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AN INVESTIGATION OF CRITICAL HEAT FLUX AND SURFACE REWET IN FLOW BOILING SYSTEMS

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Abstract—A high thermal capacity copper test section placed at the end of a heating tube is used for investigating the critical heat flux and heat transfer characteristics during rewetting process for Freon-113 upward flow at mass velocities of 357, 627, 1035 and 1465 kg m^{-2} s^{-1}. The critical heat flux coincides well with the maximum heat flux obtained by the transient cooling test. The wall superheat at the onset of rewetting changes greatly with the inlet quality, and ranges from 120 to 140°C in the subcooled and low quality region and from 60 to 80°C in the high quality region.

NOMENCLATURE

a	thermal diffusivity
c_p	specific heat
D	diameter
G	mass velocity
g	acceleration of gravity
H_{fg}	heat of vaporization
h	heat transfer coefficient
K	overall heat transfer coefficient
k	thermal conductivity
L	length of test section
L_e	characteristic length
p	pressure
Q	heat content of test section per unit length
q	heat flux at inner surface
r	radial co-ordinate
T	temperature
T_a	atmospheric temperature
$T_m(t)$	recorded temperature
T_s	saturation temperature
T_w	wall temperature at inner surface
$\Delta T_s, \Delta T$	wall superheat
t	time
x	quality

Greek symbols

λ_0	critical wavelength
μ	viscosity
ρ	density
σ	surface tension

Subscripts

c	critical heat flux condition
ex	exit end of test section
g	vapor phase
gf	vapor phase at film temperature
in	inlet end of test section
ins	inner surface of test section
l	liquid phase
max	maximum heat flux point
0	location of thermocouple
out	outer surface of test section
RW	rewet point

1. INTRODUCTION

KNOWLEDGE of heat transfer and flow structure in the region where the wall superheat has exceeded the critical heat flux condition is required in various engineering fields, such as steam generators, refrigerators and cryogenic liquid evaporators, and is especially important in the safety analysis of water-cooled nuclear reactors.

This region includes the transition boiling related to relatively low wall superheats and the stable film boiling or post-dryout dispersed flow for high wall superheats. The film boiling heat transfer under forced convection conditions [1–3] and the heat transfer mechanism in dispersed flow [4–6] have been investigated fairly well by many investigators. Study of forced convective transition boiling is, however, relatively limited due to the difficulty with measurements.

The transition boiling regime shows such a unique characteristic as the heat flux decreases with an increase in wall superheat that the boiling curve in this regime has been usually derived by means of a transient technique to record the wall temperature history in quenching. Ganic and Rohsenow [5] presented boiling curves in a region from high quality dispersed flow to transition boiling for liquid nitrogen. Cheng and Ragheb [7–10] performed an extensive measurement with high thermal capacity test sections and obtained boiling curves including transition boiling for subcooled water at near atmospheric pressure. They have shown that the boiling curves derived from the transient test are in good agreement with those from the steady-state test made with a temperature controller [11]. However, it should be noted that the experiments mentioned above were performed at low mass velocities, i.e. less than 300 kg m^{-2} s^{-1}. Green and Lawther [12, 13] have estimated the heat transfer characteristics associated with a change from pre-dryout to post-dryout conditions for Freon-12 upflow. In their study, the relationships between heat flux and wall superheat were derived from the transient wall temperature response initiated by increasing heat flux higher than the dryout condition.

In the transition boiling regime, the heated surface is wetted intermittently by the liquid flowing over it or the droplets depositing on it. Ragheb *et al.* [14] have indicated that the onset of intermittent wetting is very close to the minimum heat flux point obtained from the boiling curve and that the onset of continuous liquid contact coincides with the maximum heat flux point. The temperature boundaries of the transition boiling, i.e. the wall superheats at the critical or maximum heat flux point and the rewet point, are important factors for predicting the temperature–time history of highly superheated channels in quenching. However, little is known about the temperature boundaries and, as was pointed out by Groeneveld and Fung [15], further information is required on the rewetting in flow boiling systems.

One of the objectives of the present study is to examine the effects of mass velocity and vapor quality on the transition boiling boundaries. A high thermal capacity copper test section placed at the end of a heating tube is employed to obtain the boiling curve including transition boiling regime. Measurements are made on Freon-113 upward flow at pressures near 0.28 MPa. By the steady-state and transient tests, the critical heat flux and the heat flux vs wall superheat relation during the rewetting process are investigated in a wide range of mass velocity and quality combinations.

2. APPARATUS AND PROCEDURE

A schematic diagram of the test loop is shown in Fig. 1. The test liquid, Freon-113, was supplied by a circulating pump to a vertically arranged preheater

