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HEAT TRANSFER AUGMENTATION IN 3D INNER FINNED HELICAL TUBE

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ABSTRACT
Experiments were performed to investigate the performance enhancement of single-phase flow and boiling heat transfer in the 3D inner finned helical tubes. The tests for single-phase flow and heat transfer were carried out in the helical tubes with a curvature of 0.0663 and a length of 1.15m, the range of the Reynolds number examined varies from 1000 to 8500. In comparison to the smooth helical tube, the experimental results of two finned helical tubes with different inner fin geometry showed that the heat transfer and flow resistance in the 3D inner finned helical tube gains greater augmentation. Within the measured range of Reynolds number, the average augmentation ratio of heat transfer of the two finned tubes are 71% and 103%, compared with the smooth helical tube, and 90% and 140% for flow resistance, respectively. The tests for flow boiling heat transfer was carried out in the 3D inner finned helical tube with a curvature of 0.0605 and a length of 0.668m.Compared with that in the smooth helical tube, the boiling heat transfer coefficient in the 3D inner finned helical tube is increased by 40%~120% under varied mass flow rate and wall heat flux conditions, meanwhile, the flow resistance coefficient increased by 18%~119%.

KEY WORDS: helical tube, finned tube, convective heat transfer, and boiling heat transfer.

NOMENCLATURE

d_i helical inner diameter, m

D_c diameter of coil, m

D_n Dean number [=Re $\delta^{1/2}$]

e fin height, m

f flow drag coefficient

G mass flux，kg/(m^{2}s)

Nu Nusselt number

p Pressure, Pa

P_a axial pitch of the inner finned tube

P_c circumferential pitch of the inner fins

Q heat rate，W

q heat flux，W/m^2

Re Reynolds number

T temperature，℃

w upper width of the fin，m

x mass quality

Greek symbols

δ curvature of the helical tube[$=d_i/D_c$]

ρ density，kg/m^3

INTRODUCTION

Because helical tube heat exchanger has the same good characteristics as those commonly used shell-tube heat exchanger in strong structure, good adaptability, easy-made and low cost, it has been widely used in different applications, such as cooling and air conditioning, chemical industry and pharmacy. As the past research shown, the secondary flow in the helical tube plays a very important role for the heat transfer enhancement in the laminar flow regime. Compared with the straight tube, the heat transfer augmentation ratio of single-phase flow is up to 2.0~4.0 for laminar flow and only 1.1~1.3 for turbulent flow[1-3]. For many applications, however, the heat transfer process may no longer be single-phase flow but two-phase flow in the helical tube. The experimental results of the flow boiling heat transfer in helical tube showed that the average heat transfer coefficient augmentation ratio is only 5%~15% when compared with the straight tube[4].

It is believed, in most cases, that the heat transfer at the inner side of the helical tube is the bottleneck of the whole performance of the heat exchanger. Therefore, it is necessary to enhance the heat transfer in the inner side of the helical tube. It is worth noting that three dimensional finned enhancing technique can be applied both in the single-phase flow and two-phase flow (boiling).

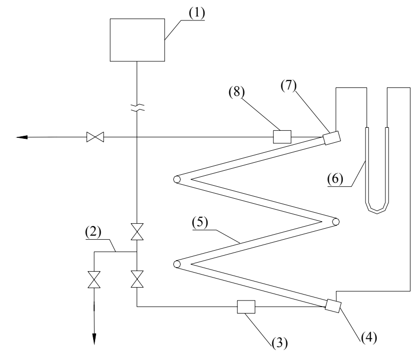
In the case of single phase flow，the 3-D inner fins were applied in the straight tube, with an aim of enhancing the single-phase convective heat transfer. The experimental results by Liao, G. and et al. [5]showed that the augmentation ratio of heat transfer to air flow in the straight tube was from 2.5 to 3.5 with the Reynolds number range from 4000 to 25,000.

Flow boiling heat transfer characteristics are extensively investigated for microfinned straight tubes with single helix geometry (so called 2-D microfinned tubes), as reviewed by A.E. Bergles[6]. The heat transfer enhancement ratio of such kind of tube can be up to 100%. L.M. Chamra and R.L. Webb [7]compared the flow boiling heat transfer performance of a 3-D microfinned tube, which has a cross-grooves geometry, to 2-D microfinned tube. Their results showed that the new geometry 3-D microfinned tube can provided 31% higher heat transfer coefficient than the 2-D microfinned tube.

