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# MECHANISM AND MODELING OF REWETTING INITIATION OF HOT DRY SURFACE IN SATURATED AND SUBCOOLED FILM BOILING

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## ABSTRACT

The behavior of rewetting on a high superheated and dry surface, focusing on rewetting temperature just as collapse of saturated and subcooled film boiling was investigated experimentally and analytically. Saturated and subcooled pool film-boiling experiments were conducted by using a Silicon wafer with 20 mm length, 20 mm width and 0.5 mm thickness and pure water at atmospheric condition. Saturated and subcooled impingement jet experiments were also preformed. Then, the model analysis of rewetting initiation of a hot dry surface in saturated and subcooled film boiling was constructed by using hydrodynamics instability on liquid-vapor interface (Rayleigh-Taylor instability) and a heat conduction model (rewetting model under a falling film). Some physical parameters on Rayleigh-Taylor instability, for example released period of bubble from vapor film on a heater, generated volume rate of vapor and so on, were estimated by using Two-Phase Boundary Layer theory of saturated and subcooled film boiling. The present analytical results also showed that as the liquid subcooling was high, MHF temperature was higher. Namely, the predictions agreed with the present experimental results and Dhir-Purohit's correlation. In addition, the present model of MHF temperature was developed by taking into account the dependence on thermal conductivity of wall of the MHF-temperature.

## INTRODUCTION

The physical mechanisms of transition boiling heat transfer are still unresolved important issues, because of the complexity of the rewetting phenomena on a superheated surface. Especially, it is not clear what temperature does contact between solid surface and liquid. In addition, the rewetting condition is very sensitive to liquid subcooling in the case of subcooled boiling.

Several attempts have been made by some investigators to examine the mechanism of wetting on a high superheated surface or a vapor film collapse in the film boiling, i.e., Minimum-Heat-Flux condition. For example, in an early work, Berenson (1962) analyzed the MHF heat flux $q_{MHF}$ based on the heat-flux-controlled hypothesis and reported the following correlation.

$$q_{MHF} = 0.09\rho_v h_{fg}\left[\frac{g(\rho_l-\rho_v)}{\rho_l+\rho_v}\right]^{1/2}\left[\frac{\sigma}{g(\rho_l-\rho_v)}\right]^{1/4} \tag{1}$$

Spiegler et al. (1963) proposed that the condition suppressed to wet on the superheated surface was the thermodynamic limit of superheat. Recently, Nishio (1987) reported the following correlation for the MHF-point temperature $T_{MHF}$ of saturated water at atmospheric pressure based on the temperature-controlled hypothesis.

$$T_{MHF} = 200\ {}^oC \tag{2}$$

However, this temperature is sensitive to the characteristics of the boiling surface, and as a result, it is very rare that the above temperature is achieved in industrial situations. Especially, this tendency is marked in the case of sub-cooled boiling, in which case the MHF temperature is known to be described by the following empirical equation by Dhir and Purohit (1987).

$$T_{MHF} = 201 + 8\Delta T_{sub}\ {}^oC \tag{3}$$

Namely, the above equation predicts values that exceed the thermodynamic limit of superheat ($T_{tls}$) for subcoolings larger than $\Delta T_{sub}$=14K; for water at one atmosphere, $T_{tls}$=310°C roughly. Therefore, the possibility of seeking a physical interpretation to the above equation seems doubtful. The authors (2004) examined effects of the temperature of a localized low-temperature region on the followings: film-boiling vapor-film collapse condition represented by MHF point temperature for a heated surface of high superheat, the way in which vapor-film collapses, and the vapor-film collapse velocity. Conclusions of the study are as follows: when the temperature of low-temperature section decreases, the MHF point temperature (namely, the average temperature of heat transfer surface at the instance of vapor-film collapse) gets elevated; conversely, the propagation velocity of vapor-film collapse decreases. The MHF point gets rapidly elevated in temperature particularly when the low-temperature section has a lower temperature than that at the thermodynamic limit of superheat (more accurately, spontaneous homogeneous nucleation temperature); the propagation velocity also gets slower in this case. Finally, we proposed one solution to explain the fact that MHF point temperature can exceed the temperature at thermodynamic limit of superheat.