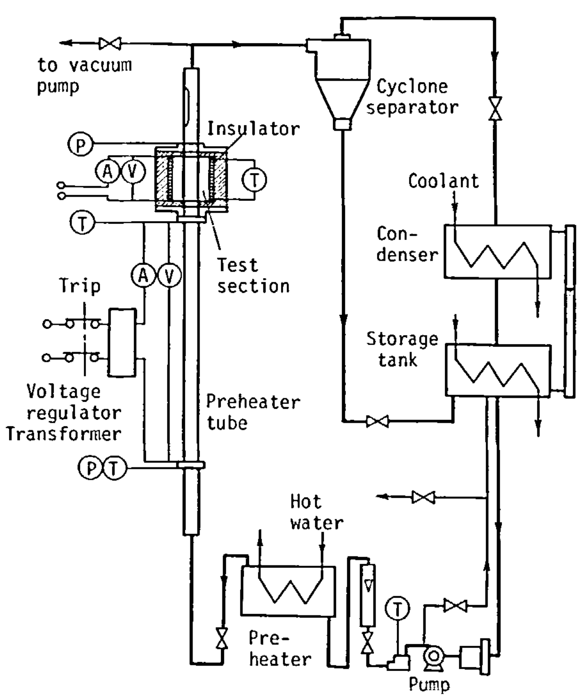

FIG. 1. Schematic diagram of experimental apparatus.

tube at a subcooled condition, flowed upwards through it and entered into the test section placed at the top of the preheater tube. The preheater tube is a uniformly heated 1.5 m long stainless-steel tube of 10 mm I.D. and 12 mm O.D. The test section is a cylindrical copper block of 10 mm I.D. and 100 or 98 mm O.D., and is heated by a sheathed heater wound round the outer surface.

Two copper blocks of heating length $L = 100$ and 60 mm were used. A total of nine and five C–A thermocouples were embedded as shown in Fig. 2. The copper block is surrounded by insulators and fixed up with bottom and top flanges. The unheated length between the preheater tube and the test section is 54 mm. For observation of the flow state, a glass tube section is provided in the upper tubing covering the length between points 72 and 142 mm away from the exit end of the copper block.

In this experiment, pressure and quality at the inlet of the preheater tube were kept at 0.32 MPa and -0.33, respectively. The inlet quality of the test section was controlled by adjusting the power input to the preheater tube. The pressure at the test section was then affected slightly by the mass velocity and inlet quality, but was in a range of 0.28 ± 0.015 MPa and the corresponding saturation temperature $T_s = 80$–84°C. Measurements were made for a steady-state heating run and a subsequent transient run.

In the steady-state run, the power input to the test section was increased in small steps while the steady temperature distribution in the copper block was measured. This procedure was repeated until a sharp rise in the copper block temperature was observed. Once the heat flux was increased over the critical heat flux, the copper block temperature began to rise sharply. The power input to the test section was reduced at this stage to settle the wall temperature down to about 250°C to prevent the inner surface fouling. After the temperature distribution in the copper block had settled down at a high temperature level, the transient run was started by cutting off the test section heater. The wall temperature history was recorded. Some studies have been made to examine under what conditions transient tests give data equivalent to those from steady-state boiling tests. Peyayopanakul and Westwater [16] performed careful transient tests for pool boiling with liquid nitrogen on a copper slab, and have shown that a slab thicker than 25 mm permits a quasi-steady-state determination of the whole boiling curve. The minimum slab thickness may change somewhat for flow boiling systems. However, since a very high thermal capacity copper block similar to the one used by Cheng *et al.* [7] is employed in the present test, its temperature variation during the transient test is expected to be slowed down enough to permit the quasi-steady-state determination.

The experimental ranges covered are:

mass velocity, $G = 357$–1465 kg m^{-2} s^{-1},
inlet quality, $x_{in} = -0.06$–0.62.

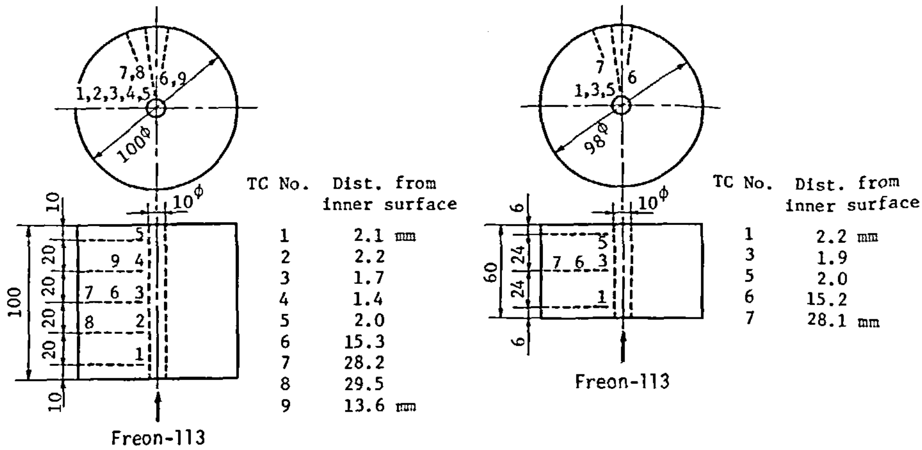

FIG. 2. Test sections.

Figure 3 shows a flow pattern map derived from Baker's chart [17] for Freon-113 at a pressure of 0.28 MPa. The experimental conditions for the test section of $L = 100$ mm are plotted on this map where the symbol ○ represents the quality at the inlet of the test section x_{in}, and the symbol × represents the quality at the exit under the critical heat flux condition

$$x_{exc} = x_{in} + \frac{4L}{D}\frac{q_c}{GH_{fg}}. \tag{1}$$

3. DERIVATION OF TEMPERATURE AND HEAT FLUX

The temperature and heat flux at the inner surface of the test section were derived by a 1-dim. analysis with the assumption that the axial heat conduction near the mid-plane was negligible. These values for the steady-state run can be easily obtained by applying two measured temperatures at the mid-plane of the test section to the steady-state heat conduction equation. The measured values of TC Nos. 3 and 7 were used for this purpose.

