



High Heat Flux Boiling Heat Transfer and Its Application on Cooling System

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(Degree)

博士 (工学)

(Date of Degree)

2011-03-25

(Date of Publication)

2011-11-09

(Resource Type)

doctoral thesis

(Report Number)

甲5307

(URL)

<https://hdl.handle.net/20.500.14094/D1005307>

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Doctoral Dissertation

**High Heat Flux Boiling Heat Transfer and
Its Application on Cooling System**

(高熱流束沸騰熱伝達とその冷却システムに関する研究)

January 2011

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In the study of high heat flux boiling heat transfer, a series of investigations on the pool and two-phase flow boiling heat transfer has been performed. The pool boiling investigations include the measurements and observations on finite horizontal heaters in a pool of dielectric liquid FC-72 by the exponential increasing heat input, $Q_0 e^{t/\tau}$, under a wide range of pressures and liquid subcoolings. It was observed three kinds of transition boiling processes to film boiling. One of the important mechanisms is the transition from the heat conduction process to film boiling, namely ‘direct transition’, which gives an explosive-like incipient boiling and then directly followed by critical heat flux (CHF) involving a significant formation of film boiling through the dominance of the vapor film and surface contact. However, during quasi-steady-state heat transfer, it was observed also that the direct transition also occurs from single phase natural convection at around atmospheric pressure and by an extent of pre-pressure on surface heater. Direct transition phenomena were confirmed to exist due to the explosive-like heterogeneous spontaneous nucleation (HSN) in originally flooded cavities on surface. In this study, the predictions of direct transition phenomena were also derived from typical incipient boiling superheat and CHF.

The two-phase flow boiling studies were performed through the investigations on the closed loop thermosyphon, which was developed for cooling of the high-power electronic devices such as IGBT module for the inverter of the electric propulsion drive onboard ship. By the experimental investigations, the CLT device could show a steady-state performance and an efficient high heat transfer process in the higher heat inputs rather than the lower ones. CLT is considered as a smart cooling device and is suited for thermal management of the shipboard high-power electronics which could perform self adjustment of pressure control and cooling process in response to the heat input to the heater.

Acknowledgement

Alhamdulillah, first of all, I have to thank Allah Subhanahu wa ta'ala for His many blesses in my life. Also I would like to thank my first and foremost supervisor, Professor Fukuda, for his great support, attention, and wonderful relationship. I have been very fortunate to have had the opportunity to work with a supervisor that is very kind and has so much to offer and is willing to share it so freely. I would like also to thank to Professor Liu for support and encouragement in this study and also to all members in the committee of final dissertation examination, and all laboratory members for their supports.

I must thank my parents, my mother (almarhumah) and father, brother and sisters for all of the love and support they have given me throughout my life, along my entire study in Japan. 'It was very hard pursuing the latest year of doctoral program, but finally, I should get to that final milestone'. Last but not least, thank you so much to my son and daughter, Yutaka and Yumi, and the younger Yusuke, also to my lovely wife Diah, there is no limit to my appreciation for your support, pray, love and sacrifice over the years.

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Nomenclature

A	constant in Eq. (4.1)
a	thermal diffusivity, m^2/s
B	constant in Eq. (4.2)
Bo	$(=q/(Gh_{lg})),$ boiling number
c_p	specific heat at constant pressure, J/kgK
D	cylinder diameter, m
F_{fl}	fluid-surface dependent parameter, $F_{fl} = 1$ (for water)
G	mass flux, $\text{kg}/\text{m}^2.\text{s}$
g	acceleration of gravity, m/s^2
h	heat transfer coefficient, $\text{W}/\text{m}^2\text{K}$; specific enthalpy, J/kg
h_c	heat transfer coefficient due to transient heat conduction, $\text{W}/\text{m}^2\text{K}$
h_{com}	combine heat transfer coefficient, $\text{W}/\text{m}^2\text{K}$
h_{lg}	latent heat of vaporization, J/kg
h_{lo}	single-phase heat transfer coefficient, $\text{W}/\text{m}^2\text{K}$
h_n	heat transfer coefficient due to steady-state natural convection, $\text{W}/\text{m}^2\text{K}$
h_{sp}	two-phase flow heat transfer coefficient, $\text{W}/\text{m}^2\text{K}$
K	Kutateladze number, constant in Eq. (1.5)
k	thermal conductivity, W/mK
Nu	Nusselt number
P	pressure, kPa
P/P_{CR}	reduced pressure, where $P_{CR} = 1830 \text{ kPa}$
Q	heat generation rate, W/m^3
Q_0	initial exponential heat input, W/m^3
q	heat flux, W/m^2
q_{CR}	$(=q_c), (=q_{max}),$ CHF, W/m^2
$q_{CR,sat}$	steady-state CHF for saturated condition, W/m^2
q_i	incipient boiling heat flux, W/m^2

Re	$(=\rho u D/\mu)$, Reynolds number
r	cylinder radius, m
T	temperature, K
T_{CR}	critical temperature of FC-72 = 450 K
T_H	homogeneous nucleation temperature, K
T_a	average temperature of the test heater, K
T_{sp}	spinodal limit temperature, K
T_w	$(=T_s)$, heater surface temperature, channel's wall temperature, K
T_{sat}	saturation temperature, K
ΔT_i	incipient surface superheat, K
ΔT_{sat}	$(=T_w - T_{sat})$, surface superheat, K
t	time, s
U	overall heat transfer coefficient, W/m ² .K
u	velocity, m/s
v	specific volume, m ³ /kg

Greek Symbols

α	surface tension, N/m
ρ	density, kg/m ³
μ	$(=[\rho_l c_{pl}/k_l \tau]^{0.5})$, m ⁻¹ , viscosity, kg/m.s
τ	exponential period, s
ν	kinematic viscosity, m ² /s

Subscripts

CR	thermodynamic critical condition
l	liquid
g	gas

Chapter 1 – Introduction

“Liquid cooling is back.”

Roger R. Schmidt (2005)

1.1 Motivation

The double increase of thermal requirement towards the development of the next generation of marine and naval vessels should be anticipated by the application of a reliable thermal management. Appropriate cooling systems dealing with large and various high heat flux sources onboard ship are needed to solve problem of lacking in thermal performance. Driven by the need of study on the thermal requirement solution or method for heat transfer process on the high-power electronic devices onboard modern ships, such as insulated gate bipolar transistor (IGBT), the research works presented in this dissertation is aimed at the correct understanding of boiling heat transfer process to design and evaluate systems that use boiling for cooling.

Boiling heat transfer with phase change processes, such as pool and flow boiling are generally very effective modes of heat transfer. However, enhanced boiling systems by the appropriate cooling method need to be developed to accommodate demands of modern thermal system. In the engineering design of the thermal systems or heat generation systems as well as safety evaluation viewpoints, correct understanding of boiling heat transfer phenomena is very important for engineers, such as nucleate boiling heat transfer process, the critical heat flux (CHF) as the limit or critical point of the effective heat transfer in the boiling systems, and also, when the heat systems involve convective boiling on a flowing liquid in such a way that the generation of vapor occurs, where usually involves variable interests such as the rates of heat transfer and the pressure losses, which might impose some limitation upon the performance of system.

In this research, a series of investigations on boiling heat transfer processes is the prime objective to perform the coherent study on the basic requirement of parameters to design heat generation systems employing phase change boiling process. It comprises study on the two-phase flow boiling and its required parameters to analyze the developed cooling system, i.e.: the two-phase closed loop thermosyphon (CLT), which includes examinations on thermal performances, pressure drops, and CHF; as well as study on the transient pool boiling heat transfer in highly wetting liquid FC-72 to collect the fundamental database which is considered as an important knowledge for correct understanding the boiling phenomena to design cooling systems in the electronic packaging technology.

1.2 Background

1.2.1 Shipboard high-power electronic coolings

Requirement of reliable cooling system for any heat sources or thermal plants onboard modern ships has been influenced by the linearly increase of the heat loads which take account the increase in demand of power based on ship's size and speed requirements, as seen in Fig. 1.1. Moreover, recent developments of the new-build 'Eco-Ship' and implementation of fully electric ship (FES) with application of the electric propulsion drive (EPD) would increase the exhaust heat loads onboard ship, which are estimated to grow about 30%-40% higher than conventional ships.

In the last two decades, the EPD has been widely evaluated and was proposed as one of the reliable, powerful, and efficient propulsion systems for modern ships (Skjoldager, 2005)(Maritime Reporter Magazine, 2002)(Sutopo, 1998). However, large implementation of advanced EPD and high-power electric propulsion equipments and installations, such as electric motor, inverter/converter, transformer, high-power cable, electronic propulsion system, high-power electronic cabinets, etc, will certainly increase the large exhaust heat loads and the cooling capacity in addition to the common exhaust heat from such auxiliary engines in the engine room. More additional heat loads are produced also by high

requirement of HVAC (Heating, Ventilating, and Air Conditioning) systems and the use of high-power electronic devices onboard modern ships such as high-end computer power and chips under marine environment, electronic communication devices, and also another typical requirement on navy combat electronic devices, etc. Thus, it makes sense to inquire thermal handling device or thermal management which is reliable to accommodate the heat load levels onboard ship.

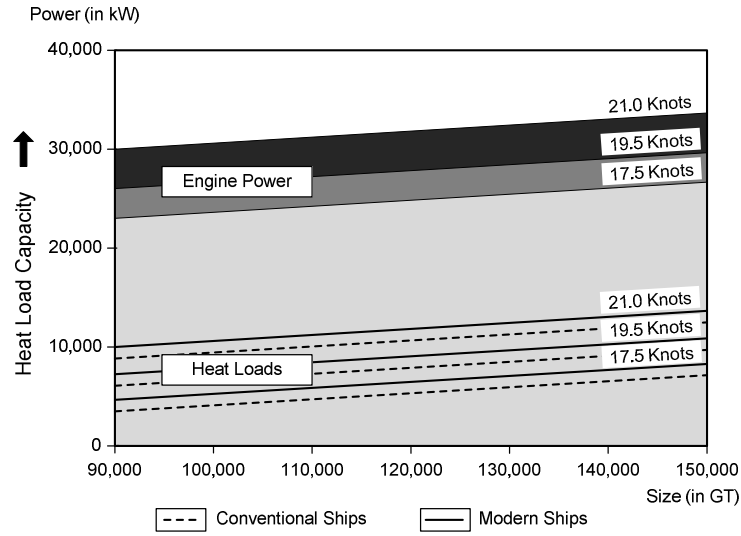


Fig. 1.1 Estimation of increasing heat load onboard modern ships.

One of the important developments in the advanced EPD is the frequency converter which includes several configurations depend on the type of electric motors, electric drive plant, and ship's maneuvering requirements, such as cyclo-converter, current source inverter (CSI), voltage source inverter (VSI), and silicon controlled rectifier (SCR). The most developed area in the inverter / converter or power conversion module is the high-power electronic IGBT module. It is a component system comprised IGBT semi conductor chips to accommodate high-power transmission and fast switching in the power conversion method. In the FES, it was found that the semi conductors accounted for over 70% of the total heat generated while occupying just over 5% of the volume (Hartenstein et al., 2007). The development of cooling method on the high-power electronic semi

conductor is driven by the continuing increases in power dissipation of electronic parts and systems. In fact, for some conditions, the traditional or standard cooling techniques such as natural and forced air cooling cannot achieve the required cooling performance due to physical limitations on the capability to handle anticipated heat dissipation requirement of high-power electronics.

The available cooling methods for IGBT module or chip include conventional air or fan cooling, bonded fin heat sinks, aluminum extrusions, etc. Nowadays, advanced cooling techniques such as spray cooling on IGBTs, impingement cooling, heat pipe with air fan-cooled heat sinks, liquid-cooled cold plates are developed due to requirement for anticipating IGBT inverter / converter with megawatts of power by switching thousands of amps at thousands of volts. IGBT has high efficiency of power conversion with small power losses during switching. Although the losses during switching are a small percentage of the conditioned power, they still result in kilowatts of heat dissipation in each IGBT. The heat losses approximately 2%-5% of power converted, so in the 98% efficient 100 kW converter, it will require at least 2 kW of cooling capacity. As the electronic devices or packages become smaller, the losses lead to very large heat fluxes at the die that might reach 300 W/cm^2 and beyond. Therefore, an appropriate cooling system is needed to satisfy the demanding cooling capacity of IGBT or the high-power semi conductors.

1.2.2 Cooling methods based on boiling heat transfer

In Fig 1.2, the comparison of various cooling methods as a function of the attainable heat transfer in terms of the heat transfer coefficient is figured (Lasance and Simons, 2005). Lasance (1997) in Lasance and Simons (2005) figured out that $20,000 \text{ W/m}^2\text{K}$ heat transfer coefficient is required to accommodate a heat flux of 100 W/cm^2 at a temperature difference of 50 K. It was concluded also that liquid cooling is needed in the future of thermal management technologies to anticipate the emerging electronics applications (Lasance and Simons, 2005).

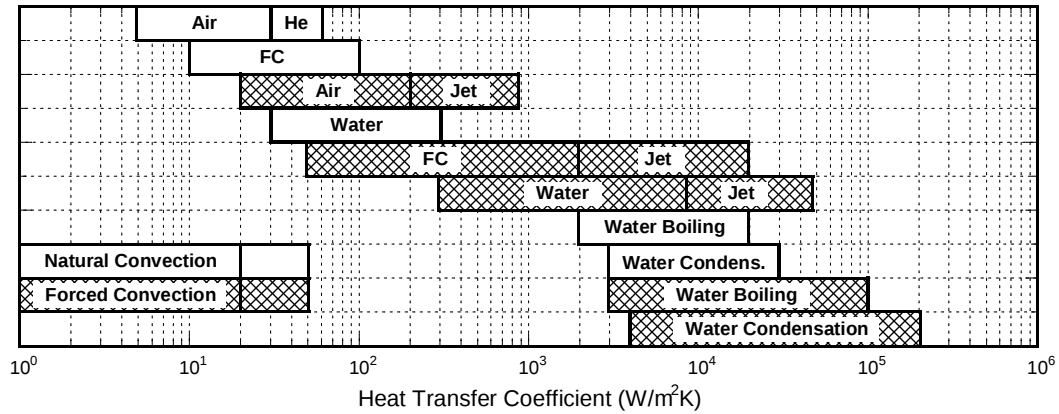


Fig. 1.2 Heat transfer coefficient attainable with natural convection, single-phase liquid forced convection and boiling for different coolants (Lasance and Simons, 2005).

Liquid cooling for application to electronics is generally separated into two categories of *direct* and *indirect* liquid cooling. Those are related to whether the liquid coolant directly contacts with the heated components to be cooled or does not directly contact with the ones to be cooled, respectively. Direct liquid cooling includes cooling techniques in the *pool boiling* arrangement where the component is directly being cooled and sits in relatively stagnant pool of coolant, such as immersion cooling and jet impingement. Indirect cooling includes heat pipes, thermosyphon, and cold plates cooling techniques.

Direct liquid cooling has been extensively studied. Two issues addressed in most of the studies are the reduction of incipience excursion and increasing the critical heat flux (CHF) with various enhanced surfaces. A result from measurement was reported by Danielson et al. (1987) showed the CHF for saturated boiling of FC-72 at atmospheric pressure on platinum wires is approximately 17 W/cm². Other measurements on classic experimental horizontal wires in a pool of saturated water and ethanol at atmospheric pressure, the steady-state CHF could attain 150 W/cm² (Sakurai, 2000) and 50 W/cm² (Fukuda et al., 2004), respectively. However, in the study of high heat flux boiling heat transfer in terms of employing boiling for cooling and involves phase change (two-phase flow) in the liquid-gas contact, flow boiling mechanism has

significant advantages over pool boiling, such as enhancing nucleate boiling, increasing CHF, and reducing the temperature overshoot. It offers also the advantage of a lower flow rate for dissipating heat fluxes compared to that of flow in a single-phase pattern.

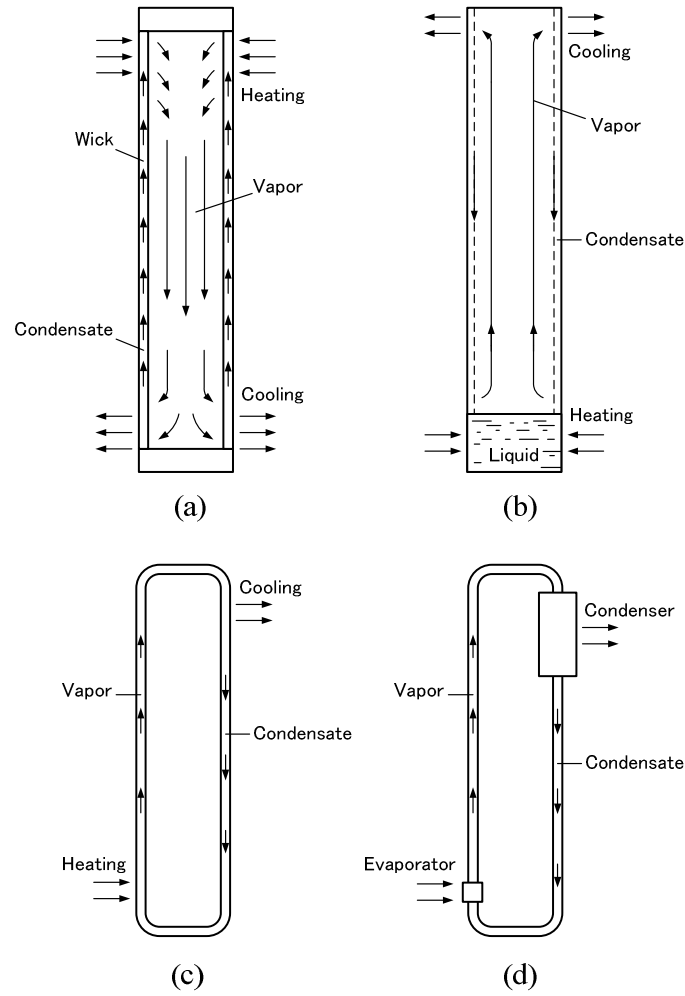


Fig. 1.3 Heat pipe and thermosyphons: (a) heat pipe, (b) thermosyphon (pipe), (c) simple loop thermosyphon, (d) advanced closed loop thermosyphon.

1.2.3 Two-phase flow closed loop thermosyphon (CLT)

Thermosyphon with its heat transfer characteristics has been widely applied in many areas of industrial applications such as gas turbine blade cooling, electrical machine rotor cooling, transformer cooling, nuclear reactor cooling,

internal combustion engine cooling, and electronic cooling. It is a passive device that offers significant cost and reliability advantages over systems requiring pumps with employing flow boiling heat transfer process. CLT is an advanced type of gravity-assisted thermosyphon as one of heat pipe types without wick. As seen in Fig. 1.3 (d), an advanced closed loop thermosyphon with attached evaporator and condenser offers possibility of reducing total cross section area of the connecting tubes and closed contact between the component and the coolant channels than a thermosyphon pipe or a heat pipe. Thermosyphonic devices can be also categorized into 3 types:

- (a) *Open thermosyphon*
- (b) *Closed thermosyphon*:
 - Pipe thermosyphon (single-phase / two-phase flow)
 - Simple loop thermosyphon (single-phase / two-phase flow)
- (c) *Advanced closed loop thermosyphon / CLT* (two-phase flow)

Gravity-assisted and pumpless loop flow thermosyphonic device constitutes a “*smart*”, self driven system (Mukherjee and Mudawar, 2003). The term smart is designated to the passively response capability of a system to enhance performance without external control input, such as a variable speed pump, to respond to the heat input. The CLT consists of an evaporator, a condenser, and connecting tubes in between, i.e.: riser (hot vapor flow) and downcomer tubes (liquid condensate flow). Heat is transferred from the source to the evaporator, where the fluid vaporizes. The vapor moves to the condenser and the released heat is dissipated to the ambient. The liquid condensate is returned to the evaporator, thus completing a closed loop. The density difference between the liquid and vapor creates a pressured head, which drives the flow through the loop. The CLT has a simple structure and reliable to transport heat in long distances with small decrease in temperature. It has been also considered as a promising heat transfer device onboard ship and is suited for robust shipboard cooling requirements (Hartenstein et al., 2007). Thermosyphon also can be set-up as to whether the evaporator is in a pool boiling or flow boiling arrangement.

Generally, thermosyphon sums up the advantages as follows:

- (a) It offers passive circulation and the ability to dissipate high heat fluxes even from small areas with low temperature differences between evaporator wall and coolant, due to high latent heats of evaporation and condensation.
- (b) The thermosyphon circumvents the capillary limitation of heat pipes by relying on gravitationally induced buoyancy to drive the flow.
- (c) Flexibility in size and cooling capacity range.
- (d) Heat can be transferred long distances without any (or with very small) decrease in temperature.
- (e) A closed loop configuration also avoids the flow impedance that occurs in a gravity fed wickless heat pipe when rising vapor interacts with falling liquid.

Nowadays, in order to accommodate the heat dissipation levels prevalent in electronic devices that continue to escalate, various cooling techniques have been proposed as a solution. Flow boiling in *small hydraulic diameter channels* is considered to be an important scheme to face future requirement of electronic cooling techniques. However, there are critical issues in the flow boiling process in the small diameter channels which exist in relation to the pressure losses and flow instability. In the reported literature, the evaporator of thermosyphon for electronics cooling has been also improved by applying narrow channels (*minichannels* or *microchannels*) to increase the thermal performance. The definition of small hydraulic diameter channels or narrow channels might refer to the ranges of diameters which are attributed to different channels, as suggested by Kandlikar (2002).

- (a) *Conventional channels*: > 3 mm
- (b) *Minichannels*: $200\ \mu\text{m}$ to 3 mm
- (c) *Microchannels*: 10 to $200\ \mu\text{m}$

1.3 Overview of the current study and organization of thesis

Facing the escalation of anticipated heat dissipation requirements on the high-power electronic devices such as IGBT module onboard ship that is due to large implementation of EPD system and FES system, the developments of new and reliable cooling methods and the improvements of cooling techniques are highly required. Cooling the high-power electronic devices with heat dissipating more than 300 W/cm^2 at the die is beyond the capability of most conventional cooling by natural and forced air-cooling. The above background and motivation led to the experimental and analytical study on the high heat flux boiling heat transfer processes from the pool boiling investigations to the application of two-phase flow boiling through the development of closed loop thermosyphon. Correct understanding of boiling heat transfer processes based on the experimental study on the pool boiling including study on two-phase flow and its mechanisms in narrow channels and specified cooling systems, can help engineers to design and evaluate heat generation based systems for different applications.

