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An experimental investigation of nucleate boiling heat transfer from an enhanced cylindrical surface

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HIGHLIGHTS

- The boiling heat transfer from the enhanced tube is divided into two heat flux regions.
- Within the low-heat-flux range, the boiling performance increases as the heat flux increases and the pressure has no effect.
- At high heat fluxes, the boiling performance remains nearly flat as the heat flux increases.
- Also at high heat fluxes, the boiling performance increases as the pressure increases.
- The boiling performance on the enhanced tube is compared with that on a plain tube.

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ABSTRACT

In this study, nucleate boiling of the refrigerant R123 on the enhanced surface with 3-D micro-structures of a cylindrical tube is experimentally investigated. Water flows inside the tube and provides heat to the refrigerant outside the tube to boil. Experiments were performed at three pressures corresponding to the saturation temperatures of 4.4, 11.1, and 17.8 °C respectively to obtain the quantitative temperature effect on the boiling heat transfer coefficient. The results showed that boiling heat transfer on the enhanced surface can be divided into the low-heat-flux region ($q'' < 25$ kW/m^2) and the high-heat-flux region ($q'' > 25$ kW/m^2). Within the low-heat-flux region, the boiling heat transfer coefficients increase linearly with increasing heat flux and the influence of the saturation temperature is insignificant. As the heat flux enters into the high-heat-flux region, the boiling heat transfer coefficients remain nearly flat as the heat flux increases within this region and the effect of saturation temperature is considerable. The boiling heat transfer coefficients at 11.1 and 17.8 °C are respectively 8.0–10.0% higher and 22.9–24.2% higher than the boiling heat transfer coefficients at 4.4 °C within the range of heat flux 25–62 kW/m^2, which is equivalent to 1.2–1.8% increase in the boiling heat transfer coefficient per degree increase in the saturation temperature. In addition, for comparison with the enhanced tube, the experiments of boiling on the smooth surface of a plain tube were also performed for the saturation temperature of 4.4 °C. Results showed that the boiling heat transfer coefficients on the enhanced tube are 6–10 times those on the plain tube within the range of heat flux of 12–62 kW/m^2.

1. Background

Increase in the efficiency and reduction in the cost of thermal energy systems drive advancements in heat transfer technologies. As an important heat transfer process, boiling on the outside surface of cylindrical tubes is widely used to exchange heat in industry applications; kettle reboilers used in the petroleum refining industries for separation of multi-component mixtures and submerged evaporators used in the desalination industry to generate clean water are examples of these applications. For the application of geothermal power generation, the working fluid boils on the outside surface of a tube bundle to extract thermal energy of water pumped from underneath the earth for power production; similar arrangements are also used in some waste heat recovery systems. In the chiller applications, chilled water is produced as a result of boiling of a refrigerant on the shell side of a flooded evaporator and then is forced to circulate in buildings for air conditioning. In most of these applications, instead of plain tubes with smooth surfaces being used, the tube surfaces are typically roughened to enhance the heat transfer. For instance, in a flooded evaporator for chiller applications, the tube inside surfaces have integral helical ridges to increase single-phase heat transfer of water flow, and the outside

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surfaces are enhanced to increase boiling heat transfer; the enhanced outside surfaces include, in the broadest sense, extruded integral fins (2-D enhanced surface) and sub-surface micro-structures (3-D enhanced surface).

The system pressure is an important parameter in these applications, and its effect on the boiling heat transfer has been investigated in various boiling configurations: nucleate boiling, flow boiling, and boiling on tube bundles. The results in the literature showed that there is an increase in the boiling heat transfer coefficient with increasing system pressure. Rainey et al. [1] observed that the heat transfer during pool boiling on microporous surfaces is improved as the system pressure is increased. An experimental study of pool boiling of R11 at two saturation temperatures of 4.4 and 26.7 °C on a 2-D enhanced surface (with extruded integral fins) was performed by Webb and Pais [2]. They showed that, as re-produced in Fig. 1, the boiling heat transfer coefficients increase by 5.5–16.0% for every 5 °C increase in the saturation temperature within the range of heat flux 15–47 kW/m^2; larger increases occur at lower heat fluxes. Measurements by Webb and Apparao [3] showed that the pressure effects on boiling heat transfer from both the 2-D enhanced surface and the smooth surface are more prominent than boiling heat transfer from the 3-D enhanced surfaces. For flow boiling, a study of boiling of R12 in a small diameter tube by Tran et al. [4] showed that the boiling heat transfer coefficient measured at 820 kPa is approximately 25% larger than that at 520 kPa; similar results were also obtained by Yen et al. [5] in a study of flow boiling in a microtube. It is known that the overall boiling heat transfer during flow boiling has contributions from nucleate boiling and convective boiling. In a study of flow boiling in a small tube, Baduge et al. [6] found that the nucleate boiling contributes more to the overall boiling heat transfer at a higher system pressure. For boiling on the tube outside surfaces in a tube bundle, the work by Jensen et al. [7] showed that the bundle boiling performance at the pressure 0.6 MPa is higher than that at 0.2 MPa, and this effect of pressure was found to depend on tubes with different surface structures. According to Jensen [8], there is an array of fundamental issues that are important to boiling on tube bundles, and these issues include heat transfer mechanisms, flow regimes, convective effect, valid correlations, non-uniformity heat transfer around the tube circumference, and critical heat flux conditions. It was pointed out that lack of local thermal and flow information for tubes in different locations in a bundle prevents a systematic approach to design of efficient heat exchangers. The numerical work by Leong and Cornwell [9] led to a profile of the local boiling heat transfer coefficient in a tube bundle, where there is an increase in the heat transfer coefficient on tubes from the bottom to the top in a bundle, later known as the *bundle effect*. Palen et al. [10] reasoned that the increase in the boiling heat transfer is due to the convective effect of gravity-introduced two-phase flow in the bundle. Fujita et al. [11] performed a study of boiling of R113 on two tube bundles (one made of smooth tubes and the other made of sintered tubes) at pressures in the range of 0.1–1.0 MPa. They observed that the bundle effect during boiling on the smooth tubes diminishes with increasing pressure, and they attributed the diminishing bundle effect to the fact that at higher pressures, a larger density of bubble nucleation leads to dominating effect of growing bubbles on heat transfer; as a result, convection has a reduced effect. Moreover, the bundle effect was not observed on sintered tubes in the study.

