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ORIGINAL

Experimental studies on the effect of tube inclination on nucleate pool boiling

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Abstract Experiments were conducted to study the effect of tube inclination on nucleate pool boiling heat transfer for different tube diameters and surface roughness values. The results show that as the tube is tilted from the vertical to the horizontal, the temperatures at the top and bottom (with respect to circumference) increase and decrease, respectively. The increase and decrease is such that they almost compensate for each other, resulting in very little variation of the average heat transfer coefficient with tube inclination. The increase in bubble sliding length at the bottom wall and decrease in bubble sliding length at the top wall are thought to be the reasons for this behaviour. Experiments have been conducted with water, ethanol and acetone at atmospheric pressure, to confirm similar effects of inclination irrespective of fluid property.

Nomenclature

A	Area of the tube, m^2
D	Diameter of the tube, m
L	Length of the tube exposed to the liquid, m
q	Heat flux, kW/m^2
R_a	Average surface roughness, μm
T	Temperature, K
ΔT	Wall superheat, K

Subscripts

ave	Average
o	Outer
i	Inner
s	Saturation
w	Wall

1 Introduction

Because of high heat transfer coefficients associated with nucleate boiling, it finds applications in power generation, heating and refrigeration processes, nuclear engineering and cooling of high energy density electronic components. Given the complex nature of the boiling process, there continues to be active research interest in this area in spite of having been investigated for decades. Fire-tube boilers, petrochemical industry and chemical process industry in general are some of the areas in which pool boiling on tubes find applications, where the heating surfaces need not always be horizontal but can be vertical or inclined.

Most of the literature available in the area of nucleate boiling seems to be for horizontal surfaces or horizontal tubes, e.g. Cornwell et al. [1] for horizontal tubes, Cooper [2] for horizontal copper and stainless steel cylinders, Cornwell and Houston [3] for horizontal tubes. However, due to its typical applications in areas such as that in a nuclear reactor, some amount of studies on inclined (most commonly with vertical configuration) surfaces have been carried out in the recent past. The Jakob and Hawkins [4] nucleate boiling correlation suggests that the heat transfer coefficient of a vertical heating surface is larger than that of horizontal heating surfaces. But, van Stralen and Sluyter [5] have reported that the heat transfer coefficient on horizontal wires

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in pure liquids is more than that on vertical wires. The first classical study on pool boiling on inclined surfaces was done by Nishikawa et al. [6] showing the effect of surface inclination on the heat transfer coefficient. Their experimental results show that the boiling heat transfer coefficient for downward facing inclined flat surfaces is higher than that for upward facing horizontal flat surfaces. Corletti et al. [7] suggested a correlation based on their experiments on three vertical tubes arranged with 38.1 mm pitch with water at 1 atm filled between them. Tube diameter was 19.05 mm. The correlation is similar to the Rohsenow [8] correlation.

Recently Kang [9] showed a pronounced effect of inclination, but only for smoother tubes (0.015 μm in this case). Since the tubes used here were in general very smooth, the application of this observation on tube inclination to commercial tubes cannot be fixed. Further, this work does not show the circumferential variation of wall temperature (top and bottom wall temperatures) with tube inclination. A variation of wall temperature along the circumference is expected as the tube is inclined from vertical due to the asymmetry of the sliding bubble motion. Moreover, this study was limited to only water boiling at atmospheric pressure. This study attributed the increase in heat transfer coefficient to liquid agitation and decrease in heat transfer to bubble coalescence in general terms.

From the above, it can be concluded that the effect of tube inclination on pool boiling needs further study to understand the phenomenon adequately. From these studies, it is evident that the effect of inclination cannot be treated independently of other parameters important in boiling. Hence, there is a need for investigating the inclination effect in conjunction with parameters like surface roughness, tube diameter and fluid properties. The aim of the present work is to investigate the effect of tube inclination on nucleate boiling for higher surface roughness with tube diameters in the transition region (where a decrease and then increase in heat transfer coefficient with increase in tube diameter was reported by Cornwell et al. [1] for horizontal tubes), with water and two organic fluids (ethanol and acetone) as the working fluids.

2 Experimental apparatus and procedure

2.1 The boiling vessel and the circuit

The experimental circuit consists of a boiling vessel with the heating tube, a condenser having cooling water circulation and an auto-transformer regulated power supply as well as appropriate measurement systems. The boiling vessel, as shown in Fig. 1, is made of a double-walled toughened glass capable of withstanding high temperature, with stainless steel (316) supports at four corners. Silicone sealant, applied in between the glass and supports, and at the corners in order to prevent leak, also absorbs thermal expansion of the glass. The inner dimensions of the glass vessel are 12 cm × 12 cm × 30 cm. Two secondary heaters (cartridge) were fixed to the bottom plate, sufficiently away from the main heater at the two corners diagonally opposite to each other. These heaters are used for preheating the boiling liquid, degasification and for maintaining the boiling liquid at saturation temperature thereby avoiding sub-cooled boiling. The boiling vessel was connected to an external condenser by a stainless steel braided Teflon hose of 20 mm diameter (for carrying vapour from boiling vessel to the condenser) and a similar hose of 8 mm diameter (for carrying condensate from the condenser to the vessel). The condenser is of reflux type. In order to maintain atmospheric pressure while boiling, a valve at the top of the condenser is kept partially open. A link (with left hand threaded bolt and right hand threaded bolt connected by a sleeve) and a hinge at the bottom of the glass vessel, as shown in Fig. 1, facilitates tilting the whole boiling vessel to different angles of inclination required for the present study (from 90° to 0° from horizontal). The power to the main heater and the secondary heaters are adjusted by auto-transformers.