In this research, study of pool boiling heat transfer is focused on the investigation of CHF and its mechanisms in the transition to film boiling from nucleate boiling and also non-boiling convection and conduction processes in a pool of highly wetting liquid FC-72. Investigation of pool boiling phenomena in the dielectric fluorocarbon liquid such FC-72 due to transient heat inputs under a wide range of pressures and subcoolings has not been extensively done yet. FC-72, the dielectric fluorocarbon liquid, is considered as the highly wetting liquid which has 0.008 N/m of low surface tension property, lower than liquid nitrogen, ethanol, and about one-eighth lower than water. FC-72 often used as the coolant of electronic package cooling system. In the microelectronic cooling, low boiling point of FC-72 at around 56°C offers advantages of an ideal temperature for operating electronics and also allowing the final heat sink or condenser to deposit its heat to ambient air without a compressor.

The pool boiling was measured on horizontal heaters due to transient heating (exponentially increasing of heat rates ranging from quasi-steady to

rapidly increasing ones), $Q_0 e^{t/\tau}$, under a wide range of pressures and liquid subcoolings. The fundamental database of CHF and pool boiling heat transfer phenomena including those predictions are obtained and presented.

The fundamental two-phase flow boiling study was conducted through the development and application of two-phase closed loop thermosyphon for cooling high-power electronic semi conductors such as IGBT module in the frequency converter of EPD system. The experimental and analytical studies of the two-phase flow boiling cover several interest parameters which are required to design thermal systems and important for correct understanding the two-phase flow phenomena in a specified application. It is focused on the examination and analysis of the thermal performance of CLT. In the current work, low pressure saturated water was considered as the working fluid.

The current work concludes all studies that are described in each chapter, where a chapter has unique descriptions and serves as the linked materials for other chapters. Following below are the outline descriptions of the work.

- (a) *Chapter 1* addresses the motivation and objective of the research including the introductory aspects of boiling heat transfer processes, variable of cooling methods, and the requirement of high-power electronic cooling onboard ship.
- (b) *Chapter 2* continues to review the related literatures. The literature survey starts from the related studies of CHF in the pool boiling heat transfer due to transient heat inputs, the effect of pressure and subcooling, and direct transition phenomena and related correlations. Review continues on two-phase flow boiling studies, starts from flow boiling heat transfer, and then CLT studies.
- (c) *Chapter 3* contains description of the experimental apparatus completed with experimental methods and procedures.
- (d) *Chapter 4* contains the experimental results and analysis in the study of pool boiling heat transfer. Data are identified, compared, analyzed, and discussed. The experimental results and discussion are related to

several subjects as follows:

- the time dependence of boiling characteristics of heat generation rate, heat flux, and surface temperature
- boiling heat transfer processes under a wide range of pressures include kinds of boiling transition process, the incipient boiling and CHF surface superheats, and the effect of pre-pressure
- the transient CHF and the predictions of incipient boiling superheat and CHF for direct transition phenomena.

(e) Chapter 5 continues the analysis of the experimental results include the discussion on several subjects in the study of two-phase flow closed-loop thermosyphon:

- thermal performances of CLT
- heat transfer coefficient

(f) *Chapter 6* contains the summary of the study and the key contribution of this dissertation.

(g) *Appendix* contains the references of data that are required in this study.

Chapter 2 – Literature Survey

“How much heat could possibly be carried away by boiling?”

Lienhard, Engines of Our Ingenuity, No. 448

The subject of boiling heat transfer with phase change and also fluid flow is an extremely important one. This arises from the fact that many industrial processes rely on these phenomena for design matters, material processing, and also energy transfer point of view. The design and evaluation of power plants, refrigeration and air-conditioning equipment, heat pumps, petroleum processing, condensers, electronic cooling systems, and many other systems are dependent upon knowledge of those two studies. There are also many examples of process dealing with phase change heat transfer and fluid flow in everyday life, such as boiling water or coffee, rain, snow, steam condensation on walls or a glass window in house during cold winter.

Related to boiling heat transfer and fluid flow, this subject is separated into two mechanisms of boiling, i.e.: *pool boiling* that refers to boiling processes without an imposed forced flow, and *flow boiling*, which includes forced and natural convection flow boiling that cover heat transfer processes with and without external power to drive flow of fluid, respectively. Flow boiling is considerably more complicated than pool boiling, owing to the coupling between hydrodynamics and boiling heat transfer processes (Ghiaasiaan, 2008).

2.1 Pool boiling heat transfer

Boiling is an effectively process carrying heat from a heater into a liquid which causes evaporation. The evaporation process occurs when the temperature of the surface T_w exceeds the saturation temperature corresponding to the liquid pressure T_{sat} . Heat is transferred from the solid surface to the liquid,

and applying the Newton's law of cooling, the heat flux, $q = h(T_w - T_{sat}) = h\Delta T$, and the heat transfer coefficient, h , is used to characterize the pool boiling process over a range of ΔT .

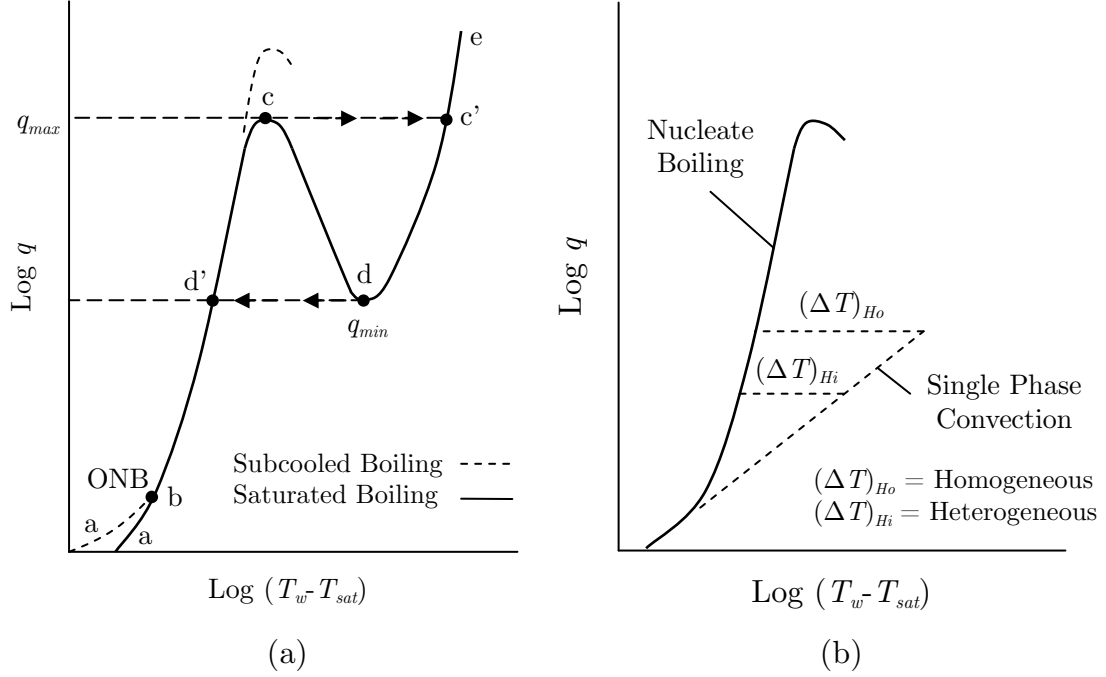


Fig. 2.1 (a) Pool boiling curve regimes, (b) wall superheat excursion (boiling hysteresis).

Pool boiling is defined as boiling from a heated surface submerged in a large volume of stagnant liquid (Collier, 1994). This liquid may be at its boiling point, in which case the term *saturated pool boiling* is employed or below its boiling point, in which case the term *subcooled pool boiling* is used. The results of investigations into heat transfer rates in pool boiling are usually plotted on a graph of heat flux, against the wall / surface superheat ($T_s - T_{sat}$), named “*boiling curve*”.

The subject of pool boiling should start with the pool boiling curve and Nukiyama's experiment in 1934 (Nukiyama, 1934). One of the important observations made during the experiment is about boiling curve that suggests five regimes of behavior, are natural convection, nucleate boiling, peak heat flux or critical heat flux (CHF), transition boiling, and film boiling (Lienhard IV and

Lienhard V, 2006). Figure 1.1 (a) shows the representation regimes on the boiling curve.

(a) *Single-phase natural convection (a-b)*. In this regime there is insufficient vapor in contact with the liquid phase to cause boiling at the saturation temperature. As the wall superheat is increased, bubble inception or the onset of nucleate boiling (ONB) at point *b* will eventually occur, but below the boiling inception point, fluid motion and heat removal are determined principally by free convection effects.

(b) *Nucleate boiling (b-c)*. The nucleation of vapor occurs at the heating surface. Starting with a few individual sites at low heat fluxes the vapor structure changes, as the heat flux is increased, as a result of bubble coalescence and finally, at high heat fluxes, vapor patches and columns are formed close to the surface (Collier 1994). A boiling hysteresis or wall superheat excursion often exists in the nucleate pool boiling of highly wetting liquids. The schematic representation of the wall superheat excursion as shown in Fig. 1.1 (b) is given in Bar-Cohen and Simon (1988).

(c) *The critical heat flux (q_{max} or CHF at point c)*. The maximum point of the nucleate boiling heat flux density is usually termed the *critical heat flux* (CHF). It marks the upper limit of nucleate boiling where the interaction of the liquid and vapor streams causes a restriction of the liquid supply to the heating surface.

(d) *Transition boiling (c-d)*. This region is termed *transition boiling*, *unstable film boiling*, or *partial film boiling* (Incropera and De Witt, 1990). It is characterized by the existence of an unstable vapor blanket over the heating surface that releases large patches of vapor at more or less regular intervals. Intermittent wetting of the surface is believed to occur (Collier, 1994).

(e) *Film boiling (d-e)*. When the heat flux reach the minimum point of heat flux, q_{min} , or the *Leidenfrost point*, at point *d*, the surface is completely

covered by a vapor blanket. Heat transfer from the surface to the liquid occurs by conduction through the vapor film instead of through a liquid film. Once a stable vapor blanket is established, the surface temperature is increased, radiation through the vapor film becomes significant and the heat flux again increases with increasing wall superheat.

2.1.1 Critical heat flux (CHF)

The heat flux, q , is a heat rate per unit area and can be expressed as Q/A , where A is an appropriate area. The q_{max} represents an important point on the boiling curve. This point is known by a variety of names: the *burnout point*, the *peak heat flux*, the *boiling crisis*, the *DNB*, or *departure from nucleate boiling*, *critical heat flux*, and the *first boiling transition*. Author designates q_{max} or q_{CR} as the critical heat flux (CHF).

The CHF gradually increases when the pressure is increased. This effect is limited by a certain maximum pressure, which is at about 1/3 of the critical pressure. This important result was found experimentally by Cichelli and Bonilla (1945) in organic liquids. They observed also that the CHF approaches zero nearing the critical pressure. Investigations on pool boiling CHF's under a wide range of pressures due to exponentially heat inputs, $Q_0 e^{t/\tau}$, in water and wetting liquids such as liquid nitrogen, ethanol, and liquid helium I have been performed also by Sakurai and his associates (1992, 1993, 1995, 1996, 2000a, 2000b, 2002). In the pool of water, the steady-state CHF's under a wide range of pressures and subcoolings on 1.2 mm diameter horizontal cylinder were measured. Steady-state CHF's correspond to the value of CHF's due to the quasi-steadily increasing heat input given by exponential time function with the period of 20 s.

The form of the preceding prediction of CHF is usually credited to Kutateladze (1959) and Zuber (1959) through their hydrodynamic instability analysis. Kutateladze did the dimensional analysis of CHF, q_{CR} , based on the flooding mechanism and obtained the following relationship but valid only for an

infinite horizontal plate:

$$q_{CR} = q_{CR,sat} = Kh_{lg}\varrho_g(\sigma g(\varrho_l - \varrho_g) / \varrho_g^2)^{1/4} \quad (2.1)$$

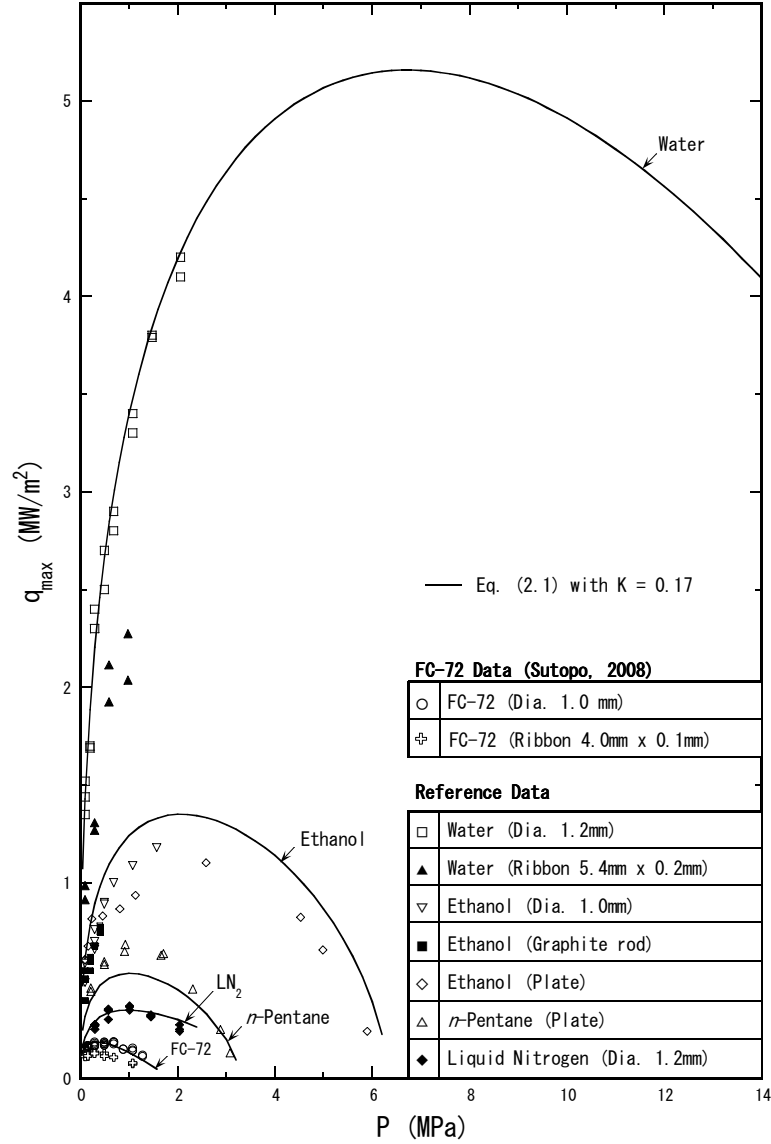


Fig. 2.2 The CHF – departure of nucleate boiling in various liquids and heater configurations (Sutopo, 2008).

Figure 2.2 shows saturated pool boiling CHF taken from experimentation with different configurations in liquids and heater geometries under saturated conditions (Sutopo, 2008). It can be seen that the maximum value of the

steady-state CHF exist at about 1/4 of the critical pressure and approaching a zero value near the critical pressure. FC-72 almost reaches a lower value of CHF 1/10 than water at atmospheric pressure. The prediction values calculated from Kutateladze's correlation in Eq. (2.1) were also plotted for comparison.

2.1.2 Transient pool boiling heat transfer

Extensive studies on the pool boiling heat transfer in non-wetting and wetting liquids under wide ranges of pressures and subcoolings due to exponentially increasing heat inputs, $Q_0 e^{t/\tau}$, were reported by Sakurai and his associates (1977a, 1977b, 1990a, 1992, 1993, 1995, 1996, 2000a, 2000b, 2002), Fukuda et al (2000, 2004), and Park (2006). The wide investigations on water and other wetting liquids such as ethanol, liquid nitrogen, and liquid helium-I have been performed include the investigation on CHF mechanisms based on transition to film boiling, incipient boiling process, typical transient CHF for a wide range of exponential periods, pressures and liquid subcoolings, effect of surface condition on CHF, and the photographic observation on vapor and bubble behavior at transitions to film boiling. The exponential periods represent increasing rate of heat inputs and surface superheats. Kasakawa et al. (1999) and Ohya et al. (2001) concluded their investigations of transient CHF in a pool of FC-72 under low subcooled conditions up to 50 K.

2.1.2.1 Typical transition to film boiling at CHF

Sakurai and his associates (1992, 1993, 1995, 1996, 2000a, 2000b, 2002) have investigated the transition boiling mechanisms to film boiling from non-boiling regimes or nucleate boiling at CHF on solid surface in water and wetting liquids such as ethanol, liquid nitrogen, and liquid helium I. They observed three kinds of transition of boiling heat transfer process are the transition from the non-boiling regime to film boiling or *direct transition* (type 1), the *transition from non-boiling to fully developed nucleate boiling (FDNB)* (type 2), and the transition from non-boiling regime to film boiling with increasing heat flux or *semi-direct*

transition (type 3).

It was observed that the direct transition to film boiling without nucleate boiling from natural convection regime due to a quasi-steadily increasing heat input at around atmospheric pressure and from transient conduction regime due to increasing heat inputs at higher pressures in liquid nitrogen. The direct transition gives lower rates of the transition surface superheat than the corresponding homogeneous spontaneous nucleation surface superheat. They also observed even for water, the semi-direct transition process to film boiling from transient conduction regime with short nucleate boiling from active cavities (the first and the second kind transitions). The direct transitions at incipient boiling surface superheat in the liquids were well explained by the assumption that the incipient boiling occurs due to the explosive boiling, named the *explosive-like heterogeneous spontaneous nucleation* (HSN) in originally flooded cavities at the HSN surface superheat.

2.1.2.2 Incipient boiling heat transfer process

It was observed there are two types of boiling incipience on cylinder surface in a liquid due to an increasing heat input; one is a bubble formation from active cavities, and the other is the explosive heterogeneous nucleation without the contribution of the active cavities. Sakurai and his associates (1992, 1993, 1995, 1996, 2000a, 2000b, 2002), Fukuda et al (2000, 2004), Park (2006), Kasakawa et al. (1999) and Ohya et al. (2001) reported in literatures the two kinds of incipient boiling process. The former type of incipient boiling from active cavities was observed in water, and the latter as in liquid nitrogen, liquid helium, ethanol, FC-72, and water pre-pressurized by appropriately high pressures. The theoretical model of HSN surface superheat and the latter boiling mechanism were suggested by Sakurai et al. (1993) to be due to the HSN from originally flooded cavities on the cylinder surface in a liquid.

Effect of pre-pressurization on the incipient boiling process has been also investigated by Sakurai (2000a) and Mizukami et al. (2001, 2005). Sakurai

(2000a) adding highly pre-pressures to eliminate the occurrence of nucleate boiling from active cavities, and obtained the existence of the other incipient boiling mechanism different from that due to active cavities. Mizukami et al. (2005) investigated the effect of the suppress pressure added on heater in a pool of FC-72 during experimental run to observe the characteristics of the incipient boiling process and suggested the required cavity mouth diameter for boiling incipience. He also found that the boiling inception condition for each vapor nucleus is independent of the system pressure.

One of important mechanisms in the boiling heat transfer process and should be responsible for occurring spontaneous nucleation is the incipient boiling process. The boiling inception model at certain required superheat limit is important to be quantified for its prediction and accuracy of evaluation on boiling heat transfer system. The predictions of the incipient boiling superheat have been suggested to include the homogeneous nucleation temperature, T_H , close to the thermodynamic spinodal limit, which was suggested by Lienhard (1982) in Eq. (2.2). Moreover, in the prediction of incipient boiling superheat for engineering approximation, it mostly takes account the heterogeneous nucleation superheat. In the pool boiling of highly wetting liquid, the prediction of heterogeneous nucleation superheat often involves the phenomena of boiling hysteresis before the onset of nucleate boiling, which considers the effect of surface tension and the prediction of effective radius of vapor bubble on surface including the Rohsenow's estimation for fully developed nucleate boiling, as seen in Eq. (2.3) for the prediction of the incipient boiling hysteresis, ΔT_{IBH} (Bar-Cohen and Simon, 1988). In the heterogeneous spontaneous nucleation, prediction of incipient boiling superheat as the lower limit HSN superheat may also refer to the contact interface temperature of cylinder surface, T_f , which closely agrees with the minimum film boiling temperature, as suggested in Eq. (2.4) by Sakurai et al. (1990b).

$$T_{sp} - T_{sat} \cong T_{CR} \left[0.923 - \frac{T_{sat}}{T_{CR}} + 0.077 \left(\frac{T_{sat}}{T_{CR}} \right)^9 \right] \quad (2.2)$$

$$\Delta T_{IBH} = T_{sat}(P_v) - T_{sat}(P) - C(q_i)^b; \text{ where } P_v = P + \frac{2\sigma}{r} - P_{gas} \quad (2.3)$$

$$\Delta T_I = T_I - T_{sat}; \text{ where } T_I = 0.92T_{CR} \left\{ 1 - 0.26 e^{\left[\frac{-20(P/P_{CR})}{1+1700/P_{CR}} \right]} \right\} \quad (2.4)$$

In the dielectric liquid, the temperature overshoot or boiling hysteresis often exists during incipient boiling process with drop of surface temperature and heat flux that may pose to be a limiting condition in the design of cooling device, as seen in Fig. 2.1(b). Bar-Cohen and Simon (1988) had discussed in detail the theoretical reasons for wall superheat overshoots at the boiling incipience of dielectric fluids. Many investigations were aimed at the reduction in incipience temperature overshoot.

