Available online at www.sciencedirect.com
SCIENCE DIRECT®

PERGAMON

International Journal of Heat and Mass Transfer 46 (2003) 3143–3152

International Journal of HEAT and MASS TRANSFER

www.elsevier.com/locate/ijhmt

An investigation of dryout/rewetting in subcooled two-phase flow boiling

R.C. Jones, R.L. Judd *

Department of Mechanical Engineering, McMaster University, Hamilton, Ont., Canada L8S 4L7

Received 18 October 2002; received in revised form 15 February 2003

Abstract

An investigation of a flow reversal that was observed to occur during narrow channel flow boiling is reported in this paper. The investigation was undertaken to determine the conditions under which the flow reversal occurred and to determine its cause. High-speed photography was used to study the sequence of events that occurred during each cycle of the flow reversal.

Two-phase flow instability models involving boiling crisis were examined and correlations for the occurrence of boiling crisis based on the flow conditions and the test section geometry were tested with the experimental data. Strong agreement was found between the onset of the flow reversal and the prediction of CHF under subcooled flow boiling conditions as well as between the predicted rewetting times and the period of the flow reversal. As a result, the instability was deemed to be caused by the onset of CHF and to be the result of dryout and rewetting of the heated surface.

1. Introduction

During an investigation into the effect of channel width on flow boiling heat transfer conducted in the summer of 1997, a flow instability was observed. When water flowed upwards at a subcooling of 5 °C across a 25 mm diameter circular heated surface through a 3 mm wide vertical channel, a localized reversal of flow was observed to occur at a velocity of 19 cm/s and a heat flux of 696 kW/m^2. Additional testing determined that similar flow reversals occurred in 1 and 5 mm wide vertical channels as well at different combinations of flow velocity and heat flux. This observation prompted the research investigation reported in this paper.

The purpose of this research investigation was to examine the conditions required to establish the onset of the two-phase flow reversal described above in order to provide an explanation for the flow reversal and its potential impact on the design of heat transfer equipment operating under similar conditions. A set of three research goals were identified:

- *To determine the limits of the observed instability*. Instabilities were observed over a wide range of flow conditions. The research apparatus was designed with the ability to control the levels of mass flux, subcooling, channel width, heat flux, and pressure drop in order to establish the range of conditions under which the flow reversals occurred.
- *To determine the type of instability under observation*. Due to the wide variety of flow instabilities that have been identified in the literature as well as the peculiarities of the flow loop, it was important to determine whether the flow reversal was hydrodynamic or thermodynamic in nature in order to determine why the flow reversal observed in narrow channel flow boiling occurred.
- *To derive the knowledge by which the onset of the flow reversal could be predicted*. Although knowledge of the type of instability and the influence of the test section and heater geometry on the onset of the flow

* Corresponding author. Tel.: +1-905-525-9140; fax: +1-905-572-7944.
E-mail address: juddr@mcmaster.ca (R.L. Judd).

0017-9310/03/$ - see front matter © 2003 Elsevier Science Ltd. All rights reserved.
doi:10.1016/S0017-9310(03)00104-2

Nomenclature

D	heater diameter (m)
f	frequency (Hz)
G	liquid mass velocity (kg/m^2 s)
g	acceleration due to gravity (m/s^2)
h	enthalpy (kJ/kg)
L	length in flow direction (m)
m	mass flow rate (kg/s)
q	power (W)
q''	heat flux (kW/m^2)
T	temperature (°C)
t	time (s)
U	local liquid velocity parallel to the heated surface (m/s)
V	velocity (m/s)
v	bubble volumetric growth rate (m^3/s)

Greek symbols

δ	channel width (m)
θ_{sub}	liquid subcooling (°C)
λ	wavelength (m)
ν	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
π	pi (–)
σ	surface tension (N/m)
T	period (s)

Subscripts and superscripts

c	critical thickness of liquid film (–)
CHF	critical heat flux (–)
d	hovering period of bubble (–)
l	liquid phase (–)
max	maximum heat flux (–)
nsv	net significant void (–)
v	vapor phase (–)

reversal is useful, it was felt to be necessary to develop a set of predictive relationships to determine the conditions under which the instability might appear, in order to predict how heat transfer systems could be designed to avoid the unexpected onset of the two-phase flow instability.

The overall goal of the investigation was to examine the flow reversal and provide a comprehensive study of its causes and effects, with the intent of improving the understanding of flow reversals in subcooled flow boiling through narrow channels.

2. Experimental apparatus

The test facility used for the study was comprised of two major components: the flow loop, and the test section. The flow loop was comprised of several lesser components, which will be described separately. Schematic diagrams of the flow loop and test section are provided in Figs. 1 and 2.

