

The two transition boiling curves and solid-liquid contact on a horizontal surface

I. Rajab and R. H. S. Winterton

Department of Mechanical Engineering, The University of Birmingham, Birmingham, UK

Transition pool boiling heat transfer from a horizontal circular surface is studied experimentally. Deionized water and refrigerant 113 are the working liquids at 1 atm pressure. The solution of the one-dimensional inverse heat conduction problem by Kastelin is extended to include a polynomial function of any degree for temperature versus time data. The existence of two transition boiling curves, i.e., separate transient heating and transient cooling curves, is confirmed. This finding is supported by independent measurement of solid-liquid contact. The heating transition boiling curve is higher than the cooling curve. Surface wettability has the same effect on both curves. As the contact angle increases, the transition boiling heat flux decreases. The two transition boiling curves are distinct even at zero contact angle. The steady-state measurements are generally higher than the transient measurements. There is no evidence of any jump among the steady-state, heating, and cooling curves.

Keywords: transition boiling; pool boiling; transient; steady state; inverse heat conduction; solid-liquid contact; wettability

Introduction

Transition boiling has received considerable attention during the past three decades under the pressure of modern technology. Application of the phenomenon to hypothetical accident scenarios in nuclear reactors has been the major impetus. The purpose of this study was to delineate the effect of surface wettability defined by the contact angle, as well as the history of the process, i.e., heating or cooling and the combined effects on the boiling curve. An important part of this study was to measure, apparently for the first time, solid-liquid contact simultaneously with the boiling curves for heating and cooling. Also, steady-state and transient measurements were investigated on the same boiling surface.

Witte and Lienhard¹ were the first investigators who argued for the presence of the two transition boiling curves. They reanalyzed the data of Berenson² and Sakurai and Shiotsu.^{3,4} Witte and Lienhard postulated the presence of two transition boiling curves even though the data were not very convincing. Also, the comments made by Winterton⁵ and the reply by Witte and Lienhard⁶ showed the need for more data to clarify the effect of surface wettability on the two transition boiling curves.

Experimental apparatus

The apparatus was designed to permit both steady-state and transient measurements. Also, the parameters influencing the boiling process, such as the history of the process (i.e., heating and cooling) and surface wettability, were anticipated in the design. Full details will be published later by Rajab.⁷

The experimental apparatus is shown in Figure 1, and test-section details are shown in Figure 2. The cylindrical test section is heated from the lower part by four Watlow cartridge heaters (300 W/240 V each). The pool of boiling liquid is

confined to the top surface of the test section by means of a glass tube and a preformed PTFE-ring seal. The diameter of the boiling surface is 27 mm. The 1.5-mm-thick flange at the top of the test section in Figure 2 is used to hold the glass tube securely in place. For the data-acquisition thermocouples, four 1-mm-diameter thermocouple holes were drilled at two levels: 2 mm and 27 mm below the boiling surface. At each level, one thermocouple was placed at the center and another halfway between the center and the side of the test section. This configuration allowed for the radial change in the temperature at each level. Two more thermocouple holes, 1 mm in diameter, were drilled at 2 mm and 34 mm below the boiling surface. The first thermocouple was used to display the surface temperature and the second to control the test section temperature. Type K sheathed thermocouples, 0.5 mm in diameter (Philips TKA 05/10/2) with a 0.05-s response time in water, were soldered in the holes with a tin-zinc mixture (Frys' metals number 703) having a melting point of about 300°C. The solder improves heat transfer between the materials of the thermocouple and the test section.

For steady-state measurements a temperature controller (Eurotherm 820), with a 0.1-s cycle time, was used to control the temperature of the boiling surface. The power consumption was determined by a kilo-Watt-hour meter.

To measure solid-liquid contact during boiling with water, a boiling heat transfer surface of anodized aluminum was utilized. The technique used is essentially that described by Dhuga and Winterton,⁸ with improvements described by Alem Rajabi and Winterton.⁹

A series of transient runs in heating and cooling were interrupted by a few steady-state runs. The boiling was conducted at 1 atm pressure with deionized water or refrigerant 113. The boiling surface was polished with emery paper (E320) and cleaned with acetone. The contact angle was measured before each experiment by means of a protractor eyepiece.

