STUDY ON HEAT TRANSFER ENHANCEMENT BY USING SPIRALLY CORRUGATED TUBES

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To study enhancement in heat transfer by corrugated tubes, one smooth tube and seven kinds of three-starts spirally corrugated tubes which had different combinations of the depth and pitch of corrugation were used in a double-pipe heat exchanger. The friction factors and heat transfer coefficients of the tube inside and annulus were measured, and the contribution of the corrugated tubes to the enhancement in heat transfer was examined.

The spirally corrugated tubes provided a secondary spiral fluid motion near the tube wall. The effects of the geometrical parameters were expressed by the tube severity factor which is a function of the depth and pitch of corrugation and the tube diameter. With increasing severity factor, both the friction factors and heat transfer coefficients of the tube inside and annulus increased. The contribution of the corrugated tubes was examined by evaluating the overall performance which was defined as the ratio of the enhancement in heat transfer to the increase in friction factor. A maximum in overall performance appeared at a relatively low severity factor.

Introduction

During the last thirty years, the need to conserve energy and materials has stimulated research into various techniques and methods to enhance heat transfer. The enhancement of convective heat transfer in tubes is the major key towards high performance of heat exchangers or condensers, some of the most important pieces of process equipment for a great number of industries such as the chemical, petrochemical, thermal and nuclear power, waste treatment industries. The main aim in the design of heat exchanger is to obtain higher heat transfer rates by using techniques for getting enhanced heat transfer surfaces which provide an enhancement in heat transfer coefficient over smooth tubes. Artificially roughened surfaces (e.g. repeated-rib tubes and spirally corrugated tubes), extended surfaces (different types of internally finned tubes, and displaced devices (twisted tape inserts, mesh and brush inserts, spiral spring inserts, and flag-type inserts) are a few examples of passive enhancement techniques¹.

There have been several studies that used spirally corrugated tubes to enhance the heat transfer. Most of them^{7,8,10-12,18} used single-start spirally corrugated tubes and the others^{6,16} used multi-starts spirally corrugated tubes. Their results showed that spirally corrugated tubes were very effective in enhancing the heat transfer coefficient, by 1.12 to 3.28 times depending on the test fluid, at constant pumping power or constant flow condition. These previous studies on smooth and spirally corrugated tubes revealed considerable information on the characteristics of turbulent flow friction factor and heat transfer of Newtonian and non-Newtonian fluids but all of them were

restricted to study of the tube inside.

The present study is undertaken with a view to investigate the performance of three-starts spirally corrugated tubes. The aims are (1) to evaluate the increase in friction factors of both the tube inside and outside, (2) to evaluate the enhancement in heat transfer coefficients of both the tube inside and outside and (3) to find the optimum tube configuration, based on both the friction factor and heat transfer performance.

1. Experimental

In the present study, experiments were carried out using one smooth tube and seven kinds of three-starts spirally corrugated tubes of different geometries as the inner tube of a double-pipe heat exchanger. Water was used as the test fluid. The smooth tube was used to standardize the experimental set-up and also to compare the increase in friction factors and the enhancement in heat transfer coefficients.

1.1 Tube configuration

All tested tubes were fabricated from plain copper tube with a 16.60 mm I. D. and 19.00 mm O. D. The spirally corrugated tubes were spiraled along their active lengths, all of them having three external helical starts. The fabrication technique embosses an internal projection, also known as a ridge, in registration with the external helical groove. This configuration was attainable without thinning the tube wall at the focus of the corrugation. The external helical groove is semicircular in cross section. A section of the three-starts spirally corrugated tubes used is shown in **Fig. 1**. In the case of multi-starts spirally corrugated tubes, the corrugation pitch is defined as the

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Fig. 1 Sectional diagram of three-starts spirally corrugated tube

Table 1. Dimensions and characteristics of smooth tube and seven spirally corrugated tubes

