International Information and Engineering Technology Association
Advancing the World of Information and Engineering

International Journal of Heat and Technology
Vol. 39, No. 2, April, 2021, pp. 329-336
Journal homepage: http://ieta.org/journals/ijht

Enhancement of Pool Boiling Heat Transfer over Plain and Rough Cylindrical Tubes

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https://doi.org/10.18280/ijht.390201

Received: 21 May 2020
Accepted: 15 March 2021

Keywords:
heating tube, heat transfer, orientation, pool boiling, surface roughness, wall superheat

ABSTRACT

This study investigates the pool boiling heat transfer of water over cylindrical heating tubes for different orientations and surface roughness of the tubes. First, two orientations of a smooth heating tube, horizontal and vertical, were used in the boiling chamber. For a given heat flux, the heat transfer coefficient achieved with the horizontal tube was always higher than that for the vertical tube. To investigate the influence of surface roughness, a rough heating tube with a fully rough outer surface was developed through a metal etching process. Under the same range of wall superheat, the rough tube enhanced the heat transfer rate significantly compared to the smooth tube. Finally, a modified heating tube (MHT) was developed by axially roughening half of the surface of an originally smooth tube. The orientation angle of the rough surface of this MHT was varied from 0° (horizontal-upward) to 180° (horizontal-downward) in the chamber. The heat flux increased significantly with the increase of orientation angles from 0° to 90° (the maximum of 80 kW/m^2 at 90°), whereas the same decreased as the orientation angle is further increased from 90° to 180°. Results revealed that the bubble dynamics over the heating tubes play a vital role in pool boiling performance.

1. INTRODUCTION

Augmentation of pool boiling heat transfer rate was the key attention of many researchers over the past years, and intensive studies continue focusing on the underlying physics and different aspects of this phenomenon. The large span of applications of pool boiling such as in thermal power plants, heat exchangers, and many other thermodynamic devices is why researchers continue studying and investigating methods and techniques to escalate the rate of heat transfer and reduce possible losses.

The orientation of the heater in the pool boiling chamber can influence the pool boiling heat transfer performance. For a given wall superheat, several researchers found enhanced heat transfer rate, characterized by increased critical heat flux (CHF) and heat transfer coefficient (HTC), when the heater orientation was changed from horizontal to vertical [1-7]. Marcus and Dropkin [1] suggested that this change in heater orientation causes an increased path length for the departing bubbles along the heater surface, and thus increases the superheated boundary region. Kang [3] changed the angle of inclination of the heating element in progressions of 15° from the horizontal direction to the vertical direction, which has shown to improve the heating at 15° from the horizontal, due to a reduction in the formation of slower, heavier bubbles at the bottom surface. In another study [5], it was observed that the addition of circumferential rectangular microgrooves on cylindrical tubes significantly augments the heat transfer performance, and horizontal orientation of the test sections resulted in 10-15% augmented heat transfer compared to that in the vertical orientation. During the experiments, the active bubbles and the rewetting process were also found to play a key role in the overall heat transfer. However, recently, Sarode et al. [6] showed an invariance in the boiling heat transfer performance for horizontal and vertical heater orientations.

A number of different techniques are employed to modify the heater surface that can enhance the critical heat flux (CHF) and the heat transfer coefficient (HTC) during a pool boiling phenomenon [8-22]. Techniques such as the use of nanofluids [8], extended surfaces called fins [9], incorporating surface roughness [10-22], etc., have been found to improve the CHF. Among these techniques, incorporating microscale surface roughness is a widely used method that can augment a boiling HTC by up to 600% [10]. Roughness can be introduced on the heater surface by chemical etching, mechanical roughening, sandblasting, etc., and the technique applied for the surface roughening plays an important role in the boiling performance. Ferjančič and Golobič [12] investigated pool boiling of water over stainless steel heater surfaces, which were roughened in two ways, chemical etching and sandblasting. For the same average roughness amplitude on the heater surfaces, the chemically etched heaters led to 51% higher CHF compared to that of the sanded heaters. Recently, Kim et al. [13] experimentally revealed a strong dependence of the CHF on the heater surface roughness as a consequence of the capillary wicking around the rough surface. This surface wickability effect on the CHF is further corroborated by Sujith Kumar et al. [16]. A heater with microscale surface roughness combines a fin action and wettabiliy effect (hydrophobic or hydrophilic) at the heater-liquid interface during the boiling. The fin action provides greater surface area for the bubbles to form and develop, thus increasing the heat flux. However, the effect of the surface wettability on the boiling is non-monotonic for both hydrophobic and hydrophilic cases [8, 16-21]. A hydrophobic surface leads to a higher HTC while resulting in a lower CHF at high heat flux conditions [8]. Conversely, a hydrophilic surface leads to a lower CHF while resulting in a lower HTC at low heat flux conditions [16]. Therefore, researchers applied partial roughening of the heater surface to produce a hybrid wettability effect, i.e., a combination of hydrophobicity and hydrophilicity on the heater surface [18-21]. Jo et al. [20] constructed a heater with hybrid wettability by producing hydrophobic dots on an originally hydrophilic surface, and from experiments, they achieved a higher HTC and CHF for the hybrid surface than that for the hydrophilic one. Kumar et al. [21] prepared a hybrid heating tube having rough interlines on the heater surface by a chemical process. The results showed that the HTC enhances with the increasing number of interlines on the heater surface and HTC is also influenced by the arrangement of the hybrid pattern of the heater surface.

