Forced Convection Boiling Heat Transfer from a Horizontal Cylinder to Subcooled Water

Sung Hong Lee*, Euk Soo Lee**

Key Words : Forced Convection Boiling, Local Temperature Surface Distribution, Boiling Curve, Degree of Subcooling at Inlet

Abstract

This investigation presents the experimental results of forced convection boiling heat transfer around a circular, electrically heated horizontal cylinder to subcooled water in cross flow. In these experiments, the following primary variables were included: heat flux, flow velocity, pressure and degree of subcooling at inlet. Local surface temperatures were measured at nine peripheral positions. Local surface temperature distributions are classified into four categories depending on the supplied heat flux. The effects of the boiling curve depending on the fluid velocity, degree of subcooling at inlet and pressure are presented.

Nomenclature

- a : constant in [Eq.(2)]
- b : thickness of the test tube, [m]
- D : outside diameter of the test tube, [m]
- h : heat transfer coefficient, [W/m²K]
- k : thermal conductivity, [W/mK]
- n : constant in [Eq.(2)]
- P : pressure, [kPa]
- q'' : heat flux, [W/m²]
- $\dot{q}$: heat generation per unit volume, [W/m³]
- r_o : outside radius of tube, [m]
- r_i : inside radius of tube, [m]
- T : temperature, [℃, K]
- ΔT_s : wall superheat, [$T_w - T_{sat}$]
- ΔT_{sub} : subcooling, [$T_{sat} - T_f$]
- V : liquid freestream velocity [m/s]
- K^* : dimensionless parameter, [$k_\infty r_o / k_w b$]
- Re : Reynolds number

* Professor, Dept. of Mechanical Engineering, Pusan National University
** Researcher, RIMT, Pusan National University

Superscript

— : average

Subscript

b : boiling
c : forced convection
f : fluid
i : inside of tube
o : outside of tube
s : superheated
sat : saturation
sub : subcooling
w : outside surface of wall
∞ : free stream fluid

1. Introduction

Boiling heat transfer is an efficient means of energy transfer. Design of heat exchanger equipment or power plants depend on knowledge of boiling. Understanding of heat transfer and fluid flow related to boiling from a tube or tube bundles has been greatly advanced in recent years, but the investigations are mostly concerned with the cases of axial flow along the tubes. Consequently, the knowledge of heat transfer in boiling from a tube or tube bundles in cross flow conditions is relatively limited. However, the fluid velocity in most heat exchanger can not be classified as purely axial flow due to the presence of the baffles which ensure that shell-side fluid will flow across the tube bundles. Examples are kettle reboiler tube bundles and shell side boiling in both horizontal forced flow and horizontal thermosyphon reboilers. Some of the studies of the problems will be mentioned below.

Fink et al.[(1)] studied the effect of flow locity and the degree of liquid subcooling on crossflow nucleate boiling of a mixture of freon-11 and freon-113 flowing normal to an electrically heated copper tube. The nucleate boiling heat transfer coefficient was evaluated from the mean heat flux and the mean wall superheat. Fand et al.[(2)] conducted an experimental study using an electrically heated horizontal cylinder of diameter approximately 12mm in a water tunnel of diameter 86.6mm with vertically upward flow. They investigated the effects of heat flux, flow velocity, degree of subcooling, surface material and pressure. An empirical correlation equation was developed to predict the performance of a cylindrical heater in crossflow boiling from the single phase crossflow heat transfer coefficient and pool nucleate boiling heat transfer coefficient. Boiling heat transfer from electrically heated cylinders at atmospheric pressure in a pool and in forced flow normal to the cylinder axis was studied by McKee and Bell[(3)]. The cylinders were 0.25 to 0.71 in. O.D. and were mounted in a 3 square in. cross-section channel. The working fluid was water. The flow velocities were 3.42 and 5.44 ft/s, and are subcooling ranged from 3 °F to 14 °F. The flow boiling curves of q'' and ΔT in cross flow are not strongly displaced from the corresponding pool boiling curves. They found that the surface temperature did not vary greatly with position around the periphery during nucleate boiling.

The flow velocity effect on the entire boiling curve was discussed by Yilmaz and Westwater[(4)]. They measured pool boiling and crossflow boiling heat transfer in Freon-113 from a horizontal cylinder at near atmospheric pressure. The cylinder was a steam-heated copper tube. The upward flow velocities were relatively high; they were 2.4, 4.0 and 6.8m/s. The entire boiling curve was found to be sensitive to flow velocity in the forced convection region. The boiling curves at different velocities did not intersect.