No.	Tube	D_o [mm]	D_i [mm]	e [mm]	w [mm]	p [mm]	p' [mm]	e/D_i [-]	p/e [-]	$\theta \times 10^3$ Key [-]
1	S	19.00	16.60	0.00	0.00	∞	∞	0.00	∞	0.00
2	C-1	19.00	16.60	0.50	2.46	60.0	20.0	0.03	40.0	0.75
3	C-2	19.00	16.60	0.75	2.84	75.0	25.0	0.04	33.3	1.36
4	C-3	19.00	16.60	0.75	2.83	60.0	20.0	0.04	26.7	1.69
5	C-4	19.00	16.60	0.75	3.14	45.0	15.0	0.04	20.0	2.26
6	C-5	19.00	16.60	0.75	3.26	30.0	10.0	0.04	13.3	3.39
7	C-6	19.00	16.60	1.50	4.37	60.0	20.0	0.09	13.3	6.78
8	C-7	19.00	16.60	2.00	6.52	60.0	20.0	0.12	10.0	12.1

axial distance between two adjacent grooves or as the axial distance through which one ridge makes a 360° turn along the tube circumference. The effective corrugation pitch is defined as the corrugation pitch divided by the number of external helical starts.

Dimensions and characteristics of the smooth and spirally corrugated tubes are listed in **Table 1**. The spirally corrugated tubes have different combinations of corrugation depth, e , corrugation width, w , corrugation pitch, p , and effective corrugation pitch, p' . The main geometrical parameters of the multi-starts spirally corrugated tubes are e , p' and the tube inside diameter, D_i . The relationship between these parameters is referred to as the tube severity factor, which expresses the degree or severity of the tube roughness. The tube severity factor^{7,8,12}, θ , is defined as :

$$\theta = e^2 / p'D_i \quad (1)$$

Gupta⁷ and Gupta and Raja Rao⁸) have used the tube severity factor to correlate with success their data for heating operation and turbulent flow of Newtonian and pseudoplastic non-Newtonian fluids in single-start spirally corrugated tubes.

1.2 Apparatus and procedure

A schematic diagram of the experimental apparatus is shown in **Fig. 2**. The actual test section consisted of a double-pipe heat exchanger 1510 mm in length. The inner tube first used was a smooth tube. This was replaced by spirally corrugated tubes in subsequent experiments. The outer tube was a 35.00 mm I. D. and 44.00 mm O. D. acrylic resin tube. The inner tube of the test section was preceded by an inlet calming section of 600 mm in length and was followed by a mixer which was used for measuring the outlet mixing-cup temperature. The inlet and outlet

Fig. 2 Schematic diagram of experimental apparatus

of both the inner tube and annulus were equipped with 320- μ m K-type thermocouples. The test section, the inlet calming section, the mixer and all the flow lines of the hot water were insulated with a 10.0 mm layer of polyurethane resin, except during the observation of flow pattern in the annulus. Static pressure taps were equipped at the inlet and outlet of the inner tube, and near the middle point* and at the outlet of the annulus.

For each experimental run the system was allowed to reach steady state conditions. Both the flow rates of fluids in the inner tube and annulus were independently controlled to give rates ranging between 0.07-0.42 and 0.07-0.97 kg s⁻¹, respectively. The inlet temperatures of fluids in the inner tube and annulus were fixed to 323 $\pm$ 0.1 and 293 $\pm$ 0.1 K respectively. After the steady state conditions were confirmed**, pressure drops and temperatures were recorded.

Friction factors for the inner tube and annulus were calculated from the pressure drops using the Fanning equation. From the steady state heat balance, the overall heat transfer coefficients based on the inside surface area, U_i were calculated. U_i was represented in terms of the individual convective heat transfer coefficients, h_i and h_o by the following equation :

$$1 / U_i A_i = 1 / h_i A_i + R_w + 1 / h_o A_o \quad (2)$$

where R_w is the wall resistance per unit area, and A_i and A_o were the inside and outside surface areas of the smooth tube respectively. The surface areas of the smooth tube were used also in the case of the spirally corrugated tubes because the surface areas were considered to be unchanged by the present fabrication method as described above. By using Eq. (2) and neglecting the tube wall resistance, it was observed how h_i changed when h_o was constant, and vice-versa. h_i and h_o were determined from

* Preliminary experiments for pressure drop of the annulus using the smooth tube showed the need for a hydrodynamic entrance length. As a result, the annulus of the heat exchanger was fitted with a hydrodynamic entrance length of about 35 hydraulic equivalent diameters of the annulus^{14,15}.

