EFFECT OF SURFACE FINISH ON FLOW BOILING HEAT TRANSFER

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ABSTRACT

It is well known that the surface structure affects the pool boiling heat transfer from a heater surface. The number and size distribution of cavities present on a heater surface affect the nucleation characteristics. This fact is utilized in developing structured and sintered surfaces for enhanced boiling performance. The nucleate boiling component in flow boiling is also expected to exhibit a somewhat similar dependence.

The present study investigates the flow boiling heat transfer of subcooled water over heater surfaces prepared with different roughness characteristics. The setup consists of a 9.5 mm diameter circular heater placed on the lower wall of a 3 mm x 40 mm horizontal channel. Four different surface finishes are tested on the same heater unit. Each surface is observed under a microscope, and analyzed with an image processing software to obtain the cavity size distribution. Relationships between the cavity size distribution and the heat transfer are then compared for the four surfaces.

The results indicate that the wall superheats for the four surfaces tested were within 0-30 percent of each other for a given heat flux. Although the roughest surface generally performed the best, no definite trend could be observed. It is also noted that the cavity size distribution of a surface is not significantly affected by simple polishing techniques. A commercially prepared sintered surface is also tested. Its performance was markedly above the four roughened surfaces. However, the sintered surface exhibited a large hysteresis effect, while the four roughened surfaces showed little hysteresis under the test conditions.

NOMENCLATURE

A	surface area, m^2
d	tube diameter, m
d_H	hydraulic diameter, m
h	enthalpy, J/kg
h_{lv}	latent heat of vaporization, J/kg
n	number or cavities; n/A – cavity density
p_g	pressure (vapor phase), Pa
p_l	pressure (liquid), Pa
q''	heat flux , W/m^2
r	bubble radius, m
r^*	non-dimensional cavity radius, eq. (4)
r_c	cavity radius, m
Re	Reynolds number, $\rho V d_H/\mu$
T	temperature, K
ΔT_{sat}	T_{wall}-T_{sat} , K
ΔT_{sub}	T_{sat}-T_{bulk} , K
T_l	liquid temperature, K
T_w	wall temperature, K
V	fluid velocity, m/s
v	specific volume, m^3/kg

Greek Symbols

α	heat transfer coefficient, W/m^{2}K
α_l	single-phase heat transfer coefficient with liquid, W/m^{2}K
δ_t	thermal boundary layer thickness, m
λ_l	thermal conductivity of liquid, W/m K
μ	kinematic viscosity, Ns/m^2
ρ	density, kg/m^3
σ	surface tension, N/m

Subscripts

b	bulk
c	cavity
g	vapor
l	liquid
sat	saturated state
sub	subcooled state
t	thermal
w	wall

INTRODUCTION

Flow boiling heat transfer consists of the convective and the nucleate boiling components. Each of these components is affected by many parameters. It is well established that the surface condition of the boiling surface affects the pool boiling heat transfer. Surface conditions such as microfins, structured surfaces, and artificial cavities can greatly influence the onset of nucleate boiling and the overall heat flux. Enhancement techniques such as sintering and specially fabricated surface geometries have been used to improve the heat transfer characteristics.

There are a number of experimental studies reported on flow boiling heat transfer. Based on these results, general correlations, such as Kandlikar (1991) have been developed for predicting heat transfer coefficient. In comparing the experimental data from various investigators, a large scatter of about 10-30 percent is observed under similar test conditions. This scatter is generally attributed to the experimental uncertainties. However, since the tubes used in these experiments are different, the differences in their surface characteristics are expected to influence their performance. To ascertain this effect, the present study focuses on measuring the influence of surface roughness on flow boiling heat transfer with water.

REVIEW OF LITERATURE

Many investigators have reported the effects of surface finish on the pool boiling curve. As early as 1931, Jakob and Fritz (Jakob, 1949) were investigating the effect of surface finish on the pool boiling curve. They used both a sandblasted surface and a surface with0.016 mm square machined grooves spaced at 0.48 mm. Both of these surfaces showed improved boiling performance. Corty and Foust (1955) obtained similar results with grit roughened surfaces.