For partially rough heating elements, the orientation of the rough surface of the heating element can influence the pool boiling heat transfer rate [4, 22, 23]. El-Genk and Guo [22, 23] changed the orientation of a copper disk quenched in water in ascending angles from 90° to 180°, where 90° indicated the vertical placement of the disk, and 180° meant that the disk was facing the downward position. The CHF decreased with the increase in inclination angle, and the CHF was significantly lower for the downward orientation (180°) compared to the other orientations of the copper disk. Besides, as the angle shifts from 90° to 180°, the CHF decreases as the vapor bubbles stick to the heater surface for a longer period, blocking the cold fluid from coming in contact with the heater surface and causing dry out. However, for the increase of inclination angle in the upward directions (from 0° to 90°), conflicting results were obtained in the previous studies regarding the impact of the inclination angle on the CHF [24-26]. In some cases, the CHF deteriorated [24, 25], but CHF remained constant in other observations [26]. Therefore, these rough surface effects on the pool boiling heat transfer performance require more efforts.

In light of the literature review, the present research work is conducted to further investigate the enhancement of boiling of water over a cylindrical heating tube at atmospheric pressure under different experimental conditions. The vertical and horizontal orientations of a plain heating tube are used to investigate the effect of heater orientation on the boiling phenomenon. The heating tube is further roughened by incorporating etching on the heating tube surface, and the resulted rough heating tube is used in pool boiling to investigate the effect of surface roughness on the heat transfer. Finally, a half-etched heating tube is employed where the orientation of the rough surface is varied from 0° to 180° to study the effect of the orientation of tube roughness on pool boiling performance.

2. METHODOLOGY OF THE STUDY

2.1 Experimental setup

An experimental setup that consists of a pool boiling chamber and a heating tube was designed and fabricated to investigate pool boiling of water over a cylindrical copper tube at atmospheric pressure. The scope of changing the orientation (vertical or horizontal) of the heating tube was also incorporated while designing the setup. Shown in Figure 1 is the design of the experimental setup, with a heating tube placed horizontally, developed by using the SolidWorks 2019. A rectangular box, which served as the pool boiling chamber, was placed on a supporting table. The boiling chamber had two glass windows to observe the boiling phenomena and both sides of the glass windows were bolstered with silicone gaskets to eliminate any risk of leakage during the experiments due to high temperatures. The boiling chamber housed a heating tube which was used to heat the water directly. Hollow cylinders made of copper 101 alloy, with thermal conductivity of 391 W/m.K at 20°C, were chosen to construct heating tubes. The placement of a cartridge heater inside a copper tube led to a complete heating tube. The rating of the cartridge heater was 1250 W and its length was 122 m. A small hole was drilled on the top cover of the glass chamber to place an auxiliary heater on the upper section to heat the water to its saturation temperature. The heating tube or copper cylinder was fixed with Nylon shafts at both ends and the Nylon shafts were firmly attached to the pool-boiling chamber by using silicon glue and M-seal (see Figure 1). The Nylon shafts applied resistance to the heat loss from the copper cylinder through its axial direction in the pool boiling chamber. The placement of thermocouples inside the copper cylinder allowed the measurement of temperature as the cylinder was heated by a cartridge heater placed inside it, via a DC power supply. A variac was used to vary the supply voltage (can vary up to 250 volts) to the heater assembly. The thermocouple probe measured the temperature of water at upper and lower sections of the chamber, and the temperature readings of the thermocouples were recorded from the thermocouple displays. The vapor of water that moved up in the chamber by convection heat transfer was condensed in a reflux condenser and the condensed liquid could return to the boiling chamber due to gravity. As a result, the liquid level in the chamber was maintained during the boiling experiments.

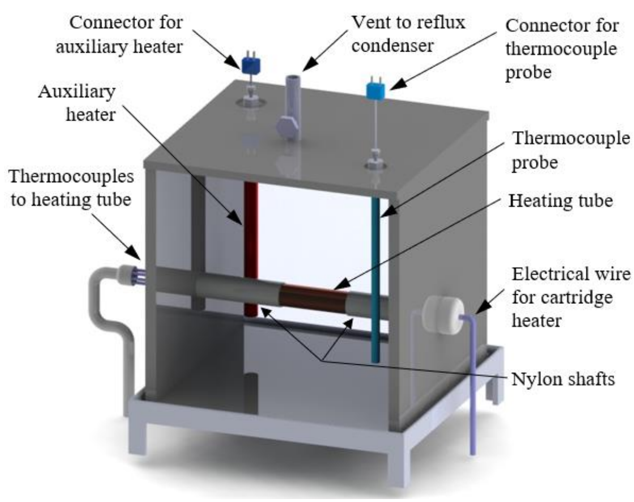


Figure 1. CAD model of the experimental setup

2.2 Construction of the heating tube

The cartridge heater, which acted as the main source of heat for the boiling of water, was inserted into a hollow copper tube to construct the heating tube (HT). The outer diameter of the copper cylinder was 34 mm and the inner diameter was 14.2 mm. The length of the cartridge heater was considered as the actual effective heat transfer length and the copper tube was made of the same length. After inserting the heater in the tube, insulating material (heat resisting glue) was used to fill the open ends of the tube. Also, the attachment of hollow nylon bars at both ends of the heating tube offered thermal insulation at those ends. Thus, the axial heat loss to the surroundings from the heating tube was restricted and thus facilitated the heat transfer in the radial direction of the heating tube surrounded by the water. Eight calibrated K-type thermocouples, which passed through the hollow nylon tube at the left side of the chamber (Figure 1), were placed inside eight equally spaced 1 mm diameter holes circumferentially at one side of the copper cylinder. Four of the thermocouples were placed at a depth of 10 mm, and the rest at a depth of 15 mm, alternately, as shown in Figure 2. This difference of depth (Δx) was introduced to determine the longitudinal heat loss from the heating tube. The internal temperature of the heating tube was measured by the inserted thermocouples whose accuracy was found to be $\pm 1\%$.

