Experimental investigation on heat transfer performance of corrugated low finned tubes

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Abstract. With the outbreak of oil crisis, heat exchangers as the energy-saving devices have gained more and more attentions. The heat transfer enhancement techniques have been developed to the third generation. Corrugated low finned tube is a kind of helically corrugated tube with trapezoidal shape groove. In this paper, the performance of convective heat transfer inside corrugated low finned tubes was studied experimentally. The results showed that the heat transfer performance could be intensified compared with the smooth tube. The reason was that helical rib generated the secondary flow and longitudinal vortex, reducing the synergy angle between velocity vector and temperature gradient. Moreover, the effects of tube structural parameter and fluid Reynolds number were analyzed. Finally, all the experimental data were fitted as the function of Reynolds number, Prandtl number, tube pitch and rib height. The research results can provide a foundation for designing heat exchangers using this enhancement technique.

1. Introduction
With the outbreak of oil crisis, heat exchangers as the energy-saving devices have gained more and more attentions. According to Bergles’s perspective [1], the heat transfer enhancement techniques have been developed to the third generation. The most popular and successful enhancement technique is the artificial roughness surface and the common enhanced tube is helically corrugated tube [2].

Yang et al [3] presented an experimental study of four spirally corrugated tubes under conditions of Reynolds number varying from 6000 to 93,000 for water, and from 3200 to 19,000 for oil. They found that the thermal performance of enhanced tubes was superior compared to a smooth tube. Vicente et al [4] studied the heat transfer performance at different Prandtl numbers. The results showed that heat transfer augmentation increased with Prandtl number. Pethkool et al [5] investigated the effects of pitch-to-diameter ratio and rib-height to diameter ratio on the thermal performance factor. The results indicated that the maximum thermal performance was 2.3.

Corrugated low finned tube (CLFT) is a kind of spirally corrugated tube with trapezoidal profile. Due to the wider spiral protrusion, its heat transfer performance is better than that of ordinary spirally corrugated tube. In this paper, the performance of convective heat transfer inside corrugated low finned tubes was studied experimentally. The effects of tube structural parameter and fluid Reynolds number were also analysed. And all the experimental data were fitted as the function of Reynolds number, Prandtl number, tube pitch and rib height.
2. Experimental apparatus

2.1 Test loop
The schematic diagram of test loop is shown in Figure 1. The experimental apparatus in present study is the identical device used in Ren’s work [6,7]. Valve #4, #5, #9 and #10 were opened while the rest valves remained close. So, only the heating water circuit, the test section, the cooling water circuit and the data acquisition system were activated during single-phase convective experiment. The test section was a double-pipe, counter current flow heat exchanger. The inner tube was the corrugated tube and outer tube was a round smooth tube. The outer tube was covered with glass fiber to reduce heat loss.

![Figure 1. Schematic diagram of experimental loop](image)

During the experiment, the cooling water was pumped by the centrifugal pump to the shell side. Its volume flow rate was kept as large as possible to reduce the temperature fluctuation of shell side. Then the shell side heat transfer coefficient could be considered as a constant. The plate heat exchanger was utilized to remove the added heat from the cooling water. The heating water was heated by the heat coils and was pumped by the magnetic pump to the tube side. Its volume flow rate was adjusted by the globe valve.

2.2 Data reduction
The overall thermal resistance during the heat transfer process consists of convective resistance inside the tube, conductive thermal resistance through the tube and convective resistance outside the tube, as shown below:

\[ R_{\text{total}} = R_i + R_w + R_o \] (1)

According to the definition of thermal resistance, the above equation can be rewritten as:

\[ \frac{1}{U_o} = \frac{A_o}{h_o} - \frac{\ln(d_o/d_i)}{2\pi \cdot \lambda_o \cdot L} + \frac{1}{h_o} \] (2)

The heat transfer coefficient of the shell side is kept constant. Then the sum of last two terms on the right of Eq. (2) can be considered a constant:

\[ R_w + R_o = \frac{\ln(d_o/d_i)}{2\pi \cdot \lambda_o \cdot L} + \frac{1}{h_o} = C_i \] (3)

