid to continue the experiment with spiral tubes at the Reynolds number of 4000 and an initial inlet temperature of 10 °C, which suggests an experiment investigating into the influences of the curvature ratio and torsion on the heat transfer characteristic of the spiral tubes, at constant flow rate and viscosity. (3) Water was replaced by the fluids with higher viscosities to continue the experiment, at a flow rate of 58.8mL/s. This part of experiment was carried out in the straight tube and any one of the spiral tube(the spiral tube with curvature ratio of 0.25 and torsion of 4 was chosen in the experiment). The fluids with higher viscosities used in the experiment include the Kerosene, 35% glycol solution and 90% glycol solution, the dynamic viscosity of which at a temperature of 10 °C are 2.4mPa.s, 13.5mPa.s and 20.2mPa.s, respectively.

Table 3. Flow rate and Reynolds numbers of the experiment

Test tube	#1 to #8										
Flow rate(ml/s)	11.2	22.4	33.6	42.0	58.8	89.6	117.6	162.4	207.2	224	280
Mean velocity(m/s)	0.04	0.08	0.12	0.15	0.21	0.32	0.42	0.58	0.74	0.8	1
Re	760	1520	2280	2850	4000	6000	8000	11000	14000	15200	19000

2.3 Calculation of the experimental data

In the experiment, thermo-physical properties of test fluids flowing through the test tubes are assumed constant along the tube length and evaluated at an average bulk temperature. Heat flux from the hot water in the water tank to the cold water through the tube is calculated from:

$$q = \frac{c_p \cdot m \cdot \left(T_{out} - T_{in}\right)}{\pi \cdot d \cdot L} \tag{2}$$

The heat flux calculated from above equation was then used to calculate the overall heat transfer coefficient, α, as

$$\alpha = \frac{q}{T_{wall} - T_{fluid}} \tag{3}$$

The symbol T_{wall} denotes the wall temperature(°C) of the test tubes, which is assumed to be equal to the temperature of the water in the water tank, 90 °C. And T_{fluid} indicates the average fluid temperature in the test tubes, which is assumed to be determined by $T_{fluid}=0.5(T_{in}+T_{out})$ due to the limited length of the test tubes.

During the calculation and latter discussion, the dimensionless parameter Nusselt number, rather than the heat transfer coefficient, is chosen to express the heat transfer characteristic. The Nusselt number can be calculated from:

$$Nu = \frac{\alpha \cdot d}{\lambda} \tag{4}$$

λ is the thermal conductivity(W/m. °C) of the fluid flowing through the test tubes.

3. Results and discussion

3.1 Effects of Reynolds number on heat transfer for test tubes

Plenty of research has revealed that the heat transfer performance can be improved by increasing the Reynolds number, for both straight and curved tubes. In this study, experiment on straight tube and spiral tubes was carried out at a series of Reynolds numbers, and finally the results were compared.

Figure 2 depicts the heat transfer characteristic of test tubes under different Reynolds numbers. It is clear that the Nusselt number increases with Reynolds number, for all the test tubes. Another result that can be achieved is that the Nusselt number for spiral tubes is higher than that for straight tube under the same Reynolds number. For example, the Nusselt number for the spiral tube, with a curvature ratio of 0.25 and torsion of 4, is 39.4% higher than that for straight tube at the Reynolds number of 19000 (Table 4). In addition, the Nusselt number in spiral tubes increases faster than that of the straight tube for Reynolds number larger than 15000. Figure 3 demonstrates the changes of pressure drop of test tubes at different Reynolds numbers. It can be observed that the pressure drop for all the test tubes increases with the Reynolds number, but the pressure drop for spiral tubes is higher than that for straight tube. What's more, the spiral tube with different curvature ratios and different torsions exhibits different pressure drop, which will be discussed in 3.2 and 3.3.