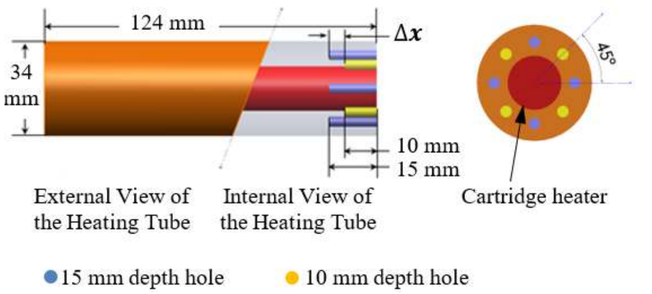


Figure 2. Positioning of thermocouples at alternate locations inside the heating tube

2.3 Preparation of the test sections

The types of the test sections were varied depending on the outer surface-roughness of the copper cylinder, whereas all other properties were kept fixed in the present study. Three different types of surfaces were investigated: plain, homogeneously rough or simply rough, and partially rough. The roughness in the copper tube was induced by a metal-etching process. The partially rough surface was prepared by applying metal-etching axially on half of the surface area and the rest remains plain surface. Hereinafter, this partially rough surface is denoted as a half-etched or modified surface. As the heating tube differs depending on its outer surface, the developed three distinct test sections are: (i) Plain heating tube (PHT), (ii) Rough heating tube (RHT), and (iii) Modified heating tube (MHT). The wetting property of the tube surface is a function of surface roughness and introducing microscale roughness on the surface reduces the spreading of the liquid on the surface and thus increases the contact angle. The contact angle of the plain copper with water is ~86° [27], so the surface of a plain copper tube is intrinsically hydrophilic. To induce roughness on the copper tube, first, the test piece of copper was manually polished for 15 min using an emery paper having a grit grade of 2000. The polished cylinder was rinsed with acetone, and de-ionized water was used to clean the polished test piece. The test piece was then placed in an oven for two hours at a steady temperature of 120℃. The area that was not to be etched was masked with a ground which is resistant to the prepared etchant solution. Subsequently, the cylinder was then immersed in a bath of Ferric chloride solution, dipping at a high velocity while rising at a relatively slower velocity of 6 mm/min. Consequently, the solution dissolves part of the metal where it was exposed, leaving behind a rough etched area into the surface. Finally, the etched tube was kept in an oven at 120°C for one hour to get the test piece for experiments.

The surface roughness of the copper cylinders was measured by a Bruker Dektak XT profilometer in the laboratory, as shown in Figure 3. The scan duration for the surfaces was about 30 seconds, and the scan length was about 50 µm. For the scans, the resolution was 0.555432 µm, and the stylus force and the scan range were 5 mgf and 524 µm, respectively. From the surface roughness profile, the mean roughness of the rough surfaces was found to be ~18 µm. Therefore, the applied etching technology was found to alter surface topography by generating micro-and nano-scale pores of varying geometries. These pores usually act as sites for bubble formation, which can contribute to the overall heat transfer effect.

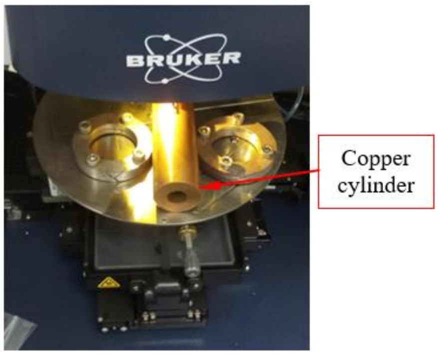


Figure 3. Surface roughness measurement of the copper cylinders using Bruker Dektak XT profilometer

2.4 Modified heating tube (MHT) with different orientations of the rough surface

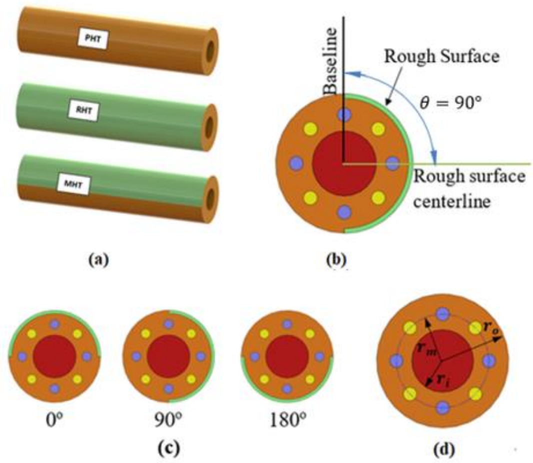


Figure 4. Preparation of test sections: (a) copper cylinders with different surface roughness criteria (PHT, RHT and MHT), (b) orientation selection criteria of the rough surface of MHT, (c) MHT with different rough surface orientations, and (d) schematic of different radii of the copper cylinder

The effect of the orientation of the rough surface of the MHT was also studied by placing it at various orientations along the axial direction (0°, 90°, and 180°) in the water pool as shown in Figure 4. Orientation was characterized by the angle between a vertical line selected as a baseline and the centerline of the etched surface and it was measured in a clockwise direction (see Figure 4(b)). Therefore, for 0° orientation, the etched surface was kept horizontally upward, and for 180° orientation, the etched surface was horizontally downward (Figure 4(c)).