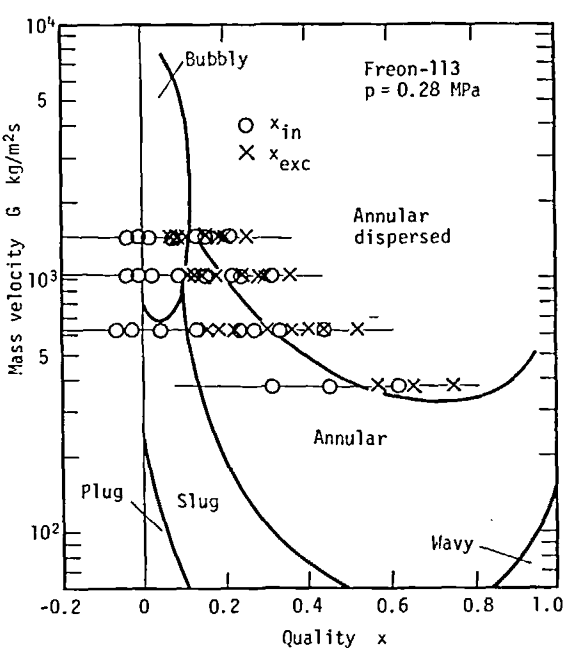

FIG. 3. Experimental range for test section $L = 100$ mm.

For analyzing the transient test results, the 1-dim. conduction model proposed by Cheng *et al.* [7], a method to compute the time-dependent temperature distribution of the test section from the recording of TC No. 3, was used.

The wall temperature $T(r, t)$ is obtained as the solution of the 1-dim. equation

$$\frac{\partial T}{\partial t} = a\left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r}\right). \tag{2}$$

The boundary conditions of the test section are

$$r = r_0: \quad T(r_0, t) = T_m(t), \tag{3}$$

$$r = r_{out}: -k(\partial T/\partial r)_{r=r_{out}} = K(T_{out} - T_a) \tag{4}$$

and the initial condition is

$$T(r, 0) = T_m(0) + \frac{r_{ins}q_{t=0}}{k}\ln\left(\frac{r}{r_0}\right). \tag{5}$$

In the above equations, r_0 and T_m are the location and reading of TC No. 3, respectively. Equation (4) shows the radial heat loss from the outer surface to the surroundings, where K denotes the overall heat transfer coefficient determined experimentally. Equation (5) represents the steady-state temperature distribution at the starting point of the transient run $t = 0$, where r_{ins} is the radius of the inner surface and $q_{t=0}$ is the inner surface heat flux at the time. The solution of equation (2) was obtained, as described in the Appendix, by reducing it to a series of finite differential equations in T.

The heat transfer rate from the inner surface to the fluid Freon-113 in a time interval Δt is assumed equal to the change in heat content of the test section, minus the radial heat loss from the outer surface during the same time interval. The heat content involved in unit length of the test section is

$$Q = \rho c_p \int_{r_{\rm ins}}^{r_{\rm out}} 2\pi r T \, dr. \tag{6}$$

Therefore, the heat flux q at the inner surface can be obtained by the following equation:

$$q = -\frac{1}{2\pi r_{\rm ins}} \frac{\Delta Q}{\Delta t} - K \frac{r_{\rm out}}{r_{\rm ins}} (T_{\rm out} - T_a). \tag{7}$$

Cheng *et al.* [18, 9, 10] have computed the temperature distribution and heat flux based on a 2-dim. model considering axial heat conduction in the test section and compared the results with those obtained by the 1-dim. model. This comparison was made by applying the temperature recordings obtained for flow boiling of subcooled water. The results for the copper test section have shown that (a) the two mid-plane boiling curves are approximately the same, and (b) the mid-plane boiling curve represents the averaged boiling curve of the entire test section.

4. BOILING CURVE

Some examples of the boiling curve thus derived are presented in Fig. 4. These are the results obtained with the test section of $L = 100$ mm at a mass velocity $G \simeq 1035$ kg m^{-2} s^{-1} and the inlet qualities of -0.008, 0.150 and 0.319. The points shown by the open symbol in Fig. 4 are the data for the steady-state run, among which the highest heat flux point is regarded in this paper as the critical heat flux condition. The solid line represents the boiling curve for the transient run computed from the recording of TC No. 3 at the mid-plane. The curve in the nucleate boiling regime is in fairly good agreement with the data for the steady-state run. For reference, the boiling curves computed by the 1-dim. analysis applying the recordings of TC No. 1 at the inlet side and TC No. 5 at the outlet side are also shown in Fig. 4.

The transient test results show a trend to move with decreasing wall superheat, from the film boiling or post-dryout dispersed flow state for high wall superheats to the nucleate boiling state through the transition boiling regime with intermittent rewetting of the surface. Although the onset point of transition boiling is not necessarily clear on the boiling curve, it is assumed as shown in Fig. 4(a) to be the point where the fluctuation settles down and the heat flux begins rising sharply with decreasing wall superheat. It is called the rewet point. The maximum heat flux point is also determined from the boiling curve as indicated in Fig. 4(a).