2.2 The heating surface characteristics and temperature measurement

A set of stainless steel (316) tubes act as heating surfaces with outer diameter ranging from 21 to 33 mm (Table 1). Stepped holes (5 mm/3 mm) were drilled on the tube at 6 different locations to insert K-type fiber glass insulated thermocouples with copper buttons (1.5 mm thick and 4.6 mm diameter). These thermocouples were pushed into the tube through the holes drilled ensuring that fiber glass insulation was not disturbed, and were drawn out from inside the tube. Two holes of 0.2 mm diameter were drilled through copper button of 1.5 mm thickness and diameter 4.6 mm. Thermocouple leads (chromel and alumel) were drawn through these holes and silver-brazed from outside. The copper button prevents the flow of liquid silver into the tube at the time of brazing which otherwise may cause a third junction inside the tube where very high temperature gradients exist. Further, a tape capable of withstanding high temperature was applied over the fiberglass insulation and a fine copper wire was wound over this below the copper button so as to keep the fibers of fiberglass insulation intact. Bulges arising from these brazings were shaved off with a very smooth file and subsequently buffed uniformly over the entire surface to remove the slugs formed on the surface. These surfaces with a surface roughness of R_a 0.08 μm act as base surfaces. These tubes were then polished circumferentially with emery cloth 120 and Silicon Carbide paper 80 to

Fig. 1 Boiling vessel

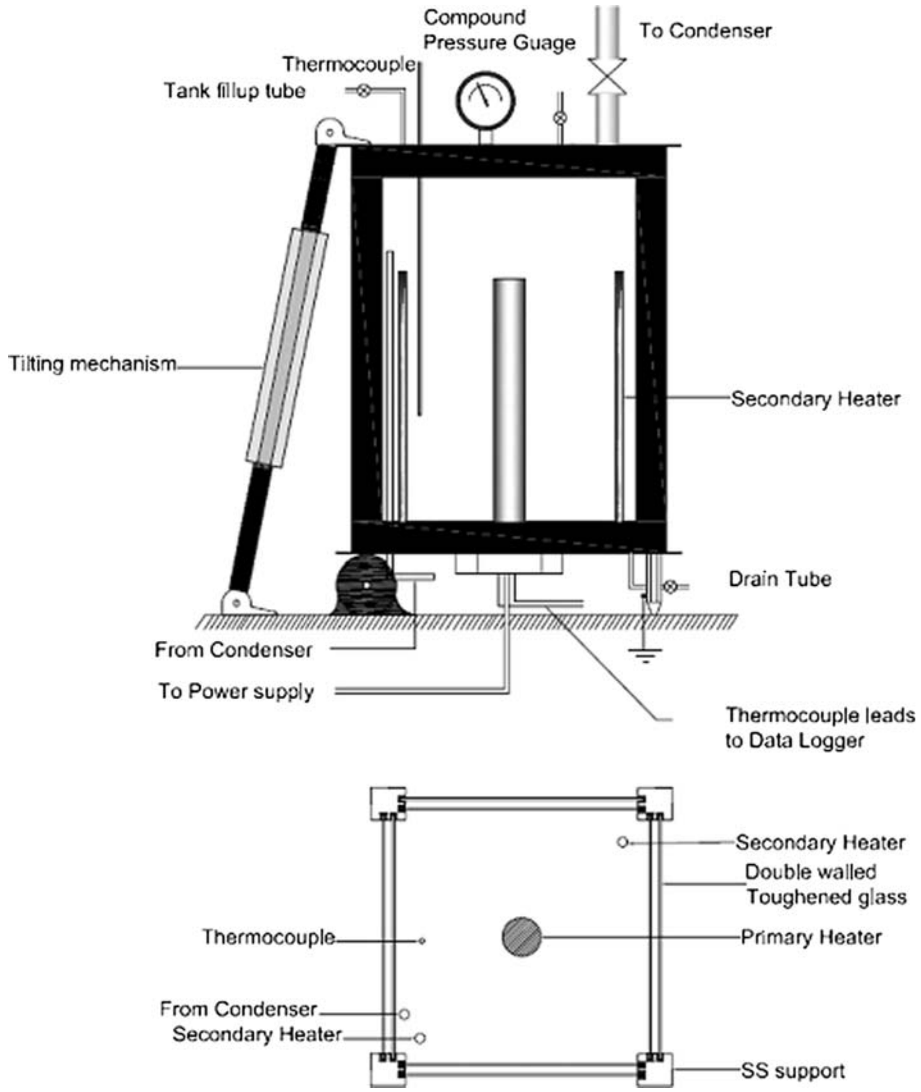


Table 1 Specification of the SS tubes and cartridge heaters

Outer diameter of SS tube (mm)	Inner diameter of SS tube (mm)	Length of SS tube exposed to water	Cartridge heater diameter (mm)	Heating length of cartridge heater (mm)	Power (220 V) (W)
33	24.6	172	16	150	2,700
26	19.1	172	10	150	1,500
21	12.3	172	6	150	700

get higher surface roughness. The roughnesses obtained in these cases are 0.29 and 0.67 µm, respectively. Here the surface roughness measured was axial surface roughness, and the instrument used was a profilometer called Perthometer S2 (Mahr). The tip of the probe of the profilometer is a conical diamond bit with a tip radius of the order of 0.1 µm and with a resolution of 0.01 µm. The traversing distance was 5.6 mm. Roughness (R_a) was measured at 6 different locations on the tube and the average value was taken. The SS tube was press-fitted into a brass holder with Teflon bush in between them to prevent heat transfer to the brass holder. This was sealed with silicone sealant to prevent any possible leak. Teflon disc was fixed at the top of the tube with silicone sealant to prevent heat transfer from the flat surface at the top.