The pump used was a G&L Goulds Pump Company centrifugal pump capable of circulating a maximum of 60 l/min of water. During the operation of the flow loop, the pump circulated water continuously throughout the

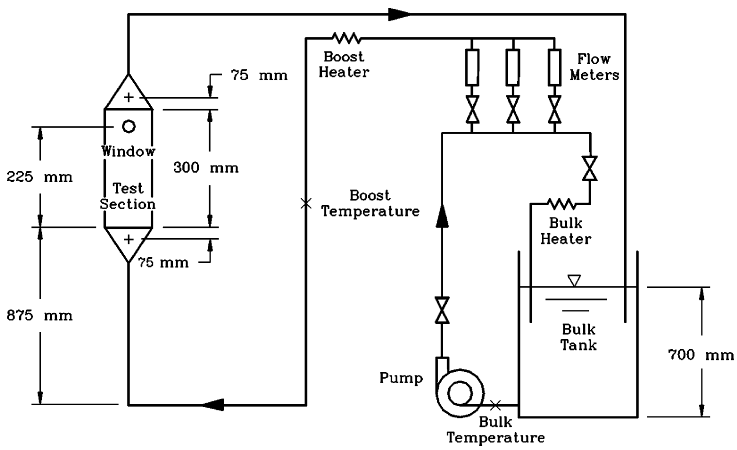

Fig. 1. Schematic flow diagram.

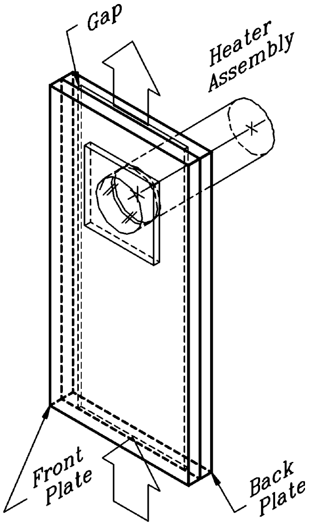

Fig. 2. Test section.

system. From the pump, the water was recirculated back to the tank, or diverted through one of the three flowmeters. Specific flow rates were obtained by adjusting a valve placed directly upstream of the flowmeter and a second valve ahead of the return pipe to the bulk tank. The maximum velocities attainable in the test section varied with channel width. For the 1, 3 and 5 mm wide channels, these values were estimated to be approximately 3.0, 1.0 and 0.6 m/s respectively.

Due to the large range of flow rates that were required for the investigation, the system was designed to utilize different flowmeters with overlapping ranges. Each of the flowmeters was connected to the flow loop by means of independent valves so that meters not in use during a specific test could be isolated.

Two heaters were used to control the temperature of the water in the system. One heater, hereafter referred to as the bulk heater, was located directly after the recirculation valve to the bulk tank. The second heater, hereafter referred to as the boost heater, was located directly after the flowmeters, in the pipe leading to the test section. The heat output for both heaters was controlled by means of two Technipower variac autotransformers.

Fluid temperature was measured at two locations, at the outlet of the bulk tank and immediately downstream of the boost heater. These temperatures were measured with type E chromel-constantan thermocouples that were connected to a data acquisition system consisting of a Data Translations 2831 board modified with thermocouple attachments and read by LabView software for use in the control system. These thermocouples were determined to be accurate to within 0.5 °C by calibration.

Pressure ports were located 7.5 cm above and below the test section. A Validine wet/dry differential strain gauge pressure transducer capable of measuring pressures ranging from 0 to 95 kPa was connected to them. In order to examine the pressure variations produced by the flow reversal, the output signal from the transducer was analyzed by a HP dynamic signal analyzer which enabled the magnitude and frequency of the pressure oscillations to be measured.

The test section consisted of a set of two aluminum plates and a heater assembly. A single glass viewing window of 5 cm $\times$ 5 cm square with a thickness of 1.25 cm was inserted in the front plate. The copper heater assembly was mounted in the back plate opposite the window. These plates were held at widths of 1, 3, or 5 mm using various sets of spacers, which were welded into position.

The viewing window was installed in an appropriately sized stepped hole in the front plate, which allowed for the use of a capping plate, combined with silicon sealant, to create an effective seal.

The heater assembly was sealed into the test sections flush with the aluminum plate using a cap plate and an O-ring to form a pressure fit with the back plate, allowing the working fluid to contact the copper surface. Using this method, the heater assembly could be transferred from one test section to another, eliminating any effect that aging might have had as a significant source of error.

The heater assembly mounted in the back plate of the test section was manufactured from a 99.995% pure copper cylinder mounted inside a stainless steel sleeve which afforded thermal insulation in the radial direction because of an air gap between the copper cylinder and the stainless steel sleeve. These two pieces were permanently attached by means of electron beam welding. The heater surface was finished with a polishing compound containing particles no larger than 1 μm in size. It is believed that the copper surface was sufficiently aged so as to prevent any further change in boiling properties.