**The steady state condition was indicated by the constancy of the flow rates and thermocouple readings.

Fig. 3 Photographs of flow pattern ($Re_o = 1250$, (a) C-3, (b) C-6)

Fig. 4 Plot of tube inside friction factors against tube inside Reynolds number for smooth and spirally corrugated tubes

experimental data by using Wilson's method.

2. Experimental Results and Discussion

2.1 Flow pattern

Flow pattern observations were carried out only inside the annulus of the heat exchanger and in conditions of laminar flow, i.e. at low Reynolds numbers. Examples of photographs of flow patterns are shown in Fig. 3. The fluid flow is observed by using an ink tracer. When the spirally corrugated tubes were used, the flow is partly spiral very near the tube wall and mainly a streamline flow at a distance from the wall. The extent of spiral motion increases with increasing degree of tube roughness, i.e. tube severity factor θ , as clearly seen by a comparison of Fig. 3(a) ($\theta = 1.69 \times 10^{-3}$) and Fig. 3(b) ($\theta = 6.78 \times 10^{-3}$). Spiral motion and main axial flow might also occur inside the spirally corrugated tubes. These secondary spiral motions are considered to cause an increase in velocities near both sides of the tube wall leading to a reduction in thickness of the laminar sublayers adjacent to the tube wall.

Fig. 5 Plot of tube outside friction factors against annulus inside Reynolds number for smooth and spirally corrugated tubes

2.2 Friction factors

Plots of the tube inside friction factors of the smooth and spirally corrugated tubes, f_{is} and f_{ic} , against the tube inside Reynolds number, Re_i , are shown in Fig. 4. Over Re_i values between 9000 and 67000, f_{is} is in good agreement with the Blasius equation²⁾. This serves the purpose of standardization of the experimental set-up. Both f_{is} and f_{ic} decrease with increasing Re_i . f_{ic} is a function of θ and significantly increases with increasing θ . Unlike for other rough tubes¹⁷⁾, the dependence of f_{ic} on Re_i decreases with increasing θ . This is because for the present spirally corrugated tubes the thickness of laminar sublayer becomes very small compared to roughness irregularities and the laminar sublayer is broken up into turbulence¹²⁾. As a result the roughness irregularities (the corrugation depth) result in increases in turbulence near the tube wall and in flow resistance. Beyond a certain value of corrugation depth, the effect of roughness irregularities is so considerable that viscous force becomes negligible compared to inertia forces which are caused by the fluid flowing over projections. In such a case, the friction factors depend only on relative roughness. This suggests that the increase in f_{ic} is not only due to the inertia effect but also due to the different fluid flow pattern. Similar observations have also been made by Gupta⁷⁾, Gupta and Raja Rao⁸⁾, and Mehta and Raja Rao¹²⁾. In the present range of Re_i , f_{ic} is found to be 1.08 to 3.13 times higher than f_{is} .

Plots of the tube outside friction factors f_{os} and f_{oc} against the annulus inside Reynolds number, Re_o , are shown in Fig. 5. Re_o is calculated using the hydraulic equivalent diameter of the annulus (D_{eq}). In the region of laminar flow, f_{os} is a little higher than the results of the Jones and Leung⁹⁾ correlation, but in the region of turbulent flow, f_{os} almost lies between correlations proposed by Brighton and Jones³⁾ and Davis⁴⁾. Here, in the Jones and Leung⁹⁾ correlation, ϕ is the geometry function which is a function of

Fig. 6 Plot of tube inside heat transfer coefficients against tube inside Reynolds number for smooth and spirally corrugated tubes

D_o/d_i . These correlation serve the purpose of standardizing the experimental set-up. Both f_{os} and f_{oc} sharply decrease with increasing Re_o , f_{oc} increases with increasing θ . This is due to roughness irregularities which give an increase in turbulence outside the tube wall. In the present range of Re_o , f_{oc} is found to be 1.04 to 2.00 times higher than f_{os} .