The final purpose of this study is to understand a rewetting condition on a superheated wall. Namely, the local and instantaneous temperature just at the rewetting is measured by using micro-electrical-mechanical system (MEMS) technology. As first step, the behavior of rewetting on high superheated surface, focusing on rewetting temperature during collapse of saturated and subcooled film boiling is investigated experimentally. Then, the rewetting temperature is examined analytically by using hydrodynamics instability on liquid-vapor interface and a heat conduction model. In addition, the present model of MHF temperature is developed by taking into account the dependence on thermal conductivity of wall of the MHF-temperature.

## NOMENCLATURE

c: specific heat
h: heat transfer coefficient
$h_{fg}$: latent heat
k: thermal conductivity
$q_w$: wall heat flux
$T_{MHF}$: minimum-heat-flux-point temperature
$T_{sat}$: saturation temperature
$T_{tls}$: thermodynamic limit of superheat
$\Delta T_{sat}$: wall superheat ( $= T_w - T_{sat}$)
$\Delta T_{sub}$: liquid subcooling ( $= T_{sat} - T_l$)
w: velocity
$\rho$: density

Subscripts
l: liquid
sat: saturation
sub: subcool
v: vapor
w: wall

## EXPERIMENTAL APPARATUS AND PROCEDURES

Saturated and subcooled film-boiling experiments were conducted by using a Silicon wafer with 20 mm length, 20 mm width and 0.5 mm thickness and pure water at atmospheric condition. (Ohtake et al., 2008)

Both pool boiling and impingement jet experiments were preformed in the present study. The pool boiling experiments were for fundamental research of boiling heat transfer; the impingement jet experiments were for industrial application. In addition, the effect of liquid velocity on the rewetting temperature was examined through both experiments.

Figure 1-1 shows the schematic diagram of the experimental apparatus on impingement jet experiments. Figure 1-2 shows that of the experimental apparatus on pool boiling experiments. Temperature of the test plate was measured by a K-type-thermocouple with 0.1 mm diameter set at back side of it. The hot junction was bonded to the test plate with special compound made from sliver particle and epoxy resin, as shown in Fig. 2. Namely, the averaged temperature of the test plate was measured in the present experiments. The test section was heated by using a copper block embedded cartridge

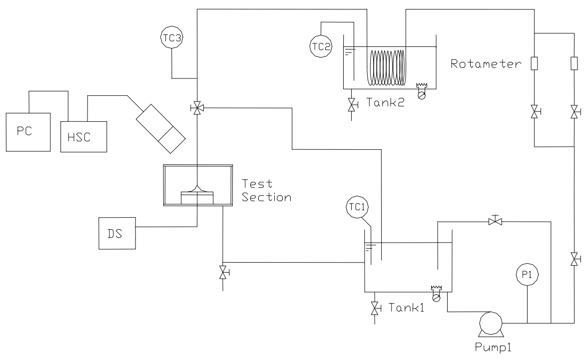



Figure 1-1 Schematic diagram of the experimental apparatus on impingement jet experiment

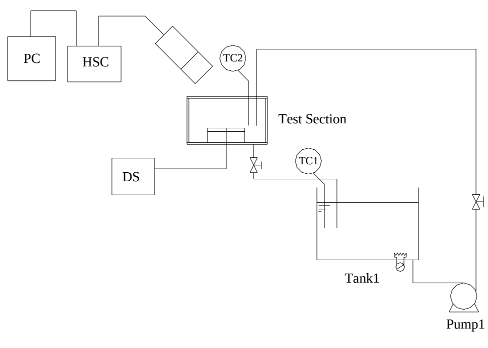



Figure 1-2 Schematic diagram of the experimental apparatus on pool boiling experiment

