Effect of Heating Boundary Conditions on Pool Boiling Experiments

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The two widely used experimental methods for pool boiling experiments are evaluated. Boiling heat transfer data employing either the water heating or the electric heating method obtained by various researchers are compared and discussed. The electric heating method is a constant heat flux process, while the water heating method represents a pseudo-exponential temperature profile for the heating water along the tube. It is concluded that the temperature profile along the tube is the major cause of the difference between results from the two methods. The parameters that shape the temperature profile and cause differences are discussed.

INTRODUCTION

In a pool boiling heat transfer setup, the method of supplying heating energy at the heat transfer surface plays a direct role in the accurate measurement of the heat transfer coefficients. The two most common methods are electric (resistive) heating and fluid heating. The electric heating involves supply of a constant heat flux to the heat transfer wall via resistive elements. The fluid heating uses heating energy of a fluid (often water) in a convective mode to heat the test section wall. During the process of experimental design and analysis of pool boiling tests concerns are: (1) how comparable are the results of water heating to resistive heating, (2) what are the advantages and disadvantages of one method over the other, and (3) which parameters critically affect the end results. In this study, both methods were investigated to provide a basis for comparison.

REVIEW OF RELATED WORK

Dhir (1991) reviewed recent advances made toward a mechanistic understanding of nucleate and transition boiling over smooth surfaces for pure liquids. The four mechanisms that contribute to the total boiling heat flux under pool boiling conditions are transient conduction at the area of influence of a bubble growing on a nucleation site, evaporation at the liquid-vapor interface, enhanced natural convection in the immediate vicinity of a growing bubble, and natural convection over the area that has no active nucleation site. The significance of these mechanisms depends on the magnitude of the wall superheat and other system variables like heater geometry, size, material, thickness, orientation with respect to gravity, surface contamination, and system pressure. It was suggested that the boiling process is an aggregate of many subprocesses that are affected by many system variables. The that the understanding of the interaction of subprocesses was very limited, and that to understand the differences between boiling data, detailed models of all the subprocesses and their interactions should be considered.

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Pasamehmetoglu and Unal (1993) investigated the heater effects on the saturation nucleate pool boiling curve, using a numerical model. Copper and nickel were considered as heater materials with thicknesses of 0.012 to 4 mm. Their parametric study showed that the heat transfer coefficient is higher for a copper heater than a nickel heater. In addition, for a given heater material, the boiling curve shifts to the left with decreasing heater thickness. It was concluded that in the study of boiling heat transfer data, numerical methods are extremely helpful in identifying the parameters controlling the nucleate boiling mechanism.

Unal et al. (1994) investigated numerically the effect of heating methods on saturated nucleate pool boiling. In their numerical investigation, they examined cases of constant heat flux and constant temperature heating methods. They showed that the heating method could significantly affect the boiling curve. They found that the boiling curve moved to the right (lower heat transfer coefficient) with the decreasing thickness of the plate heater for both the smooth and rough surfaces of nickel and copper heaters in the constant heat flux heating method. This trend was reversed in the constant temperature heating method; the boiling curve shifted left with decreasing thickness. They also found that use of alternating current instead of direct current resistive heating caused the boiling curve to move to the left. Despite the interesting findings, the authors did not provide any rationale for the trends or causes of the results.

Results of different studies for boiling heat transfer of R-134a over Turbo-B tube are depicted in Figure 1. Palm (1995) investigated boiling performance of Turbo-B tube with R-134a using the electrical heating method. The tube diameter was 19 mm and tube length was 250 mm. Operating saturation temperatures were 0.7°C and –20.4°C. His data for the operating temperature of 0.7°C are shown in Figure 1. Oh and Kwak (1996) employed the water heating method to experimentally investigate the effect of a direct current electric field on nucleate boiling heat transfer for refrigerants R-11 and R-113 in a single-tube shell/tube heat exchanger. Even though they were successful in showing the enhancement in heat transfer due to the application of high voltage electric field, in the absence of the electric field the boiling heat transfer coefficient was almost 50% lower than the resistive heating values found by both Marto and Lepere (1982) for refrigerant R-113, and Papar (1993) for refrigerant R-11. The authors acknowledged that their data is lower than others are, but did not explain the cause.

