Critical heat flux induced by flow instability in boiling channels – a review

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Abstract: Recent advanced technologies in boiler manufacturing bring about highly efficient water-tube boilers for gas firing or oil firing. These boilers have been designed mainly based on the critical heat flux criteria obtained under steady state. This is, however, not enough. In reality, physical burnout often takes place even after these 4 or 5 decades-long research history. Why? This may be mainly caused by a lack of knowledge on transient or unsteady behavior of two-phase flow in boiler tubes. In addition, there exists a certain discontinuity or gap between the laboratory test and industrial practices, and also discontinuity of knowledge and experience between senor and younger generations is much more serious. This paper describes various aspects of CHF and related phenomena with reference to practical applications in fossil-fuel-fired water-tube boilers. Starting from the various examples of accidents, important issues to be solved are discussed. Then the CHF problems under oscillatory flow condition are described. CHF - i.e. dryout in the present case, is mainly controlled by two-phase flow dynamics and is suitably simulated with a simple lumped-parameter modeling of boiling channel, while many other problems still remains unsolved. And finally a high potential of newly developed approach is demonstrated by realizing inherent void fraction fluctuation and flow pattern map of two-phase flow.

Keyword: boiling channel, critical heat flux, flow instability, density wave oscillation, quenching

1 Definition of the problem

In the design of boiling two-phase flow systems such as boilers and nuclear reactors, theoretical and/or empirical formulas related to thermal hydraulics are described, normally, in terms of steady state characteristics. However, such systems may be encountered an unanticipated dynamic behavior - e.g. flow instability problems - during the startup, partial load and/or transient operations even in a suitably designed system aiming at full-load and steady operations. Flow instabilities mean flow situations accompanied with large-amplitude fluctuations of flow, which may cause significant pressure and temperature fluctuations, departure from steady and safe operation, and mechanical vibrations of structures leading to thermal or mechanical fatigues.

Consider a natural circulation boiler consisting of downcomers, a large number of boiling channels of waterwalls and convective tube-banks, steam drum and water drum, as exemplified in Fig.1. As the water velocity in the downcomer is relatively low, the pressure drop is almost constant. Thus the constant-pressure-drop boundary condition is imposed on parallel boiling channels, even in the case of transient operations.

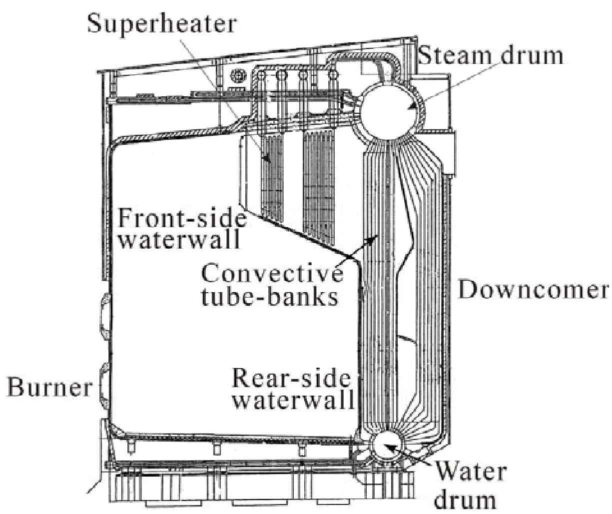


Fig.1 Natural circulation boiler.

Received date: March 20, 2010

The flow rate in each tube is, in principle, determined by the momentum balance throughout the loop. When a certain magnitude of perturbation in heat flux, flow rate or pressure happens in a boiling channel, then resultant density perturbation, in this case "void wave" or "kinematic density wave", propagates together with the mixture velocity. This density wave has relatively long time delay in propagation - residence time - in comparison with pressure wave propagation. The void fraction perturbation is converted to pressure drop perturbation in two-phase flow region. The pressure drop boundary condition across the boiling channel remains constant, so that the pressure drop perturbation in the boiling region brings about pressure drop and thus flow rate perturbations, but with negative sign so as to compensate the former pressure drop perturbation, in a subcooled region. Such a feature is well demonstrated in Fig.2.

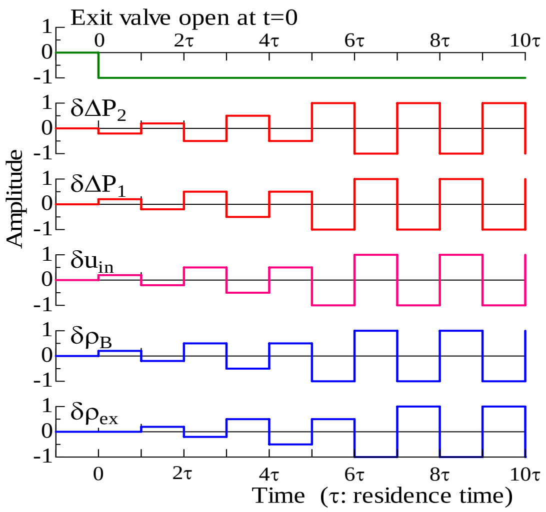


Fig. 2 Density wave oscillation. ($\Delta p_1, \Delta p_2$: pressure drop in subcooled and boiling region, respectively).

This type of flow instability is referred to as "density wave oscillation", being commonly observed in a boiling-channel including boiler, nuclear reactor and heat exchangers with phase change [1, 2].

Flow instability problems are not limited in an occurrence of such density wave oscillation, but also include many other various types of instabilities, as listed in Table 1. Among these flow instabilities, Ledinegg instability, - flow excursion - between parallel boiling channels, is first recognized one in natural- and forced-circulation boilers in 1938. The occurrence of drastic flow mal-distribution is followed by the critical heat flux (CHF) condition in a boiling channel at very low mass flux. In addition, Ledinegg pointed out the possibility of burnout due to flow stagnation or flow reversal in an evaporator tube, depending on piping arrangement and thermal and geometrical conditions of boiler furnace [3]. Such a discussion was also conducted by Muenzinger [4] in 1933, the oldest one found in my library. Ledinegg didn't mention about oscillatory flow instability, while Muenzinger mentioned about periodical pulsation during the startup of natural circulation boiler. He suggested possible piping design, so that the downward flow with boiling is avoided. If flow reversal takes place in a certain evaporator tube, which is originally designed aiming at upward flow, bubble stagnation may happen. This flow stagnation of bubble leads to burnout and physical rupture of the tube. Such a flow situation is typically exemplified in Fig.3, where bubble is almost stagnant during the period from 1.0 to 1.4 s, being a flow reversal phase of liquid flow oscillation.

Table 1 Classification of flow instabilities

Category	Pattern	Mechanism	Feature
negative resistance instability	flow excursion or Ledinegg instability	negative damping in 1st-order system	transitional, significant flow mal-distribution appears in parallel-channel system
	pressure drop oscillation	dynamic interaction between flow excursion and accumulation mechanism of mass and momentum	relaxation oscillation with large amplitude and long period
time-delayed feedback instability	density wave oscillation	propagation delay of void wave (kinematic wave) and feedback effect provide negative damping	oscillation, period comparable with residence time appears in positive resistance region of pressure drop vs. flow rate
thermal non-equilibrium instability	geysering	insufficient nucleation sites bring about large superheat followed by violent boiling	relaxation oscillation if liquid refilling mechanism exist

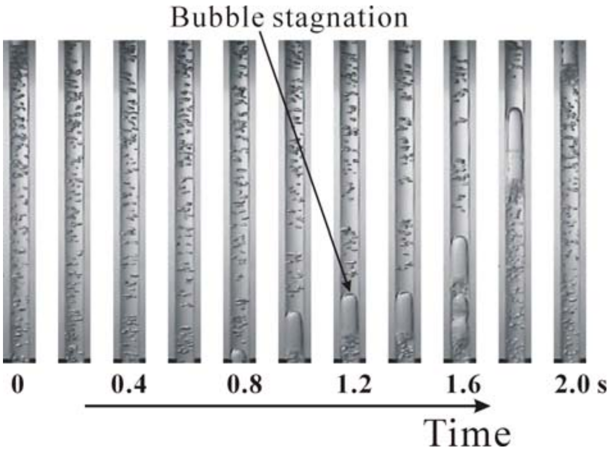


Fig. 3 Flow pattern transition in oscillatory flow.