So far, unfortunately, there are not any papers exist in the literature about the heat transfer enhancement of the inner sides of the helical tube by extended surface method. This paper aimed at examining the heat transfer performance and the flow resistance characteristic with the new 3D inner finned helical tube both in single-phase flow and in two-phase boiling flow regimes. It is well known that the 3D inner finned heat transfer enhancing technology is first applied to the straight tube and its heat transfer performance in single-phase flow is significantly better than that of other enhancement methods, such as 2D rough surface, embedded thread, twisted belt and so on. For the two-phase boiling flow, the unique structure of 3D inner fins can significantly increase the boiling nucleate number and therefore enhance the heat transfer rate. Meanwhile, it is found that the pressure drop is very small. Therefore, if we combine the 3D inner fin enhancement technology with the helical tube's strong secondary flow, the heat transfer performance of the helical tube heat exchanger should be greatly improved and the size of the heat exchanger may become more compact. Hence, this new type of the helical tube presents a new breakthrough in the heat transfer performance with respect to the common smooth helical tube. The experimental study has been conducted to this new 3D inner finned helical tube and the findings not only can be applied to the engineering practice but also will advance the development of the high efficient heat exchanger.

EXPERIMENTAL RIG AND THE PROCESS

The schematic configuration of the experiment equipments is shown in Figure 1. The test helical tube is made of two loops copper tube, whose outer diameter is 16.0mm and thickness is 2.0mm, and the helical diameter is 181.0mm, helical pitch is 75.0mm. After many experiments and the numerical analysis in the whole laminar flow regime, Manlapaz & Churchill [3] and other researchers had concluded that the heat transfer of the smooth helical tube would be remarkably influenced when the helical pitch is longer than the helical diameter. Based on this, the helical tube used in this experimental study can be approximately viewed as horizontal coiled tube because the helical pitch (75.0mm) is far less than the helical diameter (181.0mm). The test helical tube with a total length of 1147.0mm has a straight stable flow part with 500.0mm length, which locates at the inlet of the helical tube. In order to supply the helical tube with uniform heat flux, the flat heater band (4×



(1) tank (2) by pass tube (3)inlet chamber (4)gauge ring
(5) helical tube (6)pressure gauge (7) gauge ring (8)outlet chamber

Fig.1. Schematic of experimental rig

0.2mm) is placed close and tightly twisted on the outer surface of the tube. The transformer is used to adjust the heat flux by varying the power inputs from electric source. When both the voltage of the ends of the heater band and current are measured, the power, which is applied on the helical tube, can be calculated. To prevent the heat loss to the environment, the heat-insulated material, whose thickness is up to 30.0mm, is wrapped on the tube. In this experiment, the temperature is the key parameter needs to be measured accurately and thus we use the calibrated thermal couples, which are mounted on the outer surface of the helical tube's wall, as temperature sensor. In order to monitor the temperature distribution along the helical tube, the cross section of the middle and near the outlet of the helical tube are mounted 6 pairs of thermal couples, which uniformly locate around the circumference of the tube, then between these two sections, other 4 pairs of thermal couples are also uniformly mounted on the tube's outer surface through helical axis direction. The average temperatures of the inlet and outlet cooling water are measured at the inlet and outlet chamber by three thermocouples, respectively and the mass flow ratio of the cooling water is controlled by valve. In order to achieve stable flow for the cooling water, a tank is used in this experiment, which locates above the helical tube almost 10 meters. At last, all of the thermocouples are linked to the HP3457A and HP3488A through which the temperatures are automatically read.

Table 1 geometries of the 3D inner fin (mm)

	d_i	δ	e	P_a	w	P_c
Tube #1	12	0.0663	0.65	2.0	0.5	3.14
Tube #2	12	0.0663	0.95	4.0	0.5	3.14
Micro-finned tube	11.2	0.0605	0.25	1.0	0.3	0.59

Heat transfer and pressure drop of flow experiments are carried out with three different helical tubes in the single-phase flow. One of the tubes is the smooth helical tube, which is used to verify the experiment's reliability and accuracy. The others are 3D inner finned helical tubes, whose inner surface schematic drawings are shown in Fig.2 and the geometry parameters are shown on table 1.

In this experimental set-up, the uncertainty of the energy loss is between 3.8% and 6.7% and the average energy loss is only 4.6%.