The stainless steel sheathed cartridge heater was then inserted into the SS tube and the gap between the heater and the inner wall of the tube was filled with fine electrolytic copper powder and rammed tightly to promote heat transfer from the heater to the SS tube (Fig. 2). At the bottom, the entire assembly containing copper powder, cartridge heater and SS tube were sealed with silicone sealant. The length of the SS tube exposed to boiling liquid is 172 mm. Heater powers (maximum) were 700, 1,500 and 2,700 W for SS tubes with outer diameters of 21, 26 and 33 mm, respectively. However, the maximum power levels to which experiments were carried out were slightly less than their maximum capacities.

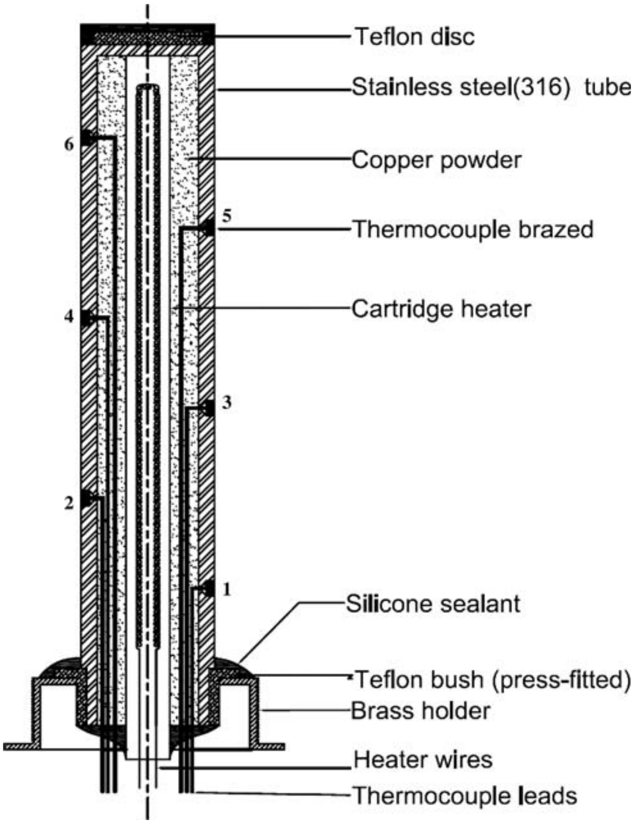


Fig. 2 Cross sectional view of the main heater

For horizontal tubes the results of Kang [9] show that middle wall or side wall (at 90° from top) superheat equals the average of top and bottom wall superheats. Also the heat transfer coefficient equals the average of two local heat transfer coefficients measured at the top and bottom wall. The results of Cornwell and Einarsson [10] for horizontal tube and zero liquid velocity (pool boiling) also show that the average heat transfer coefficient equals the average of the local heat transfer coefficients at the top and bottom of the tube. So in the present case, thermocouples were fixed (brazed) at three different axial locations but at the same circumferential position, and the other three thermocouples were brazed exactly at 180° opposite to the former, but at offset axial locations as shown in the Fig. 2. The SS tube (with main heater) is screwed to the nut welded to the bottom SS plate, and many layers of Teflon tape is applied over the threads of the tube holder before screwing such that the tube can be oriented (rotated), without leakage at the threads, so as to have three thermocouples at the top and three at the bottom (with respect to circumference) when the glass vessel (and hence the tube) is tilted.

Uncertainty of the thermocouple was found to be ±0.1°C. Uncertainties of the extreme cases were determined and it was found that the uncertainties in the wall superheat was limited to 3.5% and that in the heat flux calculated by the method of propagation of error was limited to 1%.

2.3 Data reduction

The power (Q) supplied to the heater is given by the product of Ammeter reading (I) and Voltmeter reading (V)

$$Q = VI.$$

Since heat transfer is blocked at the top flat surface of the tube by Teflon disc, and at the bottom of the tube by Teflon bush, the heat loss from these surfaces can be neglected. Therefore, heat flux is

$$q = Q/A_o$$

where the heat transfer area, A_o is

$$A_o = (\pi D_o)L$$

The wall temperatures were measured using the six embedded thermocouples and arithmetic average of these thermocouple readings was taken as the wall temperature. The temperature drop (δT) across the copper button (thickness 1.5 mm) and the silver (thickness 1 mm) is obtained with a simple conduction numerical model using FLUENT package by applying constant heat flux condition at the inner wall of the SS tube and the constant wall temperature at the outer wall. This temperature drop is deducted from the thermocouple readings so as to get the boiling surface temperature. Average of the readings of the thermocouples at different locations coupled with the correction for temperature drop is taken as the wall temperature (T_w)

$$T_\mathrm{w} = T_\mathrm{ave} - \delta T$$

Thermocouple readings are scanned by a data logger (HP34901A).

2.4 Experimental procedure

First, the liquid is heated to the boiling point by the two secondary heaters and then allowed to boil for about 1 h so as to remove the dissolved gases. During this period, the valve at the top of the condenser is kept partially open so as to maintain atmospheric pressure in the boiling vessel and to allow the dissolved gases escape through this valve. When the liquid is at steady state, thermal equilibrium will be reached between the SS tube and the boiling liquid. Subsequently, the main heater is switched on and the power is slowly increased to the maximum heat flux possible with the heater. Once the maximum heat flux is reached, thermocouple readings are monitored to ascertain the steady state. Temperature readings are scanned at 30 s intervals. Once steady state is reached, ammeter, voltmeter and thermocouples readings are recorded. Then the vessel is tilted, keeping the same power. The same process is followed for different angles of inclination. Then, the vessel (and hence the tube) is brought back to the vertical position and the power is decreased. The same procedure is repeated for different power levels in descending order. As mentioned by Kenning [11], usually heat flux in the decreasing direction is considered for laboratory purposes, and the same practice is followed in the present study.