Referring to rather old text book of BWR, the discussion on flow instability was mainly conducted, of course, in the framework of nuclear-thermal coupling - i.e. power excursion and related “chugging” - seemingly flow oscillation - observed in BORAX experiment [5]. In this book, we find the statement: “this oscillatory operation is by itself not inherently undesirable, nor necessarily dangerous. It should, however, be considered a warning that the reactor system has reached an operating condition which may lead to the chugging type operation” [5]. The oscillatory flow means density wave oscillation, and the relationship to the CHF, although being a key issue for reactor safety, had not been considered as a serious problem at that stage. This might be owing to the fact that the heat flux was far below the present level. Not only in nuclear reactor but also in boiler furnace, the heat flux successively increased with the development of high-performance systems.

In the 1960s, probably the first monogram: “Boiling Heat Transfer and Two-Phase Flow” by Tong [6] was published, which mainly focused on the boiling crisis - CHF- and flow instability in relation to the BWR. He stated clearly that “flow oscillations affect the local heat transfer characteristics and the boiling crisis”. Based on the data by Ruddick (1953), the critical heat flux is reduced 40% under flow oscillation. This 40% is quite reasonable estimation in comparison with our data [7-9].

An importance of such problems is generally claimed in literatures on two-phase flow instabilities and CHF, while the discussion and data are rather limited [10-13]. The claim remains, in most of the case, in the community of researchers, and is not hold in common by designers of boilers and heat exchangers, and possibly by nuclear people, who don’t have such experiences. Most of the people may expect that even steady flow CHF is a complicated phenomenon, not to mention oscillatory flow CHF. As is well-known, a variety of CHF models have been so far proposed [e.g. 10-13], and there exit, of course, many affecting parameters. Under steady flow condition, it is, in reality, rather hard to specify what is the triggering phenomena leading to CHF, because two-phase flow shows a fluctuating properties owing to discrete phase distribution and interfacial waves. On the other hand, during flow oscillation, relatively small or weak perturbation is put behind a large flow fluctuation, and therefore the thermal flow field becomes much simpler than so far expected.

This paper describes oscillatory flow CHF in view of, partly, practical application, so that the importance of flow stability, especially, in relation to the CHF of boiling channels may be understood by the possible readers.

2 Examples

The boiler, illustrated in Fig.1, of 150 t/h steam output generated 12MPa and 500°C superheated steam for power generation. The circulation ratio was 7.6 - i.e. corresponding approximately to exit steam quality $x_{ex} = 0.16$. This boiler was tripped after several months of commissioning operation due to physical burnout, ballooning and rupture of tubes of the furnace front-side water wall, around the burners. Through the incident investigation, rather thick silica layer was found on the inner tube wall at the damaged area. Possible cause - i.e. root to the incident - was not only bad water treatment but also flow mal-distribution [14]. The characteristic feature of this boiler was that a feed-water heater was installed in the water drum. A difference in water temperature reached about 60 K between almost saturated temperature from the downcomer and that of feed-water. This large temperature difference brought about a large subcooling at a part of water-walls, which causes a serous reduction in the water circulation in the corresponding area. The incident was limited in the zone just corresponding to the extent occupied by the feed-water heater in the water drum.

Second example is small-scale water-tube boiler. Recent advanced boiler technologies enabled to construct efficient and compact water-tube boilers with 1 to 2 t/h-steam output as illustrated in Fig. 4 [15, 16]. The design concept consists in burning clean gaseous fuel - e.g. methane, propane and dimethyl ether - in the combustor to cool the diffusion flame directly of the burning reaction zone. The heat transfer out side water tube is dominated by convection, but with radiation of about 40%, being much lower than convective heat transfer. Thus the enhancement of heat transfer in the gas side, especially, downstream the reaction zone, is essentially conducted by means of - e.g. finned tubes. The burner jet directly impinges on water tubes, and thus throughout the reaction zone and convective heat transfer zone the water circulation is a substantial factor for the safety and efficient operation of this boiler. When sufficient water is not supplied, physical burnout takes place, as typically shown in Fig.5. The water loop is rather simple as shown in Fig.6, where the water tubes have dimensions of 44.4mm I.D., 50.8mm O.D. and 800mm in length. These water tubes are connected to both upper and lower plenums with shallow rectangular cross section. The operating pressure is from 0.7 to slightly less than 1.0 MPa. The inlet velocity is normally from several centimeters per second to 10cm/s, actually very low. As the pressure is rather low, the flow in the tube is highly agitated by a large volume of generated steam, although the inlet velocity is rather low. Then, the two-phase mixture rushes into the upper plenum, where vapor is partly separated and water goes down into the water tube again, where the flow may become unsteady, i.e. irregular or alternative up and down flows may appear. Such a system is similar to that focused by Katto and Hirao [17], in which they conducted experiments with counter-current flow boiling system, similar to a closed two-phase thermosyphon. In the boiler shown in Fig. 4, rather shallow plenum, where in some case buffer plate is inserted, may enhance such flow reversal into the water tube, so that the tube wall is continuously or even periodically wetted. In this case, flow fluctuation is positively utilized to avoid CHF, without an increase in the inlet water velocity.

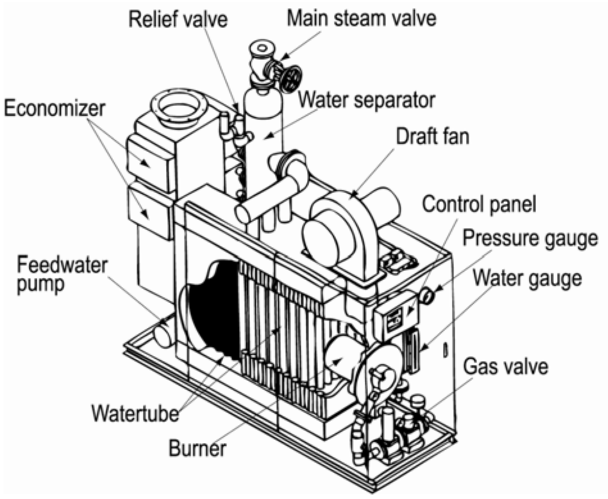


Fig. 4 Compact water-tube boiler.

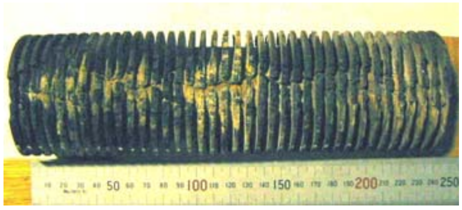


Fig. 5 Example of physical burnout of boiler tube.

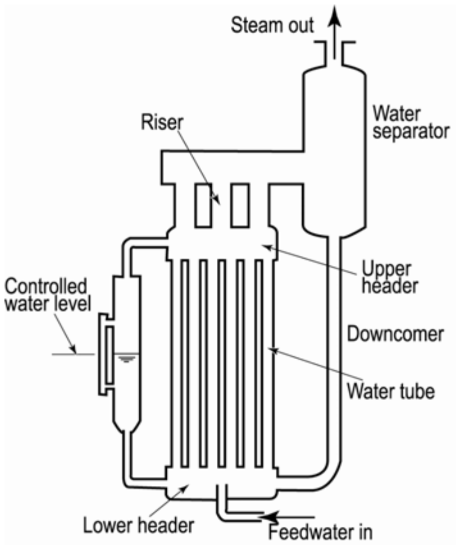


Fig. 6 Water circulation loop of compact boiler.

As mentioned above, the CHF problem is really not so clean phenomena found in university laboratories, but is rather complicated, unsteady, fluctuated and of diversity. We must, however, have a certain methodology or strategy for the CHF, otherwise the design based only on the steady state characteristics may fail and brings about serious incident as shown in Fig. 5.