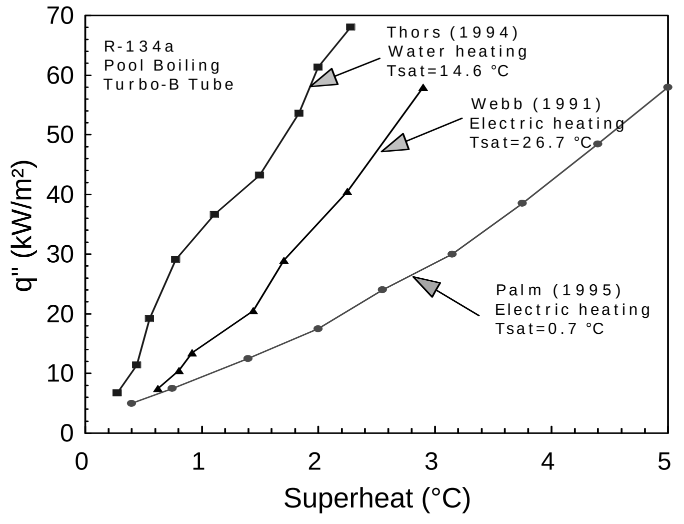

Figure 1. Comparison of data on Turbo-B tube with R-134a

Webb and Pais (1991, 1992) experimentally investigated the boiling performance of Turbo-B tube using R-134a with the electric heating method. They performed the tests at 4.4°C and 26.7°C. The tube diameter was 19 mm and the tube length was 165 mm. Their results for the operating temperature of 26.7°C are shown in Figure 1. Turbo-B tube was tested using the water heating method for a tube length of 2.4 mm and outside diameter of 19 mm (Thors 1994). The working fluid was R-134a and the operating temperature was 14.6°C. The tube-side water velocity was 1.6 m/s. For the data obtained with fluid heating, the value of the wall superheat was averaged along the tube using average heat flux and heat transfer coefficient.

From the data in Figure 1, it appears reasonable that the data of Webb falls to the left the data from Palm, as the saturation temperature is higher in Webb's case compared to Palm's case. However, the data from Thors, taken at a lower saturation temperature (14.6°C) than that of Webb case (26.7°C), should fall to the right of the data of Webb. This may suggest that the differences observed are attributed to the heating method and boundary conditions that are discussed later in this paper.

Another comparison of heating methods is reflected in the results of electric heating for boiling performance of R-114 over smooth tube by Memory et al. (1995) compared with the results of the water heating data of McManus et al. (1986) in Figure 2. Memory et al. (1995) conducted the experiments at 2.2°C saturation temperature with a tube length of 450 mm and diameter of 16 mm. McManus et al. (1986) ran the experiments at 13.8°C saturation temperature for a tube length of 304 mm and diameter of 16 mm. As can be seen from Figure 2, the results differ by as much as 50%. Also, the order does not match that of Figure 1. Although the saturation temperature is different for the two cases, it is very likely that part of the difference is due to the method of heating employed.

Kedzierski (1995) experimentally investigated the pool boiling performance of R-123 on four enhanced surfaces. The tubes were Turbo-BII, High Flux, GEWA-k, and GEWA-T. The surfaces were either machined or soldered onto a flat, thick, high conductivity copper plate. He investigated the boiling performance of the tubes by electric heating as well as water heating. He observed differences between the results. Figure 3 compares water to electric heating for the three tubes tested for the heat flux range of 10 to 70 kW/m^2. Kedzierski found that water heating

