Characteristics of Transition Boiling and Thermal Oscillation in an Upflow Convective Boiling System

H. T. Liu

S. Kakaç

Department of Mechanical Engineering, University of Miami, Coral Gables, Florida

F. Mayinger

Lehrstuhl A für Thermodynamik, Technische Universität München, München, Federal Republic of Germany

■ The characteristics of transition boiling and thermal oscillation in a single-channel forced-convection upflow system were experimentally studied. The working fluid, R-12, flows vertically upward inside the test channel of inside diameter 2.85 cm and outside diameter 3.37 cm. The test section is 339 cm long and is heated by the Joule effect. Temperature fluctuations were measured in the tube wall in the transition boiling region under both hydrodynamically stable and unstable conditions. Without exit restriction, the system was hydrodynamically stable, and the usual irregular wall temperature fluctuation in the transition boiling region was observed; with an exit restriction, the system was found to be hydrodynamically unstable, and sustained temperature fluctuations with greater magnitudes and much larger periods were observed. Wall temperature fluctuations of this type are termed thermal oscillation. During thermal oscillation the hydrodynamic instabilities were also measured. The system pressure oscillations were measured at the inlet and the void fraction oscillations were measured at the exit with a capacitance void meter. The mechanism of thermal oscillation is given, and the effects of heater wall capacitance and axial conduction on the processes of transition boiling and thermal oscillations are studied. Typical graphs of instantaneous heat flux to the fluid versus time as well as typical recordings of wall temperature oscillations are presented. The instantaneous heat flux to the fluid versus the wall temperature is also plotted, and a limit cycle is produced. The critical heat flux (CHF) values were determined under both hydrodynamically stable and unstable conditions and are compared on maps of CHF versus exit quality at different mass fluxes with constant system pressure. This study shows that the hydrodynamically unstable system has substantially lower CHF values. Thermal oscillation has been observed over the following parameter ranges: mass flux, 300–3000 kg/(m^2 s); heat flux, 25–80 kW/m^2; inlet quality, -0.2 to $+0.6$; system pressure, 10 bar.

Keywords: *transition boiling, thermal oscillation, convective boiling, two-phase flow instabilities*

INTRODUCTION

In convective boiling, as in pool boiling, different boiling mechanisms exist in different boiling regions. In saturated boiling, in the upstream region, a liquid film exists on the tube wall with entrained liquid droplets in the vapor core. Boiling in this region is termed nucleate boiling where excellent heat transfer exists between the tube wall and the fluid and very high heat fluxes can be achieved. Dryout of the liquid film causes a drastic deterioration of heat transfer and leads to a surge in wall temperature. This region is called the liquid-deficient region (or film boiling region if the vapor quality is low).

Between these two boiling regions there is a boiling zone where thermal fluctuations in the tube wall come into existence unless some additional controls are used. The boiling mechanism in this region is termed transition boiling. Although continuous research efforts have been devoted to the understanding of this transition process, it still defies a full accounting.

France et al. [1] experimentally studied the characteristics of transition boiling in sodium-heated steam genera-

Address correspondence to Professor S. Kakaç, Department of Mechanical Engineering, University of Miami, Coral Gables, FL 33124.

Experimental Thermal and Fluid Science 1994; 8:195–205

0894-1777/94/\$7.00

tor tubes. They pointed out that there exist two types of thermal fluctuations, one with high frequency, which is inherent in transition boiling, and one with low frequency, which is system-induced, but no study was performed on the system-induced thermal fluctuations.

Dougall and Rohsenow [2] noted that when the heat flux was too low to maintain film boiling, nucleate boiling would appear at the tube inlet and then spread until the entire tube was in nucleate boiling. It was assumed that this effect was governed by axial conduction. Passos and Gentile [3] studied the effect of axial heat conduction on the boiling processes. Their measurements of the wall temperature along the tube showed that under specific test conditions a wall temperature gradient as high as 33 K/mm existed. Such a high temperature gradient along the tube wall promoted axial heat conduction and the extension of the vapor film to the upstream region for a heat flux smaller than the critical value. In Ref. 2 the axial heat conduction induces the transition from film boiling to nucleate boiling; in Ref. 3 the axial heat conduction induces the transition from nucleate boiling to film boiling. Since the boiling regime is determined by the true instantaneous heat flux to the fluid and not the average flux, the effects of axial conduction are obvious. When the wall temperature fluctuates, the heater wall capacitance also affects the instantaneous heat flux to the fluid. This has not attracted much research attention.

On the other hand, thermal instabilities have been studied both experimentally and theoretically. Stenning and Veziroğlu [4] investigated and identified thermal instabilities in a single-channel upflow boiling system. Mayinger and Kastner [5] also identified thermal oscillations. Kakaç et al. [6] presented investigations of thermal instabilities in forced convection boiling in a single vertical channel. Unified studies of transition boiling and thermal oscillation under hydrodynamically unstable conditions have been very limited. It is the objective of this study to determine the effects of hydrodynamic instabilities, axial conduction of the tube wall, and the capacitance of the tube wall on transition boiling and thermal oscillation.

Transition boiling is inherently unstable. In heat flux controlled systems, the tube wall temperature fluctuates in an irregular pattern and with high frequencies. When the whole system is hydrodynamically unstable, the coupling of transition boiling and system hydrodynamic instabilities together with the effects of axial conduction and the tube thermal capacitance may lead to very different thermal fluctuations of higher magnitudes and lower frequencies. In this paper, this type of thermal fluctuation in the tube wall is termed thermal oscillation. Thermal oscillation is a very complicated coupling of hydrodynamic oscillation, transition boiling process, and the dynamics of the tube wall. On the one hand, hydrodynamic oscillation causes dryout (and thus transition boiling) to occur earlier and with greater fluctuations of wall temperature and more violent movement of the dryout boundary; on the other hand, thermal oscillation induces greater oscillations of pressure and flow.

EXPERIMENTAL SYSTEM

Figure 1 shows a schematic diagram of the test loop. The test fluid, R-12, was supplied by a centrifugal pump. A

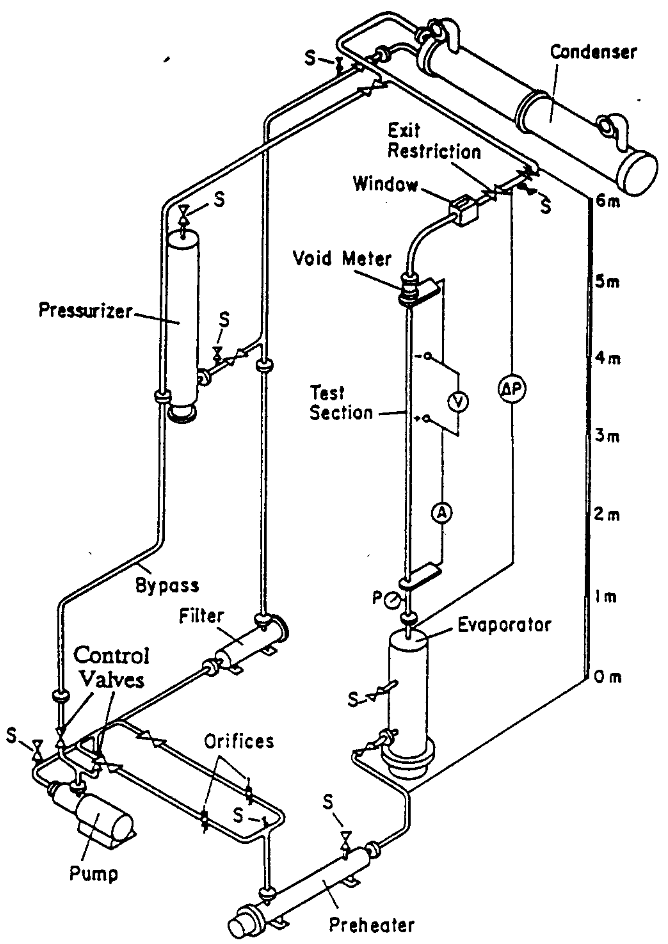

Figure 1. Schematic diagram of the experimental setup for transition boiling and thermal oscillations. S, safety valve.

bypass was incorporated into the test loop to ensure proper operation of the centrifugal pump.

