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Effects of tube shape on boiling heat transfer of low-pressure refrigerantR1233zd(E) outside horizontal tubes

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Abstract

Hydro-fluoro-olefins and hydro-chloro-fluoro-olefins, such as R1234ze(E), R1234ze(Z), and R1233zd(E), are anticipated to become low global warming potential refrigerants to replace R134a and R245fa for industrial heat pump systems. Experimental data on the heat transfer coefficient of saturated pool boiling of R1233zd(E) on a horizontal fin tube were obtained in this study. The test tube was 19.05 mm in diameter and 400 mm in heat transfer length. The heat transfer coefficients in the two kinds of fin tube were compared. The heat transfer characteristics differed according to the slit on cover of tunnel structure.

Keywords: evaporation, heat transfer coefficient, pool boiling, expansion ;

1. Introduction

Hydro-fluoro-olefins (HFOs) and hydro-chloro-fluoro-olefins (HCFOs), such as R1234ze(E), R1234ze(Z), and R1233zd(E), are expected to become low global warming potential (GWP) refrigerants, which will replace R134a and R245fa for industrial heat pump systems. The normal boiling points of most alternative refrigerant candidate in HFOs and HCFOs are higher than those of hydrofluorocarbons (HFCs). Therefore, suitable tube shape for HFOs and HCFOs is different from that for HFCs. However, the heat transfer characteristics of HFOs and HCFOs have not been investigated.

In this study, experimental data on the heat transfer coefficient (HTC) of saturated pool boiling of R1233zd(E) on a horizontal fin tube are presented.

2. Experiment

2.1. Experimental Apparatus

Fig. 1 shows an experimental apparatus to measure the condensation and pool boiling HTC on a horizontal smooth tube. This is a natural circulation system, in which the liquid refrigerant condensed on a test tube (4) and subcondensers (6) flows down to a boiling chamber, while the vapor evaporated on the test tube and electric heaters (8) returns to a condensation chamber. The flow rate of the condensate flowing down a drain pan (5) is measured by a Coriolis mass flow meter (7). The heat transfer on the test tube is obtained from the temperature change measured in the mixing chambers and the flow rate of the water flows in the test tubes. To determine the refrigerant saturation temperature, the pressure is measured very close to the test tube. The saturation temperature is checked with the temperature measured by K-type thermocouples installed in those chambers. An electric current of 40 A is conducted to the test tube, so that the mean wall temperature is calculated as a function of the electric resistance measured by a voltage meter. The wall temperature is calibrated in the preliminary test; meanwhile, the heat loss from the chambers to ambient air is also measured. The heat flux on the tube wall is defined as follows:

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$$q_{\mathrm{wall}} = \frac{V_{\mathrm{H_2O}}\rho_{\mathrm{H_2O}}\left(h_{\mathrm{H_2O,o}} - h_{\mathrm{H_2O,i}}\right) - Q_{\mathrm{loss}}}{\pi D_{\mathrm{o}} L} \tag{1}$$

where V_{H2O}, ρ_{H2O}, and h_{H2O} are the volumetric flow rate, density, and specific enthalpy of water, respectively, evaluated at the arithmetic mean of $T_{H2O,i}$ and $T_{H2O,o}$. Q_{loss} is the heat loss correlated to the temperature difference between water and ambient air. D_o and L are the outer diameter and effective heat transfer length of the test tube. Therefore, the HTC is expressed as follows:

$$\alpha = \frac{q_{\mathrm{wall}}}{T_{\mathrm{sat}} - T_{\mathrm{wall}}} \tag{2}$$

where T_{sat} is the saturation temperature evaluated at the measured pressure. T_{wall} is the tube wall temperature correlated to the electric resistance. The thermodynamic and transport properties of R1233zd(E) are calculated by REFPROP Ver. 10 [1] with the incorporated coefficients optimized by Akasaka et al. [2].

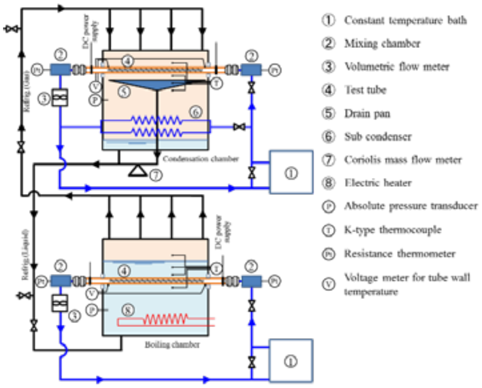

Fig. 1. Schematic diagram of experimental apparatus

2.2. Experimental condition and method

The test refrigerant was R1233zd(E). The HTC was quantified at a saturation temperature of 10°C. The measurement range of heat flux was from 3 to 20 $kW \cdot m^{-2}$ for pool boiling.