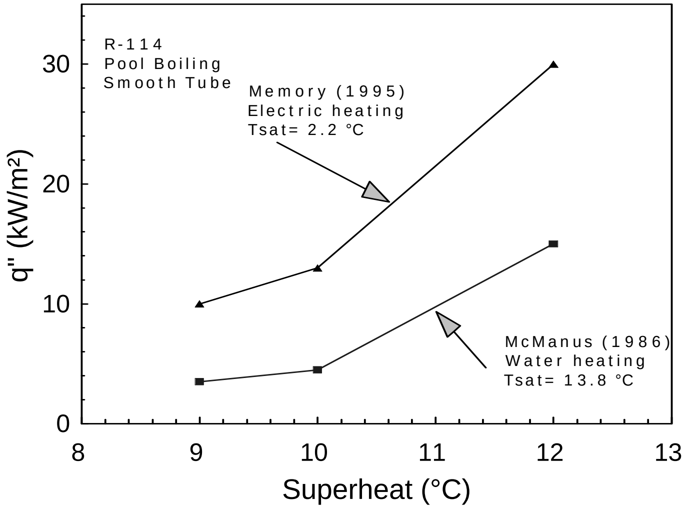

Figure 2. Comparison of data on smooth tube with R-114

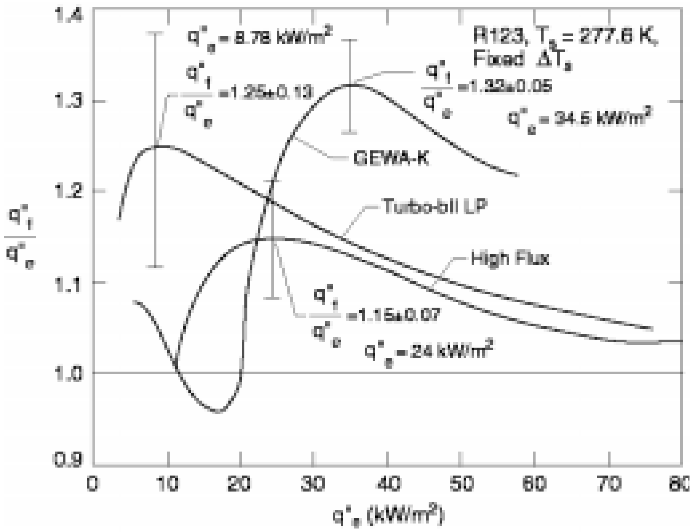

Figure 3. Comparison of water to electric heating for three tubes (Kedzierski 1995)

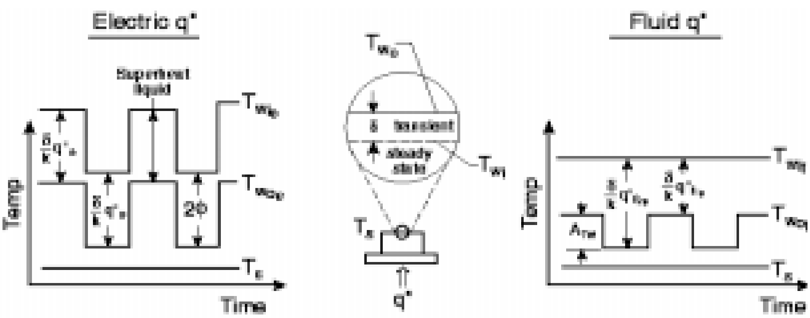

Figure 4. Speculative representation of temperature variation of the tube surface (Kedzierski 1995)

in most cases resulted in higher heat transfer coefficients compared to electric heating. The highest case was for the GEWA-K tube, in which water heating resulted in as much as a 32% greater heat flux compared to electric heating at a heat flux of 35 kW/m^2.

In an effort to explain the difference between water heating and electric heating, Kedzierski postulated that for the same time-averaged heat flux, a larger fraction of heat was used to superheat liquid in the electric heating method than in the water heating method. He approximated the transient surface temperature of the heating plate as a square wave, which was low for boiling and high for liquid superheating modes. A thin penetration depth δ in the wall near the boiling surface was defined to explain the transitional behavior of the plate temperature, defining an inner wall temperature T_{wi} at the lower edge of δ and an outer wall temperature T_{wo} at the upper edge of δ, as shown in Figure 4. The explanation was that during boiling the outer wall temperature T_{wo} dropped, since it was a more efficient means of transferring heat than natural convection (therefore having a square wave profile). The inner wall temperature was constant for the water heating method but varied for the electric heating method in phase with the same amplitude as in the outer surface temperature T_{wo} due to the constant heat flux constraint. The analysis was shown in the following equational form:

$$q''_w = q''_e - \frac{A_{TW}k}{2\delta} \tag{1}$$

Where A is the amplitude of the assumed square wave temperature profile along a period of a timed average heat flux, and k is the thermal conductivity. The above equation shows that water heating superheats the liquid less than electric heating by the amount of $Ak/2\delta$. In conclusion, for the same time-averaged heat flux, the boiling curve for the electric heating method would fall to the right, indicating a higher heat transfer coefficient for the water heating method.

The above analysis explains some of the discrepancies found in the data. However, the model of Kedzierski cannot explain all of the factors contributing to the differences found between boiling heat transfer for electric and fluid heating. For example, it did not explain the causes responsible for larger heat transfer coefficients being obtained for electrically heated data on a short test section as compared to averaged fluid heated data over a long test section. The model accounts for only boiling heat transfer differences between electric and fluid heating conditions on identical surfaces having the same length and operating conditions. In the present study a more encompassing approach to explain the differences between the heating methods is provided.

EXPERIMENTAL METHODS

In the present investigation, experiments were performed on the Turbo-BII tube using both electric and water heating methods. Two experimental setups were employed. The first setup was a small-scale unit, which was used for pool boiling heat transfer of non-CFC refrigerants, with a resistive heating method. The schematic diagram, of the setup shown in Figure 5a, consists of a high-pressure boiling chamber, an air-cooled refrigeration unit, the test section, and a pressure control system. The nominal outside diameter of the test tube was 19 mm (3/4 in.) and the length was 63.5 mm (2.5 in.). An electrical cartridge heater was imbedded in the test tube to provide heat. The heater outside diameter was 6.35 mm (1/4 in.) with a heating length of 50.8 mm (2 in.). The detailed description of the setup is given in Ohadi et al. (1994).