The experimental setup can be divided into three main segments: the fluid supply section, the test section, and the recovery section. Each of these sections is explained in detail below.

Fluid Supply Section

As shown in Fig. 1, this section comprised a pump, a pair of control valves, an orifice arrangement, a preheater, and safety valves. The control valve provided accurate control of the flow rate through the test section and protection to the pump by adjusting the flow through the bypass. Two orifices arranged in parallel, one with a diameter of 1.6 cm and the other with a diameter of 1.1 cm, were capable of measurements of mass fluxes ranging from 300 to 3000 kg/(m^2 s) corresponding to the test section. The differential pressure transducer used was in the range 0–1000 mbar. The preheater, with a capacity of 47 kW, provided different inlet liquid temperatures up to saturation temperature. Two 1-mm-diameter flow-through thermocouples, one preceding the orifice arrangement and one after the preheater, provided measurements of fluid temperatures. The safety valves ensured safe operation of the pump, the preheater, and other parts of the fluid supply system.

Test Section

As shown in Fig. 1, the test section consisted of an evaporator, a heater, an impedance void meter, a window, an adjustable exit restriction, and the main body of instrumentation. Following the preheater was the evaporator, 1143 cm in height and 355.6 cm in diameter. The evaporator, with a heating capacity of 96 kW, was able to provide a wide range of inlet vapor qualities. The heater tube was a stainless steel tube 339 cm long, with an inside diameter of 2.85 cm and outside diameter of 3.37 cm. The heater tube is heated by direct current. An impedance void meter was installed at the end of the heater tube to measure the local void fraction fluctuations. Figure 2 is a schematic drawing of the cross section of the void meter and its circuit. The void meter consisted of thin concentric tubes that were wired to constitute five separate capacitors supplied with a high-frequency voltage. On the basis of the different dielectric constants of vapor and liquid, the void fraction in each part of the cross section could be determined. The test section terminated at an exit restriction after the window. The exit restriction could be set at a specific position to create the desired exit pressure drop. During this experimental study, the exit throttling coefficient was 8.7 when pure liquid R-12 at 23°C was used, that is, $f = 8.7$ in the relationship

$$\Delta P = f\frac{\rho}{2}w^2. \tag{1}$$

Recovery Section

Following the test section was the recovery system, which consisted of a condenser, a pressurizer, and a filter. The mixture of saturated liquid and vapor discharging from the exit was directed through the condenser, which was cooled by tap water. The condenser has a capacity of 190 kW. The condensed liquid was then directed to the inlet of the pump. A pressurizer was installed between the condenser and the pump. The pressurizer, which has an internal electric heating element, was maintained at a constant pressure by controlling the heat input to ensure a constant level of system pressure during experiments. A filter was also installed between the condenser and the pump to keep the pump and all the other components free of particles. The entire test loop was well sealed, and the test liquid was used repeatedly with almost no waste.

Instrumentation and System Accuracy

The data acquisition system consisted of an IBM-compatible PC, an HP 3495A scanner (multiplexer), and an HP 3456A digital voltmeter, with all the devices on a single HP-IB (IEEE 488) interface bus. Software capable of monitoring the experiment and data recording was developed and successfully employed. The data acquisition system was used throughout the experiments, both in monitoring them and in taking the data when desired.

In the following uncertainty estimates, the confidence levels are all at 95%. For temperature measurements, ungrounded sheathed K-type NiCr–Ni thermocouples were used. Fourteen of these, 0.5 mm in diameter and point-welded to the outer surface of the heater wall, were used for heater wall temperature measurements. The last thermocouple was fixed about 1 cm from the flange to minimize the end conduction effect. Four flow-through thermocouples of 1 mm diameter were used for fluid temperature measurements: one before the orifice arrangement, one after the preheater, one at the inlet just before the heated section, and one at the exit just after the void meter. The maximum calibration error of the thermocouples was 0.2 K. Additional uncertainties were 0.5 K due to the data acquisition system and data reduction and 0.5 K due to thermal contact resistance between the thermocouples and the surface in the case of wall temperature measurements. The dynamic response time of the thermocouples is at least two orders of magnitude higher than the studied temperature oscillations; therefore, the dynamic error of the temperature measurements is negligible. The overall uncertainty of temperature measurement is 1.0 K.

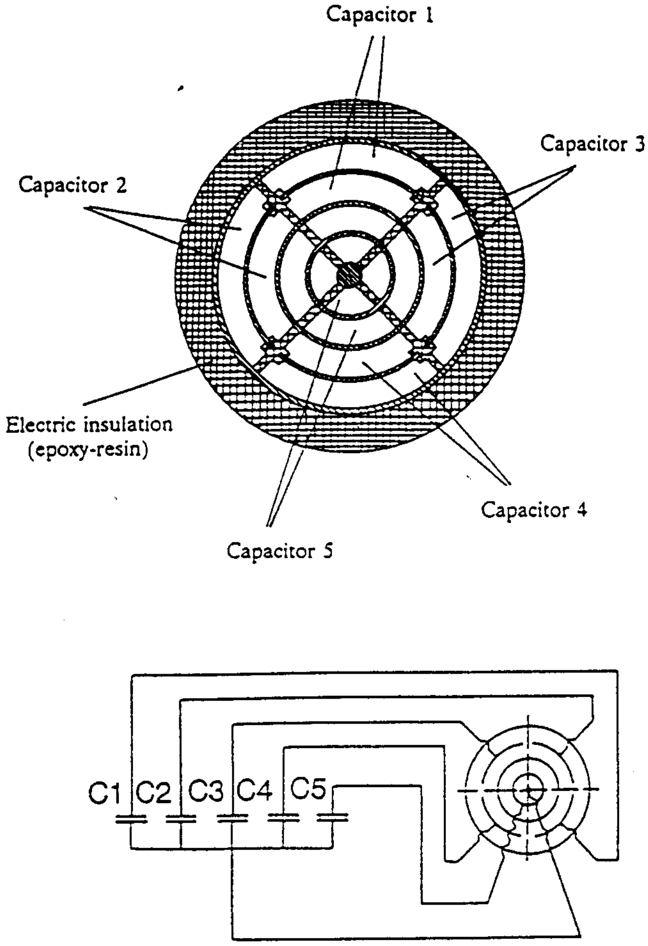

Figure 2. Schematic diagram and basic circuit of the capacitance void meter.