However, for crossflow boiling, most investigators seldom include the effect of variation in local surface temperature. Given uniform heat generation within a heater placed in a cross fluid flow boundary condition, heat flows by conduction within the wall of the heater and creates a non-uniform wall surface temperature distribution. Lee et al.[5], studied the effect of the circumferential wall conduction on heat transfer from a cylinder in pool nucleate and film boiling. They found that the local heat transfer is strongly influenced by circumferential heat conduction which is dependent on the thermal conductivities of the cylinder wall material and the fluid, the wall thickness, and the cylinder diameter.

The present study investigates the effects of velocity, degree of subcooling at inlet, and the system pressure on local boiling heat transfer from an electrically heated horizontal tube to water in cross flow.

2. Experiment

2.1 Boiling heat transfer loop

The experimental heat transfer system used in this investigation is a boiling heat transfer test loop as shown in Fig.1. The facility can handle many different kinds of working fluid. Since all of the components in contact with circulating fluid are made of Lexan, stainless steel and teflon, a variety of liquids can be used without any corrosion problem. The working fluid in this investigation is demineralized and degassed water. The fluid is circulated by a centrifugal pump through control valves, bypass circuits, flow metering devices, preheaters, test section, a water-cooled condenser, a degassing chamber and back to the suction side of the pump. The purity of the working fluid is controlled through a conditioner and filter system attached to the loop. The inlet fluid flow conditions to the test section are regulated by power input to the preheaters. The preheaters are also constructed from stainless steel casing with immersion heaters which are regulated by temperature controllers. The controller can vary the preheater output power from 0 to 60 kW. For preheater protection, a flow switch is used. This flow switch automatically turns off the preheater when the flow is lost for any reason. The condensing heat exchanger is a shell-and-tube single pass, counterflow type. The cooling water is from the city water supply in the laboratory. The vent on top of the degassing chamber is connected to a compressed air cylinder, an accumulator, and a vacuum pump through valves as shown in Fig. 1. The accumulator is made of Stainless Steel 304 and equipped with three 6 kW heaters which can be used to preheat the test fluid entering the loop. The tank has also a safety valve, a pressure gauge on top and a sight glass on the side. The maximum pressure in the accumulator is designed to be

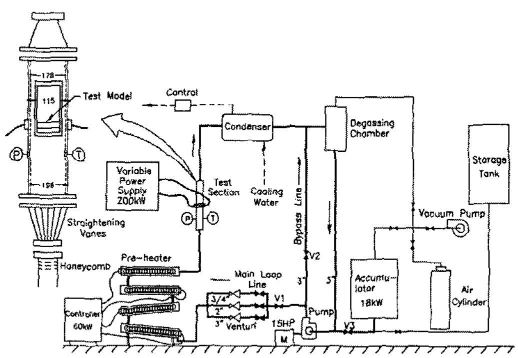


Fig.1 Schematic diagram of boiling heat transfer test loop.

2,100 kPa. The accumulator is connected to the loop and the storage tank. The volume of the storage tank is about 480 liters. Three venturies are used to cover the entire range of the test liquid flow rate from near 0 to 32 ℓ/s. The venturies are connected to a differential pressure transducer and U-type mercury manometer with a vernier scale.

The test section consists of a vertical tunnel of inside cross-sectional dimensions of 178mm ×76mm and a length of 760mm. It contains an electrically heated horizontal test heater. Two-windows, made of Lexan, are provided in the middle part of the test section for visual observation. Stainless steel honeycomb, and straightening vanes are used to recover pressure without flow separation, to reduce velocity fluctuations, and to produce a uniform velocity profile near upstream of the test section.

2.2 Experimental procedure

The boiling heat transfer loop is filled with demineralized water. To ensure accuracy of the data, the fluid is thoroughly degassed prior to any testing. This degassing is accomplished by boiling water in the preheaters and condensing in the condenser for several hours after adding water to the system. The liberated gases in the degassing chamber are vented to atmosphere.

