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International Journal of Heat and Mass Transfer 43 (2000) 4073–4085

International Journal of HEAT and MASS TRANSFER

www.elsevier.com/locate/ijhmt

Effect of surface roughness on pool boiling heat transfer

Myeong-Gie Kang

Department of Mechanical Engineering Education, Andong National University, 388 Songchun-dong, Andong-city, Kyungbuk 760-749, South Korea

Received 13 July 1999; received in revised form 31 January 2000

Abstract

In order to clarify the effect of surface roughness on pool boiling heat transfer, the experiments were carried out for the saturated pool boiling of water at atmospheric pressure. The experimental results show that increased surface roughness enhances heat transfer and its effect is magnified as the orientation of a tube changes from the horizontal to the vertical. In addition, it is identified that the increase in the ratio of a tube length to its diameter magnifies the effect of surface roughness on pool boiling heat transfer.

1. Introduction

The mechanisms of pool boiling heat transfer have been studied for a long time, since they are closely related with the design of the more efficient heat exchangers and heat removal systems. Recently, it has been widely investigated in nuclear power plants for their application to the design of new passive safety systems employed in Advanced Light Water Reactor (ALWR) designs [1,2]. The passive heat exchangers transfer decay heat from the Reactor Coolant System (RCS) to the water tank whenever the electric power becomes unavailable for heat removal.

To determine the required heat transfer surface area as well as to evaluate the system performance during postulated accidents, overall heat transfer coefficients applicable to the systems are needed. Since pool boiling heat transfer coefficient is usually the governing factor in determining overall heat transfer coefficients, many researchers have developed and published many correlations for pool boiling heat transfer. Recently, Chun and Kang [3] have summarized some published pool boiling correlations and proposed new empirical correlations containing tube diameter (D), tube surface roughness (ϵ), and tube wall superheat (ΔT) for horizontal and vertical tubes.

Although many researchers have investigated the effects of several design parameters on pool boiling heat transfer for the past two generations, the most widely studied issue in pool boiling must be the effect of heating surface conditions on heat transfer. It is one of the most effective factors in increasing heat transfer rate since it is directly related with active nucleation sites density on the heating surface. Changing surface finishes can shift the position of the boiling curve markedly, probably because it changes the cavity size distribution [4]. There have been many studies under several conditions of specially treated surface, coated surface, or polished surface. Various methods that have been employed to create heating surface conditions for these studies can be categorized as follows: (1) roughening surface itself, (2) coating the surface using special metal, porous material, or teflon, (3) adding specially manufactured geometry on the surface, (4)

E-mail address: mgkang@andong.ac.kr (M.-G. Kang).

0017-9310/00/$ - see front matter
PII: S0017-9310(00)00043-0

Nomenclature

A	heat transfer area
D	tube outer diameter
E	supplied voltage
H	ratio of the tube length to its diameter ($=L/D$)
h_b	boiling heat transfer coefficient
I	supplied current
L	tube length
q	net heat transfer
q''	heat flux
t	time
T_{sat}	saturated water temperature
T_w	tube surface temperature

Greek symbols

Δ	difference
ε'	average tube surface roughness in rms value
θ	tube inclination angle from the horizontal surface

winding wire or rope around test sections, and (5) manufacturing artificial cavity on the surface using drills or sharp edged punches. In addition to the above methods, study about polishing direction, greasing, and preparation techniques have been also discussed.

The initial controlled experiments on the surface finish effect were performed by Corty and Foust [4]. Averin [4] studied the nonwet surface effect, using a coated surface and concluded that much greater magnitudes of wall superheat are required to transfer a given heat flux. Berenson [4] studied the surface finish effect using copper and pentane and identified that the rougher surfaces give high heat transfer coefficient, at lower wall superheat, presumably because active cavity size are smaller on the smoother surfaces. Hsu and Schmidt [5] published results of water boiling, using horizontally installed stainless steel or aluminium disks to find the surface roughness effect. Young and Hummel [6] studied effect of the teflon coating on the horizontally installed 304 stainless steel surface. An experiment on porous material was performed by Marto and Rohsenow [7] by using horizontal stainless steel disk submerged in sodium. Vachon et al. [8] and Marto et al. [9] studied both roughened surface and prehistory of heating effects, using horizontal disks made of nickel or copper submerged in liquid nitrogen. The first introduction of the surface condition in the correlation was accomplished by Rohsenow [10]. Rohsenow used different constants to express the effects of various materials, surface conditions, and liquid types. A few years later, Mikic and Rohsenow [11] proposed a new correlation for heat transfer in nucleate boiling. The correlation incorporates effects of heat transfer surface characteristics and allows for different forms of heat flux versus wall superheat. Vachon et al. [8,12] studied effects of chemically etched and teflon coated stainless steel surfaces and compared its result with Young and Hummel's result [6].

The effect of drilled pores was studied by Nakayama et al. [13,14] using horizontal copper plate submerged in liquid nitrogen. Joudi and James [15] studied surface condition using horizontal stainless steel plate in water, R113, and methanol. They analyzed effects of surface contamination and rejuvenation on pool boiling heat transfer coefficient. Nishikawa and Fujita [16] published the relation between surface roughness and pressure, using refrigerants and horizontal copper plate. Bergles and Chyu [17] studied coated tubes submerged in R113 to investigate the effect of tube coating on the hysteresis. Nakayama et al. [18] performed experiment to identify pores effect on pool boiling heat transfer. Marto and Lepere [19] identified the effect of enhanced surface, using tubes submerged in R113 and FC72. Hahne and Muller [20] introduced effects of fin and tube number through the experiment, using copper tubes submerged in R11. A quantitative assessment about surface finish effect was first established by Cooper [21]. Cooper [21] published a pool boiling correlation having the parameter of surface roughness. Before Cooper published his results, many authors had introduced effects of surface condition qualitatively and, sometimes, treated it as different constants, which were dependent on the surface–liquid combinations, in the correlation. Chowdhury and Winterton [22] analyzed heat transfer characteristics while dropping a heated copper or aluminium rod into water or methanol to identify the effect of surface roughness and contact angle on pool boiling heat transfer. Ayub and Bergles [23] studied both hysteresis and surface effects using horizontally installed GEWA-T tube in the water and Yamaguchi and James [24] studied effects of wire mesh size and the number of wire layers on pool boiling heat transfer of the stainless flat plate submerged in water.