In the subcooled or low quality region, the flow type for high wall superheats seems to be an inverted annular flow with film boiling. The flow state of the vapor film in this type is probably complicated and it seems difficult to estimate the heat flux. For comparison, the heat fluxes for simple film boiling conditions calculated by the following equation are also shown in Fig. 4(a). The heat transfer coefficient in laminar film boiling under saturated liquid flow has been obtained [1]. Its averaged value for a vertical surface of characteristic length L_e is expressed as

$$h_{\rm m} = C\left[\frac{k_{\rm gf}^3 \rho_{\rm gf}(\rho_1 - \rho_{\rm gf}) g H'_{\rm fg}}{L_e \mu_{\rm gf} \Delta T_s}\right]^{1/4} \tag{8}$$

where

$$H'_{\rm fg} = H_{\rm fg} + 0.68 c_{p{\rm gf}} \Delta T_s.$$

It has been known that the constant C in the above equation is 0.667 in the case of the vapor velocity at the vapor–liquid interface being zero, and 0.943 in the case of the shear stress at the interface being zero. Here, the heat flux $q = h_{\rm m}\Delta T_s$ was calculated by applying a mean value $C = 0.81$ to the above equation. The two lines drawn in Fig. 4(a) show the results, one is obtained assuming $L_e = L$, and the other is obtained by assuming L_e is equal to the critical wavelength

$$\lambda_0 = 2\pi\left[\frac{\sigma}{(\rho_1 - \rho_{\rm gf})g}\right]^{1/2}. \tag{9}$$

The heat flux by radiation is small, estimated to be 1–4 kW m^{-2} in this case, and is not included in the two lines.

In the high quality region, on the other hand, the flow type for high wall superheats is considered to be post-dryout dispersed flow. In the dispersed flow just behind the dryout point, evaporation of entrained droplets in the vapor flow is limited and the heat added is mostly utilized for superheating the vapor flow [6]. As shown in Fig. 4(c), there was little difference between the boiling curves derived by TC No. 1 and TC No. 3 for the high wall superheats. This suggests that the dryout takes place at or near the inlet of the test section in the case of high wall superheats. The line drawn in Fig. 4(c) represents the convective heat transfer rate $q_{\rm con}$ at the mid-plane of the test section calculated by

$$q_{\rm con} = h(T_w - T_g) \tag{10}$$

and a heat transfer coefficient for vapor flow

$$\frac{hD}{k_{\rm gf}} = 0.023\left\{\frac{GD}{\mu_{\rm gf}}\left[x_{\rm in} + \frac{\rho_g}{\rho_1}(1 - x_{\rm in})\right]\right\}^{0.8} \times \left(\frac{c_p\mu}{k}\right)_{\rm gf}^{0.4}\left[1 + \frac{1.25}{(z/D)^{1.34}}\right] \tag{11}$$

where T_g is the superheated vapor temperature at the mid-plane $L/2$ calculated by assuming all the heat added is used for heating the vapor phase, and z denotes the axial distance from the dryout point and is given by $z = L/2$. The calculated value is close to the experimental result in the high quality and high wall superheat region.

The boiling curves obtained by the recordings of TC Nos. 1 and 5 are in good agreement with the curve derived by TC No. 3 in the subcooled and low quality region. However, as is seen in Fig. 4(c), a considerable difference is observed in the transition boiling regime of the high quality region. This seems to suggest that the surface rewet in the high quality region propagates