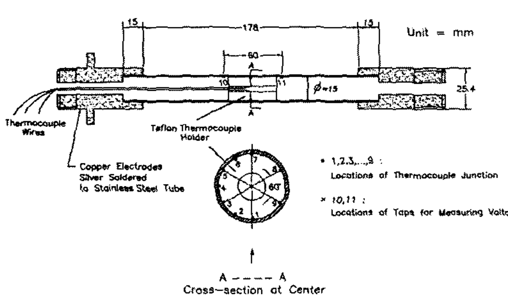


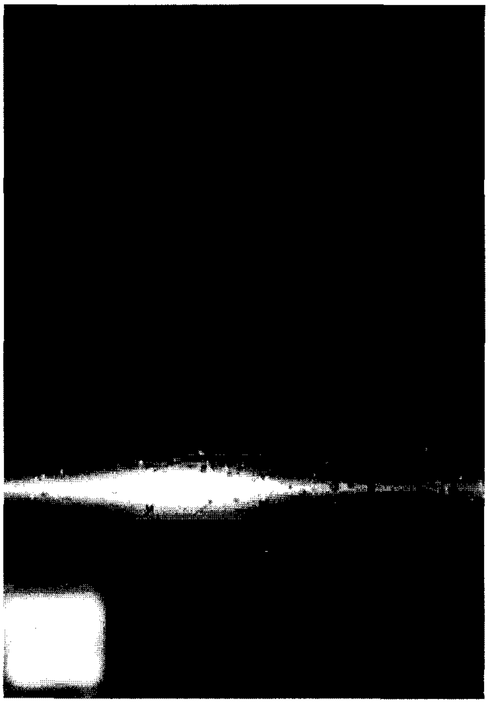
Fig.2 Details of test model (heater) and the detailed locations of thermocouples.

After a 10~12 hour long degassing, the loop is closed to prevent any air from going back into water. At this point, experiment begins. The flow rate through the test section is set at predetermined levels by adjusting the main globe valve and a bypass valve. The system pressure is maintained through the cover gas in the accumulator and by controlling the cooling water flow rate in the condenser. The upstream water temperature is controlled by the cooling water flow rate and the preheater heat flux.

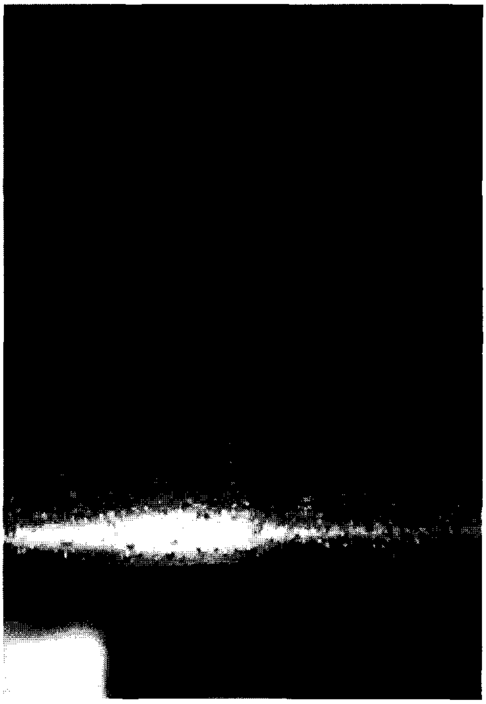
After the flow conditions are set, the flow loop is allowed to come to a steady state before any data are taken; this process takes approximately 1 hr. The heat flux is then set, the test section reaches thermal equilibrium, and data are then taken.

2.3 Test model and data reduction

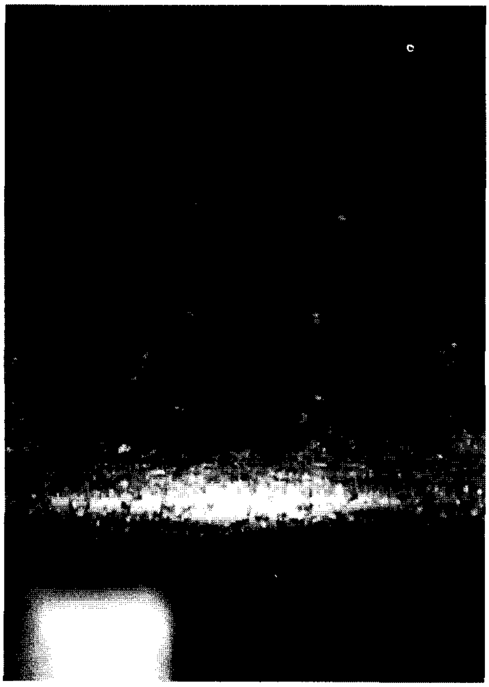
Test tube with an O.D. of 15mm and thickness of 0.8mm, made of Stainless Steel 304, is used. The test model is shown in detail in Fig. 2. The test tube is heated by using a direct electrical resistance of the metal tube itself. Nine thermocouples (T-type, 36Gage) are located around the tube. They are held tightly against the inside of tube wall by cylindrical teflon holder. Thermocouple beads are electrically insulated from the test tube with a thin coating of epoxy(thermal bond, Omega). Great care is taken to maintain the tube surface by polishing it with silicone-carbide paper of grit sizes 800~1200. The mean surface roughness is 0.03μm~0.04μm. Since the temperatures measured on the inside tube wall, the outside wall temperature may be estimated approximately by the following equation,