Three thermocouples were inserted along the centerline of the heater assembly, at depths of 1.65, 9.59, and 17.53 mm below the boiling surface. These thermocouples were also type E and were determined to be accurate to within 0.5 °C by calibration before insertion into the heater assembly. Although the temperature profile within the heater assembly was found to be somewhat non-linear, the deviation from linearity was small enough that it could be ignored in the determination of the surface temperatures by extrapolation of a best-fit relationship to the measured values. In addition, the deviation from linearity was considered to have little effect on the determination of the heat flux, which was done by multiplying the gradient obtained from the best-fit relationship, as depicted schematically in Fig. 3, by the thermal conductivity of pure copper.

Three Firecam cartridge heaters were inserted in the heater assembly, each heater capable of producing 200

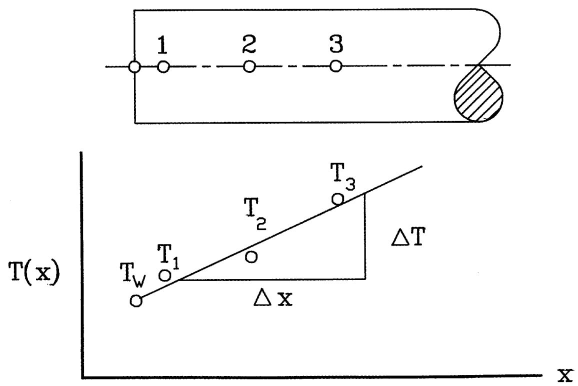

Fig. 3. Hypothetical temperature profile.

W. Power to these heaters was controlled by means of a Powerstat variable autotransformer. The electric power to the heaters was measured by means of a Conway Electronic Enterprises wattmeter, so as to provide confirmation of the heat flux calculated using the procedure described above.

3. Experimental procedure

For the purposes of this experiment, individual tests were conducted at subcoolings of 5, 10, and 15 °C. To obtain these subcoolings, the water in the system was heated from room temperature for approximately 3 h using both the bulk and boost heaters. Once the system had reached the desired level of subcooling, the power provided to the boost heater was adjusted to modify the temperature of the liquid at the heated surface. Both the bulk and boost temperatures were regulated by computer control, which was capable of holding the temperature of the water to within 0.5 °C of the desired subcooling for the duration of the tests. For each individual test, liquid subcooling, mass flux, pressure drop across the section, gap width, and heat flux were measured and recorded.

During the testing process, the occurrence of flow reversal was identified through a combination of visual evidence and frequency measurements. Once the system had stabilized at the desired flow rate and subcooling, the surface temperature and heat flux were adjusted by increasing the power input to the heater assembly in equal increments. For each increment, the system was allowed to reach steady state conditions. Once steady state operation had been attained, the temperature at the heating surface, the temperatures in the heater assembly, and the electric power were recorded. It should be noted that the temperatures and the power were not instantaneous measurements, but were weighted averages of the collected data, done so as to avoid small noise fluctuations in the thermocouple signals from significantly affecting the experimental values. In addition to recording the experimentally determined heat flux, the power provided to the heater assembly was measured by means of the wattmeter and recorded.

This process was then repeated until the maximum obtainable power to the heater assembly had been reached, or a full flow reversal had been observed. Whenever a flow reversal was observed, power to the heater assembly was reduced, and the system was allowed to return to steady state, at which point the temperature at the heated surface was measured to ensure that no significant change had occurred to the liquid subcooling during the experimental process.

A Kodak Ektapro camera operating at 500 frames/s was used to produce recordings of the phenomena occurring at the boiling surface at each power increment. In order to determine the magnitude and frequency of the pressure oscillations, the signal analyzer was used to determine the frequency and RMS magnitudes of the pressure signal associated with the flow reversal. To determine the point at which the flow reversal occurred, the frequency spectra were matched to the collected video film data. For experimental purposes, the flow reversal was judged to occur when a full reversal of flow was observed at the boiling surface, accompanied by a sudden shift in the measured frequency.

This method was used to avoid confusion related to the onset of significant void within the test section. The onset of significant void was judged to have occurred when a sudden increase in the magnitude of the measured RMS pressure was detected, typically accompanied by an observed frequency between 4 and 6 Hz. Upon the occurrence of the flow reversal, however, the emergence of a second frequency in the range of 7–8 Hz was observed, corresponding to the frequency of the full reversal of flow observed on the video. Fig. 4 depicts the variation of pressure drop across the test section for a typical set of experimental conditions and identifies the

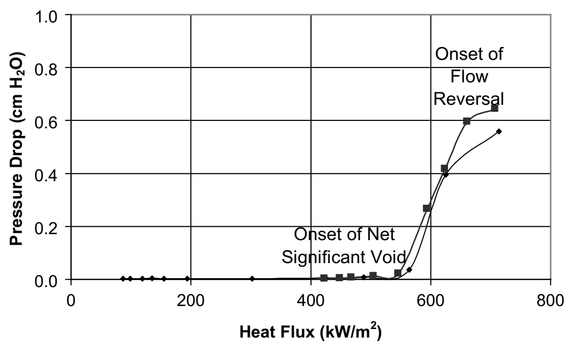

Fig. 4. Experimental pressure drop measurements: 1 mm gap, $V = 40$ cm/s.

onset of significant void and the onset of flow reversal. It should also be noted that the experimental results strongly indicate that the experimental data are highly repeatable.