3. DATA ACQUISITION AND REDUCTION

In the present study, the pool boiling heat transfer performance was assessed by determining the heat flux and heat transfer coefficient (HTC). The heat supply (Q) to the test section was varied by varying the supply voltage (V) across the cartridge heater. The corresponding current supply (I) to the heater was then measured at the heater junction and the total heat input to the test section was determined by Eq. (1).

$$Q = I \times V \tag{1}$$

Although Nylon bars were used to restrict the axial heat loss (Q_{Loss}) at the ends of the test section, an inevitable Q_{Loss} was found at both ends. This axial heat loss (Q_{Loss}) from the two ends of the heater was estimated by Eq. (2).

$$Q_{Loss} = 2 \times k_C \times [(T_{15} - T_{10})/\Delta x] \times A_c \tag{2}$$

where, k_C represents the thermal conductivity of the copper alloy, Δx (= 5 mm) is the difference of the axial positions of the thermocouples (Figure 2); and $A_c = \pi(r_o^2 - r_i^2)$ is the area of the cross-section of the copper cylinder. The average value of the temperatures measured from the thermocouples at a depth of 15 mm and the same from the thermocouples at a depth of 10 mm are represented by T_{15} and T_{10}, respectively. A steady-state condition is assumed when the variation of a temperature reading remains within ±1°C for 15 min.

The heat flux (q") to the water was determined as:

$$q" = (Q - Q_{Loss})/A \tag{3}$$

where, $A = 2\pi r_o L$ is the circumferential area and L is the length of the test section.

The average value of the temperatures of all the inserted thermocouples, *i.e.*, T_m at radial position r_m (Figure 4(d)), was used to determine the outer surface temperature of the heating tube, *i.e.*, T_o at radial position r_o (Figure 4(d)), by applying the steady-state one-dimensional heat conduction equation for radial heat transfer as:

$$T_o = T_m - [(Q - Q_{Loss})/2\pi L k_C]\, ln(r_o/r_m) \tag{4}$$

Wall superheat (ΔT_W) was determined from the difference in the outer surface temperature (T_o) and saturation temperature (T_{Sat}) of the water at the working pressure. The thermocouple probe in the water chamber was used to measure T_{Sat}. Finally, the HTC (h) was calculated by using Eq. (5).

$$h = q"/\Delta T_W \tag{5}$$

The obtained reduced data was then used to plot the curves to compare the performance of the different heating tubes under various experimental conditions.

4. RESULTS AND DISCUSSION

4.1 Effect of heating tube orientation

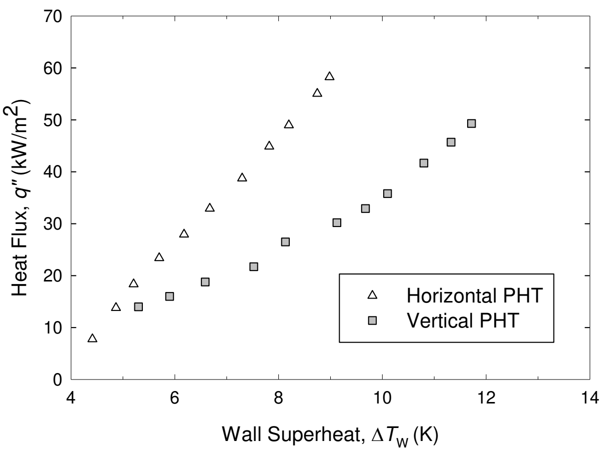


Figure 5. Nucleate boiling curves for the horizontal and vertical plain heating tube (PHT)

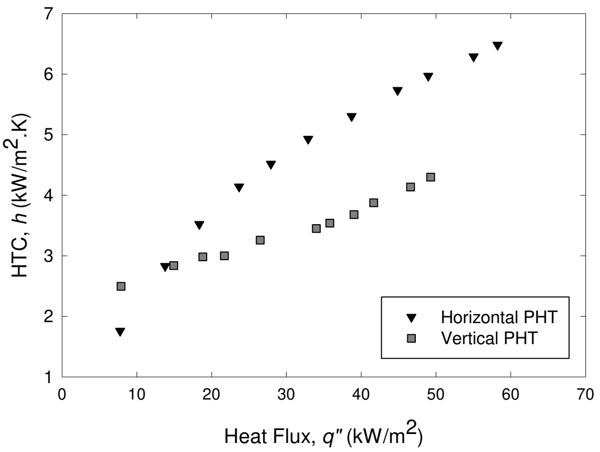


Figure 6. Comparison of the HTC for the horizontal and vertical plain heating tube (PHT)

To study the effect of tube orientation on the pool boiling heat transfer, boiling experiments were carried out with the plain heating tube (PHT) for its horizontal and vertical orientation inside the chamber. Shown in Figure 5 are the nucleate boiling curves obtained by plotting the data of the heat flux against that of the wall superheat. The equations of the previous section were used to determine the wall superheat, heat flux, and heat transfer coefficient (HTC) for different experimental conditions. The uncertainty of determining heat flux and HTC was ~4.5% with a 95% confidence level. For the purpose of visual clarity, error bars have not been included in the figures. The results show that the rate of change in heat flux is always steeper for the horizontal tube compared to that for the vertical tube. The heating tube reaches a maximum heat flux of 60 kW/m² for wall superheat of about 9 K when it is placed horizontally. In contrast, for the vertical arrangement, the maximum heat flux is 49.28 kW/m² at a wall superheat of 11.7 K. Figure 6 shows the variation of HTC against heat flux for both orientations. The monotonic rise in the values of HTC with the increasing heat flux possibly occurs due to the higher surface temperature of the test section and the rapid bubble formation at the water-copper interface. However, the horizontal tube gives higher HTC compared to the vertical one. As the primary nucleation of the bubbles occurs on the bottom surface of the tube, the horizontal tube activates more nucleation sites at its bottom and thus performed better in the case of pool boiling. Similar results have been reported in previous studies of pool boiling over cylindrical tubes [5, 7].