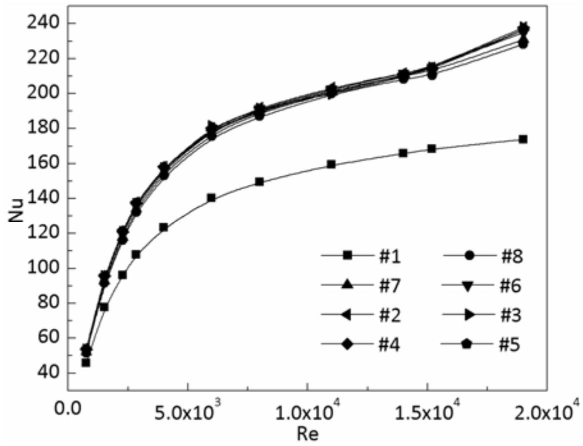

Figure 2. Nu as a function of Re

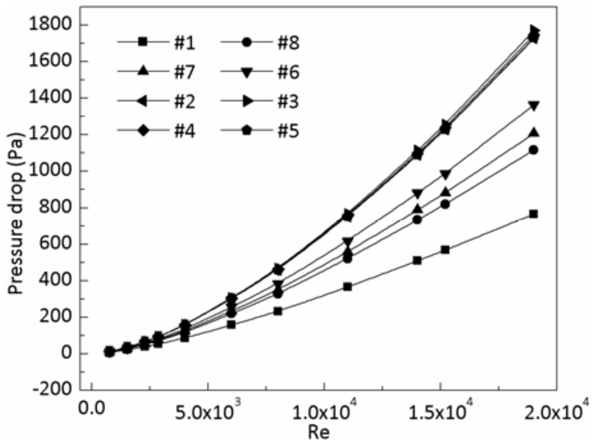

Figure 3. Pressure drop as a function of Re

Table 4. Percentage of increment in Nu compared with #1 at Re=19000

Test tube	#1	#8	#7	#6	#2	#3	#4	#5
Nu	170.7	228.2	231	235.1	237.9	238	237.4	237.5
increment	-	33.7%	35.3%	37.7%	39.4%	39.4%	39.1%	39.1%

For the straight tube, the Nusselt number and pressure drop increase with Reynolds number because the turbulence intensity of the fluid in the tube increases at high Reynolds number, thus improves the mixing effect and causes a penalty in pressure drop. However, the fluid field in the spiral tubes will not go into "chaos" due to the existence of the "subsidiary flow with certain flow pattern"(the Dean vortices), although there is an increase in Reynolds number. It is implied that an increase in Reynolds number under experiment conditions can only increase the intensity of the Dean vortices, while not destroy the vortices(However, perhaps there will be no difference in Nusselt number when the Reynolds number increases to a certain value). As result, the forced convection in the spiral tubes is improved. As for the pressure drop of the spiral tubes, the profile drag dominates. Therefore the pressure drop is closely related to the geometry factors such as curvature ratio.

3.2 Effects of curvature ratio on heat transfer of spiral tubes

Experiment was performed to investigate the influence of the curvature ratio at a certain Reynolds number. Figure 4 illustrates the variation of Nusselt number versus curvature ratio for spiral tubes (#2, #6, #7 and #8) with curvature ratios ranging from 0.1 to 0.25 and a torsion of 4. From this figure, it can be seen that the Nusselt number increases when the curvature ratio varies from 0.1 to 0.25. There is a similar tendency for the pressure drop as is shown in Figure 5.