A strain gauge type of pressure transducer with a range of 0–50 bar was used to measure both the average and instantaneous pressures at the heater inlet. This pressure transducer has an accuracy higher than 0.25%, a sensitivity of 1 mV/V ± 0.25%, and a natural frequency of approximately 20 kHz. It is well suited for performing both dynamic and static measurements with high accuracy. The error introduced by the amplifier, data acquisition system, and data reduction is less than 0.5%. The changes in the average system pressure due to unbalanced heat input to the pressurizer during experiments were less than 0.05 bar. With the average system pressure being 10 bar, this change introduced less than 0.5% of error. The overall uncertainty is 0.61%.

For mass flux measurement, the differential transducer used is in the range 0–1000 mbar. The accuracy of the differential pressure transducer is greater than 0.25%, its sensitivity is 2 mV/V $\pm$ 0.25%, and its natural frequency is 10 Hz. The overall uncertainty of mass flux measurement is estimated to be 0.5%.

The heater tube was directly heated by controlled direct current supplied from a transformer-rectifier unit with a capacity of 29 kW and was remotely controlled by a Digistant 6425 T controller. The dc voltage was applied to the heater over two copper flanges, each welded to one end of the heater connections. The heater was electrically insulated from the rest of the system by the two insulating gaskets at the two ends. Heat inputs to the heater were obtained by measuring the current into and the voltage drop across the heater. The electric current was determined by measuring the voltage drop over a shunt circuit with a high-precision resistor of 20 μohms. The fluctuation of power input during the experiments was less than 0.5%. The uncertainties of voltage measurements were less than 0.5%. The heat loss in the test section is estimated to be less than 0.1%. The overall uncertainty of the power input into the test section, including the errors introduced by the data acquisition system and data reduction, was 1.0%.

The power input to the evaporator is measured by three power meters. The heat loss of the evaporator to the environment was determined by a series of heat balance experiments. The overall uncertainty of power input to the evaporator, after compensating for the loss of heat to the environment, is estimated to be 2.0%.

Since the main interest of this measurement was in the fluctuations of void fraction and not the void fraction distributions in the cross section, only the void fraction in the center region was measured during the experiments. Calibrations of the void meter were conducted before and after the experiments under vacuum, pure vapor, and pure liquid conditions.

EXPERIMENTAL PROCEDURES

Before any tests were performed, the digital voltmeter and scanner were warmed up at least 1 h as suggested by the manufacturers. The procedure for the actual tests can be outlined as follows.

1. With the computer and data acquisition system monitoring all the important variables, the power for the pressurizer was switched on to increase the system pressure to a preset level, and the cooling tap water for the condenser was turned on.
2. When the system pressure was high enough to ensure sufficient subcooling of the fluid before and in the pump, the pump was turned on, and the flow rate was adjusted to the desired level by adjusting the control valves. The preheater and the evaporator were turned on, and the power inputs were increased gradually to the desired levels. At least 40 min was allowed for the evaporator to reach steady state. (The dynamic response characteristics of the evaporator were experimentally determined before the experiments.)
3. The heat input into the test section was increased gradually to a preset value. The system was allowed to become steady, as indicated by the recordings of system pressures, temperatures, and flow rate.
4. Allowing at least 1 h for the system to reach either steady state or sustained oscillation state at the desired parameter values, several data acquisition processes were conducted to record both the system parameters and the measured variables.
5. Following step 4, an adjustment was made that was different for each set of experiments. Three different sets of experiments were performed, and each of the adjustments is given in the following:
   (a) For experiments with varying heat input, the heat input into the test section was increased gradually to the next value.
   (b) For experiments with varying mass flow rates, the mass flow rate was decreased to the next value.
   (c) For experiments with varying inlet quality, the preheater and the evaporator power input were increased until the inlet quality reached the next value.

After each adjustment, step 4 was performed.

Steps 4 and 5 were repeated until either the wall temperature was too high or the system pressure drop was too large.

The CHF boundary was approached several times during each run with successively smaller adjustments in one of the parameters. This procedure eliminates the possibility of directly crossing into the post-dryout region, which would give misleading information regarding the CHF values.

EXPERIMENTAL RESULTS AND DISCUSSIONS

Before discussing the experimental results, it is necessary to clear a few points in convective boiling.

(a) With similar conditions, the convective boiling has characteristics similar to those in pool boiling, especially the hysteresis phenomena.

The pool boiling curve is well established. It consists of three basic parts: nucleate boiling, transition boiling, and film boiling. In pool boiling, if the heat flux is controlled (e.g., by electrical heating), a hysteresis exists when the heat flux is increased to the film boiling region and then reduced back to the nucleate boiling region. In convective boiling, however, the hysteresis is not very obvious. For instance, from the results by Bennett et al. [7], we cannot see the hysteresis when the heat flux is being increased or decreased. Note that in this type of experiment, a particular liquid flow condition to a long vertical tube heated uniformly is established. Dryout is initiated at the downstream end of the tube either by increasing the heat flux or by reducing the flow rate. Further increasing the heat flux will drive the dryout point upstream, and the downstream section of the tube will enter the post-dryout region. If the heat flux is decreased, the downstream section of the tube will reenter the nucleate boiling region. Figure 3 shows the variation of surface temperature

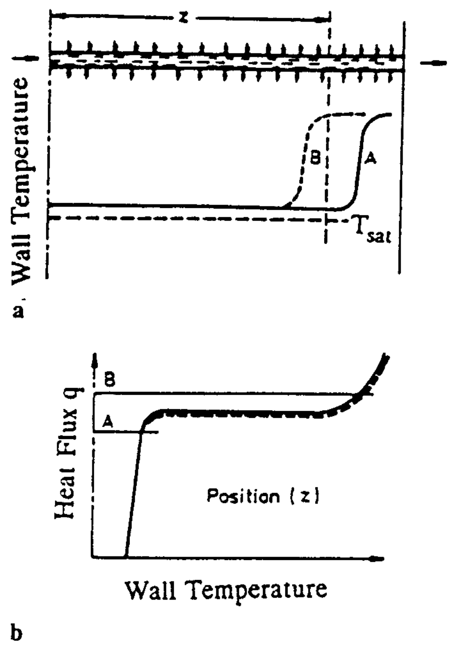

Figure 3. Dryout in a uniformly heated tube [8]. (——) Increasing q; (---) decreasing q.

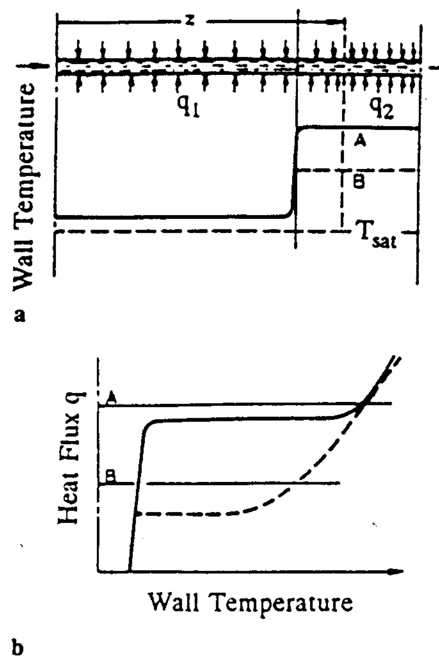

Figure 4. Dryout in a two-section heated tube [8]. (——) Increasing q; (---) decreasing q.

at a given position z as a function of heat flux [8]. There is obviously no hysteresis and no transition boiling region on this graph.

