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Experimental Investigation of the Combined Effects of Heat Exchanger Geometries on Nucleate Pool Boiling Heat Transfer in a Scaled IRWST

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열교환기 형상이 축소한 IRWST 내부의 풀핵비등에 미치는 복합적인 영향에 대한 실험적 연구

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Abstract

In an effort to determine the combined effects of major parameters of heat exchanger tubes on the nucleate pool boiling heat transfer in the scaled in-containment refueling water storage tank (IRWST), a total of 1,966 data for q''versus ΔT has been obtained using various combinations of tube diameters, surface roughness, and tube orientations. The experimental results show that (1) increased surface roughness enhances heat transfer for both horizontal and vertical tubes, (2) the two heat transfer mechanisms, i.e., enhanced heat transfer due to liquid agitation by bubbles generated and reduced heat transfer by the formation of large vapor slugs and bubble coalescence are different in two regions of low heat fluxes ($q'' \leq 50 kW/m^2$) and high heat fluxes ($q'' > 50 kW/m^2$) depending on the orientation of tubes and the degree of surface roughness, and (3) the heat transfer rate decreases as the tube diameter is increased for both horizontal and vertical tubes, but the effect of tube diameter on the nucleate pool boiling heat transfer for vertical tubes is greater than that for horizontal tubes. Two empirical heat transfer correlations for q'', one for horizontal tubes and the other for vertical tubes, are obtained in terms of surface roughness (ε) and tube diameter (D). In addition, a simple empirical correlation for nucleate pool boiling heat transfer coefficient (h_b) is obtained as a function of heat flux (q'') only.

요 약

축소한 격납용기 내부 핵연료재장전수저장탱크의 안쪽에 설치한 열교환기 튜브의 주요 매개변수들이 풀핵비등 열전달에 미치는 복합적인 영향을 규명하기 위해 튜브 외경, 표면 거칠기, 그리고 튜브 설치 방향에 대한 다양한 조합들을 활용하여 열유속 q''와 과열 온도 차이 ΔT 간의 관계에 대한 총 1,966 개의

실험값을 취득하였다. 이 실험 결과들에 의하면, (1) 표면 거칠기 증가는 수평 및 수직 튜브 모두에 대해 열전달을 향상시키고, (2) 기포 생성에 따른 두가지 열전달 기구인 주변 액체 운동 증가에 의한 열전달 향상과 기포층 및 기포 군집 형성에 의한 열전달 감소는 $50kW/m^2$의 열유속을 경계로 낮은 열유속과 높은 열유속 영역에서 서로 다르게 관찰되는데, 이것은 튜브 설치 방향과 표면 거칠기의 크기와 관련이 있으며, (3) 튜브 외경 증가는 수평 및 수직 튜브 모두에 대해 열전달을 감소시키는데, 그 영향 정도는 수평보다 수직구조에서 더 크다. 수평 및 수직 튜브들에 대해 열유속 q''와 표면 거칠기 (ε) 및 튜브 외경 (D) 사이의 관계를 결정하는 두가지 실험식을 개발하였다. 그리고, q''만의 함수로된 풀핵비등 열전달계수 (h_b) 에 대한 간단한 실험식도 부가적으로 개발하였다.

1. Introduction

One of the key features of the passive safety systems employed in the advanced light water reactor (ALWR) designs such as the Westinghouse AP600 plant is the passive residual heat removal (PRHR) heat exchanger shown schematically in Fig. 1 [1]. The PRHR system transfers decay heat from the reactor coolant system (RCS) to the containment by heating and boiling the water in the in-containment refueling water storage tank (IRWST) whenever the steam generators become unavailable for heat removal during normal operation, hot standby, heatup, or cooldown. The steam generated in the IRWST is condensed at the inner wall of the containment vessel and returned by gravity via the IRWST condensate return gutter [2]. The current version of ALWR requirements document requires the RCS temperature of 204.4°C (400 °F) to be achieved in 72 hours, with or without reactor coolant pumps operating [3].

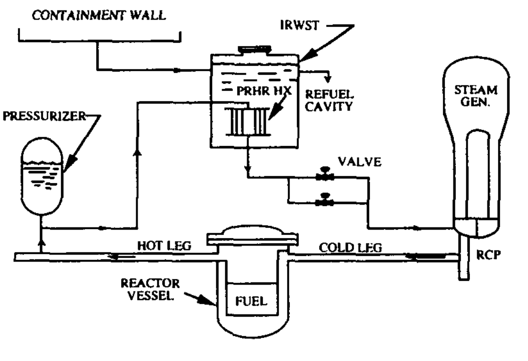


Fig. 1. Passive Residual Heat Removal(PRHR) Heat Exchanger(HX) for Advanced Light Water Reactor

The PRHR heat exchanger shown in Fig. 1 receives hot reactor coolant from the RCS pressurizer surge line and discharges to the steam generator channel head. When the PRHR flow is driven by natural circulation only, its decay heat removal capability is determined mainly by the natural circulation flow rate of the PRHR system. The natural circulation flow rate in the PRHR system, on the other hand, depends primarily on the buoyancy effects induced by the temperature difference between the cold and the hot legs and the vertical length of the PRHR system. However, the vertical length of the PRHR system is more or less limited to a narrow range and cannot easily be increased without incurring significant extra expenses. The major remaining option to improve the heat transfer capability of the PRHR system seems to be the thermal design aspect of the heat exchanger.

To determine the required heat transfer surface area as well as to evaluate the PRHR performance during postulated accidents, an overall heat transfer coefficient applicable for the PRHR system is needed. Currently, the generalized PRHR boiling correlation developed by Corletti et al. [4] is the only available correlation. However, the measured heat flux data obtained from the PRHR test facility, where the PRHR heat exchanger tubes are mounted vertically in a cylindrical tank which has the prototypic height to preserve the buoyancy effects of the IRWST, is one order of magnitude lower than conventional nu-

Table 1. Conventional Nucleate Pool Boiling Heat Transfer Correlations and Major Parameters Included