A typical experiment began with the deaeration of the deionized water by vigorous boiling in a separate beaker and with preheating of the test section to around 100°C. Then the water was poured over the surface and the deaeration continued for more than 10 minutes. Because of its low boiling point,

Address reprint requests to Dr. Winterton at the Department of Mechanical Engineering, The University of Birmingham, P.O. Box 363, Birmingham B15 2TT, UK.

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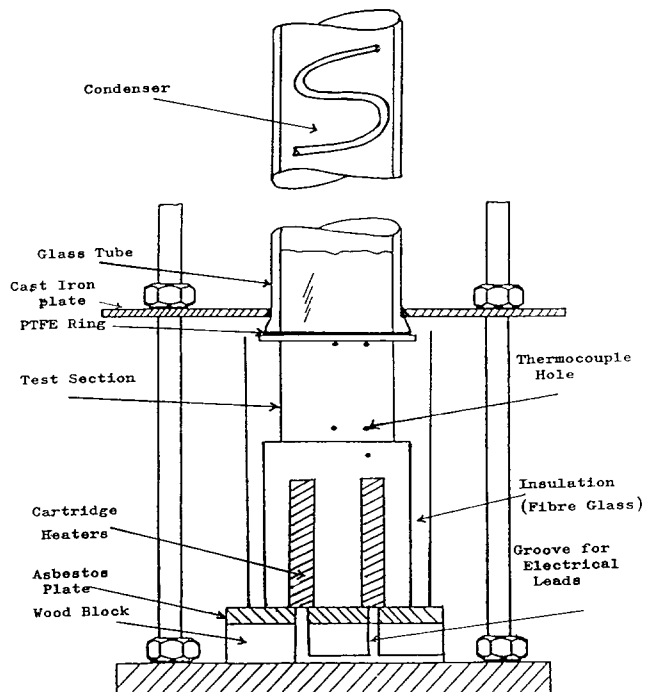


Figure 1 Configuration of the experimental apparatus

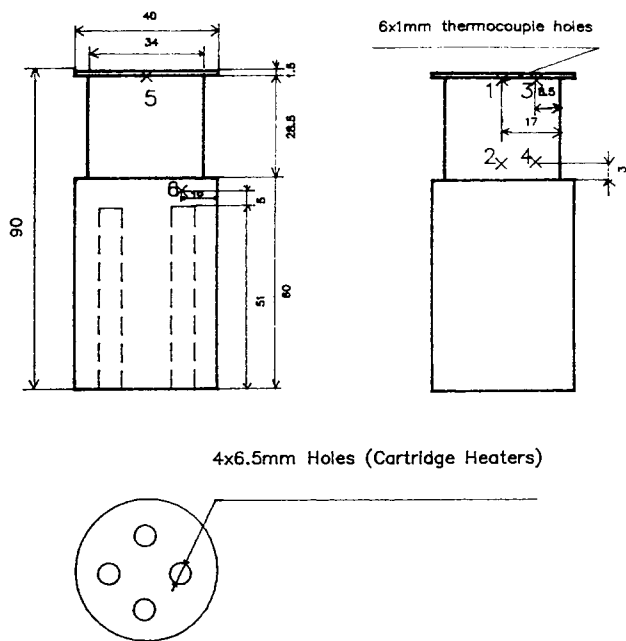


Figure 2 Test section details

refrigerant 113 was deaerated without preheating for nearly 15 minutes on the boiling surface. The heating run was conducted by increasing the set temperature of the surface to about 310°C. However, the cooling run was made by switching the power off.