The important point here is that these curves are not the same boiling curves as in pool boiling. In pool boiling, whether subcooled boiling or saturated boiling, the pool is supposed to be so large that the bulk enthalpy of the liquid remains unchanged during the entire boiling process; that is, we increase and decrease the heat flux in the liquid with the same properties. In convective boiling with uniform heat flux, however, if inlet conditions are unchanged the local parameters are functions of heat flux. Therefore, at a specific location, the conditions change as heat flux changes. For instance, as the heat flux increases, the local quality also increases. It is common knowledge that the CHF is a strong function of local quality (or subcooling). Therefore, the curves shown in Fig. 3 are not boiling curves.

If the local properties are kept unchanged during the whole process of increasing and decreasing the heat flux, a true convective boiling curve can be produced. For instance, Bailey [9] found similar hysteresis in convective boiling as in pool boiling by the following technique. The heat fluxes q_1 and q_2 upstream and downstream of the test section can be adjusted separately. Initially, the two sections are operated at equal values of heat flux and dryout is established over the whole of the downstream section. Dryout is now maintained at the downstream end of the upstream section by holding q_1 fixed. The heat flux on the downstream section q_2 is then progressively reduced, and it is found that the wall temperature at position z falls smoothly with decreasing heat flux in the manner shown in Fig. 4 [8]. At some value of the heat flux q_2, rewetting occurs, and the wall temperature drops sharply to the nucleate boiling condition. Note that since the heat flux upstream does not change, the local properties at the measuring point are almost independent of the heat flux downstream. Therefore, the conditions in this case are similar to those in pool boiling, and the conclusion is that convective boiling has characteristics similar to those of pool boiling.

(b) In convective boiling, there is a boiling hypersurface instead of a boiling curve.

The boiling curves are at least dependent on the local vapor quality of the fluid, local pressure, and local fluid velocity. So there is a boiling hypersurface instead of a boiling curve in convective boiling. Even if the pressure and fluid velocity are kept constant, the local qualities at different locations along the channel are different at a fixed heat flux, so different points along the channel correspond to different boiling curves. The entire channel corresponds to a boiling surface. Besides, at a specific location, the local quality increases as the heat flux increases. Therefore, even at a specific point, there is a boiling surface instead of a boiling curve as the heat flux increases. This is why the results of Bennett et al. [7] show no hysteresis. Their result is not a boiling curve but a projection of the boiling surface on the q–T plane.

(c) In forced convective boiling, even under heat flux–controlled conditions, if nucleate and film boiling coexist, there exists a transition boiling region.

In pool boiling, if the heat flux is controlled, the transition boiling regime cannot be reached without additional measures. In forced convective boiling, when the heat flux is controlled and is uniform, if the upstream test section is in the nucleate boiling regime and the downstream part is in the film boiling regime (or liquid-deficient region), the wall temperature in the region in between changes continuously from that of nucleate boiling to that of film boiling, and so this region must be in the transition boiling regime. Therefore, even if the heat flux is controlled, a transition boiling region may be present in convective boiling, though the region may be very short and/or unstable.

Transition Boiling

When no exit restriction was present, the system was hydrodynamically stable during the experiments, evidenced by the measurements of mass flow rate, inlet pressure, and exit void fraction. Under such conditions, "normal" transition boiling phenomena were observed.

With high vapor quality, as in the case of this study, the pertinent two-phase flow regime is annular flow in the nucleate boiling regime. In this region, a liquid film exists on the tube wall with entrained liquid droplets in the vapor core, the heat transfer is excellent, and the wall temperature is not high. The wall temperature is stable in this region. When the liquid film is depleted, the flow regime becomes mist flow with relatively poor heat transfer. The wall temperature in this region is usually much higher than that in the nucleate boiling region.

Between these two regions there is a transition boiling region where thermal fluctuations occur in the tube wall. Figure 5 shows a typical result of the wall thermal oscillation in the transition boiling region obtained in this study. The sampling rate is 10 samples per second. Note that the fluctuations are irregular and thermal oscillation of the inherent transition boiling type is present.

Figure 6 shows a schematic diagram of temperature distribution along the test section. Fixing the dryout and nucleate boiling boundaries, the wall temperature in the transition boiling region oscillates with magnitude Y_2. In well-controlled transition boiling, this is what happens during wall temperature oscillation. In most systems, the movements of both the dryout and nucleate boiling boundaries are inevitable. In such a case, a fixed location along the tube, for example, one thermocouple location, may at one time be in the transition boiling region and at another time in the liquid-deficient region or in the nucleate boiling region. This will cause the wall temperature at a fixed location to fluctuate with much greater magnitudes or even with the maximum potential magnitude Y_1.

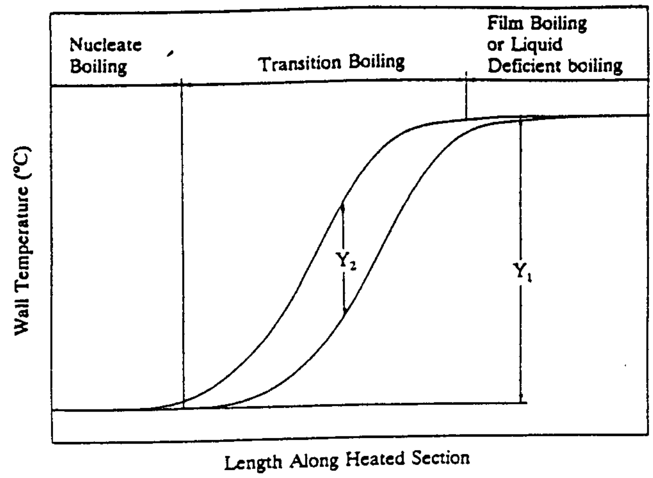

Figure 6. Schematic diagram of wall temperature distribution along the test section when upstream points are in nucleate boiling, downstream points in film boiling, and there is transition boiling in between.

Thermal Oscillation

When the exit restriction is present, under certain conditions, the system is hydrodynamically unstable. Figures 7 and 8 show typical reproduced oscillations of inlet pressure and exit void fraction. The pressure and void fraction signals were synchronized, and each was taken at a rate of 30 samples per second. The hydrodynamic oscillations are identified as instabilities of the density wave type.

In a boiling two-phase flow system, flow oscillation may lead to premature dryout and cause damages. Besides, when the downstream part of the test section is in the liquid-deficient region (or film boiling region) and the upstream part is in the nucleate boiling region, the oscillations of flow, pressure, and local quality enhance the movements of the boundaries of dryout and nucleate boiling. As explained above, this movement of the boundaries of dryout and nucleate boiling will lead to thermal oscillations of greater magnitude. Under certain conditions, at a fixed location in the transition boiling region,

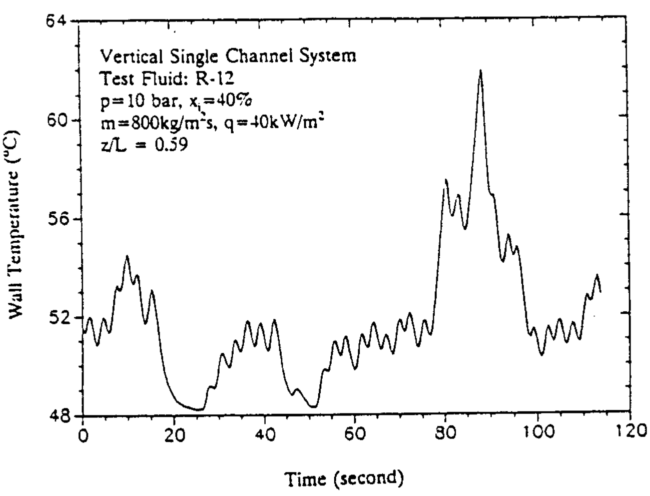

Figure 5. Typical wall temperature oscillation in transition boiling.

