Abstract
Knowledge of transient critical heat flux (CHF) for subcooled boiling of water flow is important for the design of cooling devices such as fusion reactor divertor. Heating occurs rapidly in the divertor and the heat flux is estimated to reach 10 MW/m² because of plasma disruption. To cope with such a heat flux under the transient condition, the use of a small inner diameter tube (< 3 mm diameter) is recommended, since the use of a smaller inner diameter tube results in a higher CHF. The transient CHF of subcooled boiling of water was measured for a 1.0 mm inner diameter tube, which was heated exponentially, with inlet subcoolings ranged from 82 to 145 K and a flow velocity of 10 m/s. The heat generation rate of the tube was exponentially increased according to the function $Q_0 \exp(t/\tau)$. E-folding time, $\tau$, ranged from 182 ms to 30.5 s. It was found that the transient CHF increased as the e-folding time decreased and that CHF of the small-diameter tube was higher than that of conventional channels. This indicated that CHF is affected by the e-folding time under transient conditions.

Key words: Flow boiling of water, CHF, High subcooled boiling of water, Vertical upward flow, Small diameter tube, E-folding time

1. Introduction
Knowledge of the transient critical heat flux (CHF) for subcooled boiling of water flow is necessary for the design of cooling devices such as fusion reactor divertor (Hata and Noda, 2006). Heat flux in divertor is high because of plasma disruption in the fusion reactor, and is estimated to reach 10 MW/m². Since the wall temperature of the divertor increases rapidly, it is necessary to reduce the transient heat load caused by plasma disruption, using subcooled boiling of water. Bergles (1963), Nariai et al. (1987), Celata et al. (1993), Vandervolt et al. (1994), Mudawar and Bowers (1999), and Hata et al. (2004) experimentally investigated the CHFs for subcooled boiling of water under steady-state heat loads. They clarified the effects of mass flow rate, inner tube diameter, outlet pressure, inlet and outlet subcooling, and inlet qualities on CHF. Katoaka et al. (1983) measured the transient CHF for subcooled boiling of water using a platinum wire. Celata et al. (1992) experimentally investigated the power transient CHF for subcooled boiling in vertical tubes containing R-12 (dichlorodifluoromethane). Hata and Noda (2008) and Hata and Masuzaki (2010) conducted both steady-state and transient experiments on subcooled boiling of water flowing upwards in vertical tubes with various inner diameters while exponentially increasing the heat inputs. They clarified the effects of length-to-diameter ratio ($L/d$), inlet and outlet subcooling, and the e-folding time. However, the inner diameters of the tubes were 3.0 to 9.0 mm, so the findings did not cover small-diameter tubes.

This research aimed to experimentally investigate the CHF for subcooled boiling of water flowing through a small-diameter tube under the steady-state and exponentially increasing heat load conditions. A small-diameter tube...
was used in the experiment to enhance CHF under transient conditions. Since there are no experimental data on the transient CHF of subcooled boiling of water in the heated tube with an inner diameter smaller than 3.0 mm, the aim of this study is to obtain basic data on the transient CHF in a small inner diameter tube with variations in e-folding time, flow velocity and subcooling.

Nomenclature

\[ B_0 = \frac{q_{cr, sub}}{G h_{fg}}, \text{ boiling number} \]

\[ C_1, C_2, C_3 \] constants in Eq. (18)

\[ c \] specific heat, J/kg K

\[ c_p \] specific heat at constant pressure, J/kg K

\[ d \] inner diameter of test tube, m

\[ G = \rho l u, \text{ mass flux, kg/m}^2\text{s} \]

\[ g \] acceleration of gravity, m/s²

\[ h_{fg} \] latent heat of vaporization, J/kg

\[ L \] heated length, m

\[ L_e \] entrance length, m

\[ P \] pressure, kPa

\[ P_{in} \] pressure at inlet of heated section, kPa

\[ P_{out} \] pressure at outlet of heated section, kPa

\[ P_i \] pressure measured by inlet pressure transducer, kPa

\[ P_o \] pressure measured by inlet pressure transducer, kPa

\[ Q \] heat input per unit volume, W/m³

\[ Q_0 \] initial heat input, W/m³

\[ q \] heat flux, W/m²

\[ R_0 \] normal resistance, Ω

\[ R_T \] resistance of the test tube, Ω

\[ r \] radius, m

\[ Re = \frac{G d}{\mu}, \text{ Reynolds number} \]

\[ S \] surface area, m²

\[ Sc' = \frac{c_p \Delta T_{sub,in}}{h_{fg}}, \text{ non-dimensional inlet subcooling} \]

