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REWETTING OF A HOT SURFACE BY A FALLING LIQUID FILM

TATSUHIRO UEDA, MITSURU INOUE, YASUHIRO IWATA and YOSHIYASU SOGAWA
Department of Mechanical Engineering, University of Tokyo, Bunkyo-ku, Tokyo, Japan

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Abstract—An experimental study is made on rewetting of a hot stainless steel tube by a Freon 113 liquid film at atmospheric pressure. The effects of liquid film flow rate, initial wall temperature and wall thickness on the wet front velocity are presented. Applying the temperature variation measured at the wall surface, a two-dimensional conduction equation in the wall is solved numerically to derive the boiling curve, i.e. the relation between the surface heat flux and the wall superheat. The result indicates that the intensive heat transfer to the liquid film behind the wet front is performed by transition and nucleate boiling, that the boiling curve is little affected by liquid film flow rate, initial wall temperature and wall thickness, and that the maximum heat flux is about twice the critical heat flux predicted for the pool boiling system.

NOMENCLATURE

a, thermal diffusivity [$m^2 s^{-1}$];
c_p, specific heat [$J kg^{-1} K^{-1}$];
g, acceleration of gravity [$9.8 m s^{-2}$];
H_{fg}, heat of vaporization [$J kg^{-1}$];
h, heat transfer coefficient [$W m^{-2} K^{-1}$];
k, thermal conductivity [$W m^{-1} K^{-1}$];
q, heat flux [$W m^{-2}$];
Re, liquid film Reynolds number, $4\Gamma/\mu_l$;
R_i, inside radius of tube [m];
r, radial coordinate [m];
T, temperature [°C];
T_s, saturation temperature [°C];
T_w, surface temperature [°C];
T_{wet}, rewetting temperature [°C];
T_{wi}, initial wall temperature [°C];
ΔT_w, wall superheat [°C];
t, time [s];
Δt, time interval [s];
U, wet front velocity [$m s^{-1}$];
Z, axial coordinate moving with the wet front [m];
z, fixed axial coordinate [m].

Greek symbols

Γ, liquid film flow rate per unit periphery [$kg m^{-1} s^{-1}$];
δ, wall thickness [m];
μ, viscosity [$kg m^{-1} s^{-1}$];
ρ, density [$kg m^{-3}$];
σ, surface tension [$N m^{-1}$].

Subscripts

l, liquid phase;
v, vapor phase;
w, tube wall.

1. INTRODUCTION

IN THE cooling of hot vertical surfaces by a falling liquid film, the wet front of the liquid film, which is accompanied by violent boiling and sputtering, flows down slowly as the surface ahead of it is cooled to the rewetting (or sputtering) temperature. The surface rewetting is a heat transfer process of fundamental importance in the emergency core cooling of water reactors in the event of postulated loss-of-coolant accidents.

Several experimental studies [1–4] have been made and the results show that the wet front of the liquid film moves along the surface at a constant velocity. Behind the wet front, in the wet region, the surface temperature drops rapidly to the saturation temperature due to the high heat transfer rate by boiling. The dry hot portion of the surface, ahead of the wet front, experiences in turn very poor heat transfer by vapor or vapor–mist flow. Therefore, the axial heat conduction in the wall towards the wet region plays an important role in the rewetting process.

Various 1- and 2-dim. models have been proposed for predicting the wet front velocity. Yamanouchi [3] has derived an analytical solution for the wet front velocity assuming that the heat conduction is 1-dim. in the wall and that the heat transfer coefficient is constant in the wet region and zero in the dry region. Thompson [5] presented numerical solutions of 1- and 2-dim. equations in the case where the heat transfer coefficient in the wet region was proportional to the cube of wall superheat. Dividing the wet region into two parts, Sun *et al.* [6] proposed an analytical model. Sun, Dua and coworkers [7, 8] included the effect of heat transfer in the dry region. Elias and Yadigaroglu [9] presented a computer-oriented 1-dim. conduction model in which the wet and dry regions were divided into several parts of constant heat transfer coefficient.

For predicting the wet front velocity by these analytical models, however, knowledge of the wall temperature at the wet front (the rewetting temperature) and the surface heat transfer or its axial distribution in the wet region are required. The heat transfer is considered to show a drastic change in a short distance behind the wet front. Dua and Tien [10] measured the surface temperature variation with a copper tube and liquid nitrogen system, and derived distributions of the surface heat flux behind the wet front by solving a 1-dim. equation. As for the reflooding

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process, boiling curves derived from a 1-dim. analysis were presented by Chen *et al.* [11] for the rewetting with subcooled water. However, data are limited and further investigations on the boiling phenomenon near the wet front are needed for understanding the rewetting process by a falling film.

In the present study, measurements are made of rewetting of a hot stainless steel tube by a fluorocarbon R-113 liquid film preheated near the saturation temperature. The effects of initial wall temperature, liquid film flow rate and wall thickness on the wet front velocity are investigated. Applying the temperature variations measured at the wall surface, relations between the surface heat flux and the wall superheat are derived by solving numerically the 2-dim. conduction equation in the wall. The boiling heat transfer behind the wet front is discussed and compared with the boiling curves predicted for pool boiling.

2. APPARATUS AND PROCEDURE

The experimental apparatus consists of three components, a test section, a liquid circulating system and an electric circuit for heating, as shown in Fig. 1. The test section arranged vertically in a glass tube of 124 mm I.D. is composed of a copper intake region of 150 mm long, a heating tube (SUS 304) of 16 mm O.D. and 400 mm long, and a copper tube silver-soldered to the lower end of the heating tube. The test liquid passes through a porous sintered tube provided at the upper end of the test section, flows down the outer surface of the intake region as a uniform film, and rewets the heating tube.

