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# CONJUGATED HEAT TRANSFER OF NUCLEATE POOL BOILING ON A HORIZONTAL TUBE

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**Abstract**—The nucleate pool boiling on horizontal tubes is studied experimentally as a conjugated heat transfer problem. Three non-dimensional parameters, which were originally deduced from the governing energy differential equation in the analysis of conjugated film pool boiling and modified for conjugated nucleate pool boiling, have been used to correlate the experimental data in order to accommodate the effect of the peripheral wall heat conduction on the heat transfer coefficient of nucleate pool boiling.

*Key Words:* conjugated heat transfer, nucleate boiling, film boiling, correlation

## INTRODUCTION

When an asymmetric fluid flow surrounds a solid body of moderate dimension and thermal conductivity, as in the case of a horizontally placed cylindrical heater in pool boiling, heat flows by conduction within the wall of the heater and creates a non-uniform wall surface temperature distribution. The assumption that the surface of the heater is either at constant temperature or at constant heat flux, as implied in many experimental studies, is no longer valid.

The temperature distribution in the solid depends on the temperature distribution in the fluid. Since the circumferential and radial wall temperature and heat flux distributions are not known, this is a heat transfer phenomenon of conjugated nature and the governing energy equations for both the solid and fluid must be solved for the temperature fields according to the interfacial boundary conditions.

Such conjugated heat transfer problems can be found in many single- and two-phase flow conditions. In fact, most of the engineering heat transfer problems are conjugated in nature.

Pool boiling heat transfer has been studied extensively for many years. The effects of fluid and thermal properties, of surface finish and coating, of orientation and geometry of the heater(s), of agitation of the working fluid, of the force field etc., have been investigated and a large number of correlations have been proposed. Many of the existing results on supposedly identical phenomena are, however, inconsistent or differ widely from each other.

It is obvious that to compare the experimental results obtained by different investigators, all parameters governing the heat transfer process should be set equal. Seldom included is the effect of the variation of the surface temperature due to the conjugated nature of the problem. A few studies on boiling heat transfer, however, implicitly recognize this variation of the surface temperature on the surface heat transfer coefficient (Zhukov *et al.* 1970; Berenson 1962; Magrini & Nannei 1975; Sauer *et al.* 1978; Jensen & Jackman 1984), but there seems to be no systematic study on nucleate pool boiling heat transfer which recognizes explicitly the effect of this peripheral wall conduction on the heat transfer coefficient.

An analysis is possible for the local boiling heat transfer coefficient in the conjugated film pool boiling region (Shigechi & Lee 1988), but in the regions of nucleate and transition pool boiling this is not feasible at the moment. Thus, an empirical approach to the problem together with new non-dimensional parameters to characterize the peripheral wall heat conduction, deduced from an analytical study on the film pool boiling on a horizontal heater (Shigechi & Lee 1988) and modified for conjugated nucleate pool boiling, seemed to be appropriate.

These correlating parameters, which must encompass both the solid and fluid, depend on the boundary conditions of a particular conjugated heat transfer mode, e.g. single-phase free or forced convection, nucleate or film pool, or flow boiling, condensation etc.

In the present experimental study, a non-dimensional parameter, $K^* = K_f R/K_w b$ (where $K_f$ and $K_w$ are the thermal conductivities of the fluid and heater, respectively, and $R$ and $b$ are the radius and the wall thickness of the heater, respectively), obtained from the governing energy differential equation (Lee & Kakade 1976), is used to characterize the peripheral wall heat conduction of the heaters. A new parameter, $H^*_{NB}$, deduced from the analysis of film pool boiling by Shigechi & Lee (1988) and then modified for nucleate pool boiling was used to correlate the final results.

## EXPERIMENTAL

### *Apparatus*

The experimental apparatus consisted primarily of a vertical boiling vessel of dimensions 420 × 280 × 380 mm containing an electrically heated horizontal test heater. The working fluid was 99% pure commercial grade freon-133.