A good review of this topic is provided by Webb (1981). Kurihari and Myers (1960) worked with organic fluids boiling on roughened copper surfaces. They showed that boiling performance was related to the density of active nucleation sites, (n/A), and obtained a correlation showing $\alpha \propto (n/A)^{0.43}$, where n/A is nucleation site density. Berenson (1962) obtained a 600% increase when boiling pentane on a lapped copper surface. Marto et al. (1968) presented results showing the effects of surface roughness on the pool boiling heat transfer with liquid nitrogen. Surfaces with just mirror-finish, and with cavities of various sizes drilled on the mirror-finish were tested. In general, the surfaces with the cavities exhibited a better heat transfer performance over just the mirror finish surface. Rohsenow (1985) shows the changes in the pool boiling curve with different surface finishes.

Gottzmann et al. (1971, 1973) conducted experiments with a High Flux copper sintered surface, and observed significant improvement in its heat transfer performance over a plain tube. The improvement was caused by the porous structure, which trapped more vapor-liquid interfaces with large radii. This would require much less wall superheat for nucleation. Also, a porous structure such as the High Flux surface would have a greater surface area. Wadekar (1996) reported an order of magnitude increase in flow boiling heat transfer with a UOP High Flux tube.

Heater surface structure modification has been employed successfully in enhancing the flow boiling performance of microfin tubes. However, the heat transfer enhancement in the microfin tubes is quite complex, improving both convective and nucleate boiling components. Kandlikar and Howell (1996) examined the effect of microfins on flow boiling with flat microfin heater surfaces placed in a rectangular channel. They reported an increase in bubble activity on the microfin surface when compared to a plain surface.

The onset of nucleate boiling is governed by the sizes of cavities available on a heater surface. Hsu and Graham (1961), and Hsu (1962) presented the theoretical range of active cavities that should nucleate under a given set of conditions. They considered a truncated bubble with 53.1° contact angle. Sato and Matsumura (1964) derived the nucleation criterion by considering a hemispherical shaped bubble. Following their analysis, the Helmholtz relation gives the excess pressure inside a hemispherical bubble of radius r as

$$p_g - p_l = 2s \ / \ r \qquad (1)$$

Assuming a linear profile, the temperature of the liquid, T_l, at a distance y in the boundary layer of thickness δ_t is given by-

$$T_l = T_w - \frac{y}{d_t}\left(T_w - T_b\right) \qquad (2)$$

T_w and T_b are the wall and bulk liquid temperatures. The condition for a bubble to grow on a given cavity is then obtained by equating the liquid temperature at the bubble top surface with the saturation temperature inside the bubble. The range of active cavities is obtained by solving the resulting quadratic equation and the following result is obtained in the non-dimensional form (Kandlikar and Spiesman, 1997, and Kandlikar, 1997).

$$r^*_{\max}, r^*_{\min} = \frac{1}{2}\left[\frac{\Delta T^*_{sat}}{\Delta T^*_{sub} + \Delta T^*_{sat}} \pm \sqrt{\left(\frac{\Delta T^*_{sat}}{\Delta T^*_{sub} + \Delta T^*_{sat}}\right)^2 - \frac{1}{\left(\Delta T^*_{sub} + \Delta T^*_{sat}\right)}}\,\right] \qquad (3)$$

where

$$r^* = r / d_t \qquad (4)$$

$$\Delta T^*_{sat} = \Delta T_{sat} h_{lv} d_t \ / \left(8s\, T_{sat} v_{lv}\right) \qquad (5)$$

$$\Delta T^*_{sub} = \Delta T_{sub} h_{lv} d_t \ / \left(8s\, T_{sat} v_{lv}\right) \qquad (6)$$

The wall superheat is defined as $DT_{sat} = T_w - T_{sat}$ and the localized liquid subcooling is defined as $DT_{sub} = T_{sat} - T_f$. The boundary layer thickness δ_t can be calculated approximately as $d_t = l_l/a_l$ where a_l is the convective heat transfer coefficient from the surface to the liquid calculated by $a_l = q'' / (DT_{sat} + DT_{sub})$. Variations of the above equation are presented by later investigators to account for the contact angle and the distortion in the temperature profile in the presence of a bubble (Davis and Anderson, 1966, Kenning and Cooper, 1965, and Kandlikar et al., 1997).