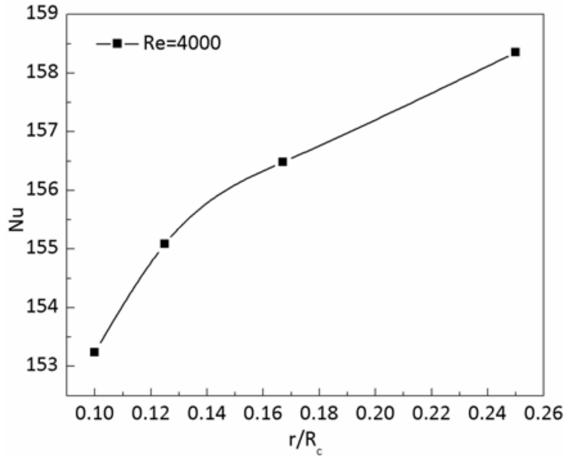

Figure 4. Nu as a function of r/R_c

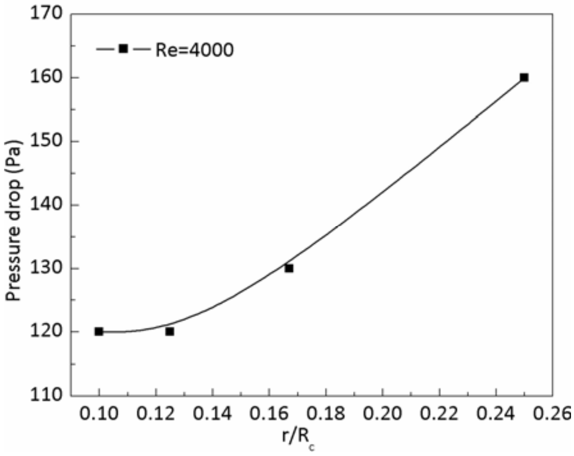

Figure 5. Pressure drop as function of r/R_c

For a spiral tube with a certain diameter, a greater curvature ratio means that the curvature is small, therefore the spiral tube has more turns than spiral tubes with smaller curvature ratios. As a result, fluid flowing through the tube has to change its main flow direction more times. Therefore the forced convection provided by Dean vortices works longer, but it inevitably increases the pressure drop.

3.3 Effects of torsion on heat transfer for spiral tubes

The spiral tubes with torsions ranging from 4 to 9 and a curvature ratio of 0.25 are used in this part of experiment. The effect of torsion on heat transfer enhancement characteristic of the spiral tubes is depicted in Figure 6 and Figure 7.

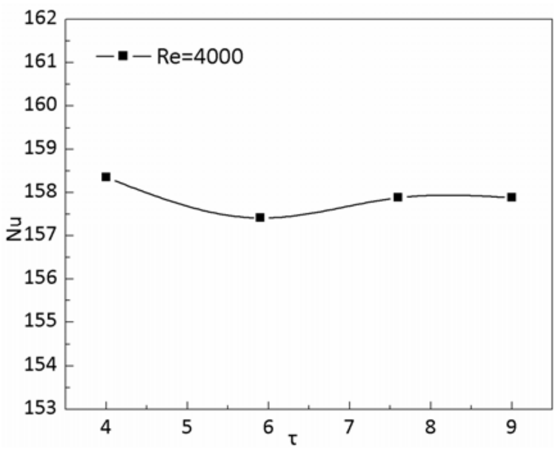

Figure 6. Nu as a function of torsion

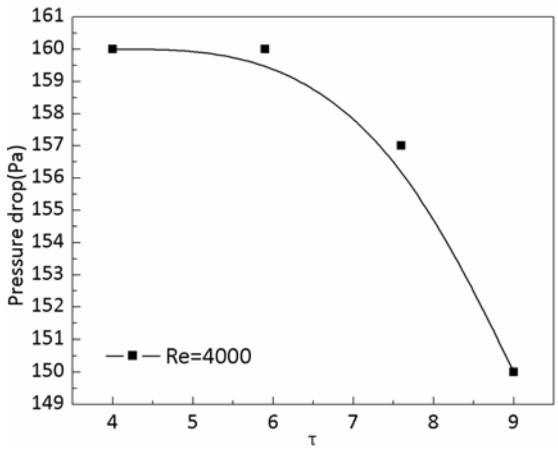

Figure 7. Pressure drop as function of torsion

The figure shows that the Nusselt numbers are fluctuating between 157 and 159, which suggests that the torsion has an insignificant influence on Nusselt number for spiral tubes. As for the influence of torsion on pressure drop, Figure 7 clearly illustrates that the pressure drop decreases from 160Pa to 150Pa as the torsion increases.