Table 1 summarizes the experimental results for the horizontal and vertical orientation of the plain heating tube (PHT) at their maximum tested heat flux conditions.

Table 1. The maximum performance achieved with the PHT in its horizontal and vertical orientations

Orientation	Performance parameters		
	q''_{max} (kW/m^2)	ΔT_W (K)	h (kW/m^{2}K)
Vertical	49.3	11.7	4.2
Horizontal	60.0	9.0	6.7

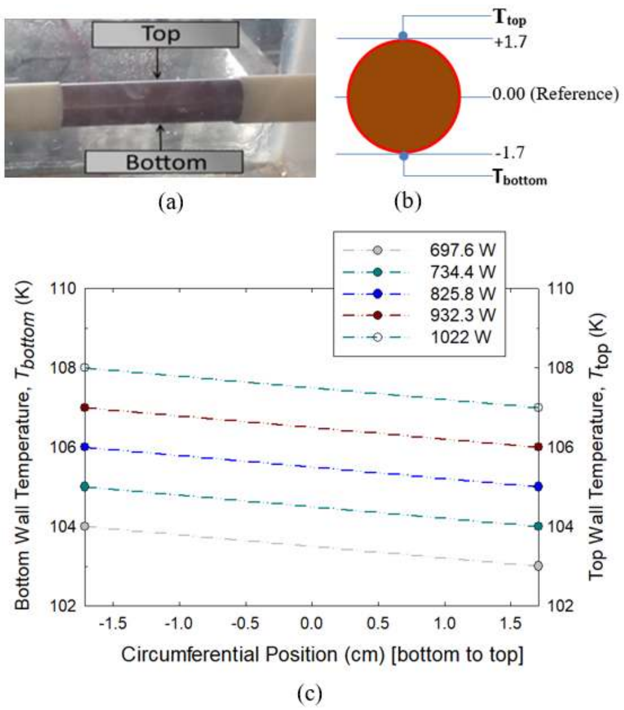


Figure 7. (a) Top and bottom surface of the PHT, (b) Schematic of the circumferential positions, and (c) Wall temperature variation along the outer circumference of the horizontal PHT for different heat supplies

From experiments, it was observed that the bubble dynamics around the horizontal PHT contributes to its better heat transfer performance (as depicted in Table 1) compared to that of the vertical PHT. Shown in Figure 7 is the variation of the surface temperature along the outer circumference of the horizontal PHT for different heat supplies to the heater. The trend of this variation depends on the formation, growth, and emission dynamics of the bubbles during the boiling process. The commencement of vapor bubbles occurs at the randomly distributed preferred sites on the surface of the test section. As the bubbles reach their maximum size, they start leaving the tube surface. However, some of the bubbles cannot move freely like the rest as they face obstruction because of the shape of the tube. The emission frequency of the formed bubbles was found to be the highest at the top-most heater surface, whereas the bubble emission was the least at the bottom-most surface of the heating tube. The emission frequency at the side positions was found to be in between those of the top and the bottom surfaces. Thus, a continuous increase in wall temperature was observed from top to side to bottom surfaces of the test section. As a result, the bottom surface of the horizontal tube activates more nucleation sites and performs better in the heat transfer than the vertical one, as seen in Figure 5. This observation corroborates the experimental results of Islam et al. [28] who found that, during pool boiling over a cylindrical tube, the wall superheat establishes on the bottom surface earlier than the top surface of the tube. The temperatures at the top and bottom positions of the horizontal PHT for different heat supplies (Q) to the heater, i.e., the values of T_{top} and T_{bottom} (Figure 7b), are summarized in Table 2.

Table 2. Temperatures at the top and bottom circumferential positions of the horizontal PHT for different conditions

Heat supply (W)	Temperatures at the bottom and top positions	
	T_{bottom} (K)	T_{top} (K)
697.6	104	103
734.4	105	104
825.8	106	105
932.3	107	106
1022.0	108	107

4.2 Effect of surface roughness

The effect of surface roughness was investigated as the heating tube was etched fully and its heat transfer performance is compared with that of the plain heating tube in Figure 8. From Figure 8, it is evident that the rough heating tube gives higher values of heat flux; however, only a heat flux of ~55 kW/m^2 could be achieved with the setup. Consequently, the HTC was also much higher when a hydrophobic surface was introduced on the heating tube, as the roughness acts as active nucleation sites and contributes to augmented heat transfer rate when compared to the plain heating tube (see Figure 9). The RHT was successfully tested up to 55 kW/m^2, where the wall superheat was 7.7 K, yielding an HTC of 7 kW/m^{2}K. On the other hand, the PHT was successfully tested up to 50 kW/m^2, where the wall superheat was 8.3 K, yielding an HTC of 6.2 kW/m^{2}K. Comparing the experimental data shown in Figure 9, it was evident that the heat transfer performance achieved with the RHT is approximately 9-34 % higher than that achieved with the PHT. The number of active nucleation sites and the bubble formation rate at each site significantly influence the pool boiling heat transfer. The roughness features on the heating tube surface serve as additional nucleation sites during the boiling process as the bubble formation usually initiates at the roughness features of the heating surface. The bubbles can form and grow at these places and thus the bubble formation rate is enhanced by the rough surface. Therefore, while used in the heating tube, the fully etched copper cylinder performed better in transferring heat to the surrounding water compared to the plain copper cylinder. Kumar et al. [21] developed heating tubes with a hybrid wettable pattern by introducing rough strips on a plain copper cylinder and found enhanced heat transfer rate from the rough surface due to the increased number of nucleation sites. Through experiments, Cooke and Kandlikar [29, 30] also found that the bubble nucleation and its growth strongly contribute to the heat transfer augmentation as observed in the present study.