The second setup was an industrially scaled liquid-to-refrigerant loop, that was designed to address the applicability of the EHD technique for refrigerant-side heat transfer enhancement in industrial chillers. The heating was applied by a water heating method. The overall schematic of the setup is shown in Figure 5b. A horizontal shell-and-tube heat exchanger was built as the EHD-enhanced pool boiling test facility to withstand pressures up to 2 MPa (~300 psi). To simulate the operating conditions of practical industrial chillers, hot water passed through the inside of the heat exchanger tubes and boiling took place on the outside of the tubes. There were two sub-loops: a hot water loop, provided the heating energy required for boiling of the refrigerant, and a cold water loop provided cooling water for the condenser loop to condense the refrigerant. Other components for the system included the instrumentation for measurement of temperature, pressure, and flow rates in the loop.

A 18.7 mm long stainless steel shell of 200 mm inside diameter with 8.1 mm thickness was designed to accommodate high pressure refrigerants. The two sight glasses are located at the middle of the shell along its length. The test section tube was a Turbo-BII copper tube with 19 mm outside diameter and 2.87 m length. Approximately 1872 mm of the tube was located inside the shell. The part of the tube that was in the shell was enhanced surface and the developing part was plain tube.

Data were taken in the electric heating setup at the saturation temperatures of 4°C, 15°C, and 19°C for a tube length of 63.5 mm and diameter of 19 mm. The data for the water heating sys-

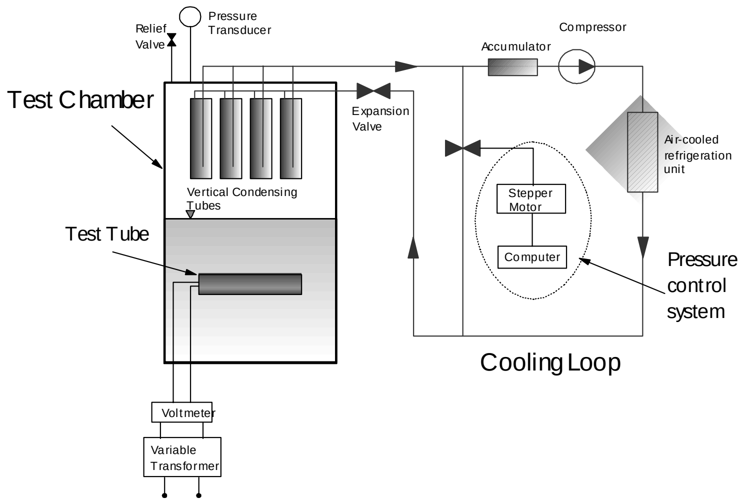

Figure 5a. Schematic of resistive heating setup

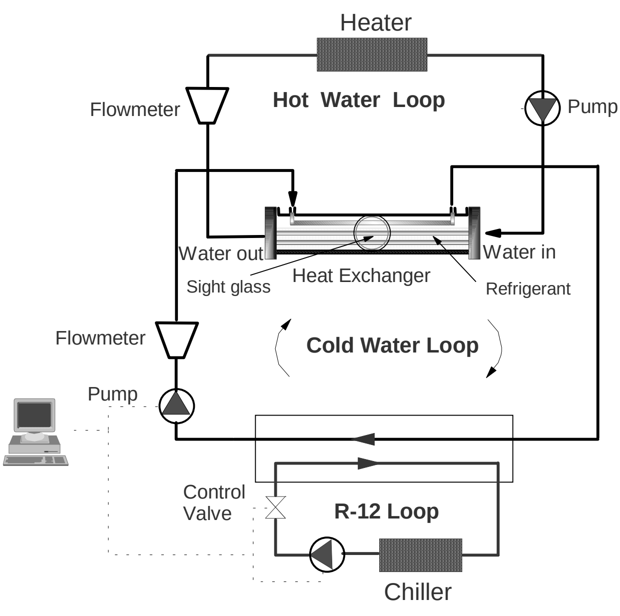

Figure 5b. Schematic of the water heating setup

tem were taken at the saturation temperature of 15°C for the same tube diameter and a tube length of 1872 mm. Figure 6 shows the results of the tests for the common saturation temperature of 15°C. It is seen that the water heating data are to the left of electric heating data. In a separate set of experiments, a 19 fins per inch (fpi) tube (fin spacing ≈ 1.34 mm) was used for the boiling experiments with R-123 as the working fluid at an operating temperature of 26.7°C using the water heating method. As shown in Figure 7, the data are to the right of the resistive heating results of Kumar (1993) for the same tube and saturation temperature. This trend does not match the order in Figure 6.