Author	Correlations	Key Parameters	Remarks
Rohsenow [5]	$q'' = \mu_l h_{fg}\left[\frac{g(\rho_l-\rho_g)}{\sigma}\right]^{\frac{1}{2}}\left(\frac{c_{pl}\Delta T}{h_{fg}Pr_l^{s}C_{sf}}\right)^3$ (a) C_{sf} = coeff. of Eq.(a)	o Pressure o Fluid-heating-surface combination	o C_{sf} = surface-fluid constant - It depends on both the surface and the fluid - Typical values ; 0.0025~0.015
Jakob and Hawkins [9]	Horizontal : $h_b = 1042\Delta T^{1/3}$ (for $q'' < 16$ kw/m^2) (b) $h_b = 5.56\Delta T^3$ (for $16 < q'' < 240$ kw/m^2) (c) Vertical : $h_b = 537\Delta T^{1/7}$ (for $q'' < 3$ kw/m^2) (d) $h_b = 7.96\Delta T^3$ (for $3 < q'' < 63$ kw/m^2) (e)	o Surface orientation	o ΔT = (Heating Surface Temp.-Saturation Temp.) o h_b = in W/m^2-K o Fluid ; water o Pressure ; 0.1 MPa
Cornwell et al. [7]	$Nu_b = C_{tb}Re_b^{2/3}$ (f) where, $Nu_b = \frac{h_b D}{k_l}$ and $Re_b = \frac{q''}{h_{fg}}\frac{D}{\mu_l}$ (tube boiling Reynolds (f) number) (C_{tb} = a constant dependent on the surface, fluid and pressure. 100 for water)	o Diameter	o Fluid ; water, refrigerants, and organics o Pressure ; $p < 0.1\ p_c$ o Tube diameter ; 6-32 mm o Surface ; commercial finish o Accuracy ; ±33 %
Cooper [13]	Copper plate or Stainless Steel cylinders ; $h_b = 55p_R^{(0.12-0.2\log_{10}\varepsilon)}(-\log_{10}p_R)^{-0.55}M^{-0.5}q''^{0.67}$ (g) Copper cylinders ; $h_b = 93.5p_R^{(0.12-0.2\log_{10}\varepsilon)}(-\log_{10}p_R)^{-0.55}M^{-0.5}q''^{0.67}$ (h) p_R=reduced pressure, M=molecular weight: ε =surface roughness in microns	o Surface roughness o Liquid property	o Orientation ; horizontal
Cornwell and Houston [8]	$Nu_b = CF(p)Re_b^{0.67}Pr^{0.4}$; $C = 9.7p_c$ (i) $F(p) = 1.8p_R^{0.17} + 4p_R^{1.2} + 10p_R^{10}$, $p_R = p/p_c$ (j)	o Diameter o General correlation	o Orientation ; horizontal o Fluid ; water, refrigerants, and organics o Pressure ; 0.001-0.8 p_c o Heat Flux ; 0.1-0.8 q_c'' o Diameter ; 8-50 mm o Surface ; machined or drawn
Present Correlation	Horizontal : $q_H'' = 0.015\varepsilon^{0.084}\Delta T^{5.508}/D^{1.318}$ (5a) Vertical : $q_V'' = 0.024\varepsilon^{0.672}\Delta T^{4.862}/D^{1.656}$ (6a) Simple form : $h_b = 0.266\ (q'')^{0.812}$ (9)	o Diameter o Surface roughness o Surface orientation	o Fluid ; water o Pressure ; 0.1 MPa o Heat flux ; $0 < q'' < 160$ kW/m^2 o Tube diameter ; 9.7-25.4 mm o Surface roughness ; 15.1-60.9 nm o Accuracy ; +35 %, -20 %

cleate pool boiling heat transfer correlations summarized in Table 1. For example, the heat flux values calculated by the generalized PRHR boiling correlation of Corletti et al. [4] are less than one-tenth of the value predicted by Rohsenow correlation [5]. That is, the coefficient (C_{sf}) value which appears in the denominator in both expressions for the heat flux reported by Corletti et al. [4] and Rohsenow [5], C_{sf} = 0.034 for the PRHR data [4] whereas C_{sf} = 0.013 for Rohsenow's value of water.

Many workers have in the past two generations investigated the effects of the diameter [6-8], the orientation of heated surface [6, 9], and the surface roughness [10-13] on nucleate pool boiling heat transfer along with the effects of pressure and fluid properties. Earlier workers have generally found that a decrease in tube diameter and an increase in surface roughness give better heat transfer at a given superheat. Also, van Stralen and Sluyter [6] have found that, in general, the heat flux on horizontal wires in pure liquids exceed the value on vertical wires. However, there seems to be some inconsistencies in their results. For example, Vachon's polished surface data [10] suggest that a maximum in nucleate boiling heat transfer may be reached with a certain surface roughness for unidirectionally polished surface. In addition, according to the Jakob's nucleate boiling correlation [9] the heat transfer coefficient of vertical heating surface is larger than that of horizontal heating surfaces as opposed to the results of van Stralen and Sluyter [6].

In an effort to investigate the potential areas for improvement of the thermal design of the PRHR system of ALWRs and to resolve the inconsistencies of the previous work by others, an experimental parametric study of a tubular heat exchanger has been performed under nucleate pool boiling conditions. That is, the present study is aimed at the determination of the combined effects of tube diameter, surface roughness, and tube orientation (horizontal and vertical) on the nucleate pool boiling heat transfer. As a result, new empirical heat transfer correlations, one for horizontal heat exchanger tubes and the other for vertical heat exchanger tubes, in terms of major heat exchanger tube parameters and the tube wall superheat have been obtained. Because of the limited practical applicability of these correlations, however, a simple empirical correlation for nucleate pool boiling heat transfer coefficient h_b in terms of only q″ has also been presented here.

2. Experimental Apparatus and Procedure

2.1. Experimental Apparatus

A schematic view of the present experimental apparatus is shown in Fig. 2. The experimental apparatus essentially consists of a scaled (1/10th in length) IRWST (i.e., a water tank), three heat exchanger tubes shown in Fig. 2(b), water and power supply systems, and associated data acquisition system to measure the temperatures of tube surfaces and the water in the IRWST.

Two major scaling parameters for AP600 PRHR system are (1) the heat flux on the tube surface and (2) the water volume of the tank (i.e., IRWST) per unit core decay power. The heat flux that corresponds to 2% of the core decay power is 101.2kW/m^2, and the water volume per unit decay power is 0.05 m^3/kW. A comparison between the prototype [4, 14] of the IRWST and the scaled model is given in Table

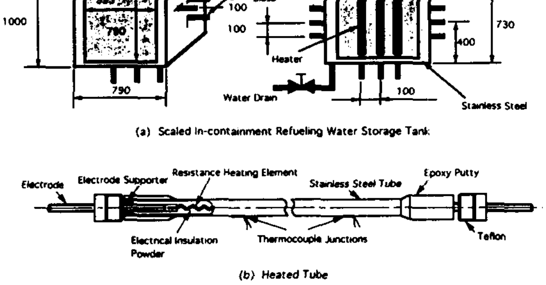


Fig. 2. Schematic Diagram of Experimental Apparatus

Table 2. Comparison Between Prototype(AP600 PRHRS) and Experimental Apparatus

Item	Prototype (P)	Scaled Model (M)	Ratio (M/P)
HX Tube Length, m	5.305	0.5305	1/10
Heat Transfer Area, m^2	382	0.0955	1/4000
Water Volume, m^3	1935	0.48375	1/4000
Power (2%), kWt	38660	9.665	1/4000
Heat Flux, kW/m^2	101.2	101.2	1/1
Water Volume per Unit Power, m^3/kW	0.05	0.05	1/1

2. The IRWST model is made of stainless steel and has a rectangular cross section (790×860mm) and a height of 1,000mm. This tank has a glass view port (595×790mm) which permits viewing of the tubes and photographing. The heat exchanger tubes are simulated by resistance heaters made of stainless steel tubes whose heating length is 530.5mm. To measure the surface temperatures of the heat exchanger one of the three heat exchanger tubes were instrumented with five thermocouples outside the surface of the tube. The thermocouple tip (about 10mm) has been bent at a 90 degree angle and brazed the bent tip on the tube wall. The thermocouple diameter is 1.5mm. The first and the fifth thermocouples are placed at 115.25mm from both ends of the heating element and the space between other thermocouples are 75mm.