In order to avoid CHF, the followings are substantial:

1. Suppress flow instabilities

2. Reduce heat flux below the safety level with a sufficient margin
3. Retain water circulation at high level
4. Suppress oscillation amplitude, even in case of oscillation

3 CHF under flow instability

Normally, power boiler and probably nuclear reactors have sufficient magnitude of inlet pressure drop (e.g. by orifice) by about 30% of the total pressure drop. Whether the magnitude is enough or not is dependent on the system configuration and operating conditions. If a boiling channel system consists of parallel channels, and the frictional component dominates the whole pressure drop, then drastic flow mal-distribution takes place, as typically exemplified in Fig. 7, between parallel channels, being caused by flow excursion or Ledinegg instability [18]. If the system has, in addition, upstream compressible volume, pressure drop oscillation may be encountered [19-23]. In such cases, flow rate in one of the channels decreases so drastically that the CHF may initiates. This is also the case in micro- and mini-channel systems [24, 25].

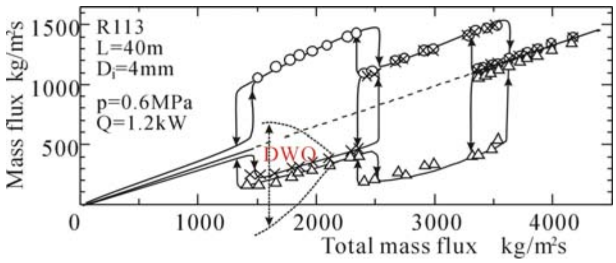


Fig. 7 Flow mal-distribution due to Ledinegg instability in parallel channel [18].

The final example in this section is works done by Mishima [26] and Mishima et al.[27]. Mishima and his group conducted experiments using a boiling channel with flexibility of arrangement and operation: up and down flows and with and without surge tank (i.e. compressibility). Fig. 8 represents an example of their results, where normalized heat and mass fluxes are defined by

$$q_{cr}^{*} = q_{cr} h_{LG}^{-1} (g\lambda_c \rho_G \rho_{LG})^{1/2} \quad (1)$$

$$G^{*} = G_0 (g\lambda_c \rho_G \rho_{LG})^{-1/2} \quad (2)$$

where $\lambda_c (\equiv \sqrt{\sigma/(g\rho_{LG})}$ the characteristic wave length. Stiff system (the term "stiff" and "soft" were denoted by Mishima) showed good agreement with Katto's correlation for high vapor quality. Soft system suffers from flow oscillation (pressure drop oscillation in this case) or flow excursion, and thus the CHF decreased drastically to about 35% of the data of stiff system. This amount is consistent to the description by Kramer [5] and Ozawa et al.[7-9].

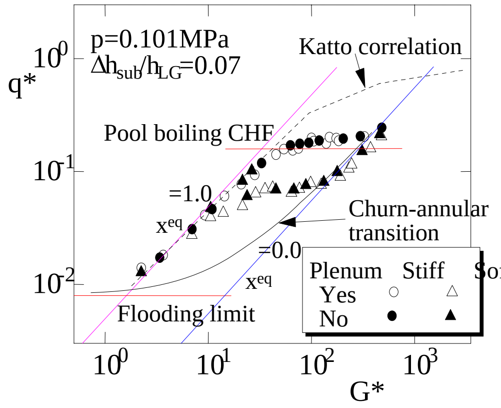


Fig. 8 Critical heat flux in stiff and soft systems[26].

Including these papers, a literature survey was conducted and listed in Table 2. As the number of literatures is limited, the list may give only a flavor of so far conducted experiments.

Hewitt gave interesting remarks [39] upon his chaired session:
"One is interested in instability in the burnout context for two reasons:
(1) Are the available burnout data obtained under truly steady state conditions?
(2) Are steady state burnout data applicable in a given real system?"

The above comments were given in EURATOM Symposium on Two-Phase Flow Dynamics in 1967, while the question, even now, remains unanswered.

4 CHF under density wave oscillation

As far as flow excursion and pressure drop oscillation are concerned, the system suffers from significant decrease in the flow rate, leading to CHF, because both types of instability is, in any case, a relaxation process between high and low mass fluxes. On the other hand, in the density wave oscillation, the flow rate oscillates, in principle, in sinusoidal form, i.e. fluctuation around a time-averaged value. This means that wall is dried

Table 2 Researches on oscillatory and transient flow CHF

Authors (Year) [ref.]	Method	Key words	Remarks
Aladyev et al. (1961)[28]	experimental	flow instability induced by upstream compressibility, heat flux distribution	water, d=8mm, L/d=15-188, P=20-200atm, G=300-5000kg/m^{2}s, vertical flow
Sato et al. (1966) [29]	experimental & analytical	imposed forced flow fluctuation, flow instability induced by upstream compressibility	water, annular channel (I.D.8mm, O.D.18, 22 & 27mm), L=600mm, vertical flow
Bergles et al. (1967) [30]	experimental	flow-pattern transition instability, choking flow	water, d=0.079-0.242in, horizontal flow
Ishigai et al. (1974) [31]	experimental & analytical	imposed stepwise transient of mass flux, premature dryout	water, d=8mm(O.D.10mm), L=750mm, P=5atm , G=200-400kg/m^{2}s, vertical flow
Mishima et al. (1985) [27]	experimental	flow instability induced by upstream compressibility, upflow and downflow	water, d=6mm, L=344mm, P=1atm, instantaneous G=0-1700kg/m^{2}s, vertical flow
Kasai et al. (1986) [32]	experimental & analytical	BWR fuel bundle, parallel channels, flow oscillation, lumped parameter model, verification of transient thermal hydraulics code	water, dout=12.3mm, rod bundle 2x2 & 4x4, L=3.708m, vertical flow
Domashev et al. (1992) [33]	experimental	natural circulation, imposed heat flux & mass flux perturbations	water, d=8mm(O.D.10mm), L=1m, P=9.8-13.7MPa, G=1000-2000kg/m^{2}s, vertical flow
Ozawa et al. (1993) [7]	experimental	imposed forced flow oscillation	water, d=5mm(O.D.6mm), L=0.9m, G=70-450kg/^{2}ms, P=0.1MPa, relative amplitude of G=0.2-3.8, τ =2, 4, 6 s
Celata et al. (1994) [34]	experimental	power & mass flux transients	R-12, d=7.57mm(O.D.9.51), L=2.1m, P=1.2-2.5MPa, G=400-1600kg/^{2}ms, vertical flow
Nair et al. (1996) [35]	analytical	stability analysis, flow excursion, density wave oscillation	downward flow, drift-flux model, linear stability analysis, CHF pattern map, similarity parameters
Ruggles et al. (1997) [36]	analytical	oscillatory two-phase flow	effect of wall heat capacity(heater dynamics), perturbation method
Umekawa et al. (1997) [37]	experimental	natural circulation, density wave oscillation	liquid nitrogen, d=3mm(O.D.4mm), d=5mm (O.D.6mm), L=0.9m, P=0.3, 0.4MPa, uin=0.05-0.35m/s, vertical flow
Kim et al. (1999) [38]	experimental	forced and natural circulations, stable and oscillating flow	water, d=5, 6.6, 9.8mm, L=0.5-0.6m, P=0.12MPa, G=50-380kg/^{2}ms, relative amplitude of G=0.05-0.65, τ =2-11 s, vertical flow
Ozawa et al. (2001) [9]	experimental & analytical	imposed forced flow oscillation, lumped-parameter model, included flow reversal	water, d=3mm(O.D.4mm, 6mm, 9mm), 4mm (O.D.4.5mm), 5.8 mm(O.D.14mm), 15mm (O.D.16mm, 19mm), annular channel(I.D.4.5mm, O.D.8mm), L=0.9m, P=0.3, 0.4MPa, G=12-1800kg/^{2}ms, relative amplitude of G=0-10.2, τ =2, 4, 6 s, vertical upward, downward flows, horizontal flow

out, if dryout type is concerned, and rewetted alternatively.

As is well-known, density wave oscillation is the most frequently appeared type of instabilities in parallel channel systems. The density wave oscillation is caused by the dynamic interaction between flow rate, void wave propagation and pressure drop, as has been explained with Fig. 2. Then the oscillation period is an order of residence time of fluid in the channel, while an amplitude, especially, of mass flux, is small and sinusoidal just beyond the threshold condition, and develops drastically with an increase in heat flux, so that the wave form becomes a limit cycle oscillation with flow reversal. The oscillation period and amplitude are determined not only by the boiling channel itself, but are affected by surrounding circumstances, such as upstream and downstream conditions, and geometrical configurations etc. Thus it is normally hard to look insight, even in a macroscopic way, into the CHF under flow instability. Our experiments are aimed at giving an answer to this problem.