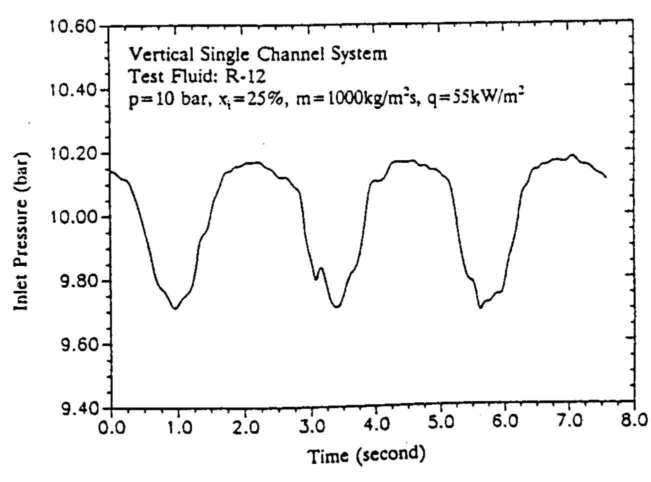

Figure 7. Typical inlet pressure oscillation when the system is hydrodynamically unstable.

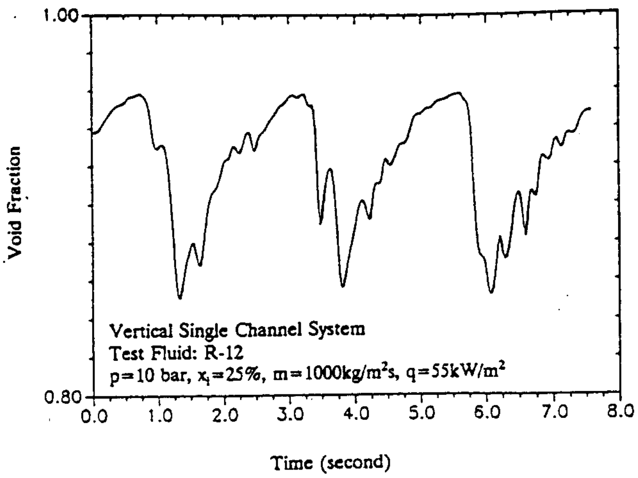

Figure 8. Typical void fraction oscillation at the exit when the system is hydrodynamically unstable.

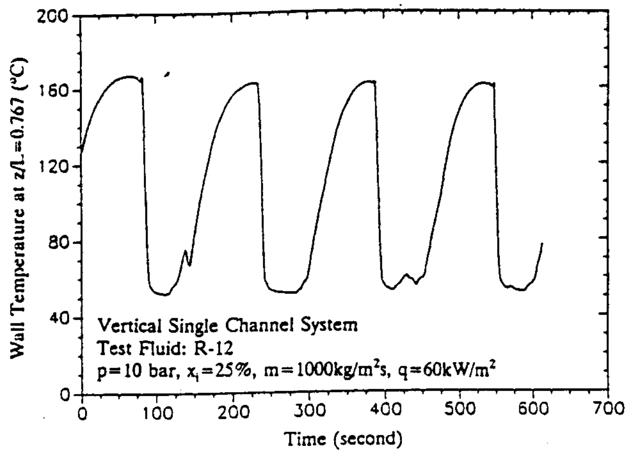

Figure 10. Typical wall temperature history during thermal oscillation at a location (point B in Fig. 9) alternately experiencing nucleate boiling, transition boiling, and liquid deficient boiling.

the wall temperature may fluctuate with the maximum potential magnitude, and this fluctuation may even be sustained. Figure 9 shows typical maximum and minimum temperature distributions along the test section when the upstream is in the nucleate boiling regime, the downstream is in the liquid-deficient regime, and a transition boiling region lies in between. The outputs of the 14 thermocouples along the test section were sampled for about 10 min, each at a rate of about 5 samples/s. The maximum and minimum temperatures at each point were found and plotted against the dimensionless length along the heater. It can be seen that both the dryout and nucleate boiling boundaries move along the test section. Figure 10 shows the wall temperature oscillation corresponding to point B in Fig. 9. Due to the movements of the dryout and nucleate boiling boundaries, point B at one time is in the nucleate boiling region and at another time in the liquid-deficient region. Therefore the wall alternatively experiences complete dryout and complete rewetting. The thermal oscillation has a fixed magnitude and a regular period that is about two orders of magnitude greater than that of inherent transition boiling.

The wall temperature oscillation corresponding to point A in Fig. 9 is shown in Fig. 11. Point A alternately moves into the transition boiling region and out to the nucleate boiling region. In the nucleate boiling region, the wall temperature is relatively stable, and in the transition boiling region it is unstable. Since this point does not move into the liquid-deficient region, it does not experience dryout, and thus the magnitudes of thermal oscillation are smaller than at point B.

Figure 12 shows wall temperature oscillation at point C. This point stays in the liquid-deficient region. The magnitudes of thermal oscillation are much smaller in this case since the point never enters the transition boiling region.

In order to isolate the source of the dynamics of thermal oscillation, the dynamic stabilities of the evaporator, which lies just upstream of the test section, were experimentally studied and found to be very stable. The evaporator's dynamic response to a stepped input was also

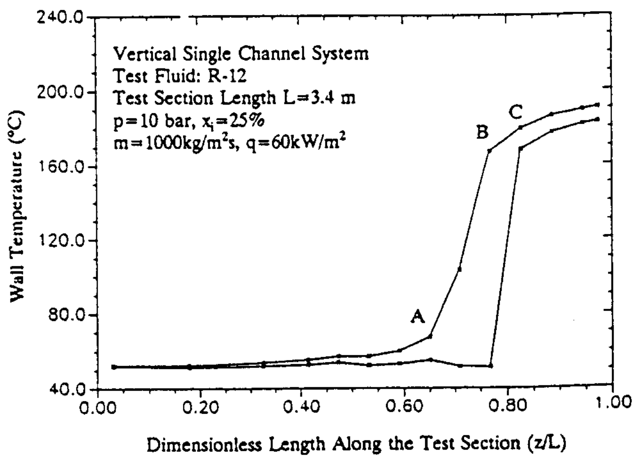

Figure 9. Typical maximum and minimum wall temperature distributions along the test section with upstream nucleate boiling, downstream film boiling, and transition boiling in between.