\[ T \] temperature, K

\[ T_a \] average temperature of the test tube, K

\[ T_{in} \] inlet liquid temperature, K

\[ T_{out} \] outlet liquid temperature, K

\[ T_{sat} \] saturation temperature, K

\[ t \] time, s

\[ T_I = \frac{T_{in} + T_{out}}{2}, \text{ average bulk liquid temperature, K} \]

\[ \Delta T_{sub,in} = (T_{sat} - T_{in}), \text{ inlet liquid subcooling, K} \]

\[ \Delta T_{sub, out} = (T_{sat} - T_{out}), \text{ outlet liquid subcooling, K} \]

\[ u \] flow velocity, m/s

\[ V \] volume, m³

\[ W_e = G^2 d / \rho \sigma, \text{ Weber number} \]

\[ \alpha \] coefficient in Eq.(2)

\[ \beta \] coefficient in Eq.(2)

\[ \lambda \] thermal conductivity, W/mK

\[ \tau = \int_0^t Q(t)dt / Q(t), \text{ e-folding time, s} \]

\[ \nu \] kinematic viscosity, m²/s

\[ \mu \] viscosity, Ns/m²

\[ \rho \] density, kg/m³

\[ \sigma \] surface tension, N/m
2. Experimental Setup
2.1 Experimental apparatus

Figure 1 shows a schematic diagram of the experimental loop of flow boiling. The loop was composed of a pump, a pre-heater, a flow meter, a test section, a pressurizer, an expansion tank and a separator, a cooler, an ion exchanger, and a sound level meter with a microphone. The water loop was made of SUS304 stainless steel tube and was capable of operating at up to 2 MPa. The working fluid was distilled water which was deionized to approximately 5-MΩcm specific resistivity. The water was cooled by the cooler before it flowed into the pump and then heated to the desired temperature by the pre-heater. Flow velocity was measured by the mass flow meter (Keyence, FD-V70) and was controlled by regulating the frequency of the triple-phase alternating power source (Mitsubishi Electric, Inverter, Model-F720-30K), which was connected to a canned-type circulation pump (Nikkiso, VNH12-C4). The circulating water was pressurized up to 800 kPa using saturated vapor in the pressurizer, and the pressure was controlled within ±1 kPa using a heat controller. To detect the boiling incipience in the test tube, a sound level meter and a microphone (Rion, NL-42) was used.
Figure 2 shows a vertical cross-sectional view of the test section of the experimental apparatus. The circular test tube made of stainless steel (SUS304 BA, Ra= 0.4 μm) has an inner diameter of 1.0 mm, a thickness of 0.5 mm, and a length of approximately 100 mm. The tube was mounted vertically in the experimental water loop. Both ends of the tube were soldered to a 5.0-mm thick silver-coated copper plate. A wire, serving as an electrode, was soldered onto the tube surface 47.4 mm apart from its upper end. The length-to-diameter ratio (L/d) is thus 47.4. To investigate the effect of L/d on CHF, we prepared another test tube with a heated length of 25.0 mm (L/d = 25.0). Both ends of the tube were isolated with 14.0-mm thick Bakelite plates. The hydrodynamic entrance length can be estimated by the following equation (Brodkey, 1988) for turbulent flow in a tube:

\[ L_e / d = 0.693 (Re_d)^{1/4} \]  

(1)

where \( L_e \), \( d \), and \( Re_d \) are the length of the tube entrance, the inner diameter of the tube, and Reynolds number, respectively. Since \( L_e / d \) is in the range from 9.2 to 12.0, the tube’s entrance length has a negligible influence in this experiment.

![Fig. 2 Cross-sectional detail of the test section of the experimental apparatus](image)

2.2 Experimental method

A direct current from an electric power source was used to heat the test tube, and the power input was exponentially increased. The heat generation rate of the tube was controlled and measured using a heating control system (Shibahara et al., 2015). The average temperature of the tube was measured by resistance thermometry, using a double-bridge circuit and including the test tube as a branch. The relationship between the electrical resistance of the test tube and its temperature was determined while water was circulated through the apparatus. The tube was cleaned with liquid acetone before the experiment. The average temperature of the tube, \( T_a \), was calculated from the resistance-temperature relationship as follows:

\[
R_f = R_0 (1 + a T_a + \beta T_a^2)
\]

(2)

where \( R_0 \) is the nominal resistance. The instantaneous mean temperature of the tube was computed using an analog computer. To prevent the burnout of the tube, the power supply to the tube was cut off when the mean temperature reached a preset value. The unbalanced voltage of the double-bridge circuit including the tube, and the voltage differences between both ends of the tube and across the standard resistor, were uploaded to a personal computer through an analog-to-digital (A/D) converter. These data were simultaneously sampled at intervals ranging from 60 μs to 200 ms. The time resolution of the A/D converter was 1 μs.