Typical examples observed in a natural circulation boiling system with liquid nitrogen [37] are shown in Fig. 9. Solid dots represent time-averaged value of inlet velocity, and triangles a maximum and a minimum values of the amplitude. The top column is the case that the density wave oscillation, indicated by 'DWO', first appeared and then CHF condition is reached. Increased the inlet restriction (in the second column of Fig. 9), the density wave oscillation was suppressed a great deal compared with the first column.

Then the CHF inception almost coincides with the threshold of oscillation. Further increase in the inlet restriction suppresses flow instability, and brings about steady flow. As the inlet restriction is rather large and the velocity must be small. A direct comparison between data in Fig. 9 is normally difficult, because not only the velocity but also amplitude are different. Neglected effects of various parameters, such as mass flux and heat flux, the critical heat flux was plotted as a function of relative amplitude in Fig. 10, where the critical heat flux is normalized by Katto's correlation [11].

An application of Katto's correlation has been well verified in comparison with the steady flow CHF. Every data has respective unique velocity, and therefore the comparison seems somewhat rough, while overall trends show the tendency that the normalized CHF decreases with an increase in the relative amplitude. The curves in Fig. 10 are simulation results, described below, for specified operating conditions indicated in the figure. The reduction in CHF seems to become saturated with an increase in relative amplitude, i.e. a certain lower limit exists. It should be noted that CHF data measured under flow instabilities have a difficulty in comparing and deducing general tendency consistent throughout the data, because mass flux is a function of heat flux and inlet restriction, especially in a natural circulation system. Thus in the next section, CHF was measured under forced flow oscillation, retaining constant time-averaged flow rate and well-defined amplitude of oscillation.

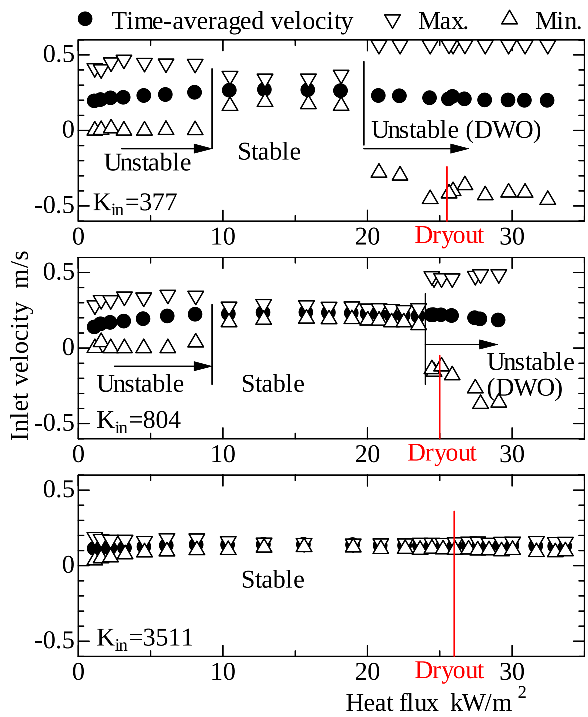


Fig. 9 Time-averaged flow rate and amplitude of flow oscillation in natural circulation system of liquid nitrogen.

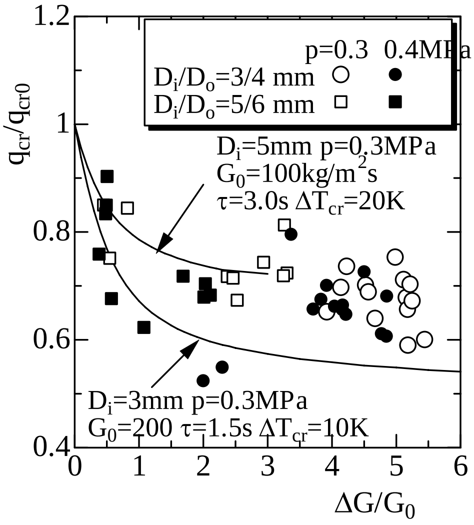


Fig. 10 CHF in natural circulation system of liquid nitrogen.

5 CHF under forced flow oscillation

5.1 Problems arisen in experiments

In discussing CHF under flow instabilities, as was mentioned above, the system configuration is quite important because the flow instabilities are significantly affected not only by the heated section but also surrounding system. In order to avoid such conjugated effects on CHF, and, in addition, to retain consistency throughout various operating conditions, forced flow oscillation was imposed on steady flow, and the CHF was measured. Such an approach is similar to dynamic response of classical control theory, in the sense obtaining transfer function of the system. This present experiment is directed to density wave oscillation, and thus the experimental setup must cover relatively wide range of oscillation behavior, e.g. amplitude and frequency, being reasonable in nuclear reactor or boilers. Still, we have a substantial difficulty in scaling boiling two-phase flow, i.e. too many scaling parameters: phase change number, subcooling number, friction number, Reynolds number, Froude number and so on [40].

Experimental setup shown in Fig.11 consists mainly of reserve tank of de-ionized water, pump, flow-metering system, vertical test section, and steam-water separator. The forced flow oscillator, consisted of a piston-cylinder and geared motor, imposes a predetermined amplitude flow oscillation with a definite oscillation period on the constant discharge from the pump. The test section was circular tubes of stainless steel heated uniformly by AC or DC current. AC power was applied to thin-walled tube, and DC power for thick walled tube. CHF, mostly dryout, was detected by thermocouples welded on the outer surface of the test tube. Approaching to the CHF condition, wall temperature becomes to fluctuate with large amplitude in accordance with the imposed flow oscillation. Amplitude of wall temperature grows up with the heat flux, and finally wall temperature excursion takes place. Experiment with downward flow was also conducted, while arrangement of piping was, of course, changed so as to meet the requirement.

As is wall-known, the wall heat capacity acts as a damping factor in temperature response, and for accurate measurement, we should solve inverse problem [41], which is, however, not easy in the present experiments. Therefore, provided the quasi-steady and one-dimensional temperature distribution across the wall, CHF was defined when the temperature fluctuation at the inner wall reached 100K beyond the saturation temperature. This critical wall superheat is, in reality, an arbitrarily determined value, while the criterion is the same as steady CHF experiment [26, 27]. Thus the same criterion was consistently used under steady and oscillatory flow conditions.

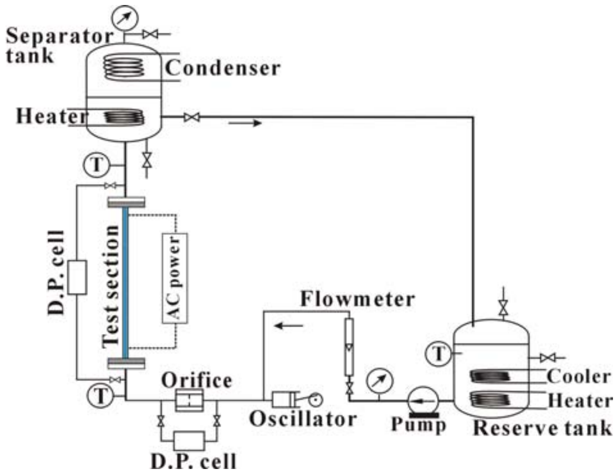


Fig. 11 Experimental apparatus.

It may be important to discuss here the effect of tube-wall heat capacity on the measured temperature fluctuation. In an oscillating temperature field, the wall heat capacity is effective within the thermal penetration depth:

$$\delta_\kappa = \sqrt{2k_W / \omega} \qquad (3)$$

The thermal penetration depth is, e.g. 7 mm for oscillation period of $\tau = 1\text{s}$. An increase in the oscillation period results in an increase in the penetration depth, and thus present wall thickness is well within the penetration depth: whole wall heat capacity has an influence on the wall temperature response.

Measured temperature on the outer wall suffers from damping and phase lag. Here such a damping and phase lag are simply estimated, for better understanding the problem, by means of a lumped-parameter approximation.