The heat transfer coefficient of the tube side can be calculated as:
$$h_i = \frac{\lambda_h}{d_i} \cdot Nu_h = \frac{\lambda_h}{d_i} \cdot (m^b \cdot Re_h^n \cdot Pr^{0.3}) = C_2 \cdot Re_h^n$$  \hspace{0.5cm} (4)$$

Submitting Eq. (3) and Eq. (4) into Eq. (2), the overall thermal resistance can be written as:

$$\frac{1}{U_o} = \frac{A_o}{C_2 \cdot A_h} \cdot \frac{1}{Re_h^n} + C_1$$  \hspace{0.5cm} (5)$$

If the $Re$ exponent $n_h$ in Eq. (5) is assumed, the experimental values of the overall thermal resistance can be represented as a linear function, in which $1/U_o$ is the dependent variable and $1/Re_h^n$ is the independent variable. $A_o/(C_2 \cdot A_h)$ is the slope of straight line and $C_1$ is the intercept [8, 9]. By applying simple linear regression, the values of constants $C_1$ and $C_2$ can be determined. Then the internal convective heat transfer Nusselt number can be evaluated from:

$$Nu_h = \frac{d_o}{\lambda_h} \cdot \frac{A_o}{(1/U_o - C_1) \cdot A_h} = \frac{d_o}{\lambda_h (1/U_o - C_1)}$$  \hspace{0.5cm} (6)$$

3. Results and discussion

3.1 Verification of the experimental apparatus

The tests inside smooth tubes were carried out to verify whether the experimental apparatus and instruments were functioning properly. The secondary-side heat transfer rate was compared to that inside tubes and only tests with differences in energy balance within 5% were considered. The experimental Nusselt numbers were compared with the predicted values calculated from Dittus–Boelter correlation. As it can be seen in Figure 2, all of the experimental Nusselt numbers agree well with the predicted values. The maximum deviation is -3.9%.

3.2 Heat transfer characteristic

Figure 3 presents the variation of Nusselt numbers with Reynolds numbers for the test tubes. Reynolds numbers range from around 15,000 to 65,000, which is under the turbulent flow condition. It can be found that the Nusselt numbers increase with the rise of Reynolds numbers both inside corrugated tubes and the smooth tube. This is because the higher Reynolds number means higher turbulence, thinner boundary layer and lower convective heat resistance. It is also noticed that all the enhanced tubes have a better heat transfer performance than the smooth tube.

Figure 4 shows the variation of Nusselt number ratio ($Nu_a/Nu_s$) with Reynolds number. It is observed that the enhancement factor of Nusselt number decreases with the increase of Reynolds number. It is because the boundary layer is thin when the Reynolds number is low and the helical rib has a significant effect. The thickness of boundary layer will increases as the rise of Reynolds number.
and gradually exceeds the rib height. The ability of generating separation vortices of helical rib becomes weaker and weaker and the heat transfer enhancement becomes worse and worse.

![Figure 3. Variation of Nu with Re](image1)

![Figure 4. Variation of Nu_s/Nu with Re](image2)

3.3 Correlation for Nusselt number

![Figure 5. Comparison between measured and predicted Nu](image3)
By fitting the experimental data, the Nusselt number can be correlated as the function of Reynolds number, Prandtl Number, pitch to diameter ratio and rib height to diameter ratio. The correlation is described in Eq. (7).

\[ Nu = 1.121 \cdot Re^{0.638} \cdot Pr^{0.33} \left( \frac{P}{d} \right)^{-0.29} \left( \frac{e}{d} \right)^{0.461} \]  

(7)

4. Conclusions

In this paper, the performance of convective heat transfer inside corrugated low finned tubes was studied experimentally. The results showed that the heat transfer performance could be intensified compared with the smooth tube. The reason was that the helical rib generated the secondary flow and longitudinal vortex, reducing the synergy angle between velocity vector and temperature gradient. All the experimental data were fitted as the function of Reynolds number, Prandtl number, tube pitch and rib height. The research results can provide a foundation for designing heat exchangers using this enhanced tube.

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