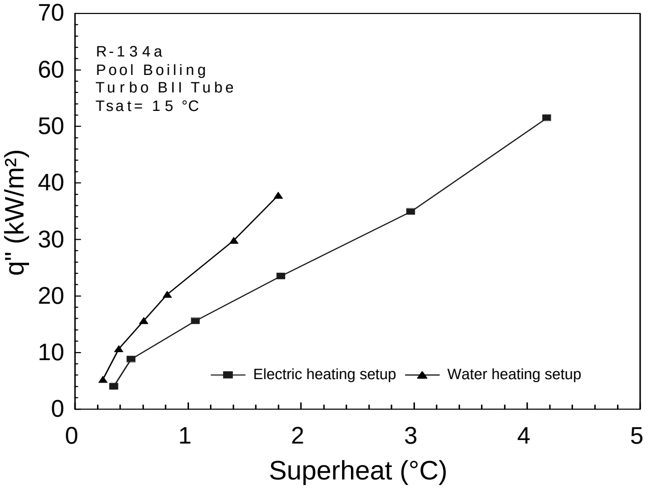

Figure 6. Comparison of data on Turbo-BII tube with R-134a

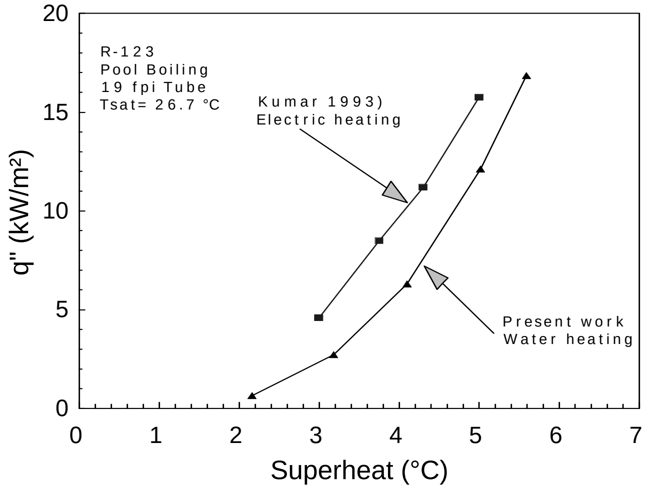

Figure 7. Comparison of data on 19 fpi tube with R-123

ANALYSIS OF WATER HEATING AND ELECTRIC HEATING PROCESSES

As discussed by Dhir (1991), in a boiling heat transfer process the heat transfer coefficient strongly depends on (1) the magnitude of heat reaching the boiling surface, (2) the conduction through the thickness of the surface, and finally (3) the outer surface structure over which the boiling occurs. In the present analysis, the comparisons between the fluid and electric heating conditions are made for identical surface structures and heat flux. Consequently, the focus of the analysis is on the conduction from the heated side of the heat transfer surface to the boiling side. In general, there are two ways that conduction and heating method can interact and cause an apparent difference in boiling heat transfer. The first conduction effect is observed for a comparison of local electric data to local fluid data. The local-local heat effect was observed by

Kedzierski (1995). The second conduction effect is observed when comparing local electric data to global fluid data. Here the averaging process contributes to the differences attributing larger heat transfer coefficients to the electrically heated boundary condition. As will be seen in the following discussion, both the first and second conduction effects are linked to the temperature profile along the heated surface.

Differences Due to Averaging

In order to address the difference between the two heating methods contributed by the averaging process, the results of electric heating (constant heat flux) were applied to water heating in a process of segmentation. This was done with the data obtained for the boiling performance of a 19 fpi tube with R-123. The length of the water heating tube (1872 mm long) was divided into 6 segments, each approximately 312 mm. Then the results of the heat transfer coefficient obtained from electric heating were applied to each segment of the tube, assuming that the heat flux on each segment remained constant. Although this approach is not strictly correct, it is an appropriate approximation.

In order to obtain a heat flux for each segment of the tube, different temperature profiles were assumed along the tube for the mean water temperature. The two ends of the temperature profiles were inlet and outlet water temperatures. A linear profile was assumed along with the two exponential profiles with two different arbitrarily selected radii of curvature. Then for each segment of the tube and for each temperature profile, heat flux and heat transfer coefficient calculations were performed and the arithmetic average of the heat transfer coefficient for the entire tube was calculated. For a superheat of 4.2°C, the heat transfer coefficient in the electrical heating system was found to be 2600 W/($m^2 \cdot$K). By applying the segmentation method to the water heating method, the heat transfer coefficient was determined to be approximately 2600, 2500, and 2200 W/($m^2 \cdot$K) for the linear and the two exponential temperature profiles, respectively. For the same superheat, the water heating data showed a heat transfer coefficient of 2200 W/($m^2 \cdot$K). This indicates the importance of the temperature profile along the tube using the segmentation process, the results of the resistive heating method approach those of the water heating method. This issue was considered in more detail by considering the analytical solution of temperature profile for the two modes of heat transfer.