Hahne et al. [25] published results of R11 boiling, using horizontal copper tubes with several fins to verify effects of fin and tube number. Webb and Pais [26] published results of GEWA type geometry. Memory et al. [27] used a TURBO-B type tube bundle to identify its effect on the heat transfer coefficient. Hsieh and Hsu [28] published results of water or refrigerants boiling, using rib-roughened horizontal copper tube. They studied both hysteresis and geometry effects. Chun and Kang [3] empirically studied effects of surface roughness on pool boiling heat transfer for the horizontal and vertical tubes and also proposed two empirical correlations containing surface roughness as a parameter. Most recently, Pioro [29] experimentally evaluated the values of constants in the Rohsenow pool boiling correlation for 13 surface–fluid combinations, and also analyzed the works of other researchers to evaluate the prediction intervals for the Rohsenow pool boiling correlation for other surface–fluid combinations.

According to the references, roughening surface or adding specially manufactured cavities on the heating surface usually has the tendency of increasing pool boiling heat transfer. In other words, as the roughness increased, the temperature difference required for a given heat flux decreased. The increase in roughness also resulted in an increase in the number of bubble-forming nuclei on the surface and the nuclei resulted in the decrease of tube surface temperature. Anyhow, the final goal of the studies for surface conditions is usually to find out a way to enhance heat transfer from the given surface of the same heat transfer area and wall superheat. The efficient methods used by the researchers can be summarized as follows: (1) to decrease the temperature for onset of nucleate boiling, (2) to generate large amount of bubbles, and (3) to reduce bubble duration time on the surface. These methods can result in the increase in active bubble sites density and liquid agitation, which are two major factors to enhance heat transfer.

If a system pressure was increased, effects of surface roughness would be decreased because active nucleation sites density on the surface was decreased with increasing pressure [30]. So, if a system is in operation at high pressure, surface roughness has no significant effect on the heat transfer coefficient. According to Chowdhury and Winterton [22], surfaces with same roughness in root mean square (rms) value can result in different data, due to the preparation technique of surface roughness. Comments similar to the above can be found in the paper published by Vachon et al. [8]. The polished surface data suggest that a maximum in nucleate boiling heat transfer may be reached with a certain surface roughness for unidirectionally polished surfaces. The chemically etched surface data emphasize the inadequacy of using rms surface roughness without specifying surface preparation. In Vachon et al. paper [8], comparison of data for the two surface finishing preparation techniques with similar rms values at a given heat flux shows the differences in superheat caused by the method of preparation.

Although there have been many studies on the effect of surface roughness on pool boiling heat transfer, tubes were seldom investigated [3]. Moreover, to the author's knowledge, nobody has published a detailed analysis result for the effect of surface roughness on pool boiling heat transfer for the vertically installed tubes. Heat exchangers with vertical tubes under pool boiling condition are usually encountered in advanced nuclear power plants [2]. It is valuable to obtain the effect of surface roughness on pool boiling heat transfer because it is significant in the sizing of heat exchangers and thermal analysis of safety related systems. Therefore, in an effort to investigate the potential areas for improvement of the thermal design of the heat exchangers and to add some data to the previous works by others, an experimental parametric study of a tubular heat exchanger has been performed under pool boiling conditions. This study is focused on (1) the detailed analyses on the parameters discussed already by Chun and Kang [3] and (2) the identification of the combined parametric effects considering new parameters. That is, the present study is focused on the determination of the effect of surface roughness on pool boiling heat transfer of a tubular heat exchanger using various combinations of relevant tube parameters.

2. Experimental apparatus and procedure

A schematic view of the present experimental apparatus and the locations of thermocouples are shown in Figs. 1 and 2, respectively. The experimental apparatus consists of a water storage tank, heat exchanger tubes, water and power supply systems, and associated data acquisition system to measure the temperatures of tube surfaces and the water in the tank.

The water storage tank is made of stainless steel and has a rectangular cross-section (790×860 mm) and a height of 1000 mm. This tank has a glass view port (595×790 mm) which permits the viewing of the tubes and photographing. The heat exchanger tubes are simulated by resistance heaters made of stainless steel tubes whose heating length is 100–530 mm. To measure the surface temperatures of the heat exchanger, one of the three heat exchanger tubes was instrumented with several (2–5) K-type sheathed thermocouples outside the surface of the tube. The thermocouple tip (about 10 mm) has been bent at an angle of 90° and brazed to the tube wall. The diameter of thermocouple is 1.5 mm. The first and the fifth thermo-

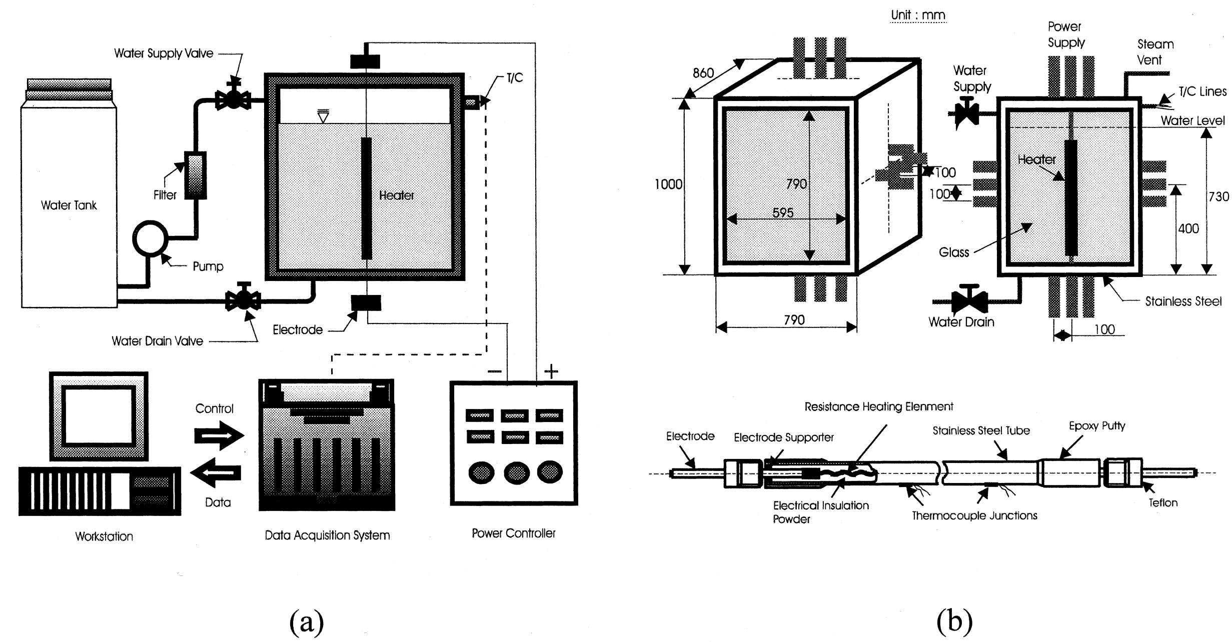


Fig. 1. Schematic diagram of the experimental apparatus: (a) overall arrangement; (b) water storage tank and heated tube.

couples are placed at 115 mm from both ends of the heating element and the space between other thermocouples is 75 mm for the 19.05 mm tube.