Seven test cylinders (direct electrical resistance heating) made of stainless steel 304 with five different wall thicknesses ($b = 0.81$, 1.02, 1.24, 1.65 and 2.87 mm, respectively), one made of monel 400 ($b = 2.92$ mm) and the other inconel 600 ($b = 2.28$ mm), were used to vary the values of the parameter $K^*$ between 0.014 and 0.075. The outside diameters of the test cylinders were kept at about 25 mm to maintain a constant heater diameter and blockage ratio so that the hydrodynamic or fluid flow pattern around the test-section would be identical for each test heater. All test cylinders were 178 mm in length.

Nine chromel–alumel thermocouples (K-type, 24 G) were spot-welded onto the centre of the inner wall periphery of the test heaters. Several additional thermocouples were positioned on the inner wall 13–26 mm away from the centre of the test heater to check the uniformity of the temperature distribution. One stainless steel 304 test heater had its thermocouples installed using the "split-junction" method (Lee & Kakade 1976) to test the accuracy of the spot-welded thermocouples. Two welding methods showed almost no difference in reading wall temperature. The circumferential wall temperature distribution was measured at every 30° interval for one half side and every 60° for the other half side. All thermocouples were calibrated *in situ*.

Great care was taken to maintain each test heater surface at the same roughness by polishing it with grit sizes 80, 180 and 320 silicon-carbide emery paper, respectively.

The test heaters were heated by a 15 kV, 1200 A a.c. power supply. The electrical power input to the test heater was measured directly with two leads embedded in the cylinder 50.5 mm apart in the central portion by a digital voltmeter and an ammeter through the measuring circuits. This eliminated the uncertainty in estimating the electrical lead losses and, thus, little error was introduced in calculating the heat flux. The probable maximum error in the wall heat flux was estimated to be about 4%.

### *Experimental procedure and data acquisition*

All tests reported here were conducted at atmospheric pressure.

The liquid preheated to a pre-set temperature in the storage tank was charged to a level of about 70 mm above the top of test heaters. The power to the heater was then turned on at a relatively high level for about half an hour to let the liquid boil off the dissolved non-condensible gases. The power to the heater was then adjusted to the desired level to start the test.

All the test data were acquired and reduced through a Hewlett-Packard data acquisition system consisting of: a 9835A computer; a 3455A digital voltmeter; a 3495A scanner; and a 7245A printer/plotter.

Since the thermocouples were spot-welded onto the inside tube wall, the solution of the "inverse" heat transfer problem was required in order to obtain the outside surface temperature. By solving the two-dimensional steady-state heat conduction equation with heat generation, the outside temperature can be obtained. The experimental results showed that the ratios of the second derivatives of the circumferential temperature distributions to those of the radial direction for the range of the present study were of the order of $10^{-4}$. Therefore, the outside wall temperature can be closely estimated by the equation

$$T_o = T_i - \frac{\bar{q}}{2bK_w}\left[\frac{1}{2}(r_o^2 - r_i^2) + r_i^2 \ln\frac{r_i}{r_o}\right], \qquad [1]$$

where $T$ and $r$ are the temperature and radial coordinate, respectively, the subscripts i and o stand for the inside and outside of the heater, respectively, and $\bar{q}$ is the average wall heat flux. Because of circumferential heat conduction, even with electrical resistance heating, the condition of constant heat flux is no longer applicable. Therefore, the local heat transfer coefficient, $h$, should be deduced from [2] which was obtained from an energy balance made on the element of the test heater wall:

$$h = \frac{\dfrac{d^2 T_w}{d\theta^2} + \dfrac{\dot{q} R^2}{K_w}}{\left(\dfrac{R^2}{K_w b}\right)(T_w - T_s)} - \epsilon\sigma\,(T_w^2 + T_s^2)\,(T_w + T_s), \qquad [2]$$

where $\dot{q}$, $\theta$ and $R$ are the specific heat generation rate, angle coordinate and radius, respectively, the subscripts w and s stand for the wall and saturation, respectively, $\epsilon$ and $\sigma$ are the emissivity of the heater surface and the Stefan–Boltzmann constant, respectively. The values of $(d^2 T/d\theta^2)$ were determined numerically from the experimental results of the circumferential temperature distribution.