The energy balance equation for internal fluid flow is:

$$\frac{dT_m}{dx} = \frac{q''P}{\dot{m}c} = \frac{p}{\dot{m}c}h(T_s - T_m) \tag{2}$$

where p is the perimeter, h is the inside heat transfer coefficient, and T_s is the wall temperature. For a constant surface temperature it is convenient to define $\Delta T = T_s - T_m$, Equation (2) is reduced to:

$$\frac{dT_m}{dx} = -\frac{d(\Delta T)}{dx} = \frac{p}{\dot{m}c}h(\Delta T) \tag{3}$$

The solution for the near temperatures is,

$$\frac{T_s - T_m(x)}{T_s - T_{m,\,i}} = \exp\left(-\frac{px}{\dot{m}c}h\right) \tag{4}$$

The mean water temperature exponentially approaches the wall temperatures. In fully developed flow the convective heat transfer coefficient is a constant and independent of x.

In a pool boiling experiment with water heating, as the water passes through the tube, water temperature decreases along the tube due to heat transfer to the outside boiling liquid. However, because temperature is constant in pool saturation boiling, the heat flux also decreases along the tube. Therefore, in this case neither the heat flux nor the temperature along the tube is constant. In contrast in resistive heating, where the heat is supplied by a heater from the tube core, the heat flux along the tube is constant.

Resistive and water heating are two different processes, with the main difference arising from the nature of the temperature profile along the tube. Electric heating is a constant heat flux process. Water heating, on the other hand, produces neither a constant surface temperature nor a constant heat flux process, and the corresponding temperature profile is neither linear nor exponential. In order to determine the shape of the temperature profile in the water heating method, the following temperature profile calculation was performed.

Data Reduction for Temperature Profile Calculation

In order to obtain the temperature profile in the water heating method, the tube was divided into 20 segments, each approximately 93.6 mm long. A computer program was used to solve five equations for each segment simultaneously for each segment. Equation (5), represents data obtained by the electric heating method for the boiling heat transfer coefficient (Kumar 1993). Equations (6) and (9) represent the convection heat transfer equations for the outside and inside of the tube, respectively. Equation (7) represents the solution of the heat conduction equation for a cylinder, and Equation (8) gives the water mean temperature for each segment of the tube.

$$h_o = -75.967 + 0.467261q'' - 2.946 \times 10^{-5} q''^2 + 8.123 \times 10^{-10} q''^3 \tag{5}$$

$$T_{so} = T_{sat} + q''/h_o \tag{6}$$

$$T_{si} = T_{so} + A_o q'' \frac{\ln(D_o/D_i)}{2\pi k l} \tag{7}$$

$$dT_m = \frac{h_i(T_{si} - T_m)dA}{\dot{m}c} \tag{8}$$

$$q'' = h_i(T_m - T_{si}) \tag{9}$$

The temperature profile along the tube, obtained from this analysis is shown in Figure 8. The calculated temperature profile has a pseudo-exponential form which emphasizes that in a water heating system the temperature profile is neither linear nor exponential. With the temperature profile along the flow direction and the flow rate, the average nucleate boiling heat flux for each segment was determined as a function of wall superheat as shown in Figure 9. It is seen that, the results of the water heating approach the data obtained under the electrical heating condition.

The actual shape of the temperature profile depends on the surface enhancement of the tube, the thickness, length, the thermophysical properties of the boiling fluid, and the amount of heat flux. Figure 10 shows the boiling heat transfer coefficient of the two tubes in an electric heating system. This figure shows the desirable performance of the Turbo-BII tube at low heat fluxes and superheat. The boiling heat transfer coefficient is higher for lower heat fluxes and decreases