Figure 2 Details of test plate and thermocouple

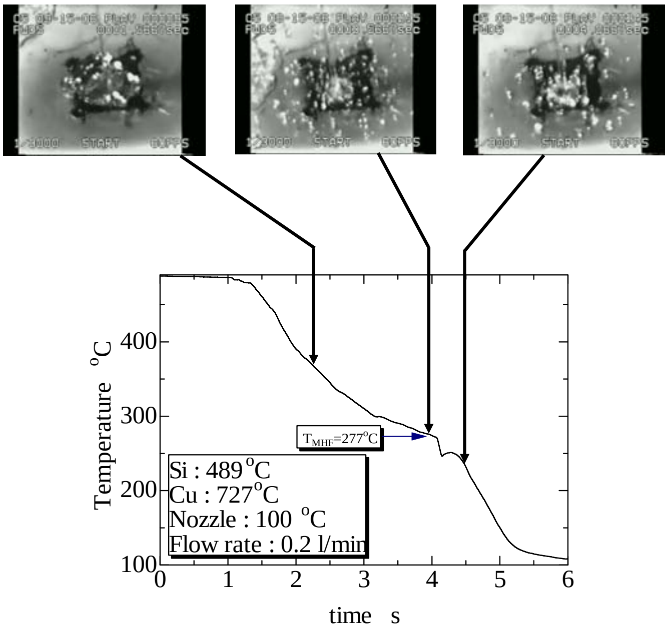



Fig. 3-1 Cooling curve for saturated condition

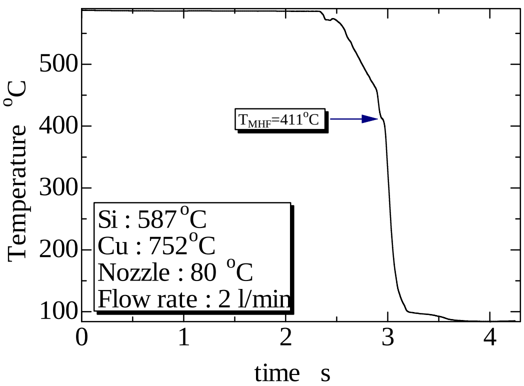



Fig. 3-2 Cooling curve for subcooled condition

type heaters. The block was connected to the test plate in heating period. Then, the block was moved out it during the experiment. After a prescribed initial wall temperature, 500 $^{o}$C, was established, an experimental vessel was filled with saturated or subcooled water, then quenching experiment was done. For the impingement jet experiments, saturated or subcooled water was injected on the superheated surface from a nozzle set above the test surface. Diameter of the nozzle was $\phi$6.3 mm. During the quenching experiments, the temperature history was measured by using the thermocouple and stored in a digital oscilloscope with 16 bits and 5 kHz. The data were post-processed with inverse heat conduction problem after the experiments. The behavior of the rewetting was also stored with a high-speed video camera at 500 frames/s. In the present study, the experiments were conducted at liquid subcooling of 0, 10 and 20 K, respectively.

## EXPERIMENTAL RESULTS AND DISCUSSION

Figure 3 shows measured results of temperature-time histories, cooling curves, for the impingement jet experiments of some experimental conditions. Figure 4 shows boiling curves for the impingement jet experiments corresponded to the cooling curve in Fig. 3. The relation between averaged wall heat flux $q_w$ and averaged wall superheat $\Delta T_{sat}$ was obtained by solving the lumped heat capacity method with the aid of measured temperature of the test section. Namely, the wall heat flux and the wall superheat were area averaged values over the test heater, 20 X 20 mm. The Biot number, Bi = $(q_w/\Delta T_{sat})\delta/k_w$, is about 0.005 at MHF-point. Thus, the uncertainty of wall heat flux at MHF-point is estimated to be about 1 %. The figure also shows the following relationships: equations of Rohsenow (1962) for nucleate boiling heat flux $q_{nb}$, Kutateladze (1950) for critical heat flux $q_{CHF}$, Berenson (1961) for coefficient of film boiling heat transfer $h_f$, and the relationship of Nishio (1987) for MHF point temperature. As shown in Figs. 4-1 and 4-2, the film-boiling heat transfer characteristic during the impingement jet experiment is similar to that of the conventional pool boiling. The critical heat flux for the saturated condition, however, is lower than calculated value by Kutateladze's equation. The result is considered that area of liquid-solid contact is limited partly about 50%, according to observation during the present experiment as shown in Fig. 3.

In Fig. 5, temperatures at minimum heat flux-point for several experimental conditions are plotted against liquid subcooling. As shown in Fig. 5, the temperature of minimum heat flux increase with the liquid subcooling. The trend is similar to that with Dhir-Purohit's equation. The present experimental results in the experimental conditions showed that temperature at incipient of rewetting were over 300 $^{o}$C (200 K) corresponded to the thermodynamic limit of superheat.