(a) q'' = 184.03 kW/m² ΔT_s = 7.65℃

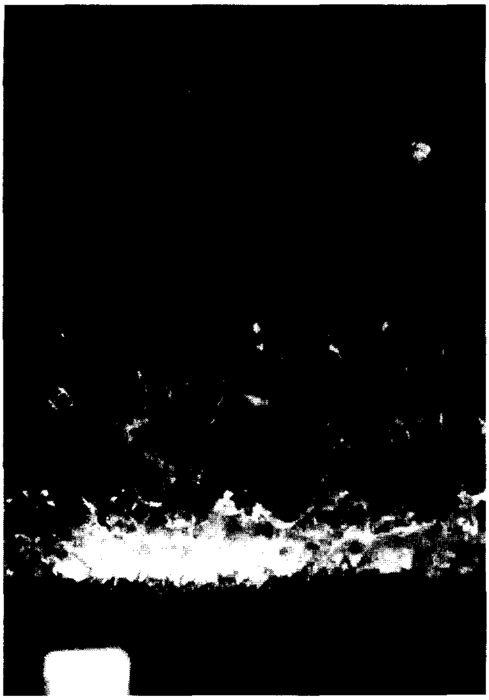


(b) q'' = 350.93 kW/m² ΔT_s = 13.69℃



(c) q'' = 553.06 kW/m² ΔT_s = 15.11℃

(d) q'' = 853.06 kW/m² ΔT_s = 16.99℃



(e) q''= 1044.92 kW/m² ΔT_s = 17.61℃

Fig.3 Photograph of forced convection nucleate boiling at P=163.4kPa, ΔT_{sub}=13.3℃, V=0.3m/s.

$$T_w = T_i - \frac{\dot{q}}{2k_w}\left[\frac{1}{2}(r_o^2 - r_i^2) + r_i^2 \ln\frac{r_i}{r_o}\right] \quad (1)$$

To heat the test tube electrically, two copper conductors are silver-soldered into the test tube. The circumferential wall temperature distribution is measured at every 30° interval for one half of circumference and every 60° for the other half. The test tubes are heated by a D.C. power supply with a 200 kW, 5 000A capacity. The electrical power input to the test heater is measured directly with two leads embedded in the tube surface 60mm apart in the central portion of the test section. This eliminates uncertainty in estimating the long electrical lead losses in the power supply line, and, thus, little error is introduced in calculating the heat flux. All of the test data are acquired and reduced through a data acquisition system consisting of Digistrip 4SPLUS(Kaye, 32 isolated status input) and a PC. The raw and reduced data are stored in the PC. Experimental uncertainty is estimated according to the Kline-McClintock method[6]. It yields maximum values of ±1.1 percent in velocity, ±3.1 percent in heater surface temperature, ±1.2 percent in heat flux. Temperature uncertainty includes the effects of thermocouple calibration and data acquisition errors. The dominant factor is inherent inaccuracy in the thermocouple itself.

3. Results and Discussion

3.1 Visual observations

Simultaneous measurements of pressure, velocity, power input to the test model, and surface and bulk water temperatures are possible. From these, all of the other relevant parameters can be calculated. Experimental conditions are given in Table 1. Experiments are done at three different velocities, V = 0.3, 0.5 and 0.8m/s, with two values of subcooling (at the test section inlet), ΔT_{sub}=13.3 and 23.3℃, at pressures of 204.8 and 163.4 kPa. The test section contains a rectangular Lexan window through which the test model can be observed and photographed. All of the photographs in this paper are taken at the camera shutter speed of 400 frames/sec. Fig. 3 shows data for Test No. 4 of table 1.

The photographs demonstrate that the number of nucleation sites on the heated tube with simultaneous boiling and forced convection in crossflow increase with increasing power input. At low heat fluxes, prior to visual boiling, the tube surface shows a polished appearance owing to interaction of light with the thermal boundary

Table 1 Experimental conditions of the test model in water. (O.D. = 15mm, b = 0.8mm, $K^* = 0.3832$)