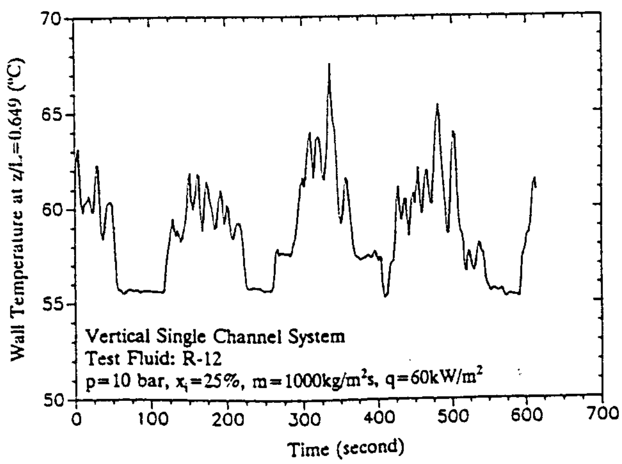

Figure 11. Typical wall temperature history during thermal oscillation at a location experiencing alternate nucleate boiling and transition boiling (point A in Fig. 9).

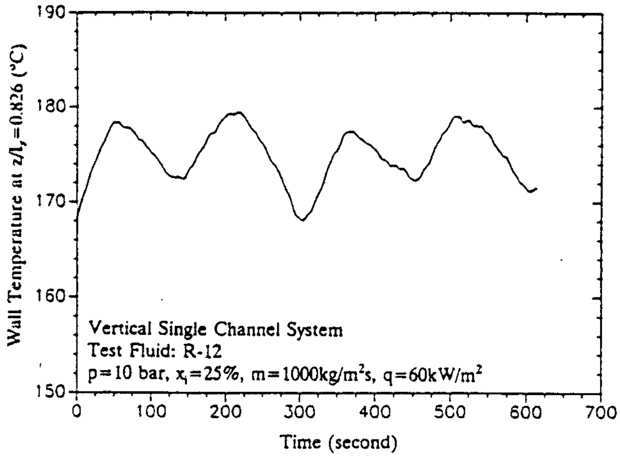

Figure 12. Typical wall temperature history during thermal oscillation at a location in the liquid-deficient boiling region on the boundary of the transition boiling region (point C in Fig. 9).

determined, and its time constant was found to be 12.5 min. So any disturbances upstream of the test section, if existent, would be smoothed out before reaching the test section. Another possible source of dynamics is the pressurizer, since it has a large compressible volume. In order to ensure that the pressurizer is not the source of dynamics, experiments were conducted with the valve connecting it and the test loop closed. Thermal oscillation was also observed under this condition. This excludes the possibility that the pressurizer acted as part of the dynamic system. Therefore the thermal oscillation could not have been caused by any external agents.

The Mechanisms of Thermal Oscillation

In the following explanations, for simplicity, no distinction will be made between film boiling and liquid-deficient boiling. Since the tube is vertical we can assume the temperature of the wall to be axisymmetric. To focus on the fundamental mechanisms of thermal oscillation, the wall temperature is lumped in the radial direction (the Fourier number is checked, and it is confirmed that the inside and outside wall temperatures oscillated in phase). The unsteady heat conduction equation is given by

$$q_f = q_0 - C\frac{\partial T}{\partial t} + K\frac{\partial^2 T}{\partial z^2} \tag{2}$$

where q_f is the instantaneous heat flux from the tube wall to the fluid, q_0 is the rate of heat generation expressed as per unit area of the tube inner surface,

$$C = \rho c_p(D^2 - d)^2/4d, \qquad K = k(D^2 - d^2)/4d$$

The Role of Wall Heat Capacitance The heat capacitance effect of the wall is to store thermal energy when the local wall temperature increases and to release thermal energy when the local wall temperature decreases. It is obvious from Eq. (2) that, at a specific location, if the wall temperature is increasing or decreasing, the effect of the wall heat capacitance is to reduce or increase, respectively, the heat flux to the fluid. A positive temperature perturbation at a point on the boundary between the nucleate boiling and transition boiling regions will lead this point to move toward the film boiling region. Note that this point is in the vicinity of the maximum heat flux point on the boiling curve. The wall stores energy and reduces the heat flux to the fluid, and this reduction in heat flux in the transition boiling region will cause further movement toward the film boiling region. On the other hand, a negative temperature perturbation at the same point will also lead this point to move toward the film boiling region.

Due to the constant heat generation rate in the tube wall, a decrease in the wall temperature will increase the heat flux into the fluid. Since the point is already at the maximum of the boiling curve, any further increase in heat flux will cause this point to "jump" to the film boiling region. Because of the continuity requirement of the wall temperature along the channel, this point cannot jump to the film boiling region but instead moves toward the film boiling region via the transition boiling region. Note that the stored energy has the potential to be released. Because of the continuity of the wall temperature, the part in the vicinity of this point will also move in the same direction, so more energy is stored and the temperature increases. This trend stops when the point on the boundary between the nucleate and transition boiling regions has an instantaneous heat flux smaller than q_{cr1} (the critical heat flux below which nucleate boiling is stable), so that perturbations will not lead this point to move toward film boiling.

If stable conditions were reached under this condition, this movement would stop. Note that this equilibrium is established with a higher wall temperature and lower heat flux condition. Once the movements stop, the wall temperature decreases to its "steady-state value" and the heat flux increases to its average value. A lower temperature and higher heat flux cause the points in the transition boiling region to move to the nucleate boiling region. Therefore, points on the boundary between the transition boiling and film boiling regions will move toward the nucleate boiling region. The points next to them will follow, and points in the film boiling region and close to the transition boiling region will move toward nucleate boiling. What really happens during this movement is that a liquid film advances downstream. The front of the liquid film is the quenching front, and the wall temperature decreases sharply at this location. This sharp decrease in wall temperature increases the heat flux to the fluid to very high values (Figs. 10 and 13). This movement stops when the point on the boundary between the film boiling and transition boiling regions has an instantaneous heat flux greater than q_{cr2} (the critical heat flux above which film boiling is stable), so that perturbations will not lead this point to move in the direction toward nucleate boiling.

Similarly, if this equilibrium were stable, there would be no more movements of the boundaries between the different boiling regions. Since during the movement from the film boiling region to the nucleate boiling region the wall releases its stored heat and increases the heat flux to a higher value than the heat generation rate in the tube wall, this equilibrium is established on the basis of higher heat flux and lower wall temperature. Once the movement stops, the wall temperature increases to its "steady-state

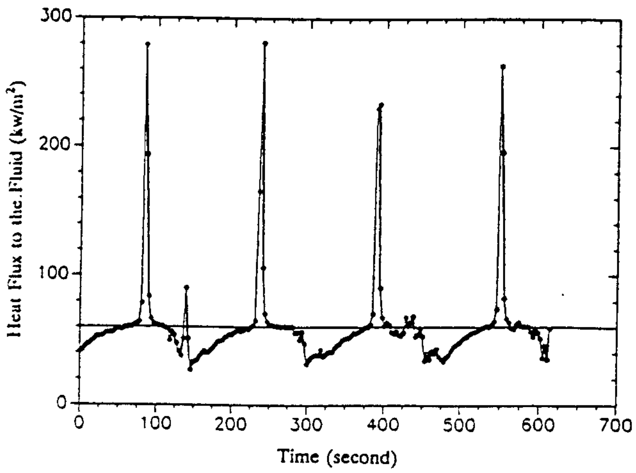

Figure 13. A typical result of instantaneous heat flux from the heater wall to the fluid with the wall heat capacitance effect included. System parameters: $p = 10$ bar, $x_i = 25\%$, $m = 1000$ kg/(m^2 s), $q = 60$ kW/m^2. Location: $z/L = 0.767$.

value" and heat flux reduces to its average value. The point on the boundary between the film boiling and transition boiling regions with a lower heat flux and higher temperature will move toward the film boiling region. This movement will be followed by the adjacent points, and another cycle starts.