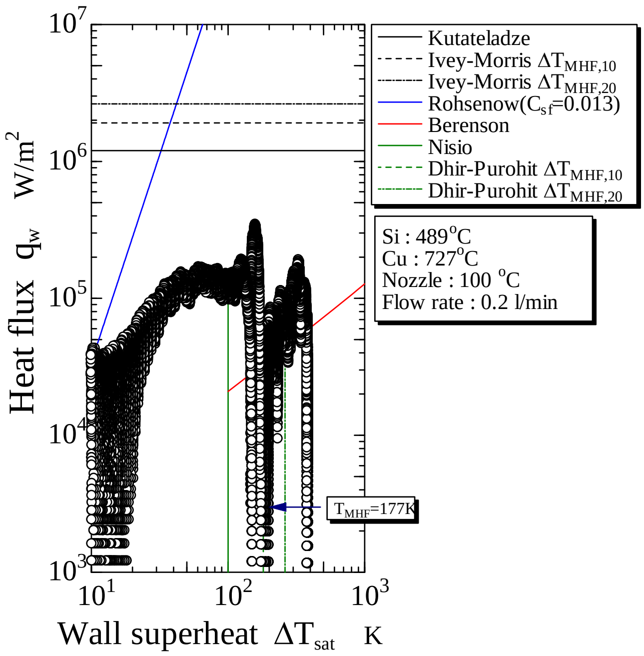



Fig. 4-1 Boiling curve for saturated condition

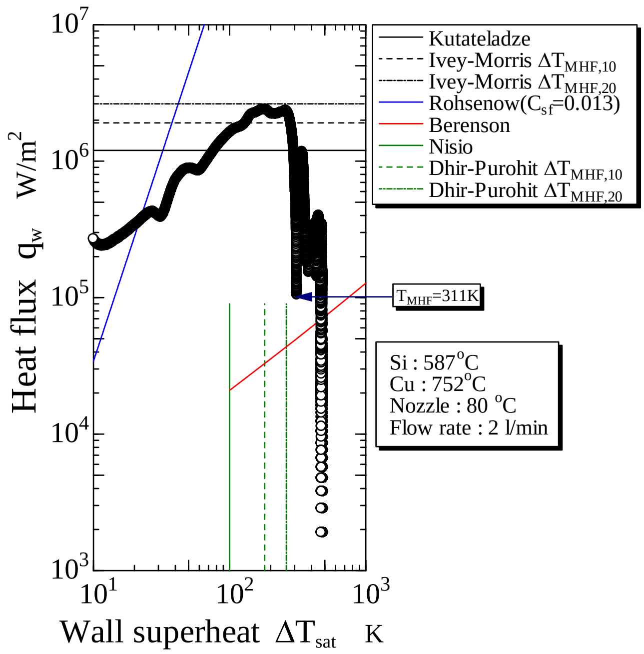



Fig. 4-2 Boiling curve for subcooled condition

## MODEL ANALYSIS

The rewetting temperature mentioned above is examined analytically by using hydrodynamics instability on liquid-vapor interface and a heat conduction model. The physical model is show in Fig. 6.

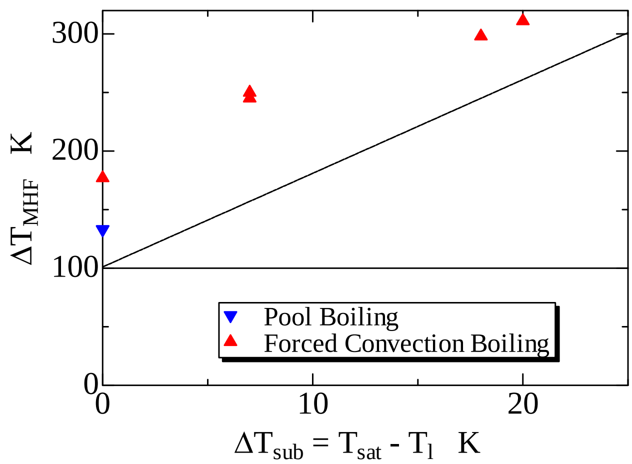



Figure 5 Minimum heat flux-temperature

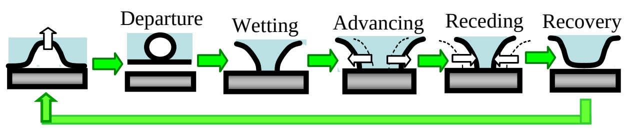



Fig. 6 Physical model for MHF

## ESTIMATION OF FILM BOILING CHARACTERISTICS

At first, we estimate film-boiling heat transfer coefficient by Berenson's equation (1961) for saturated boiling, $h_{sat}$:

$$h_{sat} = 0.425\left[\frac{k_v{}^3 g\rho_v(\rho_l-\rho_v)h_{lv}'}{\mu_v \Delta T_{sat}}\sqrt{\frac{g(\rho_l-\rho_v)}{\sigma}}\right]^{1/4} \quad (4)$$

Here, $h_{lv}$' is modified latent heat for sensible heat of vapor.