Data reduction

In steady-state experiments, heat fluxes were readily calculated from the power consumption, with an allowance for heat losses. In transient measurements, heat fluxes at the upper level, q_1 , were reduced from the temperature versus time data at the

upper and lower levels (levels 1 and 2, respectively), i.e., by solving the one-dimensional (1-D) heat conduction equation derived by Kastelin [Kudryavtsev]¹⁰. Although Maracy and Winterton¹¹ essentially followed this same procedure before, the new result (Equations 5 and 6 in Appendix A) is general and can now be applied to polynomials of any degree.

Results and discussion

All of the experiments were performed with saturated liquids, deionized water, or refrigerant 113, at 1 atm pressure and on the same boiling surface. Transient heating and cooling and steady-state measurements were attained.

Steady-state experiments could be performed in nucleate boiling, film boiling, and most of the transition boiling. But just after the critical heat flux, the very steep negative slope of the transition boiling curve made steady-state measurements impossible (see, for example, Dhuga and Winterton¹²).

Figure 3 shows the boiling curves of deionized water on a commercially pure aluminum surface having contact angles of 65°, 52°, and 0° in heating and cooling. Contact angles were measured in a static mode at a surface temperature of about 20°C. In practice, surface temperature reaches about 300°C, and the liquid-vapour interface can no longer be considered static.

The effect of the contact angle is very clear: As it decreases, the transition boiling heat flux, including the critical heat flux and the minimum film boiling heat flux, increases (for a given surface temperature). This result was confirmed by Roy Chowdhury and Winterton,¹³ who used the quenching technique with metal cylinders (aluminum and copper).

Berenson² reported similar results, but he argued that the critical heat flux does not change with surface conditions. Also, Liaw and Dhir¹⁴ showed the boiling curves at contact angles of 107° and 38°. Their results are not completely consistent with the present results. They showed that the transition boiling heat flux is higher for the higher contact angle in the upper part of the curve and lower in the lower part of the curve. But

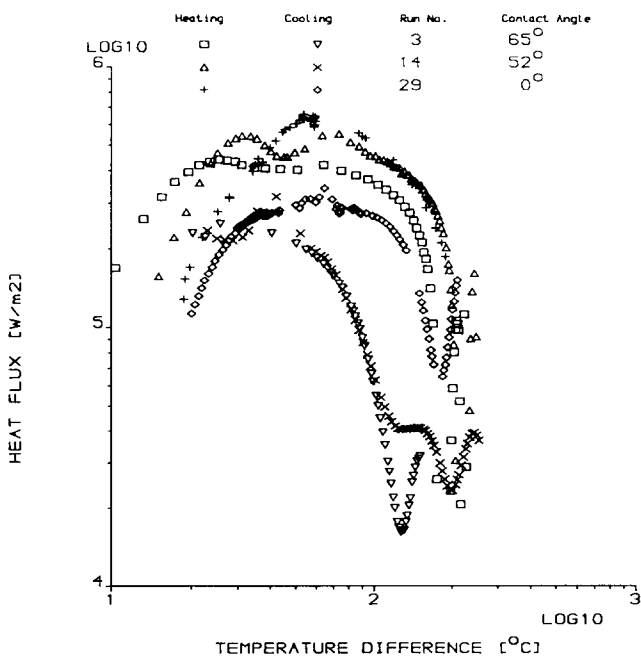


Figure 3 The boiling curves in transient heating and cooling of deionized water

this study shows that the transition boiling heat flux is higher for lower contact angles throughout the transition boiling curve. Similar results have been shown by Roy Chowdhury and Winterton^{15,16} and Maracy and Winterton.¹¹

Two different transition boiling curves have been distinguished for heating and cooling. The heating transition boiling curve is higher than the cooling curve in all of the results. The critical heat flux in cooling is 57% and 55% of that in heating at contact angles of 65° and 0°, respectively. So the difference between the two curves does not seem to be affected by the change in contact angle. Also, Figures 3 and 4 show the two transition boiling curves at zero contact angle for water and refrigerant 113, respectively. This result contradicts those of Liaw and Dhir.¹⁴ They showed that the two transition boiling curves for heating and cooling overlap at zero contact angle for refrigerant 113 on a copper surface.