At the start of the experiment, the test liquid was circulated through a by-pass tube keeping the flow rate constant and heated so that the liquid subcooling becomes about 5 K at the mixing chamber. The heating tube was heated by passing an alternating current through it. When the heating tube reached an intended temperature, the heating current was shut off, and then, the liquid was supplied to the test section by turning the three-way cock. The test liquid used was R-113 at atmospheric pressure, the saturation temperature being 47.6°C.

For measuring the surface temperature of the heated tube, five C–A thermocouples were spot-welded to the outer surface at axial intervals of 20 mm. The leads of these thermocouples (100 μm O.D.) were taken to the inside of the test section, as shown in Fig. 1, through small holes drilled in the heating tube. The clearance of

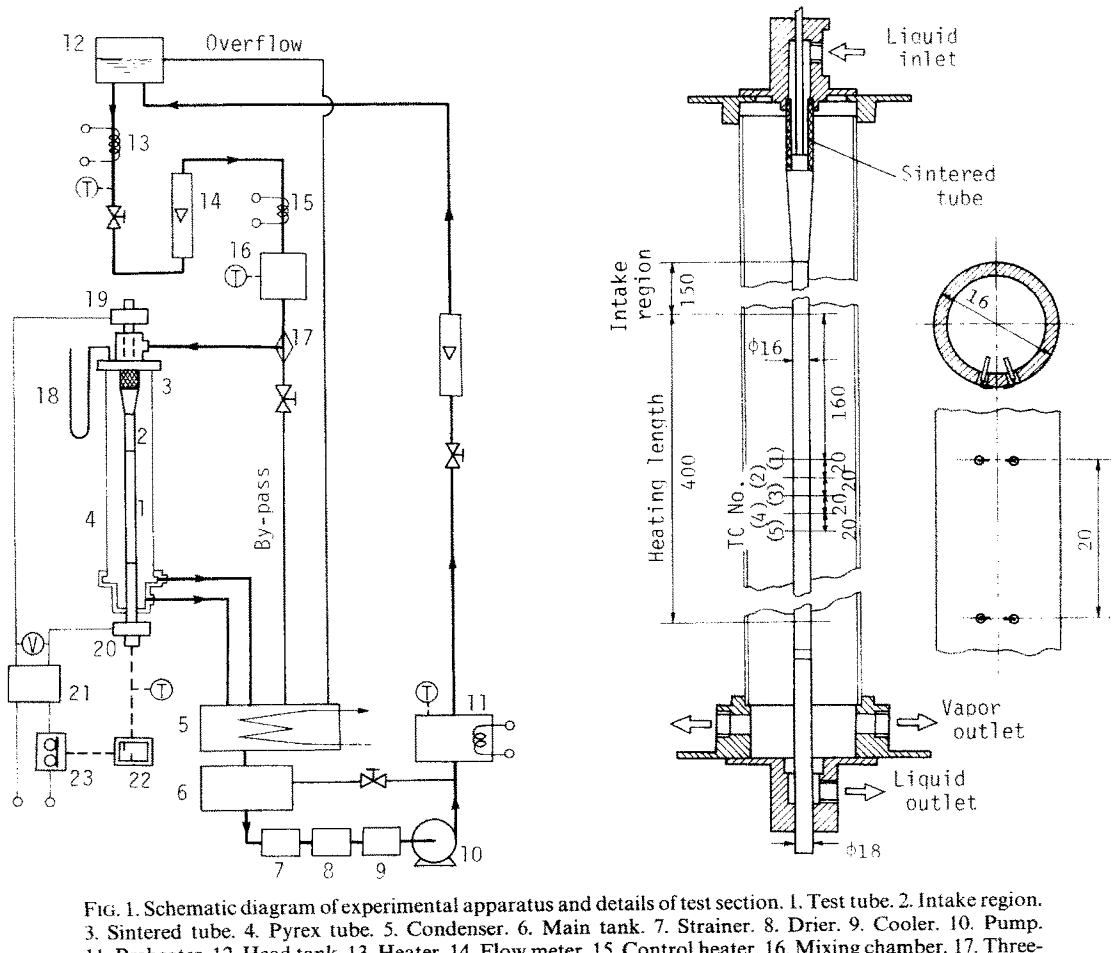

FIG. 1. Schematic diagram of experimental apparatus and details of test section. 1. Test tube. 2. Intake region. 3. Sintered tube. 4. Pyrex tube. 5. Condenser. 6. Main tank. 7. Strainer. 8. Drier. 9. Cooler. 10. Pump. 11. Preheater. 12. Head tank. 13. Heater. 14. Flow meter. 15. Control heater. 16. Mixing chamber. 17. Three-way cock. 18. Manometer. 19, 20. Electrodes. 21. Transformer. 22, 23. Excess temperature trip.

the holes was sealed up with sealing cement. The thermocouple outputs were recorded on an oscillograph chart.

Experiments were performed at six different liquid flow rates, the liquid film Reynolds number $Re = 700$, 1000, 2000, 3500, 5000 and 7000, in a range of the initial tube wall temperatures from 120 to 200°C. Four test sections of wall thickness $\delta = 0.5$, 1.0, 2.0 and 3.0 mm were used.

3. EXPERIMENTAL RESULTS

3.1. *Cooling (temperature–time) curve*

When the liquid film flows downwards on a hot surface, the wet front of the film progresses slowly accompanied by liquid film sputtering. Figure 2 shows states of boiling near the wet front of the falling film. The region of violent boiling was observed to extend its width with increasing the wall thickness δ, the width was 1–1.5 mm for $\delta = 1$ mm and 2–2.5 mm for $\delta = 3$ mm.