4. The observed instability

As indicated previously, video recordings were made of the flow reversal using a high-speed camera operating at 500 frames/s. These recordings were then transferred to videotape for the purpose of further examination. Using these recordings, it was possible to examine the sequence of events that occurred at the heated surface during the flow reversal. These stages are outlined below. A visual record of these stages is provided for conditions in the 3 mm wide channel at 10 °C subcooling, a velocity of 17.5 m/s, and a heat flux of 914.4 W/m^2 in Fig. 5.

- *Surface quenching (A)*. During the preliminary stage of the flow reversal, boiling on the surface had been quenched by a liquid front. As a result, there was a minimal amount of boiling occurring at the heated surface, with only a few active boiling sites visible.
- *Initiation of boiling sites (B)*. After the quenching front had passed, boiling reinitiated across the heated surface, resulting in the formation of small vapor bubbles that rapidly departed from the surface. While some merging of bubbles occurred, the majority

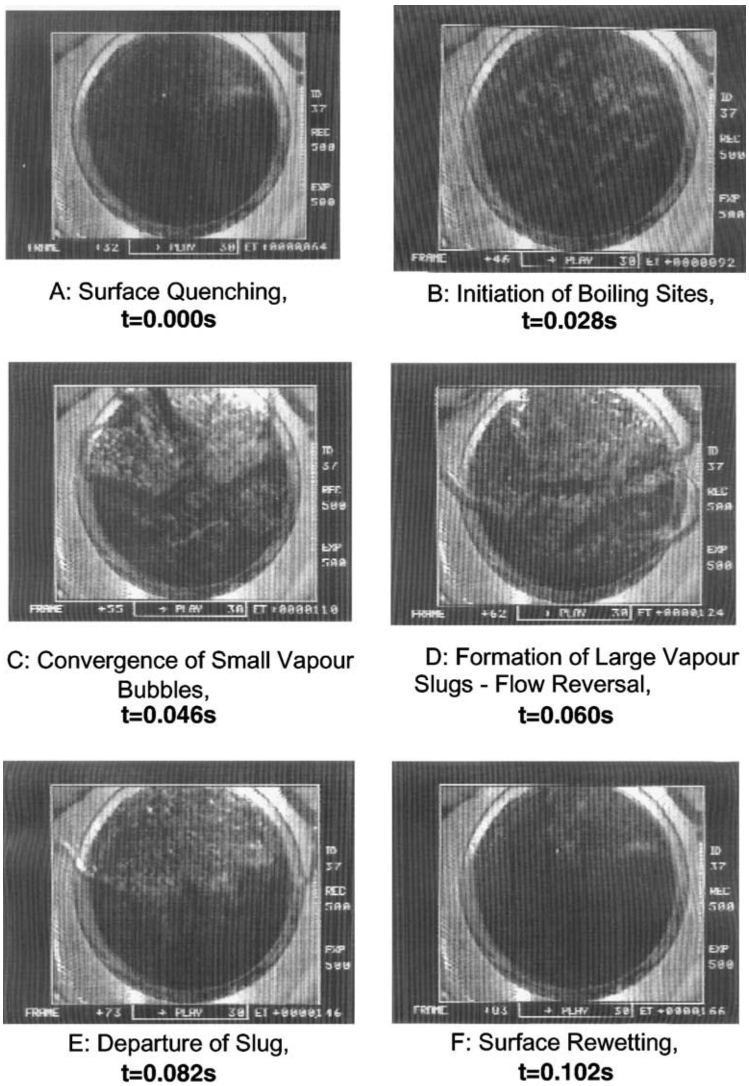

Fig. 5. Photographic images of the flow reversal.

quickly departed the surface and exited the test section.

- *Convergence of small vapor bubbles (C).* Boiling became more vigorous across the surface, resulting in an increase in the amount of vapor produced. Due to the increased amount of vapor present, the small bubbles began to interact and merge, forming larger bubbles which covered a significant amount of the surface. As the bubbles merged, the size of the merged bubbles exceeded the width of the channel.
- *Formation of large vapor slugs/flow reversal (D).* As the rate of boiling increased, the amount of vapor produced increased and the bubbles continued to merge until a large vapor slug was formed that covered the entire heated surface and filled the entire channel width. Under certain conditions, after covering the surface, the vapor slug continued to expand downwards past the lower limits of the heated surface. The video records indicated that in this phase of the flow reversal, the heated surface was undergoing dryout.
- *Departure of vapor slug (E).* At a critical point, it was speculated that the pressure increase in the test section due to buoyancy was able to overcome the surface tension holding the vapor slug, and the slug was torn away from the heated surface and forced upwards through the channel. As the slug departed, the number of active boiling sites decreased as subcooled liquid contacted the boiling surface.
- *Surface rewetting (F).* As the vapor slug departed from the heated surface, subcooled liquid rewet the heated surface, and quenched the boiling at the nucleation sites. At this point, the boiling activity occurring in the test section was again reduced to a few visible boiling sites.