3 Results and discussion

As the angle of inclination was reduced from 90° with horizontal, the top wall temperature was found to increase and the bottom wall temperature was found to decrease. In the vertical orientation, bubbles all around the tube tend to slide along the surface and parallel to the axis of the tube. When the tube is inclined at an angle from the vertical, bubbles at the top tend to move vertically upwards, i.e. at an angle to the axis of the tube, while those at the bottom tend to slide along and against the surface partly in the circumferential direction (fully circumferential for horizontal tube) and partly in the axial direction making the path helical. As a result, the sliding length of the bubbles originating from the bottom of the tube increases and hence heat transfer coefficient at the bottom increases, thereby decreasing the bottom wall temperature. This gives a difference in the wall superheat between the top and bottom wall as shown in Fig. 3 for the horizontal heater position. The increase in the average top wall temperature was nearly equal to the decrease in average bottom wall temperature. As a result there was no net effect on the heat transfer coefficient. Higher heat transfer coefficient at the tube bottom and the lower heat transfer coefficient at the top was also reported by Cornwell and Einarsson [10]. The distribution of the top and bottom wall superheats for horizontal tubes with 33 mm diameter and 0.67 μm surface roughness at various power levels is shown in Fig. 3. It was also observed (visually) that the nucleation site density at the top wall increased and that at the bottom decreased as the tube was tilted from the vertical, though it could not be recorded experimentally (by photography) due to experimental constraints (as the rising vapour column obstructed the view of the tube top wall). This decrease in nucleation site density at the bottom and increase at the top was also reported by Luke and Gorenflo [12]. The decrease in heat transfer coefficient due to the absence of sliding bubbles at the top more than offsets the increase in heat transfer coefficient due to the increase in nucleation site density at the top, while at the bottom the increase in heat transfer coefficient due to the increase in sliding length dominates over the decrease in heat transfer coefficient due to the decrease in nucleation site density. This shows the importance of sliding bubbles in the process. The vapour bubbles of larger departure diameter (as the buoyancy force is against the tube wall at the underside or bottom side) may also contribute to the increased heat transfer from the bottom wall.

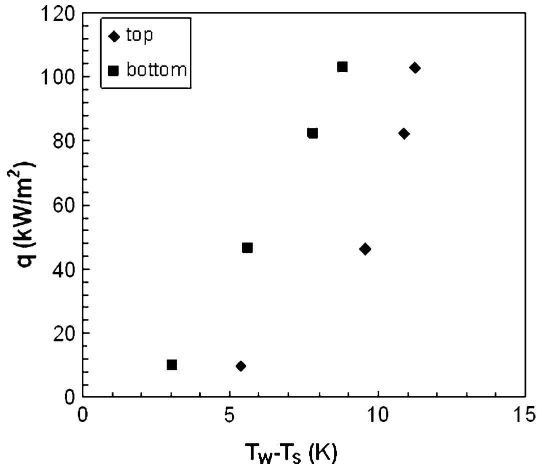


Fig. 3 Top and bottom wall superheats for water boiling on horizontal tube ($D = 33$ mm, $R_a = 0.67$ μm)

Figure 4 shows the effect of inclination on wall superheat for 90°, 60°, 45°, 30° and 0° inclination from horizontal for water boiling on a tube of 21 mm diameter and 0.08 μm surface roughness. It can be seen that the effect of inclination on heat transfer is very small. The wall superheat of the bottom and top walls for this case is given in Fig. 5. Figure 5 showing the plots on the variation of top and bottom wall superheats with the inclination angle for different heat fluxes for water boiling on 21 mm diameter tube and 0.08 μm roughness reveals interesting features of the pool boiling process. General observation from these plots is that top wall superheat increases and bottom wall