For vertical tube tests, the heat exchanger tubes are placed at 80 mm from the tank bottom and 290 mm from both sides. In horizontal tube tests, on the other hand, the tubes are situated at 400 mm from the tank bottom and 130 mm from both sides. The space between the heaters (i.e., pitch) is 100 mm for both cases.

To determine the combined effects of the major test parameters of the heat exchanger tube on pool boiling heat transfer in the tank, three different diameters (9.7, 19.05, and 25.4 mm), two different surface roughness (15.1 and 60.9 nm in rms measured by the phase measuring interferometer as shown in Table 1), three different orientations of heat exchanger tubes (horizontal: $\theta = 0°$, inclined: $\theta = 45°$, and vertical: $\theta = 90°$, and three different tube lengths (100, 300, and 530 mm) are used to obtain the heat flux (q'') versus wall superheat ($\Delta T = T_w - T_{sat}$) data for various combinations of test parameters. The two surface roughness values of the heating tubes were obtained by unidirectionally polishing with two sand papers of different grain sizes (No. 800 and No. 3000), and the 600-times magnified images of the polished surfaces by the Scanning Electron Microscope (SEM) are shown in Fig. 3. The uncertainty (errors from measurement, instruments, and environmental conditions) in the heat flux, temperature, and surface roughness is estimated to be $\pm 1.0\%$, $\pm 1.0\%$, and ± 5.0 nm, respectively.

Table 1
Values of tube suface roughness measured by phase measuring interferometer

Test section (number of sand paper)	Surface roughness in rms (nm)		
	Circumferential	Axial	Average
Rough surface (No. 800)	70.3	51.5	60.9
Smooth surface (No. 3000)	17.2	13.0	15.1

The water storage tank is filled with water until an initial water level of 730 mm is reached. The water is then heated using three tubes until it gets a saturation temperature. It is continued to boil for 30 min at saturation temperature to remove the air. Throughout the tests, two tubes are used only for the purpose of water heating and the third one is used for both heating and temperature measurement. The temperatures of the water and surfaces of the center tube are measured while the heater power is set at a constant value. Once the water temperature reaches a saturation value (i.e., 100°C since all the tests are run at atmospheric pressure condition), the AC power supply to the two heaters installed on the left and right (front and back) sides are turned off. The temperatures of the water and

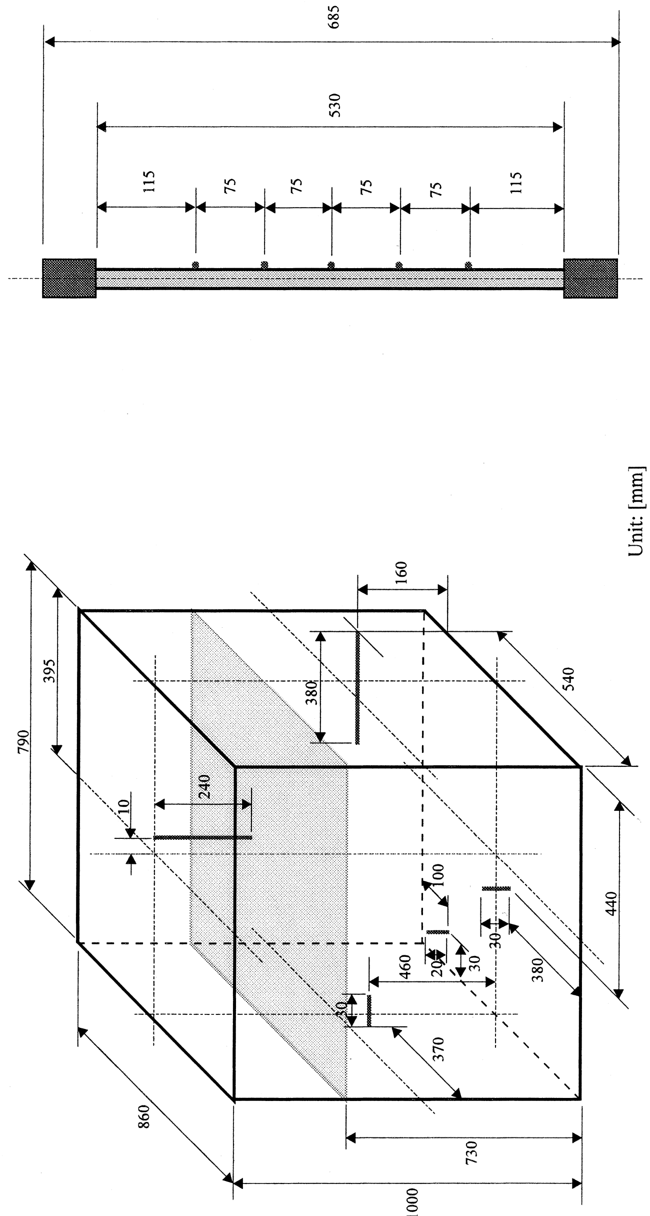


Fig. 2. Locations of thermocouples.

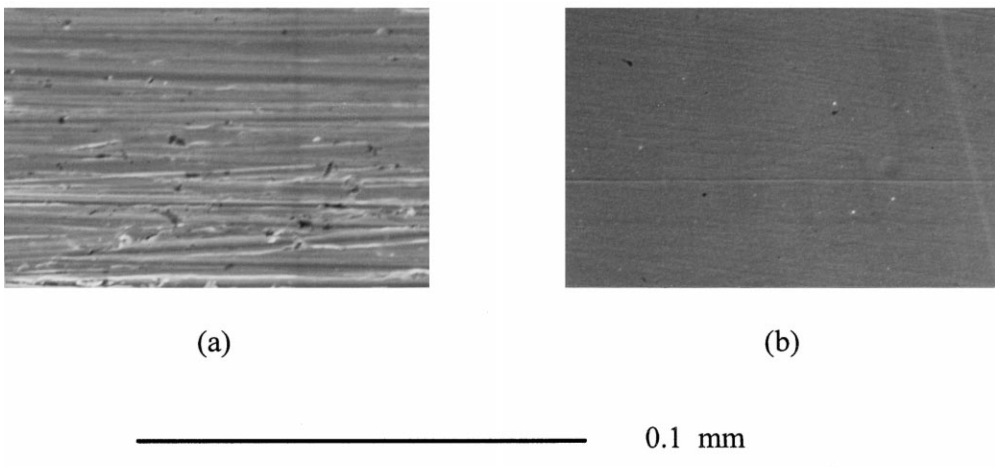


Fig. 3. SEM micrographs of specimens with different surface roughness: (a) rough surface ($\varepsilon = 60.9$ nm); (b) smooth surface ($\varepsilon = 15.1$ nm).

the heated tube surfaces are measured when they are at steady state while controlling the heat flux on the tube surface with input power. In this manner, a series of experiments have been performed for various combinations of test parameters as shown in Table 2.