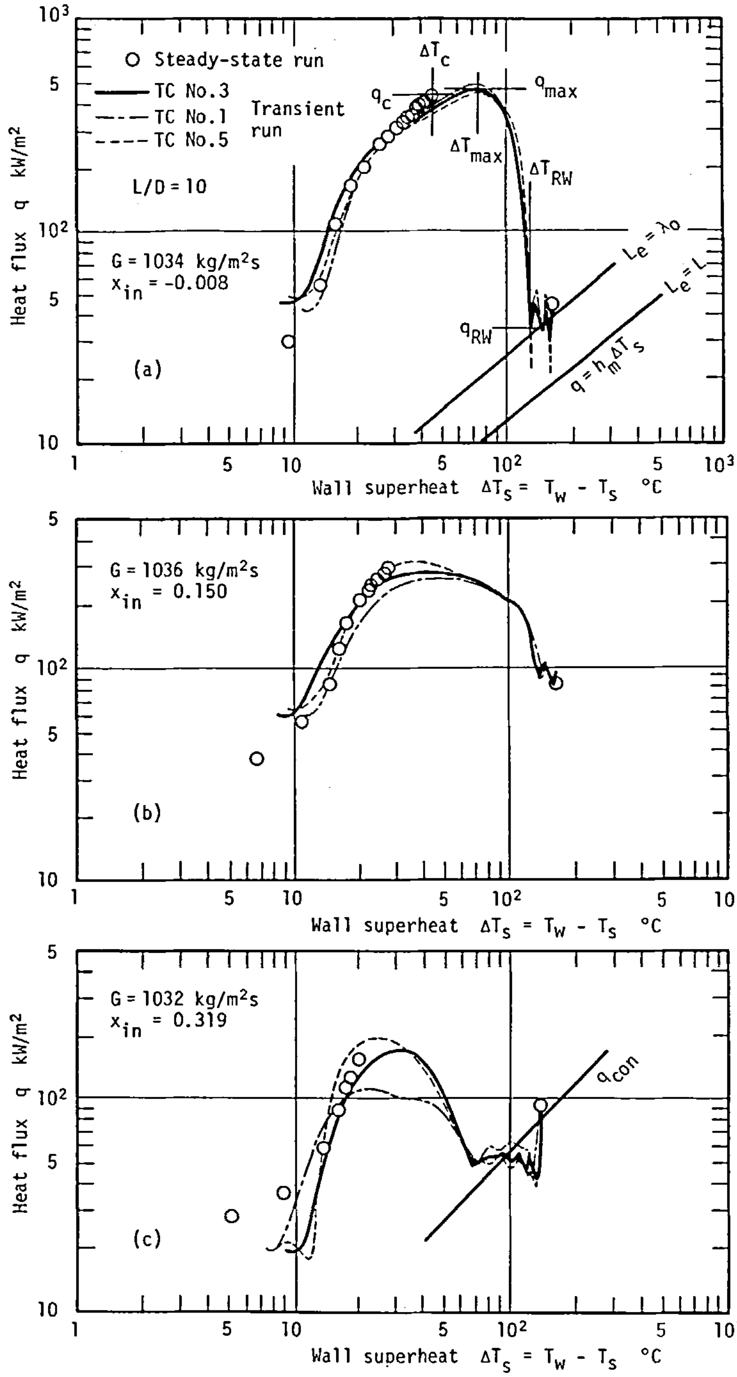

FIG. 4. Boiling curves.

downward slowly as the averaged temperature of the test section decreases, and that the axial heat conduction should be considered for obtaining the local point boiling curve in this process.

The boiling curve in the transition boiling regime is significantly affected by the inlet quality as is seen in Figs. 4(a)–(c). Figure 5 shows the boiling curves for the case of the inlet condition of the test section being slightly subcooled, and indicates that the heat flux in transition boiling increases with increasing mass velocity, in agreement with the results of Ragheb *et al.* [10].

Figure 6 shows a comparison of the critical heat flux q_c with the maximum heat flux q_{max} obtained by the transient test. Although the superheat at the maximum heat flux point ΔT_{max} is generally higher than that of the critical heat flux condition ΔT_c, the maximum heat flux is approximately equal to the critical heat flux.

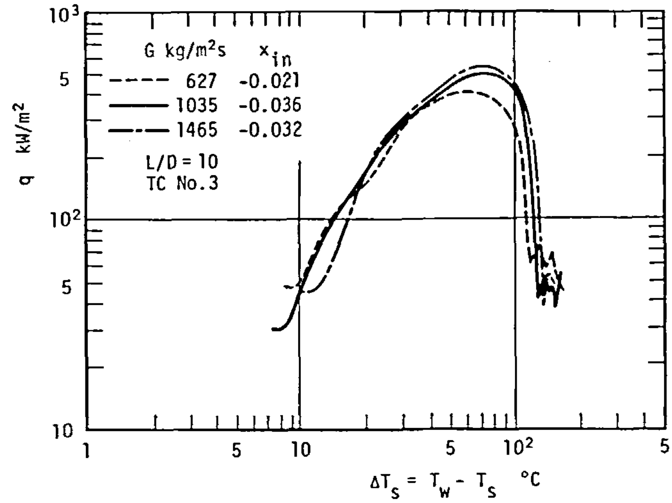

FIG. 5. Effect of mass velocity on boiling curve in subcooled region.

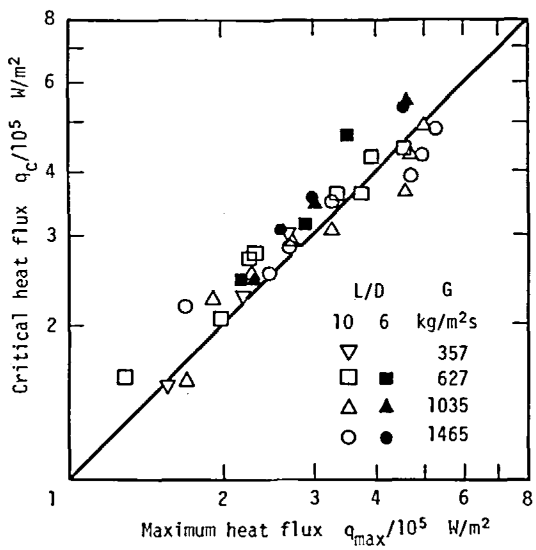

FIG. 6. Comparison of critical heat flux with maximum heat flux.

5. DISCUSSION OF RESULTS

5.1. *Critical heat flux*

Figure 7(a) shows that the critical heat flux is reduced with increasing inlet quality. The values for the test section of $L = 60$ mm are higher than those of $L = 100$ mm, and the increasing rate is raised as the inlet quality increases. Figure 7(b) represents the relation between the critical heat flux and the exit quality. A similar trend is observed to that obtained with a uniformly heated tube fed with subcooled liquid at its inlet.