5. Preliminary analysis of experimental results

In order to continue the investigation into the flow reversal observed, it was necessary to define a testing domain. This domain was required to encompass as wide a selection of flow conditions as possible in order to obtain enough information to produce an accurate representation of the conditions under which the flow reversal occurred.

To define the testing domain, three parameters were considered: pressure drop across the test section, mass flow rate, and subcooling. Pressure drop is a significant parameter in many hydrodynamic instabilities, mass flow rate is important in both thermodynamic and hydrodynamic instabilities, and subcooling assists in defining the heat input required to initiate the flow reversal. Channel width was also varied so as to determine the impact of the size of the region for bubble expansion within the test section, and to allow for the comparison of mass flow rate effects within a particular channel width and across multiple channel widths.

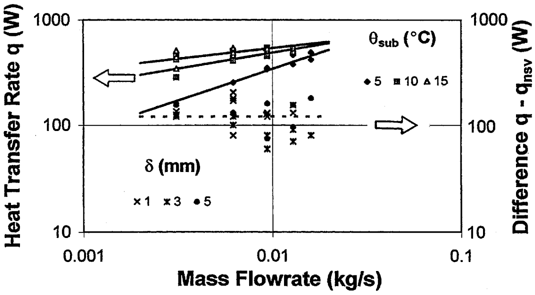

Fig. 6. Analysis of experimental results.

Using this testing domain, a series of onset of flow reversal points were obtained across multiple channel widths and liquid subcoolings. The conditions under which flow reversals occurred were then examined to determine trends and relationships that might suggest the cause of the observed flow reversal.

A graph was produced in which the heat transfer rate at the onset of the flow reversal was plotted as a function of mass flow rate, as shown in Fig. 6. The use of mass flow rate as the independent variable appears to have collapsed the data towards a series of curves independent of channel width. However, the data does not collapse into a single curve, but forms three separate curves dependent on the subcooling at the heating surface. It should be noted that each of these curves appears to be approaching a maximum heat transfer rate at which flow reversal will occur.

Fig. 6 also presents the difference between the heat transfer rate required to initiate the flow reversal, q, and the heat transfer rate required for the onset of net significant void, q_{nsv}, as a function of the mass flow rate across the heating surface. It can be seen that this treatment of the data collapses the results towards a single curve that appears to be largely independent of the mass flow rate, subcooling and channel width.

The reduction of the data towards a single curve is of particular interest as the invariant difference between the rate of heat transfer required to initiate the flow reversal and the rate of heat transfer required at the onset of net significant void suggests that the only factor required to initiate the flow reversal is the amount of heat added after the liquid has attained saturation conditions. A more detailed explanation of this conclusion will be presented in the following section.

6. Investigation of the flow reversal

The study of flow instabilities was introduced by Ledinegg [1] with the observation that under certain conditions, steady state pressure drop may decrease with increasing flow rate. This instability, referred to as the Ledinegg instability, is characterized by a sudden change in the flow in a boiling channel, which may result in a lower flow rate. Since Ledinegg's observation, interest in two-phase flow instabilities has increased greatly, because of significant applications in nuclear and steam generators, resulting in the determination of different categories of instabilities based on the causal mechanisms. The investigation of the limits of stability for each category, and the modifications that can be made to eliminate the occurrence of the instability or minimize the harmful effects, which can take the form of mechanical vibration, system control problems, or the general disruption of the heat transfer process, has been the focus on intense investigation. For the purpose of this investigation, however, only instabilities resulting from the occurrence of boiling crisis will be examined in greater detail, as the conclusion above involving the nature of the flow reversal suggests that the flow reversal is driven by a thermodynamic mechanism.

Boiling crisis is caused by a sudden change in the heat transfer mechanism, such that the heat cannot be removed from the boiling surface at a sufficient rate for stable operation. Boiling crisis is characterized by a rapid increase in wall temperature accompanied by the occurrence of a flow oscillation. These conditions have been observed for a wide range of working fluids, heater geometries, and flow conditions.

The occurrence of boiling crisis in conjunction with flow oscillations has been noted by Mathisen [2] and Dean et al. [3]. It has been proposed that this oscillation is the result of boundary layer separation during the boiling crisis creating a stagnant flow volume at the heated surface, according to Tong [4], although there is a relative lack of conclusive experimental evidence. An investigation into boiling crisis in pool boiling determined that the occurrence of boiling crisis could be linked to the intermittent behavior of the vapor masses produced and the consumption of the vapor layer across the boiling surface, according to Katto and Yokoya [5]. Similarly, the onset of boiling crisis has been linked to a shift in the boiling regime towards the conditions observed in the onset of CHF at the heated surface.