The Role of Axial Conduction

It is obvious from Eq. (2) that, at a location z along the tube, if the second derivative of temperature with respect to z is positive, the axial conduction tends to increase the local wall temperature, and if the second derivative is negative, to decrease it. Referring to Fig. 6, for a point in the vicinity of the boundary between the transition boiling and nucleate boiling regions, this second derivative is positive. The effect of axial conduction is to increase the wall temperature and thus to "pull" this point toward the transition boiling region. If at this point the local heat flux is greater than q_{cr1} it will be "pulled" into the transition boiling region; otherwise, it will stay in the nucleate boiling region. Similarly, for a point in the neighborhood of the boundary between the transition boiling and film boiling regions, this second derivative is negative. The effect of axial conduction is to decrease the local wall temperature and also to "pull" this point to the transition boiling region. If at this point the local heat flux is smaller than q_{cr2}, it will be "pulled" into the transition boiling region; otherwise, it will stay in the film boiling region. Therefore, the axial conduction enhances the transition from one boiling regime to another.

In subcooled or low-quality convective boiling, the difference between q_{cr1} and q_{cr2} is large and q_{cr1} is greater than q_{cr2} (note also that q_{cr1} and q_{cr2} are different, corresponding to different locations along the tube). If the entire tube is in film boiling but the heat fluxes corresponding to every location along the tube are smaller than q_{cr1}, incipient nucleate boiling at the tube inlet will lead the entire tube to transform to stable nucleate boiling due to axial conduction [2]. Similarly, if a portion of the tube is in nucleate boiling and the heat fluxes corresponding to this portion are greater than q_{cr2}, axial conduction will cause this part of the tube to transform to stable film boiling [3]. Similar phenomena also happen in pool boiling as evidenced in Ref. 10.

From the above explanation, the wall heat capacitance together with the characteristics of transition boiling heat transfer is seen to be the source of the dynamics, and axial conduction a destabilizing agent. The combined effect is to cause the movements of the boundaries of nucleate boiling and film boiling. Because of the microscopic instabilities of nucleate and film boiling [11] and the delays in the transitions from one boiling regime to another, the wall temperature usually experiences aperiodic fluctuations. A specific point that moves from the film boiling region into the nucleate boiling region in one cycle may stop in the middle of the transition boiling region or even stay in the film boiling region in the next cycle. Also, the cycles may not take the same length of time. If the system is hydrodynamically unstable, as in the case with the exit restriction, periodic flow oscillations with much higher frequency exist, and the movements of the boiling region boundaries are greatly enhanced. Chances are that during each cycle a point with a heat flux greater than q_{cr1} will move to the film boiling region and a point with a heat flux less than q_{cr2} will move to the nucleate boiling region. Therefore a regular pattern is formed in the wall temperature oscillations as shown in Fig. 10.

Instantaneous Heat Flux to the Fluid

To estimate the instantaneous heat flux into the fluid, Eq. (2) can be used together with the experimental data. Numerical differentiation of the experimental data is necessary to evaluate the heat flux, and large uncertainties can result from such a process. An error analysis shows that the error introduced by neglecting the conduction term in Eq. (2) is lower than the error bound for numerical differentiation of wall temperature with respect to time. So we neglect this term in determining the instantaneous heat flux to the coolant. With this simplification, Eq. (2) reduces to

$$q_f = q_0 - C\frac{dT}{dt} \tag{3}$$

Using Eq. (3), the instantaneous heat flux to the fluid can be determined on the basis of experimental data. The heat flux to the fluid corresponding to Fig. 10 is given in Fig. 13. The error bound introduced from the numerical differentiation is 8%, the error bound from neglecting the conduction term is 5%, and the overall uncertainty of the heat flux is estimated to be 10%. It can be seen that the heat flux to the fluid fluctuates greatly. When the wall temperature increases, that is, when the point moves from the nucleate boiling region to the film boiling region, the heat capacitance of the wall reduces the heat flux to the fluid by as much as 50% of the value that is generated in the tube wall. When the wall temperature decreases, that is, when rewetting or quenching happens, the instantaneous heat flux to the fluid can be as high as 4.6 times the average heat flux. It is obvious that the magnitude of the deviations of the heat flux to the fluid from its average value is so large that its importance in the processes of thermal oscillation and transition boiling has to be recognized.

Based on this experiment study, it can be deduced that even if the wall is very thin, the wall heat capacitance will still cause the heat flux to deviate substantially from its average value. Besides, with a thinner tube wall, the fluctuations of the wall temperature would be faster and so the capacitance effect would be greater. It seems that except in the cases where the wall temperature fluctuates slowly, the heater wall capacitance plays a very important role in both transition boiling and thermal oscillation.

In order to better understand thermal oscillation, heat flux to the fluid versus wall temperature corresponding to Fig. 10 is graphed as shown in Fig. 14. It is obvious that on the phase plane the graph of heat flux versus wall temperature forms a limit cycle if only a few irregular points are omitted. Note also that during thermal oscillation the local quality, pressure, and mass flux also oscillate; so this limit cycle is a multidimensional curve on the boiling hypersurface. What we see here is only a projection of this limit cycle on the q–T plane.

Comparison of CHFs

Since the hydrodynamic instabilities cause premature dryout, the CHF under this condition will be lower than that under the hydrodynamically stable condition. In order to determine the effect of hydrodynamic instabilities on the CHF, experiments were conducted both with and without the exit restriction. Without the exit restriction, the system was hydrodynamically stable; with the exit restriction, the system was unstable under certain conditions. With certain values of system pressure, mass flow rate, and inlet conditions, the heat flux was increased in very small steps. After each step the system was allowed to reach a steady state or a sustained state. When wall temperature surges at the end of the tube, the heat flux is taken as the CHF. The results for CHFs under stable and unstable conditions are shown in Fig. 15. It is obvious that at the same equilibrium quality, under the unstable condition the CHFs are considerably lower than those under stable conditions. This means that hydrodynamic stabilities have

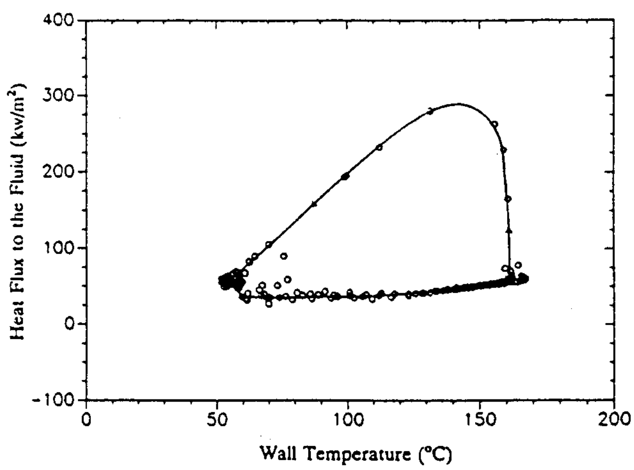

Figure 14. A typical limit cycle on the plane of heat flux versus wall temperature. System parameters: $p = 10$ bar, $x_i = 25\%$, $m = 1000$ kg/(m^2 s), $q = 60$ kW/m^2. Location: $z/L = 0.767$.

