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Experimental study of surface effect on flow boiling heat transfer in horizontal smooth tubes

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Abstract

In the present study the experiments on flow boiling heat transfer in two horizontal smooth copper tubes are carried out with pure refrigerants HFC134a, HCFC123, HCFC22, CFC114 and CFC12 using water-heated double-tube type test sections. One test tube is of 8.4 mm inside diameter, and the other is of 7.9 mm inside diameter. For both tubes the surface roughness is measured and the surface micro-geometry is observed with SEM. From the experimental results, the surface effect on flow boiling heat transfer is clarified and a correlation for the heat transfer coefficient is proposed in consideration of the surface effect. This correlation equation agrees well with the experimental data from different sources.

Nomenclature

C_1, C_2 constant
C_p isobaric specific heat [J kg^{-1} K^{-1}]
D_b bubble departure diameter [m]
D_{be} equilibrium break-off diameter [m]
d_i inner diameter [m]
F two-phase convection multiplier factor [-]
G mass velocity [kg m^{-2} s^{-1}]
h_{fg} latent heat of evaporation [J kg^{-1}]
M molecular weight [kg kmol^{-1}]
Nu Nusselt number
P pressure [Pa]
Pr Prandtl number
p_r reduced pressure
Q heat transfer rate [W]
q heat flux [W m^{-2}]
R_c nucleate cavity size [μm]
R_p roughness in Glättungstiefe [μm]
Re Reynolds number
S suppression factor
T_{wi} inside wall temperature [K]
T_{RC} calculated refrigerant temperature [K]
x quality
W mass flow rate [kg s^{-1}].

Greek symbols

α heat transfer coefficient [W m^{-2} K^{-1}]
λ thermal conductivity [W m^{-1} K^{-1}]
ΔL effective heating length of a subsection [m]
μ dynamic viscosity [Pa s]
ρ density [kg m^{-3}]
σ surface tension [N m^{-1}]
χ_{tt} Martinelli parameter $\left(=\left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1}\right)$.

Subscripts

c critical point
cal calculated
cv convective flow
exp experimental
in inlet of test section
l liquid
lo liquid only flow
nb nucleate boiling
R refrigerant
s source water
sat saturated
tp two-phase
v vapor.

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1. Introduction

Many experimental results showed that the surface condition is a very important factor in nucleate pool boiling. This factor affects strongly on not only the incipient boiling but also the nucleate boiling heat transfer coefficient. Based on the force balance on a vapor bubble and the Clausius–Clapeyron equation, a criterion equation, in which the active nucleate cavity size controls the relation between heat flux and wall superheat, was developed for the incipient boiling. Considering the effect of temperature gradient in liquid near the surface and the wall superheat, Han and Griffith [1] obtained a relation between the active cavity size and fluid properties. They found that the bubble growth rate is influenced by the thermal layer thickness and the critical wall superheat relation for the cavity. Singh et al. [2] developed a theoretical model to determine the required superheat for stable boiling from a cavity with consideration of the liquid penetration, the effect of transient heat flux and the inertial and the viscous effects of liquid moving inside the cavity.

For the nucleate boiling heat transfer coefficient, instead of nucleate cavity size, the surface roughness is often used to consider the effect of surface condition. Chowdhury and Winterton [3] studied the surface effects in pool boiling with particular emphasis on the roles of surface roughness and surface energy (measured by contact angle). They measured the surface roughness with a Talysuf-10 profilometer in terms of centerline average values. Through the experiments, they found that the nucleate boiling heat transfer appears strongly affected by surface roughness and almost unaffected by the contact angle value. To explore the effect of the surface roughness on boiling heat transfer, Nishikawa et al. [4] conducted various experiments using the refrigerants of CFC11, CFC12 and CFC114 in the range of reduced pressures from 0.08 to 0.9. They used paper to polish the boiling surface, and the surface roughness varied from 4.3 to 0.022 μm (centerline average values are approximately 2.2–0.024 μm). From over 300 data points, they noted that the surface roughness has greater effect at lower reduced pressure. They correlated their experimental data using graphical analysis and obtained

$$\alpha = 31.0q^{0.8}\frac{p_c^{0.2}}{M^{0.1}T_c^{0.9}}(8R_p)^{0.2-0.2p_r} \tag{1}$$

where R_p is the roughness in Glättungstiefe. In the above correlation, the effect of surface roughness is considered by the last term of $(8R_p)^{0.2-0.2p_r}$. Cooper [5] made a detailed review on nucleate pool boiling research, and proposed a correlation for pool boiling heat transfer using many experimental results of previous researches, as follows:

$$\alpha = 55q^{0.67}M^{-0.5}(-\log_{10}p_r)^{-0.55}p_r^{0.12-0.2\log_{10}R_p} \tag{2}$$

In his correlation, the effect of boiling surface condition is expressed as the function of the surface roughness R_p and the reduced pressure p_r. However, he did not give a clear definition on R_p.