OBJECTIVES OF THE PRESENT WORK

The objective of the present work is to investigate the effect of surface finish on flow boiling heat transfer with subcooled water.

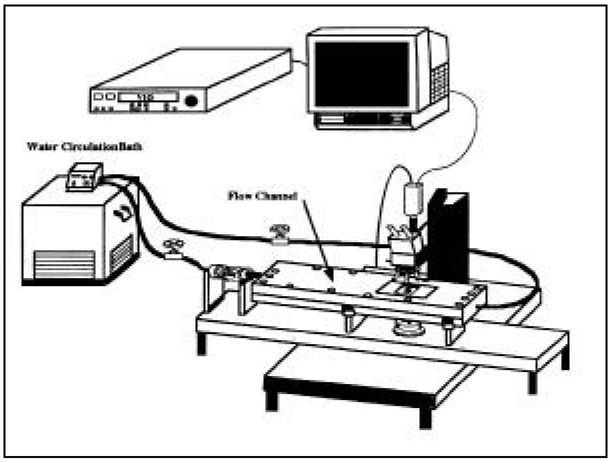

Figure 1. Experimental Setup

Experimental data will be obtained with four different surface finishes on the same heater. Each of the surfaces will be analyzed using a surface roughness tool as well as an imaging software program. Using these techniques, the cavity size distribution for the four surfaces will be obtained. The relation between the surface structure and the heat transfer performance will then be investigated. For comparison purposes, a commercially sintered surface will also be tested.

EXPERIMENTAL SETUP

The experimental setup used was similar to that of Kandlikar and Howell [1996] and is shown in figure 1. The test section consists of a 3mm x 40mm channel fabricated from 6061-T6 aluminum. It was equipped with a two piece polycarbonate window above the heater for viewing through a microscope attached to a camera. The heaters were machined out of 2024-T3 aluminum and had four E-type thermocouples along the length of the 9.4 mm rod. The heater was heated with a Watlow circumferencial electrical resistance heater at the base of the aluminum element. Figure 2 shows the details of the heater and the spacing of the thermocouples. The thermocouples were connected to the Keithley 740 System Scanning Thermometer. A torlon bushing was used to hold the heater in the test section as well as to insulate the heater from the rest of the flow section. Water was circulated through the test section by a Brinkman RC 20 constant temperature bath. It maintained the water at a constant temperature within ∀0.02 K. Atmospheric pressure was maintained in the test section by the flow control valves located at the inlet and the outlet of the flow loop. The flow rate was monitored by an Omega FL-1503A rotameter with a maximum flow of 2.53 GPM.

ERROR ANALYSIS

The heater surface temperature was determined by conducting a finite difference analysis of the heater assembly with the known thermocouple temperatures as the input. The thermocouples were calibrated, and the readings were accurate to within ∀0.1 °C. The