2.1.2.3 Transient CHF for period

Sakurai (2000a) have reported an experimental procedure by adding the high pre-pressure for a while before each experimental run on the surface heater and observed a dissipation of active cavities on surface then induce the direct transition to film boiling without nucleate boiling. The occurrence of direct transition was assumed due to HSN in originally flooded cavities on the solid surface.

Sakurai (2000a) also have reported typical of transient CHF, q_{CR} , for the exponential periods, τ , in the cases without and with pre-pressurization (cases 1 and 2, respectively) in the pool of water. The rates of CHF gradually increase, then decrease, and again increase with the decrease in exponential period. They were classified clearly into the first, second, and third groups for long, short, and intermediate periods, respectively. The third group lies between the first and the second groups. The values of q_{CR} for periods under saturated condition at pressures ranging from 101.3 to 2063 kPa for the cases 1 and 2 are shown in Fig. 2.3. The q_{CR} of the first group at the pressures are expressed by the following empirical equation given by Sakurai et al. (1995).

$$q_{CR} = q_{CR,sub} (1 + 0.21\tau^{-0.5}) \quad (2.5)$$

where $q_{CR,sub}$ is the correlation for steady-state CHF that was derived by modifying the Kutateladze's correlation in Eq. (2.1) taking into account the non-linear effect of high liquid subcoolings on the q_{CR} . In the saturated conditions, $q_{CR,sub}$ equals Kutateladze correlation.

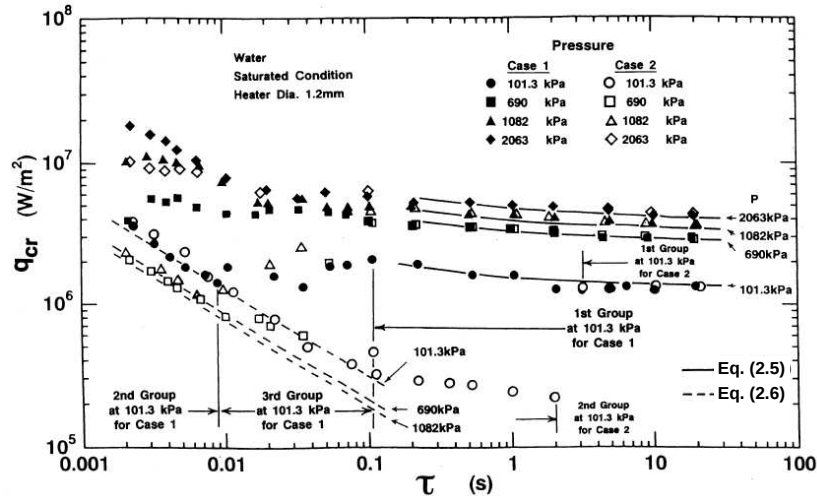


Fig. 2.3 Transient CHF for periods at various pressures for cases 1 and 2.
(Sakurai, 2000a).

The CHF in the short periods belong to the second group were expressed by the linear asymptote line obtained from Eq. (2.6). The asymptote was given by Sakurai and Shiotsu (1977b) as a function of the non-boiling combined heat transfer coefficient, h_{com} , for exponential heat input, the HSN surface superheat as the incipient boiling surface superheat for period, $\Delta T_i(\tau)$, and the liquid subcooling, ΔT_{sub} . h_{com} is a combination of the conduction heat transfer coefficient h_c and the natural convection heat transfer coefficient h_n (Takeuchi et al., 1995). The h_c in Eq. (2.8) is a function of $\mu (= [\rho_l c_{pl} / k_l \tau]^{0.5})$, and K_0 and K_1 , which are the modified Bessel functions of the second kind of zero and first orders.

$$q_{cr} = h_{com} [\Delta T_i(\tau) + \Delta T_{sub}] \quad (2.6)$$

$$h_{com} = [h_c^4 + h_n^4]^{1/4} \quad (2.7)$$

$$h_c = (k_l Q_l c_{pl} / \tau)^{0.5} K_1(\mu d / 2) / K_0(\mu d / 2) \cong (k_l Q_l c_{pl} / \tau)^{0.5} \quad (2.8)$$

$$h_n = (k_l / d) \times Nu \quad (2.9)$$

where Nu is the Nusselt number as the antilog of the general correlation for natural convection heat transfer coefficient obtained from numerical solutions by Takeuchi et al. (1995).

Figure 2.4 shows the q_{CR} for period for the subcoolings over range of 0-80 K for the cases 1 and 2. For subcoolings of 0, 20, and 40 K, the corresponding values of CHF for period were derived from Eq. (2.5). However, for the subcoolings of 60 and 80 K, the rates of q_{CR} are lower than those predicted by Eq. (2.5), thus can be expressed by the following empirical equation suggested by Sakurai et al. (1995).

$$q_{cr} = q_{cr,sub} (1 + 0.023\tau^{-0.7}) \quad (2.10)$$

The direct transition curve representing the value of q_{CR} for short periods for the pre-pressurized case 2 was obtained for each subcooling from Eq. (2.6).

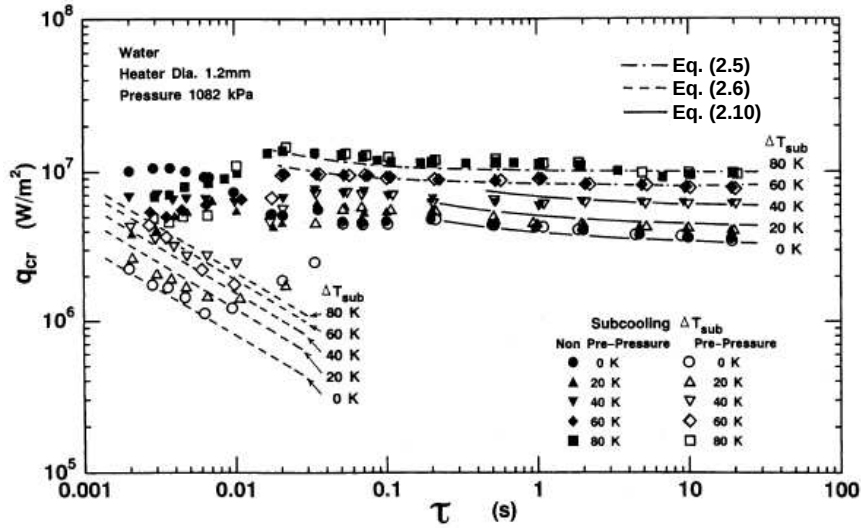


Fig. 2.4 Transient CHF for periods at various subcoolings for cases 1 and 2. (Sakurai, 2000a).

2.2 Two-phase flow boiling

In the classical thermodynamic, it tells that a phase is a macroscopic state of

matter which is homogeneous in chemical composition and physical structure; e.g., a gas, liquid or solid of a pure component (Corradini, 1997). *Two-phase flow* is the simplest case of multiphase flow in which two phases are present for a pure component. A sequence of two-phase and boiling heat transfer regimes takes place along the heated channels during flow boiling, as a result of the increasing quality. In boiling and condensation, the mass quality, x , is often defined as the fraction of the total mass rate of flow consists of vapor, $\dot{m}_g$, and liquid, $\dot{m}_l$, in the liquid-gas flows.

$$x = \frac{\dot{m}_g}{\dot{m}_l + \dot{m}_g} \quad \text{or} \quad x = \frac{h - h_l}{h_{lg}} \quad (\text{at thermodynamic equilibrium}), \quad (2.11)$$

$$(1 - x) = \frac{\dot{m}_l}{\dot{m}_l + \dot{m}_g}$$

An instantaneous area (or volume) average gas fraction may be defined as the area (or volume) of the channel occupied by the gas phase, A_g , divided by the cross sectional area (or total volume) of the channel, A . The area-average gas fraction will be referred to as the *void-fraction*, and will be denoted by a .

$$a = a_g = \frac{A_g}{A}, \quad a_l = 1 - a \quad (2.12)$$

The rate of change of static pressure in the direction of flow will be represented by (dp/dz) . Between the position to another in the flow direction of fluid, the loss of pressure is denoted by p , and positive values for this quantity represent a condition with a rise of pressure with respect to axial distance.

$$\Delta p = -\int_1^2 \left(\frac{dp}{dz} \right) dz \quad (2.13)$$

Analysis of the two-phase boiling heat transfer process and its prediction include complex phenomena in the aspect of phase change and will involve wide variables such as: mass flow rate, mass dryness fraction, liquid density, gas density, liquid viscosity, gas viscosity, surface tension, surface roughness, and pipe inclination. Related to the most topics in the prediction of two-phase flow phenomena, the examinations will focus on the heat transfer rate, critical heat flux,

and the pressure losses. Many literatures in relation to this topic and the correlations developed to explain the phenomena. In this research project, related correlations required for the analysis of two-phase flow boiling through an application on closed loop thermosyphon are listed in Table A1, A2, and A3, for summaries of the correlations of heat transfer coefficient, CHF, and pressure drops.

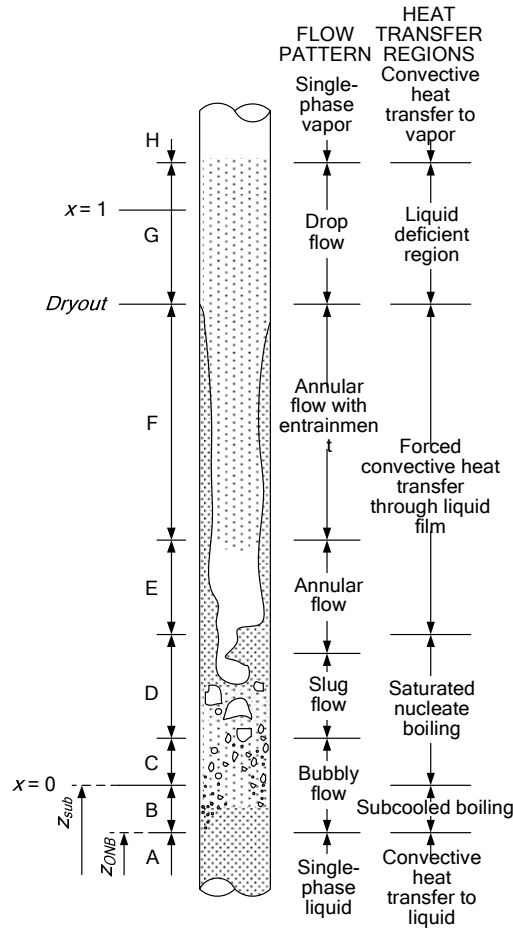


Fig. 2.5 Regimes of heat transfer in convective flow boiling.

2.2.1 Flow boiling heat transfer regimes

Figure 2.5 shows the various flow patterns encountered over the length of the heated tube, together with the corresponding heat transfer region. The vertical

tube is considered also to be heated uniformly over its length with a low heat flux and fed with subcooled liquid (Collier, 1994).

- (a) *Region A*. A single-phase convective heat transfer to the liquid phase at region A, the liquid is being heated up to the saturation temperature while the wall temperature remains below that necessary for nucleation.
- (b) *Region B*. Subcooled nucleate boiling occurs with vapor formation in the presence of subcooled liquid. Degree of superheat, T_{sat} , is considered to exist when the wall temperature exceeds the saturation temperature.
- (c) *Region C*. It is the point at which the liquid reaches the saturation temperature ($x = 0$). A thermodynamic non-equilibrium might exist at the transition between region B and C, where the vapor generated in the subcooled region flows in the centre of the channel.
- (d) *Region C – G*. The heat transfer process depends on the thermodynamic mass quality (x) of the fluid. Refer to Eq. (2.13), the quality of vapor-liquid mixture at a distance, z , is given by

$$x(z) = \frac{h(z) - h_l}{h_{lg}} \quad (2.14)$$

$$\text{or } x(z) = \frac{4q}{DGh_{lg}}(z - z_{sub}) \quad (\text{in terms of heat flux and length})$$

where z_{sub} is the subcooled length required to bring the enthalpy of the liquid up to the saturated liquid enthalpy, h_l . In the region $0 < x < 1$ and for complete thermodynamic equilibrium, x represents the ratio of the vapor mass flow-rate to the total mass flow-rate.

- (e) *Region E – F*. The process of boiling is replaced by the process of evaporation as the quality increase along saturated nucleate boiling region and the vapor-liquid formations are followed by change of flow pattern from bubbly or slug flow to annular flow. The region also refers to as the two-phase forced convective region since the nucleation is completely suppressed and the heat is often transferred to the thin liquid film in the wall and then will make the boiling process no longer exist.

- (f) *Region G – H.* In this transition, the complete evaporation ($x = 1$) occurs induced by critical value of quality is often called ‘*dryout*’, which is accompanied by a rise in the wall temperature.

2.2.2 Two-phase flow Closed Loop Thermosyphon

Advanced closed loop thermosyphon mainly consists of evaporator, condenser, and the connection tubes. The closed loop flow of fluid relies on the gravity assistant and the head difference. Successive studies on advanced closed loop flow have been performed by several authors. However, wide configurations and two-phase flow boiling applications require different analysis due to its flexible arrangement, equipments and loop flow couplings. Closed loop thermosyphon is one of heat pipe types and have been widely applied in industries. The originally invention of thermosyphon system was a boiler system such as Perkins tubes boiler, which was developed in 1827 (Reay and Kew, 2006). However, application of the thermosyphon system as a compact device for advanced electronic cooling system needs to be examined thoroughly to derive the optimum performance. Following reviews are related to literatures reported by authors on the subject of closed loop thermosyphon’s configuration.

Haider et al (2002)

A model for the two-phase flow and heat transfer in the closed loop, two-phase thermosyphon (CLTPT) involving co-current natural circulation was presented. The configuration of the CLTPT is shown in Fig. 2.6. The present model was based on mass, momentum, and energy balances in the evaporator, rising tube, condenser, and the falling tube. The homogeneous two-phase flow model was used to evaluate the friction pressure drop of the two-phase flow imposed by the available gravitational head through the loop. The saturation temperature dictates both the heat source (chip) temperature and the condenser heat rejection capacity. Thermodynamic constraints were applied to model the saturation temperature, which also depends upon the local heat transfer

coefficient and the two-phase flow patterns inside the condenser.

The boiling characteristics of the enhanced structure were used to predict the chip temperature. The model is compared with experimental data for dielectric working fluid PF-5060 and is in general agreement with the observed trends. The degradation of condensation heat transfer coefficient due to diminished vapor convective effects, and the presence of subcooled liquid in the condenser are expected to cause higher thermal resistance at low heat fluxes. The local condensation heat transfer coefficient is a major area of uncertainty. The key of successful model laid in using correlations for small diameter horizontal tubes and validated with fluids similar to PF-5060, such as R-113.

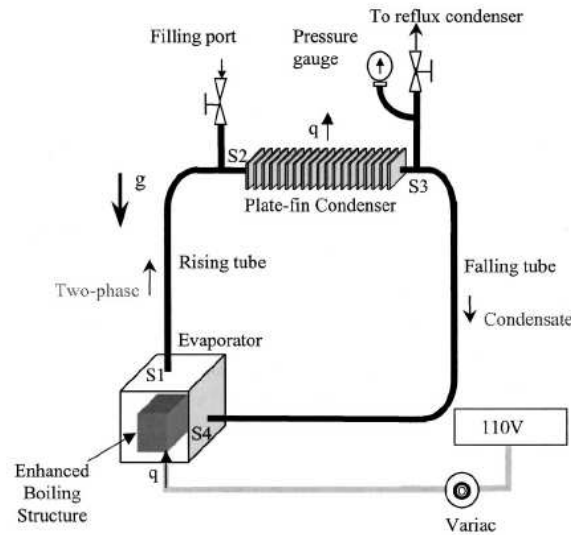


Fig. 2.6 Schematic of the experimental setup of the co-current, closed loop, two-phase thermosyphon (Haider et al., 2002).

Mukherjee and Mudawar (2003).

A pumpless cooling system based on open thermosyphon specifically tailored to compact, high-flux electronic cooling systems. Experiments were performed on a flat vertical boiling surface with water and FC-72 using different boiler gaps varied from 0.051 to 21.46 mm. For large gaps, CHF showed insignificant dependence on the gap for both fluids. However, small gaps produced CHF

variations that were both drastic and which followed opposite trends for the two fluids. A numerical model is constructed to determine how the gap influences the various components of pressure drop, velocities, coolant flow rate, and hence system response to heat input. The velocity of the two-phase mixture exiting the boiling surface, u_L , was given by:

$$u_L = \frac{\dot{m}}{A_c} (v_l + x_L v_{lg}) \quad (2.15)$$

Khodabandeh (2005a, 2005b)

Advanced closed loop thermosyphon for cooling three parallel high heat flux electronic components. The tested evaporators were made from small blocks of copper with 7, 5, 4, 3, 2 vertical channels with the diameter of 1.1, 1.5, 1.9, 2.5, and 3.5 mm, respectively and the length of 14.6 mm. The working fluid is isobutene at heat fluxes ranging between 28.3 and 311.5 kW/m².

For prediction the pressure drop, in the riser, different combinations of frictional pressure drop and void fraction correlations were tested. Homogeneous model has been used to predict the pressure drop.

Hartenstine et al. (2007)

A loop thermosyphon device (~1 meter tall) was fabricated and tested that included several copper porous wick structures in cylindrical evaporators, as seen in Fig. 2.7. It was intended for designing the high heat flux electronic cooling onboard USA naval ships. The loop thermosyphon was considered as the thermal requirements for the next generation naval vessels, which capable of acquiring high heat fluxes (> 1 kW/cm²). The first two were standard annular monoporous and biporous wick designs. The third wick consists of an annular evaporator wick and an integral secondary slab wick for improved liquid transport.

The biporous wick performed better with respect to critical heat flux capability compared to the monoporous design. The biporous wick did not provide heat transfer coefficients as high as the monoporous wick at high heat

fluxes as liquid was not able to wet the wall well enough due to vapor occupying the large voids within the powder. In the pressure drop model, the mass flow rate to loop was defined as:

$$m = \int_0^{L_{tot}} \frac{\pi D^2}{4} [\rho_l(1 - a) + \rho_g a] dz \quad (2.16)$$

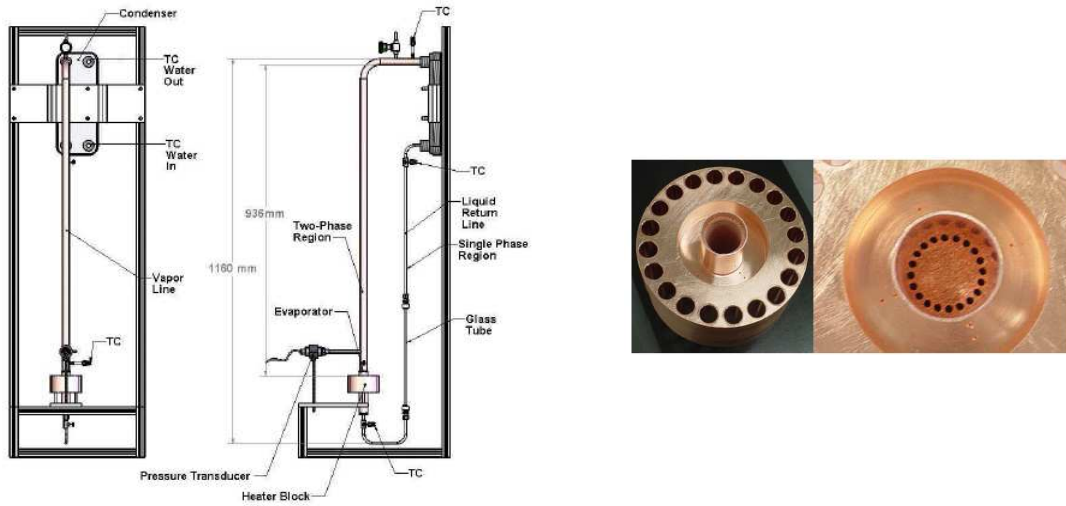


Fig. 2.7 Loop thermosyphon (Hartenstine et al., 2007).

Overview studies of the CLT cooling system were also reported by Na et al. (2001), Pal et al. (2002), Khodabandeh and Palm (2002), Kishimoto and Harada (1994), Stauder and McDonald (1986), and Bar-Cohen and Schewietzer (1985). Next development of CLT technology for high-power electronic cooling device is becoming a large attention because of the high heat transfer capability and flexible configuration for various applications. Investigations on its performance by narrow channels will be interesting phenomena for dissipating high heat flux. In this study, investigations on the experimental transient pool boiling in FC-72 and thermal performance of CLT configuration by a miniature heat sink with 1.0 mm diameter minichannels are intended to contribute correct understanding of boiling heat transfer processes, which will be a basis for the further analysis on pool boiling and two-phase flow boiling heat transfer.