Bergles and Rohsenow [6] made an analysis on the condition of incipient boiling. They pointed out that the incipient boiling should be independent of surface condition for most commercially finished surfaces due to the existence of a wide range of cavity sizes. Extending the above study, Davis and Anderson [7] proposed a criterion correlation for the incipient flow boiling which is referred to that of incipient pool boiling developed by Han and Griffth [1]. They reduced the critical cavity size as the function of pressure, heat flux and physical properties. From the measurements of the surface characteristics of a copper tube and the incipient boiling heat transfer data taken in that tube, they concluded that the active cavity size can be of the order of 1.0 μm under some normal engineering conditions. On the other hand, the relation between the cavity size and the wall superheat for a bubble is the same both in its generation and in its disappearance. The suppression of nucleate boiling in the annular flow would have the same characteristics as the incipience of nucleate boiling. In flow boiling, the wall superheat for a given heat flux decreases with increasing quality or mass flow rate. When the wall superheat falls below that required to form bubbles, nucleate boiling is fully suppressed. With this concept, Jung [8] found that the calculated suppression point is very close to the experimental data when the maximum cavity radius is set to 0.28 μm.

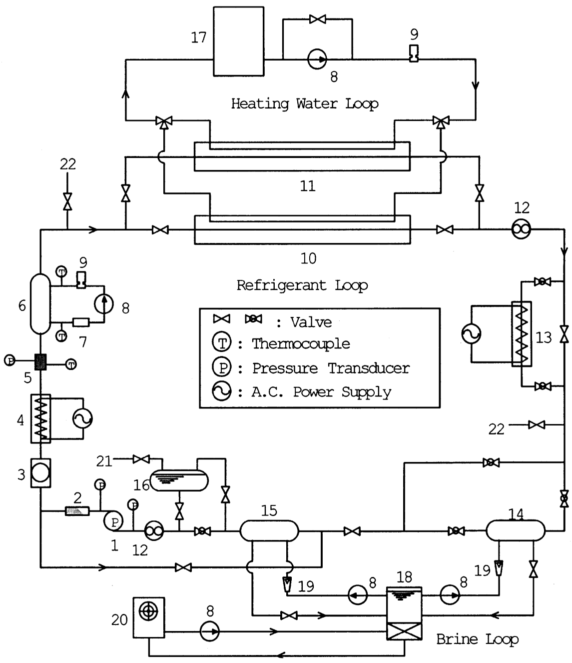
For flow boiling heat transfer characteristics, Chen [9] postulated that the heat transfer coefficient in tube can be expressed as a simple addition of forced convective flow and nucleate boiling parts. On this hypothesis, he developed a so-called 'Chen-type' correlation for in-tube boiling heat transfer characteristics. After that, Jung [8], Takamatsu et al. [10] and Yoshida et al. [11] proposed their correlations based on this hypothesis. On the other hand, Rhee and Young [12], Shah [13], Kandlikar [14] and Liu and Winterton [15] also proposed their correlations for boiling heat transfer characteristics, which are different in type with Chen's. All of these correlations only express the effect of mass flow rate, pressure, quality and heat flux on boiling heat transfer coefficient, but do not include the effect of surface condition.

In the present paper, the experimental study on flow boiling heat transfer in two horizontal smooth tubes is carried out using five kinds of pure refrigerants HFC134a, HCFC123 HCFC22, CFC114 and CFC12 as test fluid. From the present and many other experimental results, the surface effect on the flow boiling heat transfer is examined. Using the same model of Takamatsu et al. [10] and considering the effect of surface condition, a prediction correlation for flow boiling heat transfer coefficient is proposed.