Figure 3 shows a record of temperature–time history of TC Nos. 1–4 located on the tube surface. Although fairly large fluctuations due to splashed films and droplets are observed in the thermocouple output, the decrease in wall temperature is gradual in the dry region. However, a sharp temperature drop takes place when the wet front reaches the thermocouple location. Inspection of correspondence between the variation of thermocouple output and the wet front position photographically observed showed that the moment of the wet front covers the thermocouple location coincided well with the start point of the sharp drop in wall temperature. Therefore, the temperature at this point was defined as the rewetting temperature T_{wet}. The output of the thermocouple located 20 mm below (TC No. 2) at this moment was regarded as the initial wall temperature T_{wi} in this experiment.

The wet front velocity was determined by

$$U = L/\Delta t \quad (1)$$

where Δt is the time interval taken for the wet front to travel the distance $L = 20$ mm between the location of TC No. 1 and that of TC No. 2.

The wall superheat at the rewet point $\Delta T_{wet} = T_{wet} - T_s$ is currently used as a matching parameter for correlating the experimental data on the wet front velocity. However, experimentally determined data on the superheat are limited. Figure 4 shows the superheat ΔT_{wet} obtained in this experiment. Although the values are scattered in the range 34–47 K, showing a slight trend to increase with increasing initial wall temperature, the average is about $\Delta T_{wet} = 40$ K, independent of the liquid film Reynolds number and the wall thickness. Dua and Tien [10] have reported the value $\Delta T_{wet} \approx 25$ K for a copper tube and liquid nitrogen film system. The present result is comparable with the value $\Delta T_{wet} \approx 30$ K obtained by Simopoulos

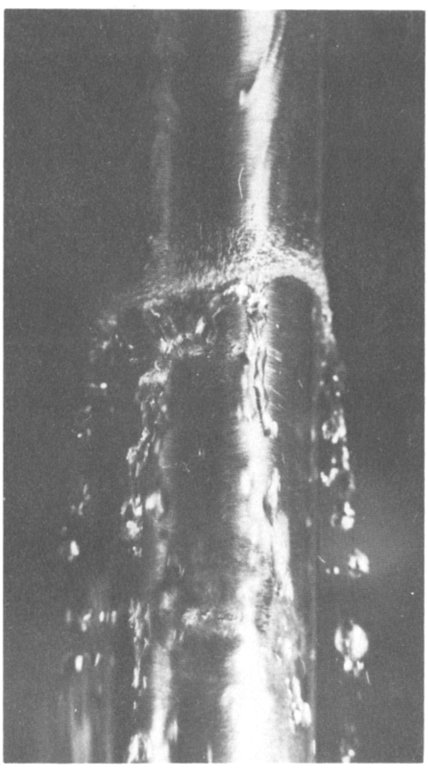

FIG. 2(a). $\delta = 1$ mm.

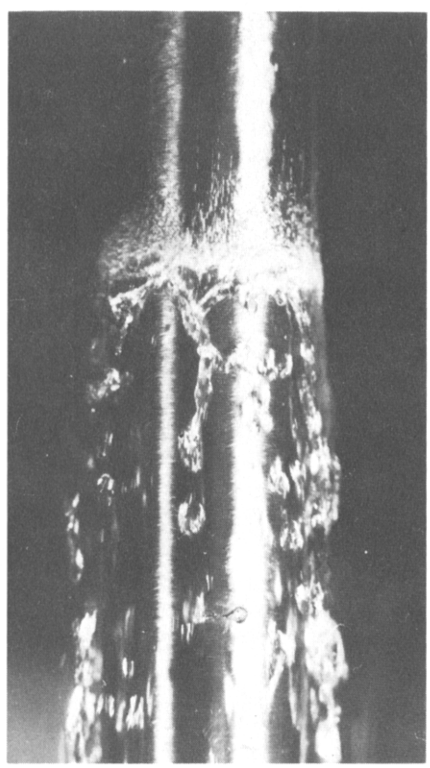

FIG. 2(b). $\delta = 3$ mm.

FIG. 2. Boiling of liquid film near the wet front, $Re = 2000$.

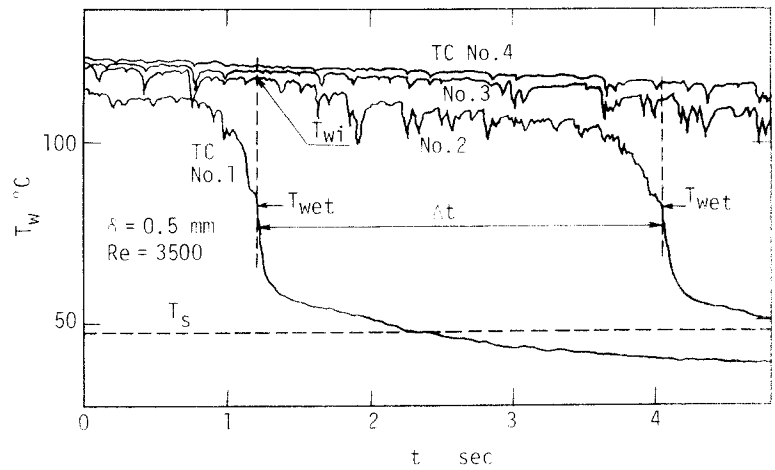

FIG. 3. Temperature–time history of thermocouples located on the tube surface.

et al. [12] from measurements of a stainless steel and R-113 film system carried out at 1.7–5.2 bar.

3.2. *Wet front velocity*

Figure 5(a) shows the wet front velocity U plotted against the initial wall superheat $\Delta T_{wi} = T_{wi} - T_s$. The wet front velocity decreases with increasing initial wall superheat, and is expressed approximately by $U \propto \Delta T_{wi}^{-1.6}$ as the lines drawn in the figure. The velocity decreases as the wall thickness is increased, but is only slightly affected by the film flow rate. The experimental results of the velocity U presented so far for subcooled liquid films have shown a trend to increase with increasing the flow rate [1, 3, 4]. However, the effect of the flow rate is not clear in the present data obtained with the film preheated near the saturation temperature, the same as the result of Bennett *et al.* [2].

