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Condensation heat transfer in horizontal three dimensional two-layer two-side enhanced tubes

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ABSTRACT

Condensation heat transfer characteristics of near-azeotropic refrigerant R410A inside the horizontal copper and stainless steel EHT series tubes were measured. All three dimensional two-layer two-side enhanced tubes have deep-textured surface structures composed of petal-shaped background patterns and dimpled protrusions or boat-shaped cavities. Tests were conducted at an average saturation temperature of 45 °C; over a mass flux range of 100–450 kg/(m^2·s); with an inlet quality of 0.8 and outlet quality of 0.2. Condensation heat transfer coefficient of the copper 1EHT1 tube is about 39–47% higher than that of a stainless steel one, showing a beneficial effect of high thermal conductivity material. Effects of saturation temperature and tube diameter on the copper 1EHT tubes were also discussed. Compared with a smooth tube, the heat transfer enhancement ratio of these stainless steel EHT tubes is up to 1.19. As expected, the 1EHT1 tube exhibits the highest condensation heat transfer coefficient due to its larger dimpled protrusions. In addition, the heat transfer performance of 1EHT2 and 3EHT tubes are similar. An explanation could be the condensate retention in boat-shaped cavities on the bottom of the 3EHT tube.

1. Introduction

Heat exchange and energy recovery are commonly involved in a wide variety of industrial facilities and household appliances. In the past few decades, for sake of promoting the development of high performance thermal systems (i.e. biochemical reactor, heat pump, sea water desalination equipment, HVAC, etc.), various enhanced surface structures have been designed and applied. The desired goal is to make heat transfer units compact and generate a higher heat transfer coefficient with a limited increase to the pumping power. Over time, enhanced heat transfer tubes are being the main method applying to improve the thermal efficiency of heat transfer systems.

Ji et al. [1] reported their experimental data on condensing heat transfer of R134a outside five smooth tubes and six enhanced tubes with three-dimensional integral-fins. Five tube materials were used in their research, including titanium, cupronickel (B10 and B30), stainless steel and copper. Within the heat flux range tested, the external condensation heat transfer coefficients of enhanced tubes were 7.5–8.5 times those of smooth tubes with the same materials. In addition, they found that the heat transfer enhancement ratio of enhanced copper tubes was about 1.6–2.1 when compared to low thermal conductivity tubes with the same surface structures. Wan et al. [2] tested stainless steel edge-shaped finned tubes fabricated by a plowing-extruding process. Under the same test condition, the shell-side condensation heat transfer coefficients of edge-shaped finned tubes were about 1.7–2.6 times that of a smooth tube. They also found that the enhancement of heat transfer performance increased with the increasing plowing-extruding depth. An experimental investigation on convective condensation of R600a inside a horizontal smooth tube and a dimpled tube was performed by Sarmadian et al. [3]. Their results showed the heat transfer coefficients of the dimpled tube were 1.2–2.0 times that of the smooth tube. Additionally, they noted that the transition between intermittent flow and annular flow in dimpled tubes occurred at a lower vapor quality. Wang et al. [4] studied the heat transfer characteristics using cold air in the dimpled tubes with different pattern arrangements. Results indicated that the tubes with parallel or staggered dimples arrangement have the similar performance.

Recently, Kukulka et al. [5] introduced a series of novel Vipertex enhanced tubes (EHT series tubes) with petal-shaped background

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patterns in a certain arrangement. The results of fouling tests suggested that fouling occurred at a faster speed for one smooth tube than EHT series tubes. Thus, this type of surface design can efficiently inhibit the in-tube fouling. Guo et al. [6] investigated the condensation heat transfer in horizontal smooth, herringbone and 1EHT tubes using refrigerants R22, R32, R410A. They pointed out that the condensation heat transfer coefficient of 1EHT tube was nearly 0.30–0.95 times higher than that of a smooth tube. Li et al. [7] tested the condensation heat transfer characteristics of R410A inside one smooth and two 2EHT tubes. Compared with a smooth tube, the measured condensing coefficients of 2EHT tubes were 10–16% higher. Although the tube-side heat transfer efficiency of EHT copper tubes has been confirmed, it still remains unknown for the EHT series tubes made of low thermal conductivity materials. Therefore, a further research is of great importance for the development of enhanced surfaces.

2. Experimental details

In this paper, the experimental rig consists of two main closed circuits: (1) a refrigerant loop; and (2) a cooling water loop of the test section. The same test system was introduced in detail by the authors' group [6,7] when it was utilized in the investigation on internal heat transfer characteristics of copper tubes. The whole test section is a horizontal counter-flow tube-in-tube heat exchanger with an effective length (L) of 2 m. Both platinum RTDs-100 and absolute pressure transducers are installed at the inlet and outlet of the test section to monitor the state of working fluid.

Three dimensional deep-textured surfaces of all the two-layer two-side enhanced tubes tested are presented in Fig. 1. Obviously, the 1EHT1 tube has larger dimpled protrusions/cavities on its surface than the 1EHT2 tube. These surface structures, composed of dimpled protrusions or boat-shaped cavities and staggered petal arrays, can extend the active heat transfer area and provide a longer product usage life. In order to reduce energy and metal material consumption, these enhanced tubes were fabricated using a multiple high-pressured extruding process. Consequently, the actual wall thickness (δ) varies from 0.46 mm to 0.71 mm due to different degrees of extrusion force in this process. Using the results measured by a Nanovea ST400 3D Profilometer, the ratios of the actual inner surface area to that of an equivalent smooth tube were calculated to be 1.20, 1.09 and 1.25, respectively for the 1EHT1, 1EHT2 and 3EHT tubes. All test tubes have the same nominal outer diameter (d_o) of 9.52 mm. For the sake of simplicity, the six test tubes are named by Cu-ST, Cu-1EHT1, SS-ST, SS-1EHT1, SS-1EHT2 and SS-3EHT, respectively, where two types of tube materials are represented by Cu (pure copper) and SS (Type 304L stainless steel).