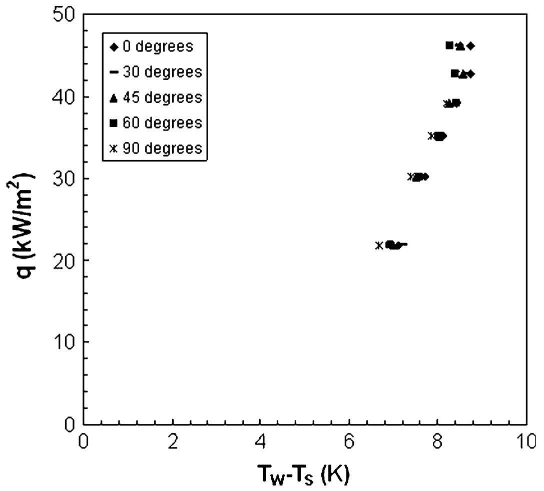
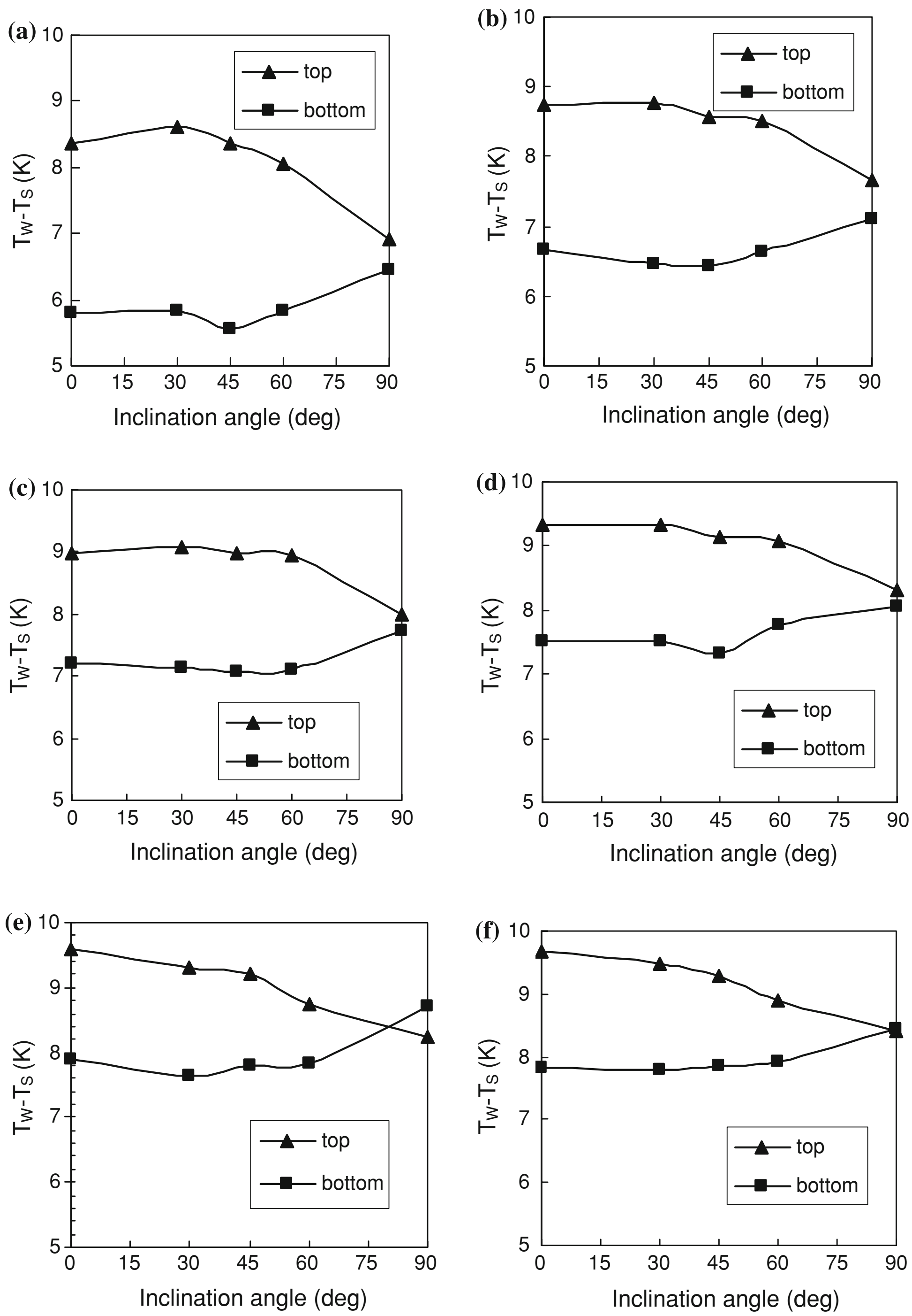


Fig. 4 Nucleate boiling curve for water boiling on tube ($D = 21$ mm, $R_a = 0.08$ μm) at various inclinations

Fig. 5 Variation of *top* and *bottom* wall superheats with the angle of inclination from horizontal, for different heat fluxes (q (kW/m^2) = 21.8 (**a**), 30.2 (**b**), 35.1 (**c**), 39.1 (**d**), 42.7 (**e**) and 46.1 (**f**)) for water boiling on the tube of D = 21 mm and R_a = 0.08 μm



superheat decreases as the inclination is changed from 90° to 0° from the horizontal. The difference between the top wall superheat and the bottom wall superheat will be maximum for 0° inclination angle (horizontal), and minimum (almost zero) for 90° inclination (vertical). Further, it can be observed that most of the change in top wall superheat and bottom wall superheat takes place as the tube is tilted from 90° to 45° from the horizontal, and on further tilting towards 0°, the change in the top and bottom wall superheats is relatively small. It can also be seen that in some of the plots (5a, 5b and 5d) the bottom wall superheat reaches its distinct minimum at 45° inclination angle. This is probably due to the fact that sliding bubbles from bottom of the tube will have larger sliding distances (axial plus circumferential distance components of the tube) when the tube is inclined at 45°, and hence more heat transfer, resulting in lower wall superheats.

However, with increase in surface roughness, the experimental data points for heat transfer appear to be even closer as shown in Fig. 6 for water boiling on 21 mm

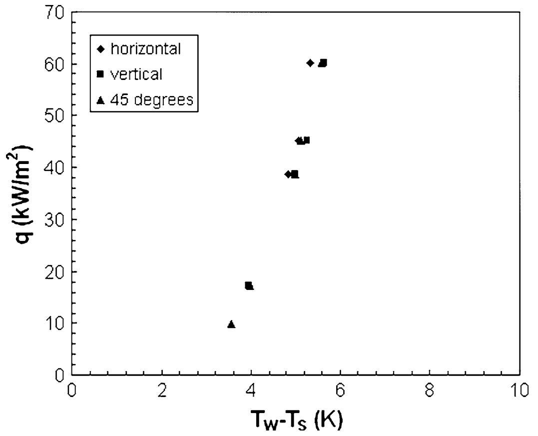


Fig. 6 Nucleate boiling curve for water boiling on tube ($D = 21$ mm, $R_a = 0.67$ μm) at various inclinations

diameter tube and 0.67 μm surface roughness. This is possibly due to the increase in nucleation site density with the increase in surface roughness. Figure 7 shows a similar trend for water boiling on 33 mm diameter tube and 0.29 μm surface roughness. Boiling with Ethanol (Figs. 8, 10, 11) and Acetone (Figs. 12, 14, 15) also gave the similar results for the effect of inclination.