Assuming the heat transfer coefficient is constant in the wet region and zero in the dry region, Yamanouchi [3] solved the 1-dim. equation and derived theoretically the wet front velocity as follows:

$$U = \frac{1}{\rho_w c_{pw}} \left(\frac{hk_w}{\delta}\right)^{1/2} / F(\Delta T), \tag{2}$$

$$F(\Delta T) = [(T_{wi} - T_s)(T_{wi} - T_{wet})]^{1/2}/(T_{wet} - T_s)$$

where h is the heat transfer coefficient on the surface in a wet region. Figure 5(b) shows the variation of the wet front velocity for the temperature term in the above equation, $F(\Delta T)$, obtained by substituting the rewetting superheat temperature, $T_{wet} - T_s = 40$ K. The data were well correlated with a form of $U \propto 1/F(\Delta T)$ as is indicated in equation (2). However, the relation of $U \propto 1/\delta^{0.5}$ given in equation (2) did not fit the experimental result. Figure 6 represents the variation of the parameters derived from Fig. 5 for the wall thickness. The effect of the wall thickness on the wet front velocity is not so simple as indicated by equation (2). This discrepancy can probably be ascribed to the over-simplified assumptions of Yamanouchi's model.

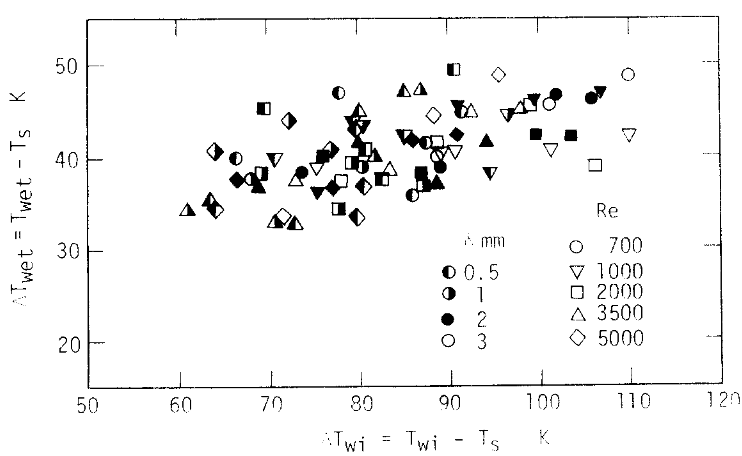

FIG. 4. Superheat at the rewet point.

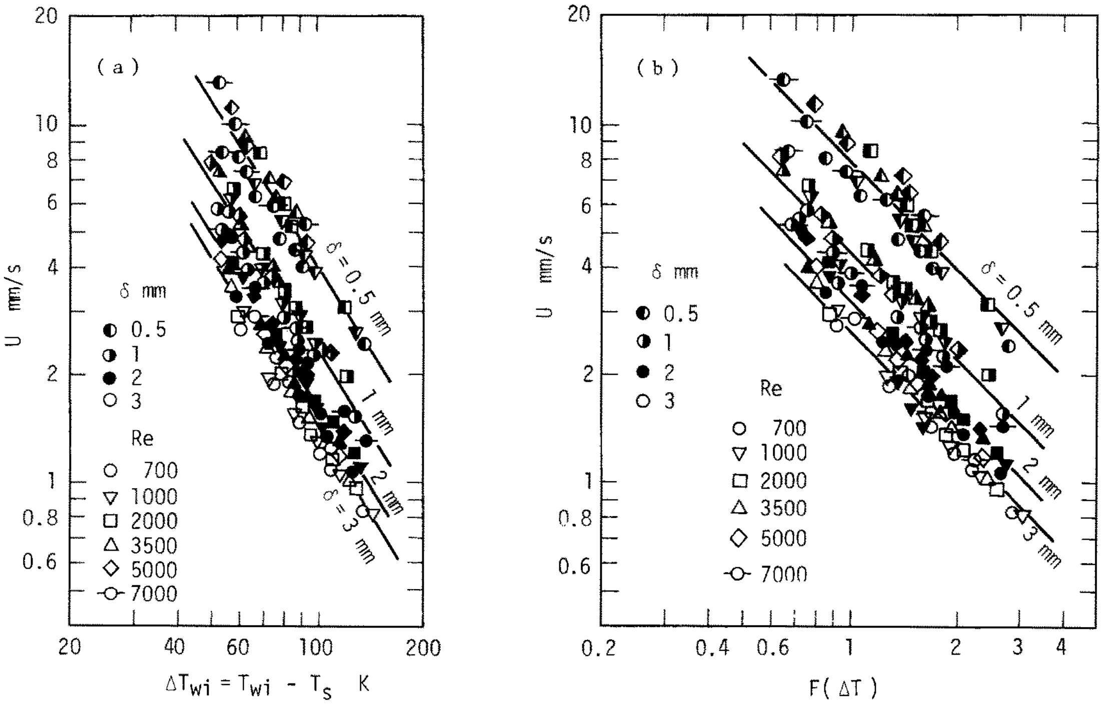

FIG. 5. Variation of the wet front velocity with initial wall temperature and wall thickness.

in which the temperature variation across the wall thickness is neglected and the surface heat transfer coefficient is assumed to be constant in the wet region. For reference, the value of h in equation (2) was calculated by applying the present experimental data for U, T_{wi} and T_{wet}. The heat transfer coefficient h thus obtained, was scattered in the range 15–40 kW m^{-2} K^{-1}, showing a trend to increase with decreasing δ and increasing Re.