The flow state observed through the glass tube section at the critical heat flux condition was as follows: When the exit quality was less than 0.15, the flow state was bubbly with many small bubbles. When the exit quality was higher than 0.20, the flow state was annular. The liquid film in the middle quality region was observed to flow along the surface periodically contacting the wall and then separating from it. This contrasts with the observations for the liquid film in the high quality region, which was accompanied by intermittent film disruption with a part of it being divided into rivulets.

5.2. *The rewet point*

The wall superheat at the onset of rewetting is affected strongly by the inlet quality. As shown in Fig. 8(a), the value ΔT_{RW} is ranged from 120 to 140°C in the subcooled and low quality region and from 60 to 80°C in the high quality region.

Although experimental data on the rewetting in flow boiling systems are limited and there exists a noticeable lack of knowledge of the rewetting mechanisms, Iloeje *et al.* [19] suggested three different controlling mechanisms, i.e. collapse of vapor film, axial conduction controlled rewet and dispersed flow rewet. It seems that the flow state has an important effect on the onset of rewetting. The flow state observed through the glass tube section at the starting point of the transient run was as follows: The flow in the subcooled and low quality region was a bubbly flow of large bubbles which appeared to be generated from the vapor film in the test section, whereas the flow state in the high quality region was dispersed. Figure 8(a) suggests that the wall superheat at the rewet point by collapse of vapor film is much higher than that by dispersed flow rewet. In the middle quality region, the rewet point was not clear and ΔT_{RW} was scattered in a wide range.

It has been shown that a droplet impinging on a hot surface rebounds away but a fraction of it begins to adhere to the surface as the wall superheat decreases below a boundary value [20]. The value of ΔT_{RW} in the high quality region approximately corresponds to the boundary value obtained with a Freon-113 droplet and copper system. Recently, Iloeje *et al.* [21] have

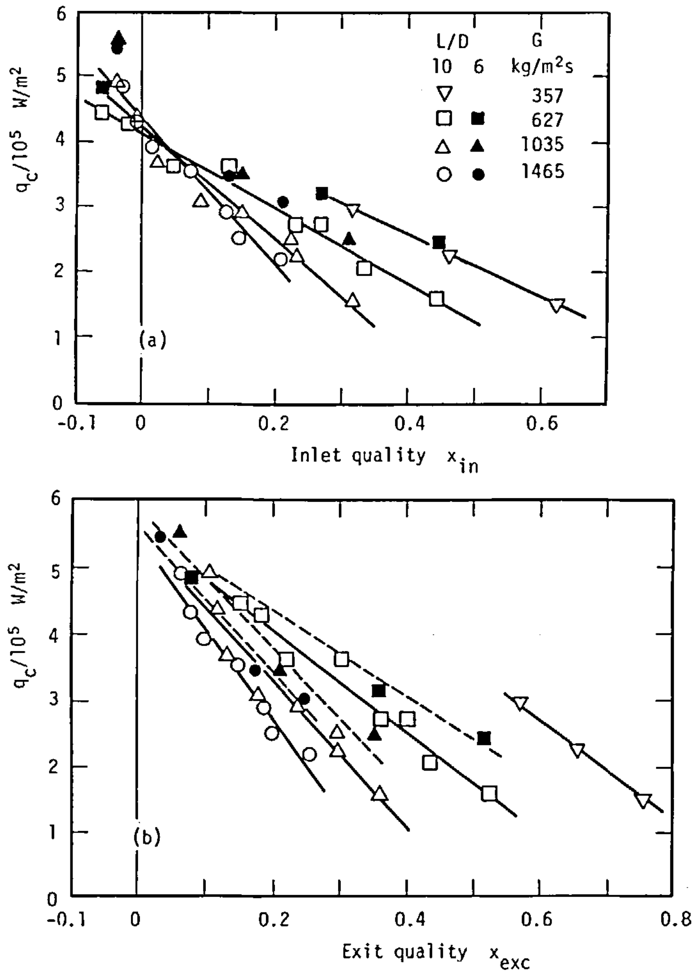

FIG. 7. Variation of critical heat flux with inlet quality and exit quality.

investigated the dispersed flow rewet with liquid nitrogen, and indicated that the rewet wall superheat decreases with increasing flow quality.

Along with ΔT_{RW}, the wall superheat at the maximum heat flux point ΔT_{max} is plotted in Fig. 8(a). Both wall superheats are insensitive to the length of the test section. Figure 8(b) shows the heat flux at the rewet point q_{RW} plotted against the inlet quality. The heat flux takes a peak value in the middle quality region and the value for $L = 60$ mm is considerably higher than that for $L = 100$ mm.

5.3. *Maximum heat flux point*

Figure 9 shows the wall superheats at the maximum heat flux point and the critical heat flux condition, ΔT_{max} and ΔT_c, plotted against the exit quality x_{exc}. ΔT_{max} increases with increasing quality in the low quality region, while it decreases with increasing quality and mass velocity in the high quality region. ΔT_c shows a trend to decrease with increasing exit quality and mass velocity in a manner similar to q_c shown in Fig. 7(b), but ΔT_c is little affected by the test section length.

6. CONCLUSIONS

The critical heat flux and the boiling curve for Freon-113 forced convective upflow were examined by a steady-state heating test and a subsequent transient test with a high thermal capacity copper test section. The conclusions derived from this study are as follows:

(1) The critical heat flux coincides well with the maximum heat flux obtained by the transient test.