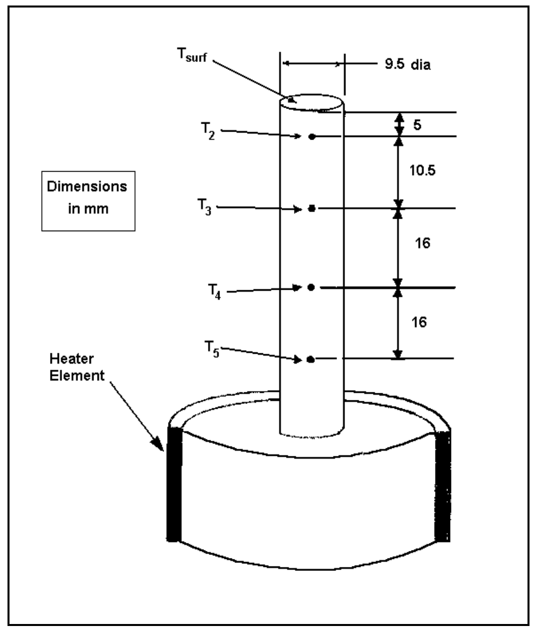

Figure 2. Details of Heater Section

heater surface temperature was estimated to within ∀0.15 °C, and the heat flux was estimated to be accurate to within ∀8.5% by considering the effect of errors associated with thermocouple placement, and the heat flux estimate along the heater rod. The effect of other variables such as the thermal entry region over the heater surface was minimized by using the same heater and applying different surface finishes to it. The relative performance of the four surfaces tested is believed to be more accurate than the heat flux estimates.

Another source of error was that the preparation of each surface left inherent scratches in a preferred direction. To deal with this, each of the four surfaces were placed in the flow section so that these scratches were at a 45° angle with the direction of the flow. This allowed in keeping the effect of the surface structure orientation to a minimum.

EXPERIMENTAL PROCEDURE

The heater surfaces were prepared as follows. Three of the four surfaces were prepared with different grit wet-dry silicon carbide grinding papers. The fourth surface was polished with 1 μm silica particles in a water solution and then with 0.03 μm silica particles in a water solution to give it a mirror-like finish. The same heater was used to minimize the relative errors in the heat transfer calculations. By doing so, the thermocouple locations and the heater geometrical variables did not influence the relative performance. Table 1 shows the surface roughness calculations for the four surfaces that were tested and their respective average roughness values measured with a FEDERAL PocketSurf® I Portable Surface Roughness Gage. Multiple readings were taken on each surface to get an average roughness value.

Table 1. Roughness Characteristics of Test Surfaces

Surface	Preparation Method	Average Roughness (μm)
A-188	0.03 μm soluton	0.188
B-363	600 grit	0.363
C-716	400 grit	0.716
D-3064	120 grit	3.064
E-UOP	UOP High Flux Surface	NA, sintered layer thickness
		225-300 μm, porosity 40-60%

The Image-Pro PLUS™ software was used to analyze the surfaces. After a surface was prepared, the images were obtained with the help of an image grabber board. Using the counting feature of the software, an approximate count of cavities could be obtained. Since the images obtained were black and white, the cavities appeared as dark spots. The software was used to measure the average diameter of each dark spot. With this data, the size distribution of cavities was plotted. Prior to each count, the image was calibrated for the proper magnification. This was accomplished by placing a slide with tic marks at known spacing under the same magnification and obtaining an image. Then using the software package, the image could be calibrated for that magnification.

Before taking any data, the constant temperature water bath was heated to about 98°C. It was then allowed to circulate at this temperature for 3 to 4 hours to ensure that the water was as de-gassed as possible. The bath was then allowed to cool to the desired degree of subcooling and the desired flow rate was adjusted. The flow rate was adjusted by both changing the pumping rate and by adjusting the two valves connected to the test section. These two valves also regulated the pressure inside the test section. The pressure was set at one atmosphere. The power supply was then turned on which gave a heat flux through the heater. After the whole system reached a steady-state temperature, the readings were taken. After all pertinent information was recorded, the heater power input was increased by increasing the voltage supplied to the resistance wrap heater. The voltage was usually raised in 5V increments. This caused an increase of about 0.07 amps in the current through the heater resulting in a 0.35 W increase in power for each step. In the cases where decreasing heat flux data were also obtained, the decreasing step size was varied between 2V and 5V.

From the four thermocouple readings along the length of the heater and the thermal resistance of the heater material, the heat flux through the heater could be determined. Then the surface temperature of the heater and the heat transfer coefficient could be determined from the heat flux.

RESULTS AND DISCUSSION

The ranges of active cavity sizes for the given flow conditions were obtained by employing eq. (3). The results are shown in Table 2. It shows the theoretical ranges of cavity sizes that should nucleate for two different flow rates and various wall superheat values. These two cases correspond to the two flow conditions for which the results are presented.

