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Technical Note

Experimental validation of Cooper correlation at higher heat flux

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ABSTRACT

An experimental investigation on the validation of Cooper correlation at the heat flux of 10,000 to 370,000 W/m^2 is conducted with refrigerant R134a. The slope of best-fitting straight line for heat transfer coefficient versus heat flux is mostly of 0.67 at heat flux less than 250 kW/m^2. A sustainable decrease of slope is observed when heat flux is larger than 250 kW/m^2. At the higher heat flux greater than 250 kW/m^2, a significant deviation of experimental data and the prediction result of Cooper's correlation is observed. The valid range of heat flux for Cooper's correlation to predict the pool boiling heat transfer of R134a should be less than 250 kW/m^2.

1. Introduction

To predict the pool boiling heat transfer of refrigerant outside plain surface, Cooper [1,2] reformulated many pre-existed equations into a particular simple and convenient form:

$$h_o = Cq^{0.67} M_r^{-0.5} P_r^m (-\lg P_r)^{-0.55} \quad (1)$$

where:

$$C = 90\mathrm{W}^{0.33}/(m^{0.66}\ \mathrm{K})$$

$$m = 0.12 - 0.2\lg\{R_\mathrm{p}\}_{\mu\mathrm{m}}$$

The principal physical parameters characterizing the pool boiling heat transfer were considered in this equation. It is now widely used in predicting the pool boiling heat transfer of refrigerant outside plain surface. However, the scope of the heat flux where it is obtained was mainly in the range of 1–100 kW/m^2 and the predicting accuracy was not clearly identified at higher heat flux. In this paper, the investigation on the pool boiling heat transfer outside plain tube is performed at heat flux up to 370 kW/m^2. The objective is to obtain the valid range of Cooper correlation at higher heat flux.

2. Experimental apparatus

The experimental test rig consists of refrigerant, heating water and cooling water circulating systems. The tested tubes are fixed in the boiler. Internal diameter and length of boiler are 257 mm and 1100 mm, respectively. Three tubes can be configured in the boiler at each experiment. When the heating water flows through inside of boiling tubes, refrigerant boils and converts to vapor, rising upward through the tube welding on the top of boiler and entering the condenser. In the condenser, refrigerant vapor is condensed; latent heat is removed by the cooling water flows through the tube-side of the tubes fixed in the condenser. The pool boiling heat transfer coefficient is determined in the heat transfer process. Finally, the condensate flows back to boiler by gravity. The details of the experimental apparatus and test procedure can refer to [3,4].

3. Data reduction

The pool boiling heat transfer coefficient can be determined with the measured temperatures and water flow rates. Heat balance of the system is firstly examined through the test results.

Power input from heating water:

$$\phi_\mathrm{e} = m_\mathrm{e} c_\mathrm{p} (t_{\mathrm{e,in}} - t_{\mathrm{e,out}}) \quad (2)$$

Power output by cooling water:

$$\phi_\mathrm{c} = m_\mathrm{c} c_\mathrm{p} (t_{\mathrm{c,out}} - t_{\mathrm{c,in}}) \quad (3)$$

The properties of water are taken from [5]. The allowed maximum difference between these two heat transfer rates is mostly within 3%, 5% at most. The heat transfer rates, ϕ_e, is used to determine the overall heat transfer coefficient in pool boiling.

$$k = \frac{\phi_\mathrm{e}}{A_\mathrm{o} \cdot \Delta t_\mathrm{m}} \quad (4)$$

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List of symbols

A	area, m^2
c_p	specific heat capacity, J kg^{-1} K^{-1}
C	coefficient of Cooper correlation
d	diameter of tube, mm
h	heat transfer coefficients, W m^{-2} K^{-1}
k	overall heat transfer coefficients, W m^{-2} K^{-1}
L	tube's tested length, m
m	mass flow rate, kg s^{-1}; coefficient in Cooper's correlation
M_r	molecular weight of refrigerant
P_s	saturate pressure
P_r	reduced pressure in Cooper's correlation
q	heat flux, W m^{-2}
r	latent heat, J/kg
Re	Reynolds number
R_f	thermal resistance of foul, m^2 K W^{-1}
R_p	average surface roughness of plain tube, μm
R_w	thermal resistance of tube wall, m^2 K W^{-1}
t	temperature, °C