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

2.2. Major Test Parameters and Procedures

To determine the combined effects of the major test parameters of the heat exchanger tube on the nucleate pool boiling heat transfer in the scaled IRWST, four different diameters (ranging from 9.7mm to 25.4mm), three different surface roughness (15.1, 26.2, and 60.9 nm in RMS measured by the phase measuring interferometer as given in Table 3) and two different orientations of heat exchanger tubes (horizontal and vertical) are used to obtain the heat flux (q″) versus wall superheat ($\Delta T = T_w - T_{sat}$) data for various combinations of test parameters as summarized in Table 4. The three surface roughness of the heating tubes were obtained by polishing unidirectionally with three different grain size sand papers (#800, #2000, and #3000).

Table 3. Values of Tube Surface Roughness Measured by Phase Measuring Interferometer

Test Section (Sand Paper)	Grain Size, μm	rms Surface Roughness, nm		
		Circumferential	Axial	Average
#800	31.75	70.3	51.5	60.9
#2000	12.70	30.6	21.8	26.2
#3000	8.47	17.2	13.0	15.1

The scaled IRWST is filled with water until the initial water level is reached at 730mm and the water is then boiled for 30 minutes at saturation temperature to remove the air. The temperatures of the water and heater surfaces are measured while the heater power is set at constant value. However, once the water temperature is reached at saturation value (i.e., 100°C since all the tests are run at atmospheric pressure condition), the temperatures of the water and 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 has been performed for various combinations of test parameters.

2.3. Measurements of Heat Flux and Tube Surface Temperature

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

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

Table 4. Test Matrix and Experimental q″ versus ΔT Data for Correlation Development

Test	Heated Tube Number of Tubes	Tube Orientation	D (mm)	ε (nm)	L (mm)	Water Level (mm)	Heat Flux Range (kW/m^2)	Number of Thermocouples Water	Tube	Number of Data
	1	H	19.05	60.9	530.5	730	0-160	5	5	
	2	H	19.05	60.9	530.5	730	0-160	5	5	373
Nucleate	3	H	19.05	60.9	530.5	730	0-160	5	5	
Pool	1	V	19.05	60.9	530.5	730	0-160	5	5	131
Boiling	1	H	19.05	26.2	530.5	730	0-160	5	5	121
Heat	1	V	19.05	26.2	530.5	730	0-160	5	5	147
Transfer	1	H	19.05	15.1	530.5	730	0-160	5	5	85
Tests	1	V	19.05	15.1	530.5	730	0-160	5	5	121
	1	H	9.7	60.9	300.0	730	0-80	5	3	90
	1	V	9.7	60.9	300.0	730	0-80	5	3	76
	1	H	9.7	15.1	300.0	730	0-80	5	3	63
	1	V	9.7	15.1	300.0	730	0-80	5	3	87
	1	H	14.0	60.9	300.0	730	0-90	5	3	116
	1	V	14.0	60.9	300.0	730	0-90	5	3	121
	1	H	25.4	60.9	300.0	730	0-110	5	3	148
	1	V	25.4	60.9	300.0	730	0-110	5	3	123
	1	H	25.4	15.1	300.0	730	0-110	5	3	80
	1	V	25.4	15.1	300.0	730	0-110	5	3	84

- q″ versus ΔT test data with Horizontal Tube (H) = 1076 points
- q″ versus ΔT test data with Vertical Tube (V) = 890 points
- Total Number of q″ versus ΔT test data = 1966 points

where E and I are the supplied voltage (in volt) and current (in ampere), and D and L are the outside diameter and the length of the heated tube, respectively. 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 thermocouples brazed on the tube surface.

3. Correlations of Experimental Data

As summarized in Table 4, a total of 1,966 data (1,076 with horizontal tubes and 890 with vertical tubes) has been obtained for heat flux (q″) versus wall superheat(ΔT) for various combinations of the diameter, surface roughness, and tube orientation. In Figs. 3~7 the experimental results for nucleate pool boiling heat transfer are plotted as the heat flux (q″) versus wall superheat (ΔT) using the diameter, the surface roughness, and the orientation of heated tubes as major test parameters.

A close review of literatures on the nucleate pool boiling heat transfer has revealed the following facts :

(1) It is not realistic to obtain any general theoretical correlation for heat transfer coefficients in nucleate boiling.

This is because the boiling occurs at nucleation sites, and the number of sites is very dependent upon (a) the physical condition and preparation of the surface ; and (b) how well the liquid wets the surface and how efficiently the liquid displaces air from the cavities [15].

(2) For a given fluid on a surface of given roughness, at constant pressure an empirical correlation may be developed in the following form [13] :

$$h_b \propto (q/A)^n \quad (2)$$

(3) Two practical approaches to develop a nucleate boiling heat transfer correlation are possible: The first approach is to take account of both effects of the surface roughness and the diameter of the heat exchanger tube. The second approach is to ignore both effects and develop a correlation that gives a typical boiling heat transfer coefficient at the particular heat flux for the fluid since the surface effects are often very difficult to define quantitatively.

Using the first approach, simple correlations are sought in the following form:

$$q'' = C_1 \varepsilon^{C_2} \Delta T^{C_3} D^{C_4} \quad (3)$$

$$h_b = C_5 (q'')^{C_6} \quad (4)$$

As a result, two empirical correlations, one for vertical tubes and the other for horizontal tubes, have been obtained using present experimental data and the statistical analysis system (SAS) computer program (which uses the least square methods as a regression technique) as follows:

$$q''_H = 0.015\varepsilon^{0.084} \Delta T^{6.508}/D^{1.318} \text{ (for horizontal tubes)} \quad (5a)$$

$$h_{bH} = 0.015\varepsilon^{0.084} \Delta T^{4.508}/D^{1.318} \text{ (for horizontal tubes)} \quad (5b)$$

$$q''_V = 0.024\varepsilon^{0.672} \Delta T^{4.862}/D^{1.656} \text{ (for vertical tubes)} \quad (6a)$$

$$h_{bV} = 0.024\varepsilon^{0.672} \Delta T^{3.862}/D^{1.656} \text{ (for vertical tubes)} \quad (6b)$$

From visual observations of boiling behavior and also from the analysis of present experimental data it has been confirmed that the effectiveness of two competing heat transfer mechanisms (i.e., enhanced heat transfer due to liquid agitation by bubbles generated and reduced heat transfer by the formation of large vapor slugs and bubble coalescence) is different in two regions of low heat fluxes and high heat fluxes depending on the degree of surface roughness and the tube orientation. Therefore, two empirical heat transfer correlations are derived for two different regions of heat fluxes to fit present experimental data more closely and for convenience in discussion as follows:

For low heat flux region ($q'' \leq 50 kW/m^2$):

$$q''_H = 0.006\varepsilon^{0.490} \Delta T^{6.116}/D^{1.318} \text{ (for horizontal tube)} \quad (7a)$$

$$q''_V = 0.002\varepsilon^{0.910} \Delta T^{6.600}/D^{1.656} \text{ (for vertical tube)} \quad (7b)$$

For high heat flux region ($q'' > 50 kW/m^2$):

$$q''_H = 0.054\varepsilon^{0.086} \Delta T^{4.962}/D^{1.318} \text{ (for horizontal tube)} \quad (8a)$$

$$q''_V = 0.031\varepsilon^{0.640} \Delta T^{4.780}/D^{1.656} \text{ (for vertical tube)} \quad (8b)$$

The above correlations alone, however, may be of no use for general application unless the surface roughness ε(in nm) of the tubes in question is either given or a priori known. Therefore, using the second approach, another simple correlation has been obtained as follows:

$$h_b = 0.266 (q'')^{0.812} \quad (9)$$

Predictions made by the above correlations are shown by solid lines and compared with experimental data from Fig. 3 to Fig. 7.

4. Summary of Parametric Effects and Discussion

4.1. Effects of Surface Roughness on Nucleate Boiling Heat Transfer

Figures 3(a) and 3(b) are q'' versus ΔT curves for a given tube diameter when the tube surface roughness is used as a major test parameter. In these figures the following observations can be made:

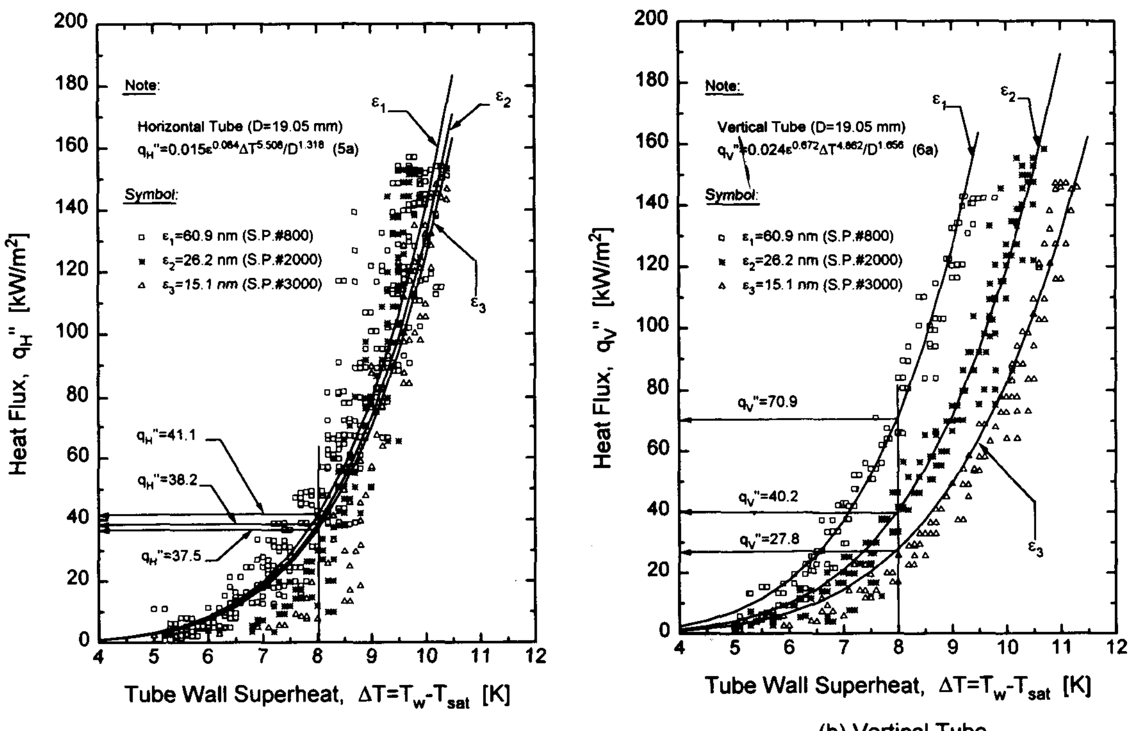
(1) Increased surface roughness gives better heat transfer at a given superheat for both horizontal and vertical tubes which is in general agreement with previous investigators [10-13]. As noted by earlier workers, the main reason for the increase of heat transfer in rough surface is because the rough surface has usually more cavities over a wide range of radius than the smooth surface has, while the nucleate boiling heat transfer coefficient depends on the nu-

cleation sites density.

(2) Figure 3(a) shows that the effect of surface roughness on the nucleate boiling heat transfer for horizontal tubes is very small. For example, q''_H increases only 9.6% (from 37.5 to 41.1 kW/m^2) when ε is increased by 300% (from 15.1 to 60.9nm) according to Eq.(5a) at the given wall superheat ($\Delta T = 8$K) and the tube diameter (D = 19.05mm). For vertical tubes, on the other hand, Fig.3(b) shows that the effect of surface roughness on the nucleate boiling heat transfer is significantly larger than that for horizontal tubes. According to Eq.(6a), q''_V increases more than 150% (from 27.8 to 70.9kW/m^2) under the same conditions used for horizontal tubes. The reason for this is partly because the liquid agitation effect of the bubbles generated is more pronounced in vertical tubes with rough surface.

4.2. Combined Effects of Tube Orientation and Surface Roughness On Nucleate Boiling Heat Transfer

To examine the combined effects of tube orientation and surface roughness on nucleate pool boiling, q'' versus ΔT data obtained from the tests with horizontal and vertical tubes (for a given tube diameter D = 19.05mm) are plotted in Fig. 4 for three different surface roughnesses. It is particularly interesting to note that the slope of q'' versus ΔT curve of the vertical tube becomes smaller than that of the horizontal tube as the surface roughness decreases from $\varepsilon = 60.9$ to $\varepsilon = 15.1$nm. That is, when the surface roughness is $\varepsilon = 26.2$nm, the slopes of two q'' versus ΔT curves shown in Fig. 4(b) shift and cross each other at about $\Delta T = 8.4$K and $q'' = 50.0$kW/m^2.