3.4 Effects of viscosity on heat transfer for test tubes

In this part of experiment, Kerosene, 35% glycol solution and 90% glycol solution are used, the dynamic viscosities of which are 2.4mPa.s, 13.5mPa.s and 20.2mPa.s respectively, at a temperature of 10 °C. However, the dimensionless Prandtl number is employed instead of viscosity to indicate the fluid viscosity in the discussion. The Prandtl numbers of Kerosene, 35% glycol solution (35% glycol and 65% water, in volume) and 90% glycol solution(90% glycol and 10% water, in volume) are presumed to be 34, 66 and 150 respectively at 10 °C.

Figure 8 shows the variation of Nusselt number with Prandtl number in test tube #1 and #2, at flow rate of 58.8ml/s. As indicated in the figure, the Nusselt number tends to increase with the rise of Prandtl number for all cases. Meanwhile, the Nusselt number for #2 is higher than that for #1 under the same Prandtl number. It can be concluded that the spiral tube exhibits a better heat transfer performance than straight tube when the fluid has a high viscosity. The influence of the Prandtl number on pressure drop of the tube is shown in Figure 9. As indicated in the figure, the pressure drop of #1 and #2 increases with the rise of Prandtl number. It is also found that the pressure drop for #2 is higher than that for #1 under the same Prandtl number.

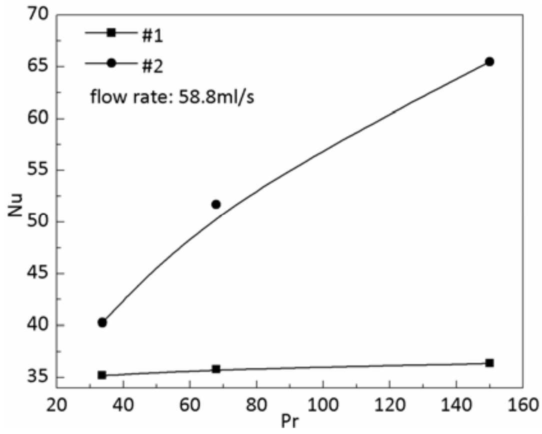

Figure 8. Nu as a function of Pr

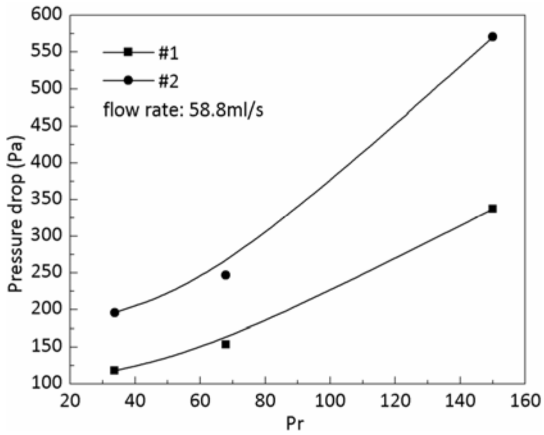

Figure 9. Pressure drop as a function of Pr

The spiral tubes in this part of experiment are characteristic of a higher Nusselt number due to the inertial force(centrifugal force) that drives the Dean vortices. In the spiral tube, the inertial force dominates, while in the straight force the reverse is true(which means that the viscous force dominates). Therefore, the heat transfer between the center and the boundary layer continues in the spiral tube even with a high viscosity fluid. Whereas in the straight tube, the heat exchange between the fluid in boundary layer and the core of the tube is inadequate. As for the pressure drop, the profile drag plays an important role, therefore the spiral tube exhibits higher pressure drop than straight ones.

4. Correlation equation of heat transfer enhancement by Dean vortices

4.1 Effects on Dean number on heat transfer of spiral tubes

The results above revealed that the heat transfer performance of spiral tubes is influenced by Reynolds number, curvature ratio and viscosity of the fluid. According to Eq.(1), the Dean number is a function of the Reynolds number and curvature ratio of the spiral tube. As a result, it can be predicted that the heat transfer performance is influenced by Dean numbers.