Test No.	Pressure [kPa, abs.]	T_{sat} [℃]	Velocity [m/s]	Degree of subcooling [℃]
1	204.8	120.81	0.3	13.3
2	204.8	120.81	0.5	13.3
3	204.8	120.81	0.8	13.3
4	163.4	113.74	0.3	13.3
5	163.4	113.74	0.3	22.3

layer. With a sufficient heat flux, nucleation starts first on the rear half of the tube at random locations. These first-stage nucleation sites are few, and they are independent of one another. They emit vapor bubbles at random frequencies [see Fig. 3(a)]. As heat flux is increased, the density of sites increases and becomes more uniform, and nucleation spreads to forward regions of the tube against incoming flow direction [see Fig. 3(b)]. Bubbles thus initiated on the forward half grow and move around the tube due to liquid drag and buoyant effects. They separate individually from a tube somewhere downstream of the 90-degree position [see Fig. 3(c)]. With a further increase in heat flux, the percentage of vapor in the two phase mixture in the wake region behind the tube increases to the point that a vapor cavity forms in this region. The transition from the individual bubble separation to this type of cavity formation takes place gradually[see Fig.3(d)]. In the forward region of the tube, the influence of moving fluid is evident. In crossflow nucleate boiling, an increase in both flow velocity and degree of subcooling decreases the bubble diameter. Individual bubbles appear to be spherical and separated from each other by varying distances. Bubble coalescence in the vicinity of an active site also occurred, intermittently forming long vapor streams[Fig. 3(e)]. The bubbles are apparently moving at the velocity of the outer flow very shortly after they leave the surface.

3.2 Local temperature distribution

Circumferential surface temperatures at nine different positions on the test tube, for three values of flow velocity(V = 0.3 m/s, 0.5 m/s, 0.8 m/s) are shown in Fig. 4. The effect of flow velocities on the local surface temperature distribution is significant for a fixed value of K^* [13, 14]. K^*, which may be deduced from the governing energy equation, should be used to characterize peripheral wall heat conduction of the heater(tube) in addition to other parameters governing the heat transfer process. The special cases, $K^* = \infty$ and 0, yield the constant heat flux and constant temperature boundary conditions, respectively. As shown in this figure, there is no significant difference between measurements on the right and left sides of the test tube. The amplitudes of variations in local surface temperature distribution are significant (depending on velocity) and increase with increasing flow velocity. While the minimum surface temperature seems to occur at the forward stagnation point for the cross flow velocity of 0.3m/s, the minimum temperature point moves toward the angle of about 120° with a decreasing flow velocity in Fig. 4.

Circumferential wall temperature distribution depends greatly on the heat flux as illustrated in Fig. 5. Four different patterns are observed.

(1) For the heat flux range from 75 to 108 kW/m², local temperature distributions are mainly influenced by the cross flow velocity. This result is similar to those observed in a single phase flow study for Re $\leq$ 100,000 [7]. Starting at the stagnation point, wall temperature increases with increasing θ due to laminar boundary layer development. However, a maximum is reached near the 90 degree location. From this point, the wall temperature decreases with θ due to vortex formation.

(2) For the heat flux range from 149 to 192 kW/m², isolated bubbles form on the rear half of the tube at random nucleate sites and separate from the surface[see Fig. 3(a)]. Wall temperature variation is characterized by 4 minima around a whole(360 degree) surface. For the conditions of Re$\geq$ 100,000 [7], the local wall temperature distribution in cross flow of air is similar to that in cross flow boiling of water in this range.

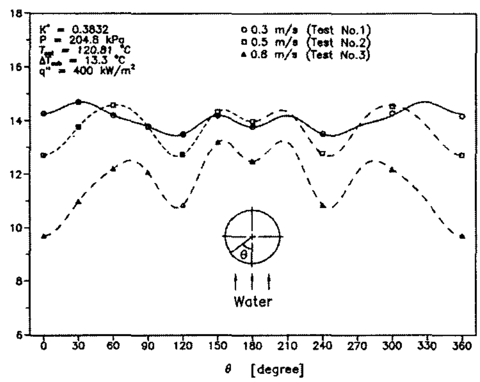


Fig.4 Circumferential surface temperature distribution in flow boiling.

(3) For the heat flux range from 236 to 260 kW/m², more nucleation sites become active and increased bubble formation causes bubble interference and coalescence. The highest local wall temperature may happen near the 60 deg location, and surface temperature distribution shows a relatively small variation over the surface compared with the lower heat flux cases or may show a transition phenomena.

(4) For the heat flux q″ range from 350 kW/m² to 956 kW/m², the minimum wall temperature shifts from about 0 deg to about 120 deg. This change may be caused by the increasing fluid and vapor velocity near the 90 deg location, the high density of nucleation sites, and high vapor quality in the forward stagnation region. Fand et al.[2] found that the forward stagnation region and the rear stagnation region of the cylinder are the regions of the highest density of nucleation sites. Wege and Jensen [8] showed that heat transfer coefficient was the highest at the ±90 deg positions.