(a) Horizontal Tube (b) Vertical Tube

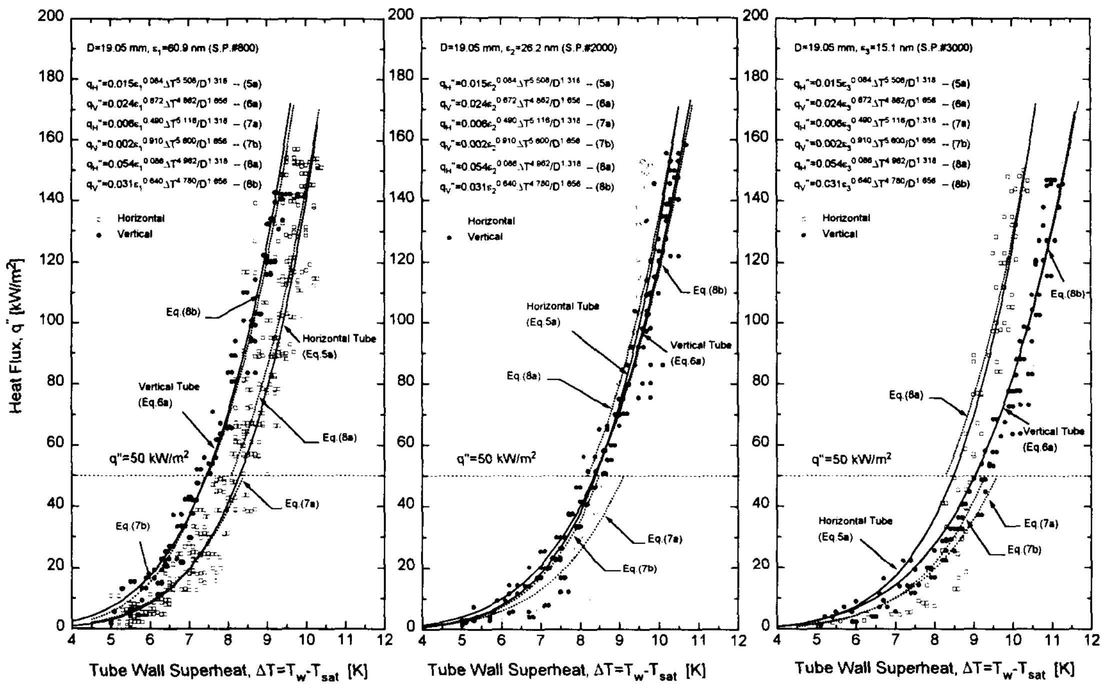
Fig. 3. Heat Flux versus Wall Superheat for Various Surface Roughness with Different Tube Orientation

The reason for this may be attributable to the following phenomena :

(1) From visual observations (shown in Photos 1 and 2) the relationship between the combined effects of the orientation and the surface roughness of the tube and the heat transfer mechanisms in nucleate boiling region is first inferred as follows : There seems to be two competing effects on the nucleate boiling heat transfer. One is the effect of liquid agitation by bubbles generated on the surface which enhances heat transfer and the other is the effect of bubble coalescence and the formation of large vapor slugs in the high heat flux region (e.g., $q''>50kW/m^2$) in particular which reduces heat transfer from the tube surface.

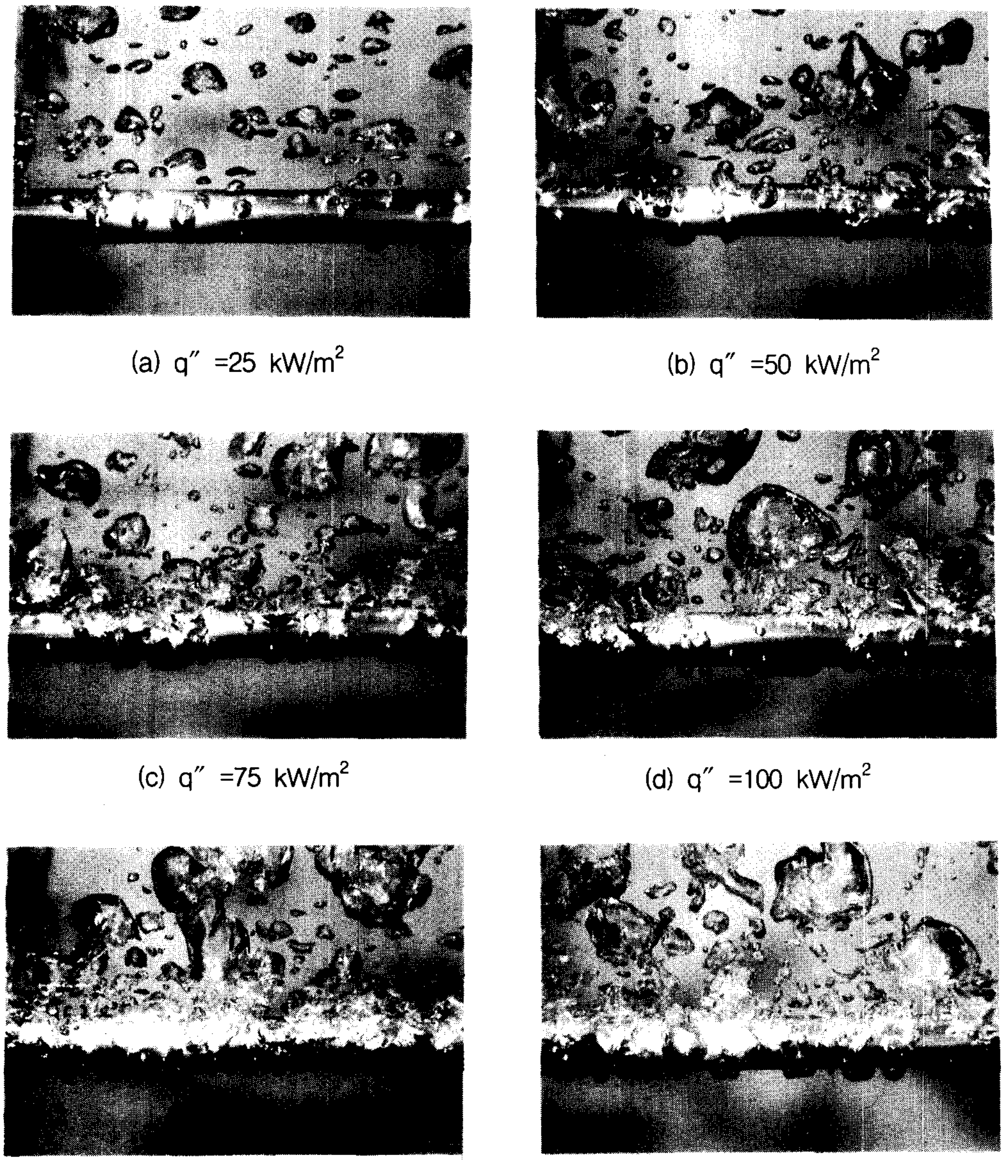
(2) For vertical tubes, particularly in the high heat flux region ($q''>50kW/m^2$), the coalescing bubbles originate from nuclei which are distributed over the entire heating surface as noted by earlier workers [6]. For horizontal tubes, on the other hand, bubble coalescence is generally limited to only part of the upper surface while the lower half is almost entirely free from large bubbles or bubble coalescence. Therefore, the horizontal tube enhances nucleate boiling heat transfer much more than the vertical tube particularly in the high heat flux region and when the surface is more smooth (e.g., $\varepsilon<26.2$nm). For vertical tubes with smooth surface ($\varepsilon<26.2$nm), the effect of reductions in the effective heat transfer area and in the number of active nucleation sites due to the bubble coalescence and formation of large bubble slugs becomes more predominant than the liquid agitation effect and the net effect is the decrease in nucleate boiling heat transfer in comparison with horizontal tubes.