Figure 5 shows the solid-liquid contact, i.e., wetted-area fraction, versus surface superheat in steady state and transient and in heating and cooling. The experiments were conducted on a new surface of anodized commercially pure aluminum with saturated deionized water. The contact angle was about 40°. The steady-state results are distinguished as "steady-state cooling"; i.e., the surface was previously at a higher temperature, but measurements are performed in a steady state, or "steady-state heating," where the surface was previously at a lower temperature. Obviously, solid-liquid contact in heating is higher than it is in cooling for both steady-state and transient measurements. Also, near the critical heat flux, contact is somewhat higher in steady-state heating than in transient heating. This approach confirms the presence of the two transition boiling curves: At a given surface temperature, the solid-liquid contact is higher in heating. The instrumentation used for the solid-liquid contact measurements is quite independent of that used to obtain heat flux values. The coincidence of the contact in steady-state cooling and transient cooling suggests that the speed of the transition has no effect on the contact. But the contact in steady-state heating is not quite the same as it is in transient heating. This difference suggests some contact dependence on transition speed. However,

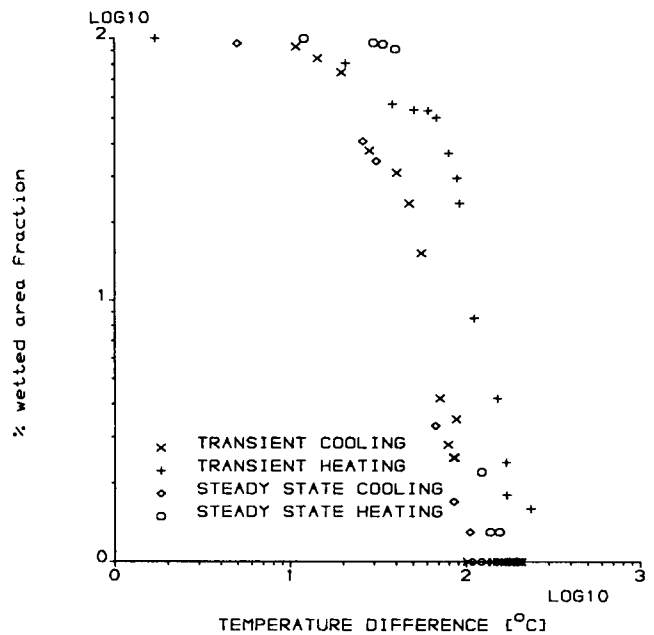


Figure 5 Solid-liquid contact of deionized water for steady-state (run no. 32) and transient runs (run no. 34) in heating and cooling on an anodized aluminum surface

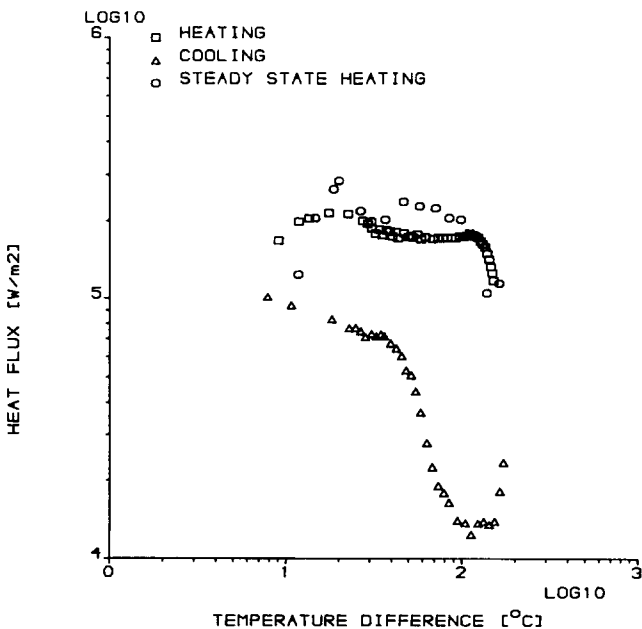


Figure 4 The boiling curves of arcton-113 for steady state (run no. 109) and transient heating and cooling (run no. 108)