The average surface roughness values for the four surfaces are shown in Table 1. Figures 3–6 show the actual cavity distributions for these surfaces obtained in this work. In each plot, the numbers of cavities present in each 0.5 μm range are plotted.

Table 2. Theoretical Range of Active Cavity Diameters

Theoretical Active Cavity Diameters (all dimensions μm)		
Wall Superheat	Re = 2253	Re = 7163
2.6°	ONB @ 51	None
3.4°	24 - 98	ONB @ 39
5°	14 - 136	16 - 82
10°	7 - 194	7 - 122
12.5°	5 - 212	5 - 134
15°	4 - 222	4 - 141

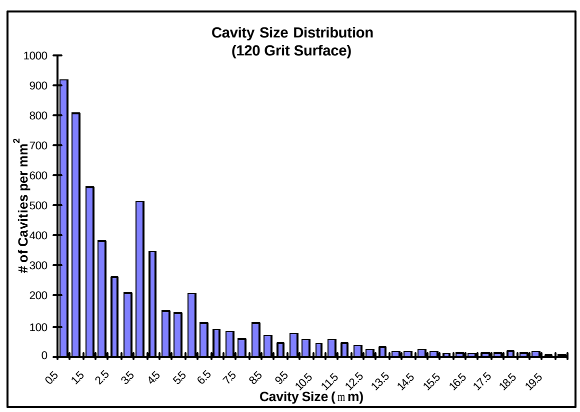

Figure 3. Cavity distribution for surface D-3064

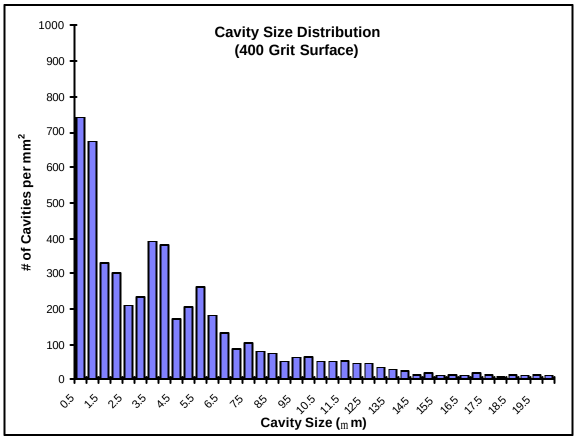

Figure 4. Cavity distribution for surface C-716

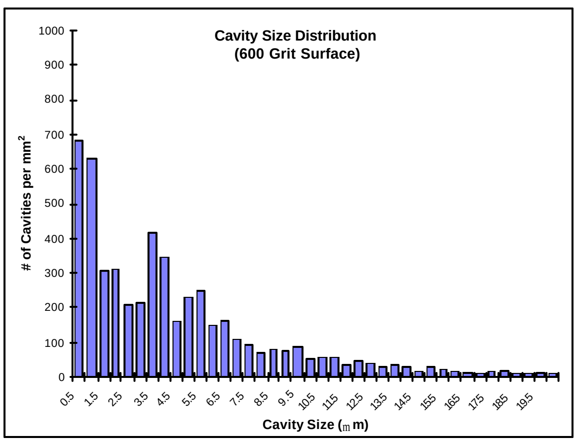

Figure 5. Cavity distribution for surface B-363

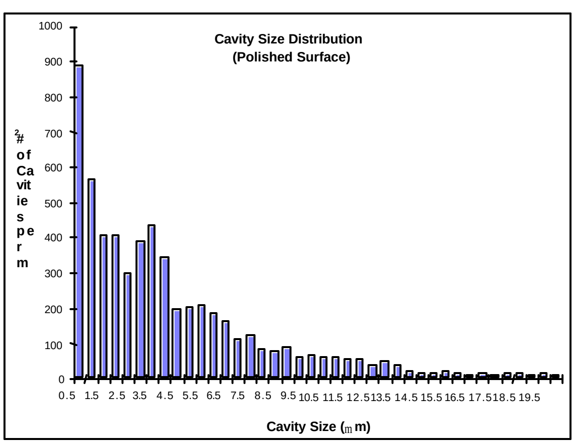

Figure 6. Cavity distribution for surface A-188

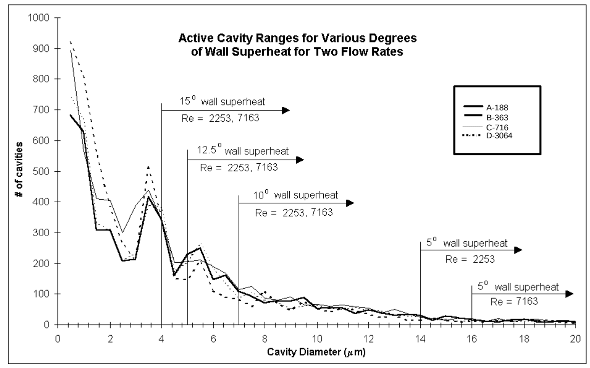

Figure 7. Comparative cavity distributions