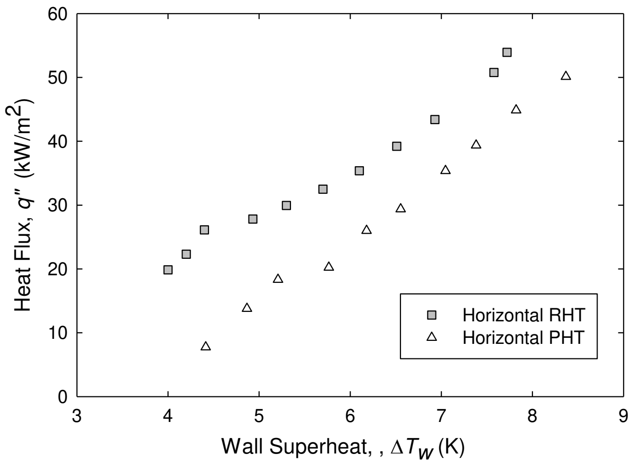


Figure 8. Nucleate boiling curves for the horizontal PHT and RHT

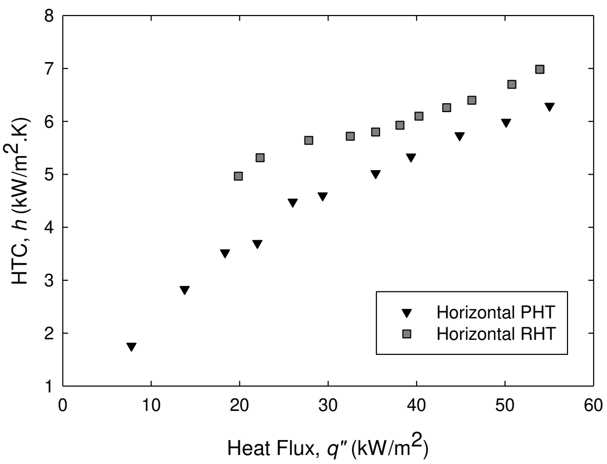
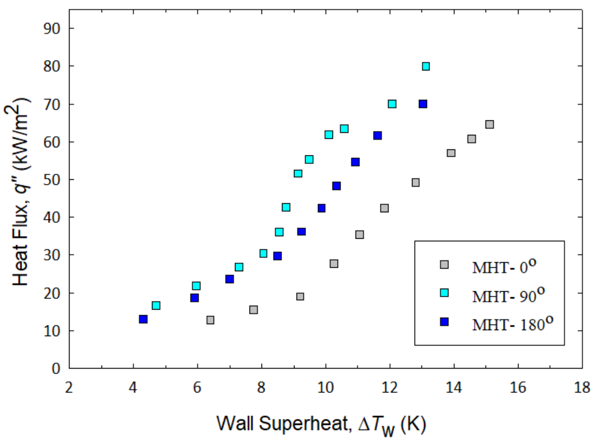


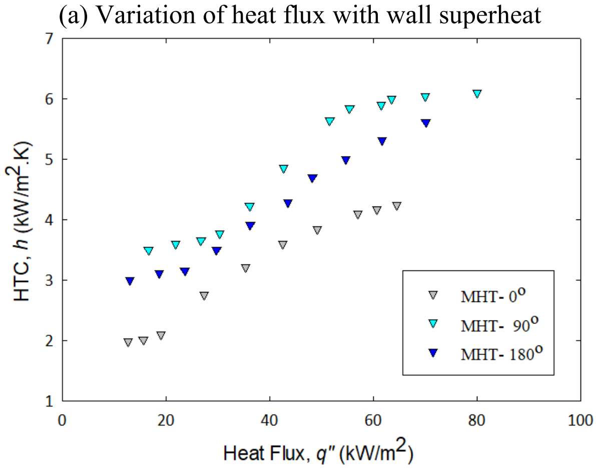
Figure 9. Variation of heat transfer coefficient (HTC) for the horizontal PHT and RHT

4.3 Effect of rough surface orientation for a partially rough heating tube

As it was observed that both orientation and surface roughness can contribute to heat transfer performance, the experiments were extended by changing the orientation of the etched surface area of the half-etched modified heating tube (MHT). The MHT was placed horizontally with its rough surface at 0° (horizontal upward), 90° and 180° (horizontal downward) and the corresponding change in heat transfer performance was explored. Figure 10(a) shows the variation of heat flux with wall superheat for different orientation of surface roughness on the heating tube, and the variation of HTC with heat flux is shown in Figure 10(b). The curves show a similar trend in all cases, i.e., with the increase of wall superheat, there is an increase in heat flux, and HTC is seen to be increased with the increasing heat flux. The heat fluxes are much higher for the MHT with 90° rough surface orientation. With the increase of angle from 0° to 90°, an increase in the heat flux with wall superheat is observed, although as the angle of orientation increases from 90° to 180°, there is a decrease in the heat transfer. The increased hindrance for the 180° oriented rough surface led to coalescence of the neighboring bubbles to form a larger bubble, which resulted in an increased vapor-surface contact time and consequent reduction in the heat flux in the present study. Kwark [4] also investigated the effect of the rough surface orientation on the pool boiling heat transfer by using a nanocoated heating surface and found a similar decrease in the critical heat flux for change of the orientation from 90° to 180°. Table 3 summarizes the experimental results for the three rough surface orientations of the modified heating tube (MHT) at their maximum tested heat flux conditions.