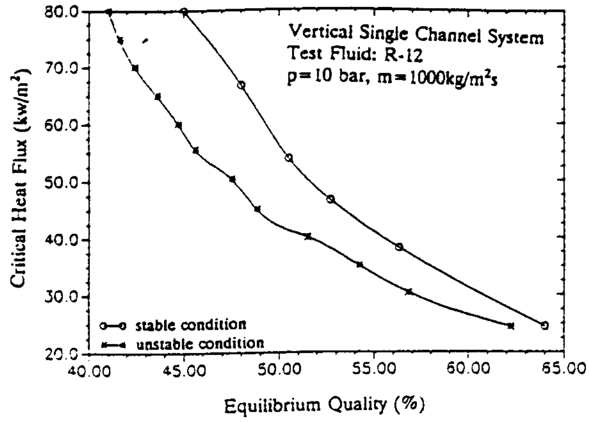

Figure 15. Comparison of critical heat fluxes under hydrodynamically stable and unstable conditions.

to be taken into consideration in the CHF predictions in the design of two-phase flow equipment.

PRACTICAL SIGNIFICANCE / USEFULNESS

The study of this type of thermal oscillation is very important because it leads to premature dryout. Most experimental data on CHF were obtained under very closely controlled laboratory conditions; that is, the experimental systems were made as stable as possible to obtain those data. In real industrial situations, a very stable system is not always possible due to either the structure of the system or energy considerations or simply due to the present state of knowledge. Therefore experimental data often need a large "safety factor" when they are applied to the design and operation of industrial systems. The "safety factor" includes not only known safety factors but also many factors that people are not able to determine at the time. If some of the uncertain factors become known, the "safety factor," and therefore the cost, can be reduced and still keep the system within the same safety margin, if not better. Besides, in certain applications, transition boiling and post-CHF regions are normal operating conditions. For instance, many sodium-heated steam generators used in liquid metal fast breeder reactor electric power plants operate with CHF and transition boiling in the water tubes. In such situations, the potential for high cycle fatigue due to thermal stress is an important consideration in determining the lifetime of the equipment.

CONCLUSIONS

In the convective transition boiling region, the wall temperature fluctuations have two distinct modes, one with high frequency and small amplitudes and the other with low frequency and large amplitudes. The mode with high frequency is inherent to transition boiling. It is a local phenomenon and also happens in pool boiling. The mode with low frequency is a system mode and depends on the heater wall capacitance, axial conduction, and transition boiling characteristics. When transition boiling occurs in a hydrodynamically unstable system, the system mode of temperature fluctuations in the heater wall is enhanced, and this oscillation may even be sustained and periodic—a phenomenon referred to as thermal oscillation in this paper.

Thermal oscillation is a complicated phenomenon resulting from the unique characteristics of convective boiling heat transfer in the transition boiling region, the effect of heat capacitance of the heater wall, and the effect of axial heat conduction and their interactions with the flow hydrodynamic instabilities. The wall heat capacitance and the characteristics of convective transition boiling heat transfer are the source of dynamics of thermal oscillation, and axial conduction is a destabilizing agent. The hydrodynamic instabilities enhance the movements of the boundaries between the boiling regions and thus lead the otherwise chaotic oscillation into periodic oscillation. Hydrodynamic instabilities also cause premature dryout and reduce the CHF considerably.

We gratefully acknowledge the financial support of the Deutsche Forschungsgemeinschaft and Alexander von Humboldt Stiftung. We also thank the numerous researchers, technicians, and staff members at Lehrstuhl A für Thermodynamik, Technische Universität München, for their invaluable assistance during the entire process of this research project.

NOMENCLATURE

- C capacitance of the heater wall, KJ/(m^2 K)
- c_p specific heat at constant pressure, kJ/(kg K)
- CHF critical heat flux in boiling heat transfer, kW/m^2
- d inside diameter of the heater tube, m
- D outside diameter of the heater tube, m
- k thermal conductivity, kW/(m K)
- K heater wall conductance, kW/K
- L total length of the test section, m
- m mass flux, kg/(m$^2 \cdot$ s)
- p pressure, bar
- q average heat flux into fluid, kW/m^2
- q_{cr1} maximum heat flux for stable nucleate boiling, kW/m^2
- q_{cr2} minimum heat flux for stable film boiling or liquid deficient boiling, kW/m^2
- q_f instantaneous heat flux into fluid, kW/m^2
- q_0 rate of heat generation expressed per unit area of the tube inner surface, kW/m^2
- t time, s
- T temperature, K or °C
- x_i inlet vapor quality, dimensionless
- z axial distance along the test section, m

REFERENCES

1. France, D. M., Carlson, R. D., Chiang, T., and Priemer, R., Characteristics of Transition Boiling in Sodium-Heated Steam Generator Tubes, *J. Heat Transfer* **101**, 270–275, 1979.
2. Dougall, R. S., and Rohsenow, W. M., Film Boiling on the Inside of Vertical Tubes with Upward Flow of the Fluid at Low Qualities, MIT Rep. TR-907926, September 1963.
3. Passos, J. C., and Gentile, D., An Experimental Investigation of Transition Boiling in Subcooled Freon-113 Forced Flow, *J. Heat Transfer* **113**, 459–462, 1991.
4. Stenning, A. H., and Veziroğlu, T. N., Flow Oscillation Modes in Forced-Convection Boiling, in *Proceedings of the 1965 Heat Transfer and Fluid Mechanics Institute*, Stanford Univ. Press, pp. 301–306, June 1965.
5. Mayinger, F., and Kastner, W., Berechnung Instabilitäten in Zweiphasenströmungen, *Chem. Ing. Tech.* **40**, 1185–1192, 1968.
6. Kakaç, S., Veziroğlu, T. N., Padki, M. M., Fu, L. Q., and Chen, J. X., Investigation of Thermal Instabilities in a Forced Convective Upward Boiling System, *Exp. Thermal Fluid Sci.* **3**, 191–201, 1990.
7. Bennett, A. W., Hewitt, G. F., Kearsey, H. A., and Keeys, R. K. F., Heat Transfer to Steam-Water Mixtures Flowing in Uniformly Heated Tubes in Which the Critical Heat Flux Has Been Exceeded, AERE Rep. R5373, 1967.
8. Bergles, A. E., Collier, J. G., Delhaye, J. M., Hewitt, G. F., and Mayinger, F., *Two-Phase Flow and Heat Transfer in the Power and Process Industries*, Hemisphere, Washington, D.C., 1981.
9. Bailey, N. A., The Interaction of Droplet Deposition and Forced Convection in Post Dryout Heat Transfer at High Subcritical Pressures, Presented at the European Two-Phase Flow Group Meeting, Rome 1972.
10. Kovalev, S. A., An Investigation of Minimum Heat Fluxes in Pool Boiling of Water, *Int. J. Heat Mass Transfer* **9**, 1219–1226, 1966.
11. Zuber, N., On the Stability of Boiling Heat Transfer, *J. Heat Transfer* **80**, 711–720, 1958.

Received January 26, 1993; revised November 8, 1993