Greek alphabet

ϕ	heat transfer rate, W
λ	thermal conductivity, W m^{-1} K^{-1}
Δt_m	logarithmic mean temperature difference, °C
σ	surface tension, N/m
ρ_l	density of refrigerant liquid, kg/m^3
ρ_V	density of refrigerant vapor, kg/m^3

Subscript

c	condensing
e	evaporating
i	inside of tube
in	inlet of tube
max	maximum heat flux
o	outside of tube
out	outlet of tube
s	saturation
w	wall

In this equation, Δt_m is the log-mean temperature difference of refrigerant in shell-side and water in tube-side, which is defined as follows:

$$\Delta t_m = \frac{|t_{e,in} - t_{e,out}|}{\ln\left(\frac{t_s - t_{e,in}}{t_s - t_{e,out}}\right)} \tag{5}$$

Thermal resistance separation method is used to determine the shell-side heat transfer coefficient:

$$\frac{1}{h_o} = \frac{1}{k} - \frac{A_o}{A_i}\frac{1}{h_i} - R_f - R_w \tag{6}$$

where, h_o and h_i are the boiling and convection heat transfer coefficients in the shell-side and tube-side, respectively. The tube-side heat transfer coefficient is calculated by Gnielinski correlation [6].

The uncertainty analysis according to literature [7,8] is performed to estimate the experimental uncertainties. The uncertainties of h_i is considered of 10% [6,9,10]. The thermal resistance of tub-side to the overall thermal resistance is within 34.3%. The uncertainties of the shell-side boiling heat transfer coefficient is in the range of 6.9–19.8%.

4. Results and discussion

The pool boiling heat transfer of refrigerant R134a outside plain tube at saturation temperature of 6 °C, 10 °C, 16 °C and 22 °C (±0.2 °C) are investigated. The parameters that influence the heat transfer rate include the roughness and shapes of heating surface, fluid, saturate pressures, and heat flux. In the calculation, the surface roughness is 0.3 μm as suggested by [1,2] for commercial plain tubes. Figs. 1–4 show the comparison of Cooper's correlation and experiment results at different saturate temperatures and heat flux. The heat transfer coefficient is plotted against heat flux in log–log plot. The relative deviation of test results from Cooper's correlation lies within ±10% at lower heat flux less than 250 kW/m^2 K. As the increment of heat flux, the deviation is increasing. At saturation temperature of 22 °C, a deviation up to −44.8% is observed at heat flux 368.0 kW/m^2. The error bars are also shown in the graphs. Considered the experimental uncertainties, the pool boiling heat transfer coefficient at heat flux higher than 250 kW/m^2 is still significantly lower than the prediction result.

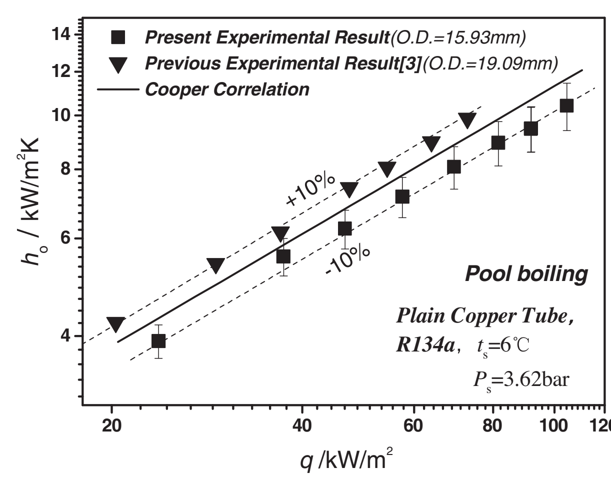

Fig. 1. Pool boiling heat transfer coefficient versus heat flux outside plain tube at saturate temperature 6 °C.

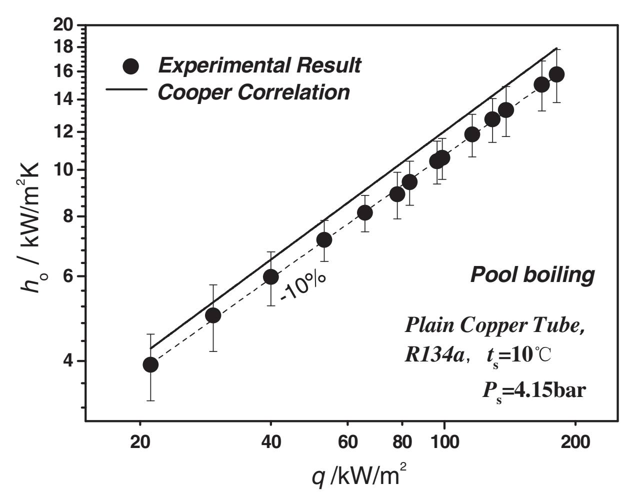

Fig. 2. Pool boiling heat transfer coefficient versus heat flux outside plain tube at saturate temperature 10 °C.