The heat flux from the electrically heated tube surface is calculated from the measured values of the power input as follows:

$$q'' = \frac{q}{A} = \frac{EI}{\pi DL} = h_b(T_w - T_{sat}) = h_b \Delta T \tag{1}$$

The tube surface temperature T_w used in Eq. (1), on the other hand, is the arithmetic average value of the temperatures measured by five (for the case of D = 19.05 mm) thermocouples brazed on the tube surface as shown in Fig. 2.

Table 2
Test sections and test matrix

Orientation	Heated tube			Heat flux range (kW/m^2)	Number of thermocouples	
	D (mm)	ε (nm)	L (mm)		Water	Tube
Horizontal	9.7	60.9	300	0–80	5	3
Vertical	9.7	60.9	300	0–80	5	3
Horizontal	9.7	15.1	300	0–80	5	3
Vertical	9.7	15.1	300	0–80	5	3
Horizontal[a]	19.05	60.9	530	0–160	5	5
Horizontal[b]	19.05	60.9	530	0–160	5	5
Horizontal[c]	19.05	60.9	530	0–160	5	5
Vertical	19.05	60.9	530	0–160	5	5
Horizontal	19.05	15.1	530	0–160	5	5
Vertical	19.05	15.1	530	0–160	5	5
Inclined	19.05	15.1	530	0–130	5	5
Inclined	19.05	60.9	530	0–130	5	5
Vertical	19.05	15.1	100	0–120	5	2
Vertical	19.05	60.9	100	0–120	5	2
Horizontal	25.4	60.9	300	0–110	5	3
Vertical	25.4	60.9	300	0–110	5	3
Horizontal	25.4	15.1	300	0–110	5	3
Vertical	25.4	15.1	300	0–110	5	3

[a] Means the test to measure surface temperatures of bottom side of the horizontal tube.
[b] Means the test to measure surface temperatures of middle side of the horizontal tube.
[c] Means the test to measure surface temperatures of top side of the horizontal tube.

3. Results and discussions

3.1. Reproducibility of the experimental data

To verify the reproducibility of the experimental data, several tests for a vertical tube (D = 19.05 mm and $\varepsilon = 60.9$ nm) were conducted with changing heat flux. Three tests were done and their results are shown in Fig. 4. Fig. 4(a) shows results when the heat flux increases and Fig. 4(b) shows results when the heat flux decreases. Each test result shows a similar trend and shows no significant difference in comparison with other test results. Therefore, it can be concluded that the tests were reproducible throughout this work.

3.2. Temperature distribution on tube surface

An experimental investigation was carried out to determine the distribution of the local pool boiling heat transfer coefficients on a horizontal tube (D = 19.05 mm and $\varepsilon = 60.9$ nm) in saturated water at atmospheric pressure. The temperature distribution and the local pool boiling heat transfer coefficients on the horizontal tube surfaces are shown in Fig. 5. For the horizontal tube shown in Fig. 5(a), surface temperatures are measured at the top, middle, and bottom of the tube circumference. According to Fig. 5(a), the temperature increases as it goes from bottom to top regions for the same heat transfer. In horizontal tubes, bubbles generated at the bottom side of the tube circumference get buoyancy and slide up towards the top side of the tube circumference along the sides of the tube. These bubbles coalesce with other bubbles during movement, continue growing to large bubbles, and, finally, depart from the tube surface near the top region. Thus, a bubble-layer is formed around the tube to decrease active nucleation sites on the tube surface, and, accordingly, to reduce the amount of heat transfer to the relevant liquid. Since heat flux transferred from the surface can be assumed to be equal in all directions for the given electric power supply, the circumferential distribution of the local heat transfer coefficients of the tube revealed that maxima and minima occurred at the bottom and top, respectively, as shown in Fig. 5(b). This result is similar to that of Gupta et al. [31]. Since the measured temperature at the middle side of the horizontal tube can be considered as an average value of the temperatures