2. Experimental apparatus and data reduction

2.1. Experimental apparatus

The experimental apparatus used in this study is shown in Fig. 1. It consists of three main loops: a refrigerant loop, a water loop and brine loop. In the refrigerant loop, the subcooled liquid refrigerant is delivered with a pump (1) through a desiccant filter (2), a mass flow meter (3), an electrical preheater (4), a mixing chamber (5) and a heat exchanger (6) to the test section (10) or (11). The refrigerant vapor generates in the test section, condenses in two condensers (14) and (15). The pump (1) used in the refrigerant loop is a positive-displacement pump that was used instead of a compressor to eliminate the effect of lubricating oil. The desiccant filter (2) is used to exclude the water dissolved in the refrigerant. The electrical preheater (4) and the heat exchanger (6) are used to adjust the thermodynamic state of refrigerant at the entrance of the test section. An after-heater (13) in the refrigerant loop is used to control the pressure in the refrigerant loop. The water loop, including a heat source tank (17), a centrifugal pump (8) and a gear-type flow meter (9), is used to supply heating water to the test section. The brine loop, which consists of a brine tank (18), a chilling unit (20), two centrifugal pumps (8) and two float-type flow meters (19), is used to condense the refrigerant. The temperature in the brine tank is usually set at -2.0°C in the flow boiling experiments.



1	Positive displacement pump	12	Sight glass
2	Desiccant filter	13	After-heater
3	Mass flow meter	14	Condenser 1
4	Preheater	15	Condenser 2
5	Mixing chamber	16	Liquid reservoir
6	Heat exchanger	17	Heat source tank
7	Heat sink tank	18	Brine tank
8	Centrifugal pump	19	Float-type flow meter
9	Gear-type flow meter	20	Chilling unit
10	Test section (Smooth tube)	21	Charging port
11	Test section (Microfin tube)	22	Sampling port

Fig. 1. Experimental apparatus.

The test section shown in Fig. 2 is a horizontal double-tube heat exchanger in which the refrigerant flows inside an inner tube, and the heating water flows in an outer annulus. In the test section, two straight smooth copper tubes A (8.4 mm I.D. and 10.0 mm O.D.) and B (7.9 mm I.D. and 9.5 mm O.D.), each of which is 6.0 m in length, are used as test tube. The outer tube is formed by two polycarbonate resin blocks each of which has a half-round ditch with a radius of 8 mm. To measure the local heat transfer rate, the annulus is divided into 14 subsections along the tube axis by brass blocks. In these subsections, nos 1, 2, 13 and 14 are 250 mm long, while other subsections are 500 mm long. The heat transfer rate in each subsection is treated as a local value. Depending on the requirement of the experiments, the flow condition in the double-tube heat exchanger can be set to counter or parallel flow by means of valves in the refrigerant loop.

2.2. Data reduction

The heat flux on the heat transfer surface in one subsection is calculated as

$$q = \frac{Q}{\pi d_{\mathrm{i}} \Delta L} \tag{3}$$

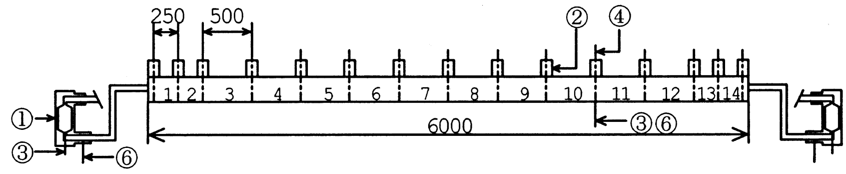
where $Q = C_{\mathrm{ps}} W_{\mathrm{s}} \Delta T_{\mathrm{s}}$. Q is the total heat transfer in one subsection. ΔL is the effective heating length of the subsection. C_{ps} is the isobaric specific heat of heating water. W_{s} is the flow rate of heating water. ΔT_{s} is the temperature difference of heating water in the subsection

The flow boiling heat transfer coefficient in the smooth tube is defined as

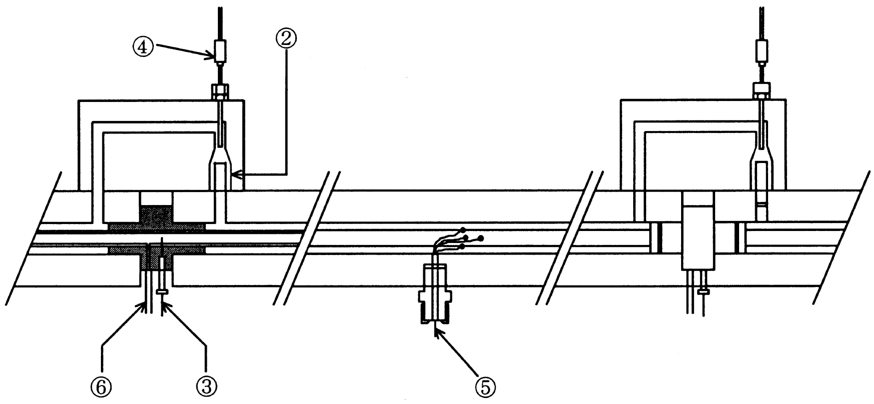
$$\alpha = \frac{q}{T_{\mathrm{wi}} - T_{\mathrm{RC}}} \tag{4}$$

where T_{wi} is the inner wall temperature, which is estimated from the measured outer wall temperature by one-dimensional heat conduction equation. T_{RC} is the saturated refrigerant temperature corresponding to the measured pressure.