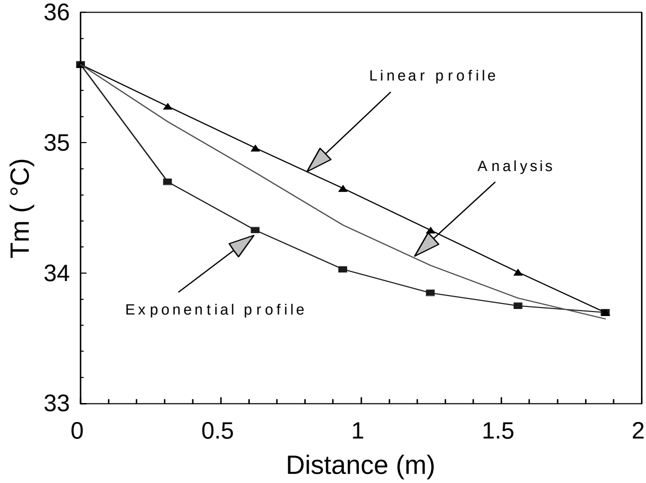

Figure 8. Mean water temperature along the tube

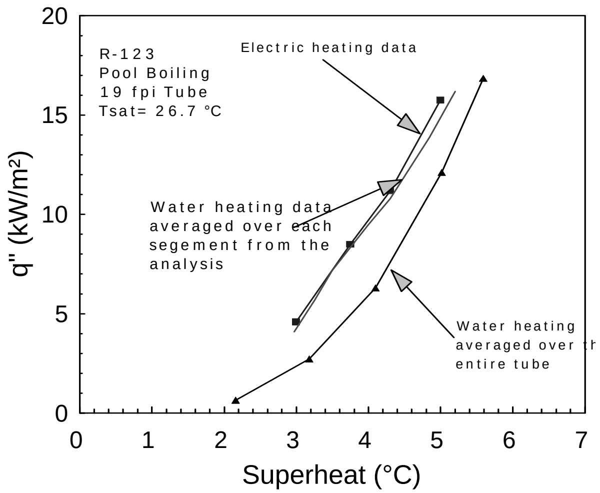

Figure 9. Comparison of the analysis and the electrical heating data

as the heat flux increases. For the 19 fpi tube, which is not recommended as a high performance tube for low heat fluxes, the trend is almost the opposite. The heat transfer coefficient is lower for the lower heat fluxes and has an ascending trend.

This observation clarifies why the water heating for the Turbo-BII tube (Figure 6 is to the left of electric heating. The pseudo-exponential water temperature profile along the tube creates a pseudo-exponential temperature difference between the hot water in the tube and the saturation temperature of the boiling refrigerant. The heat transfer coefficient measured in a water heating system is an average value for the entire tube length, as the local heat transfer coefficient for most of the tube length is at a smaller superheat (due to its exponential nature) and therefore higher heat transfer coefficient. Thus, the overall (average) heat transfer coefficient for the water heating method will be higher than that of the electric method for the same superheat for the

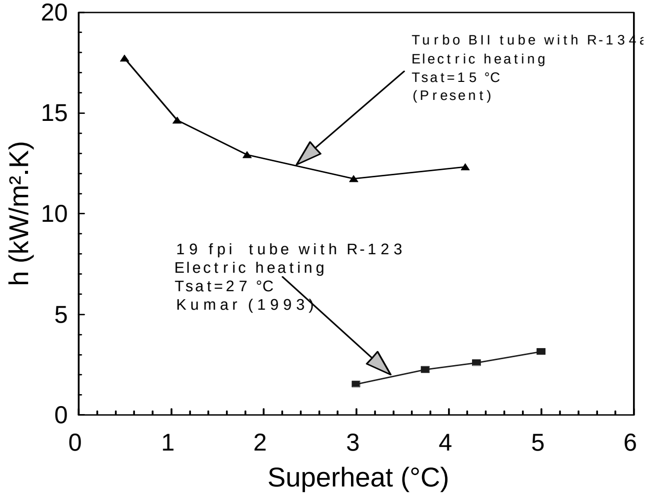

Figure 10. Boiling heat transfer coefficient for Turbo-BII and 19 fpi tubes

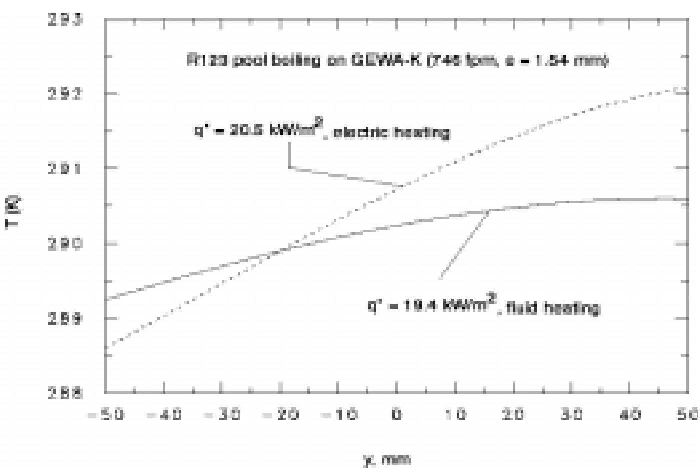

Figure 11. Temperature profile along the test section (Kedzierski 1995)

Turbo-BII tube. Because the heat transfer coefficient of the 19 fpi tube has an ascending trend (as shown in Figure 10), the results of the heat transfer coefficient for the water heating method is lower than that of the electric method. This is indicative of the important role the water temperature profile and the heat transfer coefficient play on a given tube at a certain heat flux.