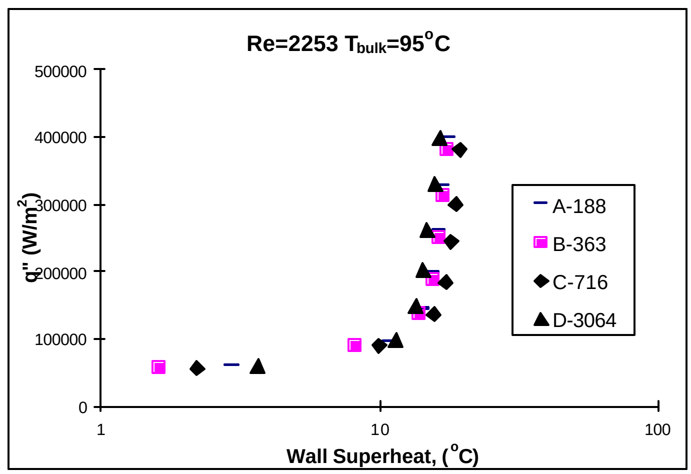

Figure 8. Performance of four surfaces at Re=2253

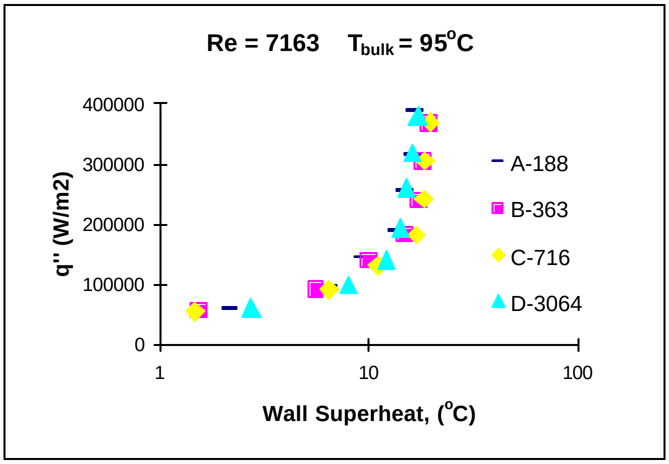

Figure 9. Performance of four surfaces at Re=7163

The cavity size distribution curves presented in Figs. 3-6 are compared with each other as well as with the ranges in Table 2. The results of comparison are presented in Fig. 7. From this graph, the relative performance for the surfaces can be estimated. The surface with larger number of active cavities under a given set of flow conditions is expected to perform better from a nucleate boiling standpoint. It must be kept in mind that there are many factors affecting the overall heat transfer in flow boiling, and the nucleation is only one of them. The lower bounds of active cavities for various degrees of wall superheat are also shown in Fig. 7. Since the curves are quite close, and cross each other, it is difficult to predict which surface will perform better from this graph.