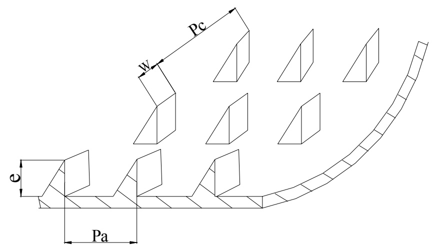


Fig.2. Schematic of inner surface of 3D inner finned

For in-tube flow boiling heat transfer, the detailed experimental apparatus was shown in reference [4]. The inner surface outline of 3D micro-fine test tube is shown in Figure 3, and its geometries are also tabulated in table 1. The experimental tests were performed under the conditions of pressure 0.49~0.67, mass flux 70~320kg/(m^2s), heat flux kW/m^2 and vapor qualities 0.05~0.95. The uncertainties of the main parameters in this experiment are: heat transfer coefficient h ±10.6%; mass flux G ±3.0%; quality x ±2.6%.

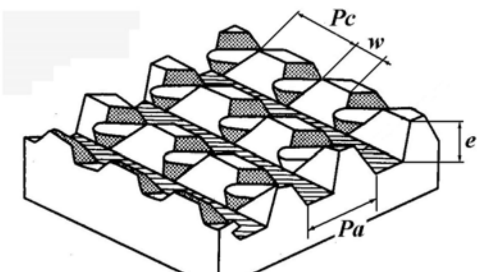


Fig.3. Schematic of inner surface of 3D micro-fin tube

SINGLE-PHASE EXPERIMENT RESULTS AND ANALYSIS

From the prior experiments and the conclusions, the secondary flow tend to stabilize the laminar flow; therefore the critical Reynolds number from laminar to turbulent flow in helical tube is far higher than that in the straight tube. Schmidt [8] recommended that the critical Reynolds number could be calculated with the following formula:

$$Re_{crit}=2300(1+8.6\delta^{0.45}) \quad (1)$$

Srinivasan *et al.* [9] introduced another formula to calculate the critical Reynolds number:

$$Re_{crit}=2100(1+12\delta^{0.5}) \quad (2)$$

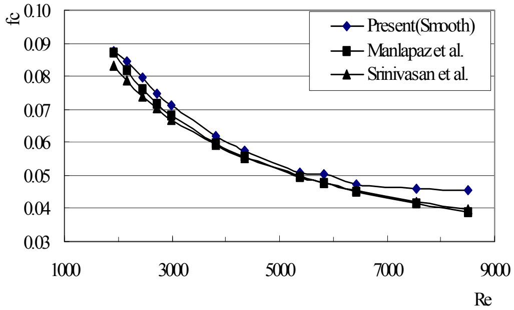


Fig.4 Flow resistance vs. Re in smooth helical tube

In this study, the curvature of the helical tube is 0.0663, from the above two formulas, the calculated critical Reynolds numbers are 8133 and 8588, respectively. In this experiment, the working material is water and the Reynolds number is varying in between 1900 and 8500, and thus the flow in this helical tube is still in the laminar flow regime. Figure 4 depicts the water's flow resistance vs. Re in smooth helical tube where the flow is in the laminar flow regime. From Fig.4, we can conclude that this paper's experiments are very consistent with the prior results. Note that larger deviation in friction factor measurement was observed at high Reynolds number. This is reasonable since the flow is approaching turbulent flow regime when the Reynolds number is near the upper limit of our experiments. Compared with the Srinivasan's results [9], the average error is 5.48% and the maximum error is less than 7%; compared with the Manlapaz and Churchill's results [3], the average error is 4.25% and the maximum error is less than 5.5%.

The prior researchers had concluded some experiential formulas about the heat transfer performance in the smooth helical tubes. Janssen and Hoogendoorn [10] introduced following formula:

$$Nu=0.7Re^{0.43}Pr^{1/6}\delta^{0.07}, \qquad Pr > 20 \quad (3)$$

Dravid *et al.* [2] brought another empirical formula, which is suitable for wider conditions.

$$Nu=(0.76+0.65 \cdot Dn^{0.5})\, Pr^{0.175} \quad (4)$$

Where, $50<Dn< 2000$ and $5<Pr< 175$.

Figure 5 illustrates Nu number vs. Re number in the smooth helical tube. From the figure, the calculated Nu number from measurements is differed by 0.7% than the formula of

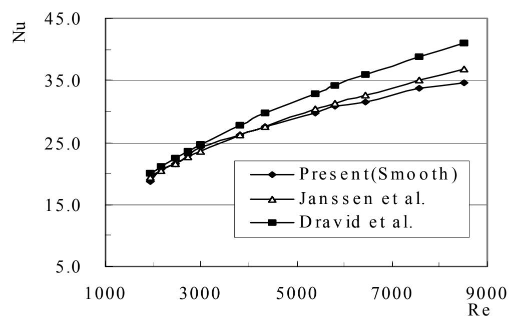


Fig.5 Nu vs. Re in smooth helical tube

Janssen and Hoogendroorn [10], the maximum error is less than ±4%. But for the formula of Dravid *et al.* [2], the result is differed by 6.88%; and the maximum error is less than 10%.