4. ANALYSIS OF SURFACE HEAT FLUX

A physical model of the rewetting by a falling liquid film is shown in Fig. 7. Since the wet front moves at a constant velocity U, the surface temperature profile along the tube length is obtainable from the recorded temperature–time history. In this system, it is convenient to adopt a coordinate Z which moves with the wet front and has its origin at the wet front point,

$$Z = z - Ut. \tag{3}$$

The curve in Fig. 7 shows the surface temperature profile thus transformed to the Z coordinate, in which the value at $Z = 0$ represents the rewetting temperature T_{wet}. Here, based on the currently used assumption that the temperature distribution in the wall moves with the

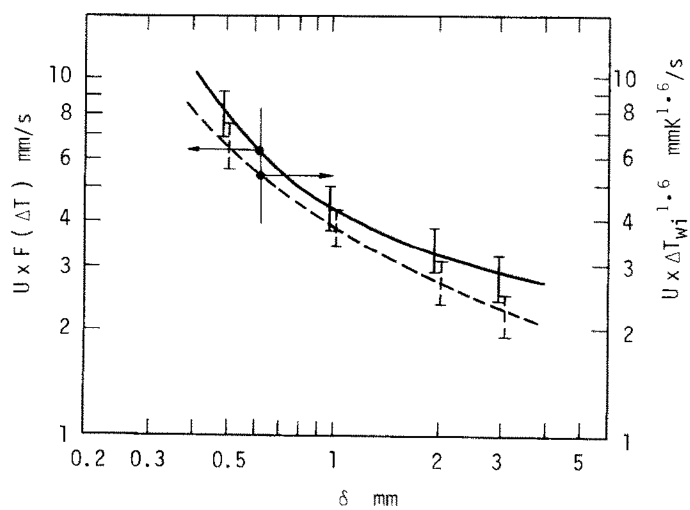

FIG. 6. Effect of wall thickness on the wet front velocity.

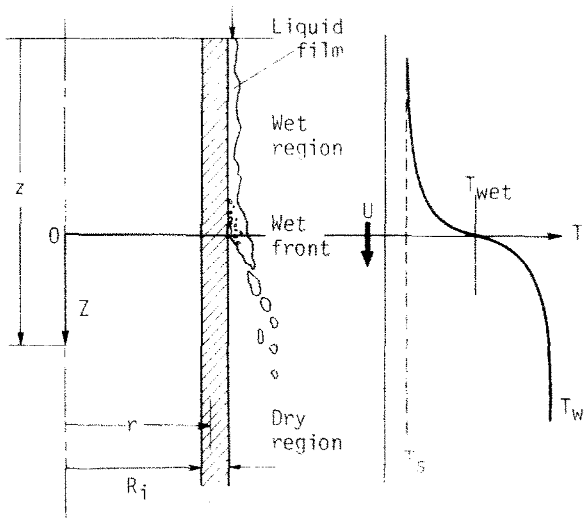

FIG. 7. Physical model of rewetting.

wet front without changing its profile, the surface heat flux was calculated by solving numerically the 2-dim. conduction equation in the wall.

Provided the tube wall is assumed to have a uniform temperature throughout its thickness δ, the problem becomes 1-dim. In the present experiment, the heat flux through the inner surface is zero, then, the outer surface heat flux q is expressed from an energy balance on a differential element of the tube as follows:

$$q = \delta\left(k_w \frac{\partial^2 T_w}{\partial z^2} - \rho_w c_{pw} \frac{\partial T_w}{\partial t}\right). \tag{4}$$

The space and time gradients are related as

$$\frac{\partial^2 T_w}{\partial z^2} = \frac{\partial^2 T_w}{\partial Z^2}, \quad \frac{\partial T_w}{\partial t} = -U \frac{\partial T_w}{\partial Z}. \tag{5}$$

Then, equation (4) is transformed to

$$q = \delta k_w\left(\frac{\partial^2 T_w}{\partial Z^2} + \frac{U}{a_w} \frac{\partial T_w}{\partial Z}\right). \tag{6}$$

Therefore, it is possible to calculate the surface heat flux from the surface temperature profile.

However, for evaluating the accurate heat flux, a 2-dim. analysis considering the wall temperature variation in the radial direction will be required. The 2-dim. conduction equation in the wall is

$$k_w\left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2}\right) = \rho_w c_{pw} \frac{\partial T}{\partial t}. \tag{7}$$

Then, the governing equation transformed to the Z coordinate is expressed as

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial Z^2} = -\frac{U}{a_w}\frac{\partial T}{\partial Z}. \tag{8}$$

The boundary conditions are

$$\begin{aligned} r = R_i: &\quad \partial T/\partial r = 0 \\ r = R_i + \delta: &\quad T = T_w(Z) \end{aligned} \tag{9}$$

where, $T_w(Z)$ denotes the surface temperature obtained by transforming the recorded temperature–time history to the Z coordinate. The recorded temperature fluctuated in the dry wall region, so that a smooth curve was approximated for evaluating $T_w(Z)$ in this region. There seems to be little influence arising from this approximation on the wall temperature distribution in the wet region.

In the present study, equations (8) and (9) were converted to finite difference equations as described in the Appendix, and computed with an iteration method to find the whole temperature distribution in the wall. The computation was terminated when the converged temperature distribution was obtained. The outer surface heat flux q was then derived from the wall temperature distribution thus obtained. The heat transfer coefficient on the surface defined by

$$h = q/(T_w - T_s) \tag{10}$$

was also calculated.

5. RESULTS AND DISCUSSION

The relations between the surface heat flux and the wall superheat (the boiling curves) derived from 1- and 2-dim. analyses are compared in Fig. 8. The curves of the 1-dim. analysis are shifted to higher wall superheats than those of the 2-dim. analysis, showing higher maximum heat fluxes. The difference in the maximum heat flux increases with increasing the wall thickness, the value by 1-dim. analysis being about three times higher than that by 2-dim. analysis, in the case of δ = 3 mm. This trend indicates that the wall temperature gradient in the radial direction cannot be neglected and that the 2-dim. analysis is necessary for evaluating the surface heat flux near the wet front accurately.