Energy balance across the wall is written as

$$q_0 - q = c_W \rho_W \frac{D_0^2 - D_i^2}{4D_i} \frac{dT_W}{dt} \qquad (4)$$

$$q = \alpha(T_W - T_f) \qquad (5)$$

The wall temperature perturbation is given as a function of fluid temperature perturbation, and is expressed by a first-order delay. Then the gain and phase lag are given by

$$\text{Gain}=\frac{1}{\sqrt{1+(\omega\tau_W)^2}}, \quad \text{Phase lag}=-\tan^{-1}(\omega\tau_W) \qquad (6)$$

where time constant is defined by

$$\tau_W \equiv c_W \rho_W (D_0^2 - D^2)/(4D\alpha) \qquad (7)$$

In estimating the time constant in a dryout region, the heat transfer coefficient in equation(7) may be a value for post-dryout heat transfer, while for simple estimation the heat transfer coefficient is estimated by the Dittus-Boelter equation. When thick-walled tube is the case, the gain and phase lag become 1/37 and 88°, respectively, for 3 mm I.D. and 9 mm O.D. at the oscillation period 2 s, and 1/4 and 75° at the oscillation period 2 s for thin-walled tube, 3 mm I.D. and 4 mm O.D. Thus experiments with thick-walled tube should be very careful. We may add the following to Hewitt's comments:

(3) Are oscillatory flow burnout data in a laboratory applicable to industrial system?

5.2 Limiting value based on lumped-parameter model

Suppose a rectangular wave, similar to Fig. 2, of alternative switching between dryout and rewetting. During a half period, the wall is heated up and in the next half period the wall is cooled down by the liquid film. During the wetted condition, wall temperature may be very close to the saturation temperature owing to very high heat transfer coefficient relative to the dryout or post-dryout condition. If this wall superheat reaches the critical temperature difference, $\Delta T_{cr} = T_W - T_S$ (=100K in the present case) during heat-up process, we say CHF condition arrives. Thus the following simplified relationship holds:

$$\tau_D = \tau_W \log(1/[1-\alpha\Delta T_{cr}/q_{cr}]) \qquad (8)$$

The period τ_D is referred to as dryout period in this paper. This equation is rewritten as

$$q_{cr} = \alpha_0 \Delta T_{cr}/[1-\exp(-\tau_D/\tau_W)] \qquad (9)$$

where heat transfer coefficient is simply given as a steady state value at time-averaged mass flux, e.g. by the Dittus-Boelter equation, being less than the actual value. Then the limiting case is given by

$$\tau_D/\tau = 0.5 \qquad (10)$$

5.3 Experimental observation and discussion

Experimental results are shown in Figs.12 to 15. Throughout the data, it is noted consistently that the CHF decreases drastically with an increase in the oscillation amplitude. In addition, longer the oscillation period is, lower the CHF becomes. Based on the simplified model, long period oscillation means long dryout period leading to heating up the tube wall. The CHF, as shown in Fig. 12, decreases, first, drastically and tends to saturate at a unique value determined by the oscillation period, i.e. limiting value calculated from Eqs. (9) and (10). The relative amplitude of $\Delta G/G_0 = 3$ seems rather large, while the density wave oscillation grows rather quickly so as to have such large amplitude as is clearly observed in Figs. 9 and 10. Thus the range of relative amplitude in the data is realistic values.

Figure 12 shows effect of flow orientation as well, i.e. vertical upward and downward flows. When the mass flux is relatively high, e.g. 200 and 400 kg/m^{2}s in the present experiment, there exists no significant difference between the upward and downward flow even in the case of large relative amplitude.

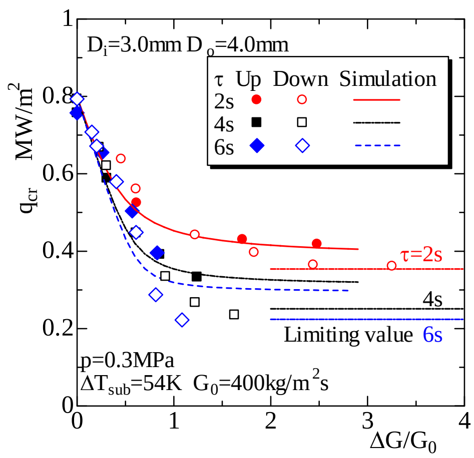


Fig. 12 Oscillatory flow CHF.

The effect of time-averaged mass flux on CHF is shown in Fig. 13. As the steady flow CHF is different at each mass flux, the oscillatory flow CHF is normalized simply by the respective steady flow data. The normalized CHF at 200kg/m^{2}s seems higher than that at 400kg/m^{2}s. The absolute value of CHF at 200 kg/m^{2}s is about 460kW/m^2. Reduction in CHF is about 30% at $\tau = 2$s (absolute value is 310kW/m^2) from the steady flow CHF, while for 400kg/m^{2}s the reduction reaches almost 50% - i.e. absolute value is 390kW/m^2 at $\tau = 2\,\mathrm{s}$ while steady flow CHF 760 KW/m^2.

When tube wall thickness is large, the heat capacity becomes large as well, and therefore the wall heat-up or fluctuation may be damped, so that the critical heat flux retains higher value than that in the thin-walled tube, as shown in Fig. 14. This description may be qualitatively correct, while plotted data in Fig. 14 has definite uncertainty caused by temperature measurement on the outer wall. It is worthwhile to draw attention to the fact that steady flow CHF data of both tubes occupy nearly the same level. In addition to circular tubes, experiment was conducted with annular channel, in which inner tube was heated. General feature in Fig. 15 is consistent with those of circular tubes.

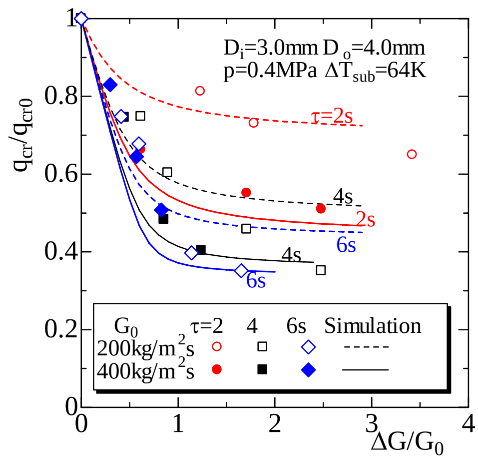


Fig.13 Oscillatory flow CHF (effect of mass flux).

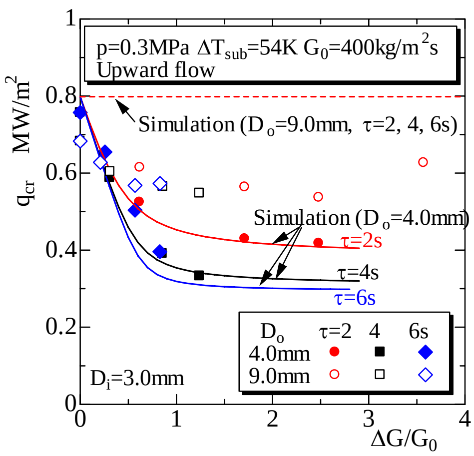


Fig. 14 Oscillatory flow CHF (effect of tube wall thickness).

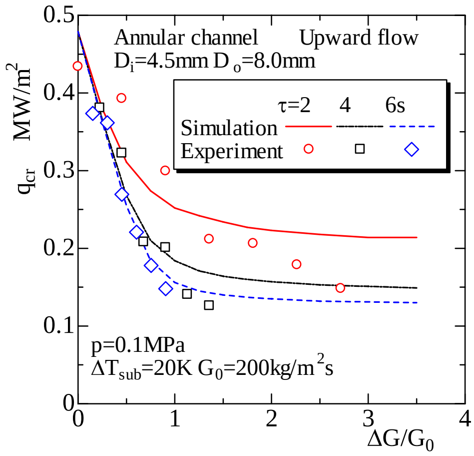


Fig. 15 Oscillatory flow CHF in annular channel.

5.4 Numerical simulation

CHF under oscillatory flow condition at relatively high quality, i.e. dryout, is mainly controlled by two-phase flow dynamics. Authors have already conducted numerical simulation of density wave oscillation based on a lumped-parameter model [42].