(a) Variation of heat flux with wall superheat



(b) Variation of HTC with heat flux

Figure 10. Effect of rough surface orientation on the pool boiling heat transfer over the MHT

Table 3. The maximum performance achieved with the MHT for its different rough surface orientations

Orientation of the rough surface	Performance parameters		
	q''_{max} (kW/m²)	ΔT_W (K)	h (kW/m²K)
0°	64.5	15.1	4.3
90°	80.0	13.1	6.1
180°	70.0	13.0	5.4

5. CONCLUSIONS

Pool boiling of liquid water over cylindrical copper tubes was experimentally investigated for different orientations and surface roughness of the tubes. Initially, the heat transfer over a smooth heating tube was investigated for the horizontal and vertical orientations of the tube inside the pool boiling chamber. Then, surface roughness was introduced on the outer surface of the smooth heating tube by a metal etching process to investigate the influence of surface roughness and the orientation of the rough surface on the heat transfer. For all the experiments, the heat flux and the heat transfer coefficient were found to increase with the increase in wall superheat. The following conclusions can be drawn based on the analysis of the experimental results:

- The horizontal orientation of the plain heating tube (PHT) facilitated enhanced heat transfer compared to its vertical orientation due to the enhanced bubble nucleation at the bottom surface of the horizontal PHT. For the horizontal PHT in the present setup, a maximum heat transfer coefficient (HTC) of 6.7 kW/m^2K was achieved at a heat flux of 60 kW/m^2. On the other hand, the vertical PHT in the setup was able to achieve a maximum HTC of 4.2 kW/m^2K at a heat flux of 49.3 kW/m^2.
- The surface roughness of the heating tube played a key role in enhancing the heat transfer rate due to the increased nucleation sites on the rough surface. The fully rough heating tube (RHT) showed approximately 9-34% better performance compared to that of the plain heating tube (PHT).
- The orientation of the rough surface area of the partially rough (half-etched) modified heating tube (MHT) significantly influenced the heat transfer performance. An increase in the heat flux was observed as the orientation angle increases from 0° to 90°. However, the heat flux was found to decrease as the orientation angle was increased from 90° to 180°. This decrease in heat flux can be attributed to the merging of individual bubbles and the formation of larger bubbles as the rough surface blocks the upward movement of the bubbles at its 180° orientation. The maximum heat flux of 80 kW/m^2 was achieved for the 90° orientation of the half-etched heating tube.

All these experimental findings provide a reliable insight into the factors that can augment the pool boiling heat transfer in numerous engineering applications.

ACKNOWLEDGMENT

The authors would like to acknowledge the Ahsanullah University of Science & Technology, Dhaka, Bangladesh, for providing the necessary laboratory facilities to conduct the experiments.

REFERENCES

[1] Marcus, B., Dropkin, D. (1963). The effect of surface configuration on nucleate boiling heat transfer. International Journal of Heat and Mass Transfer, 6(9): 863-866. https://doi.org/10.1016/0017-9310(63)90069-3

[2] Chang, J.Y., You, S.M. (1996). Heater orientation effects on pool boiling of micro-porous-enhanced surfaces in saturated FC-72. Journal of Heat Transfer, 118(4): 937-943. https://doi.org/10.1115/1.2822592

[3] Kang, M.G. (2003). Effects of tube inclination on pool boiling heat transfer. Nuclear Engineering Design, 220: 67-81. https://doi.org/10.1016/S0029-5493(02)00322-9

[4] Kwark, S.M., Amaya, M., Kumar, R., Moreno, G., You, S.M. (2010). Effects of pressure, orientation, and heater size on pool boiling of water with nanocoated heaters. International Journal of Heat and Mass Transfer, 53(23-24): 5199-5208. https://doi.org/10.1016/j.ijheatmasstransfer.2010.07.040

[5] Mehta, J.S., Kandlikar, S.G. (2013). Pool boiling heat transfer enhancement over cylindrical tubes with water at atmospheric pressure, Part I: Experimental results for circumferential rectangular open microchannels. International Journal of Heat and Mass Transfer, 64: 1205-1215. https://doi.org/10.1016/j.ijheatmasstransfer.2013.03.087

[6] Sarode, A., Raj, R., Bhargav, A. (2020). Effect of confinement and heater surface inclination on pool boiling performance of patterned wettability surfaces. Journal of Enhanced Heat Transfer, 27(8): 711-727. https://doi.org/10.1615/jenhheattransf.2020033852

[7] Narayan, G.P., Kanjirakat, A., Sateesh, G., Das, S.K. (2008). Effect of surface orientation on pool boiling heat transfer of nanoparticle suspensions. International Journal of Multiphysics, 34: 145-160. https://doi.org/10.1016/j.ijmultiphaseflow.2007.08.004

[8] Barber, J., Brutin, D., Tadrist, L. (2011). A review on boiling heat transfer enhancement with nanofluids. Nanoscale Research Letters, 6: 280. https://doi.org/10.1186/1556-276X-6-280

[9] Ji, W.T., Zhang, D.C., Feng, N., Guo, J.F., Numata, M., Xi, G., Tao, W.Q. (2010). Nucleate pool boiling heat transfer of R134a and R134a-PVE lubricant mixtures on smooth and five enhanced tubes. Journal of Heat Transfer, 132(11): 111502. https://doi.org/10.1115/1.4001931

[10] Liang, G., Mudawar, I. (2019). Review of pool boiling enhancement by surface modification. International Journal of Heat and Mass Transfer, 128: 892-933. https://doi.org/10.1016/j.ijheatmasstransfer.2018.09.026

[11] Dong, L., Quan, X., Cheng, P. (2012). An analysis of surface-microstructures effects on heterogeneous nucleation in pool boiling. International Journal of Heat and Mass Transfer, 55(15-16): 4376-4384. https://doi.org/10.1016/j.ijheatmasstransfer.2012.04.006