In the case of the subcooled boiling, we calculate the film-boiling heat transfer coefficients, $h_{sub}$, from the following expressions by using as a result of two-phase boundary layer theory by Nishio and Ohtake (1991):

$$h_{sub}/h_{sat} = 1.27 \quad \text{for } \Delta T_{sub}=10\text{K} \quad (5)$$

$$h_{sub}/h_{sat} = 1.67 \quad \text{for } \Delta T_{sub}=20\text{K} \quad (5)'$$

$$h_{sub}/h_{sat} = 2.11 \quad \text{for } \Delta T_{sub}=30\text{K} \quad (5)''$$

Next, volume of generated vapor at the condition of film boiling is given by

$$Q_v\,[m^3/s] = (q_{vapor} \times A_{w)}/(\rho_v h_{lv}) \quad . \quad (6)$$

Here, $q_{vapor}$ is heat flux for evaporation. For saturated boiling, it is given by

$$q_{vapor,sat} = h_{sat}\,\Delta T_{sat} \quad (7)$$

Here, $\Delta T_{sat}$ is wall superheat: $\Delta T_{sat} = T_w - T_{sat}$. On the other hand, in the case of subcooled boiling, although $h_{sub}$ is higher than $h_{sat}$, most of heat transfer from a heater is spent on sensible heat transfer to liquid phase, according to analytical results by Nishio et al. (1987). Namely, $q_{vapor,sub}$ is smaller than $q_{vapor,sat}$. Therefore, we consider to estimate a contribution of sensible heat transfer to liquid phase and heat fluxes for evaporation in the subcooled boiling are given by

$$q_{vapor} = 1.27 \times h_{Berenson} \Delta T_{sat} \times (1 - 0.720) \quad \text{for } T_{sub}=10\text{K} \quad (8)$$

$$q_{vapor} = 1.67 \times h_{Berenson} \Delta T_{sat} \times (1 - 0.933) \quad \text{for } T_{sub}=20\text{K} \quad (8)'$$

$$q_{vapor} = 2.11 \times h_{Berenson} \Delta T_{sat} \times (1 - 0.970) \quad \text{for } T_{sub}=30\text{K} \quad (8)''$$

Here, 0.720, 0.933 and 0.970 is from a result of two-phase boundary layer theory (Nishio and Ohtake) for 10 K, 20 K and 30 K liquid subcooling, respectively.

Finally, frequency of released bubbles is able to estimated by

$$f = Q_v / V . \quad (9)$$

Here, $V$ is a volume of released bubble, as shown in Fig. 7.

Therefore, it is assumed that the volume of released bubble is given as a hemispherical bubble with diameter of the most dangerous wavelength of Rayleigh-Taylor instability, $\lambda_d$, for saturated boiling:

$$V_{sat} = 0.5 \times \left( \pi \lambda_d^{\ 3} / 6 \right) \ . \quad (10)$$

For subcooled boiling, the volume of released bubble is assumed to that of a spherical cap bubble:

$$V_{sub} = \frac{\pi \lambda_d}{12} \left( \frac{3 \lambda_d^{\ 2}}{4} + h^2 \right) \ . \quad (10)'$$

The height of bubble is modified by using the two-phase boundary layer theory:

$$h = \frac{\delta_{v,sub}}{\delta_{v.sat}} \frac{\lambda_d}{2} \ . \quad (11)$$

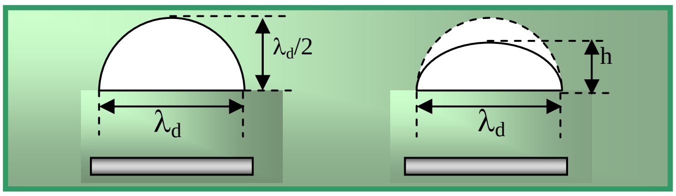
We assume the $f$ $(=Q_v/V)$ to be frequency for occurrence of wet on heating surface.

## ESTIMATION OF WETTING PARAMETERS

Period of wetting is determined by f mentioned above.