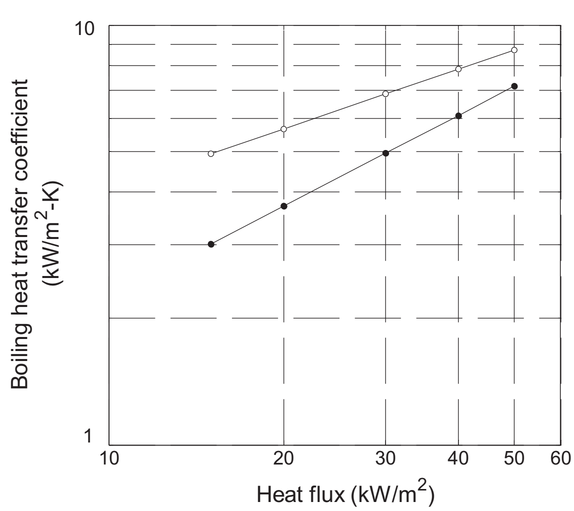

Fig. 1. Boiling heat transfer coefficients (R11) obtained by Webb and Pais [2].

This work was motivated by the need to predict the evaporator performance in a chiller while taking into account the effect of the system pressure on the boiling heat transfer coefficient. Chillers are typically operated at a partial and varying load. The variation in the cooling load leads to variation in the evaporator pressure, and thus the refrigerant saturation temperature. Although it is known that the boiling heat transfer coefficient increases with increasing system pressure and thus saturation temperature, a quantitative measure of the saturation temperature effect on the boiling heat transfer coefficient is not available, especially for boiling on enhanced surfaces. In this work, the heat transfer coefficients during boiling of the refrigerant R123 on the 3-D enhanced surface of a *single* evaporator tube (hereafter called the enhanced surface) were experimentally determined. The experiments were performed at three system pressures respectively corresponding to saturation temperatures of 4.4, 11.1, and 17.8 °C to obtain a quantitative measure of the saturation temperature effect on the boiling heat transfer coefficient on the enhanced surface. In addition, the experiments of boiling on the smooth surface of a plain tube were also performed for the saturation temperature of 4.4 °C for comparison with the enhanced surface.

2. Test section

The test section is an evaporator tube manufactured by Wolverine Tube, Inc. The tube has a total length of 3.048 m that consists of one 2.438-m test segment and two end 0.305-m smooth segments for sealing, plumbing, and instrumentation. Made out of a plain copper tube (raw stock), both the inner surface and the outer surface of the 2.438-m test segment were roughened, as shown in Fig. 2(a), to increase heat transfer. The tube manufacturing process follows the standard evaporator tube manufacturing technologies consisting of sequential steps of drawing, finning, notching, rolling down, and flattening. The tube inner surface was manufactured to have integral helical ridges with a circumferential pitch of approximately 1.5 mm to enhance single-phase heat transfer of water flow inside the tube. The tube outer surface is finished with pocket-like sub-surface micro-structures to enhance the refrigerant boiling heat transfer. A zoom-in view of the micro-structures is given in Fig. 2(b), where the structure pattern has a pitch of nearly 0.5 mm. A cut-away view of the tube wall is shown in Fig. 2(c) that shows the helical ridges on the inner surface and the pocket-like micro-structures on the outer surface, marked with the tube outside diameter D_o and the tube inside diameter D_i; the values of D_o and D_i are respectively 25.20 and 22.66 mm. The tube has an actual outside surface area with micro-structures of 311.2 mm^2 per mm of tube length, which is nearly 4 times the outside footprint area of $\pi D_o = 79$ mm^2 per mm of tube length. It is reasoned that, despite the