[12] Ferjančič, K., Golobič, I. (2002). Surface effects on pool boiling CHF. Experimental Thermal and Fluid Science, 25(7): 565-571. https://doi.org/10.1016/s0894-1777(01)00104-2

[13] Kim, J., Jun, S., Laksnarain, R., You, S.M. (2016). Effect of surface roughness on pool boiling heat transfer at a heated surface having moderate wettability. International Journal of Heat and Mass Transfer, 101: 992-1002. https://doi.org/10.1016/j.ijheatmasstransfer.2016.05.067

[14] Arenales, M.R.M., Kumar, S., Kuo, L.S., Chen, P.H. (2020). Surface roughness variation effects on copper tubes in pool boiling of water. International Journal of Heat and Mass Transfer, 151: 119399. https://doi.org/10.1016/j.ijheatmasstransfer.2020.119399

[15] Gheitaghy, A.M., Saffari, H., Zhang, G.Q. (2019). Effect of nanostructured microporous surfaces on pool boiling augmentation. Heat Transfer Engineering, 40: 762-771. https://doi.org/10.1080/01457632.2018.1442310

[16] Sujith Kumar, C., Suresh, S., Praveen, A., Santhosh Kumar, M., Gopi, V. (2016). Effect of surfactant addition on hydrophilicity of $ZnO–Al_2O_3$ composite and enhancement of flow boiling heat transfer. Experimental Thermal and Fluid Science, 70: 325-334. https://doi.org/10.1016/j.expthermflusci.2015.09.014

[17] Nguyen, T., Liu, D., Kayes, M.I., Wang, B., Rashin, N., Leu, P.W., Tran, T. (2018). Critical heat flux enhancement in pool boiling through increased rewetting on nanopillar array surfaces. Scientific Reports, 8(1): 1-9. https://doi.org/10.1038/s41598-018-22693-z

[18] Takata, Y., Hidaka, S., Masuda, M., Ito, T. (2003). Pool boiling on a superhydrophilic surface. International Journal of Energy Research, 27(2): 111-119. https://doi.org/10.1002/er.861

[19] Takata, Y., Hidaka, S., Uraguchi, T. (2006). Boiling feature on a super water-repellent surface. Heat Transfer Engineering, 27(8): 25-30. https://doi.org/10.1080/01457630600793962

[20] Jo, H., Ahn, H.S., Kang, S.H., Kim, M.H. (2011). A study of nucleate boiling heat transfer on hydrophilic, hydrophobic and heterogeneous wetting surface. International Journal of Heat and Mass Transfer, 54: 5643-5652. https://doi.org/10.1016/j.ijheatmasstransfer.2011.06.001

[21] Kumar, C.S.S., Chang, Y.W., Chen, P.H. (2017). Pool-boiling heat-transfer enhancement on cylindrical surfaces with hybrid wettable patterns. Journal of Visualized Experiments, 122: e55387. https://doi.org/10.3791/55387

[22] Guo, Z., El-Genk, M.S. (1992). An experimental study of saturated pool boiling from downward facing and inclined surfaces. International Journal of Heat and Mass Transfer, 35: 2109-2117. https://doi.org/10.1016/0017-9310(92)90056-X

[23] El-Genk, M.S., Guo, Z. (1993). Transient boiling from inclined and downward-facing surfaces in a saturated pool. International Journal of Refrigeration, 16: 414-422. https://doi.org/10.1016/0140-7007(93)90058-G

[24] Reed, S.J., Mudawar, I. (1997). Enhancement of boiling heat transfer using highly wetting liquids with pressed-on fins at low contact forces. International Journal of Heat and Mass Transfer, 40: 2379-2392. https://doi.org/10.1016/S0017-9310(96)00286-4

[25] Chang, J.Y., You, S.M. (1996). Heater orientation effects on pool boiling of microporous-enhanced surfaces in saturated FC-72. Journal of Heat Transfer, 118: 937-943. https://doi.org/10.1115/1.2822592

[26] Yang, S.H., Baek, W., Chang, S.H. (1997). Pool-boiling critical heat flux of water on small plates: Effects of surface orientation and size. International Communications in Heat and Mass Transfer, 24: 1093-1102. https://doi.org/10.1016/S0735-1933(97)00103-6

[27] Evgeniya, O., Feoktistov, D., Kuznetsov, G. (2015). Investigation of drop dynamic contact angle on copper surface. In: EPJ Web Conference, France, 82: 01053(1-5). https://doi.org/10.1051/epjconf/20158201053

[28] Islam, M., Haque, K., Saha, S. (2011). An experimental investigation of pool boiling at atmospheric pressure. Daffodil International University Journal of Science and Technology, 6: 80-86. https://doi.org/10.3329/diujst.v6i1.9337

[29] Cooke, D., Kandlikar. S.G. (2011). Pool boiling heat transfer and bubble dynamics over plain and enhanced microchannels. Journal of Heat Transfer, 133: 052902 (1-9). https://doi.org/10.1115/1.4003046

[30] Cooke, D., Kandlikar, S.G. (2012). Effect of open microchannel geometry on pool boiling enhancement. International Journal of Heat and Mass Transfer, 55: 1004-1013. https://doi.org/10.1016/j.ijheatmasstransfer.2011.10.010

NOMENCLATURE

A	circumferential area, m^2
I	current, A
k	thermal conductivity, $W.m^{-1}. K^{-1}$
L	length, m
Q	heat input, W
q	heat flux, kW/m^2
r	radius, m
T	temperature, K
V	voltage, v
x	distance, m

Greek symbols

α	heat transfer coefficient, $kW/m^2.K$
θ	angle, °

Subscripts

c	cross-section
$LOSS$	loss
m	mean
o	outer
sat	saturation
w	wall