"Lumped-parameter" means that flow characteristics are first integrated in each subsection, assuming linear distribution of enthalpy, specific volume and velocity. Then, the two-phase flow dynamics, described originally as partial differential equation system, is converted to ordinary differential equation system, so that non-linear simulation becomes possible [43].

Assumptions are as follows:

1. One-dimensional homogeneous flow with thermodynamic equilibrium.
2. Wall heat capacity is expressed by a lumped-parameter model.
3. The flow field is subdivided into subcooled, two phase flow and superheated regimes.
4. Linear distributions of enthalpy, specific volume and thus velocity.

Thermal flow field are given by mass and energy conservation equations:

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u}{\partial z} = 0 \tag{11}$$

$$\frac{\partial \rho h}{\partial t} + \frac{\partial \rho h u}{\partial z} = q_v \tag{12}$$

Applied the linear distribution assumption, the velocity distribution in two-phase flow region is given by

$$\frac{du}{dz} = q_v \frac{v_{LG}}{h_{LG}} \tag{13}$$

Integration of energy conservation equation over the subcooled region gives the dynamics of subcooled length λ_L - i.e. boiling boundary - with the help of Leibniz rule,

$$\frac{d\lambda_L}{dt} = 2u_{in} - \frac{2q_v \lambda_L}{\rho_L \Delta h_{sub}} \tag{14}$$

In the same manner, the superheated boundary λ_G, where vapor quality becomes unity, is given by

$$\frac{d\lambda_G}{dt} = \frac{\rho_{TP} - \rho_L}{\rho_{TP} - \rho_G}\frac{d\lambda_L}{dt} - \frac{\rho_G u_{ex} - \rho_L u_{in}}{\rho_{TP} - \rho_G} \tag{15}$$

where $\rho_{TP} = \frac{1}{v_{LG}} \log\left(\frac{v_G}{v_L}\right)$ is an integrated-averaged density over the two-phase flow region. When the exit quality is less than unity, the exit specific volume of two-phase mixture is given by

$$\frac{dv_{ex}}{dt} = \frac{\rho_L u_{in} - \rho_{ex} u_{ex} - \left\{\frac{1}{v_L} - \frac{\log(v_{ex}/v_L)}{v_{ex} - v_L}\right\}\frac{d\lambda_L}{dt}}{\frac{L_T - \lambda_L}{(v_{ex} - v_L)^2}\left\{1 - \frac{v_L}{v_{ex}} - \log(v_{ex}/v_L)\right\}} \tag{16}$$

The equation system is numerically solved by means of Runge-Kutta method with an input of inlet velocity oscillation.

Wall temperature behavior is expressed, based on a lumped-parameter model, by

$$\frac{dT_m}{dt} = \frac{1}{c_W \rho_W}\left(q_W - \frac{4D_i q}{D_o^2 - D_i^2}\right) \tag{17}$$

where T_m is mean temperature of tube wall, and heat flux q is determined by

$$q = \alpha(T_m - T_f) \tag{18}$$

The heat transfer coefficient is given by the Dittus-Boelter equation in subcooled and superheated regions, and by Steiner's correlation [44] in two-phase region. It should be noted here that the wall temperature dynamics was determined by the flow in the channel, i.e. one-way coupling was used in this simulation. CHF was determined, in the same manner as in the experiment, by detecting outer wall temperature. This outer wall temperature was estimated as a function of the mean temperature determined from the simulation and the temperature gradient across the tube wall under quasi-steady state.

Simulation results are also drawn in Figs. 12 to 15. As the flow model is based on homogeneous flow assumption, the simulation results are independent on whether the flow direction is vertical upward or downward. Through the comparison with respective experimental data, present simplified numerical simulation is well confirmed its applicability to the dryout under flow oscillation, except the case with thick-walled tube shown in Fig. 14. The data of thick-walled tube must be carefully examined taking into account the thermal diffusion across the wall and heat capacity.

5.5 Simplified correlation of oscillatory flow CHF

As mentioned above, CHF is given by equation(9) based on the lumped-parameter assumption for wall heat capacity. Consider a critical mass flux G_{cr} just as the vapor quality becomes unity at the tube under a specified heat flux condition. Then, the tube wall becomes heated up during the period τ_D of instantaneous mass flux less than this critical value, so that the CHF is brought about. Assuming thermodynamic equilibrium, and quasi-steady state, the critical mass flux G_{cr} is determined by

$$G_{cr} = 4Lq_{cr} / [(h_{LG} + \Delta h_{sub}) D_i] \tag{19}$$

Provided a sinusoidal mass flux oscillation, the dryout period in this case is given by

$$\tau_D = 0.5\tau\{1 + (2/\pi)\sin^{-1}[(G_{cr} - G_0)/\Delta G]\} \tag{20}$$

Limited to the dryout at high quality, the steady flow CHF is roughly expressed by a simple function of the heat transfer coefficient defined for saturated vapor flow at the time-averaged mass flux G_0 and the critical wall-superheat ΔT_{cr} [45],

$$q_{cr0} \propto \alpha_0 \Delta T_{cr} \tag{21}$$

Equation (9) is normalized by this steady relationship (21). Then, we have

$$q_{cr} / q_{cr0} = C / [1 - \exp(-\tau_D / \tau_W)] \tag{22}$$

This equation implies that the CHF is correlated with a function of dryout period relative to the time constant of the wall, τ_D / τ_W. Experimental results of high-quality CHF, normalized by the steady CHF data in experiments, are plotted as a function of τ_D/τ_W in Fig. 16. Normalized CHF decreases, first, drastically, and then tends to saturate. Equation (22) with $C=0.15$, 0.2 and 0.3 are shown by blue curves. The empirical constant C is within an experimentally determined values, when α_0 is given by the Dittus-Boelter equation and critical wall superheat $\Delta T_{cr} = 100\,\mathrm{K}$. The plotted data is, at low τ_D/τ_W, close to the curve of $C=0.3$, while approaches to the curve of $C=0.15$ with an increase in τ_D/τ_W. Considering that the constant C is a correction factor for heat transfer, the time constant τ_W, being a function of heat transfer coefficient, needs also correction. Then, we modify equation(22) to

$$q_{cr}/q_{cr0} = C_1/[1-\exp(-C_2\tau_D/\tau_W)] \qquad (23)$$

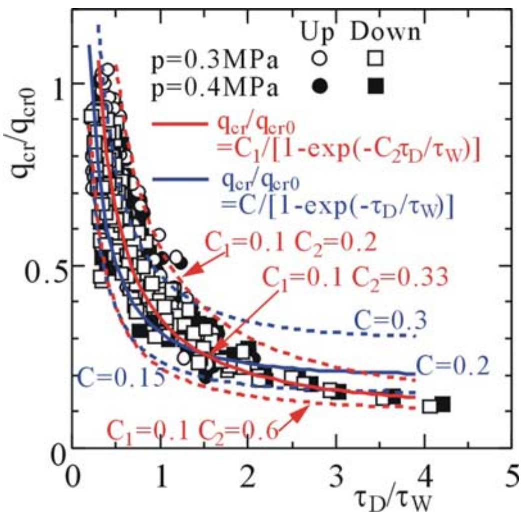


Fig. 16 Scaling of CHF at high quality.

Most of the data is within two red curves with $C_1=0.1, C_2=0.2$ and $C_1=0.1, C_2=0.6$, while averaged trend is well expressed with $C_1=0.1, C_2=0.33$. Although thick-walled tube data in the range $\tau/\tau_W > 0.8$ (corresponding to D_i =3 mm, D_o =9 mm, $G_0 \le 600$ kg/m^{2}s) is excluded, other whole data obtained in the experiments are plotted in Fig. 16, including data of D_i =3.0 mm, D_o =4.0, 6.0 mm, D_i =4.0 mm, D_o =4.5 mm, G_0 =60-1800 kg/m^{2}s, $\Delta G/G_0$ =0.0-10.2, p =0.3, 0.4 MPa [45].