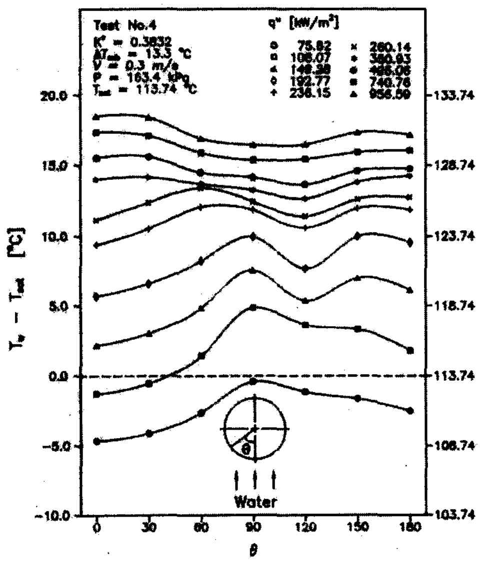


Fig.5 Circumferential surface temperature distribution depending on heat flux in flow boiling.

The local temperature distribution for Test No. 4 condition is shown in Fig.6. Wall temperature measurements for water indicate a large variation in the circumferential wall temperature. The heat transfer rate in forced convection boiling varies greatly at heat flux q″ of about 260 kW/m², or at a ΔT of about 12℃ on the boiling curve. Two distinct regions are found. Below or left side of this point the forced convection heat transfer is dormant. Beyond (right side of) this point the heat transfer is seems to be governed by nucleate boiling. ⓐ~ⓔ data points in Fig. 6 refer to photographs in Fig. 3.

3.3 Effect of velocity, pressure and subcooling on boiling curve

Figures 7, 8, and 9 show the relationship between the wall heat flux and the wall super heat at different parameters. The full lines in Fig. 7 and Fig. 8 are for the wall heat flux, q_b'', based only nucleate boiling. In this region heat transfer is customarily expressed by the following equation [8].

$$h_b = a\,(q_b'')^n \tag{2a}$$

thus

$$q_b'' = (a\,\Delta T_s)^{\frac{1}{1-n}} \tag{2b}$$

The dashed lines are for the wall heat flux q_c'' based only on forced convection at the flow velocities shown and evaluated from

$$q_c'' = h_c(T_w - T_f) = h_c(\Delta T_s + \Delta T_{sub}) \tag{3}$$

The curve-fit values of the constants a and n in Eq. (2b) and h_c in Eq. (3), are given in the figures. Fig.7 represents the effect of the flow velocity on the surface heat flux in the crossflow nucleate boiling for a fixed value of K*. It can be seen that the effects of flow

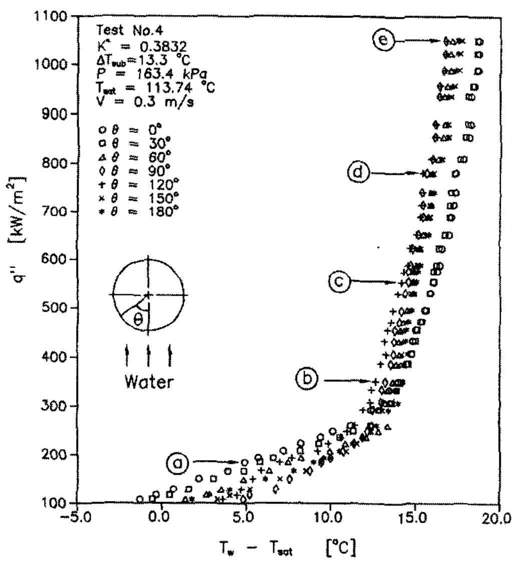


Fig.6 q″ vs. ($T_w - Tsat$)distribution around a horizontal circular tube in flow boiling of water.(refer to Fig.3 a, b, c, d, e)

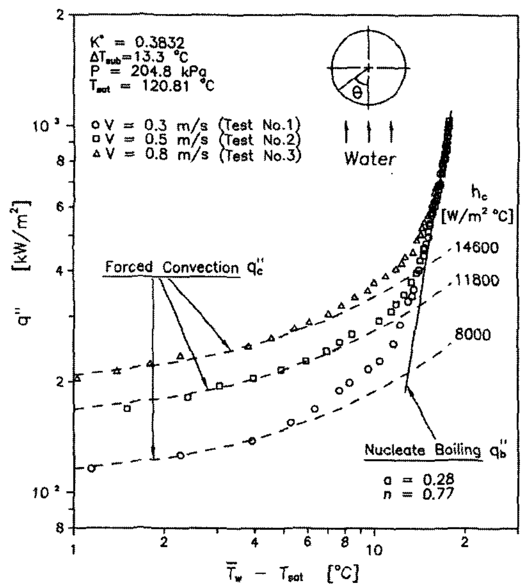


Fig.7 Effect of velocity on heat flux in flow boiling of water.

velocity at low wall superheat range ($q'' \leq 700$ kW/m²) are significant. In this region, heat transfer is governed by forced convection. An increase in forced convection improves the nucleate boiling heat transfer, and the boiling curves shift upward with increasing flow velocity. However, at higher wall superheat range, the effect of flow velocity seems to disappear, and there appears to be a single curve representative of fully developed nucleate boiling.