(2) The heat flux vs wall superheat relation in the transition boiling regime is affected strongly by the inlet quality.

(3) The wall superheat at the onset of rewetting changes greatly with the inlet quality, and ranges from 120 to 140°C in the subcooled and low quality region and from 60 to 80°C in the high quality region.

(4) The wall superheat at the maximum heat flux point increases with increasing exit quality in the low quality region, while it decreases with increasing exit quality and mass velocity in the high quality region.

(5) The heat fluxes at the critical heat flux point, the maximum heat flux point and the rewet point are all increased with decreasing heating length, whereas the

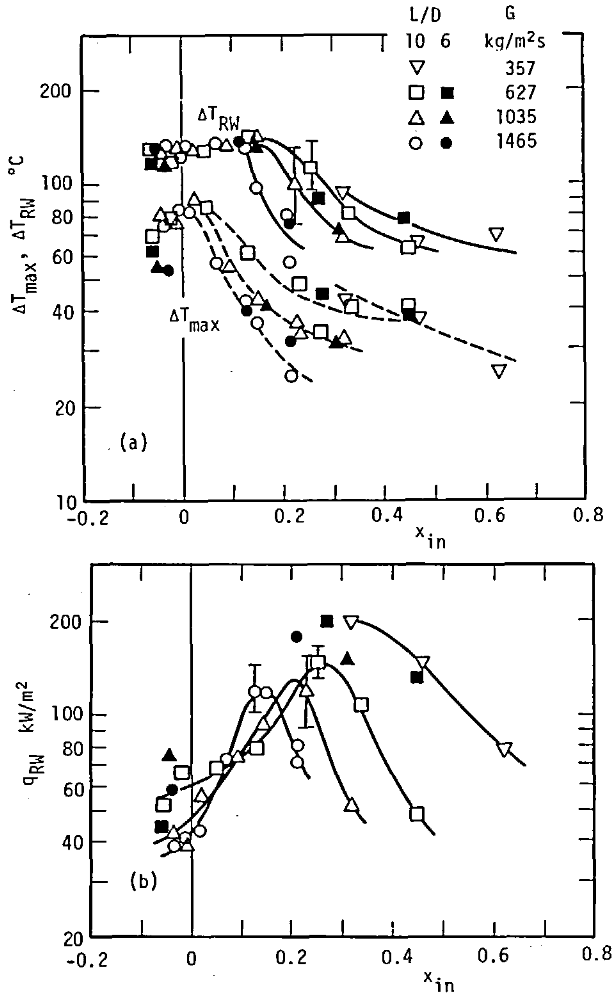

FIG. 8. Wall superheat and heat flux at rewet point.

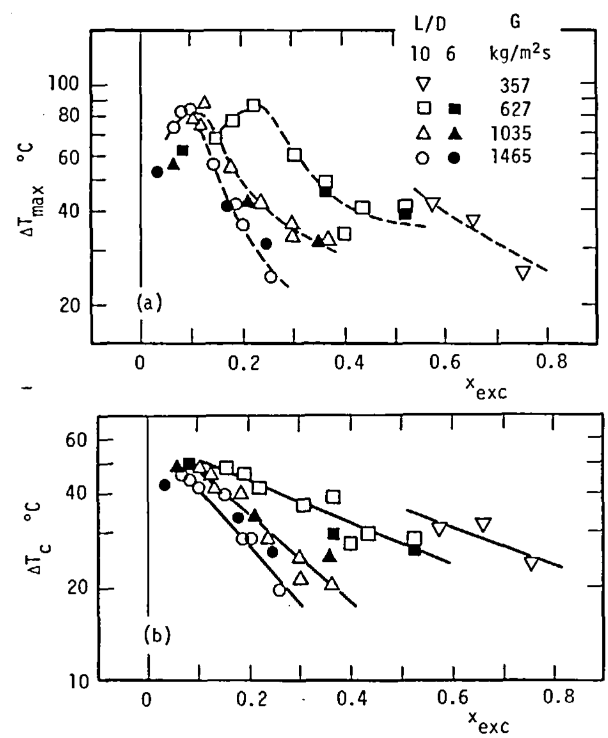

FIG. 9. Wall superheats at maximum heat flux point and critical heat flux condition.

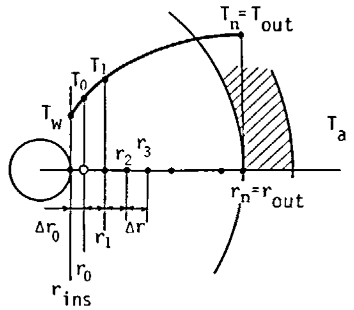

FIG. 10. Distribution of nodal points.

wall superheats at these points are little affected by the heating length.