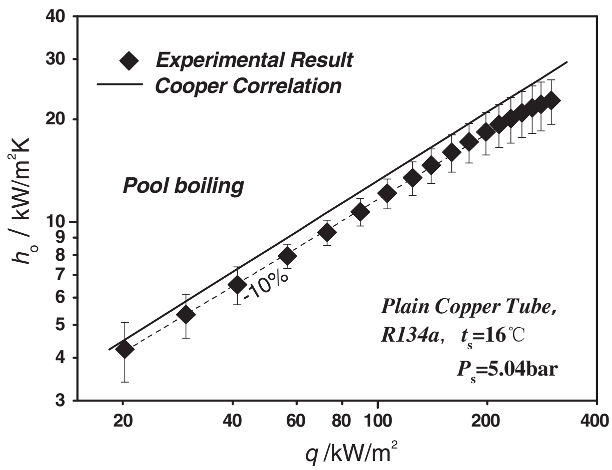

Fig. 3. Pool boiling heat transfer coefficient versus heat flux outside plain tube at saturate temperature 16 °C.

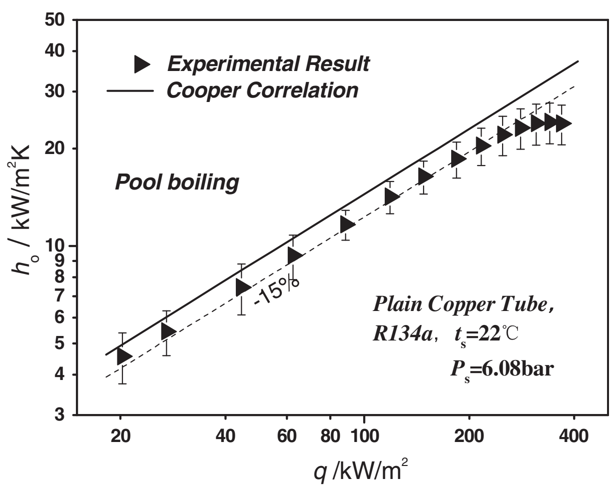

Fig. 4. Pool boiling heat transfer coefficient versus heat flux outside plain tube at saturate temperature 22 °C.

The slope of best fitting line is generally maintained constant at the heat flux less than 250 kW/m^2. It is 0.67 at lower heat flux. A sustainable decrease of slope for the tube is observed from the figures when heat flux is larger than 250 kW/m^2. The reasons might be that as the increasing of heat flux, the bubbles gradually change into vapor column and block. It is difficult for liquid inflow and vapor ejection outside the heating surface. It might have effects on the heat transfer.

The correlation presented by Lienhard et al. [11] is used to calculate the reference critical heat flux of R134a:

$$q_{\max} = 0.149 r \rho_v^{1/2} [\sigma g(\rho_l - \rho_v)]^{1/4} \quad (7)$$

At the saturation temperatures of 6 °C and 22 °C, the reference critical heat flux for refrigerant R134a are respectively of 412.2 and 460.8 kW/m^2. The highest heat flux in the investigation for plain tube is 368 kW/m^2, which is approaching the critical heat flux. It indicated that the deviation of Cooper's correlation and experiment result appears to be associated with the instabilities of bubble column rising from the tube surface. This equation might need some revisions to compensate for this effects, because Cooper's correlation is derived basically from the data with the heat flux in the range of 1–100 kW/m^2 [1,2].

5. Conclusions

The pool boiling heat transfer of R134a outside plain tube at higher heat flux is tested at different saturation temperatures. The major findings are as follows:

(1) At the higher heat flux greater than 250 kW/m^2, a larger deviation of experimental data and Cooper's correlation is observed.
(2) The slope of best-fitting straight line for pool boiling heat transfer coefficient versus heat flux is mostly of 0.67 at heat flux less than 250 kW/m^2 for plain tube. A sustainable decrease of slope is observed when heat flux is larger than 250 kW/m^2.
(3) The valid range of heat flux for Cooper's correlation to predict the pool boiling heat transfer of R134a should be 1–250 kW/m^2.

Conflict of interest

None declared.

Acknowledgments

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