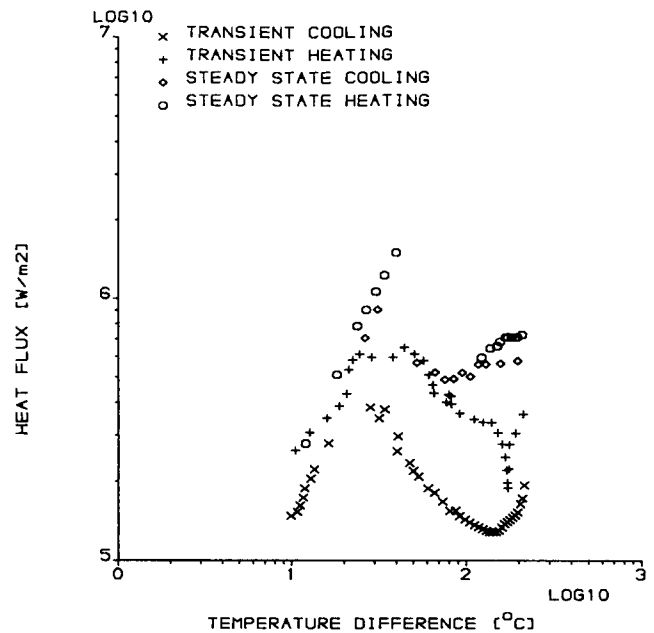


Figure 6 The boiling curves of deionized water for steady-state (run no. 32) and transient runs (run no. 34) in heating and cooling on an anodized aluminum surface

a steady-state run takes about nine hours, so a change in surface condition is quite possible.

Figure 6 shows the corresponding boiling curves. Comparing Figure 6 with Figure 5, we see that the critical heat flux occurs with high solid-liquid contact. In the nucleate boiling region, the main parameter in determining heat flux is surface temperature. At the minimum film boiling point, contact is 2% or less. The steady-state measurements are higher than those of transient heating at surface temperatures higher than 225°C and in the nucleate boiling region near the critical heat flux (a similar effect was obtained with refrigerant 113; Figure 4).

Also, the steady-state heat flux in heating is higher than that in cooling. Making steady-state heating measurements (i.e., increasing temperature, but still steady state) was rather difficult in the range 140–225°C, where the stored heat in the test section makes the surface temperature jump over that range.

For the steady-state results, a substantial part of the transition boiling region is missing. However, if lines are drawn linking the critical heat flux point with the nearest point in the transition region, the curves are similar to those for transient measurements but shifted to higher heat flux at a given wall temperature. That is, the steady-state curves also show a distinct difference between increasing temperature and reducing temperature.

The steady-state critical heat flux (heating) is found to be 110% for water and 114% for refrigerant 113 of the values predicted by Zuber's hydrodynamic theory.¹⁷

For the results with water, the speed of the transition in heating and cooling is around 1°C/s in most parts of the boiling curve. But near the critical heat flux, the speed is about 15°C/s in heating and 8°C/s in cooling.

If we define transition time as the time required to cover the temperature difference range 120–300°C, the transition time in heating is almost constant (100 s). But in cooling it drops from 639 s to 262 s as the contact angle falls. However, if the criterion of Peyayopanukul and Westwater,¹⁸ who used the quenching technique of cylindrical copper blocks in liquid nitrogen, is applied in this study, all of the curves presented are independent of the transition time, which is greater than 1.5 s (1.7 s for water and 8 s for freon-113). But the quasi-steady-state case is not valid, because the transient measurements are only 55% of the steady-state values at the critical heat flux and zero contact angle. Therefore another criterion for the quasi-steady-state case is required.