In essence, the effectiveness of the two heat trans-



(a) ε_1=60.9 nm (b) ε_2=26.2 nm (c) ε_3=15.1 nm

Fig. 4. Heat Flux versus Wall Superheat for Three Different Roughness with Different Tube Orientation



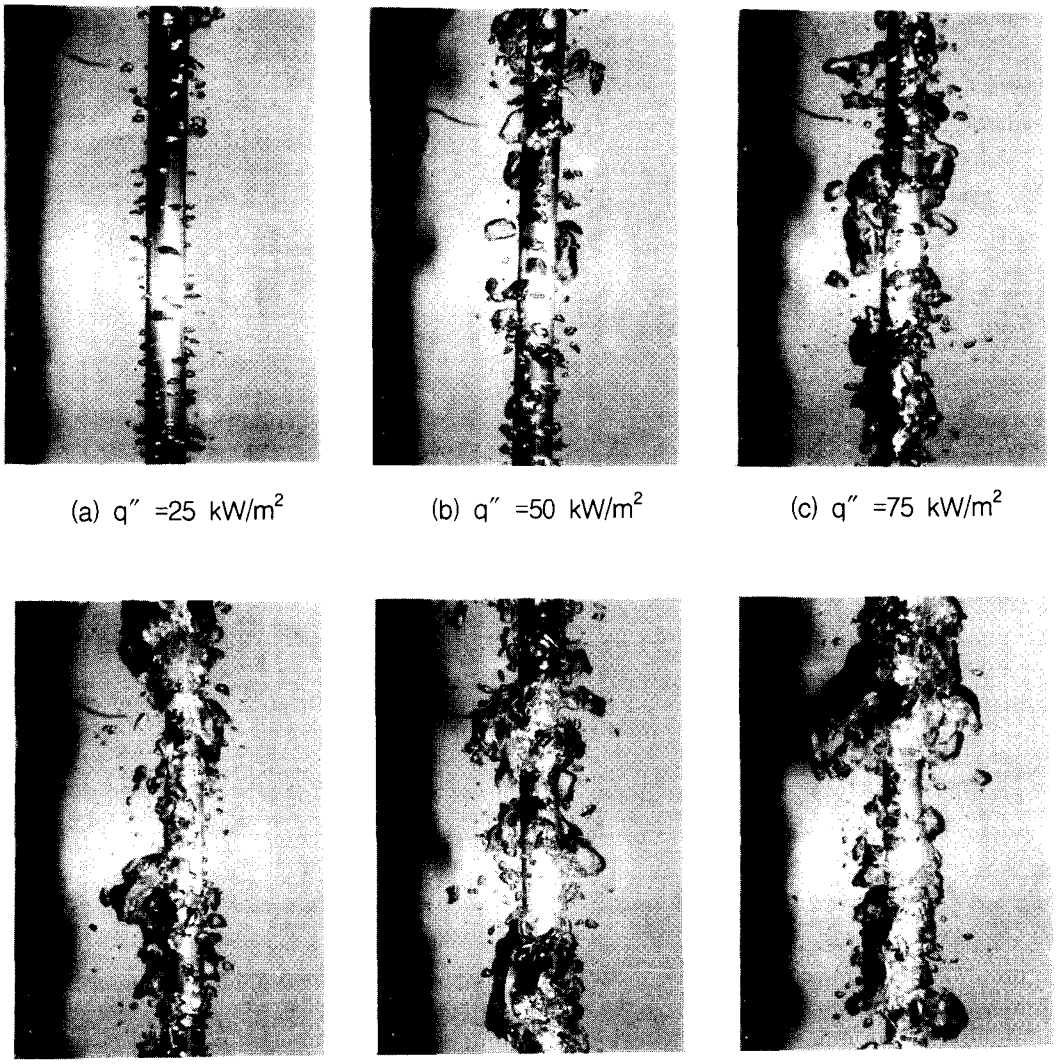
(a) q″ =25 kW/m^2 (b) q″ =50 kW/m^2

(c) q″ =75 kW/m^2 (d) q″ =100 kW/m^2

(e) q″ =125 kW/m^2 (f) q″ =146 kW/m^2

Photo. 1. Bubble Generation and Coalescence On the Horizontal Heated Tube Surface (D=19.05mm, ε=60.9nm) for Various Heat Fluxes

fer mechanisms, i.e., (1) enhanced heat transfer due to liquid agitation by bubbles generated and (2) reduced heat transfer by the formation of large vapor slugs and bubble coalescence is different in two regions of low heat fluxes and high heat fluxes depending on the orientation of the tube and the degree of surface roughness.



(a) q″ =25 kW/m^2 (b) q″ =50 kW/m^2 (c) q″ =75 kW/m^2

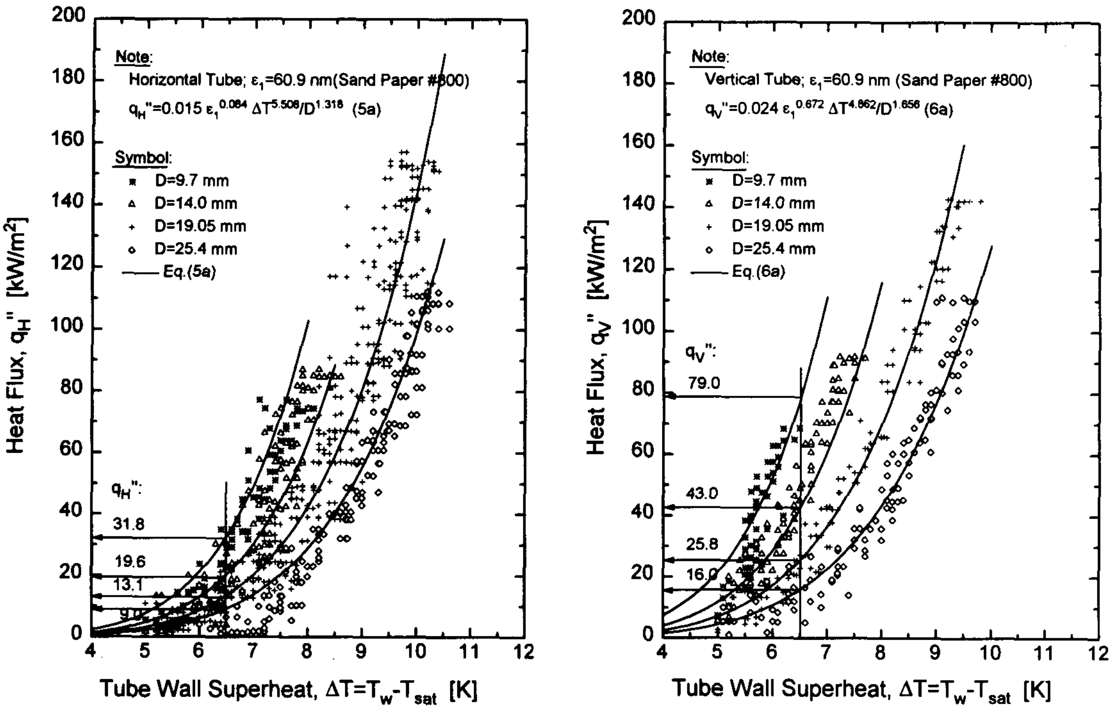
(d) q″ =100 kW/m^2 (e) q″ =125 kW/m^2 (f) q″ =144 kW/m^2