Finally, the performance curves of the four roughened surfaces are presented in Figs. 8 and 9. Figure 8 is a plot of heat flux vs. wall superheat for a Reynolds number of 2253 (based on the channel hydraulic diameter) and a bulk temperature of 95°C. It can be seen from the plot that the four surfaces tested in the present study performed very similar, which agrees with the findings based on Fig. 7. Under these conditions, the roughest surface D-3064 performed slightly better than the others. It is seen in the plot, however, that the surfaces did not perform in the same order of their surface roughness. Figure 9 shows a similar comparison at a Reynolds number of 7163. Under these flow conditions, the roughest surface D-3064 and the smoothest surface A-188 performed just about equal to each other, and slightly better than the other two surfaces.

In both Figs. 8 and 9, the single-phase performance at low wall superheats is seen to be about the same for each surface. Only at the higher values of wall superheat, slightly different performance is observed for the surfaces in response to the differences in their nucleation characteristics. Although the surface roughness values for these surfaces are different, the cavity size distribution plots are quite similar as seen from Fig. 7. For a given heat flux, the corresponding wall temperatures differ by 0% – 30 % from each other for the four surfaces tested. This difference is representative of the scatter observed in the flow boiling experiments of earlier researchers with water, refrigerants, and other fluids.

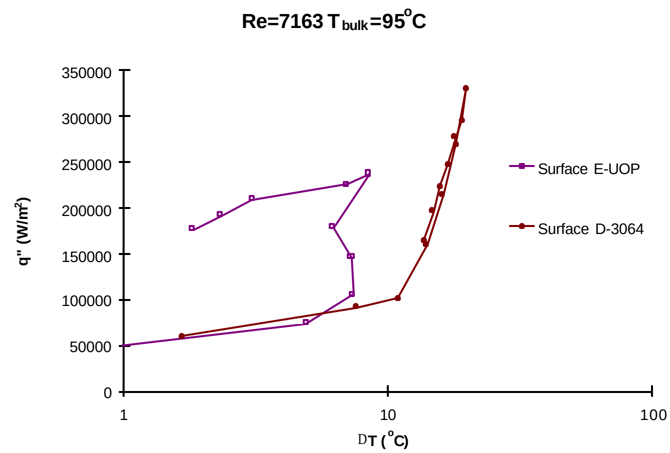

Figure 10. Hysteresis effect on a sintered surface

Figure 10 shows the performance of a sintered surface supplied by UOP in Tonawanda, NY. A 3 mm thick piece of the sintered surface was soldered on top of the heater. The data was reduced by considering the additional resistance introduced by the base and solder material. The heat flux was increased, starting from a low value, and then it was decreased. The huge amount of hysteresis is clearly seen with the sintered surface, although its performance dramatically improves during the decreasing heat flux. For comparison, the rough surface D-3064 is also shown. Very little hysteresis effect is seen for this surface.

CONCLUSIONS

The effect of surface characteristics on flow boiling heat transfer was evaluated experimentally with subcooled flow of water at atmospheric pressure. Four surfaces of different average surface roughness values were tested. The heat transfer performance was compared for these surfaces, and relationship between the performance and average roughness and cavity size distributions was investigated. Based on the present study, the following conclusions are drawn.

- The effect of roughness was very complex but the heat transfer performance seemed to depend on the number of cavities and the cavity size distributions rather than just the average roughness indicator.
- A variation in the range of 0% - 30% in the wall superheat for a given heat flux was observed for the four surfaces tested. This behavior to some extent explains the "scatter" in the flow boiling data reported by different investigators employing the same set of conditions, as the tube surface characteristics may be different.
- The active cavity range criterion was used to explain the effect of the surface roughness. The number of cavities available in the active cavity range directly relates to how many of these cavities will nucleate on a surface and thus affect the heat transfer

performance. Simple surface treatments, such as sandpaper or other polishing techniques are not adequate to provide any significant improvement in heat transfer performance.

- Sintered surfaces clearly outperform the simple roughened surfaces. However, hysteresis, higher initial cost, and other factors such as degradation over a period of time and pressure drop increase need to be considered in selecting a specific heater surface.

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