## RESULTS AND DISCUSSION

Because the material of the heating surface has a strong effect on nucleate boiling, only the stainless steel 304 tubes with different values of $K^*$ are discussed in this paper.

An example of the circumferential surface temperature of the test cylinders for five values of $K^*$, is shown in figure 1. Similar plots showed that there was no significant difference between temperatures at symmetric angles around the circumference.

The effect of $K^*$ on the local heat transfer coefficients can be seen in figure 2. It is clear that the heat transfer is generally more efficient in the bottom portion than in the top.

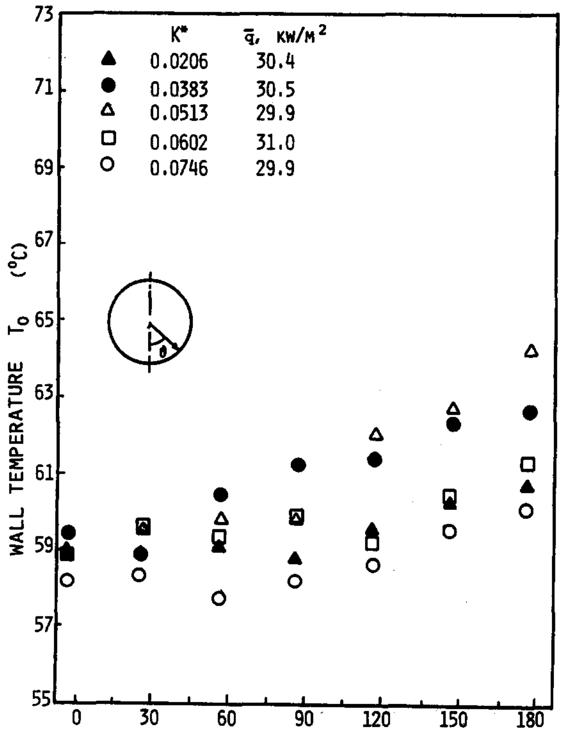


Figure 1. Wall temperature distribution.

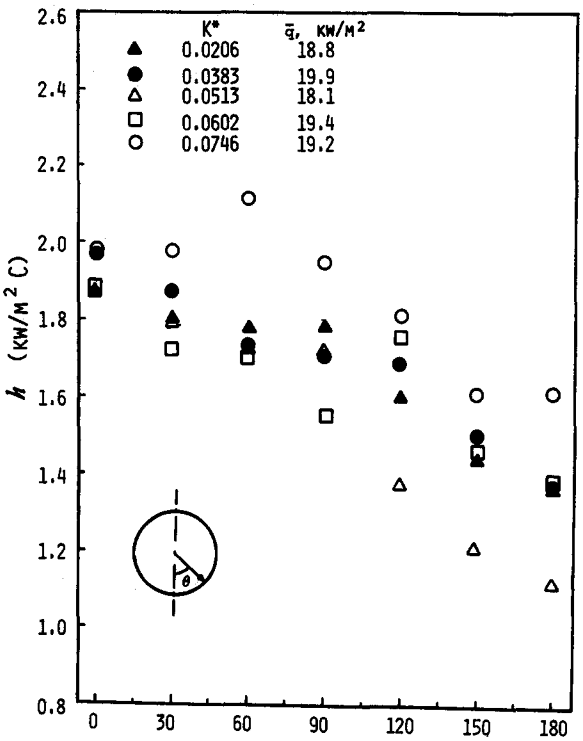


Figure 2. Local heat transfer coefficient.