The boiling curves derived from the 2-dim. analysis are presented in Fig. 9. Figure 9(a) shows the result of test runs performed at different film flow rates, and Fig.

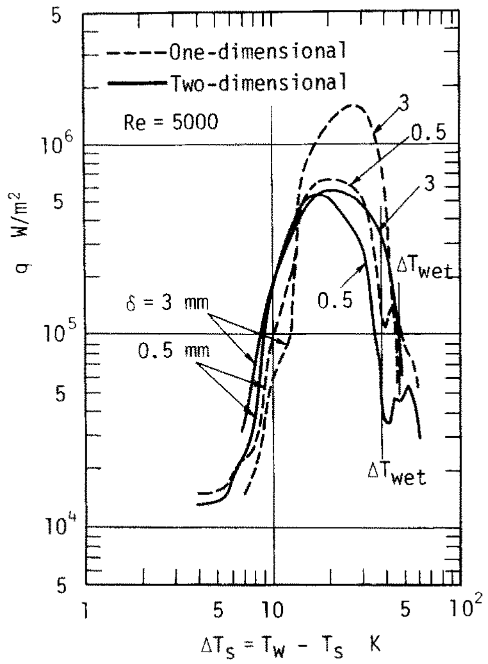

FIG. 8. Boiling curves (comparison of 1- and 2-dim. analyses).

9(b) the result obtained for different values of the initial wall temperature. It should be noted that the boiling curve is considered to be independent of the film flow rate and the initial wall temperature, and that, as shown in Fig. 8, the boiling curve derived from the 2-dim. analysis is little affected by the wall thickness. In the present test with a stainless steel and R-113 film system, the maximum heat flux appeared near $\Delta T_s = 25$ K, and its value was about 5–6 × 10^5 W m^{-2}.

As is seen in Fig. 9(a), the present result was compared with some correlations proposed for pool boiling. Nishikawa and Fujita [13, 14] presented a general correlation for nucleate boiling heat transfer to fluorocarbon refrigerants. The dotted lines written in the nucleate boiling region represent the relations obtained by assuming the nucleation factor $f_\zeta = 4.0$. In the present experiment, the region of violent boiling behind the wet front was observed to be 1–2.5 mm in width, so that two values of the characteristic length of heating surface, $R = 1$ and 2 mm, were applied to the correlation. The value of Zuber [15] and Kutateladze [16] shows the critical heat flux in pool boiling predicted from their equation

$$\frac{q_{\text{crit}}}{\rho_v H_{fg}} = 0.16\left[\frac{\sigma g(\rho_l-\rho_v)}{\rho_v^2}\right]^{1/4}\left(\frac{\rho_l}{\rho_l+\rho_v}\right)^{1/2}. \quad (11)$$

The maximum heat flux in the rewetting process is about twice the value predicted by the above equation. The three dotted lines written in the high wall superheat region represent the relations for the film boiling, which were derived from Bromley's equation [17] presented for the averaged heat transfer coefficient in laminar film boiling from the vertical surface,

$$h_f = 0.943\left[\frac{k_v^3\rho_v(\rho_l-\rho_v)gH'_{fg}}{L\mu_v(T_w-T_s)}\right]^{1/4} \quad (12)$$

where

$$H'_{fg} = H_{fg}[1+0.68c_{pv}(T_w-T_s)/H_{fg}]$$

and L denotes height of the vertical surface. Also shown in this figure are the superheats at the rewet point ΔT_{wet} observed in the test runs. The condition of the rewet point is located in the transition boiling region, and

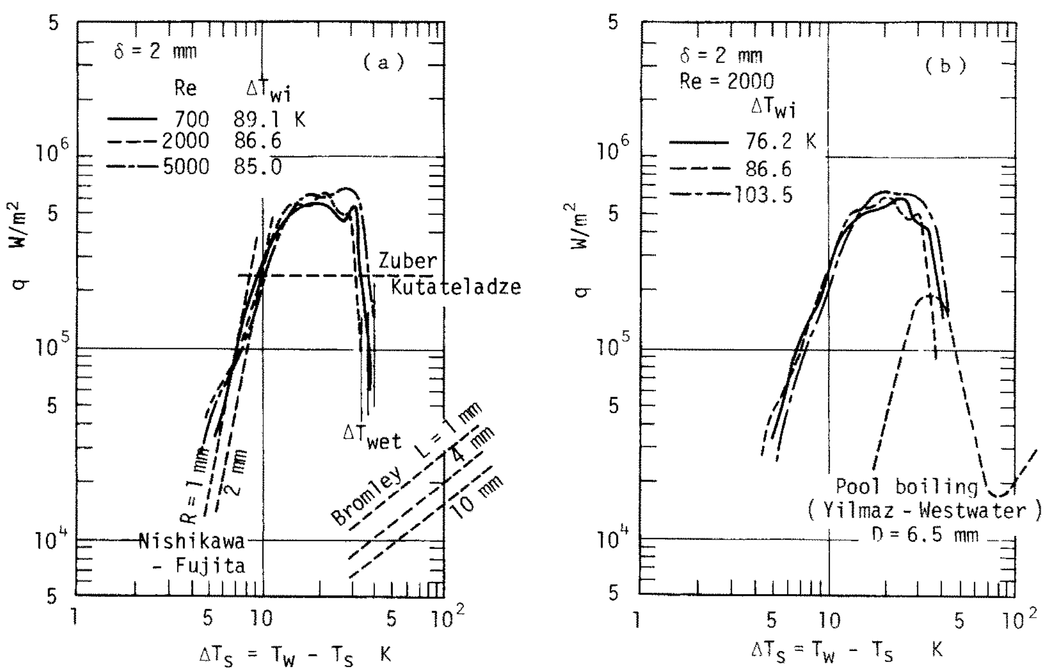

FIG. 9. Boiling curves, (a) effect of film flow rate, (b) effect of initial wall temperature.

does not seem to be corresponding to the minimum heat flux of the film boiling. The dotted line depicted in Fig. 9(b) for reference, is the boiling curve measured by Yilmaz and Westwater [18] for R-113 pool boiling from a horizontal tube of diameter $D = 6.5$ mm at atmospheric pressure.