For liquid-wall contact time, $t_{wet}$ = 10 ms, (12)

based on measured results by Yao and Henry (1978). Although measured value of liquid-wall contact time for subcooled boiling did not report through past researches, prediction of MHF-Temperature obtained by the present model




(a) Saturated boiling (b) Subcooled boiling

Fig. 7 Released bubble in the present model

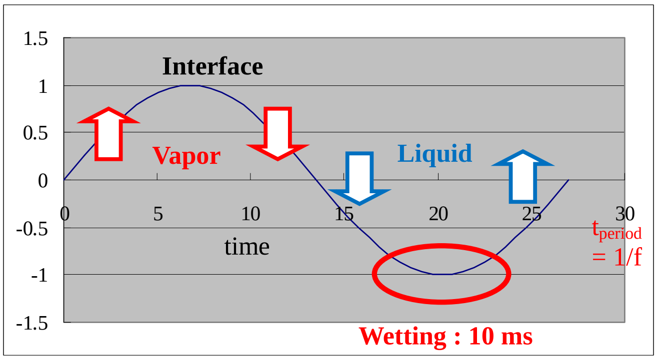



Fig. 8 Behavior of wave on liquid-vapor interface

was independent of the liquid-wall contact time fortunately.

Wetting area is estimated as a quantity based on characteristics of wave on liquid-vapor interface, i.e, Rayleigh-Taylor instability. The critical wavelength is calculated by

$$\lambda_{cr} = 2\pi \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} , \quad (13)$$

where $\lambda_{cr}$ is the critical wavelength of Rayleigh-Taylor instability.

If wave on liquid-vapor interface behaviors as Rayleigh-Taylor instability, it is assumed to size of the wetting area is estimated by the following equation, as shown in Fig. 8.

$D_{wet} = \lambda_{cr} \times [t_{wet}/t_{period}] = \lambda_{cr} \times [t_{wet} \times f]$

Here, $D_{wet}$ is diameter of wetting area.

Finally, averaged advancing and receding velocities during the wetting are predicted by

$w_{advancing} = w_{receding} = (D_{wet}/2) / (t_{wet}/2)$

based on hydrodynamics of wave on liquid-vapor interface.

## ESTIMATION OF REWETTING

For thermal wetting characteristics, we assume to be Yamanouchi's model which represents rewetting under a falling-flow of a liquid layer:

$$w_{rewet} = \left( 1/\rho_w c_w \right) \sqrt{h_{wet} k_w / \delta_w} \left( T_0 - T_l \right) \Big/ \sqrt{\left( T_w - T_0 \right)\left( T_w - T_l \right)} \quad (14)$$

In this attempt, we needed the following parameters: temperature at wet front $T_0$ and typical heat transfer coefficient on the wetted-side $h_{wet}$.

Now, we made the following assumptions.

(1) Dried-out surface temperature $T_w$ was equal to initial temperature.

(2) Temperature at the wet front $T_0$ was considered to be referred to a two-body contact problem based on heat conduction.

$$\frac{T_w - T_0}{T_0 - T_l} = \sqrt{\frac{(\rho c_p k)_l}{(\rho c_p k)_w}} \quad (15)$$

Here, $T_w$ is initial temperature. $T_l$ is temperature in bulk liquid.

(3) Heat transfer coefficient on the wetted-side $h_{wet}$ was given by the critical heat flux conditions as specified by equations for Kutateladze's equation for CHF in saturated boiling,

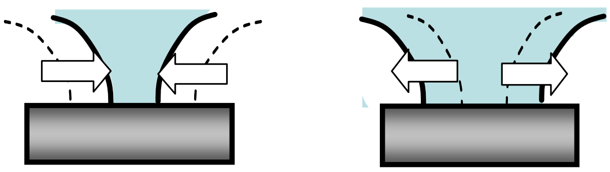
$$q_{CHF} = 0.13\rho_v h_{fg}\left[\frac{g\sigma(\rho_l-\rho_v)}{\rho_v^{\,2}}\right]^{1/4} \qquad (16)$$

For subcooled boiling, the CHF is estimated by using equation of Ivey-Morris (1962):

$$q_{CHF,sub} = q_{CHF,sat}\left\{1+0.102\left(\frac{\rho_l}{\rho_v}\right)^{\frac{3}{4}}\frac{C_{pl}\Delta T_{sub}}{h_{lv}}\right\} \qquad (16)'$$

(4) Heat transfer coefficient on the dried-out side was neglected.