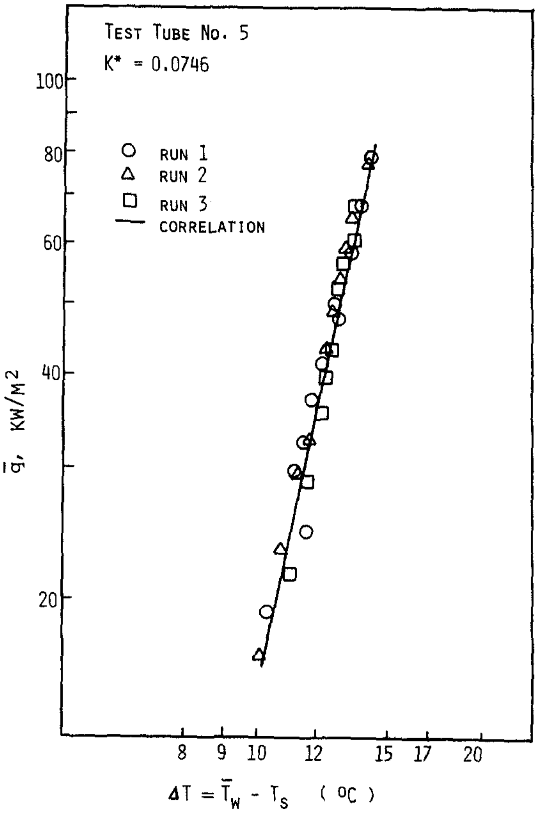


Figure 3. Reproducibility of data.

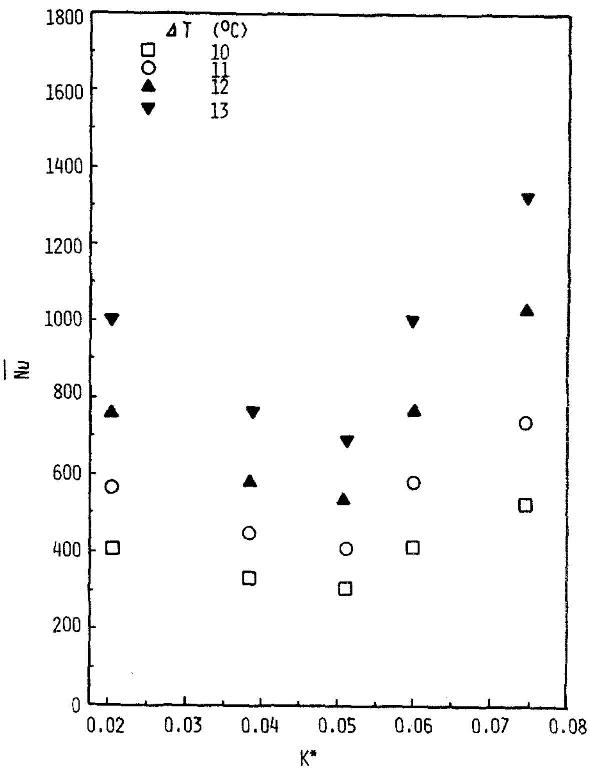


Figure 4. Effect of $K^*$ on $\overline{\mathrm{Nu}}$.

Figure 3 represents typical average heat transfer rates and the reproducibility of the test results is excellent. The line of experimental correlation shown in the figure was obtained by least-squares regression analysis and from these lines information such as shown in figure 4 is deduced.

A large value of $K^*$ implies a poor conductor and should add a large circumferential surface temperature variation. But the experimental results show that this is not necessarily so. It seems there exists a critical value of $K^*$ at which the resulting heat transfer rate becomes a minimum, as can be clearly seen in figure 4.

The existence of a critical $K^*$ may be explained from the governing energy equation, which is given in dimensionless form as

$$\frac{\partial^2 T^+}{\partial r^{+2}} + \frac{1}{r^+}\frac{\partial T^+}{\partial r^+} + \frac{1}{r^{+2}}\frac{\partial^2 T^+}{\partial \theta^2} + \frac{1}{2}K^*\overline{\mathrm{Nu}} = 0, \qquad [3]$$

where $T^+$ and $r^+$ are defined as $(T - T_s)/(T_w - T_s)$ and $r/R$, respectively, Nu is the Nusselt number and the overbar stands for "average". It should be noted that $K^*$ is not the only parameter which governs the temperature distribution. The boundary condition at the outside wall as well as $\overline{\mathrm{Nu}}$ affect the surface temperature distribution. Since they are coupled to each other, the dependence of the surface temperature distribution on $K^*$ could not be simply monotonic.