Differences Found Locally

From the preceding analysis it is evident that the temperature profile at the heated wall influences the local boiling heat transfer for two fluid heated surfaces. Because the electric and fluid heating methods produce different wall temperature profiles for the same heat flux, the boiling heat transfer coefficient produced by the two boundary conditions are different. Figure 11 compares the wall temperature profile at the heated surface for fluid heated and electrically heated data. The measurements were obtained from Kedzierski (1995) for R-123 pool boiling on an enhanced heat transfer surface. The figure illustrates that the wall temperature profile for electric and fluid heating are significantly different even for approximately the same heat flux. Consequently, it should be expected that the different wall temperature profiles would support different boiling heat transfer coefficients for the same heat flux. In the following section, parameters affecting the temperature profile will be discussed.

Parameters Affecting the Temperature Profile

In a practical situation, the process of water heating produces neither constant surface temperature nor a constant heat flux and the temperature profile is the determining factor in the heat transfer process. The temperature profile depends on the inside and outside surface in terms of heat transfer capability, the tube thickness and material, thermophysical properties of the fluid (water), the size of the system, and the boundary conditions. One important parameter is the water velocity as it passes through the tube. In a water heating process, the overall heat transfer coefficient U is a function of both outside and inside heat transfer coefficients h_o and h_i. The inside coefficient h_i is a function of Reynolds number, which depends on the water velocity. By increasing the water velocity, Reynolds number and therefore h_i will increase. As a result h_o and U will change due to the fact that the heat flux and ΔT will change as the water velocity changes. This suggests that in a water heating process the results will vary depending on water velocity. Therefore, these parameters need to be considered before a meaningful comparison between the experimental boiling heat transfer data from various sources can be made.

In addition, there are other possible causes of difference in water heating as well as electric heating:

1. The boiling curve of a higher operating saturation temperature falls to the left of a lower operating saturation temperature. The amount of difference depends on the boiling fluid and the heating surface enhancement.
2. Unal (1994) reported that the boiling curve moved to the right with decreasing wall thickness in constant heat flux, and moved to the left with decreasing thickness for constant temperature heating. Thus the results for different tube thicknesses could vary.
3. A circumferential temperature difference around the tube for more than 1EC has been reported by Webb (1991) and McKee and Bell (1968). This temperature difference, depending on how it was accounted for in the data reduction, could cause a rather substantial discrepancy when comparing the various results.
4. Length, diameter, and boundary conditions also affect the results. Usually in a small test section, the homogeneity of the test conditions is preserved, and the number of measurements per unit length is higher compared to a larger test section where the reported result is typically an average value. Therefore, for the same number of measurements, the small test section may yield more accurate results if precaution is made to prevent any axial heat transfer.
5. Boiling hysteresis exists in most boiling processes due to uncertainties in the nature of the boiling process in terms of nucleation sites, bubble size and frequency. This phenomenon affects the average heat transfer coefficients at low temperature approach.
6. Uncertainties inherent in any experimental measurement could be partly responsible for the difference between the results.

In summary, the issues discussed in this paper help explain some of the causes of discrepancies between the results of various investigations. It is very difficult to provide a direct comparison of phase change data against each other. In practice tube and equipment manufacturers have established their own test rigs to simulate near practical design operation conditions. This problem needs to be dealt with in much greater depth by the research community.

CONCLUSIONS

In this study, the effect of the electric and fluid heating methods on the pool boiling process have been discussed. Several boiling heat transfer data employing water or resistive heating methods by different researchers have been compared and reviewed. The electric heating is a constant heat flux process, whereas the water heating is neither a constant heat flux nor a constant surface temperature process. Water heating method produces a pseudo-exponential temperature profile along the tube. The pseudo-exponential temperature profile along the tube, which depends on the variation of the heat transfer coefficient with heat flux, is postulated to cause the main difference between the results of the two methods. The parameters that shape the temperature profile and other possible causes of differences are given. Until more universal modeling of the problem is achieved, converting the data from one experimental facility to another will need to take into account the uncertainties and parameters discussed in this paper.

NOMENCLATURE

A	heat transfer surface area
A_{TW}	amplitude of wall temperature fluctuation
c	specific heat
D	diameter
fpi	fins per inch
h	heat transfer coefficient
k	thermal conductivity of test section
L	test section length
$\dot{m}$	mass flow rate
p	the perimeter of the tube
q	heat transfer rate
q''	heat flux
T	temperature
U	overall heat transfer coefficient
x	distance along the tube
δ	penetration depth

Subscripts

e	electric heating
f	fluid
i	inside
m	mean
o	outside
s	surface
sat	saturation
w	water heating

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