Based on the results above, the Dean numbers are calculated with different Reynolds numbers and curvature ratios. Then the Nusselt numbers and pressure drop obtained in the experiment are matched with the Dean numbers. Figure 10 demonstrates the variation of Nusselt number under different Dean numbers. From the figure it can be seen that the Nusselt numbers is approximately in parabolic increase with the Dean numbers, which means an improved heat transfer performance. As to the changes in pressure drop under different Dean numbers, Figure 11 provides a clear depiction. As is shown in the figure, the pressure drop increases with the Dean numbers.

However, there is a lack of evidence to judge whether a spiral tube with Dean vortices has a better heat transfer performance than straight tube, by comparing only the Nusselt numbers and pressure drop. Therefore, the thermal enhancement factor, η, is introduced to better evaluate the heat transfer performance of the spiral tubes($\eta=(Nu/Nu_{ref})(f/f_{ref})^{-1/3}$) [24]. In this literature, the straight test tube is regarded as the reference tube. A value of η greater than 1 indicates better performance of heat transfer. Figure 12 illustrates the variation of η versus Dean number for all the spiral tubes in the experiment.

The figure shows that spiral tubes perform better than straight tubes. The comprehensive heat transfer performance of spiral tube improves with increasing Dean number before the Dean number reaches approximately 2000. When Dean numbers are greater than 2000, the value of η starts to decrease, but still greater than 1. It implies that the spiral tube with Dean vortices has a better heat transfer enhancement performance than straight tubes, especially when the Dean number is approaching 2000. (The pressure drop increases much faster when Dean number is great than 2000, which decrease the comprehensive heat transfer performance, although the Nusselt number keeps increasing).