Photo. 2. Bubble Generation and Coalescence On the Vertical Heated Tube Surface (D=19.05mm, ε=60.9nm) for Various Heat Fluxes

4.3. Effect of Tube Diameter On Nucleate Boiling Heat Transfer

The effect of heat exchanger tube diameter on the nucleate boiling heat transfer for both horizontal and vertical tubes can be observed in Fig. 5 where q″ versus ΔT is plotted for four different tube diameters from D = 9.7 to D = 25.4mm. The curves obtained by Eqs.(5) and (6) show that the heat transfer rate decreases as the tube diameter is increased for both horizontal and vertical tubes. This result is in general agreement with previous results obtained by others [6, 7].

For a given surface roughness of the tube, the ef-



(a) Horizontal Tube (b) Vertical Tube

Fig. 5. Heat Flux versus Wall Superheat for Various Diameters with Different Tube Orientation

fect of tube diameter on the nucleate boiling heat transfer for vertical tubes is greater than that for horizontal tubes. For example, as shown in Fig. 5, when the tube diameter is decreased from 25.4 to 9.7mm (~162%), q''_H increases from 9.0 to 31.8 kW/m^2 (~253%) whereas q''_V increases from 16.0 to 79.0 kW/m^2 (~394%).

4.4. Comparison with Existing Studies

A comparison between the measured heat flux q''_{exp} and the calculated heat flux q''_{corr} by Eqs.(5a) and (6a) is shown in Fig. 6. This figure indicates that the scatter of the present experimental data is between +30% and −20%, with some exceptions, from the fitted curve of Eqs.(5a) and (6a). In Fig. 7, present experimental data for q'' versus h_b are plotted along with the fitted curve of Eq.(9). This figure also shows that the scatter of the present data for both horizontal and vertical tubes ranges from +35% to −20% considering from the fitted curve of Eq.(9). The scatter of the present data is of similar size to that found in other existing pool boiling data. As noted by others [8], there seems to be some inherent randomness in pool boiling due to the uncertainties associated with nucleation site density, physical conditions of the tube surface and others. This fact precludes greater accuracy of both theoretical and empirical correlations for heat transfer coefficients in nucleate boiling.

In Fig. 8, on the other hand, q'' versus ΔT curves obtained by typical existing correlations of Rohsenow [5], Cornwell [7], Cornwell and Houston [8], and Jakob [9] are compared with the calculated values obtained by present empirical correlations for water boiling on heated surfaces at atmospheric pressure. The wide scatter between the curves shown in Fig. 8

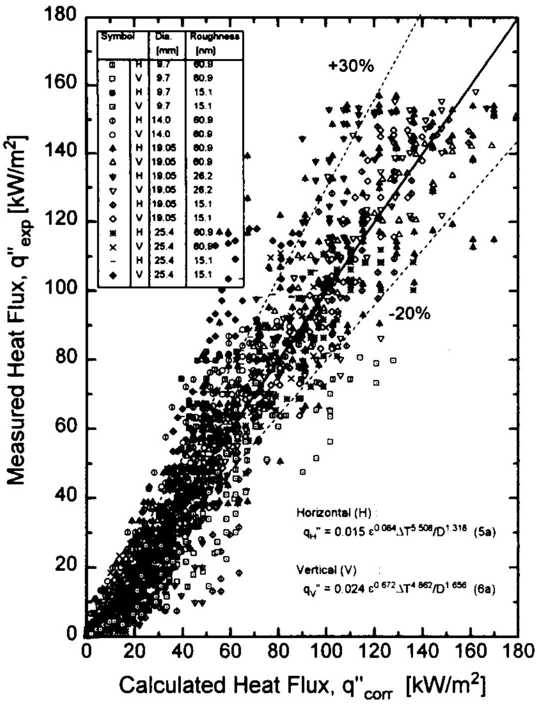


Fig. 6. Measured Heat Flux(q''_{exp}) versus Calculated Heat Flux(q''_{corr}) for Both Horizontal and Vertical Tubes

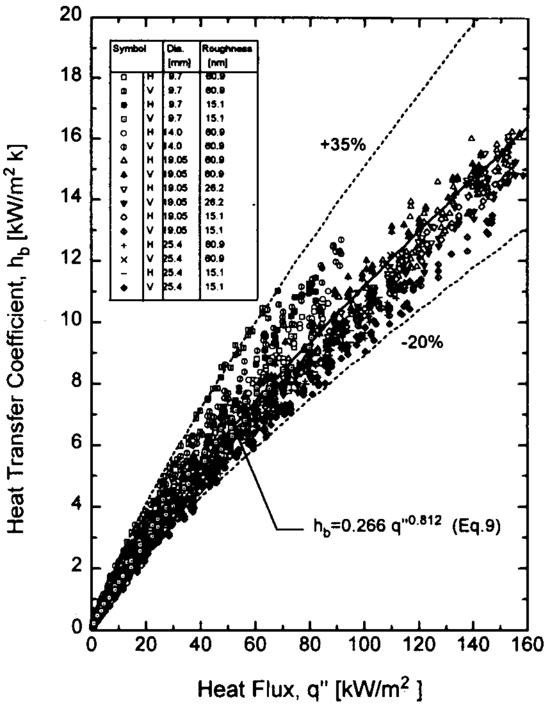


Fig. 7. Variation of Nucleate Pool Boiling Heat Transfer Coefficient versus Heat Flux

is mainly due to the difference in surface roughness ε and orientation of heated tubes. For example, in Rohsenow's correlation [5] $C_{sf} = 0.0132$ is for mechanically polished stainless steel, whereas $C_{sf} = 0.0080$ is for ground and polished stainless steel. Also, in Cornwell's correlation [7], $C_{tb} = 100$ and $C_{tb} = 200$ are for very smooth surface and very rough surface, respectively.