The heat flux on the surface of the test tube, \( q \), was calculated using the following equation:

\[
q(t) = \frac{V}{S} \left( \dot{Q}(t) - \rho C_v \frac{dT_a}{dt} \right)
\]

(3)
where $V, S, \rho_s$, and $c_h$ are the volume, inner surface area, density, and specific heat of the tube, respectively. $Q$ is the internal heat generation rate and $T_a$ is the average temperature of the tube. Since CHFs are affected by the thermal diffusivity of the test tube (Golobic, 1997), the heat generation rate is controlled by the heating control system in the experiment. Radiation loss from the outer tube surface was within 0.07%. Using the measured average temperature of the test tube, the temperature distribution was calculated using the unsteady heat conduction equation by using the experimental heat flux calculated by Eq. (3) and assuming that the surface temperature was uniform around the circumference.

$$\rho \frac{\partial T}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r} \left( r \lambda \frac{\partial T}{\partial r} \right) + \frac{1}{r} \frac{\partial}{\partial \theta} \left( \lambda \frac{\partial T}{\partial \theta} \right) + \lambda \frac{\partial^2 T}{\partial z^2} + Q(t)$$  \hspace{1cm} (4)

Boundary conditions and the average temperature are as follows:

$$-\lambda \frac{\partial T}{\partial r}|_{r=0} = q(t)$$  \hspace{1cm} (5)

$$\frac{\partial T}{\partial r}|_{r=r_0} = 0$$  \hspace{1cm} (6)

$$\frac{1}{\pi(r_o^2-r_i^2)} \int_{r_i}^{r_o} 2\pi r T(r) dr = T_a$$  \hspace{1cm} (7)

For the steady-state condition, the average temperature of the tube was calculated using the following equation. The inner and outer surface temperatures of the tube were then calculated.

$$\frac{d^2T}{dr^2} + \frac{1}{r} \frac{dT}{dr} + \frac{Q}{\lambda} = 0$$  \hspace{1cm} (8)

$$T(r) = \frac{Qr^2}{4\lambda} + \frac{Qr_i^2}{2\lambda} \ln r + C$$  \hspace{1cm} (9)

$$T_a = \frac{1}{\pi(r_o^2-r_i^2)} \int_{r_i}^{r_o} 2\pi r T(r) dr$$  \hspace{1cm} (10)

$$T_0 = T(r_i) = T_a - \frac{Qr_i}{4(r_o^2-r_i^2)\lambda} \times \left[ 4r_i^2 \left( \ln r_i - \frac{1}{2} \right) - r_i^3 \left( \ln r_i - \frac{1}{2} \right) \right] - \frac{Qr_i^2}{2(2r_o^2-r_i^2)\lambda} (r_i^2 - 2r_i^2 \ln r_i)$$  \hspace{1cm} (11)

$$T_o = T(r_o) = T_a - \frac{Qr_o}{4(r_o^2-r_i^2)\lambda} \times \left[ 4r_o^2 \left( \ln r_o - \frac{1}{2} \right) - r_o^3 \left( \ln r_o - \frac{1}{2} \right) \right] - \frac{Qr_o^2}{2(r_o^2-r_i^2)\lambda} (1 - 2\ln r_o)$$  \hspace{1cm} (12)

$$C = T_a - \frac{Qr}{4(r_o^2-r_i^2)\lambda} \times \left[ 4r^2 \left( \ln r - \frac{1}{2} \right) - r^3 \left( \ln r - \frac{1}{2} \right) \right] - \frac{Qr^2}{2(r_o^2-r_i^2)\lambda} (1 - 2\ln r)$$  \hspace{1cm} (13)

The inlet and outlet temperatures, $T_{in}$ and $T_{out}$, were measured with K-type thermocouples at the centerline of the tube. The uncertainty of the temperature measurement was ± 0.65 K. The inlet and outlet pressures, $P_{in}$ and $P_{out}$ were measured by the strain gauge pressure transducers set up at the inlet and outlet of the stream conduits, as shown in Fig. 2. The uncertainty analysis was conducted on the experimental results. Measurement uncertainty was estimated according to the ANSI/ASME performance test codes (1987). The maximum uncertainties of heat generation rate, heat flux, and surface temperature were estimated to be 2.0% rdg, 2.4% rdg, and 2.8% rdg, respectively. An uncertainty analysis was conducted on the experimental results. Measurement uncertainty was estimated according to the ANSI/ASME performance test codes (1987). The maximum uncertainties of heat generation rate, heat flux, and surface temperature were estimated to be 2.0% rdg, 2.4% rdg. Results derived from Eqs. (14) and (15) were within ±10% of those predicted by the pressure drop model.