Postulate of the two transition boiling curves

The two distinct transition boiling curves explain the nature of the transition boiling regime. Approached from the nucleate boiling side by heating, growth and detachment of the bubbles reach their critical value at the critical heat flux, where the bubbles start to join each other, forming patches of vapor. The patches collapse under the dynamic behavior of the boiling liquid and new patches appear. At surface temperatures above 200°C, the solid-liquid contact drops to less than 10%. This change indicates that the patches join each other to form a film of vapor, which becomes complete at surface temperatures of about 270°C. Conversely, approached from the film boiling side by cooling, the vapor film starts to collapse into bigger patches than those in heating at the same surface temperature. Those bigger patches explain the lower heat flux when the rate of heat transfer declines. This phenomenon persists until the critical heat flux is reached and the normal growth of bubbles reappears. Therefore we postulate that:

Heating transition boiling is that boiling with unstable vapor patches, and cooling transition boiling is that boiling with bigger unstable vapor patches.

Figure 5 shows that the contact is the same in steady-state cooling and transient cooling, although the heat fluxes in the steady state are higher than those in transient. This difference could be attributed to greater energy being needed to support the patches of vapor.

Conclusions

A general solution for the Kastelin method of solving the inverse heat conduction problem has been obtained. Hence the temperature polynomial functions can be of any degree.

There are two distinct transition boiling curves on both wetting and nonwetting surfaces for heating and cooling. Both are affected in the same way by the contact angle. As the contact angle decreases, the transition boiling heat flux, including the critical heat flux and the minimum film boiling heat flux, increases.

The existence of two distinct transition boiling processes has been confirmed by separate measurement of the degree of solid-liquid contact. The steady-state measurements are higher than the transient values, and at the critical heat flux they agree very well with hydrodynamic theory.

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Appendix

Solution of the one-dimensional heat conduction equation by the Kastelin method

By having temperature vary with time at two levels in a test section, separated by a distance δ_1 , Kastelin [Kudryavtsev]¹⁰

derived the following equation for determining the heat flux:

$$q = -\frac{k}{\delta_1} [f_2(t) - f_1(t)] - 2\frac{k}{\delta_1} \sum_{n=1}^{\infty} e^{-sn} \int_0^t e^{sn} [(-1)^n f_2'(t) - f_1'(t)] dt \quad (A.1)$$

where

$$f_2(t) = \sum_{i=0}^m b_i t^i \quad (A.2)$$

$$f_1(t) = \sum_{i=0}^m a_i t^i$$

$$f_2'(t) = \frac{\partial f_2(t)}{\partial t} \quad (A.3)$$

$$f_1'(t) = \frac{\partial f_1(t)}{\partial t}$$

$$s = \left(\frac{\pi n}{\delta_1}\right)^2 \alpha \quad (A.4)$$

and $f_1(t)$ and $f_2(t)$ are the temperature polynomial functions of time at the upper and lower levels, respectively; m is the

degree of the polynomial; k is the thermal conductivity; and $\alpha = k/\rho c$ is the thermal diffusivity.

Therefore the heat flux could be evaluated analytically at each time interval, if the two temperature functions of time $f_2(t)$ and $f_1(t)$ were available. For this reason a curve-fitting program was necessary to evaluate the two functions from the available temperature-time history obtained from experiments. So we evaluated the heat flux at the upper level, 2 mm below the boiling surface, from Equation 1 by integrating by parts:

$$q = -\left(\frac{k}{\delta_1}\right) [f_2(t) - f_1(t)] - \frac{2k}{\delta_1} \sum_{n=1}^{\infty} e^{-sn} [(-1)^n B - A] \quad (A.5)$$

where

$$A = \sum_{i=1}^m \frac{(-1)^{i+1} f_1^i(t) e^{sn|t}}{s^i} \Big|_0$$

$$B = \sum_{i=1}^m \frac{(-1)^{i+1} f_2^i(t) e^{sn|t}}{s^i} \Big|_0 \quad (A.6)$$

The heat flux at the boiling surface, q_s , is evaluated by the heat balance method, using the control volume of thickness δ (4 mm) below the boiling surface. Also, the surface temperature, T_s , is evaluated by extrapolation of the temperature at the upper level, T_u , and the temperature at the lower level, T_l . The equations are presented by Maracy and Winterton.¹¹