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APPENDIX

Discretizing the test section from TC No. 3 location r_0 to the outer surface r_{out} as shown in Fig. 10, and denoting

$$r_i = r_0 + i\Delta r \quad (i = 0, 1, 2, \ldots, n)$$
$$t_j = j\Delta t \quad (j = 0, 1, 2, \ldots)$$
$$T(r_i, t_j) = T_{i,j},$$

equation (2) is reduced to the following difference equation:

$$\frac{T_{i,j} - T_{i,j-1}}{\Delta t} = a\left[\frac{T_{i+1,j} - 2T_{i,j} + T_{i-1,j}}{(\Delta r)^2} + \frac{1}{r_i}\frac{T_{i+1,j} - T_{i,j}}{\Delta r}\right].$$

Then,

$$T_{i-1,j} - \left[2 + \frac{\Delta r}{r_i} + \frac{(\Delta r)^2}{a\Delta t}\right]T_{i,j} + \left(1 + \frac{\Delta r}{r_i}\right)T_{i+1,j} = -\frac{(\Delta r)^2}{a\Delta t}T_{i,j-1} \quad (i = 1, 2, \ldots, n-1) \quad \text{(A1)}$$

where the temperatures $T_{0,j}$ are the measured values as shown in equation (3). The boundary condition, equation (4), is written in finite difference form as follows:

$$-k\frac{T_{n,j} - T_{n-1,j}}{\Delta r} = K(T_{n,j} - T_a).$$

Then

$$T_{n-1,j} - \left(1 + \frac{K\Delta r}{k}\right)T_{n,j} = -\frac{K\Delta r}{k}T_a. \quad \text{(A2)}$$

Equations (A1) and (A2) represent a system of n first-order equations with n unknowns $T_{i,j}$, and are readily solved when the temperatures $T_{i,j-1}$ are known. In the present case, the temperatures $T_{i,0}$ are known from the initial condition, equation (5), so that starting from $j = 0$ the temperatures T_i at each time increment can be obtained in turn. In this computation $n = 20$ and $\Delta t = 0.2$ s were selected. Once the values of T_1–T_n have been obtained, the inner surface temperature T_w can be determined from the following conduction equation at the recorded point, i.e. the nodal point $i = 0$:

$$\frac{dT_{0,j}}{dt} = a\left[\frac{(T_{1,j}-T_{0,j})/\Delta r-(T_{0,j}-T_{w,j})/\Delta r_0}{(\Delta r+\Delta r_0)/2}+\frac{1}{r_0}\,\frac{T_{1,j}^{*}-T_{0,j}}{\Delta r}\right] \tag{A3}$$

where, $dT_{0,j}/dt$ is the temperature gradient derived from the recording of TC No. 3.

ETUDE DU FLUX THERMIQUE CRITIQUE ET DU REMOUILLAGE DE LA SURFACE DANS LES SYSTEMES D'ECOULEMENT AVEC EBULLITION

Résumé—Une section de mesure en cuivre à forte capacité thermique, placée à l'extrémité d'un tube chauffé, est utilisée pour étudier le flux thermique critique et les caractéristiques de transfert thermique pendant le processus de remouillage, pour du Freon 113 en écoulement ascendant à des débits surfaciques de 357, 627, 1035 et 1465 kg m^{-2} s^{-1}. Le flux critique coincide bien avec le flux maximal obtenu par les essais de refroidissement transitoire. La surchauffe de la paroi à l'endroit du remouillage change beaucoup avec la qualité à l'entrée et elle est comprise entre 120 et 140°C pour la région à faible qualité, contre 60 et 80°C dans la région à forte qualité.

EINE UNTERSUCHUNG DER KRITISCHEN WÄRMESTROMDICHTE UND DER OBERFLÄCHENWIEDERBENETZUNG BEIM STRÖMUNGSSIEDEN

Zusammenfassung—Eine Teststrecke aus Kupfer mit hoher thermischer Kapazität ist am Ende eines beheizten Rohres angebracht. Sie wird zur Untersuchung der kritischen Wärmestromdichte und der Wärmeübertragungscharakteristiken bei Wiederbenetzungsprozessen verwendet, dabei strömt Freon-113 aufwärts mit Massenstromdichten von 357, 627, 1035 und 1465 kg m^{-2} s^{-1}. Die kritische Wärmestromdichte stimmt gut mit dem Maximum der Wärmestromdichte beim instationären Abkühltest überein. Die Wandüberhitzung bei Beginn der Wiederbenetzung ändert sich stark in Abhängigkeit vom Eintrittsdampfgehalt. Sie liegt im Bereich von 120 bis 140°C im unterkühlten Gebiet bei niedrigen Dampfgehalten und zwischen 60 und 80°C im Gebiet hoher Dampfgehalte.

ИССЛЕДОВАНИЕ КРИТИЧЕСКОГО ТЕПЛОВОГО ПОТОКА И ПОВТОРНОГО СМАЧИВАНИЯ ПОВЕРХНОСТИ В КИПЯЩИХ ПРОТОЧНЫХ СИСТЕМАХ

Аннотация—Для исследования критического теплового потока и характеристик теплопереноса в процессе повторного смачивания поверхности при восходящем течении фреона-113 с массовыми скоростями 357, 627, 1035 и 1465 кг $м^{-2}с^{-1}$ использовалась медная вставка с высокой теплоемкостью, помещенная в конце нагревающей трубы. Значения критического теплового потока хорошо согласуются с данными, полученными для максимального теплового потока в режиме неустановившегося охлаждения. На перегрев стенки в начале повторного смачивания оказывает существенное влияние паросодержание потока на входе. Перегрев изменяется в диапазоне от 120 до 140°C в области с недогревом и низким паросодержанием и в диапазоне от 60 до 80°C в области с высоким паросодержанием.