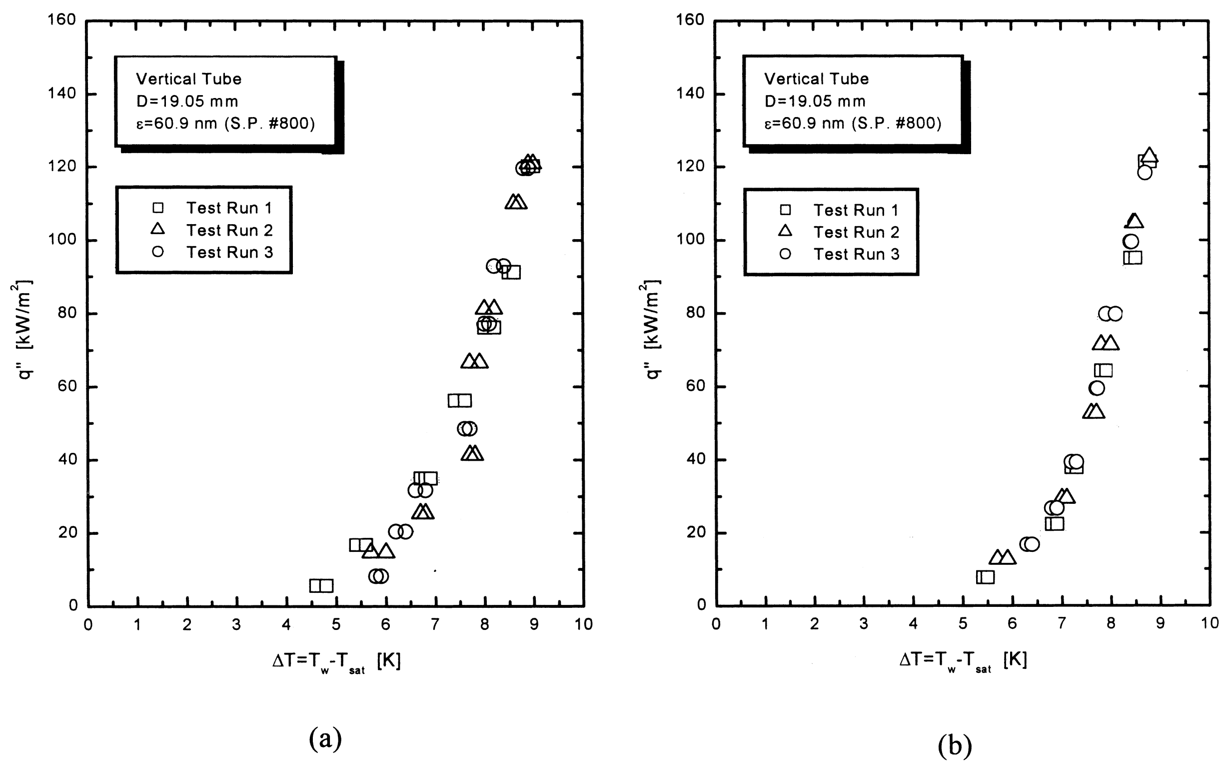


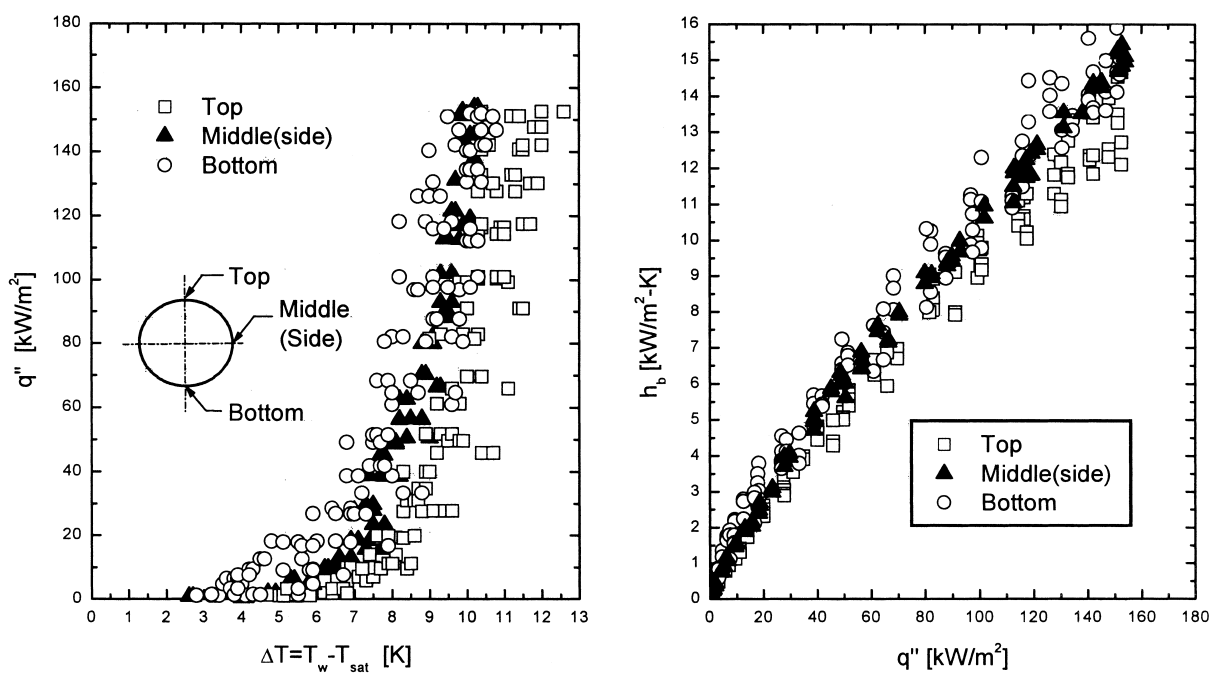
Fig. 4. Heat flux versus tube wall superheat to verify the reproducibility of the experimental data: (a) heat flux increasing direction; (b) heat flux decreasing direction.

measured at the top and bottom sides, it was selected as a standard wall temperature for the given power supply throughout this study.

Results of the temperature distribution through the height of the vertical tube (D = 19.05 mm and $\varepsilon =$ 60.9 nm) are introduced in Fig. 6. In the figure, temperature changes for two heat fluxes of 1.42 and 107.9 kW/m^2 are introduced as time elapses. As shown in the figure, a large difference between two temperatures of bottom and top regions of the vertical tube length is observed as the heat flux increases. At low heat flux (i.e., q'' = 1.42 kW/m^2) it is observed that the temperature of the thermocouple 1 installed at the upper side of the tube is lower than the temperature of the thermocouple 5 installed at the lower side of the tube. The temperature of the thermocouple 1, on the other hand, is higher than the temperature of the thermocouple 5, more than 3 K at high heat flux region (i.e., q'' = 107.9 kW/m^2). In vertical tubes, bubbles generated at the bottom can develop into large bubbles while moving along the tube length by coalescing with relevant bubbles and finally depart at the top region. At low heat flux, these bubble slugs agitate relevant liquid and increase heat transfer rate at the top region. At high heat flux, on the other hand, these bubble slugs prevent the access of relevant liquid to the heating surface and also result in rapid flow on the surface, which could act to suppress a sufficient bubble growth near the top regions. That is, as the heat flux increases, the temperature near the top region increases more than the bottom region due to decrease in active sites density at the top side. Since $h_b = q''/\Delta T$, $h_b = 9.5$ and 14.4 kW/m^2 · K can be obtained from the measured values of thermocouples 1 and 5, respectively, in case of q'' = 107.9 kW/m^2.

3.3. Surface roughness effect

Figs. 7 and 8 are q'' versus ΔT curves for the horizontal and vertical tubes (D = 9.7, 19.05, and 25.4 mm), respectively, when the tube surface roughness is used as a major test parameter. As shown in the figures, increased surface roughness gives better heat transfer at a given superheat, which is in general agreement with previous investigators. As noted by earlier researchers, the main reason for the increase in heat transfer on rough surface is because the rough surface has usually more cavities over a wide range of radii



(a) (b)

Fig. 5. Comparison of the results due to changes in thermocouple location for the horizontal tube of D = 19.05 mm and $\varepsilon = 60.9$ nm: (a) heat flux versus tube wall superheat; (b) local pool boiling heat transfer coefficient versus heat flux.

than the smooth surface, whereas pool boiling heat transfer coefficient depends on the nucleation sites density.

However, as shown in Fig. 7, it is observed that the effect of surface roughness on pool boiling heat transfer for horizontal tubes is very low. For example, q'' increases only 71.4% (from 35 to 60 kW/m^2) when ε is increased by 300% (from 15.1 to 60.9 nm) at the given wall superheat ($\Delta T = 8.5$ K) and the tube diameter (D = 19.05 mm). For vertical tubes, on the other hand, Fig. 8 shows that the effect of surface roughness on pool boiling heat transfer is significantly higher than that for horizontal tubes. For example, q'' increases more than 230% (from 30 to 100 kW/m^2) under the same conditions used for horizontal tubes.