The calculation method for heat transfer rate Q, inside wall temperature T_{wi}, refrigerant temperature T_{RC} and quality x in boiling condition is described in detail in [16]. The thermophysical properties used in the present study are estimated from the thermophysical property tables suggested by JAR [17–20]. The measurement error in the present study is about 0.1 K for refrigerant temperature



(a) Scheme of test evaporator.



(b) Detail of subsection.

1 Mixing chamber (Refrigerant)
2 Mixing chamber (Heating water)
3 Thermocouple (Refrigerant)
4 Resistant thermometer (Heating water)
5 Thermocouple (Tube surface)
6 Pressure measuring port

Fig. 2. Test section.

T_{RC} and 0.2 K for inter wall temperature T_{wi}. The uncertainty for heat flux q is less than 2%, and about less than 10% for heat transfer coefficient α. The experimental ranges are listed in Table 1. The experimental data in the present study were tabled in [16].

3. Experimental results

3.1. Surface condition of test tubes

In general, the surface condition is an important parameter in boiling heat transfer process. The nucleate cavity size controls the degree of superheat at the initial boiling point. The number density and the size of active nucleate cavity have a direct influence on heat flux in nucleate boiling region. In most previous studies, the surface roughness is often used to estimate the effect of surface condition on the nucleate boiling heat transfer. Figure 3(a) and (b) shows the typical results of surface roughness measured along the axial direction for tube A and tube B, where R_{max}, R_{cla}, R_{rms}, skewness and flatness are the maximum height roughness, the centerline average height roughness, the root mean square roughness, the moments of the third order and the moments of the fourth order, respectively. It is clarified from these figures that tube A is rougher than tube B.

Figure 4(a) and (b) illustrates the surface microgeometry observed by SEM (Scanning Electron Microscope) for the two tubes. It is found in Fig. 4(a) that the heat transfer surface of tube A can be divided into three areas: smooth area, rough area and some large cavity points. The smooth area looks like really smooth and almost no cavity on it. In the rough area, the cavity is small and shallow. For some large cavity points, the cavity size ranges from 3 to 6 μm. Some deep pits with the width of 5 μm are longer than 20 μm. For tube B, as shown in Fig. 4(b), no large cavity can be found on the surface, and many long pits are shallow and only of 1–2 μm in width.

Table 1
Experimental ranges

	Tube A				Tube B			
Refrigerant	G_R [kg m^{-2}s^{-1}]	$P_{R,in}$ [MPa]	x	q [kW m^{-2}]	G_R [kg m^{-2}s^{-1}]	$P_{R,in}$ [MPa]	x	q [kW m^{-2}]
HCFC22	300–312	0.57–1.12	0.0–1.0	5–89	223–361	0.67–1.14	0.0–1.0	4–55
HCFC123	100–310	0.25–0.46	0.0–1.0	2–60	—	—	—	—
HFC134	100–356	0.55–0.74	0.0–1.0	2–111	243–306	0.63–0.69	0.0–1.0	10–54
CFC114	—	—	—	—	222–349	0.43–0.45	0.0–1.0	3–40
CFC12	—	—	—	—	305	0.62	0.0–1.0	10–37

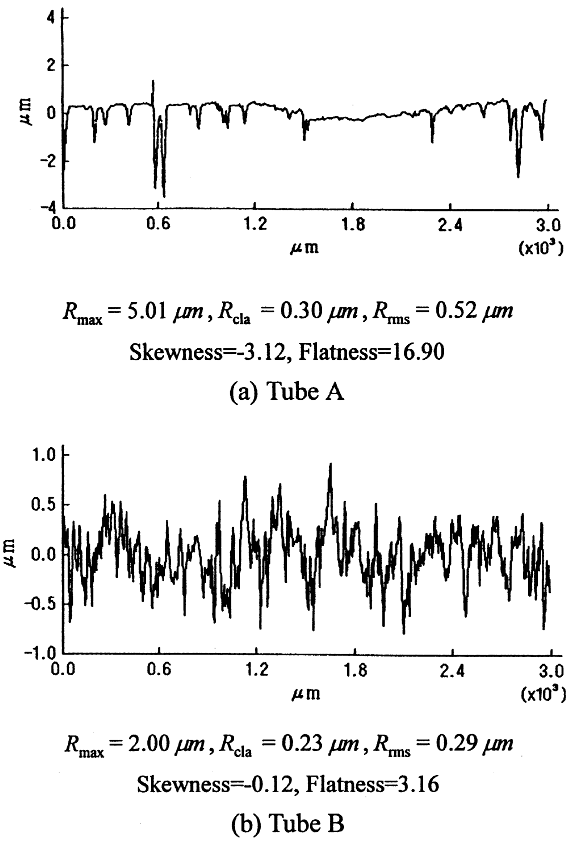


Fig. 3. Roughness of test surface.