Based on the Blasius equation²⁾, the following equation for f_{ic} was derived:

$$f_i = 0.079 \exp(-64\theta) Re_i^{-(0.25 \exp(-99\theta))} \quad (3)$$

where, $0 \leq \theta \leq 12.1 \times 10^{-3}$ and $9000 \leq Re_i \leq 60000$. The results of Eq. (3) are also shown in Fig. 4 by the dotted lines. It can be seen that the obtained correlation agrees reasonably well with the experimental data and the deviation of Eq. (3) is $\pm 15\%$ for all the data. We also tried to correlate our data by the additional type of equation, ($f_i/f_{is} = 1 + 78 \theta^{0.83}$, $f_{is} = 0.079 Re_i^{-0.25}$), but its deviation is $\pm 20\%$. From this, it is considered that the effect of θ on the friction factor is dependent on Re_i , i.e. the flow pattern. On the other hand, it was difficult to correlate f_{oc} as simply as f_{ic} . This is because the flow mode changes under the present experimental conditions as shown in Fig. 5.

2.3 Heat transfer coefficients

Plots of the tube inside heat transfer coefficients, h_{is} and h_{ic} , against Re_i are shown in Fig. 6. Over the range of Reynolds number 9000-67000, h_{is} is in good agreement with the Dittus-Boelter equation⁵⁾ for cooling operation. h_{is} and h_{ic} increase with increasing Re_i . Here, h_{ic} is a function of θ and h_{ic} considerably increases with increasing θ . This is because with increasing θ the spiral motion of the fluid flowing over the projections increases and the thickness of the boundary layer decreases. h_{ic} is found to be 1.02 to 3.69 times higher than h_{is} .

Plots of the tube outside heat transfer coefficients, h_{os} and h_{oc} , against Re_o are shown in Fig. 7. In the region of turbulent flow, h_{os} is a little higher than the results of the

Fig. 7 Plot of tube outside heat transfer coefficients against annulus inside Reynolds number for smooth and spirally corrugated tubes

Monrad and Pelton¹³⁾ correlation. Also h_{os} and h_{oc} are found to increase with increasing Re_o . h_{oc} increases with increasing θ only at lower Re_o (laminar flow region) but tends to be the same as that of the smooth concentric annulus at higher Re_o . This is because with increasing θ and Re_o , the enhancement provided by the spiral motion decreases inside the annuli, which means that for the present type of annuli* these spirally corrugated tubes are more effective in the laminar flow region. In the present range of Re_o , 1600-23000, h_{oc} is found to be 1.00 to 2.63 times higher than h_o .

For h_{ic} , the following equation was derived based on the Dittus-Boelter equation⁵⁾:

$$\begin{aligned} h_i / h_{is} &= 1 + 71 \theta^{0.74} \\ h_{is} &= 0.023 Re_i^{0.8} Pr_i^{0.3} k / D_i \end{aligned} \quad (4)$$

where $0 \leq \theta \leq 12.1 \times 10^{-3}$ and $9000 \leq Re_i \leq 60000$. The results of Eq. (4) are also shown in Fig. 6 by the dotted lines. The results show good agreement with the experimental data and the deviation of Eq. (4) is $\pm 25\%$ for all the data. We also tried to correlate our data with an equation of the same type as Eq. (3) but its deviation is $\pm 35\%$. For heat transfer, the contribution of θ is considered to be scarcely affected by the flow pattern. For h_{oc} it was difficult to correlate the experimental data with an equation of similar type to Eq. (4).

2.4 Evaluation of tube configuration

In the present study, the effects of the corrugation depth, e , effective corrugation pitch, p' , and tube severity factor, θ , are examined by evaluating the overall performances of both the inner tube and annulus, defined as enhancement ratios, by the following equations:

$$\begin{aligned} \eta_i &= (h_{ic} / h_{is}) / f_{ic} / f_{is} \\ \eta_o &= (h_{oc} / h_{os}) / f_{oc} / f_{os} \end{aligned} \quad (5)$$

The effect of e on η_i and η_o is shown in Fig. 8. The

* One wall of the annuli was a rough surface and the other a smooth surface.

Fig. 8 Effect of corrugation depth on overall performances of inner tube and annulus ($p' = 20.0$ mm)

Fig. 9 Effect of corrugation pitch on overall performances of inner tube and annulus ($e = 0.75$ mm)

data of the same p' ($=20.0$ mm) are plotted. It is found that η_i increases with increasing Re_i and that η_i has a maximum for e of 0.75 mm. Spirally corrugated tube C-3 ($e = 0.75$ mm) gives the best overall performance, ranging from 1.13 to 1.26. For η_o the maximum appears for a little smaller e ($=0.50$ mm). However, the trend in Re_o is the opposite and η_o decreases with increasing Re_o . This is because, as seen in Fig. 7, the contribution spiral motion to the enhancement in h_o becomes less at high Re_o . From Fig. 8 it is seen that too large an e is not effective for performance improvement.