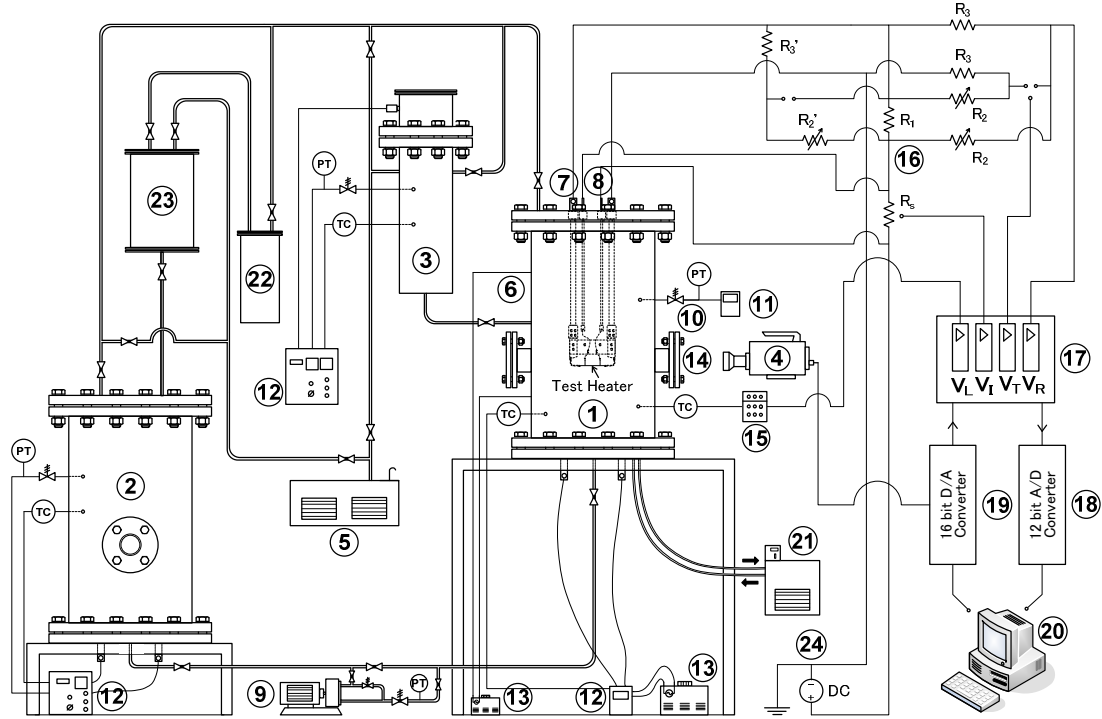
Chapter 3 – Experimental Apparatus and Procedure

3.1 Pool boiling apparatus

The representation drawing of the pool boiling experimental apparatus is shown in Fig. 3.1, which was developed by Sakurai and Shiotsu (1977a). The test chamber or boiling vessel with observation windows is made of stainless steel with 200 mm of inner diameter and 600 mm in height. The vessel is designed for an application of pressure up to 2 MPa. Two current conductors and two potential conductors are mounted at the upper side of the vessel. The liquid temperature in the boiling vessel is heated by a sheathed heater installed at downside of the vessel. The vessel is kept warm by micro heaters and thermally isolated with lagging materials. The liquid temperature in boiling vessel is measured by K-type thermocouples. A pressure transducer to measure the system pressure inside vessel is equipped with a pressure relief valve.

As seen in Fig. 3.1, the test section or test heater is mounted horizontally inside the boiling vessel. In this study, finite horizontal cylinder heaters with a diameter of 1.0 mm and a 0.1 mm thickness vertically oriented horizontal ribbon were used as the test section or heater. The specification parameters of the heaters are listed in Table 1. On the heater, two fine 50 μm diameter of wires were spot-welded on the heater surface as the potential taps. Thus, the effective length of test heater can be determined as the distance between the potential taps.

Heaters were annealed in order to maintain the material at even properties, and the electrical resistance versus temperature relation was calibrated in water and glycerin baths using a precision double bridge circuit. The calibration accuracy was estimated to be within ± 0.5 K. Test section is a “Commercial Surface (CS)” material without any treatments or finishes on surface. However, before they were mounted inside the boiling vessel, necessary cleaning on surface was taken to minimize any possible solid carbons or dirt.



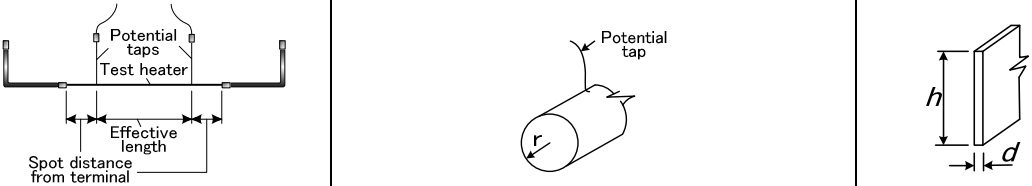
- | | | |
|-----------------------------|-----------------------------|--------------------------|
| (1) Boiling vessel | (9) Booster pump | (17) Heat control system |
| (2) Auxiliary tank | (10) Pressure relief valve | (18) A/D converter |
| (3) Pressurizer | (11) Pressure gauge | (19) D/A converter |
| (4) High-speed video camera | (12) Temperature controller | (20) PC |
| (5) Vacuum pump | (13) Slide rheostat | (21) Tube cooler |
| (6) Micro heater | (14) Observation window | (22) Condenser |
| (7) Current conductor | (15) Cold junction | (23) Receiver tank |
| (8) Potential conductor | (16) Bridge circuit | (24) DC power supply |

Fig. 3.1 Representative of pool boiling experimental apparatus.

3.1.1 Experimental methods and procedure

The experiments of pool boiling heat transfer on horizontal surfaces with transient power input apply several required systems such as power supply and power control system, very low resistance and voltage drop measurements through bridge circuit, data amplification, and data sampling and processing system. Figure 3.2 shows the block diagram of experimental method on heating system and data measurement and processing system.

Table 3.1 Specification parameters of test section

Geometry	Cylinder			Ribbon
Heater No.	AU163	AU162	PT147	PT169
Material	Gold	Gold	Platinum	Platinum
Size	$\phi 1.0$ mm	$\phi 1.0$ mm	$\phi 1.0$ mm	4.0mm $\times$ 0.1mm
Effective length	54.7 mm	30.4 mm	30.9 mm	31.5 mm
Distance from each terminal	15.0 mm	10.0 mm	10.0 mm	15.0 mm
				

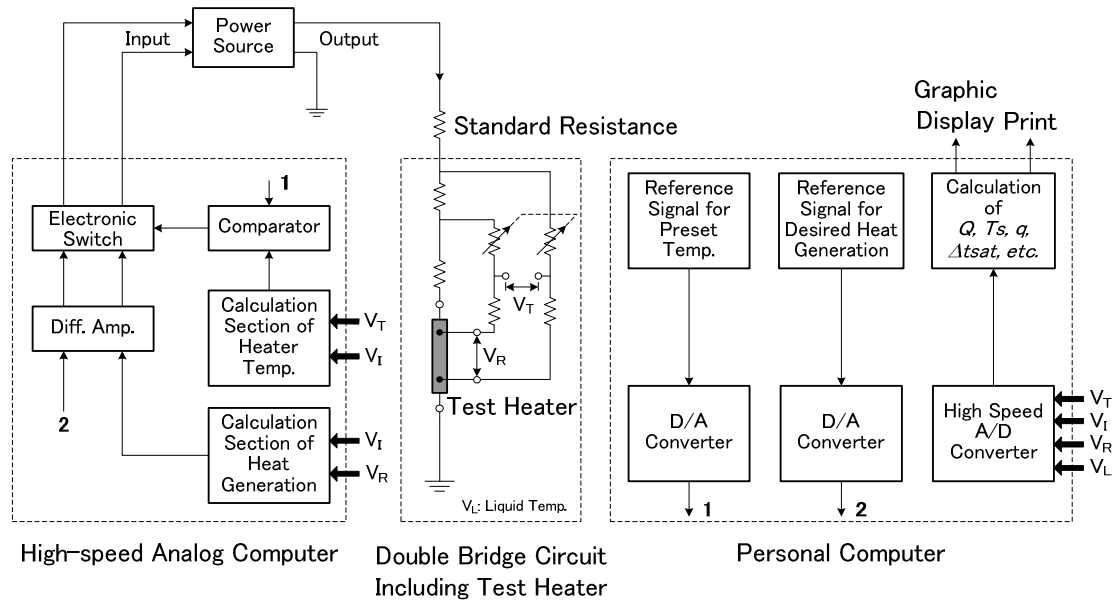


Fig. 3.2 Block diagram of heat input control system and data measurement and processing system.

The test heater was heated electrically by using a fast response, direct current source (max. 700 A) controlled by a digital computer so as to give

increasing heat input with a period. The average temperature of the test heater was measured by resistance thermometry using the heater itself. A double bridge circuit with the heater as a branch was first balanced at the bulk liquid temperature. The output voltages of the bridge circuit, together with the voltage drops across the potential taps of the heater and across a standard resistance, were amplified and passed through analog-to-digital converters of a personal computer. These voltages were simultaneously sampled at a constant time interval. The fastest sampling speed of the A/D converter is 5 μ s/channel. The average temperature was obtained by using the previously calibrated resistance-temperature relation.

The heat generation rate of the test heater was determined from the current to the heater and the voltage difference between potential taps on the test heaters. The surface temperature was obtained by solving the conduction equation in the heater under the conditions of the average temperature and heat generation rate. The instantaneous surface heat flux was obtained from the heat balance equation for a given heat generation rate. The experimental error was estimated to be about ± 1 K in the heater surface temperature and ± 2 % in the heat flux.

$$Q = Q_0 e^{t/\tau} \quad (3.1)$$

The electric current was supplied to the test heater, and the heat generation rate, Q , increased with exponential function. The heat input control system then controlled and measured the heat generation rate of the heater. The average temperature of test heater was measured by resistance thermometry using a double bridge circuit including the test heater as a branch. The heat flux of the heater was calculated by the following equation for heat balance.

$$q = \frac{d}{4} \left(Q - \rho c_p \frac{dT_a}{dt} \right) \quad (3.2)$$

The test heater surface temperature can be calculated from unsteady heat conduction equation of the next expression by assuming the surface temperature around the test heater to be uniform. The equations below are the temperature distribution and the existing boundary conditions we have for the horizontal

cylinder.

$$\frac{\partial T}{\partial t} = a \left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \right) + \frac{Q}{\rho_h c_{ph}} \quad (3.3)$$

Boundary conditions for cylinder are as follows,

$$\left. \frac{\partial T}{\partial r} \right|_{r=0} = 0, \quad -k \left. \frac{\partial T}{\partial r} \right|_{r=D/2} = q \quad (3.4)$$

$$T_a = \frac{\int_0^R T(2\pi r) dr}{\int_0^R (2\pi r) dr} = \frac{2}{R^2} \int_0^R T r dr \quad (3.5)$$

The experiment was carried out as follows. First, a boiling process for degassing of the liquid used in the experiment was performed at least for 30 minutes. Then, the liquid was fully filled in the boiling vessel with the free surface only in the pressurizer and liquid feed tank. Liquid temperatures and pressures in the boiling vessel and in the pressurizer were separately controlled to realize the desired saturated condition. At pressures more than atmospheric one, the experimental conditions were controlled by increasing the liquid temperature, and the pressurization was carried out by the steam itself.

Pre-pressure on heater surface before each experimental running at atmospheric pressure was given to the surface of heater in the saturated liquid pool system for 30 s with quasi-steady-state increasing heat input. The pre-pressure with various hydrostatic pressures up to 1.2 MPa were provided using a booster pump.

Experimental running is executed from high-speed analog computer based on required experimental condition and on each exponential period, τ . A high-speed video camera system (1000 frames/s with a rotary shutter exposure of 1/10000 s) was used to observe the boiling phenomena and to confirm the start of boiling on the test heater surface. The high-speed video camera was started before the starting of a measurement. A video timer started simultaneously with the starting of measurement, and then the boiling phenomenon was recorded with the passage of time.

3.2 Closed loop thermosyphon

Development of CLT was based on the concept of flexible configuration of thermal management with capability of dissipating heat from parallel heat sources as seen in Fig. 3.3, and the requirement of reliable cooling device for high heat flux sources such as high-power electronic devices onboard ship. CLT has been considered as a flexible device to transport heat in long distances, has simple structure, and also reserves as a ‘maintenance free’ device that is due to smart response to the heat load and does not involve external power.

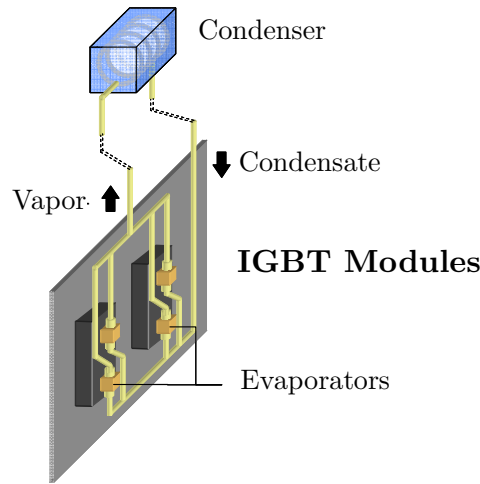


Fig. 3.3 The concept of thermal management with parallel cooling of IGBT module by CLT.

3.2.1 Experimental set-up

Figure 3.4 shows the experimental set-up of CLT for IGBT cooling to examine a cooling device which capitalizes fluid density differences and gravitationally induced buoyancy to drive the pumpless loop flow. The CLT set-up is a closed loop system with has 600 mm in height, 2233 in length, 220 ml total volume, and using water as working fluid. Main loop is a copper pipe with inner diameter of 10.7 mm. The developed CLT set-up and the main components are pictured in Fig. 3.5. The geometry of system loop and its dimension are figured in Fig. 3.6.

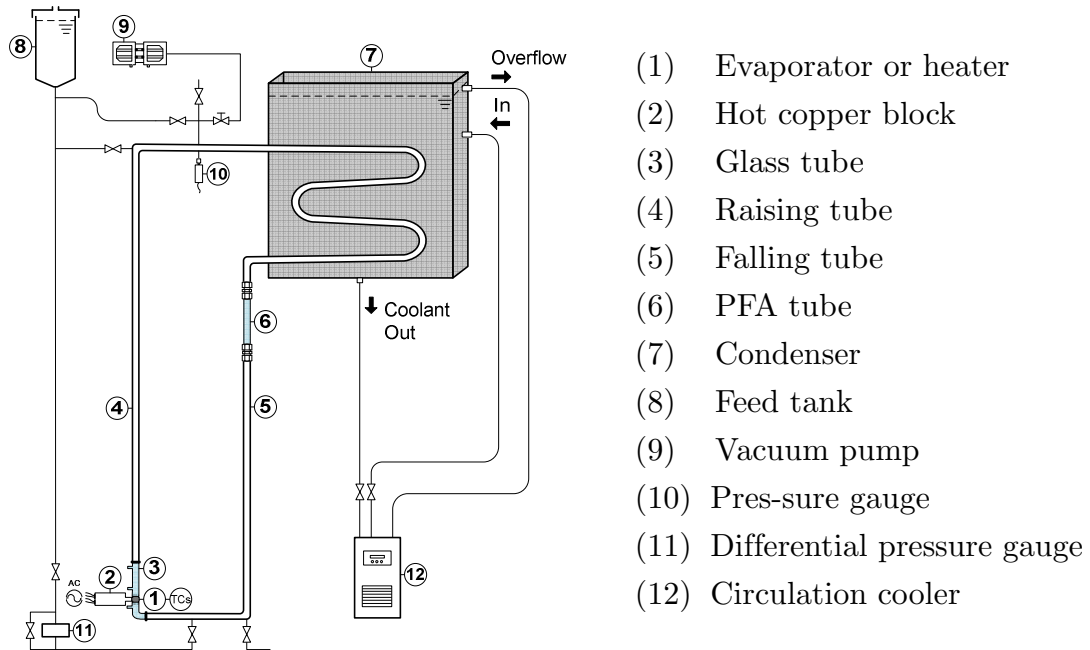


Fig. 3.4 Representation drawing of CLT set-up.

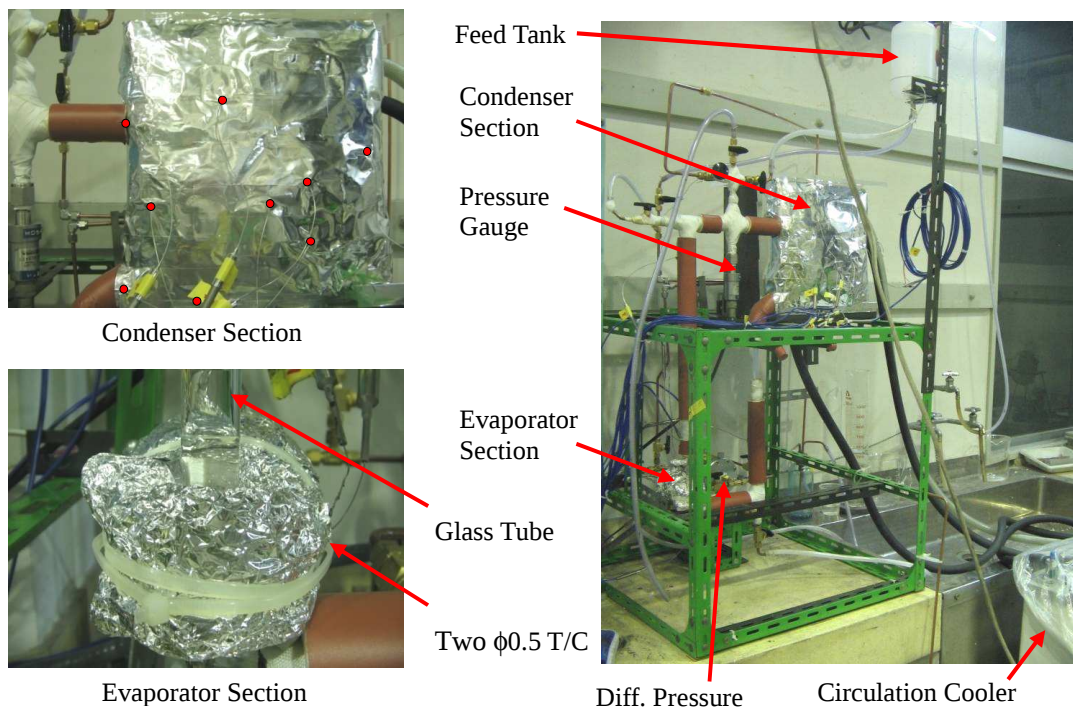


Fig. 3.5 Components of CLT set-up.

The heat losses from heater and condenser sections were minimized by applying heat insulation materials along the outer side of the inserted cartridge heater block and the evaporator section and also covering the flat panel tube on fore tank of condenser. The temperature of system is maintained by heat insulation along the loop except on glass and PFA tubes.



As seen in Fig. 3.6, fourteen thermocouples probe of type K were set including eight T/Cs on eight locations along the loop, three ones on the condenser plate, one on the hot copper block, and two 0.5 mm diameter ones on evaporator. Two signal amplifiers were provided to collect data of pressures. The system pressure and the differential pressure of running system were measured by an electronic pressure transducer (Kyowa, max. 10 kg/cm²) and a differential pressure gauge (Kyowa, max. 200 g/cm²), respectively. All analog data of temperature and amplified voltage signal are read by a data logger (Keyence, NR250, 16 ch., 14 bit) and convert them to digitally data for further required analysis, and were monitored by a data logging software in a PC (Windows).

3.2.1.1 Evaporator section

The evaporator section and its joint section with hot copper block are illustrated in Fig. 3.7. The set-up of evaporator section included main evaporator or heater block, a copper block heater with inserted four cartridge heaters or the simulated component for IGBT cooling, and glass tubes. The arrangement of this section was based on the requirement of simulating the IGBT module cooling to accommodate high heat fluxes. CLT with advanced heater and condenser structures are expected to acquire high heat transfer rate in an extent limit operation temperature. The picture of evaporator with staggered array minichannels is shown also in Fig. 3.7.

The evaporator or heater was made of copper block (15 mm × 15 mm) and then cut, shaped, and milled to make the the core and shoulder body of evaporator. At the center of core body, nineteen 1.01 mm diameter vertical circular channels were drilled to make 20 mm length hole. The channels were composed in staggered array facing the simulated component or die heating area. They were divided into five layers consisted of 3, 4, 5, 4, and 3 channels at the first, second, third, fourth, and fifth layers, respectively. At the mid of shoulder body, two holes-one at 1.5 mm from die area with horizontal 7.5 mm depth parallels with the array layer, and one on the opposite end tip of body with free convective cooling, were drilled for 0.5 mm thermocouples type K to measure temperature of heater surface. At the end treatment, heater block was cleaned

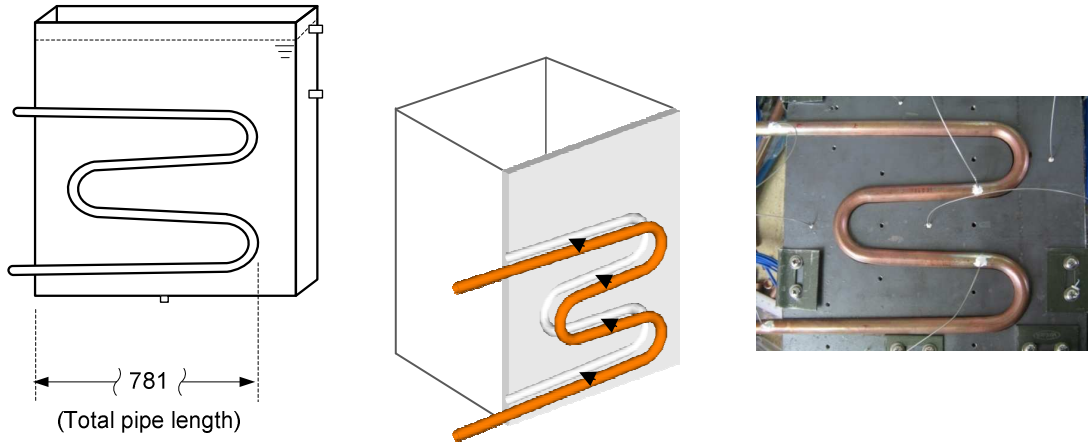


Fig. 3.8 Direct hull cooling method based flat tube panel water-cooled condenser.

3.2.1.2 Condenser section

Condenser is a simulated direct hull cooling method. It serves as a water-cooled condenser made of 12 mm thickness of steel plate with 300 mm × 250 mm surface area on the waterside and has total volume of tank is 2370 cm³. Flat tube panel from CLT's main loop was embedded on fore tank surface in 6 mm depth and 781 mm in length and has 156 cm² heat transfer area along its groove. Condenser was designed with the required length capable to accommodate the high heat fluxes two-phase boiling heat transfer through evaporation process based on power to dissipate of 1 kW at low pressure assuming laminar condensation and uniform surface temperature with an inner pipe diameter of 10.7 mm. The developed condenser with embedded tube panel is illustrated in Fig. 3.8.

3.2.2 Experimental methods and procedure

Measurements were taken based on several variable system parameters such as pressure, saturation temperature, heat load, fill charge rate, and condenser's coolant temperature. Experimentations are based on the controlled heat flux system to measure and collect data.