Finally, Eq.(6a), which represents a statistical mean value of the present experimental data for vertical tubes, is compared with PRHR test data of Corletti et al. [1, 4] and Rohsenow's correlation [5] in Fig. 9. From this figure it can be observed that the heat flux q'' for a given ΔT obtained by Corletti et al. [4] is about one order of magnitude smaller than that of the present or Rohsenow's correlations. The main reason for this difference may be attributable to the difference in heat exchanger tube geometries. That is, when the tube length(L) and the space between the tubes (i.e., pitch) of the PRHR heat exchanger test tubes of Corletti et al. [1] are compared with the present work, their tube length is about 10 times longer (i.e., L = 5,486.4mm) while their tube pitch (38.1mm) is only 38% of the present work. When the heated tubes are extremely long and their pitches are very small, the effect of bubble coalescence and the formation of large vapor slugs could be magnified, which could effectively reduce heat transfer from the tube surface. One possible explanation for the reduced boiling heat flux offered by Corletti et al. [4] is the vertical orientation of the tubes, which prevents the tube from being in a fully developed nucleate boiling regime.

5. Conclusions

An experimental parametric study of a tubular heat exchanger has been carried out under nucleate pool

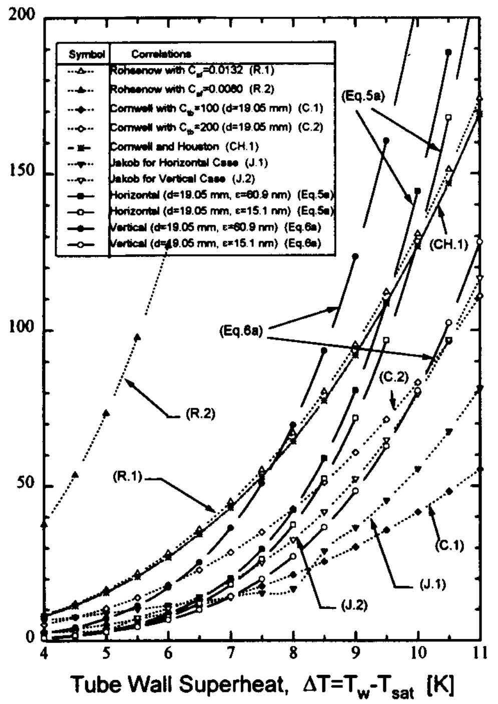


Fig. 8. Comparison of Present Correlation with Typical Exostomg Cprre;atopms fpr q'' versis ΔT

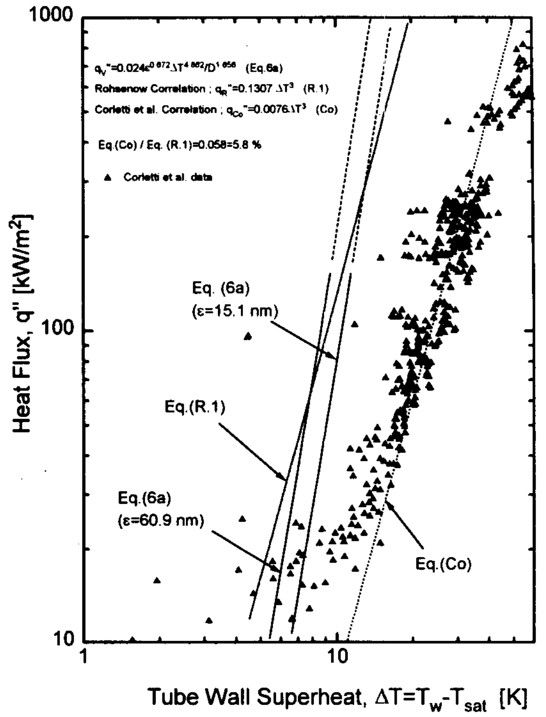


Fig. 9. Comparison of the Present Work with Exisiting Correlation [5] and PRHR Test Data [4]

boiling conditions for an application to the thermal design of a passive residual heat removal system of advanced light water reactors. To determine the combined effects of major parameters of heat exchanger tubes on the nucleate pool boiling heat transfer in a scaled in-containment refueling water storage tank, a total of 1,966 data (1,076 with horizontal tubes and 890 with vertical tubes) for heat flux versus wall superheat has been obtained with various combinations of test parameters of tube diameter, surface roughness, and tube orientation. Main conclusions of the present experimental results are as follows :

(1) Increased surface roughness enhances heat transfer for both horizontal and vertical tubes. However, the effect of surface roughness on the nucleate boiling heat transfer for vertical tubes is significantly greater than that for horizontal tubes. The reason for this is partly because the liquid agitation effect of bubbles generated is more pronounced in vertical tubes with rough surface.

(2) The effectiveness of the two heat transfer mechanisms, i.e., enhanced heat transfer due to liquid agitation by bubbles generated and reduced heat transfer by the formation of large vapor slugs and bubble coalescence is different in two regions of low heat fluxes ($q'' \leq 50$ kW/m^2) and high heat fluxes ($q'' >$ 50 kW/m^2) depending on the orientation of tubes and the degree of surface roughness.

(3) The heat transfer rate decreases as the tube diameter is increased for both horizontal and vertical tubes. For a given surface roughness of the tube, the effect of tube diameter on the nucleate pool boiling heat transfer for vertical tubes is greater than that for horizontal tubes.

Two empirical heat transfer correlations for q'', one for horizontal tubes and the other for vertical tubes, are obtained in terms of surface roughness (ε) and tube diameter (D) as given by Eqs.(5) and (6). In addition, a simple empirical correlation, Eq.(9), for nucleate pool boiling heat transfer coefficient h_b is obtained as a function of heat flux (q″) only. The overall scatter of the present data ranges from +35% to −20% from the fitted curve of Eq.(9).

Acknowledgement

The authors gratefully acknowledge the financial support of the Korea Science and Engineering Foundation.

Nomenclature

A heat transfer area, m^2
C_p specific heat, J/kg-K
C_1 empirical constant between q″ and geometric parameters
C_5 empirical constant between h_b and q″
C_{sf} empirical constant to express effects of liquid and heating surface combination in Rohsenow's correlation
C_{rb} empirical constant to express surface roughness effect in Cornwell's correlation
D tube outer diameter, m or mm
E supplied voltage, V
h_b boiling heat transfer coefficient, kW/m^2-K
h_{fg} enthalpy of vaporization, J/kg
I supplied current, A
k thermal conductivity, W/m-K
L tube length, m
Nu Nusselt number
Pr Prandtl number
p liquid pressure, Pa
p_R reduced pressure
q″ heat flux, kW/m^2
q total heat transfer, kW
Re Reynolds number
T temperature, K
△T degree of superheat of the heating surface (Tw-Tsat), K

Greek Symbols

ε average tube surface roughness in rms value, nm
μ viscosity, kg/m-s
ρ density, kg/m^3
σ surface tension of liquid-vapor interface, N/m

Subscripts

b boiling heat transfer
corr correlation
exp experiment
f fluid
g vapor
H horizontal tube
sat saturation state
t tube
V vertical tube
w tube surface wall

Superscripts

C_2 empirical constant to express surface roughness effect
C_3 empirical constant to express wall superheat effect
C_4 empirical constant to express tube diameter effect
C_6 empirical constant to express heat flux effect

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