From the above smooth helical tube experimental data comparisons, it can be concluded that the experimental method used in this study is accurate and reliable. Thus, it is believed that the subsequent tests' results are significant and reliable.

The flow resistance and the heat transfer characteristics of two different types of 3D inner finned helical tubes are shown in Fig.6 and Fig.7, respectively. For the two types of 3D tube, it can be seen that the friction factor and Nusselt number of the fin of No.1 tube are lower than that of No.2 (see Figs. 6 and 7). It is also worth pointing out that the height of the fin plays very importance role for the flow resistance and the heat transfer coefficient. With the height of the fin longer, both the flow resistance and the heat transfer coefficient are increased in the 3D inner finned helical tubes. In Figure 7, compared with

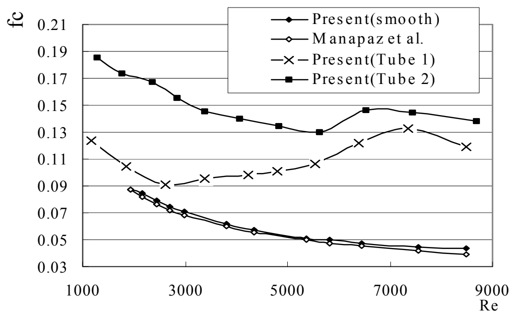


Fig.6 Flow resistance vs. Re in 3D inner finned helical tube

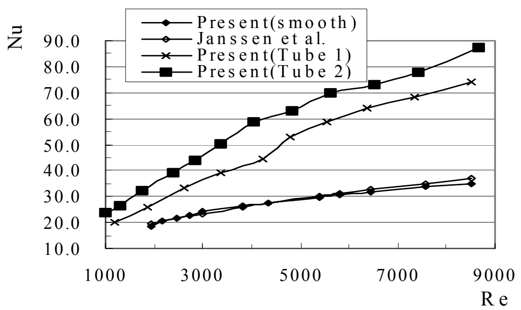


Fig.7 Nu vs. Re in 3D inner finned helical tube

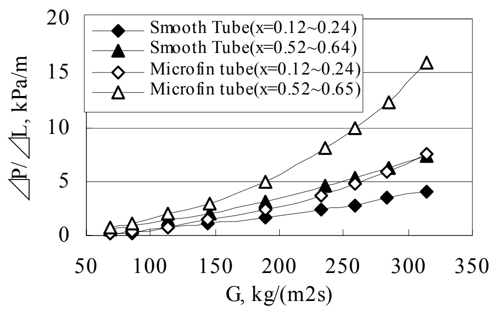


Fig.8 Pressure drops in two-phase flow

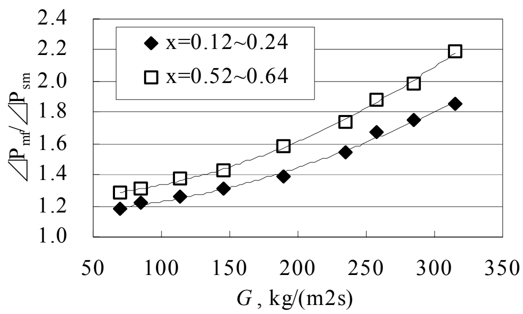


Fig.9 Pressure drops in 3D helical tube

smooth helical tube, the average heat transfer augmentation ratio of No.1 tube is 1.71 with the maximum up to 2.1. The average flow resistance augmentation ratio is 1.9 (see Fig. 6). For the No.2 tube, the average heat transfer augmentation ratio is 2.03 and the maximum is up to 2.3, but the average flow resistance augmentation ratio is about 2.4. It is well known that the heat transfer enhancement of the 3D structure depends on the area of heat transfer surface in the laminar flow regime. As the Reynolds number increases toward the turbulent flow regime, the intensity of flow increased, the effect of coarse surface on the flow disturbances will be increased, which may results in early flow transition (from the laminar flow to turbulent flow). Although the secondary flow can stabilize the flow in the laminar flow regime so that the critical Reynolds number can reach 8500 in the smooth helical tube, the transition of the flow may arrive earlier because the inner fins disturb the flow in the 3D inner finned helical tube. In 3D finned helical tube, the flow pattern is extremely complicate, not only there exists the effect centrifugal and Coriolis forces on the secondary flow and main flow, but also the surface disturbance due to the existence of the finned surface. This type of complex flow study has not been seen in the literature.