3.2. Heat transfer characteristics

Figure 5(a) and (b) shows the comparison between the experimental results for two smooth tubes and the prediction of the correlation for tube B proposed by Takamatsu et al. [10]. It is found in these figures that for tube A the experimental results of HFC134a and HCFC22 are higher than the predicted values, while the experimental results of tube B agree well with the predicted values. The difference between tube A and tube B is only the surface condition. Other conditions, such as experimental conditions and ranges, are almost the same. Although the inner diameter of these two tubes is different the effect of difference in inner diameter on heat transfer coefficient is less than 1% according to the correlation of Takamatsu et al. [10].

Figure 6(a), (b) and (c) shows the comparison between the measured and predicted heat transfer coefficients of HCFC123, HFC134a and HCFC22 in the counter flow condition for tube A. The broken line and the dot-dash line represent the convective heat transfer component α_{cv} and the total heat transfer coefficient α_{cal}, respectively, both of which are calculated by correlation of Takamatsu et al. [10]. The difference between α_{cal} and α_{cv} denotes the calculated nucleate boiling component $\alpha_{nb,cal}$. The difference between α_{exp} and α_{cv} expresses the real nucleate boiling component $\alpha_{nb,exp}$ if the prediction of α_{cv} is reliable. The expression of α_{cv} is developed by Yoshida et al. [21] from their large data source and checked by Takamatsu et al. [10] with different experimental conditions. So we have enough confidence to its reliability. In Fig. 6(a), for HCFC123, it is revealed that the heat transfer coefficient is predicted well by the correlation of Takamatsu et al. [10]. This reason is that for HCFC123 the vapor velocity is higher and the liquid film is thinner, due to lower pressure and lower latent heat. Therefore, the nucleate boiling heat transfer component is suppressed in a marked degree and the convective heat transfer component is dominant. On the other hand, as shown in Fig. 6(b) and (c), for HFC134a and HCFC22, the nucleate boiling component takes an important part and is not fully suppressed in the high quality region under the counter-current flow condition. The nucleate boiling component of experimental data is much higher than the predicted value. As a result, the heat transfer coefficient for tube A is higher than that of tube B.

Based on the above consideration, we can get rid of the effect of all factors except surface condition, that is,

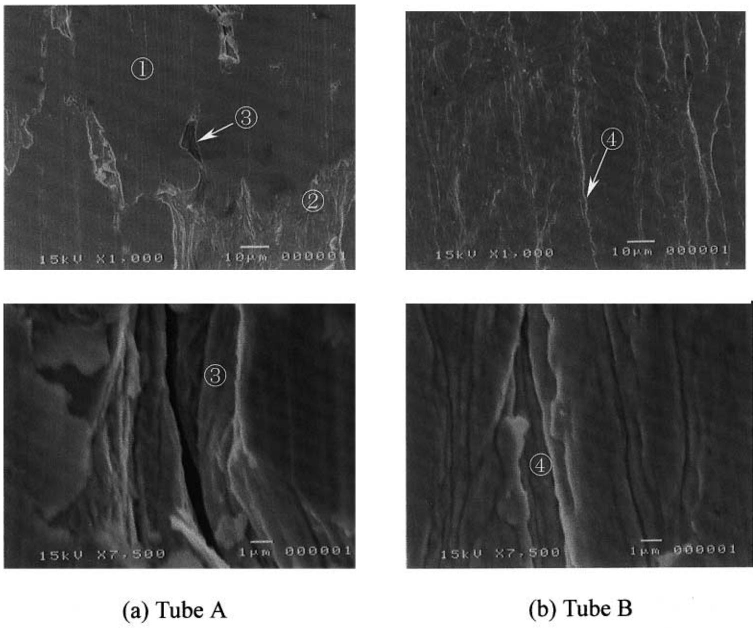


Fig. 4. Micro-geometry of test surface: 1. smooth area, 2. rough area, 3. large cavity, 4. long pit.

the difference of the flow boiling heat transfer characteristics in tube A and tube B is caused by the difference in surface condition of these two smooth tubes.