5.6 CHF at very low velocity

When the mass flux is very low, 38.3 kg/m^{2}s, being a similar level to the above-mentioned compact water-tube boiler, the CHF data obtained in the experiment with a test tube of 15 mm I.D. and 16 mm O.D. are plotted in Fig. 17. The data contains both vertical upward and downward flow. The general trends are the same as those of small bore tubes, while the CHF in upward flow is lower than that in the downward flow, at such very low mass flux condition. This tendency becomes reversed at slightly higher mass flux, 82.1 kg/m^{2}s, in Fig. 18, with a larger dispersion of data than that in Fig. 17. Such a feature is mainly due to the flow pattern. Flow pattern in the downward flow with very low velocity is easy to become annular form - i.e. wall is well wetted by liquid film. On the other hand in the upward flow, the flow may be highly agitated so that the liquid film breakdown easily takes place. It should be further noted about these data that the CHF is first observed at the exit at very low mass flux in Fig. 17, while the downward-flow data in Fig. 18 involve both upstream and downstream CHFs, where “upstream CHF” means CHF in the middle along the test tube, and “downstream CHF” initiates at the tube exit.

CHF data of downward flow are plotted as a function of time-averaged mass flux in Fig. 19 with an indication of the location of CHF. At very low mass flux, the CHF is almost included in the category “downstream CHF”, and the data are limited at relatively high exit quality. With an increase in the time-averaged mass flux, the dispersion of data, including steady and oscillatory condition, becomes large. The plotted data seems to have lower and upper limits bounded by dashed curves, typically 250 and 550 kW/m^2, respectively, in the range of mass flux from 50 to 120 kg/m^{2}s. “Steady flow” in downward flow is no more than the flow condition without imposing forced flow oscillation at the inlet. The flow in boiling or two-phase region may suffer from above-mentioned phase separation, bubble stagnation and fluctuation caused by buoyancy force counteractive to the downward flow of liquid. Even if a sophisticated mechanistic model is constructed, the verification of the model is generally rather hard - neutron radiography may have a possibility [46]. In viewing whole data, however, the situation is not so pessimistic, and we notice, at least, the CHF to be limited within the bounded area indicated by dashed lines, even under fluctuating conditions. “Not seeing the forest for the trees" is an important strategy at the present stage of understanding.

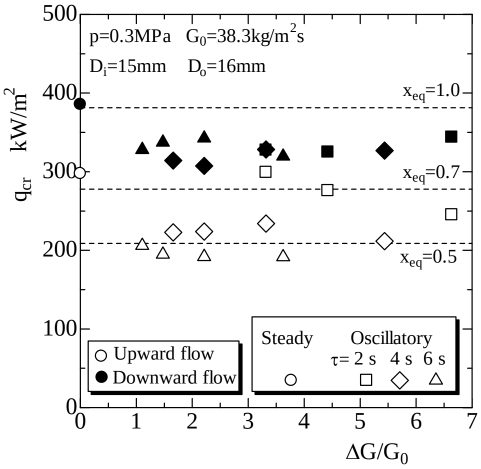


Fig. 17 Oscillatory flow CHF at low mass flux.

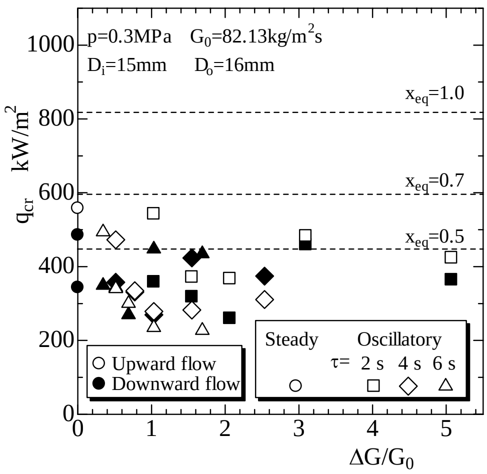


Fig. 18 Oscillatory flow CHF at low mass flux.

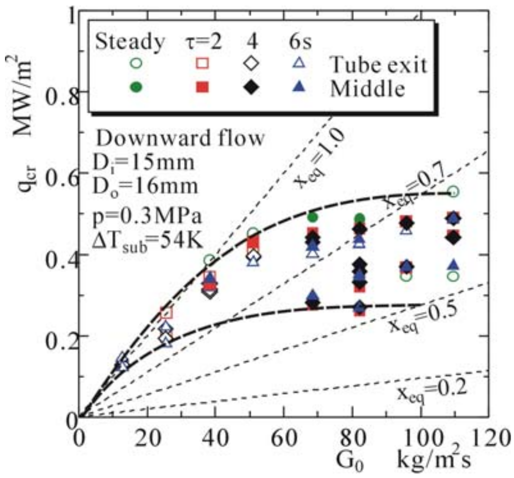


Fig. 19 CHF pattern map classified by the axial position of occurrence.

5.7 Stability of inverted annular flow and quenching

Gravity-feed emergency core cooling [47] is one of the important safety systems for passive-safety nuclear reactor. Related to this subject, Erbacher and Wiehr [48] have clamed that the flow oscillation deteriorated cooling and thus delayed quenching. When subcooled liquid is supplied into a channel at very high temperature, so-called inverted annular flow appeared first, and then flow pattern changes through agitated regime to dispersed flow in accordance with cooling, which is followed by partial or temporal quenching, and finally totally quenched state arrives. Such gravity-feed cooling phenomena were investigated with a liquid-nitrogen natural circulation system [49, 50].

The experimental apparatus consisted of head column of liquid nitrogen, downcomer, flow metering system with turbine flowmeter and heated section. The test section of D_i =5.0 mm, D_o =6.0 mm and 900 mm in heated length was initially heated up to a predetermined temperature, and then liquid nitrogen was allowed to enter the test section. Typical traces of wall temperature and inlet velocity are exemplified in Fig. 20. After the initial fluctuation followed by abrupt opening the valve of the downcomer, the flow rate decreased gradually until 100s, and then density wave oscillation was initiated. This density wave grew up quickly to become limit cycle oscillation.

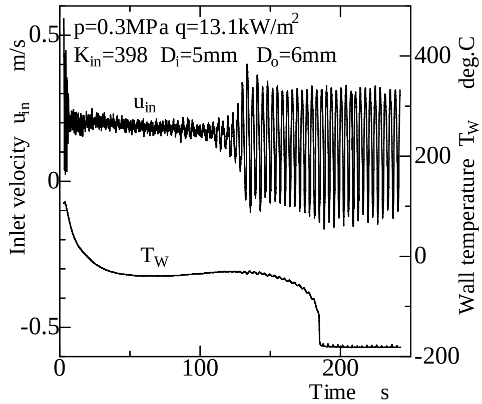


Fig. 20 Inlet velocity and wall temperature traces during quenching process (experiment with liquid nitrogen).

During the course of such transient, the wall temperature had a plateau after the initial decrease, and then got down significantly to nearly saturation temperature. The first stage of temperature transient roughly corresponds to inverted annular flow, the drastic decrease followed by after the plateau to transition boiling, and final stage of almost constant temperature to nucleate boiling - i.e. quenched state.

Figure 21 demonstrates the effect of flow oscillation on the quench time. The black curve with lower inlet restriction K_{in} is the case of quenching with density wave oscillation, while that of higher value of K_{in} (red curve) corresponds to non-oscillating condition. Until about 100 s, the temperature traces are smooth in both of the traces, while temperature oscillation, induced by density wave oscillation, appears in the case with smaller value of K_{in}. The quench time without oscillation (red curve) is definitely short in comparison with that of oscillatory flow. This means flow oscillation deteriorates the cooling, being quite similar phenomena to oscillatory flow CHF. In any case, heated or unheated wall at high temperature is subjected to a sufficient amount of liquid during a half or less than half period during the flow oscillation, while the wall is subjected to very low heat transfer during the another half period of flow oscillation.