In Fig.8, heat flux versus wall superheat are plotted for different degrees of subcooling. In the low wall superheats of $q'' \leq 400\,\mathrm{kW/m^2}$, heat transfer rate increases as the degree of subcooling. But in the region of higher wall superheats of $q'' \gg 400\,\mathrm{kW/m^2}$, the effect of subcooling is negligible. The effects of velocity and subcooling dominate at lower wall superheat range; however, at higher wall superheats, these effects are negligible. The effects of system pressure, together with a pool boiling correlation from the literature[8] on the boiling curve, are shown in Fig. 8. At moderate to low wall superheats, an increasing pressure increases the heat transfer rate. But the pressure effects in the present range are smaller than those of velocity and subcooling.

Two methods have been suggested for correlating heat transfer with simultaneous boiling and forced convection. The first method, proposed by Rohsenow[10], consists of a simple superposition of effects as expressed by the following equation

$$q'' = q_c'' + q_b'' \tag{4}$$

where q'' is the total heat flux, and q_c'' and q_b'' represent forced convection and nucleate pool boiling heat fluxes, respectively. This method requires knowledge of nucleate pool boiling. The second approach, formulated by Kutateladze[11], is embodied in the following equation

$$\frac{h}{h_c} = \left[1 + \left(\frac{h_b}{h_c}\right)^2\right]^{0.5} \tag{5}$$

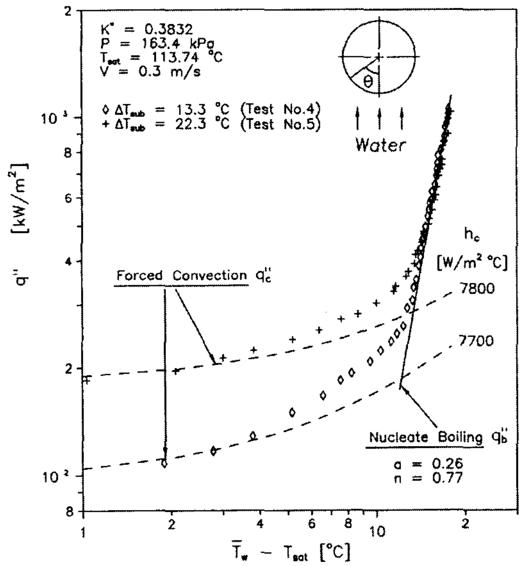


Fig.8 Effect of subcooling on heat flux in flow boiling of water.

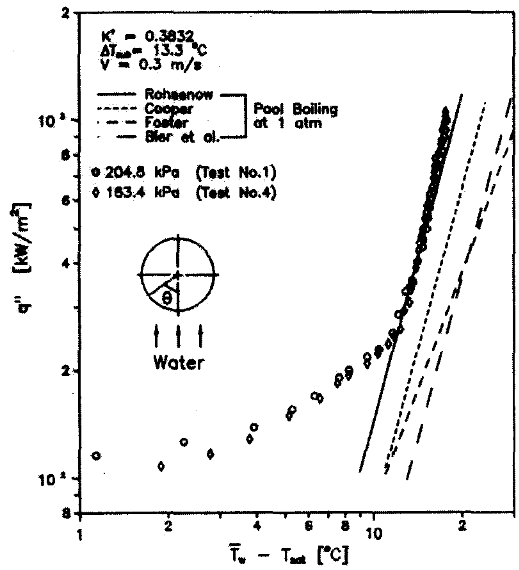


Fig.9 Effect of pressure on heat flux in flow boiling of water.