An uncertainty analysis was conducted on the experimental results. Measurement uncertainty was estimated according to the ANSI/ASME performance test codes (1987). The maximum uncertainties of heat generation rate, heat flux, and surface temperature were estimated to be 2.0% rdg, 2.4% rdg, and 2.8% rdg, respectively. Experimental procedures were carried out in the following way: The water was deionized by the ion exchanger for 30 min at least. Then the water was circulated by the pump and deaerated by boiling in the pressurizer. When the flow
velocity and system pressure in the water loop were in a steady-state an electric current was transmitted to the test tube, and the power input was raised exponentially. Succeedingly, the surface temperature of the test tube and the power input over time were measured.

3. Experimental Results

The transient CHFs of the subcooled boiling of water were measured for the e-folding times ranging from 182 ms to 30.6 s. The inlet liquid subcoolings ranged from 82 to 145 K and flow velocities were 10 to 12 m/s. The heat generation rate is given by the exponential function, \( Q = Q_0 \exp(t/\tau) \), where \( Q \) is heat generation rate, \( Q_0 \) is initial heat generation rate, \( t \) is time, and \( \tau \) is the e-folding time, respectively. A smaller \( \tau \) implies a faster rate of heat generation. The CHF defined as the surface temperature excursion of the test tube from fully developed nucleate boiling to film boiling. Hence, the jump of the \( \Delta T_{sat} \) was detected as the CHF.

Figure 3 presents typical experimental data on the changes over time of the heat generation rate, \( \dot{Q} \), the heat flux, \( q \), and the surface temperature of the tube, \( T_s \), at an inlet subcooling of 145 K, an e-folding time of 30.6 s, a flow velocity of 10 m/s, and an initial heat input of 0.136 \times 10^9 W/m^3. As shown in Fig. 3, the surface temperature of the tube and heat flux increases exponentially with the heat generation rate. The inflection point in the surface temperature of the tube indicates that CHF occurred at approximately 140 s after the initiation of power input. The CHF is approximately 21 MW/m^2.

Figure 4 shows the relationship between heat flux and surface superheat, \( \Delta T_{sat} \) (= \( T_s - T_{sat} \)) at a flow velocity of 10 m/s, \( \tau = 30.6 \) s, and an inlet subcooling of 145 K. Heat flux increases with increasing \( \Delta T_{sat} \) and begins to ascend steeply from the onset of nucleate boiling (ONB) at a \( \Delta T_{sat} \) of 15 K. The ONB was detected by the sound level meter and the microphone. The dashed line represents the ONB model Eq. (16) proposed by Bergles and Rohsenow (1964) and the chain line the ONB model Eq. (17) proposed by Sato and Matsumura (1963):

Bergles and Rohsenow

\[
q_{ONB} = 5.3P^{0.16}[1.8(T_s - T_{sat})_{ONB}^{-0.0234}] 
\]

Sato and Matsumura

\[
q_{ONB} = \frac{\lambda h_{fg}(T_s - T_{sat})^2}{8\sigma T_{sat}(\nu_g - \nu_l)} 
\]

where \( P \), \( \lambda \), \( h_{fg} \), \( \sigma \), \( \nu_g \), and \( \nu_l \) are system pressure, thermal conductivity of water, latent heat of vaporization, surface tension of water, kinematic viscosity of vapor, and viscosity of liquid, respectively. Calculations by Eqs. (16) and (17) were carried out using these thermal properties of water, and the saturated temperature at the system pressure. The
ONB point of this study is plotted in Fig. 4 and compared with Eqs. (16) and (17). The present experimental result agrees with the Bergles and Rohsenow model (Eq. (16)).