Similarly, Figs. 9 and 13 show the plots on the variation of top and bottom wall superheats with inclination angle for boiling of ethanol and acetone, respectively, on a tube of 21 mm diameter and 0.08 μm roughness. In these cases also, the increasing trend of top wall superheat and the decreasing trend of bottom wall superheat, as the tube is tilted from vertical, are similar to water except that the superheats here are higher (due to the lower surface tension, latent heat and thermal conductivity of the organic fluids as compared to that of water). These results show that it is important to measure or consider both top and bottom wall superheats to estimate the average heat transfer coefficient of the tube when conducting experiments for boiling on inclined tubes.

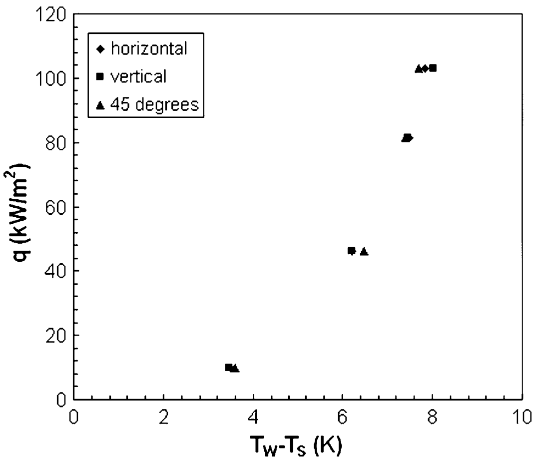


Fig. 7 Nucleate boiling curve for water boiling on tube ($D = 33$ mm, $R_a = 0.29$ μm) at various inclinations

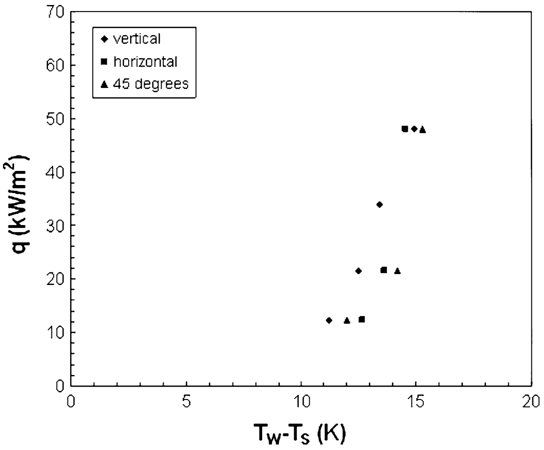


Fig. 8 Nucleate boiling curve for ethanol boiling on tube ($D = 21$ mm, $R_a = 0.08$ μm) at various inclinations

That the effect of tube inclination is relatively high (no matter how small) for smooth tubes (Fig. 4) as compared to rougher surfaces (Fig. 6) in the present work is in agreement with the results of Kang [9] for very smooth surfaces (15 and 60 nm). Kang [9] reported that the tubes with 15 nm roughness showed a greater effect of inclination on heat transfer coefficient than the tube with 60 nm roughness. In the present work, tubes with 90° inclination (from the horizontal) has a slightly higher heater transfer coefficient compared to that with 0° inclination for 0.08 μm roughness (Fig. 4).

However, the present results are different from those reported by Kang [13] for water boiling on 12.7 and 19.1 mm diameter tubes with buffed surface. Surface roughness was not considered in that study and the results show that for 0° tube inclination, 19.1 mm tube has a higher heat transfer coefficient as compared to 12.7 mm tube (and this is contrary to the trends reported by Cornwell et al. [1] for horizontal tubes), and the opposite is the case for 90° of inclination. Also, the variation of heat transfer coefficients with the tube (of diameter 12.7 mm) inclination is rather irregular—increase, decrease, increase, decrease, decrease and then increase for 15°, 30°, 45°, 60°, 75° and 90° of tube inclination, respectively. In his work Kang [13], all the thermocouples are in line-along the length and there were no thermocouples to take care of the circumferential variation of temperature in the inclined position. It is not clear whether only temperatures for one azimuthal angle (along the length) were measured and average was taken as average tube wall temperature, or the tube was rotated to measure wall temperatures at the top and bottom walls (with respect to circumference). How top and bottom wall superheats (along the circumference) vary