4. Proposed correlation on flow boiling

As mentioned before, there are several types of correlations for flow boiling heat transfer characteristics. In these correlations, the effect of surface condition is not considered. In this study, we select the correlation of Takamatsu et al. [10] that belongs to the Chen-type correlation as our model correlation, and supplement the surface effect in it. In common knowledge, the heat transfer coefficient is almost independent of the surface roughness for the forced convective heat transfer. On the other hand, for the nucleate boiling the heat transfer coefficient is sensitive to the surface condition, which was proved by many experiments under pool boiling condition and also found in the present experiment under flow boiling condition. For pool boiling, Nishikawa et al. [4] studied this effect systematically and proposed a correlation to consider this effect in pool boiling condition, as follows:

$$F_R = (8R_p)^{(0.2-0.2p_r)} \tag{5}$$

where R_p represents the roughness in Glättungstiefe. This value of R_p is usually not equal to nucleate cavity size, but it may be the same order as the nucleate cavity size for the artificial surface polished by emery papers. In the present study, due to the similarity between pool nucleate boiling and flow nucleate boiling, we introduce equation (5) into the nucleate boiling component to evaluate the effect of surface condition, in which R_p is replaced by the nucleate cavity size R_c. A new correlation including the surface effect is proposed as follows:

$$\alpha = \alpha_{cv} + \alpha_{nb} \tag{6}$$

$$\alpha_{cv} = 0.023 Re_{tp}^{0.8} Pr_l^{0.4}\left(\frac{\lambda_l}{d_i}\right) \tag{7}$$

$$Re_{tp} = F^{1/0.8} Re_{lo} \tag{8}$$

$$Re_{lo} = \frac{G(1-x)d_i}{\mu_l} \tag{9}$$

$$F = 1 + 2\chi_{tt}^{-0.88} \tag{10}$$

$$\alpha_{nb} = K^{0.745} S \alpha_{pb} \tag{11}$$

$$K^{0.745} = \frac{1}{1+0.875\eta+0.518\eta^2-0.159\eta^3+0.7907\eta^4} \tag{12}$$

$$\eta = \frac{\alpha_{cv}}{S\alpha_{pb}} \tag{13}$$

$$S = \frac{1}{\xi}(1-e^{-\xi}) \tag{14}$$

$$\xi = C_l\left(\frac{\rho_l}{\rho_v}\frac{Cp_l}{h_{fg}}T_{sat}\right)^{1.25} La\frac{\alpha_{cv}}{\lambda_l} \tag{15}$$

$$La = \sqrt{\frac{2\sigma}{g(\rho_l-\rho_v)}} \tag{16}$$

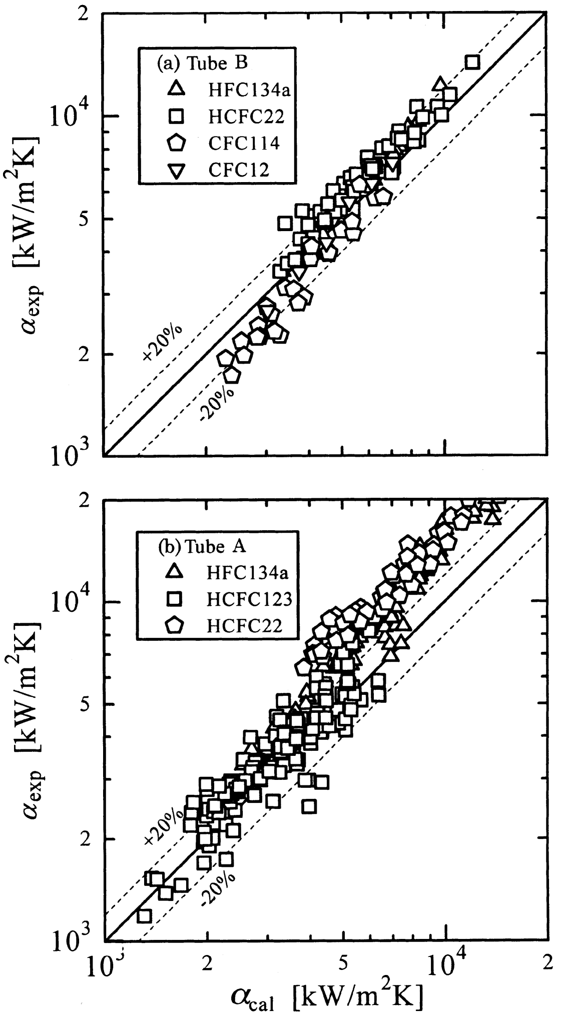


Fig. 5. Comparison between experimental results and correlation of Takamatsu et al.