Kutateladze found that Eq.[5] correlated the results of several experimental studies of boiling water flowing inside tubes. Bergles and Rohsenow[12] stuided both pool boiling and in forced convection surface boiling. Pool boiling data were taken under saturated and subcooled conditions for surfaces similar to those used in forced convection surface boiling. They found that the curves for forced convection boiling cannot be based on the data for saturated pool boiling but must rather be based on the actual forced-convection data. So, a simple interpolation equation which satisfied the characteristics of the boiling curve was suggested

$$q'' = q_c''\left[1 + \left\{\frac{q_b''}{q_c''}\left(1 - \frac{q_{bi}''}{q_b''}\right)\right\}^2\right]^{0.5} \qquad (6)$$

where q_{bi}'' is the fully developed boiling heat flux at incipient boiling. The heat flux q_{bi}'' is calculated from Eq. (2b) at the point of boiling incipience. Since $q_{bi}'' \ll q_b''$ for most of the nucleate boiling range, q_{bi}''/q_b'' may be set to zero in Eq.[6] without a significant error. Fig.10 shows a comparison between the experimental data and the calculated prediction based on super-position model for Tests No. 1 and No. 5. The maximum deviation between the prediction and the experimental data is about 11 percent.

4. Conclusion

Experiments on forced convection boiling heat transfer from a horizontal tube to water have been performed. All of the following prim ary quantities were varied: heat flux, crossflow velocity, pressure and subcooling. Local and average temperature distribution have been reported for variety of fluid conditions. Some high speed motion picture studies were made of bubble motions during boiling. The following conclusions can be drawn from this investigation.

(1) Local surface temperature distribution depends on the boiling heat flux. Surface temperature distributions are classified into four different groups depending on the supplied heat flux.

(2) For low wall superheats, increases in flow velocity and degree of subcooling at inlet increase both single phase convection heat transfer and boiling heat transfer. But at higher wall superheats, neither flow velocity nor subcooling has significant effect on nucleate boiling heat transfer. The pressure effect in the present range is small.

References

(1) Fink, J., Gaddis, E. S. and Vogelpohl, A., 1982, " Forced Convection Boiling of Mixture of Freon-11 and Freon-113 Flowing Normal to a Cylinder", *Proc. 7th Int. Heat Transfer Conf. Munich*, Vol. 4, pp. 207-212.

(2) Fand, R. M., Keswani, K. K., Jotwani, M. M. and Ho, R. C. C., 1976, " Simultaneous Boiling and Forced Convection Heat Transfer from a Horizontal Cylinder to Water", *Trans. ASME J. Heat Transfer* Vol. 98, pp. 395-400.

(3) McKee, H. B. and Bell, K. J., 1969, " Forced Convection Boiling from a Cylinder Normal to The Flow", *Chemical Engineering Progress Symposium Series*, Vol. 65,. pp. 222-230.

(4) Yilmaz, S. and Westwater,J. W., 1980, " Effect of Velocity on Heat Transfer to Boiling Freon-113", *Trans. ASME J. Heat Transfer* Vol. 102, pp. 26-31.

(5) Lee, Y., Zeng, Y. and Shigechi, T., 1990, "

Conjugated Heat Transfer of Nucleate Pool Boiling on a Horizontal Tube", *Int. J. Multiphase Flow*, Vol. 16, No. 3, pp. 421-428.

(6) S.J. Kline and McClintok, 1953, "Estimating Uncertainty in Single-Sample Uncertainty Analysis", *Mechanical Engineering*, Jan., pp. 3-8.

(7) Giedt, W. H., 1949, "Investigation of Variation of Point Unit Heat Transfer Coefficient around a Cylinder Normal to an Air Stream", *Trans. Am. Soc.* Mech. Engrs 71, pp. 375-381.

(8) Wege, M. E. and Jensen, M. K., 1984, "Boiling Heat Transfer from a Horizontal Tube in an Upward Flowing Two-Phase Crossflow", *Trans. ASME J. of Heat Transfer*, Vol. 106, pp. 849-855.

(9) Hewitt, G. H., Shiress, G. L. and Bott, T. R., 1994, Process Heat Transfer, *CRC Press*, London.

(10) Rohsenow, W. M., 1953, Heat Transfer with Evaporation, University of Michigan Press.

(11) Kutateladze, S. S., 1961, "Boiling Heat Transfer", *Int. J. Heat Mass Transfer*, Vol. 4, pp. 31-45.

(12) Bergles, A. E. and Rohsenow, W. M., 1964, "The Determination of Forced Convection Surface Boiling Heat Transfer", *Trans. ASME J. of Heat Transfer*, Vol. 8, pp. 365 -372.

(13) Lee, Y. and Kakade, S.G., 1976, " Effect of Peripheral Wall Conduction on Heat Transfer from a Cylinder in Cross Flow ", *Int. J. Heat Mass Transfer*, Vol. 19, pp. 1031-1037.

(14) Lee, S.H. and Lee, E.S., 1999, " Effect of Wall Heat Conduction on Convection Heat Transfer from a Circular Tube in Cross Flow ", 2nd Int. Symp. Twp-Phase Flow Modelling and Experimentation, Pisa.