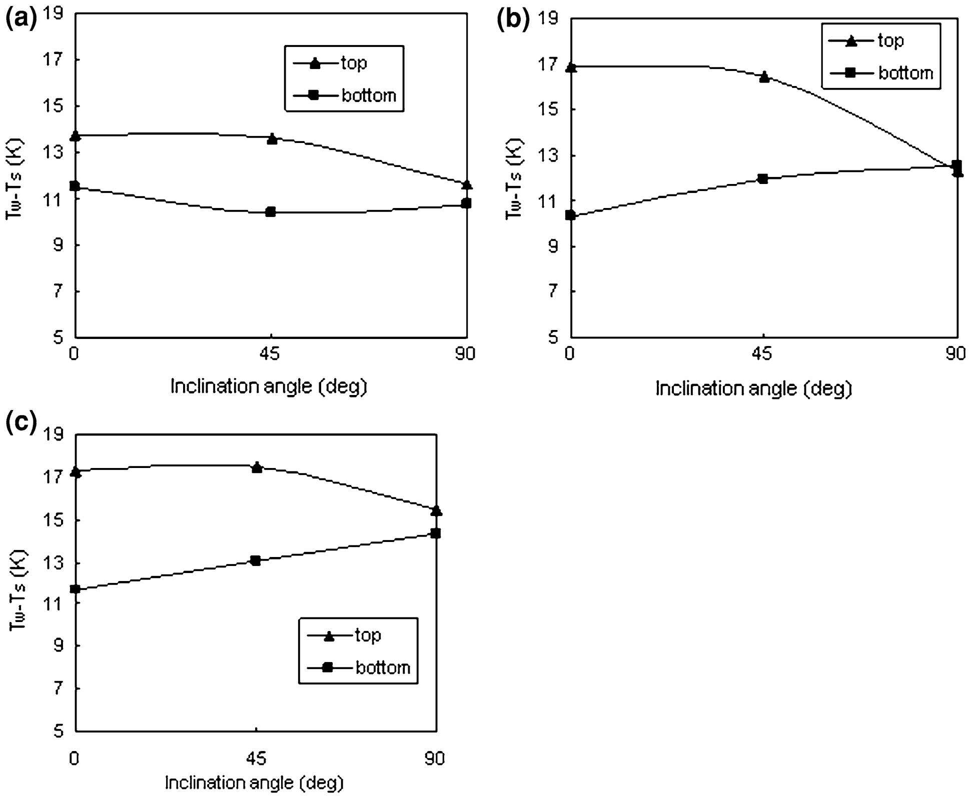


Fig. 9 Variation of wall superheat with the angle of inclination from horizontal, for different heat fluxes (q (kW/m^2) = 12.4 (**a**), 21.6 (**b**), 48 (**c**)) for ethanol boiling on the tube of $D = 21$ mm and $R_a = 0.08$ μm

with different angles of tube inclination was also not shown. So it is difficult to make a comparison of the present results with that of Kang [13] and the present results are more comprehensive because the circumferential variation is measured here.

The results of Nishikawa et al. [6] are for flat downward facing surfaces. Although a direct comparison can not be made between the present results for tube and Nishikawa et al.'s results for flat downward facing surface, the observation that as the tube is tilted from 90° to 0° (from vertical to horizontal) the bottom wall temperature decreases is in agreement with Nishikawa et al.'s results wherein the wall temperature decreases as the downward facing flat surface is tilted from 90°. However,

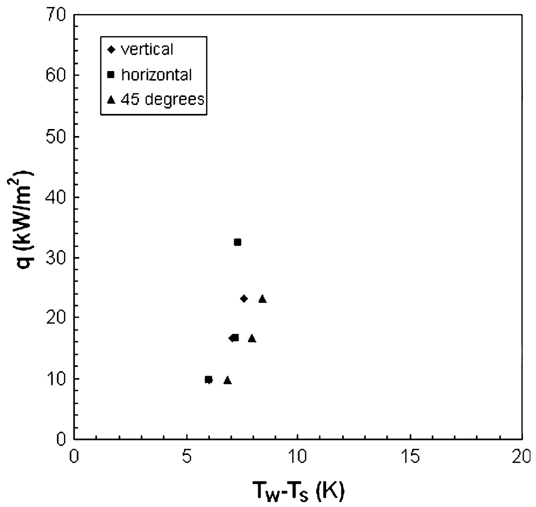


Fig. 10 Nucleate boiling curve for ethanol boiling on tube ($D = 33$ mm, $R_a = 0.29$ μm) at various inclinations

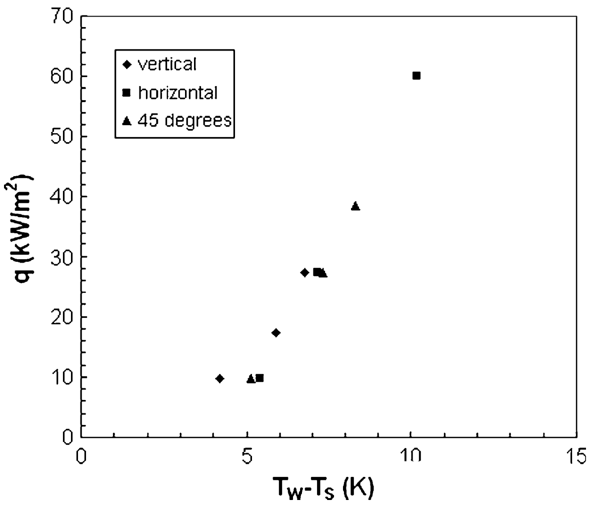


Fig. 11 Nucleate boiling curve for ethanol boiling on tube ($D = 21$ mm, $R_a = 0.67$ μm) at various inclinations