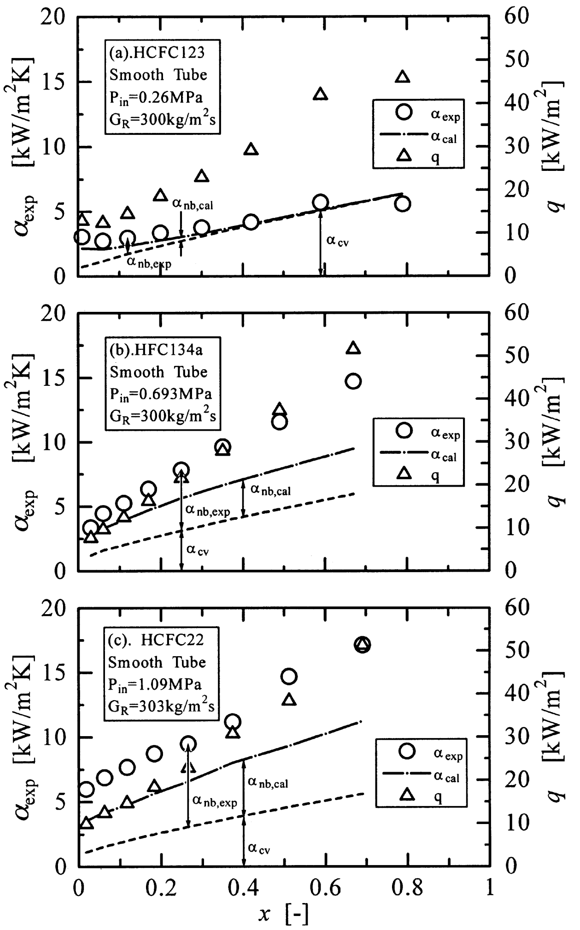


Fig. 6. Axial distribution of convective evaporation and nucleate boiling component.

$$\alpha_{pb} = C_2 \times 207 \frac{\lambda_l}{D_b}\left(\frac{qD_b}{\lambda_l T_{sat}}\right)^{0.745}\left(\frac{\rho_v}{\rho_l}\right)^{0.581} Pr_l^{0.533} F_r \tag{17}$$

$$D_b = 0.51 La \tag{18}$$

$$F_r = (8R_c)^{(0.2-0.2p_r)} \tag{19}$$

In the above correlation the experimental constants C_1 and C_2 are independent of the kind of refrigerant and the surface condition, and the nucleate cavity size R_c depends only on the surface condition of test tube. This cavity size is different from the measurement result by surface contact profile meter, and influenced by the tube material, manufacturing process and treatment before experiments. Since there are many different kinds and sizes of cavities on a real surface, it is difficult to determine a representative value for cavities from the direct observation by SEM. In the present stage, this nucleate cavity size R_c can be only determined from the experimental heat transfer data, and the measured value of R_c from the microphoto is used as a reference.

Based on the above criterions and the experimental data obtained by Altman et al. [22], Anderson et al. [23], Chaddock and Noerager [24], Jung [8], Yoshida [25], Khanpara et al. [26], Murata and Hashizume [27], Takahashi et al. [28] and the present study, the optimum values of the two constants are determined as $C_1 = 5.0 \times 10^{-5}$ and $C_2 = 1.25$. The estimated nucleate cavity size for each tube is listed in Table 2. It is found from these tubes that the nucleate cavity size R_c is 0.3 μm for six tubes, about 1.0 μm for three tubes, and over 2.0 μm for two tubes. It is noted here that the value of R_c for Jung's tube is estimated as 0.3 μm in the present study, very close to the 0.28 μm value that was found by Jung [8].

Table 2
Estimated nucleate cavity size for different tubes

Author	Tube	L [m]	d_i [mm]	G [kg m^{-2}s^{-1}]	P [MPa]	q [kW m^{-2}]	R_c [μm]
Altman et al. [22]	Copper	1.22×2	8.7	146–570	0.57–1.07	9–36	0.3
Anderson et al. [23]	Copper	2.286	16.9	202–350	0.57	8–27	0.8
Takahashi et al. [28]	Copper	—	7.9	194–455	0.56–0.62	25	0.3
Khanpara et al. [26]	Copper	3.81, 1.0	7.9	248–520	0.33	5–5.5	0.3
Murata-Hashizume [27]	Copper	—	10.3	200–300	0.2	10–30	1.0
Yoshida [25]	Copper	3.0, 4.0	11.0	300–500	0.59	10–40	0.3
Chaddock–Noerager [24]	SUS	1.93	11.7	351–587	0.44	7–35	0.3
Yoshida [25]	SUS	—	10.6	200–400	0.59	20	2.0
Jung [8]	SUS	4.0×2	9.1	240–750	0.25–0.67	10–45	0.3
Present (tube A)	Copper	6.0	8.4	150–350	0.25–1.12	2–111	3.5
Present (tube B)	Copper	6.0	7.9	2.20–360	0.43–1.14	3–54	1.0