Investigations into the occurrence of CHF have been conducted under a wide variety of flow conditions and system geometries with the intent of developing theoretical and empirical models capable of predicting the conditions required for the onset of the flow reversal. Initial models represented attempts to extend the existing knowledge of CHF in pool boiling to flow boiling conditions through empirical relationships, such as those presented by Bergles and Rohsenow [6].

The prediction of CHF in flow boiling has been achieved for saturated flow conditions, resulting in the development of correlations dictating the impact of specific geometries. An initial theoretical model for determining CHF under saturated conditions was proposed by Haramura and Katto [7] for parallel flow over a flat plate. Similar work has been performed by Katto and Kurata [8]. Using dimensionless groups similar to those used by Haramura and Katto, Yagov and Puzin [9] were able to develop an empirical correlation for the prediction of CHF for parallel saturated flow over a circular boiling surface.

Two-phase flow in narrow passages has also been studied by many investigators, with the intent of determining the influence of the passage dimensions on the onset of CHF for applications typically involving the cooling of computer chips or accident control in nuclear reactors. A study performed by Mishima and Nishihara [10] investigated the impact of flow direction and magnitude on the onset of CHF under low pressure conditions for channels with hydraulic diameters from 4 to 30.5 mm. Results from the study indicated that under certain conditions, CHF could occur significantly below the level predicted by conventional correlations. Similar work into the influence of flow direction on the onset of CHF in narrow channels was performed by Kaminaga et al. [11] for a non-uniform heat flux. Similarly, it was observed by Celata et al. [12] that in circular channels with diameters under 15 mm, at high heat fluxes and flow velocities, the onset of CHF may become very high. It was proposed that under these conditions, CHF is inversely proportionate to the channel diameter. In considering the effect of a narrow channel on the onset of CHF in subcooled flow boiling, Mudawwar and Maddox [13] extended their model to include the effect of the velocity profile on the ability of the incoming flow to penetrate the liquid sublayer. Investigations into the effects of channel diameter on the occurrence of CHF were also performed by Celata et al. [14,15] in which empirical relationships between CHF in narrow channels and unrestrained flow were recommended.

Based on a recent investigation into the present state of CHF research by Weisman [16], theoretical research into the onset of CHF in flow boiling is focusing on more exotic heater geometries and more extreme flow conditions and flow patterns, often involving bubble modeling.

After examining the general literature on the occurrence of CHF and considering the flow conditions and the geometry of the test section under investigation, two independently developed correlations for the onset of CHF were selected for further examination.

7. Haramura and Katto correlation

The correlation developed by Haramura and Katto [7] was designed for use in determining CHF in forced convective boiling over a flat plate subjected to parallel flow. For the occurrence of CHF on a flat plate, it was

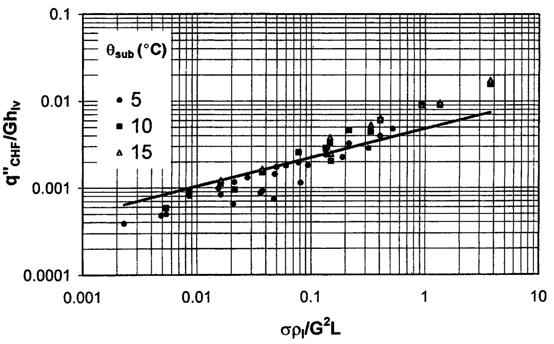

Fig. 7. Haramura and Katto correlation.

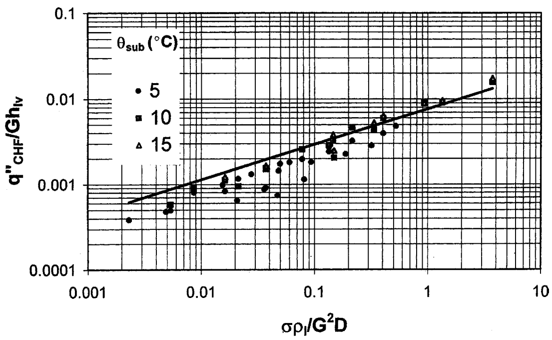

Fig. 8. Yagov and Puzin correlation.

assumed that the heat being transferred from the heated surface balanced the latent heat required to evaporate the liquid flowing into the liquid film, yielding the following energy balance:

$$q''_{\max} = \rho_l \delta_c U h_{lv}/L \tag{1}$$

where δ_c is the liquid film layer thickness defined as

$$\delta_c (q''/h_{lv})^2/(\sigma\rho_{lv}) = 0.00536(1 + \rho_v/\rho_l)(\rho_v/\rho_l)^{0.4} \tag{2}$$

Substitution of the sublayer thickness into the energy balance at the heated surface yields a prediction for the onset of CHF:

$$q''_{\max}/Gh_{lv} = 0.175(\rho_v/\rho_l)^{0.467}(\sigma\rho_l/G^2L)^{1/3} \tag{3}$$

Using this correlation, the experimental data obtained during the present study was used to produce Fig. 7.