3.2.2.1 Power input system and measurement

Heating of the evaporator is performed by four cartridge heaters (Rama Corp., 32A31, 120 V, 400 W) that were inserted into the copper block. The electric power to each cartridge heater is equally supplied by four regulated volt sliders (Yamabishi, 100 V/0-130V, Max. 5.0 A). Wattage of cartridge heaters was derated as required. Each electric line for cartridge heaters is provided with electric resistance wire, R , to measure the potential, V_R , and in order to calculate the electric current, I . The R values are: $R_1=15.1725 \text{ m}$, $R_2=15.2878 \text{ m}$, $R_3=15.3138 \text{ m}$, $R_4=14.6250 \text{ m}$.

All voltages from adjusted rate of volt slider and through resistance wire were measured by a digital multi meter (Advantest, R6452) within an error range of $\pm 0.03\%$. Total dissipated heat load, Q , is derived by calculating currents and measured voltages, V , from volt sliders. Independent and a series of measurements were carried out in response to the variable heat load as a function of voltage distribution given by quadruple 10 V increment from 30 V to 80 V.

Heat inputs are considered to be dissipated by the heat sink of the test section. However, the heat losses, Q_{loss} , through insulations, and natural convection to ambient along hot copper block gave a range of losses 10% - 7.5% of the total dissipated heat. Heat losses at around evaporator section were considered negligible. Thus, the net heat dissipation to the evaporator, Q_{net} , was derived from reduction of total dissipated heat by its losses.

$$Q = V \frac{V_R}{R} \quad (3.6)$$

$$Q_{net} = Q - Q_{loss} \quad (3.7)$$

$$q_{eff} = \frac{Q_{net}}{A_{die}} \quad (3.8)$$

One of the attributed heat fluxes in the study of heat transfer to the heat sink is the uniformly effective heat flux, q_{eff} , in Eq. (3.8), which is defined as the total of net dissipated heat to the evaporator, Q_{net} , divided by the die area, A_{die} , or the contact surface area between the evaporator and the hot simulated component

or hot copper block. However, at the extent distances in every row of array from die, the wall heat fluxes on each of wetted perimeter of channels are being non-uniform ones. The wall heat fluxes also are extremely influenced by the flow boiling process. Therefore, the second parameter of heat flux required for analysis is the mean heat flux, $\bar{q}_c$, as the averaged value of convective heat flux over the heated channel that is due to convective flow boiling heat transfer process.

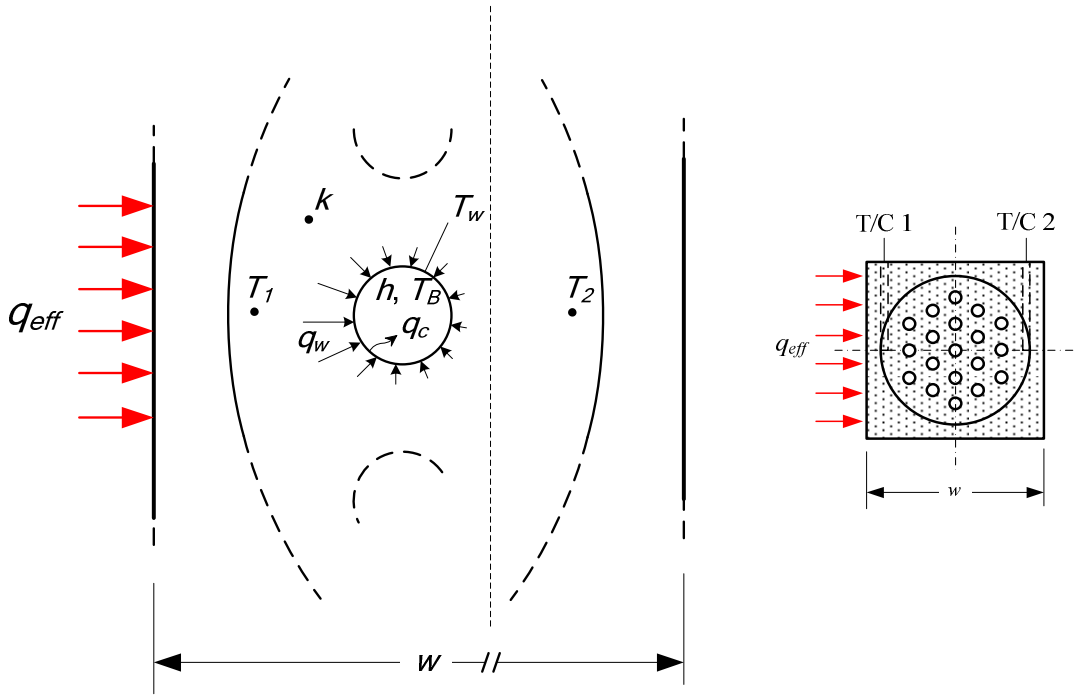


Fig. 3.9 Heat transfer profile on the unit channels of evaporator

Based on the evaporator geometry, the averaged convective heat flux, q_c , at saturated liquid flow along the length, z , of the heated channels equal to the mean wall heat flux, q_w , and can be calculated as the approximate experimental heat flux measured between two thermocouples as a function of heat balance in the one-dimensional heat transfer conduction process assuming the absence of heat loss, as seen in Fig. 3.9.

$$q_c = q_w = \frac{q_{eff} A_{die}}{nL\pi D} \quad (3.9)$$

where T_1 and T_2 are the measured temperatures by T/C 1 and 2 respectively. The local two-phase heat transfer coefficient in the saturated boiling regions is given by:

$$h_{tp} = \frac{2q_w}{T_1 + T_2 - 2T_{sat}(z)} \quad (3.10)$$

3.2.2.2 Experimental procedure

The experimental procedure was carried as follows. First, pure water was degassed by a short time boiling before filling it to feed water tank. Main loop of CLT was filled up at a certain volume, and then system loop pressure was vacuumed down to about 12.35 kPa to reach an extent saturation water temperature of 50 °C at cold condition. Coolant inside circulation cooler was setting up on 15 °C to simulate the direct hull cooling water temperature and then pumped and circulated to fully water-filled condenser tank to make steady temperature condition.

Electric heat input starts to supply to the evaporator / heater by adjusting the voltage for CLT under a saturated pressure and steady temperature of condenser water coolant conditions. The steady-state data measurements were considered to be taken when system attained a stable temperature performance of heaters after about 10-15 minutes from start of each series of applied heat input. Steadily measured data for each series of rated heat input were registered by data logger for 30 minutes length period and in another case the measurements were performed for 120 minutes experimental running at each item condition. Monitoring of measurements of data and sampling graphics are viewed through PC.

3.2.2.3 Uncertainties

Overall thermal performance of CLT is determined by temperature reading

of thermocouples probed in several locations. K-type thermocouples with diameter of 1.0 mm were used except two 0.5 mm ones that were probed in the evaporator test section at 1.5 mm from hot die simulated module and at the opposite end tip of the shoulder body of evaporator.. Temperatures have error reading ± 1 °C. The measured analog system pressure and also differential pressure rates during operation were calibrated, and sent to be amplified first, and then read by data logger, which have an experimental error range of $\pm 0.3\%$. The experimental set-up has also another possible uncertainty during high-heat loads due to lack of silicon seal (high-heat resistance type, ShinEtsu, KE-3418) on sealing of between glass tubes and evaporator section. It has possible limit of continuous use at high temperature up to 250 °C.

Chapter 4 – Transient Pool Boiling in FC-72

4.1 Experimental conditions

In this study, the experimental data including CHF were obtained due to the exponentially increasing heat inputs, $Q_0 e^{t/\tau}$, as function of e-fold time or the exponential periods that were ranged from 10 ms up to 50 s. In the further analysis, a range of periods above around 1 s will correspond to the quasi-steady-state periods, and the periods above 10 s correspond to the steady-state periods. Thus, the periods below around 1 s will correspond to the short periods.

The system parameters for the measurements also include a wide range of pressures and subcoolings. The pressure parameter was ranged from around atmospheric pressure up to about 1.3 MPa, and the liquid subcooling parameter was ranged from saturated condition up to 140 K. For the pre-pressurization to the heater, a range of hydrostatic pressures was ranged up to 1.2 MPa. All requirements of the experimental condition are summarized in Table 4.1.

Table 4.1 Experimental conditions

Liquid	FC-72 (C_6F_{14})
Pressure	79.5 kPa ~ 1278.1 kPa
Pre-pressure	~ 1.2 MPa
Liquid subcooling	0 ~ 150.0 K
Exponential period	10.0 ms ~ 50.0 s

In this study, the ‘fluorinert’ or ‘dielectric fluorocarbon’ FC-72 was used as the stagnant coolant for the pool boiling experiments. Compared to water and

other wetting liquids such as ethanol, or liquid nitrogen, FC-72 has low boiling point and high dielectric strength that are being physically advantaged properties required by the operation of electronic liquid cooling system. However, FC-72 is known for its poor thermal properties than water and may give a boiling hysteresis for certain configurations. Table 4.2 shows the physical properties of FC-72 at 25°C from 3M Company (3M, 2000) and also water for comparison.

Table 4.2 Physical properties of FC-72 and water

Properties	FC-72	Water
Boiling point (1 atm) [°C]	56.6	100.0
Liquid specific heat [kJ/kg.°C]	1.1	4.2
Latent heat of vaporization [J/g]	88.0	2257.0
Liquid thermal conductivity [W/m.°C]	0.057	0.61
Surface tension [$\times 10^{-3}$ N/m]	10.0	72.0
Critical temperature [°C]	176.0	374.0
Critical Pressure [MPa]	1.83	22.0

4.2 Pool boiling heat transfer processes in FC-72

4.2.1 Time dependence of pool boiling characteristic

In Fig. 4.1, it was plotted the graph for the time dependence of q , Q , and T_s , measured for period of 10 s with a transition to film boiling or the direct transition. Measurements were taken through a treatment to the heater by giving a hydrostatic pre-pressure of 1.2 MPa. Figure 4.1 is a representation of boiling process with direct transition due to quasi-steadily increasing heat inputs plotted in Fig. 4.3 for case 2. Compared to the previous studies on the pool boiling heat transfer process through nucleate boiling (Sutopo, 2008), it was observed that the direct transition process also exist not only from transient heat conduction but

also from the increasing quasi-steadily natural convection. In Fig. 4.1, even the surface temperature speed during the incipient boiling is almost same with one in case 1 without pre-pressure, a simultaneously process of CHF after boiling incipience was found as a new mechanism of CHF by inducing of the direct transition process without any occurrence of nucleate boiling from active cavities. It was assumed due to the explosive-like heterogeneous spontaneous nucleation (HSN). After reached the CHF, heat flux decreases down while the surface temperature continues to increase. The heat flux continues to increase until the heat flux reach the steady-state film boiling (see the reference of boiling curve in Figs. 4.2 and 4.3).

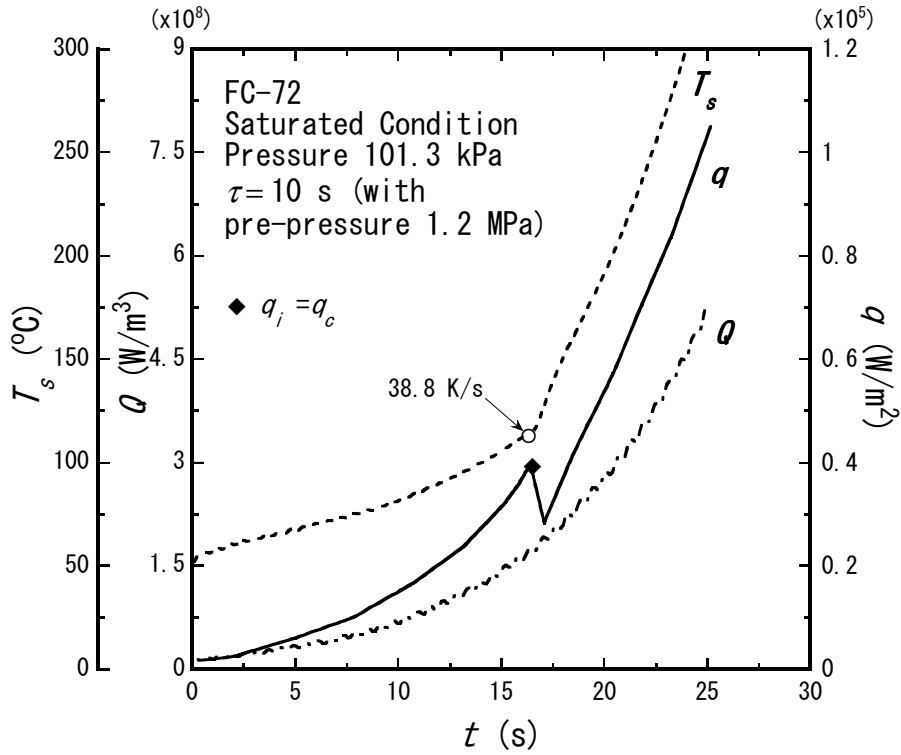


Fig. 4.1 Time dependence profile of q , Q , and T_s for direct transition (with pre-pressure of 1.2 MPa)

4.2.2 Transition from non-boiling to film boiling

The typical heat transfer processes for horizontal finite heater in a pool of

FC-72 due to transient heat inputs, $Q_0 e^{t/\tau}$, are shown in Fig. 4.2 for the log-log graphs of heat flux, q , versus surface superheat, ΔT_{sat} . Generally, the boiling heat transfer processes from non-boiling regime to film boiling due to transient heat inputs at pressures can be classified into three types of transition, i.e.: the transition from non-boiling to fully developed nucleate boiling (FDNB), the direct transition to film boiling, and the transition from non-boiling to film boiling through nucleate boiling or semi-direct transition.

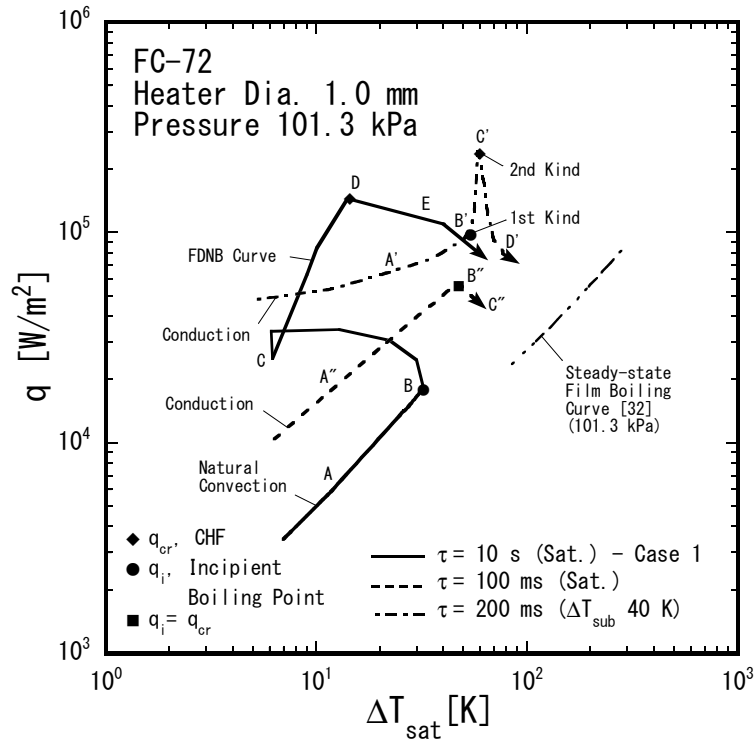


Fig. 4.2 Pool boiling heat transfer processes in FC-72

Typical of boiling heat transfer processes on direct transition at CHF are shown on curves in Fig. 4.3, which represent characteristics of boiling heat flux, q , versus surface superheat, ΔT_{sat} . The curves for periods, τ , of 10 s and 100 ms that were measured at atmospheric pressure correspond to the quasi-steadily increasing heat input and rapidly increasing one, respectively. The applied pre-pressure of 1.2 MPa on measurement at $\tau=10$ s is the hydrostatic pressure

added to the cylinder surface for a while prior to each experimental running to enable flooded cavities and to dissipate the activation of cavities for nucleate boiling by elimination of the possibility of entrapped vapor inside cavities.

It was observed that the direct transition process in the pool boiling of FC-72 predominantly occurs from the transient conduction process by rapid increasing heat input. In Fig. 4.3, a black dash line curve representing the direct transition at CHF for $\tau=100$ ms appears from increasing non-boiling conduction heat transfer up to the boiling inception and then followed by CHF simultaneously at surface heating rate of 3720 K/s, which results an equal value of incipient boiling and critical heat fluxes ($q_{CR}=q_i$) at 6.0×10^4 W/m². The typical CHF and incipient boiling heat flux that were measured for the exponential period by rapid increasing heat input can be seen also in Figs. 7 and 8 for the graphs of q_{CR} and q_i versus the periods, τ .

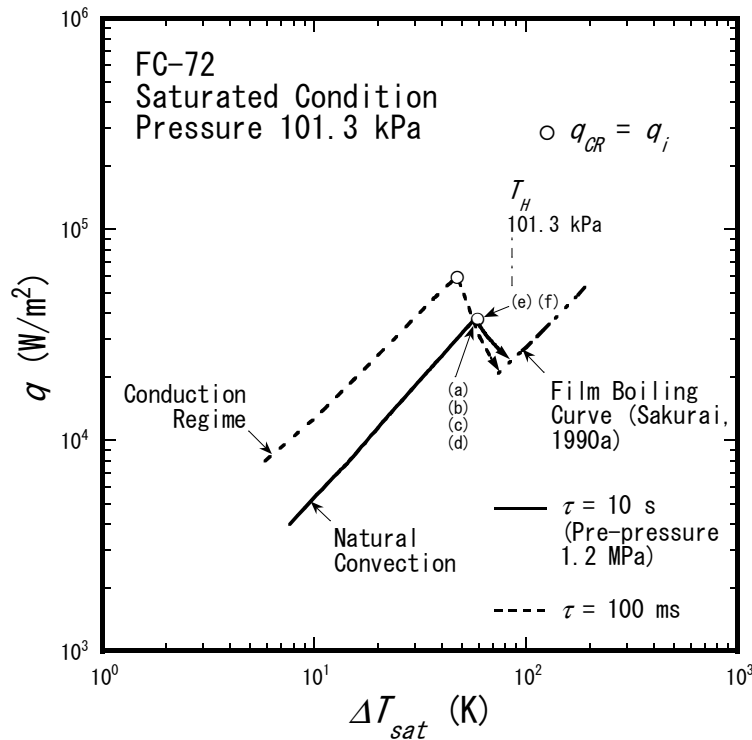


Fig. 4.3 Direct transition processes at periods of 100 ms and 10 s

However, authors observed that the direct transition processes exist not only from conduction process but also from single phase natural convection. A same typical direct transition process has been confirmed to exist from quasi-steadily increasing natural convection on solid surface in the pool of FC-72 by measurements at around atmospheric and sub-atmospheric pressures, and by applied hydrostatic pressure or pre-pressure to the heater surface. In Fig. 4.3, it is represented by a black solid line curve for $\tau=10$ s with a given pre-pressure of 1.2 MPa. The curve has surface heating rate at the direct transition point at CHF, q_{CR} , is 38.8 K/s and the value of q_{CR} and q_i is 3.7×10^4 W/m². In the figure, it is seen also that the transition surface superheat of direct transitions in a pool of saturated FC-72 at atmospheric pressure is clearly lower than the corresponding homogeneous nucleation temperature, T_H , at 87 K, which was derived from Eq. (7) for predicting the homogeneous nucleation temperature near its spinodal limit as suggested by Lienhard (1982). The typical direct transition process at CHF is also represented by photographic observations shown in Fig. 4.4 that were taken at points (a) to (f) on the boiling curve for $\tau=10$ s (with a pre-pressure of 1.2 MPa) in Fig. 4.3.

4.2.3 Photographic observations on vapor film behavior at direct transition

The corresponding phenomena of direct transition process that are shown in Figs. 4.4(a) to 4.4(f) start from a non-boiling regime of quasi-steadily increasing natural convection and proceed up to the incipient boiling point that is seen in the first photograph at point 4.4(a). A vapor film about 20 mm in length on the solid liquid contact appears at surface temperature of 387.74 K. Figures 4.4(a) to 4.4(d) show the vapor film behavior with a rapid growth of film covering the cylinder surface. Vapor film fully blankets the cylinder surface in only 3 ms from boiling incipience at point (a). It was assumed due to an explosive-like spontaneous nucleation (HSN) in originally flooded cavities on the cylinder surface and then forms the vapor tube concentrically covering the cylinder. Figure 4.4(e)

was taken at 30 ms from point (a) just prior to the CHF and shows the detachment of bubbles from top of vapor tube which moves upward by buoyancy force from the position shown in the former photograph.

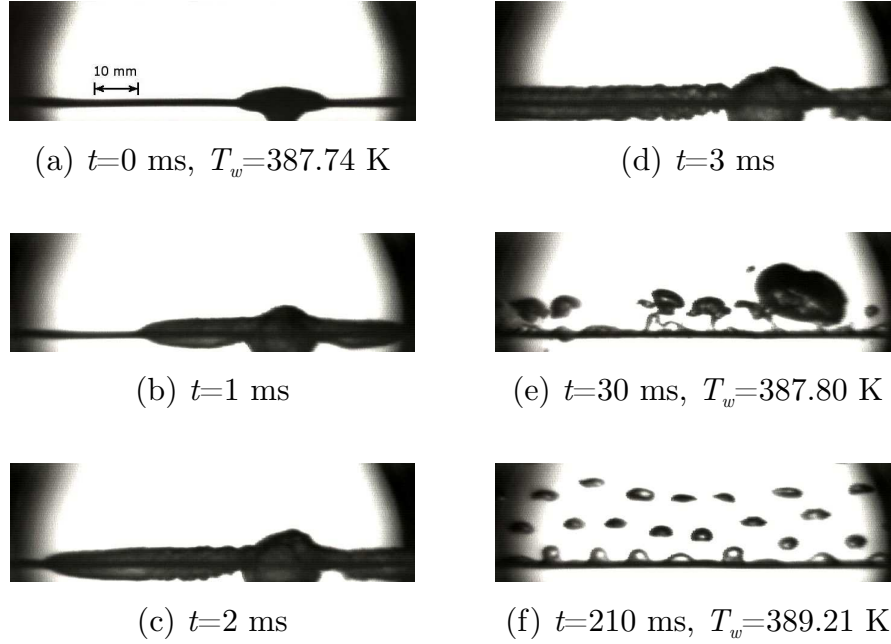


Fig. 4.4 Vapor film behaviors during direct transition to film boiling at 101.3 kPa under saturated condition for period of 10 s with a pre-pressure of 1.2 MPa

Direct transition occurs with a wavy pattern on vapor film surface. This process becomes significant and diminishes heat flux as the surface superheat increases. It was observed also that direct transition exists without boiling hysteresis or nucleate boiling occurrences as seen also on the boiling curve in Fig. 4.3. Transition boiling heat flux steeply decreases and almost reaches theoretical minimum film boiling at its contact interface temperature, T_f , which was suggested by Sakurai et al.(1990b). A thin, wavy vapor film continues to form while heat transfer process approaches the steady-state film boiling curve, which was derived from film boiling prediction by Sakurai et al. (1990a). The facts of direct transitions at incipient boiling surface superheat confirm the assumption of the incipient boiling that is due to the explosive-like HSN in originally flooded

cavities on cylinder surface at HSN surface superheat. Figure 4.4(f) at 210 ms from point (a) shows existing Taylor instability wave after detachment of bubbles and then continues to form new film boiling on the solid-liquid interface.