The effect of p' on η_i and η_o is shown in Fig. 9. Data of the same e ($=0.75$ mm) are plotted. First η_i increases with increasing p' until a value of about 15 mm and then decreases beyond this value. Spirally corrugated tubes C-4 ($p' = 15.0$ mm) and C-5 ($p' = 10.0$ mm) give the best overall performances, ranging from 1.18 to 1.31. In the case

Fig. 10 Effect of tube severity factor on overall performance of inner tube

Fig. 11 Effect of tube severity factor on overall performance of annulus

of η_o any maximum is not seen within the present experimental conditions. However, from the fact that the smooth tube corresponds to the case of infinite p' , it is considered that η_o has a maximum for a larger p' .

The effect of θ on η_i and η_o is shown in Figs. 10 and 11 respectively. Figure 10 shows that η_i has a maximum for θ of $2.26 - 3.39 \times 10^{-3}$. The maximum value of η_i increases with increasing Re_i , from 1.18 to 1.31. Also, the data of Gupta⁷⁾ are plotted in Fig. 10 for comparison. His data show the same tendency while the maximum appears for a little lower θ . For η_o , Fig. 11 shows the same trend as that in Fig. 10 but the value of θ which gives a maximum becomes lower and the peak is sharper than those of η_i . When θ is greater than 2.5×10^{-3} , η_o gradually decreases with θ and, especially at high Re_o , η_o is below unity. That is, too higher θ has a negative contribution at high Re_o .

Conclusion

Using seven kinds of three-starts spirally corrugated tubes, the flow pattern was observed and the friction factor and heat transfer coefficient were measured. The effects of geometrical parameters were examined by evaluating overall performances. The following conclusions were obtained :

1. The spirally corrugated tubes provide a secondary spiral motion of fluid and the degree increases with increasing θ .

2. Both f_{ic} and f_{oc} increase with increasing θ and decrease with increasing Re_i or Re_o . The dependence of f_{ic} on Re_i decreases with increasing θ , while such a tendency is not seen for f_{oc} .

3. h_{ic} increases with increasing θ and Re_i , but h_{oc} increases with increasing θ only at lower Re_o and tends to be the same as h_{os} at higher Re_o .

4. Both η_i and η_o have maxima in terms of e . The effects of p' on η_i and η_o are different. The effects of geometrical parameters are expressed by θ , and both η_i and η_o have a maxima at θ values of $2.26 - 3.39 \times 10^{-3}$ and $0.75 - 1.36 \times 10^{-3}$, respectively.

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Nomenclature

A	= surface area	[m ²]
D	= diameter of inner tube	[m]
D_{eq}	= hydraulic equivalent diameter of the annulus (= $d_i - D_o$)	[m]
d	= diameter of outer tube	[m]
f	= friction factor	[-]
e	= corrugation depth	[m]
h	= heat transfer coefficient	[W m ⁻² K ⁻¹]
k	= fluid thermal conductivity	[W m ⁻¹ K ⁻¹]
p	= corrugation pitch	[m]
p'	= effective corrugation pitch	[m]
Pr	= Prandtl number	[-]
R_w	= tube wall resistance per unit of area	[K m ² W ⁻¹]
Re	= Reynolds number	[-]

T	= temperature	[K]
U	= overall heat transfer coefficient	[W m ⁻² K ⁻¹]
W	= fluid flow rate	[kg s ⁻¹]
w	= corrugation width	[m]
θ	= tube severity factor, defined by Eq. (1)	[-]
η	= overall performance of the spirally corrugated tubes, defined by Eq. (5)	[-]
ϕ	= geometry function (= $\{ 1 + (D_o / d_i)^2 - [1 - (D_o / d_i)^2] / \ln (d_i / D_o) \} / (1 - D_o / d_i)^2$)	[-]

<Subscripts>

c	= spirally corrugated tubes
i	= tube inside
in	= inlet conditions
o	= tube outside (annulus inside)
s	= smooth tube

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