Condensation tests were performed using near-azeotropic refrigerant R410A over the mass flux range of 100–450 kg/(m^2·s); at a saturation temperature of 45 °C; with the vapor quality along the test section changing from 0.8 to 0.2. All thermodynamic properties of R410A were obtained from NIST REFPROP 9.0 database [8].

3. Data reduction

Data were reduced in order to obtain the tube-side heat transfer coefficient. The heat transfer rate in the test section, Q_{ts}, can be calculated by an energy conservation equation for the cooling water flowing through the annulus:

$$Q_{ts} = c_{pl,w,ts} \cdot m_{w,ts} \cdot (T_{w,ts,out} - T_{w,ts,in}) \tag{1}$$

where $c_{pl,w,ts}$, $m_{w,ts}$, $T_{w,ts,out}$, $T_{w,ts,in}$ are the specific heat taken at an average temperature $((T_{w,ts,out} + T_{w,ts,in})/2)$, mass flow rate, inlet temperature and outlet temperature of the recycled water through the whole test section, respectively.

The logarithmic mean temperature difference ($LMTD$) is expressed by

$$\Delta T_{LMTD} = \frac{(T_{ref,ts,out} - T_{w,ts,in}) - (T_{ref,ts,in} - T_{w,ts,out})}{\ln\left[(T_{ref,ts,out} - T_{w,ts,in})/(T_{ref,ts,in} - T_{w,ts,out})\right]} \tag{2}$$

Here, $T_{ref,ts,in}$ and $T_{ref,ts,out}$ are the refrigerant inlet and outlet temperatures of the test tube. Assuming no fouling resistance, the internal heat transfer coefficient can be obtained from the following thermal resistance equation:

$$h_i = \frac{1}{A_i\left[\frac{LMTD}{Q_{ts}} - \frac{1}{h_w A_o} - \frac{\ln(d_o/d_i)}{2\pi L k_{wall}}\right]} \tag{3}$$

where the outer surface area ($A_o = \pi d_o L$) is on basis of the nominal outside diameter (d_o), and inner surface area ($A_i = \pi d_i L$) is the nominal area based on the maximum inside diameter ($d_i = d_o - 2\delta$). Besides, k_{wall} is the thermal conductivity of tube material.

For smooth tubes, the water-side heat transfer coefficient (h_w) can be calculated by the empirical correlation provided by Gnielinski [9]. In view of those outer surface enhancement structures of EHT tubes, the Gnielinski equation [9] cannot apply to the EHT tubes directly. Therefore, a correction factor C was adopted as the heat transfer enhancement ratio of the enhanced surface tube to a smooth tube. Accordingly, the actual water-side convection coefficient can be obtained from the modified equation:

$$h_w = C \cdot \frac{(f/2)(Re_w - 1000)Pr_w}{1 + 12.7(f/2)^{1/2}(Pr_w^{2/3} - 1)} \left(\frac{\mu_{bulk}}{\mu_{wall}}\right)^{0.14} \frac{k_w}{d_h} \tag{4}$$

Here, a power of dynamic viscosity ratio $(\mu_{bulk}/\mu_{wall})^{0.14}$ reflects the influence of non-homogeneous temperature distribution on fluid properties. d_h is the hydraulic diameter of the test tubes. Something else to note is that experimental data points are valid only for $3000 < Re_w < 5 \times 10^6$ and $0.5 < Pr_w < 2000$. The Fanning friction factor (f) is determined from the Petukhov equation [10] as given in Eq. (5).

$$f = (1.58 \ln Re_w - 3.28)^{-2} \tag{5}$$

To calculate the unknown multiplier C, the Wilson plot method [11] was applied by varying the mass flow rate of water under fully developed turbulent liquid flow. According to the experimental results, the correction factor C is 2.78, 2.04, 1.24 and 1.85 respectively for the Cu-1EHT1, SS-1EHT1, SS-1EHT2 and SS-3EHT tubes. In this study, all uncertainties in calculated parameters were obtained by the procedure described by the authors' group [7]. The maximum uncertainty of h_i is 10.22% when the relative error of h_w is assumed to be 10%, showing a high test accuracy.

4. Results and discussion

Fig. 2a reveals the relationship between the condensation heat transfer coefficient (HTC) and refrigerant mass flux inside the tested copper 1EHT1 and smooth tubes. Measured HTC of the 1EHT1 tube is 1.53–1.62 times that of a smooth copper tube at mass fluxes varying from 150 to 450 kg/(m^2·s). Under the same operating conditions, better performance of the 1EHT1 tube is mainly attributed to the larger heat transfer surface area, stronger turbulence effect and condensate film disruption caused by dimpled protrusions. Moreover, staggered petal arrays on the internal surface can continue to induce secondary flow mixing and separation with the help of inertia force, thus further enhancing the effect of turbulence. With the increasing inner diameter, the condensation process is gradually dominated by gravity. More dimpled protrusions on the lower part of the tube will be submerged in the