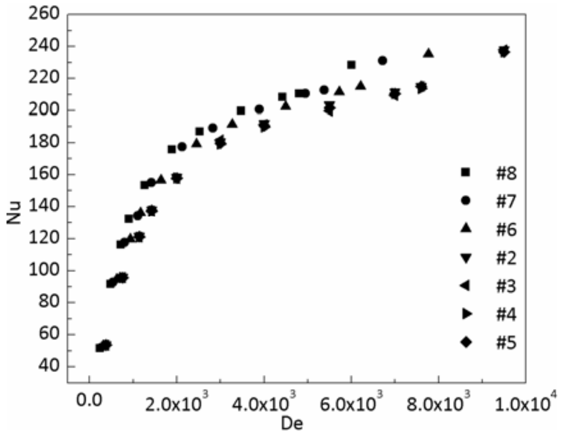

Figure 10. Nu as a function of De

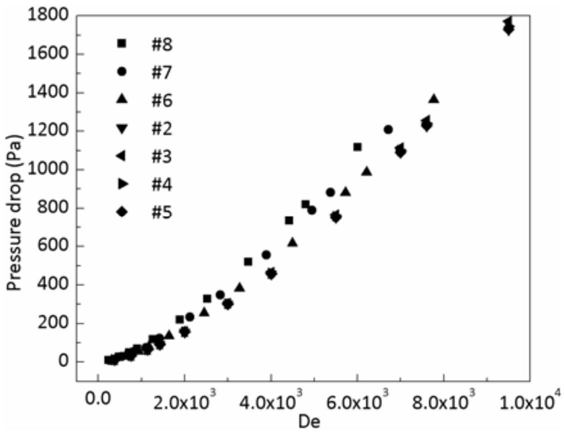

Figure 11. Pressure drop as function of De

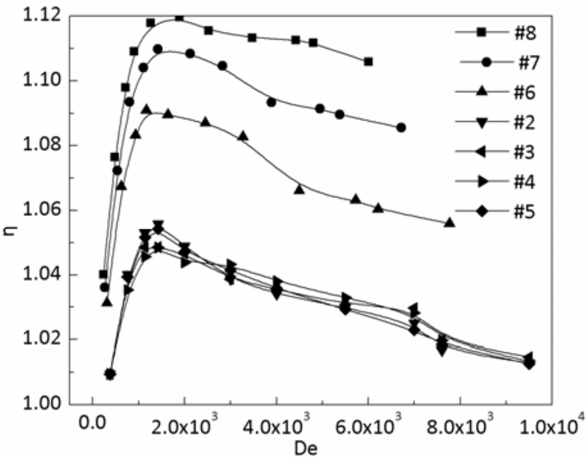

Figure 12. The value of η as a function of De

4.2 Deduction of the correlation equation

As mentioned in 4.1, the heat transfer enhancement performance is closely relevant to the Dean number and the viscosity that is indicated by dimensionless Prandtl number. As a result, the Nusselt number can be expressed as:

$$Nu \propto f(De \cdot \mathrm{Pr}) \tag{5}$$

Meanwhile, the heat transfer enhancement is achieved by forced convection caused by Dean vortices. Therefore, the mathematic model of forced convection in tubes can be assumed to be:

$$Nu = C \cdot De^{m} \cdot \mathrm{Pr}^{n} \tag{6}$$

where C, m and n are the undetermined constants which is influenced by the flow and heat transfer characteristics.

According to Eq.(6), multiple regression was performed using STATISTICA with the data obtained from the experiment. The results can be expressed as the following equations:

$$Nu = 5.3611 \cdot De^{0.553637} \cdot \mathrm{Pr}^{0.497215} \tag{7}$$

$$\Delta P = 0.003 \cdot De^{1.503735} \cdot \mathrm{Pr}^{0.118736} \tag{8}$$

Eq.(7) and Eq.(8) are believed to be effective when Dean numbers ranges from 240 to 9500, and Prandtl number ranges from 9 to 150, and the correlation coefficient for Eq.(7) and Eq.(8) are 0.9737 and 0.9825, respectively (9 denotes the Prandtl number of water at 10 °C).

5. Conclusion

The heat transfer enhancement characteristics for spiral tubes were experimentally investigated in terms of Dean vortices. From the data obtained from the experiment, it is found that the Dean vortices induced by centrifugal force in the spiral tube plays an significant role in the heat transfer enhancement. A few important conclusions are stated below.

Results depict that both the Nu and pressure drop increase with Re, and pressure drop in all spiral tubes is higher than that of the straight tube. In addition, Nu and pressure drop for spiral tubes increase with the curvature ratio at constant flow rate, viscosity and torsion. And the torsion has been found to be ineffective to heat transfer performance, but it reduces the pressure drop of the flow in spiral tubes. What's more, it was observed that the Nu for both spiral tube and straight tube increase with the Prandtl number and it was also found that the Nu for the spiral tube was much greater, which suggests that the spiral tube has a better heat transfer performance even under high viscosity fluid. Finally, considering the influence of Re, curvature ratio and Pr simultaneously, the effect of Dean number on heat transfer was investigated. The result reveals that both Nu and pressure drop increase with De. And what deserves attention is that η is greater than 1 for all the spiral tubes and reaches its top value when De approximates 2000. Based on the results above, correlation equations were obtained via multiple regression, which are believed to be effective when Dean number ranges from 240 to 9500 and Prandtl number ranges from 9 to 150.

Acknowledgements

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Nomenclature

De	Dean number
Re	Reynolds number
r	inner radius of the tube, m
d	inner diameter of the tube, m
L	length of the tube, m
R_c	curvature of spiral tube, m
S	pitch of spiral tube, m
V	main velocity of the fluid, m/s
q	heat flux, W/m2
c_p	specific heat capacity, J/(kg. °C)
m	mass flow rate, kg/s
T	temperature, °C
Nu	Nusselt number
Pr	Prandtl number
f	drag coefficient
ΔP	pressure drop, Pa

Greek symbols

υ	kinetic viscosity, m^2/s
α	heat transfer coefficient ,W/(m^2. °C)
λ	thermal conductivity, W/(m. °C)
η	comprehensive evaluation factor
τ	torsion of spiral tube

Subscripts

in	inlet
out	outlet
wall	tube wall
fluid	fluid
ref	reference tube

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