The hypothesis that there exists a critical $K^*$ seems to agree well with the results of two contradicting studies on nucleate pool boiling made by Sharp (1964) and Magrini & Nannei (1975). In the study by Sharp, heat transfer decreased with increasing values of $K^*$ (possibly in the range $K^* < K^*_{\mathrm{crit}}$ of the present study; the subscript "crit" stands for "critical"), but Magrini & Nannei reported that heat transfer increased with increasing values of $K^*$ (possibly in the range $K^* > K^*_{\mathrm{crit}}$ of the present study).

Our plausible explanation on the existence of the critical $K^*$ is as follows.

The visual study on boiling phenomena by Bartolino & Fantini (1973) indicated that with decreasing heater wall thickness, there was a sharp increase in the bubble population on the heater surface at a given surface heat flux.

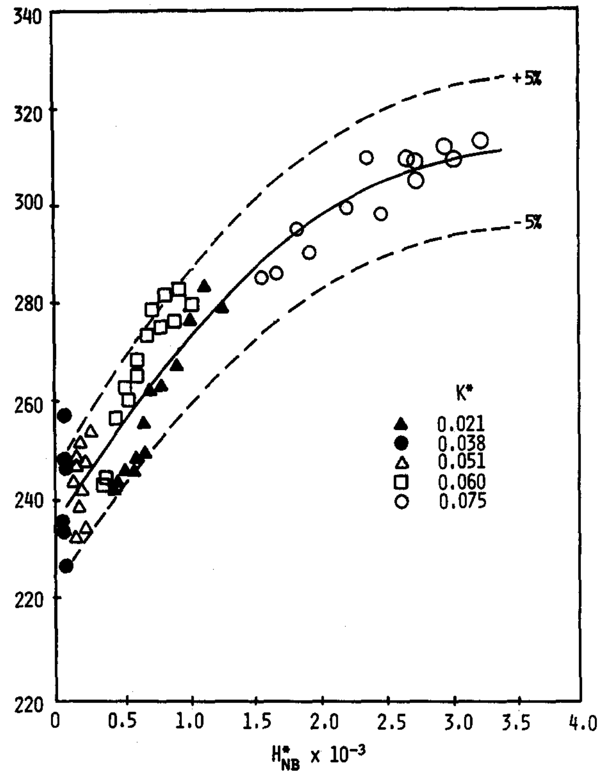


Figure 5. A new correlation of conjugated nucleate pool boiling.

For a thinner wall, the specific heat density, $\dot{q}$, is larger than that in a thicker wall for a given surface heat flux because $\bar{q} = \dot{q} \cdot b$. If conditions other than the heater wall thickness, $b$, are set equal, the effect of $K^*$ on nucleate boiling is equivalent to the effect of wall thickness on nucleate pool boiling. Since the specific heat density in a thinner wall is larger than that in a thicker wall for a given surface heat flux ($\bar{q} = \dot{q} \cdot b$), the thinner wall would have a larger bubble population in light of the finding of Bartolini & Fantini (1973). The higher bubble population would produce a stronger stirring motion near the wall surface and hence result in a higher heat transfer rate at the same surface heat flux. A small value of wall thickness makes a higher value of $K^*$, and therefore the heat transfer rate increases with increasing values of $K^*$ in the range of $K^* > K^*_{\text{crit}}$.

On the other hand, when the wall thickness is large, the transverse heat flux directed to the base of a bubble from the surrounding area would become large because of low thermal resistance to the transverse flux. In this case, the growing bubble extracts heat not only from the wall beneath the bubble base but also from the surrounding area of the bubble. Consequently, the bubble on the thicker wall could extract more heat than that on the thinner wall and therefore, the total heat transfer rate would be higher. A larger wall thickness results in a smaller value of $K^*$ and hence the heat transfer rate increases with decreasing values of $K^*$, or, the heat transfer rate decreases with increasing values of $K^*$ in the range of $K^* < K^*_{\text{crit}}$.