2.3. Test tubes

The three kinds of test tubes, which were made of copper, had 19.12-mm outer diameter and 400-mm heat transfer length (tubes A, B, and C), and they were set in a perfectly horizontal configuration. As an example, Fig. 2 shows the external appearance of tube B. The three enhanced tubes had different fin geometries, including the tunnel structure. The difference of the tubes was the size of slit on cover of tunnel structure, as shown Fig. 3. In the order of tubes A, B, and C, the slit on the cover became smaller. For all tubes, the tunnel structure sizes were approximately the same.

Fig. 2. External appearance of tube B

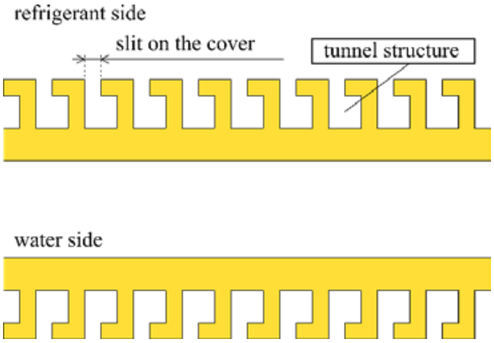

Fig. 3. Schematic diagram of test tubes

3. Results and discussion

Fig. 4 shows the variation in boiling HTC with heat flux in test tubes A, B, and C. The circle, triangle, and square represent tube A, tube B, and tube C, respectively. The symbols are the experimental values of HTC, with the vertical bars indicating the measurement uncertainty of HTC. In general, boiling HTC increases with increase in heat flux.

Boiling HTC of tube B was considerably higher than that of tube A. This was caused by the promotion of the nucleate boiling and the boiling by a thin liquid film. In tube B, when the refrigerant boiled, local pressure became higher by the cover of the tunnel structure. However, in tube A, local pressure did not increase, because the slit on the cover was wide. Then, the nuclear boiling in tube B was promoted. Furthermore, the heat transfer in tube B was promoted by the thin liquid film. In tube B, most of the generated bubbles were exhausted from the upper part of the tube through the tunnel structure, because the slit on the cover was small (a small number of generated bubbles were exhausted by the slit on the cover). A thin liquid film was formed by surface tension as bubbles moved along the tunnel structure. However, in tube A, the generated bubbles were exhausted from the slit on the cover. Then, the heat transfer in tube A was not promoted.

Boiling HTC of tube B was considerably higher than that of tube C. The slit on the cover of tube C was smaller than that of tube B. Then, nuclear boiling in tube C was promoted, as with tube B. However, the HTC of tube C decreased because of a dry patch. In tube C, dryout occurred on part of the heat transfer surface because the slit on the cover was too small, making it difficult for the bubbles to be exhausted.

Therefore, boiling HTC of tube B was considerably higher than those of tubes A and B, because there was an appropriate size in the slit on the cover.

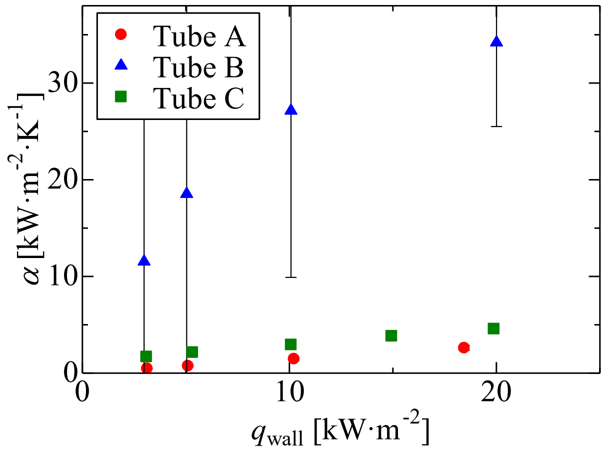

Fig. 4. Variation in boiling HTC with heat flux

4. Conclusion

The experimentally determined HTCs of low-GWP refrigerants of R1233zd(E) on three horizontally installed enhanced tubes were presented. The main conclusions are as follows:

- The HTC was improved by nuclear boiling as well as boiling by the thin liquid film promoted by the tunnel structure and slit on the cover of tunnel structure.
- The slit of the cover of tunnel structure had an appropriate size.

References

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[2] Akasaka, R., and Lemmon, E. W., 2018, Helmholtz equation of state for R-1233zd(E), Refprop Ver. 10.