The reason for this is presumed to be the liquid agitation due to bubbles being more pronounced in vertical tubes of rougher surface. Since the amount of heat transfer can be determined by the combined effects of the nucleation sites density and liquid agitation, the increase in the intensity of liquid agitation can increase the heat transfer rate. Once a bubble is departed from the heating surface it agitates relevant liquid and this results in rapid mixing between liquid and bubbles. Accordingly, this accelerates bubble detachment from the heating surface.

According to the published results [1–3], it is presumed that there are possibly two factors to decrease heat transfer rate from the boiling surface. One of them is bubble agglomeration on the surface and the other one is a generation of a rapid flow through the heating surface. Once bubbles are coalesced together on the heating surface, they have a tendency to remain on the surface longer than the case of a single bubble. They also make a big bubble slug on the surface to prevent the access of liquid to the surface. A rapid flow also reduces the amount of heat transfer rate since it prevents the generation of a normal size bubble.

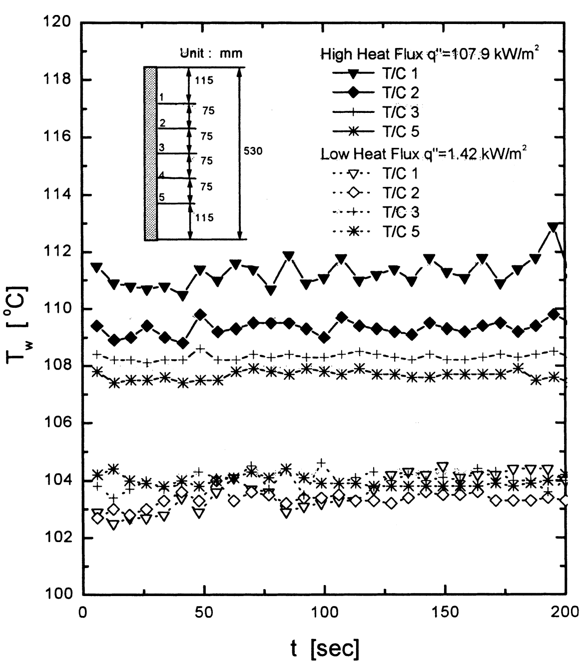


Fig. 6. Temperature distribution on the surface of the vertical tube (D = 19.05 mm and $\varepsilon = 60.9$ nm) submerged in the saturated water.

As explained above, if a heating surface gets rougher, more nucleation sites are generated and much larger sized bubbles can be produced in a short time period. As a result of this, much heat can be transferred from the heating surface as latent heat. For horizontal tubes, the generated bubbles moving along the tube circumference (shown in Fig. 5(a)) coalesce near the top regions. This prevents the access of relevant liquid. Since only a small number of bubbles are detached from the surface before the bubbles get on the top region, a small amount of liquid agitation can be expected. Hence, the increase in bubble generation due to the increase in nucleation sites density, cannot contribute much in heat transfer enhancement. The experimental results show that the effect of surface roughness is somewhat remarkable at low heat fluxes (i.e., $q'' \leq 50$ kW/m^2) since early bubble generation gives a sufficient liquid agitation to enhance heat transfer. When heat flux is low, it is observed that stirring action of single bubbles is dominant and bubble coalescence to large bunches of bubbles is somewhat unusual. As heat flux increases to more than 50 kW/m^2, the intensity of liquid agitation to the enhancement of heat transfer is decreased due to bubble coalescence on the surface and the two curves of different surface roughness become closer. If heat flux is increased, roughness effect will be decreased because of active sites density reduction followed by bubble coalescence around heating surface. This effect will be pronounced as the heating surface gets rougher.

On the other hand, the increase in heat transfer rate for the vertical tubes due to the roughened heating surface is amazing. The generated bubbles for the lower heating surface are moving along the surface and agitate relevant liquid strongly to enhance heat transfer rate. Fig. 8(a)–(c) show the experimental results for the vertical tubes of D = 9.7, 19.05, and 25.4 mm. Since tube lengths are 300, 300, and 530 mm for the diameters of 9.7, 19.05, and 25.4 mm, the values of H for the tubes are 30.9, 27.8, and 11.8, respectively. Since liquid agitation is highly effective for relatively short tubes of $H \leq 70$ [2] the slope for the q'' versus ΔT gets larger as the ratio of L/D has smaller value. The intensity of liquid agitation is increased as tube length increases and many more bubbles generate from the surface, such that, the effect of tube surface roughness is highly visible as H has a high value (i.e., 30.9 for the case) as shown in Fig. 8(a). Once, a tube gets longer and has a higher H value, the contribution of liquid agitation on heat transfer increases remarkably until H reaches a value of 70 [2]. If the value of H is greater than 70, the effect of bubble coalescence and the formation of a rapid flow along the tube surface becomes dominant. Therefore, the effect of the liquid agitation on heat transfer decreases due to the formation of bubble coalescence and rapid flow. In any case, the effect of surface roughness on pool boiling heat transfer decreases as the heat flux increases.

To identify the relation between the surface roughness and tube length, some tests have been done for a very short tube of $H = 5.3$ (i.e., $L = 100$ mm, and $D = 19.05$ mm) and their results are shown in Fig. 9. Two surface roughnesses (i.e., $\varepsilon = 15.1$ and 60.9 nm) are selected for the tests. As shown in Figs. 8(b) and 9, the effect of surface roughness for the case of smaller H is relatively small in comparison with the case of higher H, although it has the same tube diameter of 19.05 mm. For the same tube wall superheat ($\Delta T = 9$ K) the ratios between two heat fluxes for the tubes of different surface roughness (i.e., $\varepsilon = 15.1$, and 60.9 nm) are approximately 1.1 and 2.9 for the shorter and longer tubes, respectively. Therefore, it can be concluded that the effect of the surface roughness for the vertically installed tubes increases when the tube diameter decreases and tube length increases. The heat transfer coefficient versus heat flux curve is shown in Fig. 10 for various H values with different surface roughness. As shown in the figure, the increase in the ratio of a tube length to its diameter magnifies the effect of surface roughness on pool boiling heat transfer.