Figure 5 shows the effect of inlet subcooling, $\Delta T_{\text{sub, in}}$, on CHF under steady-state conditions. The flow velocities are 10 m/s and 12 m/s. The CHF increases monotonically with $\Delta T_{\text{sub, in}}$ at each flow velocity. The dashed lines show the calculated values by the following correlation of Hata et al. (2004), which can be applied to for an inner diameter in the range from 3.0 to 9.0 mm and a $\Delta T_{\text{sub, in}}$ of greater than 40 K.

$$B_{0e} = C_1 \left( \frac{d}{\sigma / (g \rho_l - \rho_v)} \right)^{0.5} W_{e}^{0.5} \left( \frac{L}{d} \right)^{-0.1} e^{-13.4(10^3 d^3 x^3) S c^*}$$

(18)

where $C_1 = 0.082$, $C_2 = 0.53$, $C_3 = 0.7$, and $B_{0e}$, $W_{e}$, $\sigma$, $g$, $\rho_l$, $\rho_v$, $Re$ and $Sc^*$ are boiling number ($= \frac{q_{cr, sub}}{G h_{fg}}$), Weber number ($= \frac{G^2 d}{\rho_l \sigma}$), surface tension, acceleration of gravity, density of liquid, density of vapor, Reynolds number, and non-dimensional inlet subcooling ($= c_{pl} \Delta T_{\text{sub, in}} / h_{fg}$), respectively. The thermal properties of the saturated water are determined from the outlet pressure. As shown in Fig. 5, the CHFs are higher than the values predicted by Eq. (18) for both flow velocities. This may be attributable to the following: (i) small vapor bubbles formed under the high pressure and high flow velocities. 

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**Fig. 4 Relationship between heat flux, $q$, and surface superheat, $\Delta T_{\text{sat}}$.**

- SUS304
  - $d = 1.0 \text{ mm}$
  - $L = 47.4 \text{ mm}$
  - $P_{in} = 963 \text{ kPa}$
  - $\Delta T_{\text{sub, in}} = 145 \text{ K}$

**Fig. 5 Effect of inlet subcooling ($\Delta T_{\text{sub, in}}$) on CHF under steady-state conditions.**

- SUS 304
  - $d = 1.0 \text{ mm}$
  - $L = 47.4 \text{ mm}$
  - $P_{in} = 936 - 1028 \text{ kPa}$
  - Flow velocity
    - 12 m/s
    - 10 m/s
  - Eq. (18)
subcooling and the high pressure conditions; (ii) increase of a superficial velocity of the vapor bubble relative to the liquid velocity due to the onset of net vapor generation (NVG) or the onset of significant void (OSV); and (iii) rapid condensation of vapor bubbles caused by the highly subcooled bulk fluid with the outlet subcooling from 1.7 to 51.5 K. (Chen, 2012). Since it appears that small vapor bubbles occupy a large proportion of the cross-section due to the NVG or the OSV, it is considered that the apparent flow velocity increased in the test tube. Moreover, the saturated temperature of small vapor bubbles is relatively higher than that of large bubbles since the internal pressure of small vapor bubble is higher than that of large vapor bubble. Therefore, it is considered that the small vapor bubbles cannot exist, and condense in the highly subcooled bulk fluid flowing in the test tube.

Figure 6 shows the effect of $L/d$ on CHF at a flow velocity of 10 m/s and inlet subcoolings ranged from 133 to 145 K. For comparison, the experimental data of Vandervort et al. (1994) are plotted in Fig.6. It was found that the CHF increases with decreasing of $L/d$.

Figure 7 shows the relationship between CHF and the e-folding time at a flow velocity of 10 m/s and an inlet subcooling of 145 K. As shown in Fig. 7, CHF increases as the e-folding time decreases. For longer e-folding time, the CHF seems to approach a constant corresponding to a steady-state condition, contrarily it increased for shorter e-folding time, which corresponds to an unsteady state. Since the heat generation rate of the test tube increased rapidly under the transient condition, it is considered that the heat flux of the test tube increased until vapor film covered the heated surface of the test tube. In Fig.7, a dashed line represents the transient CHFs calculated by the following equation, which takes into account the e-folding time (Hata et al., 2004) but can be applied to an inner tube diameter of 3 to 9 mm. It is found that the CHF is higher than that predicted by Eq. (19) between the long e-folding time (steady-state conditions) and the short e-folding time (transient conditions) in the small tube.

$$Bo_{\mu} = Bo_{\text{st}} \left[1 + 11.4 \left(\frac{gT}{\sqrt{g/\rho L(\rho_f - \rho_v)}}\right)^{-0.6}\right]$$

for inlet subcooling ($\Delta T_{\text{sub,in}} \geq 40$ K ) \hspace{1em} (19)

4. Conclusions

The transient CHF of subcooled boiling of water flowing through a small inner diameter tube with exponentially increasing heat inputs was experimentally investigated. The fundamental data on the transient CHF in the small diameter tube ($d < 3$ mm) were obtained. The experimental results revealed that the CHF was higher than that predicted by Eq. (19) obtained by Hata et al. (2004) under steady state and transient conditions. Moreover, it was found that the
CHF was affected by the e-folding time under transient conditions.

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