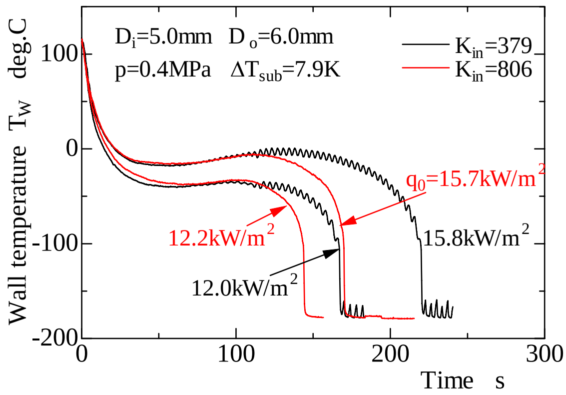


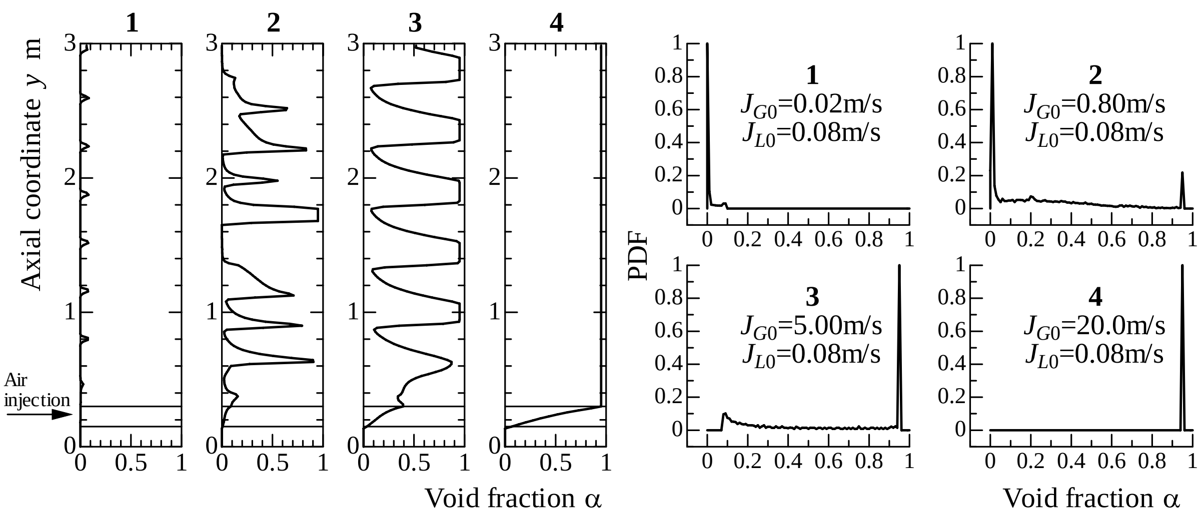
Fig. 21 Effect of flow stability on quenching process.

6 Future problem

Generally, downward flow boiling is not appropriate for industrial boiling channel system. There is, however, the possibility of appearance of downward boiling flow, having the form of flow mal-distribution in parallel-channel boiling system, depending on the tube arrangement and heat flux distribution. Widely spread feature is not limited in downward flow, but is often encountered in horizontal or nearly horizontal boiling channels [51], also being not appropriate for boiling system. Horizontal serpentine tubes of exhaust-gas boiler of Diesel engine is a typical example, which suffers from CHF and physical burnout, if sufficient amount of water is not circulated on the occasion of so-called soot fire, being often encountered in startup or shut down processes of engines.

In both of downward and horizontal tubes, bubble stagnation may trigger the CHF at low vapor quality, but with a variety of feature - everything is possible in such a system, and thus modeling or formulation of the thermal flow field is really a hard task. One promising approach is proposed by Ito et al. [52, 53] and Ozawa et al. [54-57]. The detailed discussion on this subject is beyond the present paper, while only a brief and typical example is described here.

As has been discussed above, the CHF problem in oscillatory flow, and more broadly including transients in general, becomes simple in the sense that two-phase flow dynamics of a large scale dominates the thermal



(1) Snap shot (2) PDF

Fig.22 Void fraction fluctuation by discrete bubble mode and resulting probability density function.

flow field rather than detailed structure. In other words, sophisticated two-phase flow and heat transfer mechanisms plays a minor role under transients. This concept is supported by the fact that the oscillatory flow CHF at high quality is well simulated and formulated in rather simple manner. On the other hand in the downward flow at low mass flux, the well-suited simulation to high quality CHF is not a suitable prediction methodology any more. Well-known two-fluid or multi-fluid modeling does not work as well, because these modeling are based on "continuous flow hypothesis" [54].

The new approach, referred to as the discrete bubble model [52-57] in the framework of pattern dynamics approach by the authors, consists of void propagation equation with limited number of relationship of bubble movement, gas phase compressibility, and phase redistribution based on a frame of reference with length scale of tube diameter. Fig. 22 shows typical void fraction fluctuations and respective probability density functions (PDF). Then the flow regime map, identified based on the PDF patterns is exemplified in Fig.23. The void fraction fluctuation represents local and time dependent behavior of two-phase flow. By introducing such a modeling into the discussion of CHFs, we have a powerful tool for everything-happening region, e.g. downward flow at low mass flux.

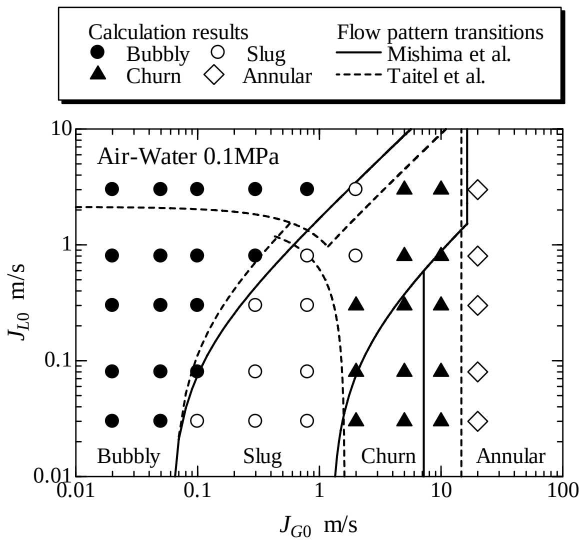


Fig.23 Flow pattern map.

7 Concluding remarks

CHF in well-designed and well-controlled boiling channel systems, such as nuclear reactor, has been well predicted so far with sufficient accuracy, while in a small-scale boiler and newly designed boiling channel systems often suffer from CHF problems. This is mainly because of the CHF researches based on steady state flow conditions. Under transient including startup, shutdown and flow instabilities, the CHF prediction is far below an acceptable level. The present discussion is to clam engineers' and/or designers' attention to such a problem and to enhance discussions for further development in CHF researches. In this paper, discussions were mainly focused on the most often encountered density wave oscillation. An overview of oscillatory flow CHF and new strategy for so far unsolved CHF problems were described.

Nomenclature

c	specific heat [J/kgK]
D_i, D_o	inner and outer diameter [m]
G	mass flux [kg/m^{2}s]
g	gravitational acceleration [m/s^2]
h	enthalpy [J/kg]
h_{LG}	latent heat of vaporization [J/kg]
J	Volumetric flux
k_W	thermal diffusivity of tube wall [m^2/s]
L	heated length [m]
p	pressure [Pa]
q	heat flux at inner wall [W/m^2]
q_0	heat generation rate per unit inner surface area [W/m^2]
q_W	volumetric heat generation rate in tube wall [W/m^3]
T	temperature [deg.C, K]
t	time [s]
u	velocity [m/s]
v	specific volume [m^3/kg]
v_{LG}	specific volume difference between saturated vapor and liquid [m^3/kg]
z	axial coordinate [m]

Greek symbol

α	heat transfer coefficient [W/m^{2}K], void fraction [-]
Δh_{sub}	subcooled enthalpy [J/kg]
ρ	density [kg/m^3]
σ	surface tension [N/m].
τ	oscillation period [s]
τ_D	dryout period [s]
τ_W	time constant of wall heat capacity [s]
ω	angular frequency [1/s]

Suffix

0	steady state value
cr	critical condition
ex	exit
f	fluid
G	vapor
in	inlet
L	liquid
m	mean value
S	saturation condition
W	wall

Acknowledgement

This research was supported in part by the Kansai University Research Grants: Grant-in-Aid for Joint Research, 2005. The present authors wish to express their sincere thanks to those who have supported present investigation. Thanks are extended to Prof. Emeritus K. Mishima, Kyoto University, and Prof. Emeritus K. Akagawa, Kobe University, for their permission to use their figures. The main body of this paper was presented at the ECI International Conference on Boiling Heat Transfer, Spoleto, 7-12 May 2006.

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