If a tube is properly inclined ($\theta = 45°$) as shown in Fig. 11, a somewhat larger liquid agitation can be expected in comparison with the horizontal tube as shown in Fig. 7(b). In this case, bubbles can detach from the surface before they reach the top region of the tube, such that bubble agglomeration near the top region can be prevented. Moreover, detached bubbles agitate the relevant liquid to enhance the amount of heat transfer rate from the heating surface. Whereas

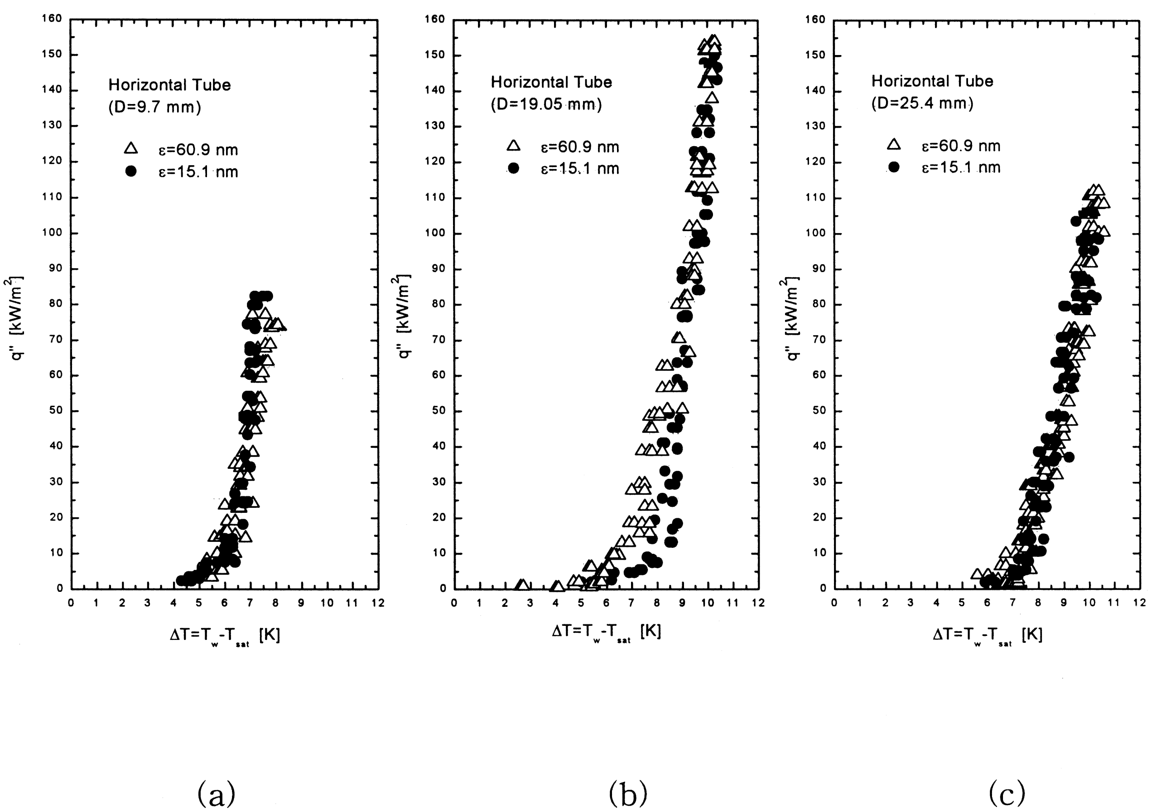


Fig. 7. Heat flux versus tube wall superheat for various surface roughness with different tube diameters (horizontal tubes): (a) $D = 9.7$ mm; (b) $D = 19.05$ mm; (c) $D = 25.4$ mm.

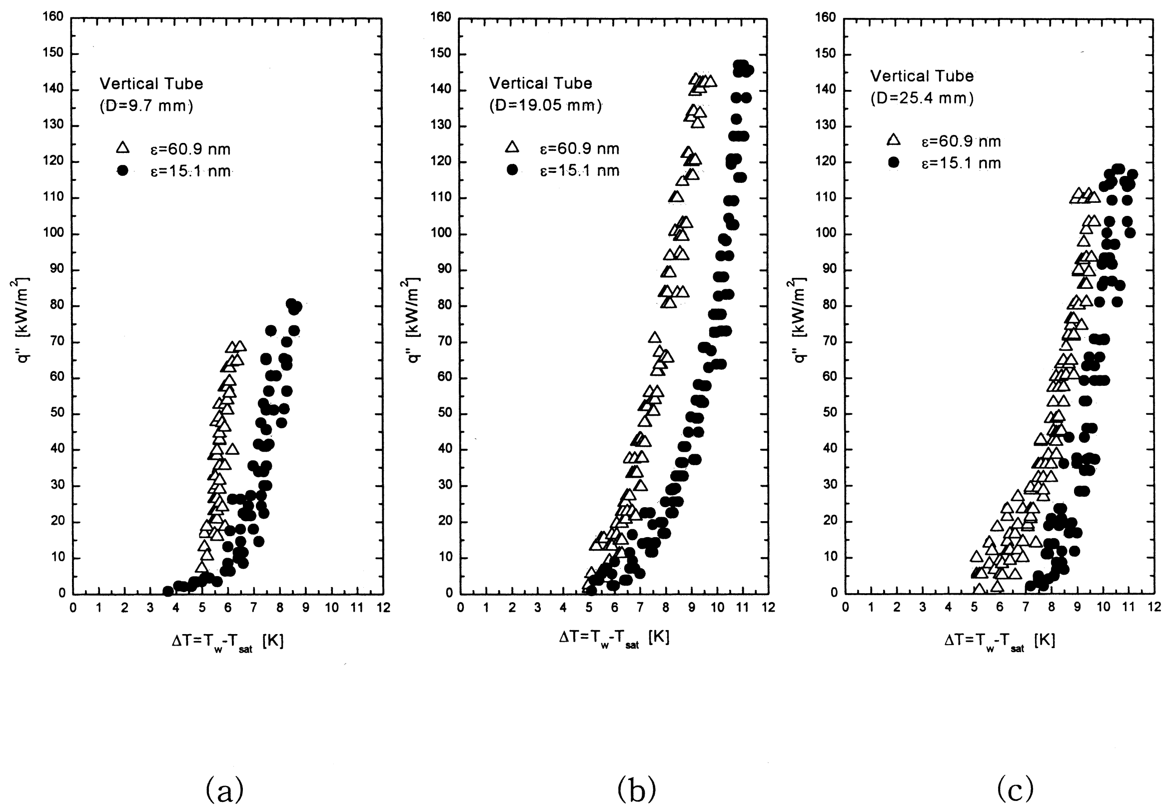


Fig. 8. Heat flux versus tube wall superheat for various surface roughness with different tube diameters (vertical tubes): (a) $D = 9.7$ mm; (b) $D = 19.05$ mm; (c) $D = 25.4$ mm.

no significant difference is observed between two curves of q'' versus ΔT at low heat flux, the difference is amplified as heat flux gets increased. At high heat flux, bubble agglomeration on the tube surface is suppressed due to a higher liquid agitation intensity for the inclined tubes. However, the effect of surface roughness for the inclined tubes is less than that for the vertical tubes as shown in Figs. 8(b) and 11. The main cause is presumed to be the decrease in the intensity of liquid agitation.