(5) Wetted-side temperature Ts was set to liquid temperature $T_l$.



(i) film boiling is maintained (ii) film boiling is collapsed

Fig. 9 Criterion of MHF

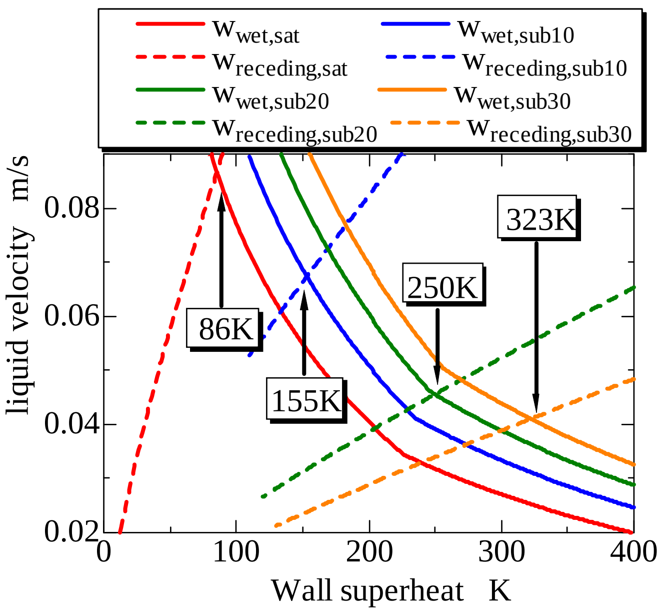



Fig. 10 $w_{receding}$ vs $w_{wet}$

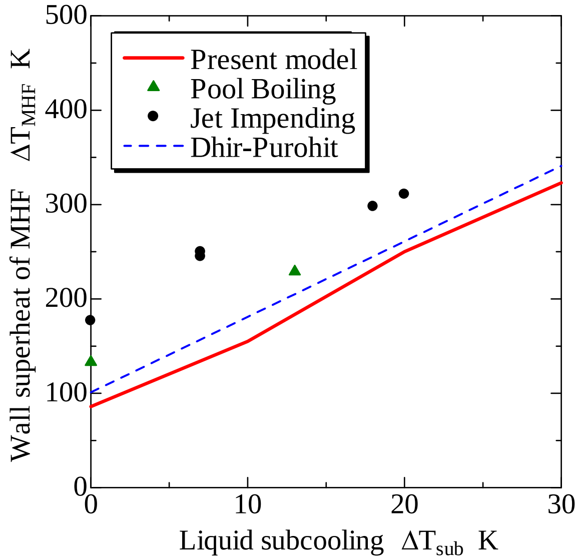



Fig. 11 Prediction by present model

## PREDICTION OF MHF-TEMPERATURE

In the present model, minimum heat flux-temperature is developed through comparison between the prediction on thermal wetting and the estimation based on hydrodynamics of wave on liquid-vapor interface, as shown in Fig. 9. Namely,

$w_{rewet}$ is compared with $w_{receding}$:

for $w_{receding} > w_{wet}$, film boiling is maintained,
for $w_{receding} < w_{wet}$, film boiling is collapsed.

When $w_{receding} = w_{wet}$, $T_w = T_{MHF}$. (17)

Figure 10 shows calculated results of $w_{receding}$ based on liquid-vapor interfacial hydrodynamic instability and $w_{wet}$ by using the rewetting model. From Fig. 10, wall superheat at point of intersection of $w_{receding}$ and $w_{wet}$ is high with increasing of liquid subcooling.

A prediction by the present model is illustrated in Fig. 11. The experimental results for minimum heat flux-temperature obtained in the present experiments are also shown in this figure. The present experimental results showed that as the liquid subcooling was high, Minimum Heat Flux temperature (i.e., Minimum Film Boiling, rewetting temperature or quenching temperature) is higher. Furthermore, the effect of the rewetting temperature on liquid subcooling is similar to that of Dhir-Purohit's correlation quantitatively. As seen from Fig. 11, the present analytical results also showed that as the liquid subcooling is high, MHF temperature is higher. Namely, the predictions agree with the present experimental results and Dhir-Purohit's correlation.