If the wall thickness happens to be the value at which both the effects discussed above are the least, then the lowest heat transfer rate would result and this critical wall thickness should correspond to the critical $K^*$.

Regardless of whether or not the above explanation is probable, the added asymmetry of the thermal boundary conditions, due to the value of $K^*$, affected the average heat transfer rate up to a maximum of about over 100% for the ranges of $K^*$ and heat flux studied in the nucleate pool boiling region.

*A new correlation for conjugated nucleate pool boiling*

The classical solution of Bromley (1950) for *film* pool boiling does not recognize the variation of wall surface temperature. The present authors made an analysis for conjugated *film* pool boiling from a horizontal heater with uniform heat generation (Shigechi & Lee 1988). The assumptions made in the analysis are identical to those of the Bromley solution except that the temperature field in the fluid is coupled with the two-dimensional temperature distributions on the heater wall. The solution for the governing energy equations for the two-phase flow and the solid wall requires the conditions of equality in temperature and heat flux at the interface between the solid and fluid flow.

The problem is solved by the integral method. The integration of the governing energy and momentum equations for the tube wall and for the vapour film with the appropriate boundary conditions, together with necessary velocity and temperature profiles, yields two ordinary differential equations. Introducing new dimensionless parameters, the governing differential equations are made non-dimensional and solutions are obtained. From the results of this analysis, it was attempted to deduce a new correlation for conjugated nucleate pool boiling which would shed light on some of the scatter in the data observed among many different pool boiling studies.

**Deduction**

(a) For the conjugated *film* pool boiling from a horizontal heater, it was shown in the above mentioned analysis that there exists a relationship

$$\frac{\mathrm{Nu}_{\mathrm{FB}}}{2B_{\mathrm{FB}}^{-1/3}} = fn_1(H_{\mathrm{FB}}), \qquad [4]$$

where

$$\mathrm{Nu}_{\mathrm{FB}} = \frac{2Rh}{K_{\mathrm{G}}}, \qquad [5]$$

$$H_{\mathrm{FB}} = \left(\frac{RK_{\mathrm{G}}}{bK_{\mathrm{w}}}\right) B_{\mathrm{FB}}^{-1/3} = K^* B_{\mathrm{FB}}^{-1/3} \qquad [6]$$

and

$$B_{\mathrm{FB}} = \frac{\mu_{\mathrm{G}}}{g(\rho_{\mathrm{L}} - \rho_{\mathrm{G}})\,\rho_{\mathrm{G}} h_{\mathrm{LG}}} \frac{\dot{q}b}{R^2}. \qquad [7]$$

The subscripts FB, L and G stand for film boiling, liquid and vapour or gas, respectively, $\rho$ and $\mu$ are the density and viscosity, respectively, and $h_{\mathrm{LG}}$ and $g$ are the latent heat of vaporization and gravity acceleration, respectively.

(b) For conjugated *nucleate* pool boiling, although the physical aspects involved are quite different from that of film pool boiling, it is reasonable to assume that there may exist a similar relationship (with appropriate parameters to accommodate the different physical aspects):

$$\frac{\mathrm{Nu}_{\mathrm{NB}}}{2B_{\mathrm{NB}}^{-m}} = fn_2(H_{\mathrm{NB}}) = F, \qquad [8]$$

where

$$\mathrm{Nu}_{\mathrm{NB}} = \frac{2Rh}{K_{\mathrm{L}}} \qquad [9]$$

and

$$H_{\mathrm{NB}} = \left(\frac{RK_{\mathrm{L}}}{bK_{\mathrm{w}}}\right) B_{\mathrm{NB}}^{-m} = K^* B_{\mathrm{NB}}^{-m} \qquad [10]$$

The subscripts NB and L stand for nucleate boiling and liquid, respectively.