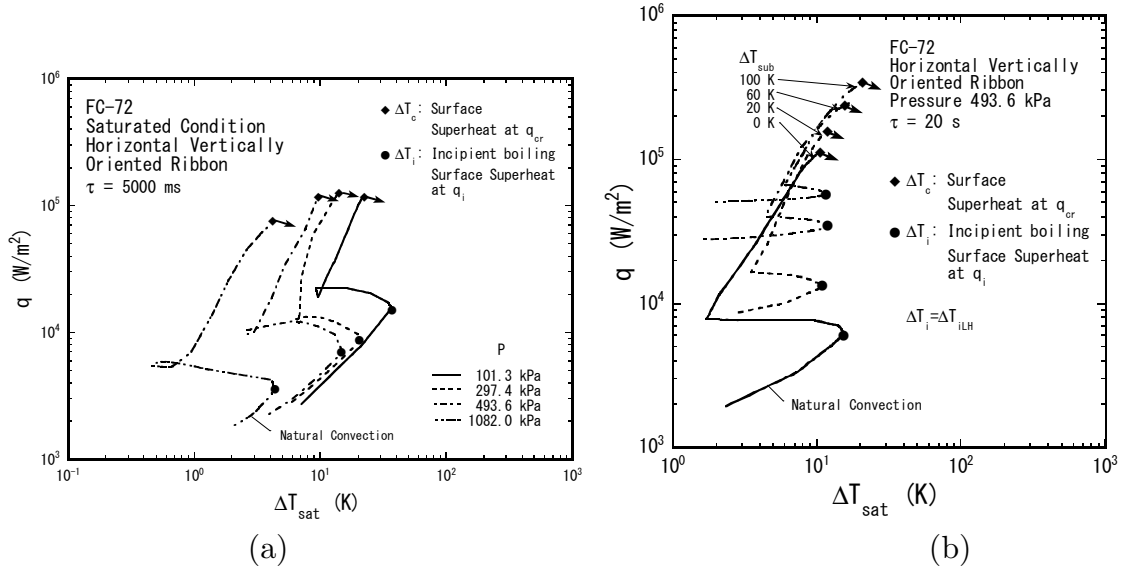


Fig. 4.5 Lower limit HSN Surface Superheat (ΔT_{iLH}): for a wide range of pressures (a) and liquid subcoolings (b)

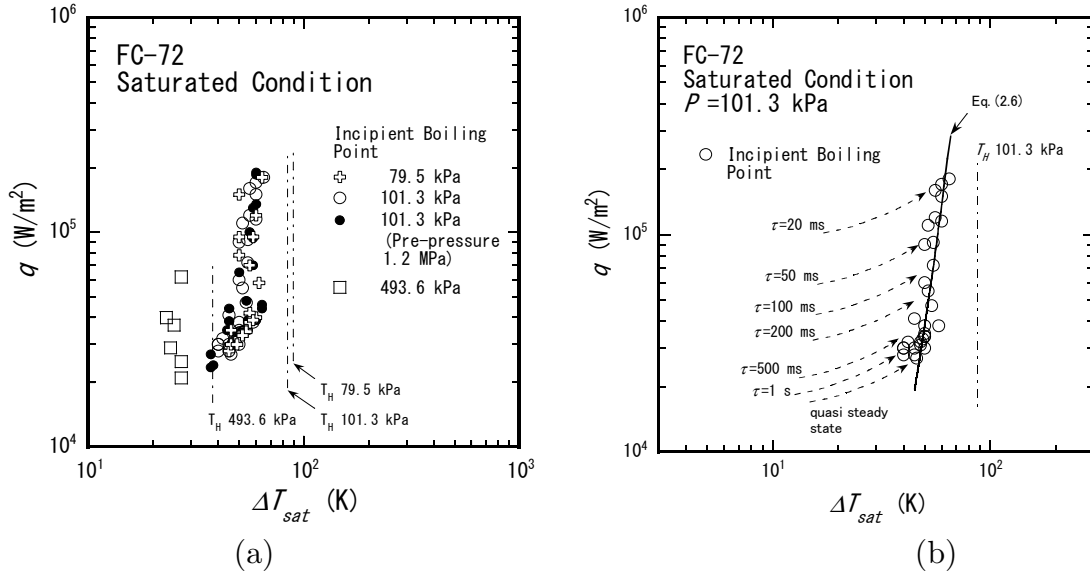


Fig. 4.6 Incipient boiling points for direct transition process

4.3 Incipient boiling and CHF surface superheats

4.3.1 Lower limit HSN surface superheat

Figure 4.5 (a) and (b) shows the type 1 of typical of boiling heat transfer processes up to CHF point from FDNB regime through the incipient boiling point due to the transient heat input measured on the horizontal vertically oriented ribbon under a wide range of subcoolings ranged from 0 to 100 K at the pressure of 493.6 kPa. It is seen that at saturated condition, the ΔT_c is lower than ΔT_i . However, at temperatures above saturated condition, which is seen on graph that it starts from subcooling 20 K, the ΔT_c agree with the ΔT_i , and by increasing in liquid subcoolings, ΔT_c are becoming higher than ΔT_i . In this phenomenon, the ΔT_i corresponds as the lower limit of HSN superheat (ΔT_{iLH}) for inducing the CHF. Thus, it was suggested that the CHF at the pool boiling of FC-72 at liquid subcooling condition above saturation are caused by the HSN process of incipient boiling. The theoretical of the lower limit HSN surface superheat has been widely discussed in Sakurai et al. (1993) and Sakurai (2000).

In Fig. 4.6(a), the incipient boiling points that are due to HSN for inducing direct transitions are shown on the graph of heat flux, q , versus surface superheat, ΔT_{sat} . They were measured for periods ranging from 10 ms to 20 s at a wide range of pressures. The incipient boiling surface superheat, ΔT_i , from both transient conduction and quasi-steadily increasing natural convection processes in non-boiling regimes for direct transition at CHF, q_{CR} , are lower than the values of homogeneous nucleation temperature, T_H .

At pressures of 79.5 kPa and 101.3 kPa, the direct transitions at q_{CR} were derived from both transient conduction and natural convection processes by the rapid increasing heat inputs in short periods ($\tau \leq 500$ ms) and the quasi-steady-state increasing ones in long periods ($500 \text{ ms} < \tau < 20 \text{ s}$), respectively. At heat fluxes below $4.5 \times 10^4 \text{ W/m}^2$, the q_{CR} increase almost linearly with the increase in ΔT_i at each pressure. Those predominantly belong to direct transitions from natural convection by quasi-steady-state increasing heat inputs and a combine of transient conduction and natural convection, in a range of

periods of $\tau > 500$ ms. However, at heat fluxes higher than around 4.5×10^4 W/m², which exist from transient conduction and by rapid increasing heat inputs for periods of $\tau \leq 500$ ms, the q_{CR} increase at an almost constant value of ΔT_i .

In Fig. 4.6(b), it is seen the incipient boiling points for direct transition process in saturated pool of FC-72 at atmospheric pressure. They were observed to exist at a range of periods, $10 \text{ ms} < \tau < 5 \text{ s}$, by rapid increasing heat inputs and quasi-steady-state increasing ones. The incipient boiling points seem to have less dependence on the increasing rate of heating or decreasing rate of periods, τ , however, by shorter periods, the incipient boiling superheats are tending upward approaching the homogeneous nucleation temperature. The maximum heat fluxes or CHF's for direct transition boiling processes at the incipient boiling points increase with the decrease in periods and agree with the corresponding curve obtained from the correlation that was suggested by Sakurai and Shiotsu (1977a).

Figure 4.7 shows the relation of surface superheats, ΔT_i and ΔT_c for the exponential period, τ , which were measured on a 1.0 mm diameter horizontal gold cylinder at atmospheric pressure. Related to the thermal requirement in the typical microprocessor applications, the allowable maximum chip temperature is approximately at 373 K, i.e.: around 44 K of surface superheat temperature for saturated FC-72 at atmospheric pressure. In the measurement on gold cylinder surface in a pool of FC-72, it is seen that the ΔT_i under saturated or subcooled conditions require an almost constant rate at around periods above 1 s due to the quasi-steady-state increasing heat inputs (region I and II). However, in the region III below 1 s that of due to rapid increasing heat inputs at direct and semi-direct transitions, ΔT_i increase with the decrease in periods. It was found also that several ΔT_i on gold surface at saturated condition for quasi-steady-state period above 1 s, in the region I, require higher values than the allowable rate of chip cooling. However, at subcooled conditions of 20 K and 40 K in the region II, the ΔT_i show lower rates than the allowable rate.

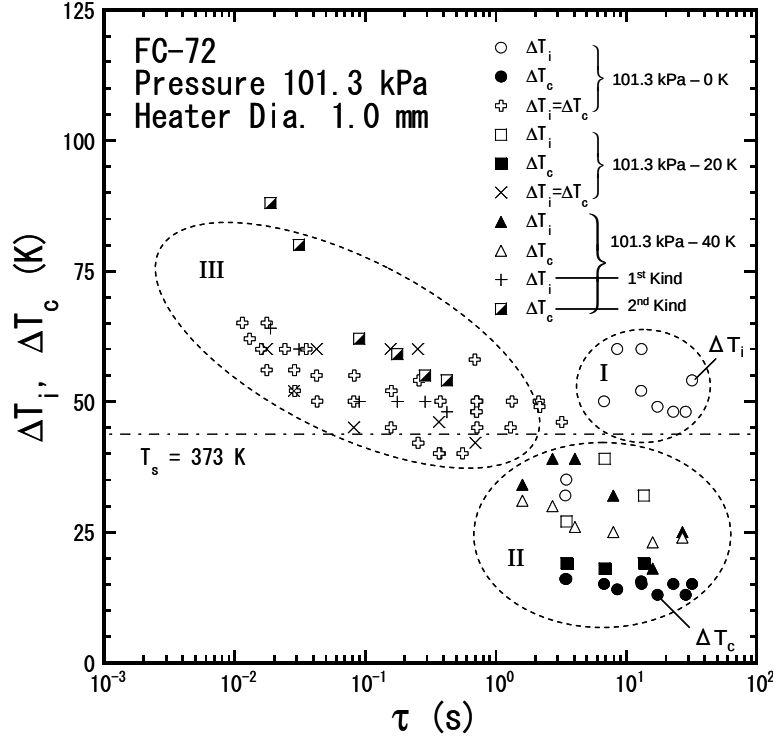


Fig. 4.7 Surface superheats for exponential periods on a horizontal cylinder

4.3.2 Effect of pre-pressure on the incipient boiling surface superheat

In the current work on horizontal gold wire, the incipient boiling surface superheats, ΔT_i , were measured by quasi-steady-state increasing heat inputs with periods, τ , of 10 s and by adding the pre-pressures up to 1.2 MPa in saturated pool of FC-72. As seen in Fig. 4.8, the incipient boiling superheats lie on a limit range of temperatures from 30 K to 60 K and seem to have a less dependence on the pre-pressure.

However, in higher pre-pressures, authors observed the spread of the potential incipient boiling superheats for dissipating nucleate boiling and inducing direct transition process. Figure 4.8 shows a constant boundary line at $\Delta T_i = 48$ K, which represents limit of incipient boiling surface superheat for inducing direct transition process. It divides the graph into region 1 and region 2 of ΔT_i for representing a different heat transfer process. The below region is the region 1 for ΔT_i with white circle marks that gives potential nucleate boiling process

from the cavities or nucleation sites on cylinder surface heater after the boiling incipience during transition to FDNB process. The above one is the region 2 for T_i with black solid circle marks, which are followed simultaneously by CHF surface superheat for direct transition processes to film boiling. The ΔT_i for inducing FDNB processes appear up to pre-pressure of 1.0 MPa. However, the given pre-pressures of 1.2 MPa result the potential ΔT_i for fully direct transition processes with no FDNB processes. Another investigation on boiling inception in pool of FC-72 on horizontal platinum heater has been reported also by Mizukami et al. (2005) and observed a mechanism of incipient boiling process that was due to spontaneous nucleation and the incipient superheat was generally independent of the pre-pressure given to the solid surface.

The pre-pressure confirms also that quasi-steady-state heat transfer could induce a direct transition to film boiling at CHF. It gives an equal value of heat flux between the CHF, q_{CR} , and the incipient boiling heat flux, q_i . These characteristics can be seen in Fig. 4.9 for graphs of q_{CR} and q_i versus exponential periods, τ , that were measured at atmospheric pressure.

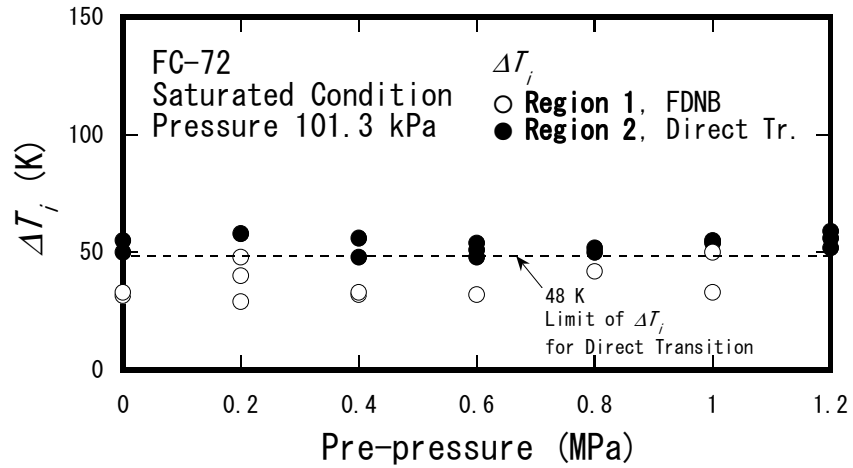


Fig. 4.8. Incipient boiling surface superheat vs. pre-pressure

4.4 Transient CHF

4.4.1 Transient CHF for a wide range of liquid subcoolings and pressures

Figures 4.9 shows the log-log graphs of q_{cr} and q_i versus to show the phenomena of the transient CHF and the incipient heat fluxes due to exponentially increasing heat inputs in subcooled conditions of 60 K with the comparisons of data and corresponding curves measured from the 1.0 mm diameter horizontal cylinder and the horizontal vertically oriented ribbon. At high subcooling of 60 K at pressure of 1082.0 kPa, the CHF from a horizontal vertically oriented ribbon also give lower values than ones from horizontal cylinder. However, as seen in Fig. 4.9, the shortest period of the first group belong to horizontal vertically oriented ribbon is shorter than one at the first group belong to the horizontal cylinder. The quasi-steady-state CHF and the transient CHF at the second group from horizontal ribbon give average values at about 40 % and 30 % respectively lower than ones from the horizontal cylinder. At low subcoolings, the quasi-steady-state CHF were expressed by the correlation in Eq. (2.5), however, at high subcoolings, the CHF were expressed by the correlation in Eq. (2.10) due to enabling CHF from the nucleate boiling process at shorter periods. The incipient heat fluxes, q_i , from the boiling process due to quasi-steadily increasing heat inputs at low and high subcoolings show an almost constant rate for periods for each pressure at a range of rates of 80-120 % lower than CHF.

The effect of liquid subcoolings is also clearly in the graph of q_{cr} versus τ in Fig. 4.10. The CHF in all typical groups of CHF for periods increase with the increase of liquid subcooling. However, the third groups between the first and second groups widen with the increase of liquid subcooling. Thus, it was found that the CHF at high subcoolings of 60 K and 100 K for the direct transition process are scarce in the second groups. The quasi-steady-state CHF at low subcoolings up to 20 K, were expressed by the correlation in Eq. (2.5), however, the CHF at high subcoolings, the quasi-steady-state CHF were expressed by the

corresponding curves obtained from Eq. (2.10). The CHF's in the second groups at short periods were expressed by the asymptote curves obtained from Eq. (2.7).

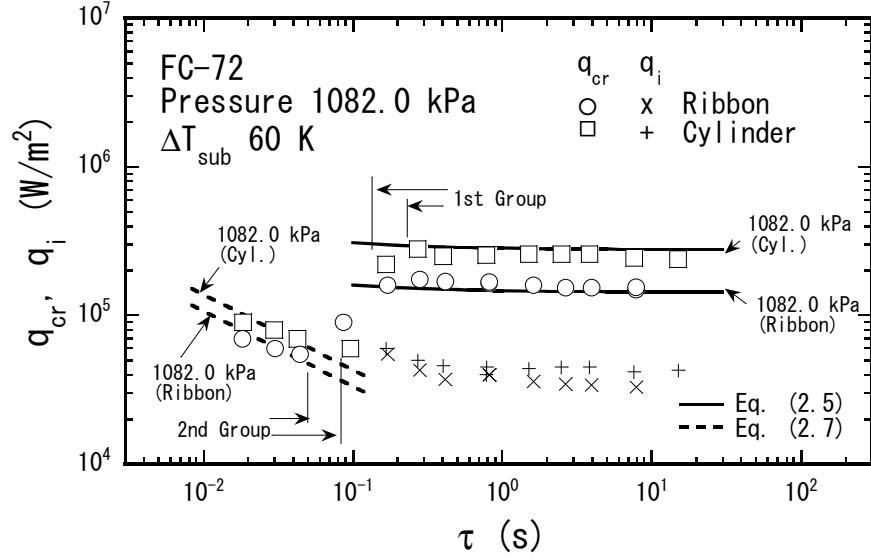


Fig. 4.9. The q_{cr} and q_i at T_{sub} 60 K and Pressure of 1082.0 kPa from horizontal cylinder and vertically oriented ribbon heaters

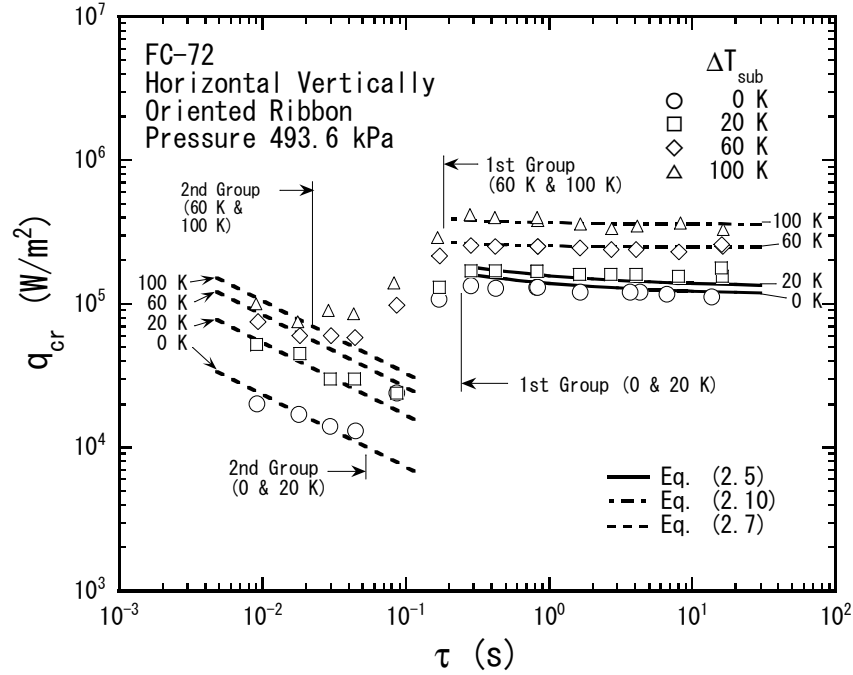


Fig. 4.10. Transient CHF's for subcooled conditions at pressure of 493.6 kPa

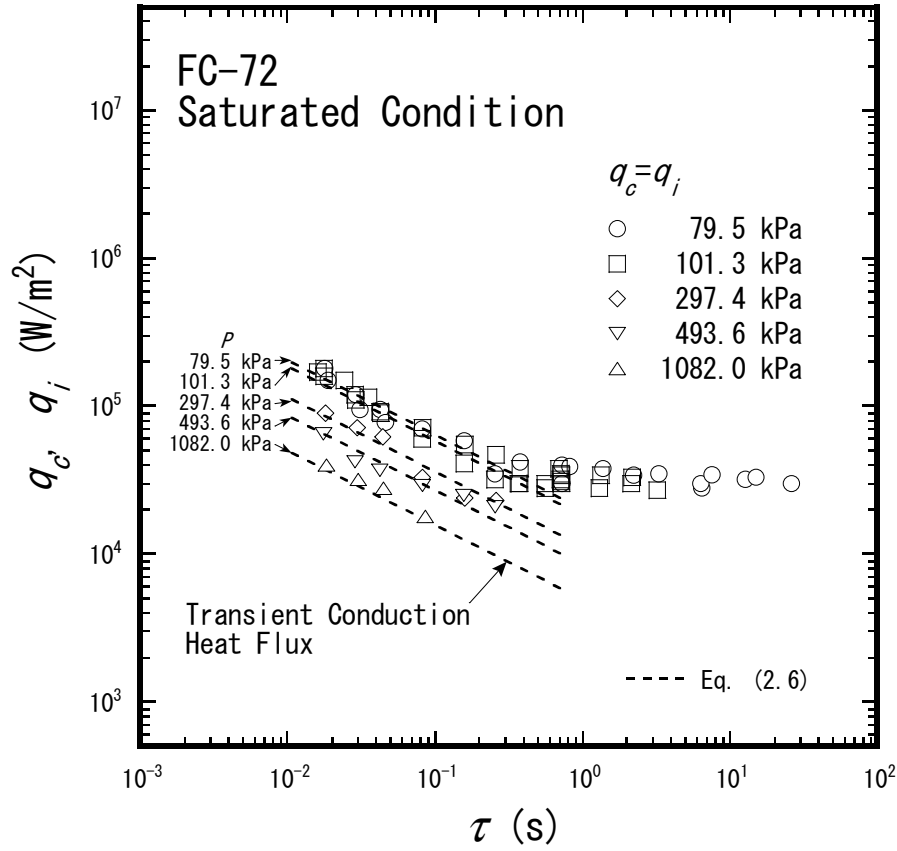


Fig. 4.11. CHF and incipient boiling heat flux for direct transition process

The critical heat fluxes for direct transition process by transient heat inputs from quasi-steadily increasing heat inputs up to rapidly ones, were plotted on graphs of q_{CR} or q_c and q_i for the exponential period, τ , in Figs. 4.11 and 4.12. They were measured under saturated conditions at a wide range of pressures from 79.5 kPa up to 1082.0 kPa and the exponential periods ranging from 10 ms up to 20 s.