With respect to comparison of the collected data and the correlation, it was observed that the data collected matches the expected slope. There is some separation of the data into separate bands dependent on subcooling, which was not accounted for in this model, in as much as it assumes all liquid to be at saturated conditions. Although the correlation developed by Haramura and Katto provides good agreement with the experimental data, the geometry, a heated square is somewhat different than the geometry investigated in the present experiment.

8. Yagov and Puzin correlation

A correlation similar to that developed by Haramura and Katto was proposed by Yagov and Puzin [9]. The test facility used to obtain the experimental data consisted of a cylindrical bar inserted flush with the surface of a flat plate, designed to examine the occurrence of CHF on a circular boiling surface under forced convective boiling.

Similar to the present research, the temperature gradient in the heater assembly was used to determine the heat flux being applied to the boiling surface. For the purposes of developing the correlation, however, Yagov and Puzin determined the onset of CHF through the occurrence of a sudden increase in the temperature of the heater assembly, rather than through the visual observation of a flow reversal at the heated surface. For convenience and consistency, non-dimensional groupings similar to those proposed by Haramura and Katto were used to develop the following correlation for the onset of CHF:

$$q''_{\max}/Gh_{lv} = 0.66(\rho_v/\rho_l)^{0.604}(\sigma\rho_l/G^2D)^{0.415} \tag{4}$$

Considering the similarities of the geometries used with respect to the test sections and heated surfaces, it is not unexpected that this correlation would produce a strong match with the experimental results. The experimental results collected during the present study were plotted against the correlation, as seen in Fig. 8. It is noted that for all channel widths, the experimental results showed good agreement at all subcoolings. Additionally, the slope of the experimental data at each subcooling matched that of the correlation, with the data closely surrounding the correlation, strongly suggesting that the flow reversal was the result of the onset of CHF over a circular heated surface.

9. Frequency of the flow reversal

In addition to providing an accurate predictor for the required heat flux at the onset of the flow reversal, the implication of the occurrence of instability as a result of achieving dryout at the heated surface also provides some insight into the observed instability frequency.

During the occurrence of the flow reversal, both visual observation and vibrational analysis indicated that the onset of the flow reversal was accompanied by a sudden shift in measured frequency to a value within the range of 7–9 Hz. In order to evaluate the accuracy of the measured frequency of the instability, a prediction was made of the time required for the rewetted heated surface to reach CHF, using the model proposed by Haramura and Katto [7]:

$$\tau_d = (3/4\pi)^{1/5}[4(\xi\rho_l - \rho_v)/g(\rho_l - \rho_v)]^{3/5} v_l^{1/5}, \quad \xi = 11/16 \tag{5}$$

where

$$v_l = \lambda_d^2 q''/\rho_v h_{lv} \tag{6}$$

and

$$\lambda_d = 2\pi 3^{1/2}[\sigma/g(\rho_l - \rho_v)]^{1/2} \tag{7}$$

Using the known conditions of the flow, the time required for the heated surface to reach CHF was calculated and compiled in tabular form. All experimental values were obtained at a subcooling of 10 °C and a channel width of 3 mm (Table 1).

Although the periods presented above result in a frequency that exceeds the oscillation frequency observed during the experimental process, it should be noted that achieving CHF is only a part of the observed flow reversal cycle. When combined with the time interval required for the heated surface to rewet and boiling to reinitiate, the overall time period produces an estimated oscillation frequency consistent with the experimentally observed frequency. A schematic diagram showing the duration of the various components of the dryout/rewetting cycle is presented in Fig. 9.

This figure shows a hypothetical cycle for the flow reversal. The cycle is considered to begin with a subcooled surface upon which no nucleation sites are active. It is assumed that the local flow rate across the heated surface matches the bulk flow rate for the system. As the boiling at the heated surface increases, nucleation sites begin to activate. This site activation is considered the beginning of the period required for the onset of CHF. The bubbles begin to merge and the flow rate across the surface decreases until the direction of flow reverses, at which point CHF is judged to occur, which is considered to be the ending of the period required for the onset of CHF. After the flow reversal, the vapor slug moves away from the heated surface, and subcooled liquid contacts the surface once again, ending the oscillation cycle, and returning the system to its original state. The estimated period for a complete cycle of the flow reversal, 132 ms, was determined from visual records of the instability. As can be seen, the observed oscillation period encompasses the period of time required for the heater surface to reach CHF, while still providing time for the heating surface to rewet.

Based on the agreement between the experimentally observed frequency and the frequency estimated through theoretical means assuming that the instability is caused by the onset of CHF in the test section, it is proposed that the observed instability is the result of the onset of CHF.