Now, the new boiling parameter, $B_{\mathrm{NB}}$, which must be able to accommodate the different physical aspects, i.e. nucleate pool boiling, and the exponent $m$ can be deduced from Rohsenow's (1952) correlation for nucleate pool boiling, given as

$$\frac{c_{\mathrm{L}}(\bar{T}_{\mathrm{w}} - T_{\mathrm{s}})}{h_{\mathrm{LG}}} = C_{\mathrm{sf}} \left\{ \frac{\bar{q}}{\mu_{\mathrm{L}} h_{\mathrm{LG}}} \left[ \frac{\sigma}{g(\rho_{\mathrm{L}} - \rho_{\mathrm{G}})} \right]^{0.5} \right\}^{1/3} \mathrm{Pr}_{\mathrm{L}}^{n}, \qquad [11]$$

where $n = 1$ for water and $n = 1.7$ for all other liquids. $C_{\mathrm{L}}$ is the specific heat of the liquid, $C_{\mathrm{sf}}$ is Rohsenow's constant, Pr is the Prandtl number and $\sigma$ is the surface tension. Since $\bar{h} = \bar{q}/(\bar{T}_{\mathrm{w}} - T_{\mathrm{s}})$

and $\bar{q} = \dot{q} \cdot b$, [11] can be rewritten as

$$\bar{h} = \frac{1}{C_{sf}} \left(\frac{K_L}{R}\right) \left\{ \frac{\dot{q}^2 b^2 R^3}{\mu_L^2 h_{LG}^2} \left[\frac{g(\rho_L - \rho_G)}{\sigma}\right]^{1/2} \mathrm{Pr}_L^{3(1-n)} \right\}^{1/3}. \quad [12]$$

Since Rohsenow's constant, $C_{sf}$, depends on the nature of the heating surface–fluid combination, we can expect that there may exist the following functional relation:

$$\mathrm{Nu}_{NB} \sim \left\{ \frac{\dot{q}^2 b^2 R^3}{\mu_L^2 h_{LG}^2} \left[\frac{g(\rho_L - \rho_G)}{\sigma}\right]^{1/2} \mathrm{Pr}_L^{3(1-n)} \right\}^{1/3} \frac{1}{C_{sf}}. \quad [13]$$

If we let the factor $(1/C_{sf})$ be included in the fuction $F$, we can now obtain for the parameter $B_{NB}$, from [8] and [13]:

$$B_{NB} = \frac{\mu_L^2 h_{LG}^2}{\bar{q}^2 R^3} \left[\frac{\sigma}{g(\rho_L - \rho_G)}\right]^{1/2} \mathrm{Pr}_L^{-3(1-n)}. \quad [14]$$

Now that $B_{NB}$ is defined, the value of $m$ can be deduced as $+1/3$ by comparing the non-dimensionalized governing equation for *film* pool boiling (Shigechi & Lee 1988) for the case of an isothermal surface [i.e. $F(H) = \text{const}$], with Rohsenow's correlation for *nucleate* pool boiling, [12] above.

In the present experimental study of heaters made of different materials, the constant, $C_{sf}$, is also seen as a function of $K^*$. Therefore, from results similar to those plotted in figure 4, we modified the parameter $H_{NB}$, [10], as follows:

$$H_{NB}^* = (K^* - K_{crit}^*)^2 B_{NB}^{-1/3}. \quad [15]$$

Our experimental results show that the critical value of $K^*$ is about 0.043.

The final equation of the new correlation for the experimental data of the present study is

$$F - 237 + 42.3\,(H_{NB}^* \cdot 10^3) - 6.0\,(H_{NB}^* \cdot 10^3)^2 \quad [16]$$

The comparison of the new correlation with the experimental data is shown in figure 5; the coefficient of correlation was 0.949. It can be seen that the experimental data fall within $\pm 5\%$ of the correlation.

## CONCLUDING REMARKS

The results from the present study lead to the conclusions that the parameters $K^*$ and $H_{NB}^*$ have a significant effect on the nucleate pool boiling heat transfer process, and that these parameters should be included in the correlation for such conjugated boiling heat transfer phenomena.

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