In Fig. 4.11, the CHF, q_{CR} , at pressures of 79.5 kPa, 101.3 kPa, 297.4 kPa, 493.6 kPa, and 1082 kPa were plotted for periods, τ . On the direct transition, the CHF occurs simultaneously after the incipient boiling, thus it has equal value with the incipient boiling heat flux, q_i , which were plotted in same marks with the CHF at every pressure. Ones observed the direct transitions at CHF at pressure of 79.5 kPa were obtained not only from non-boiling transient conduction but also

single phase natural convection. They appear in whole range of short periods of $\tau \leq 1000$ ms and even in quasi-steadily and steady-state ones at $\tau > 1000$ ms and $\tau \geq 20$ s, respectively. The q_{CR} in quasi-steady-state periods up to steady-state show almost a constant value. On the other hand, the q_{CR} in short periods increase with the decrease in periods. Measurements at atmospheric pressure also result direct transitions from quasi-steady-state increasing heat input up to periods of $\tau < 5$ s.

At higher pressure than atmospheric, the direct transitions appear predominantly in shorter periods of $\tau < 300$ ms from transient conduction by rapid increasing heat inputs. Within short periods, the q_{CR} increase with the decrease in periods, and the period of the minimum CHF at each pressure shortens with the increase in pressure. The possible of nucleate boiling processes in the short periods or by rapid increasing heat inputs were observed to exist at higher pressures. It was assumed that boiling process at high pressure near its homogeneous temperature makes it possible for inducing an insufficient HSN process and then initiates nucleate boiling without the direct transition boiling process to film boiling.

In Fig. 4.11, the CHF for direct transition boiling process at short periods of $\tau < 1000$ ms are dependent on pressure. This phenomenon can be expressed by the corresponding value of CHF for periods in linear asymptote lines that were plotted in Fig. 4.11 by black dash lines. The corresponding lines were obtained from a correlation in terms of non-boiling heat transfer coefficient from transient conduction process that was suggested by Sakurai and Shiotsu (1977a) in Eq. (2.6). The conduction heat transfer coefficients, h_c , by exponentially rapid increasing heat inputs were observed to increase rapidly with the decrease in τ and approach a certain asymptotic value.

Figure 4.12 shows the effect of pre-pressure on the CHF for direct transition process to film boiling. It is shown on graph of q_{CR} and q_i for the period, τ , on measurements of case 2 with pre-pressure of 1.2 MPa at atmospheric pressure under saturated condition. The case 1 of q_{CR} for direct transitions without

pre-pressure is also plotted as a comparison. In case 1 without pre-pressure, the boiling heat transfer at quasi-steady-state periods, $\tau > 3$ s, gives CHF at the transition from fully development nucleate boiling (FDNB) to film boiling that was assumed due to the hydrodynamic instability (HI). It gives an almost constant of q_{CR} at average value of 1.5×10^5 W/m², which are plotted on graph with hollow square marks. From the observations of boiling phenomena by photograph and high speed video, it was observed that the FDNB process is induced by the activation of cavities entraining vapor after detachment of bubbles from top of vapor film covering cylinder surface. It was observed to occur during the incipient boiling process, which is followed by boiling hysteresis with rapid change of heat flux and surface superheat just prior to detachment of bubbles. The incipient boiling heat fluxes, q_i , belong to case 1 were plotted on graph by white diamond marks.

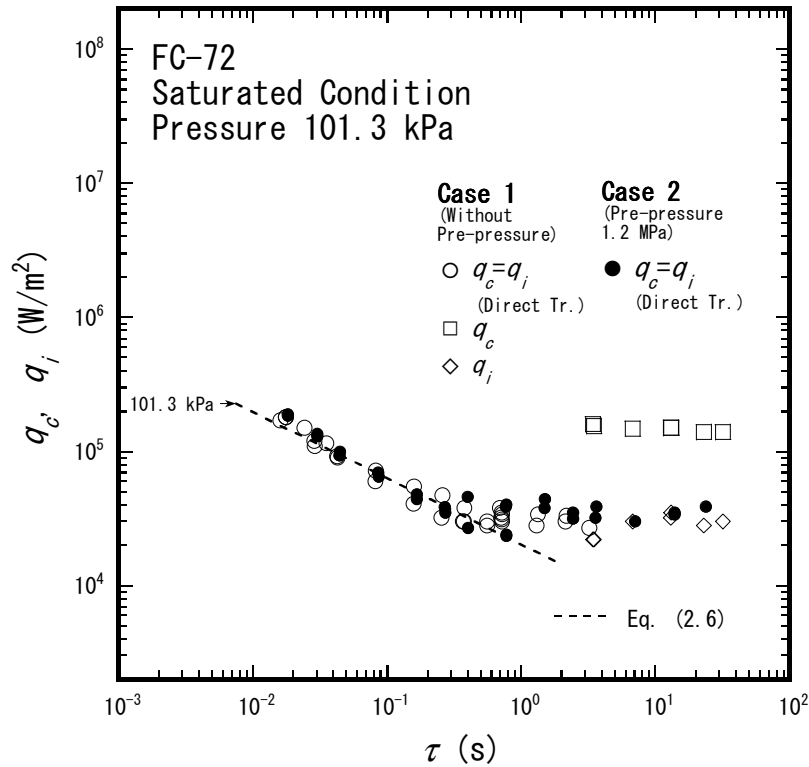


Fig. 4.12 Effect of pre-pressure on CHF and incipient boiling heat flux for direct transition process

As seen in Fig. 4.12, the measurements with pre-pressure 1.2 MPa in case 2 give the direct transitions at CHF from not only transient conduction in short periods by rapid increasing heat input but also from natural convection in long periods by quasi-steadily increasing heat inputs, and result an equal value of q_{CR} and q_i from simultaneously process of incipient boiling and CHF. The q_{CR} and q_i were plotted with same solid black circle marks. The CHFs, q_{CR} , from transient conduction heat flux in short periods of $\tau \leq 1000$ ms increase with the decrease in periods. However, during quasi-steady-state periods up to steady-state ones at $\tau > 1000$ ms, the q_{CR} approach a constant value at around 3.5×10^4 W/m². Compared to the CHF in case 1 for quasi-steady-state periods, the effect of pre-pressure on CHF in the case 2 shows a quite different phenomenon.

The values of CHF belong to direct transitions in case 2 by quasi-steady-state increasing heat inputs are about 77% lower than ones in the case 1 belong to CHF from FDNB process. However, on both cases 1 and 2, the direct transitions from transient conduction by rapid increasing heat inputs within short periods show a same typical increasing rate of CHF that increase with the decrease in periods, and can be expressed by the corresponding linear asymptote line.

4.4.2 Predictions of incipient boiling surface superheat and CHF for direct transition boiling process

A bunch of predictions of CHF for boiling heat transfer has been reported in literatures. However the preceding prediction of CHF should be credited to Kutateladze (1959) and Zuber (1959) for their hydrodynamic instability analysis. In the transient boiling heat transfer with a wide range of exponential periods under various pressures and liquid subcoolings, it was suggested that the CHF could be explained by the HSN model at its lower limit superheat instead of the hydrodynamic instability (HI) model (Sakurai, 2000)(Sakurai et al., 1992)(Fukuda et al., 2004). In the current work, direct transition process by exponentially rapid increasing heat inputs at short periods of $\tau < 1000$ ms, the CHF and its mechanism for transition boiling from non-boiling to film boiling were confirmed due to the

heterogeneous spontaneous nucleation with an explosive-like boiling inception and then followed by CHF simultaneously to proceed the transition boiling. This kind of transition boiling phenomenon shows a significant correlation of the boiling inception and CHF mechanisms.

The prediction of CHF by exponentially increasing heat input has been suggested by Sakurai and Shiotsu (1977a) for the correlation of the heat flux at the incipient boiling superheat with predominant heat transfer coefficient from transient conduction process. The heat flux is maximum heat flux or CHF, q_{CR} , and is equal with the incipient heat flux, q_i , for direct transition process. This correlation could explain the phenomena of direct transition process by rapid increasing heat input at short period region and shows well agreement with the experimental data as seen in Figs. 4.6(b), 4.11, and 4.12.

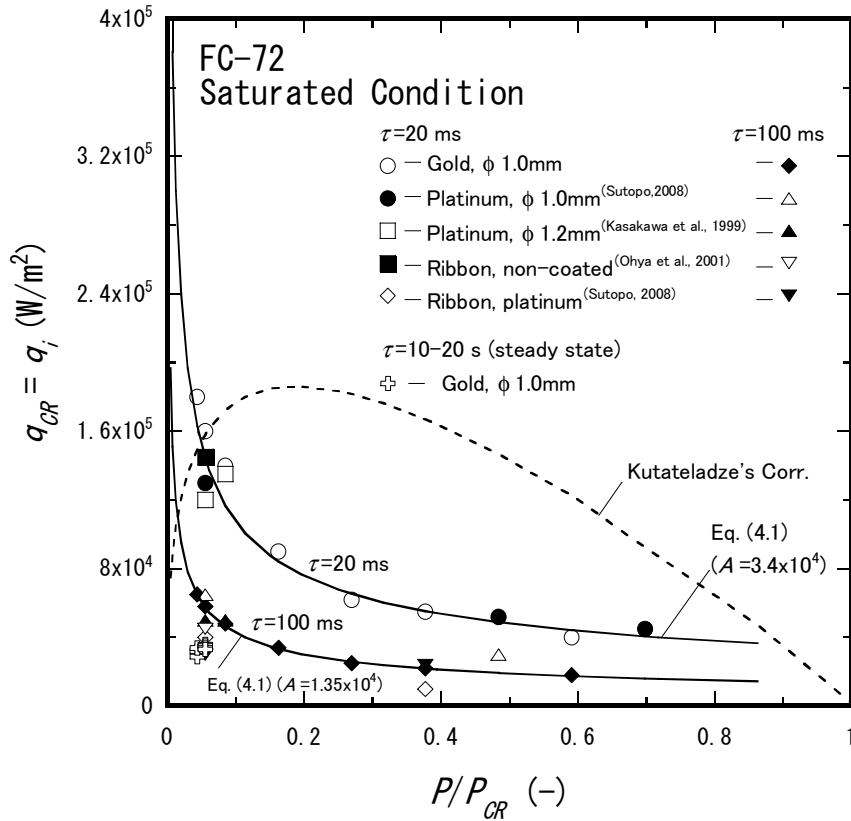


Fig. 4.13 q_{CR} vs reduced pressure

In Fig. 4.13, the current data of CHF, q_{CR} , and incipient boiling heat flux, q_i , from direct transition boiling process on 1.0 mm horizontal gold cylinder at the representation of short periods of 20 ms and 100 ms by rapid increasing heat inputs and at quasi-steady-state periods of 10 s and 20 s were plotted for the reduced pressure, P/P_{CR} . Other data from measurements by transient heating on different horizontal geometry and material of the heater were also plotted. It shows that the increasing rates of q_{CR} and q_i at short periods increase non-linearly with the decrease in reduced pressure. The heat flux also significantly depends on the rate of the exponential period, τ , at each pressure, which is also shown by the typical heat flux for period in Figs. 4.6(b), 4.11, and 4.12. However, direct transition processes from natural convection at steady-state periods of 10 s and 20 s for pressures of 79.5 and 101.3 kPa seem to give q_{CR} with a constant of increasing rate.

$$q_{CR}(\tau) = A (P/P_{CR})^{-0.5} \quad (4.1)$$

$$\Delta T_i(\tau) = B (P/P_{CR})^{-0.5} \quad (4.2)$$

In order to express the typical increasing rate of CHF for direct transition at short periods by rapid transient heat transfer, a corresponding curve in black solid lines derived from Eq. (4.1) was plotted in graph based on the experimental data. The empirical correlation in Eq. (4.1) is proposed to predict the corresponding CHF and incipient heat flux, q_{CR} and q_i , for the direct transition process by rapid increasing heat inputs at short periods of τ . It is a function of the non-dimensional parameter of the reduced pressure, P/P_{CR} , to the power of -0.5 to express non-linearly the increasing rate of CHF, and a constant, A , which was obtained from least square fit to experimental data. The corresponding curve shows a good agreement with the experimental data at each of short periods for various reduced pressures, with constants A are equal to 3.4×10^4 and 1.35×10^4 for periods of 20 ms and 100 ms, respectively. The corresponding curve for steady state CHF by HI model was also plotted for comparison from Kutateladze's correlation.

One of important mechanisms in the boiling heat transfer process and should

be responsible for occurring spontaneous nucleation is the incipient boiling process. The boiling inception model at certain required superheat limit is important to be quantified for its prediction and accuracy of evaluation on boiling heat transfer system. The predictions of the incipient boiling superheat have been suggested to include the homogeneous nucleation temperature, T_H , close to the thermodynamic spinodal limit, which was suggested by Lienhard (1982) in Eq. (2.2). Moreover, in the prediction of incipient boiling superheat for engineering approximation, it mostly takes account the heterogeneous nucleation superheat. In the pool boiling of highly wetting liquid, the prediction of heterogeneous nucleation superheat often involves the phenomena of boiling hysteresis before the onset of nucleate boiling, which considers the effect of surface tension and the prediction of effective radius of vapor bubble on surface including the Rohsenow's estimation for fully developed nucleate boiling, as seen in Eq. (2.3) for the prediction of the incipient boiling hysteresis ΔT_{IBH} (Bar-Cohen and Simon, 1988). In the heterogeneous spontaneous nucleation, prediction of incipient boiling superheat as the lower limit HSN superheat may also refer to the contact interface temperature of cylinder surface, T_i , which closely agrees with the minimum film boiling temperature, as suggested in Eq. (2.4) by Sakurai et al (1990b).

However, in the boiling heat transfer phenomena by rapid exponentially increasing heat inputs with short periods of $\tau < 1000$ ms for a wide range of pressures, the mechanism of incipient boiling that is assumed due to spontaneous nucleation contributes significantly the initiation process of boiling for inducing the direct transition to film boiling. In the pool of saturated FC-72, the incipient superheats, ΔT_i , have been measured for a wide range of system pressures up to 1082 kPa and were plotted in graph for the reduced pressure, P/P_{CR} , as seen in Fig. 4.14. The current data for 1.0 mm diameter horizontal gold cylinder by quasi-steady-state increasing heat inputs at periods 10 s and 20 s, and by rapid increasing ones at short periods of 20 ms and 100 ms were plotted in the figure with bold plus, white circle, and black solid diamond marks, respectively. The ΔT_i for short periods increases non-linearly with the decrease in reduced pressures and

approaching the homogeneous nucleation temperature. These data were accompanied also for comparison by other measured data with transient heating from different horizontal geometries and materials of heaters. However, the incipient surface superheats seem to have less dependence on the exponential periods, τ , at each pressure.

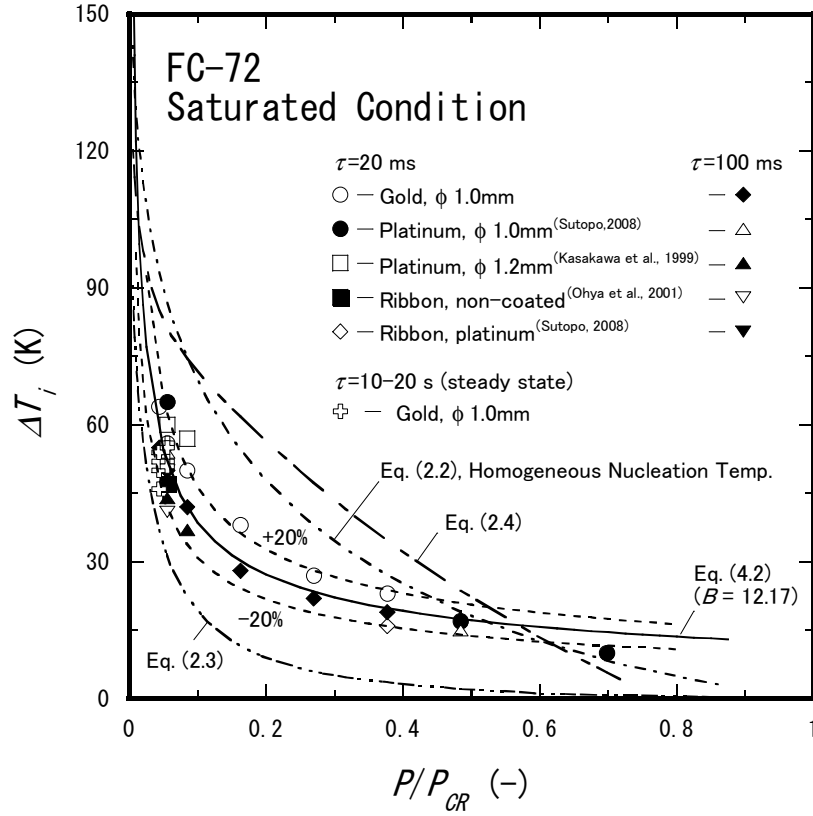


Fig. 4.14. ΔT_i vs reduced pressure

This typical incipient boiling superheat for direct transition process by rapid increasing heat input at short periods can be also predicted by a corresponding non-linear curve on solid line plotted in the figure. It was obtained from the empirical correlation in Eq. (4.2) as a function of non-dimensional parameter of the reduced pressure, P/P_{CR} , and a constant, B , which is the corresponding value of incipient boiling superheat and was obtained from least square fit to the experimental data. As seen in the figure, the corresponding curve almost agrees

with the measured data, which are lying within an error range of $\pm 20\%$, with $B=12.17$. The other predictions of incipient boiling superheat were also plotted for comparison, which were calculated from Eqs. (2.2-2.4) and based on the corresponding saturation temperature. The incipient boiling superheats for a wide range of system pressures are mostly lower than the homogeneous nucleation temperature. In Fig. 4.11, the curve of ΔT_{IBH} from Eq. (2.3) was obtained without taking into account the third component in the equation for Rohsenow's estimation of FDNB in the boiling hysteresis and then remains the first and second components, which reserve as the incipient boiling superheat in the heterogeneous nucleation process. In this calculation, the radius of cavity or the effective radius of vapor bubble embryo on surface, r , was using a fixed value of $0.1 \mu\text{m}$.

Chapter 5 – Performance of Closed Loop Thermosyphon

Examination and analysis of the performance of two-phase closed loop thermosyphon (CLT) include the assessment of the collected data and calculations based on the required parameters i.e: experimental thermal performances and thermal resistances.

5.1 Experimental conditions

The developed CLT device for IGBT cooling with high power dissipation from hot simulated component was examined based on several variable system parameters such as pressure, saturation temperature, heat load, fill charge rate, and condenser's coolant temperature. The experimental conditions were summarized in Table 5.1.

Table 5.1 Experimental conditions

Working fluid	Pure water
System pressure	18.0 kPa ~ 36.0 kPa
Fill charge rate	60.0 ml, 100.0 ml
Condenser coolant temperature	15.0 °C
Heat load	88.0 W ~ 670.0 W

In this research, water was chose as the working fluid due to its high thermal properties and has figure of merit (FOM) higher than FC-72. FOM for CHF under flow boiling in saturated liquid can be defined as (Yeh and Chu, 2002):

$$FOM = h_{lg} \varrho_g^{0.239} \varrho_l^{0.396} \sigma^{0.365} \quad (5.1)$$

5.2 Thermal performance of CLT

The thermal performances of the CLT device were measured mostly based on

temperature distributions. The performance is the response to the corresponding heat input to the evaporator and the occurrences of pressure losses and the difference of the buoyancy head between two kinds of phase running in the loop. The examinations include the measurements of pressures, temperature distribution of the closed loop system and the requirement of the heat transfer coefficient.

In Fig. 5.1, it is seen that the CLT loop reach the stable running with balance of pressure to drive the flow. The fluctuations of temperature profile were suggested due to the effect of pressure losses along evaporator channel and the riser. It is shown also that system could reach the steady-state at about 10-15 minute after employing heat input. This phenomenon was observed also in the graph of temperature profile of the hot simulated component and the evaporator in each fill charge rate and pressure, as seen in Figs. 5.2 and 5.3.