Table 1
Predicted period for onset of CHF

m (kg/s)	0.0031	0.0062	0.0094	0.0129
q''_{CHF} (kW/m^2)	829	809	898	977
τ_d (ms)	75.2	74.8	76.4	77.7
f (Hz)	13.3	13.4	13.1	12.9

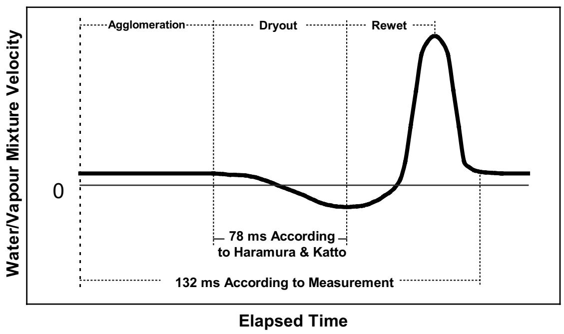

Fig. 9. Hypothetical variation of water/vapour mixture velocity with time.

10. Conclusions

An investigation into the occurrence of a previously little known flow reversal was performed in order to determine the nature of the instability being observed and the conditions required for it to occur.

Based on the analysis of the experimental results, it was determined that the flow reversal observed was due to the onset of CHF. In consideration that the flow reversal might be the result of the onset of CHF, two correlations were examined. A plot of the experimental data against the correlation produced by Haramura and Katto for saturated flow boiling over a square heated surface produced a strong agreement for the experimental results, as well as a match for the slope of both the experimental data and theory. A similar result was obtained using a correlation for saturated flow boiling over a circular surface, as proposed by Yagov and Puzin. Estimates of the time required for the development of CHF from ONB supported the frequency of the flow reversal observed.

Based on the experimental investigation, the following conclusions were drawn:

- The flow reversal was caused by the onset of CHF at the heated surface.
- The onset of the reversal is suppressed by an increase in mass flow rate, or an increase in the liquid subcooling.
- The onset of the flow reversal can be predicted through the correlation proposed by Yagov and

Puzin for saturated flow boiling across a circular heated surface.

References

[1] M. Ledinegg, Die Warme 61 (1938) 891.
[2] P.P. Mathisen, Out of pile channel instability in the LoopSkaluan, in: Proc. Symp. on Two Phase Dynamics, Eindhoven, 1967.
[3] R.A. Dean, R.S. Dougall, L.S. Tong, Effect of vapor injection on critical heat flux in a subcooled R-113 (Freon) flow, in: Int. Symp. Two Phase Flow Systems, Israel, 1971.
[4] L.S. Tong, Boundary-layer analysis of the flow boiling crisis, Int. J. Heat Mass Transfer 11 (1968) 1208–1211.
[5] Y. Katto, S. Yokoya, Principal mechanism of boiling crisis in pool boiling, Int. J. Heat Mass Transfer 11 (1968) 993–1002.
[6] A.E. Bergles, W.M. Rohsenow, The determination of forced convection surface boiling heat transfer, ASME Series C 86 (1964) 365–372.
[7] Y. Haramura, Y. Katto, A new hydrodynamic model of critical heat flux, applicable widely to both pool and forced convective boiling on submerged bodies in saturated liquids, Int. J. Heat Mass Transfer 26 (1983) 389–399.
[8] Y. Katto, C. Kurata, Critical heat flux of saturated convective boiling on uniformly heated plates in a parallel flow, Int. J. Multiphase Flow 6 (1980) 575–582.
[9] V.V. Yagov, V.A. Puzin, Critical heat fluxes in forced-convective boiling of refrigerant-12 under conditions of local heat sources, Heat Transfer—Sov. Res. 16 (1984) 47–52.
[10] K. Mishima, H. Nishihara, The effect of flow direction and magnitude on CHF for low pressure water in thin rectangular channels, Nucl. Eng. Des. 86 (1985) 165–181.
[11] M. Kaminaga, Y. Sudo, Y. Murayama, Experimental study of the critical heat flux in a narrow vertical rectangular channel, Heat Transfer—Jpn. Res. 20 (1991) 72–85.
[12] G.P. Celata, M. Cumo, A. Mariani, H. Naria, F. Inasaka, Influence of channel diameter on subcooled flow boiling burnout at high heat fluxes, Int. J. Heat Mass Transfer 36 (1993) 3407–3410.
[13] I. Mudawwar, D.E. Maddox, Critical heat flux in subcooled flow boiling of fluorocarbon liquid on a simulated electronic chip in a vertical rectangular channel, Int. J. Heat Mass Transfer 32 (1989) 379–394.
[14] G.P. Celata, M. Cumo, A. Mariani, Burnout in highly subcooled water flow boiling in small diameter tubes, Int. J. Heat Mass Transfer 36 (1993) 1269–1285.
[15] G.P. Celata, M. Cumo, A. Mariani, The effect of tube diameter on the critical heat flux in subcooled flow boiling, Int. J. Heat Mass Transfer 39 (1996) 1755–1757.
[16] J. Weisman, The current status of theoretically based approaches to the prediction of the critical heat flux in flow boiling, Nucl. Technol. 99 (1992) 1–21.