## EFFECT OF WALL THERMAL CONDUCTIVITY ON MHF-TEMPERATURE

According to published experimental evidence on minimum heat flux-temperature, as thermal conductivity of wall was lower, MHF-temperature was higher. In this section, the effect of wall thermal conductivity on MHF-temperature is examined through developing the present model.

In present report, the wetting area, $D_{wet}$, is focused for a discussion of the effect of wall thermal conductivity on MHF-temperature. Namely, if thermal conductivity of wall is high, vaporization mass is larger. Thus, it is considered that wetting is suppressed. In the above model, although averaged advancing velocity during the wetting is predicted by using a heat conduction model included thermal conductivity, receding velocity during the wetting is estimated by using hydrodynamics instability on liquid-vapor interface without effect of thermal conductivity. Therefore, it is assumed that the receding velocity during the wetting is influenced by wall thermal properties (see Table 1): as thermal conductivity of wall is higher, the receding velocity during the wetting, $w_{receding} = (D_{wet}/2) / (t_{wet}/2)$, is faster. The wetting area, $D_{wet}$, is large.

Any modified factors of the wetting area, C', are taken into account in the developing present models:

$$D_{wet}' = C' D_{wet} \qquad (18)$$

In regard of the modified factors, three models are examined:

I) modification of heat flux based on the contact problem of semi-infinity bodies on heat conduction:

$q=k(T_w-T_s)/(\pi a t)^{0.5}$ ,

II) modification of thermal conductivity,

III) modification of thermal diffusivity.

Simplest linear correlation is utilized for all modifications. For the modified process, the MHF-temperature of Silicon condition for the present experimental result is adopted as a basic MHF-temperature. For example, $C'_{II,\ Cu} = k_{Cu}/k_{Si}$ for copper of case II, modification of thermal conductivity.

The present modified results are summarized in Fig.11. For the effect of wall thermal conductivity on MHF-temperature, the best estimation is modification of thermal diffusivity, case III, as shown in Fig. 11.

Table 1 Physical Properties

| Material | | Copper | Brass (70Cu, 30Zn) | Carbon steel (0.45C) | Si |
|---|---|---|---|---|---|
| Thermal properties at 250℃ | $\rho$ kg/m$^3$ | 8805.6 | 8530 | 7783.1 | 2324 |
| | $c_p$ kJ/kgK | 0.411 | 0.4298 | 0.539 | 0.8166 |
| | k W/mK | 381.8 | 112.2 | 37.76 | 83.86 |

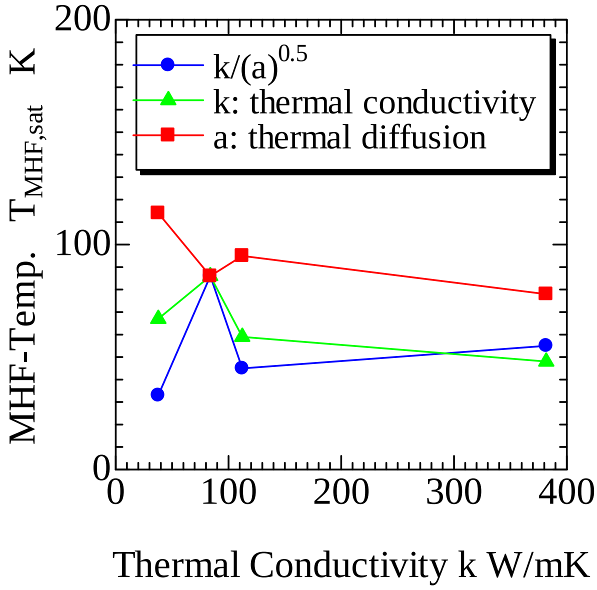



Fig. 12 Effect of wall thermal conductivity on MHF-temperature

## CONCLUSIONS

As first step, the behavior of rewetting on high superheated surface, focusing on rewetting temperature during collapse of saturated and subcooled film boiling was investigated experimentally. Then, the rewetting temperature was examined analytically by using hydrodynamics instability on liquid-vapor interface and a heat conduction model with Two-Phase Boundary Layer theory for saturated and subcooled film boiling. . The present analytical results showed that as the liquid subcooling was high, MHF temperature was higher. Namely, the predictions agreed with the present experimental results and Dhir-Purohit's correlation. In addition, the present model of MHF temperature was developed by taking into account the dependence on thermal conductivity of wall of the MHF-temperature. The best estimation was modification of thermal diffusivity.

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