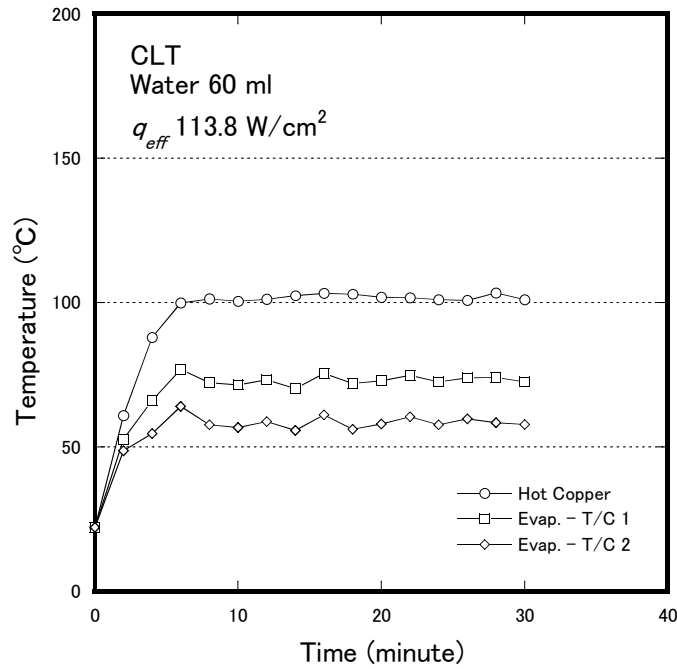


Fig. 5.1. Temperature profile of hot simulated component and evaporator at heat flux 113.8 W/cm^2 with 60 ml fill charge rate.

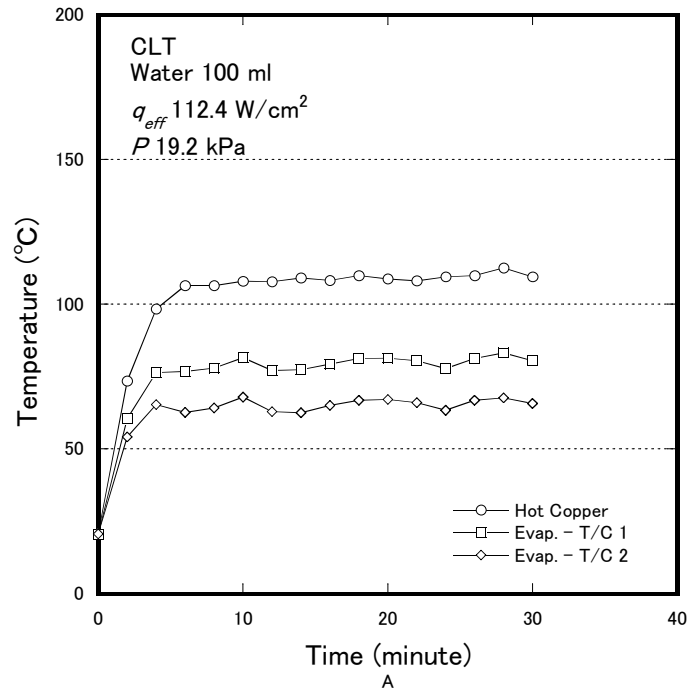


Fig. 5.2. Temperature profile of hot simulated component and evaporator at heat flux 112.4 W/cm² with 60 ml fill charge rate.

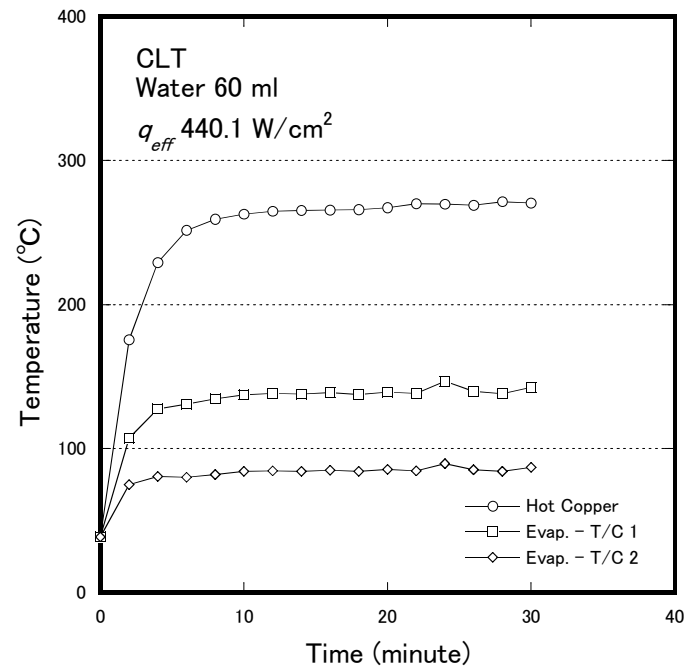


Fig. 5.3. Temperature profile of hot simulated component and evaporator at heat flux 440.1 W/cm² with 60 ml fill charge rate.

The steady-state conditions of two-phase flow boiling process were observed often to exist in the higher heat loads compared to the lower ones. It is considered that the power at above around 350 W will induce a stable performance of CLT device. This phenomenon also was observed in the measurements in other fill charge rate and pressures, as seen in Figs. 5.4, 5.5, and 5.6.

It is seen in Fig. 5.6 that the overall performance of CLT depends on the increasing of system pressure by increasing of heat input to the heater. The system pressure will give most significant effect since the closed loop thermosyphon can adjust by itself the required evaporation and condensation based on the temperature and pressure in response to the heat input. During the steady-state running, CLT could maintain the system pressure and produced the vapor bulk temperature at around saturation boiling regime. The heat flux in Fig. 5.7 is the one dimensional steady-state conduction heat transfer measured between two probed 0.5 mm thermocouples (T/C wall) in the evaporator.

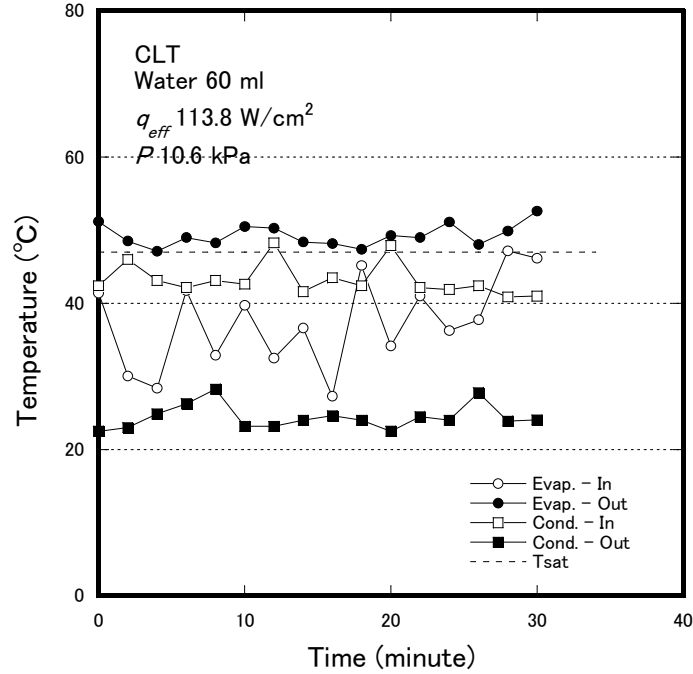


Fig. 5.4. Temperature profile of working fluid at Evaporator and Condenser for inlet and outlet conditions at $P 10.6 \text{ kPa}$.

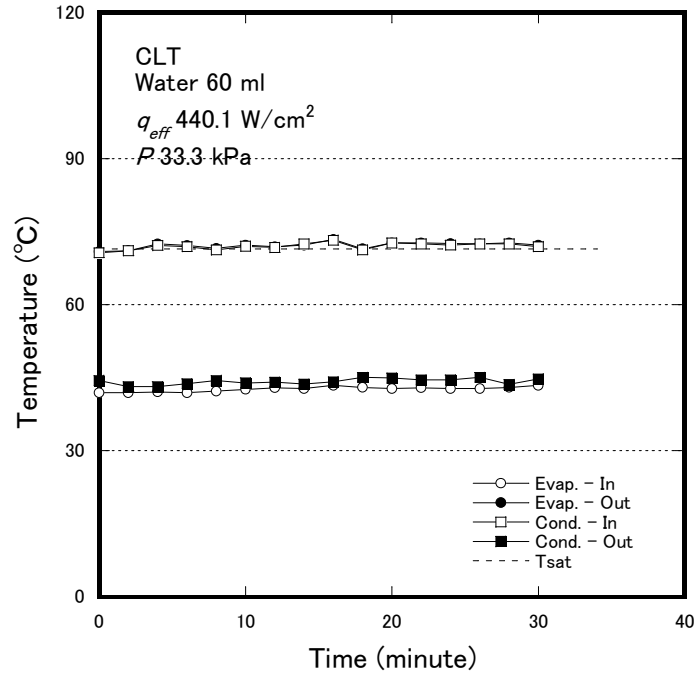


Fig. 5.5. Temperature profile of working fluid at Evaporator and Condenser for inlet and outlet conditions at $P 33.3 \text{ kPa}$.

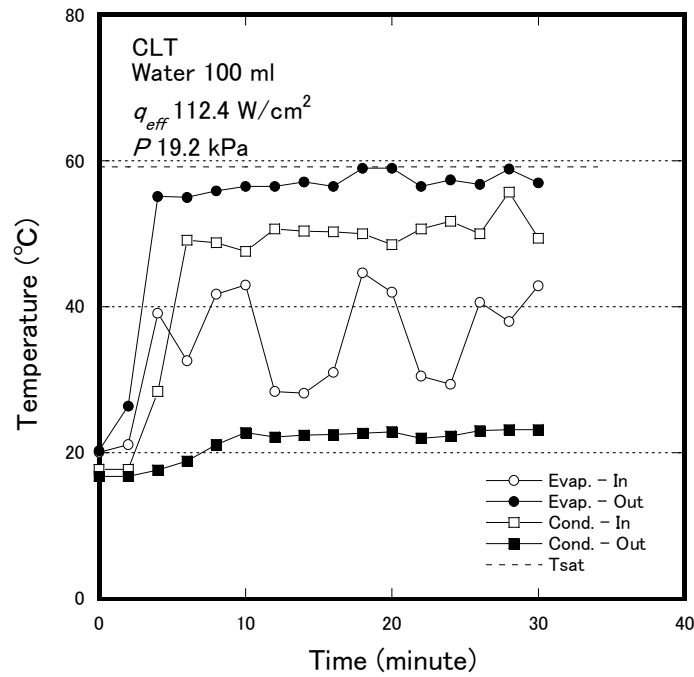


Fig. 5.6. Temperature profile of hot simulated component and evaporator at heat flux 112.4 W/cm^2 with 100 ml fill charge rate.

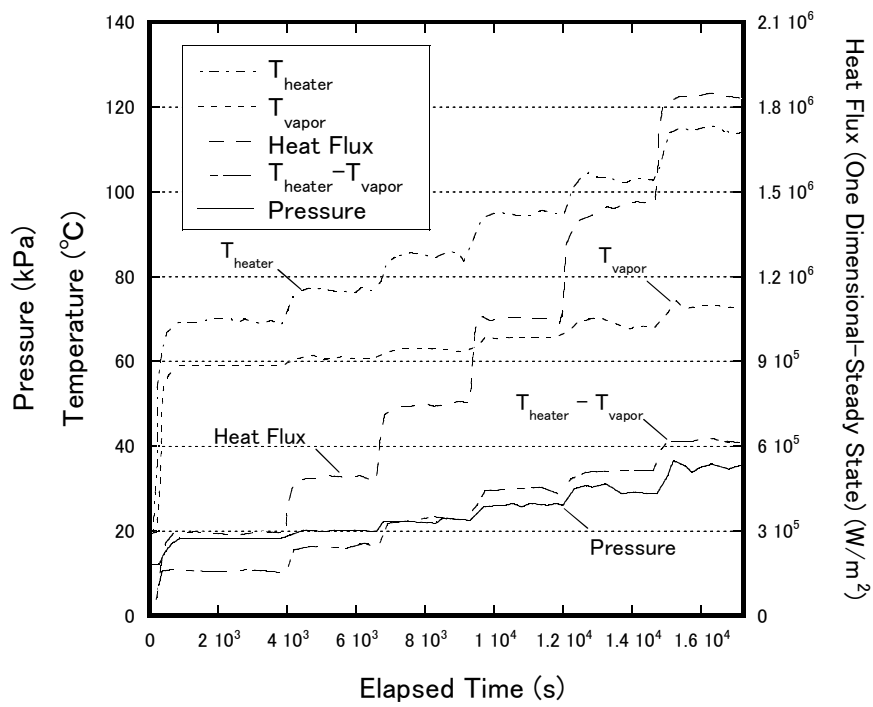


Fig. 5.7 Time dependence profile of CLT for 60 ml fill charge rate

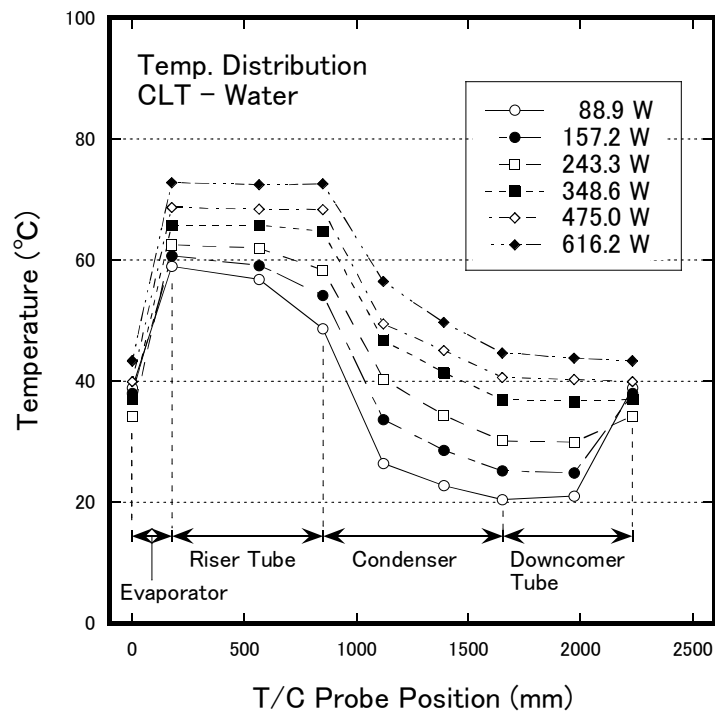


Fig. 5.8 Temperature distribution for 60 ml fill charge rate

The evaporator and condenser give stable performances which seem occur at higher heat load above around 350 W with higher mass fluxes that could keep the bulk vapor in an almost constant temperature from evaporation process up to inlet position of condenser. The condenser of direct hull cooling method could also maintain the condensation process at temperature of condensate at around 30 °C lower than the inlet vapor temperature of the condenser.

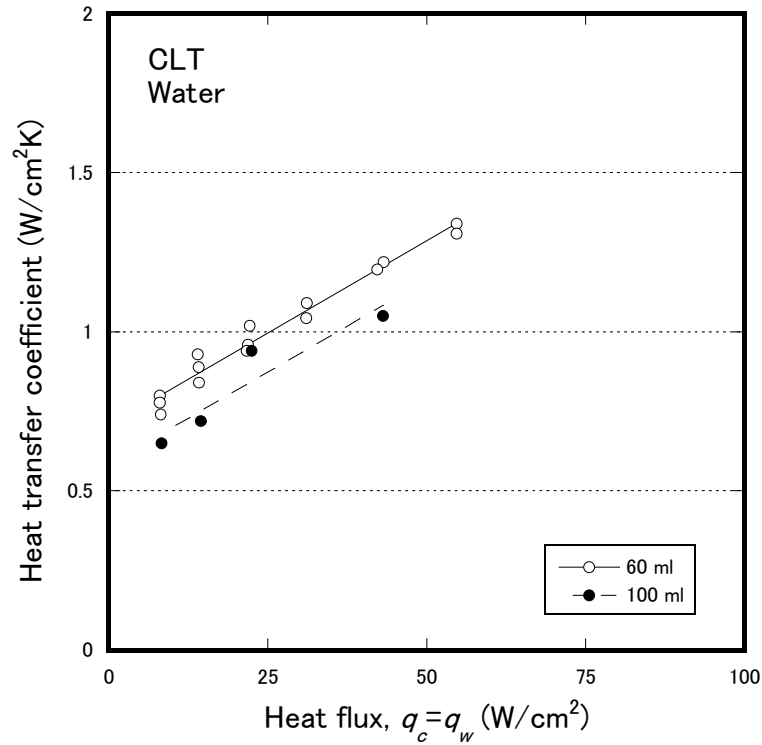


Fig. 5.9. Heat flux vs. Heat transfer coefficient

The heat transfer coefficient increase almost linearly as the heat input increase. The flow boiling process in the channel could maintain the saturated condition and give heat dissipation process through channels configuration. However, the fill charge ratio at 100 ml showed lower values than the 60 ml fill charge ratio. It is considered due to the filling or flooding effect of top region of the riser of CLT.

Chapter 6 – Conclusions

Investigations on the boiling heat transfer phenomena were performed, which include basic knowledge of transient pool boiling and the application study of flow boiling heat transfer through examination on the developed cooling device. The general conclusion in this research covers the requirements of further study on boiling heat transfer. In the investigations, the major conclusions are as follows:

1. The transition of boiling heat transfer to film boiling in the pool boiling heat transfer in wetting liquid of FC-72 can be categorized into three kinds of mechanisms. They are mainly related to the kind of initial boiling process and nucleation process, and the transition to film boiling at CHF. Those are: the transition from non-boiling to film boiling through fully developed nucleate boiling or FDNB, the transition to film boiling without nucleate boiling or direct transition, and transition from non-boiling to film boiling through a short nucleate boiling process or the semi-direct transition, that were named as the type 1, 2, and 3. The incipient boiling process in the pool boiling of FC-72 due to rapid increasing heat input during direct transition to film boiling was observed to be different from the mechanism of boiling process due to active cavities. It was confirmed that the incipient boiling process was induced by the explosive-like heterogeneous spontaneous nucleation (HSN) from the originally flooded cavities on solid surface.
2. The incipient boiling surface superheat obtained by a pre-pressure to the heater was suggested to induce the process of direct transition to film boiling even from quasi-steadily increasing natural convection due to quasi-steadily increasing heat input. This fact confirmed the new mechanism of boiling incipience in the pool boiling phenomena due to the explosive-like HSN.

3. The direct transition process has been investigated and the prediction related to the incipient boiling and the CHF due to rapid increasing heat input was derived.
4. The flow boiling heat transfer process in the CLT that was developed gave several important mechanisms required for further studies:
 - (a) The CLT provide an efficient of heat transfer process in higher heat inputs rather than the lower ones. The subcooled inlet conditions and the back flow phenomena was observed in the lower heat input below about 350 W.
 - (b) CLT is suited for thermal management which could give self adjustment of pressure control and cooling process in response to the heat input to the heater.
 - (c) The flooding mechanism should be controlled to avoid the dryout or the CHF early on the high heat dissipation.
 - (d) Higher fill charge ratio might give lack of performance that was related to flooding limitation.
 - (e) Heat flux and heat transfer coefficient is increase as the heat load increase in the system.

From this study, the key of contributions were addressed to the development of experimental configurations to obtain more predictive method of critical heat flux on finite heater in a pool boiling heat transfer due to exponentially increasing heat inputs especially for designing thermal system and concerning the direct transition involving the explosive-like incipient boiling process. In the flow boiling studies, the development and the examination of the CLT device contribute a basic understanding of the two-phase system, which is required for further analysis on the CLT as the cooling system for high-power electronic devices.

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Appendix

Table A.1 Physical properties of FC-72

Pressure [kPa]	T_{sat} [C]	ρ_l [kg/m ³]	ρ_v [kg/m ³]	h_{lv} [J/kg]	σ [N/m]
8.6	-1.00	1.79E+03	1.30	9.89E+04	1.34E-02
14.6	9.60	1.75E+03	2.12	9.51E+04	1.25E-02
23.5	19.90	1.72E+03	3.31	9.13E+04	1.15E-02
36.6	30.30	1.69E+03	5.00	8.74E+04	1.06E-02
54.7	40.50	1.66E+03	7.29	8.35E+04	9.66E-03
79.5	50.70	1.63E+03	10.37	7.95E+04	8.78E-03
101.3	57.70	1.61E+03	13.03	7.67E+04	8.18E-03
112.4	60.80	1.60E+03	14.38	7.54E+04	7.92E-03
154.9	70.80	1.57E+03	19.53	7.12E+04	7.09E-03
208.9	80.70	1.53E+03	26.06	6.69E+04	6.28E-03
276.2	90.50	1.49E+03	34.25	6.24E+04	5.50E-03
297.4	93.20	1.48E+03	36.84	6.12E+04	5.29E-03
358.8	100.30	1.44E+03	44.44	5.78E+04	4.74E-03
458.9	110.00	1.39E+03	57.12	5.30E+04	4.02E-03
493.6	113.00	1.38E+03	61.61	5.15E+04	3.80E-03
578.9	119.70	1.34E+03	72.89	4.79E+04	3.32E-03
689.7	127.30	1.30E+03	88.18	4.37E+04	2.79E-03
721.1	129.30	1.28E+03	92.66	4.25E+04	2.65E-03
885.9	138.87	1.23E+03	117.40	3.68E+04	2.02E-03
888.7	139.00	1.23E+03	117.84	3.67E+04	2.01E-03
1082.0	148.60	1.17E+03	150.27	3.03E+04	1.42E-03
1085.2	148.80	1.16E+03	150.84	3.02E+04	1.41E-03
1278.1	157.09	1.11E+03	186.10	2.39E+04	8.59E-04
1313.2	158.50	1.10E+03	196.16	2.28E+04	8.58E-04
1578.6	168.40	1.04E+03	266.42	1.32E+04	3.65E-04
1841.1	177.02	9.81E+02	363.90	0.00E+00	0.00E+00

Table A.2 Physical properties of gold and platinum

Properties	Gold	Platinum
Phase	Solid	Solid
Density	19.3 g/cm ³	21.45 g/cm ³
Melting point	1064.18 °C	1768.3 °C
Boiling point	2856 °C	3825 °C
Heat of vaporization	324 kJ/mol	469 kJ/mol
Heat capacity	25.418 J/(mol·K) (25°)	25.86 J/(mol·K) (25°)
Electrical resistivity	22.14 nΩ·m (20°C)	105 nΩ·m (20°C)
Thermal conductivity	318 W/m.K (300 K)	71.6 W/m.K (300 K)