4. Conclusions

An experimental parametric study of a tubular heat exchanger has been carried out to determine effects of surface roughness on saturated pool boiling heat transfer in the water tank at atmospheric pressure. Throughout the tests, several data for q'' versus ΔT have been empirically obtained using various combinations of tube diameters ($D = 9.7$, 19.05, and 25.4 mm), heating surface roughness ($\varepsilon = 15.1$ and 60.9 nm), tube orientations ($\theta = 0°$, 45°, and 90°), and tube lengths ($L = 100$, 300, and 530 mm). The main conclusions for the present study are as follows:

1. Increased surface roughness gives better heat transfer at a given superheat because the rough surface has usually more cavities than the smooth surface, while pool boiling heat transfer coefficient depends on the nucleation sites density.
2. To obtain the net effect of surface roughness on pool boiling for the tubes, active nucleation sites density, the intensity of liquid agitation, bubble agglomeration on the surface, and the formation of a rapid flow around the tube surface must be considered. The increase in active sites density and the intensity of liquid agitation increases the heat transfer, while increase in bubble agglomeration on the surface and the formation of a rapid flow results in decrease in heat transfer.
3. The increase in surface roughness gives no plausible change on pool boiling heat transfer, especially in high heat flux region, for the horizontal tubes because liquid agitation is not so strong and large bubble slugs are formed on the tube surface as heat flux gets increased. However, its effect is magnified

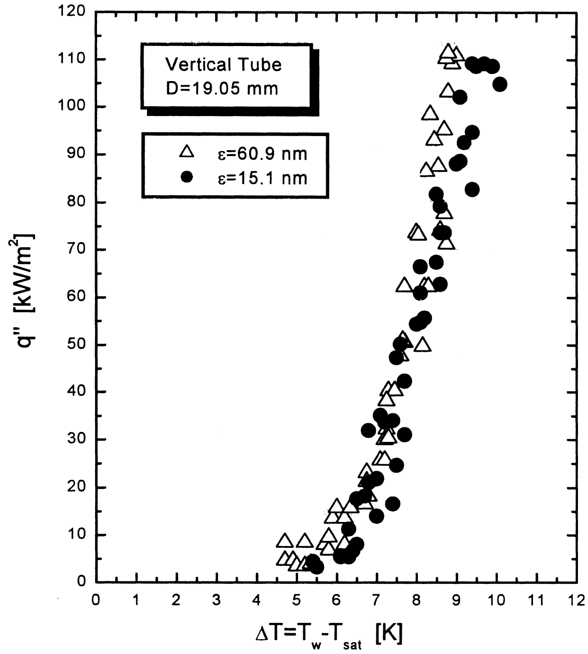


Fig. 9. Heat flux versus tube wall superheat for the vertical tube of shorter length (D = 19.05 mm and L = 100 mm).

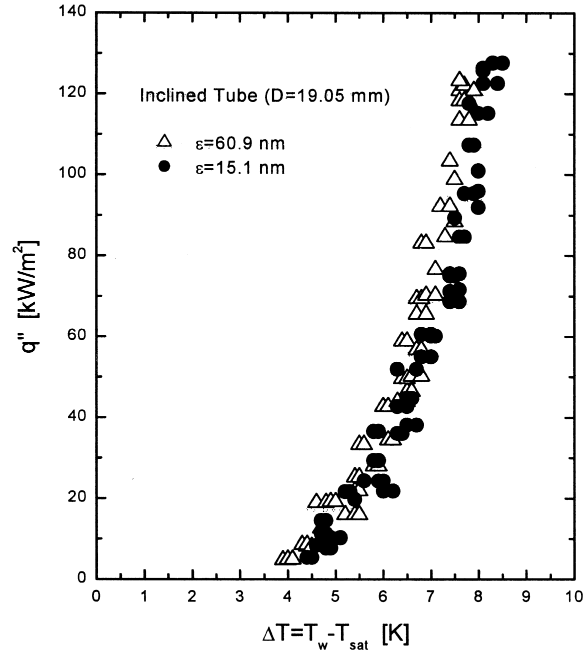


Fig. 11. Heat flux versus tube wall superheat for the inclined tubes of different surface roughness (D = 19.05 mm).

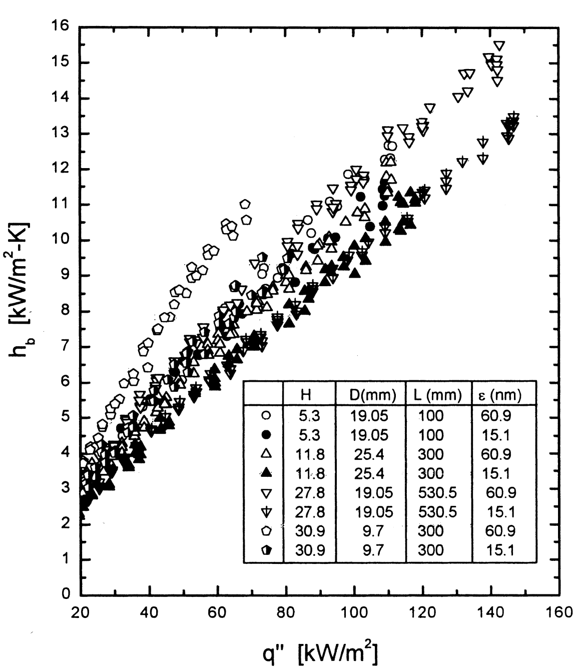


Fig. 10. Heat transfer coefficient versus heat flux for various H $(=L/D)$ values with different surface roughness.

as the orientation of a tube changes from the horizontal to the vertical because the change in tube orientation gives much stronger liquid agitation and smaller bubble coalescence.

4. It is identified that the increase in the ratio of a tube length to its diameter (i.e., $H = L/D$) magnifies the effect of surface roughness on pool boiling heat transfer for the vertically installed tubes because the intensity of liquid agitation is presumed to be increased in accordance with the increase in tube length. Once the tube gets longer the formation of a rapid flow and bubble slugs increases and the contribution of liquid agitation on the heat transfer decreases.

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