Figure 7 shows the comparison between the present experimental results and the previous correlation equation. It is found that most of the experimental data agree with the present correlation within $\pm 20\%$. For the present two smooth tubes, the nucleate cavity size R_c is 3.5 μm for tube A and 1.0 μm for tube B. These values are very close to the width of many larger pits observed by SEM.

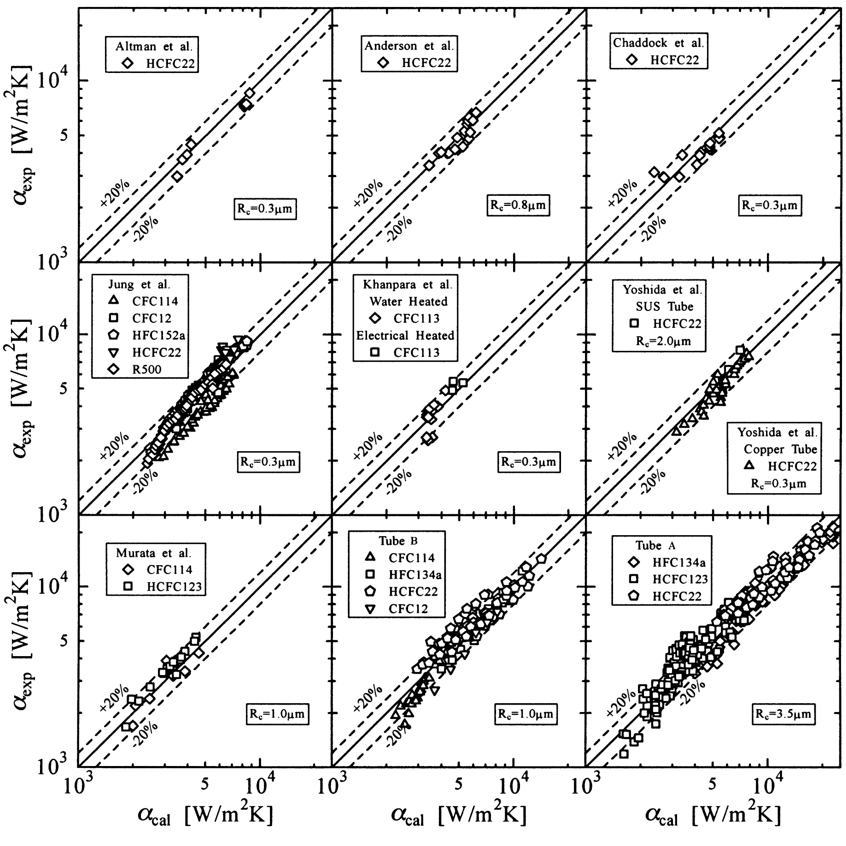


Fig. 7. Comparison of experimental and calculated results.

It should be pointed here that tube B is a normal commercial heat transfer tube ever widely used in air-conditioner, while tube A is a special heat transfer tube without final smoothing treatment. Using above results, we can design a simpler manufacture process for heat transfer tube. In the viewpoint of heat transfer, the last smoothing process is not necessary and can be cut down. The produced heat transfer tube with relatively large cavity size will have higher heat transfer performance, while the pressure drop is not increased [16].

5. Conclusions

1. The experimental studies on the flow boiling heat transfer in two smooth tubes were conducted using five kinds of refrigerants HFC134a, HCFC123, HCFC22, CFC114 and CFC12. The surface roughness was measured and surface micro-geometry was obtained with SEM for both tubes.
2. From these experiments the effect of surface condition on flow boiling was clarified. This effect comes from the nucleate boiling component and relates to the nucleate cavity size. This nucleate cavity size of boiling surface seems mainly controlled by the cavity width.
3. Based on the experimental data from the present study and other sources, a new correlation considering the surface effect was proposed for flow boiling in annular and semi-annular flow region. This correlation agrees well with the experimental data.
4. Relatively rough surface can lead higher heat transfer characteristics. In the viewpoint of heat transfer, the last smoothing process in manufacturing heat transfer tube is not necessary, and cheaper tube with higher performance can be produced.

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