In the analysis for predicting the wet front velocity, as mentioned in the Introduction, several models have been presented of the surface heat transfer distribution behind the wet front. However, the measurement of the surface heat transfer is limited and no model has considered the effect of wall thickness on the surface heat transfer distribution. Figure 10 shows the axial distributions of the surface heat transfer coefficient derived from the 2-dim. analysis, in which the wet front is located at $Z = 0$. The distribution curves represent the averaged values obtained from test runs of various ΔT_{wi} under a condition of $Re = 5000$. It is found from this figure that the heat transfer in the wet region ($Z < 0$) consists of the transition boiling direct behind the wet front and the following nucleate boiling, and the boiling region extends with increasing the tube wall thickness.

6. CONCLUSIONS

Rewetting of a hot surface by a falling film was studied experimentally at atmospheric pressure with R-113 liquid preheated near the saturation temperature, and the relation between the surface heat flux and the surface superheat behind the wet front was examined by solving numerically the 2-dim. conduction equation in the wall. The conclusions derived from this study are as follows:

(1) The wet front velocity decreases with increasing initial wall temperature and wall thickness, but is little affected by the liquid film flow rate. The effect of the initial wall temperature on the wet front velocity is well expressed by Yamanouchi's solution for the 1-dim. equation [equation (2)], but the effect of the wall thickness is not simple, as is shown in Fig. 6.

(2) The rewetting temperature is found to be about 40 K above the saturation temperature for the present stainless steel and R-113 liquid system.

(3) Near the wet front in the wet region, the heat transfer process consists of the transition boiling direct behind the wet front and the following nucleate boiling. The axial distribution of the surface heat transfer coefficient is dependent upon the wall thickness.

(4) However, the relation between the surface heat flux and the wall superheat, namely, the boiling curve is little affected by the wall thickness, and is independent of the liquid film flow rate and the initial wall superheat. The maximum heat flux is about twice the critical heat flux predicted for the pool boiling system.

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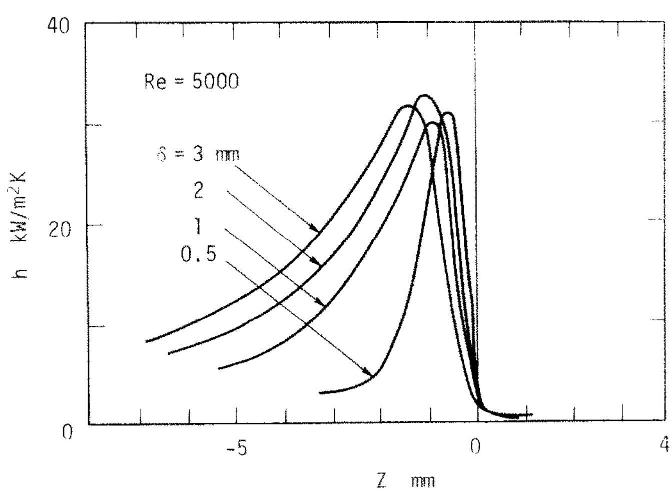

FIG. 10. Distributions of heat transfer coefficient behind the wet front.

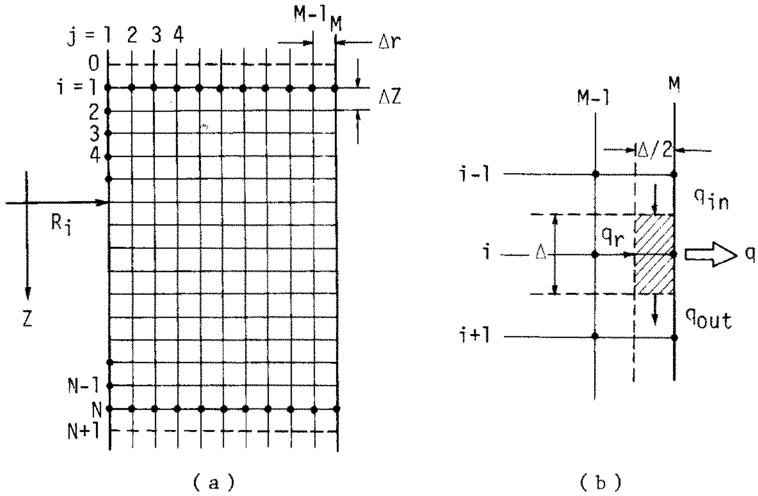

FIG. 11. Nodal network for 2-dim. analysis and calculation of surface heat flux.

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APPENDIX

Discretizing the tube wall along r- and Z-directions as shown in Fig. 11(a), equation (8) is reduced to the following difference equation:

$$\frac{1}{\Delta r}\left[\frac{T_{i,j+1}-T_{i,j}}{\Delta r}-\frac{T_{i,j}-T_{i,j-1}}{\Delta r}\right]+\frac{1}{R_i+(j-1)\Delta r}\frac{T_{i,j+1}-T_{i,j-1}}{2\Delta r}+\frac{1}{\Delta Z}\left[\frac{T_{i+1,j}-T_{i,j}}{\Delta Z}-\frac{T_{i,j}-T_{i-1,j}}{\Delta Z}\right]=-\frac{U}{a_w}\frac{T_{i+1,j}-T_{i-1,j}}{2\Delta Z}. \tag{A1}$$