EXPERIMENTAL AND THEORETICAL ANALYSIS OF FLOW BOILING HEAT TRANSFER

In the experimental investigation, the flow boiling heat transfer and flow resistance are measured using refrigerant R134a for the smooth helical tube and 3D inner finned helical tube. The test conditions are as follows: Mass flow ratio G=70~380 kg/(m^{2}s); Heat flux q=2.0~22kW/m^2 and the Quality x=0.05~0.95.

The flow pressure drops per unit length in smooth and 3D inner finned helical tube are shown in Fig.8. In this figure, the average quality of the refrigerant varies from 0.12 to 0.25 and from 0.52 to 0.64, where the flow is in the two-phase flow regime. As can be seen from the figure, with the mass flow ratio increased, the flow boiling resistance is increased both in smooth and 3D inner finned helical tube, and the resistance of the high quality flow is larger than that of the lower quality flow. Therefore, all of the resistances in 3D inner finned helical tube are higher than that of the smooth one and the increase in 3D tubes is faster than that of smooth one with the mass flow ratio increased.

Under the same conditions, the flow resistance ratio of 3D and smooth tube vs. mass flow ratio are shown in Fig.9. From this figure, with the mass flow ratio increased, the flow resistance of 3D and smooth tube, which is between 1.18 and 2.19, is also increased, but the value in the high quality is lower than that in the low quality flow region.

The average heat transfer coefficient in 3D tubes and the heat transfer enhancement ratio (h_{mf} / h_{sm}) between 3D and smooth tube vs. mass flow ratio are shown in Figs.10 and 11, respectively. In Fig.10, with the mass flow ratio increased; the average heat transfer coefficient is increased. At the same time, the higher the heat flux, the larger the heat transfer coefficient. From Fig.11, the enhancement ratio, which is in between 1.4 and 2.2, is decreased when the mass flow ratio increases and the value under the lower heat flux condition is higher than that of higher heat flux condition.

CONCLUSION

This paper is the first study to apply the 3D inner fin to the helical tube, which combines the 3D fin with the strong secondary flow in the helical tube to enhance the single phase and boiling heat transfer. The experiments of heat transfer and flow resistance are performed both in 3D inner finned and

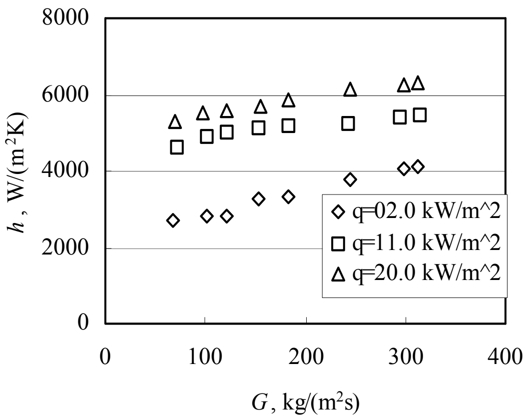


Fig.10 Average boiling heat transfer coefficients

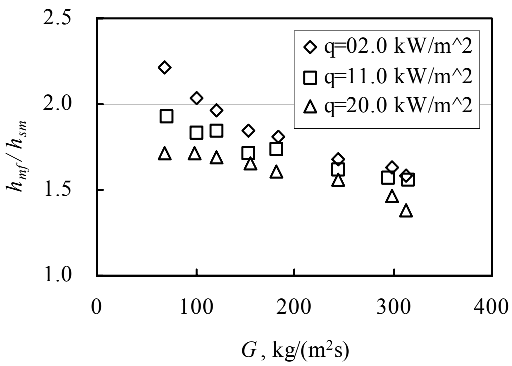


Fig.11Augmentation ratio of boiling heat transfer vs. mass flux in 3D micro-finned helical tube

smooth helical tube to investigate the heat transfer and flow resistance performance. The baseline study of the smooth helical tube agrees very well with the existing experimental data. Some brief conclusions can be written as follows:

(1) The surface of 3D inner fins plays a very important role in the heat transfer enhancement of the helical tube in the single-phase flow regime. Within the range of Reynolds number (1000~8500), compared with the smooth helical tube, the average heat transfer coefficient of the two different geometry 3D inner fin tubes are increased by 71% and 103% and the flow resistance is only increased by 90% and 140%, respectively.

(2) Compared with the two different geometry 3D inner fin tubes, the higher the fins, the better the heat transfer performance.

(3) The special geometry of the 3D inner fins can significantly enhance the flow boiling heat transfer in the helical tube. In this paper, compared with the smooth helical tube, the flow boiling heat transfer coefficient of the 3D inner finned helical tube is increased by 40%~120%, and the flow resistance is only increased by 18%~119%

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