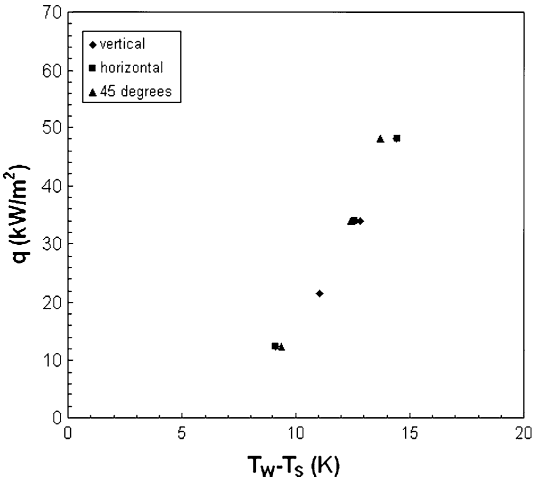


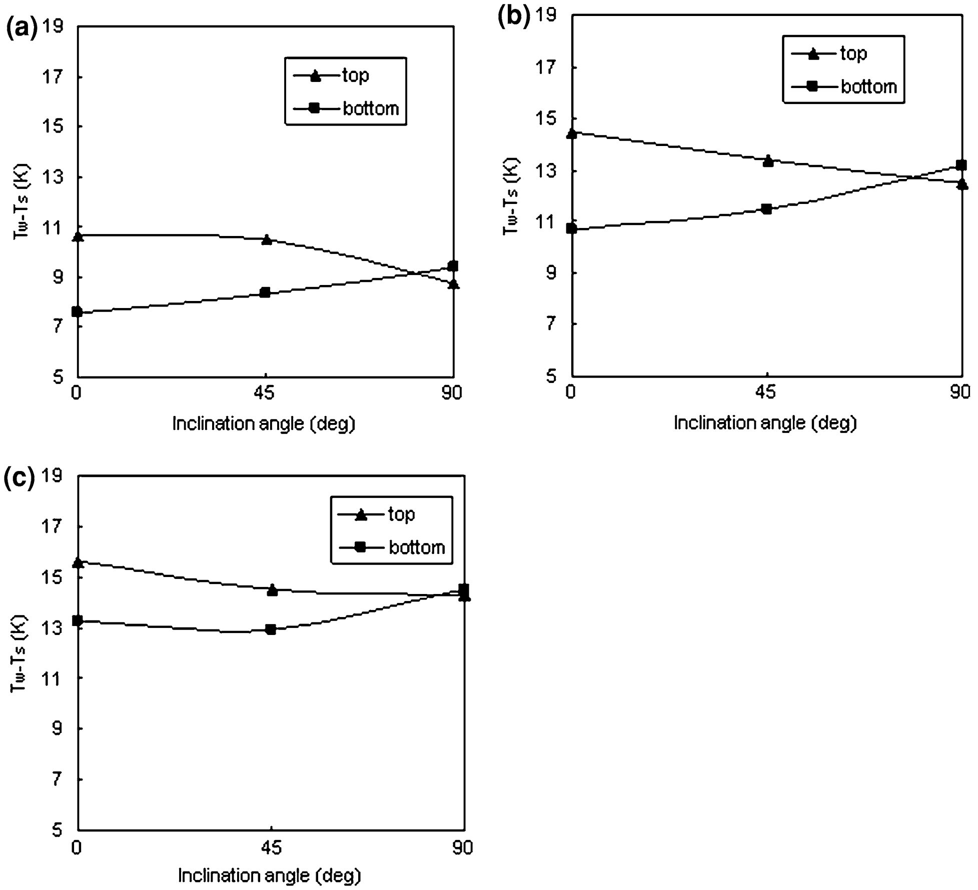
Fig. 12 Nucleate boiling curve for acetone boiling on tube ($D = 21$ mm, $R_a = 0.08$ μm) at various inclinations

as the tube has partly bottom wall (downward facing) and partly top wall (upward facing), the decrease in wall superheat at the bottom wall is nearly offset by the increase in wall superheat at the top wall (with respect to circumference).

4 Conclusions

In the present work, variation of top and bottom wall superheats (with respect to the circumference) with the tube inclination, which has so far not been reported in the literature, has been presented for different surface roughness values, tube diameters and fluids (water, ethanol and acetone) in nucleate pool boiling. As the tube is tilted from vertical to horizontal, the wall temperature at the top wall increases (as bubbles tend to depart directly without sliding) and that at the bottom wall temperature decreases (due to the increase in sliding length) resulting in higher heat transfer coefficients at the bottom wall and lower heat transfer coefficients at the top wall. This self-compensating effect of increase and decrease in wall temperatures at the top and bottom, respectively, results in very little effect of tube inclination on the average boiling heat transfer coefficient for the range of the parameters investigated. There seems to be a small effect of inclination for smooth surfaces and almost negligible for rougher surfaces. Further, it can be concluded that for accurately measuring the average heat transfer coefficient in pool boiling on inclined tubes, it is important to

Fig. 13 Variation of wall superheat with the angle of inclination from horizontal, for different heat fluxes (q (kW/m^2) = 12.4 (**a**), 34 (**b**), 48 (**c**)) for acetone boiling on the tube of $D = 21$ mm and $R_a = 0.08$ micron



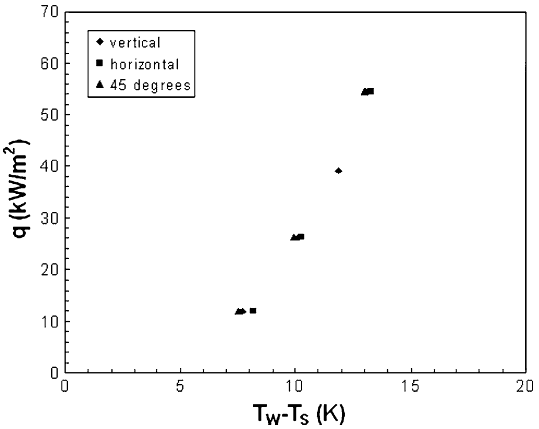


Fig. 14 Nucleate boiling curve for acetone boiling on tube ($D = 21$ mm, $R_a = 0.67$ µm) at various inclinations

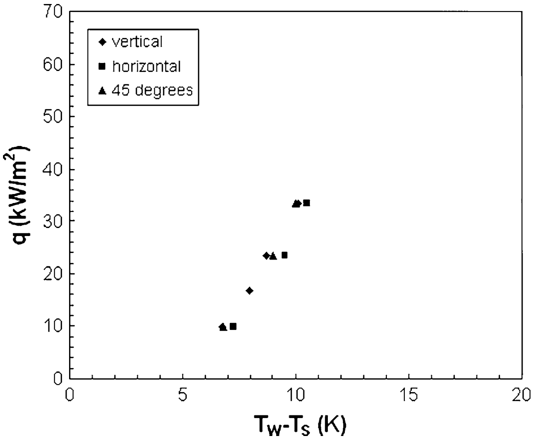


Fig. 15 Nucleate boiling curve for acetone boiling on tube ($D = 33$ mm, $R_a = 0.08$ µm) at various inclinations

measure wall temperatures at both top wall and bottom wall (with respect to circumference) as they differ significantly for inclinations other than 90° (vertical).

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