To simplify the computation, we assume

$$\Delta r = \Delta Z = \Delta \tag{A2}$$

then, equation (A1) is rearranged as

$$T_{i,j} = \tfrac{1}{4}[(1-\alpha_j)T_{i,j-1}+(1-\alpha)T_{i-1,j}+(1+\alpha_j)T_{i,j+1}+(1+\alpha)T_{i+1,j}] \tag{A3}$$

where

$$\alpha_j = \frac{1/2}{R_i/\Delta+(j-1)}, \qquad \alpha = \frac{U}{a_w}\frac{\Delta}{2}.$$

The boundary conditions, equation (9), are written in finite difference form as follows:

On the inner surface

$$T_{i,2} = T_{i,0}$$

this gives

$$T_{i,1} = \tfrac{1}{4}[2T_{i,2}+(1-\alpha)T_{i-1,1}+(1+\alpha)T_{i+1,1}]. \tag{A4}$$

On the outer surface

$$T_{i,M} = T_w(i) \tag{A5}$$

where, $T_w(i)$ represents the temperature at the ith nodal point on the outer surface. In this paper, the temperature distribution in the tube wall was computed from equations (A3)–(A5) by substituting the measured surface temperature into $T_w(i)$. The interval size $\Delta = 0.01$ mm was used in this computation.

As for boundary conditions in Z-direction, the following relationships derived by imposing zero net axial conduction at the nodal point $i = 0$ and $N+1$

$$\begin{aligned} T_{0,j} &= 2T_{1,j}-T_{2,j} \\ T_{N+1,j} &= 2T_{N,j}-T_{N-1,j} \end{aligned} \tag{A6}$$

were used.

Once the temperature distribution in the tube wall has been solved, the surface heat flux q is obtainable. From the heat balance for the hatched area in Fig. 11(b),

$$q = \tfrac{1}{2}(q_{in}-q_{out})+q_r. \tag{A7}$$

Then, the surface heat flux was determined by calculating the heat fluxes q_{in}, q_{out} and q_r from the following equations:

$$q_{in} = -\frac{k_w}{\Delta}[\tfrac{1}{4}(3T_{i,M}+T_{i,M-1})-\tfrac{1}{4}(3T_{i-1,M}+T_{i-1,M-1})],$$

$$q_{out} = -\frac{k_w}{\Delta}[\tfrac{1}{4}(3T_{i+1,M}+T_{i+1,M-1})-\tfrac{1}{4}(3T_{i,M}+T_{i,M-1})],$$

$$q_r = -\frac{k_w}{\Delta}(T_{i,M}-T_{i,M-1}).$$

REMOUILLAGE D'UNE SURFACE CHAUDE PAR UN FILM LIQUIDE TOMBANT

Résumé—On étudie expérimentalement le remouillage d'un tube chaud en acier inoxydable par un film liquide de Freon 113, à la pression atmosphérique. On présente les effets du débit du film, de la température initiale de la paroi et de l'épaisseur de la paroi sur la vitesse du front de mouillage. En appliquant la variation de température mesurée sur la surface pariétale, une équation de conduction bidimensionnelle est résolue numériquement pour obtenir la courbe d'ébullition, la relation entre le flux thermique et la surchauffe de la paroi. Le résultat montre que le transfert intense au film liquide derrière le front humide est réalisé par l'ébullition nucléée et de transition et que la courbe d'ébullition est peu affectée par la vitesse du film, la température initiale de la paroi et l'épaisseur de celle-ci, et le flux thermique maximal est environ deux fois plus grand que le flux critique prédit pour l'ébullition en réservoir.

WIEDERBENETZUNG EINER HEISSEN OBERFLÄCHE DURCH EINEN FALLENDEN FLÜSSIGKEITSFILM

Zusammenfassung Die Wiederbenetzung eines heißen Rohres aus rostfreiem Stahl durch einen Film aus flüssigem R 113 bei Atmosphären-Druck wird experimentell untersucht. Der Einfluß des Film-Massenstroms, der anfänglichen Wandtemperatur und der Wandstärke auf die Ausbreitungsgeschwindigkeit der Benetzungsfront werden dargestellt. Unter Verwendung der gemessenen Temperaturänderung an der Wandoberfläche wird die zweidimensionale Wärmeleitungsgleichung in der Wand numerisch gelöst, um die Siedekurve zu erhalten ; diese stellt die Beziehung zwischen der Wärmestromdichte an der Oberfläche und der Wandüberhitzung dar. Das Ergebnis zeigt, daß die intensive Wärmeübertragung an den Flüssigkeitsfilm hinter der Benetzungsfront durch Übergangs- und Blasensieden hervorgerufen wird. Die Siedekurve wird von Film-Massenstrom, anfänglicher Wandtemperatur und Wandstärke nur wenig beeinflußt. Die maximale Wärmestromdichte ist ungefähr doppelt so groß wie die kritische Wärmestromdichte, die sich für Behältersieden errechnen läßt.

ПОВТОРНОЕ СМАЧИВАНИЕ НАГРЕТОЙ ПОВЕРХНОСТИ НАТЕКАЮЩЕЙ ПЛЕНКОЙ ЖИДКОСТИ

Аннотация—Проведено экспериментальное исследование повторного смачивания нагретой трубы из нержавеющей стали жидкой пленкой фреона-113 при атмосферном давлении. Рассмотрено влияние расхода жидкой пленки, начальной температуры и толщины стенки на скорость фронта смачивания. В результате измерения температуры на поверхности стенки и численного решения двухмерного уравнения теплопроводности в стенке, получена кривая кипения, т.е. соотношение между поверхностным тепловым потоком и перегревом стенки. Показано, что интенсивный теплоперенос к пленке жидкости за фронтом смачивания осуществляется переходным и пузырьковым кипением, кривая кипения слабо зависит от расхода жидкой пленки и начальной температуры и толщины стенки, а максимальный тепловой поток примерно в два раза превышает критический тепловой поток, рассчитанный для системы кипения в большом объеме.