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Enhanced nucleate boiling in an angular geometry found in structured surfaces

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Abstract—Nucleate boiling within an angular geometry, a basic geometry found in many structured boiling heat transfer surfaces, is investigated. Pool boiling tests are conducted using water at atmospheric pressure and a horizontal heated wall with an inclined wall attachment making a line contact to form an angular geometry. It is found that nucleate boiling from the horizontal heated wall is greatly enhanced by a favorable thermal environment within the angular geometry. The effects of the angle between the two walls, and roughness and length of the wall attachment are studied. The results reveal the importance of the angular geometry to the enhanced nucleate boiling heat transfer of structured surfaces, and provide information to the development of more effective surfaces.

1. INTRODUCTION

It is known that nucleate boiling heat transfer can be enhanced by various types of structured surfaces. A scrutiny of these surfaces reveals a common angular geometry included in many structures. This common geometry features an angle of void space formed by two solid walls. For example, in the surface proposed by Ragi [1], featuring spaced restricted openings provided by a formed cover sheet bonded to the substrate as depicted in Fig. 1, an angular geometry is formed between the cover sheet and the substrate. In the commercial boiling surface ECR-40 [2], an angular geometry is found between the integral surface structure and the base surface, as shown in Fig. 2. In a nucleate boiling study using a surface having V-shaped grooves covered with perforated plates [3], it was observed that the vapor phase always started from one of the corners of a groove, as shown in Fig. 3. Apparently, the angular, or wedge-shaped, geometry has a special vapor formation capability which is as yet unknown to us. This geometry may play an important role in providing the high heat transfer performances of these structured boiling surfaces. It is therefore the objective of this study to investigate nucleate boiling within such angular geometry.

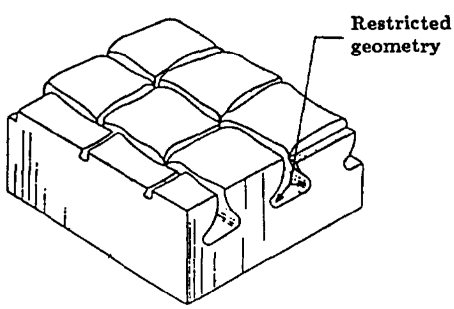

Fig. 2. ECR-40 surface structure by Fujikake [2].

In a surface structure, the two walls forming the angle may not be symmetrically heated. This makes it difficult to define the heating condition in the experiment. In this study, experiments were conducted with only one wall of the angular geometry heated, and the other wall being inactive, as depicted in Fig. 4. If nucleate boiling is enhanced under such test con-

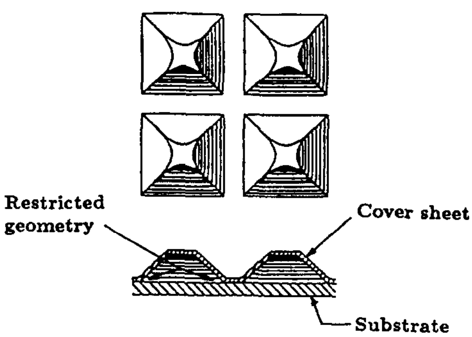

Fig. 1. Composite surface structure by Ragi [1].

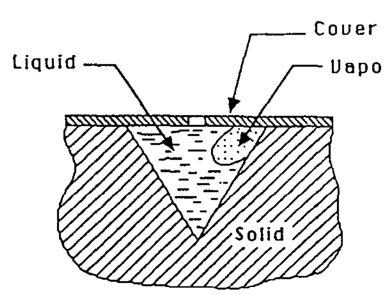

Fig. 3. Grooved surface covered with perforated plate, by Arshad and Thome [3].

NOMENCLATURE

C_d	drag coefficient	Greek symbols	
F	force [N]	β	liquid contact angle [rad]
g	acceleration due to gravity, 9.8 m s^{-2}	θ	angle between the heated wall and the wall attachment, as defined in Fig. 4 [deg]
l	length of the wall attachment, as defined in Fig. 4 [mm]	σ	surface tension [N m^{-1}].
q''	heat flux [W m^{-2}]		
r	cavity radius [mm]	Subscripts	
R	bubble radius [mm]	B	buoyancy
T	temperature [°C]	b	bubble
ΔT	wall superheat, $T_w - T_s$ [°C]	d	drag
U	velocity [m s^{-1}]	l	liquid
x	distance from the contact line of the wall attachment and the heated wall, as defined in Fig. 4 [mm].	m	mean
		s	saturation; surface tension
		w	wall.

ditions, greater enhancement is expected with both walls heated, whether the heating is symmetrical or not. On the other hand, the test result can provide information to the development of a boiling heat transfer enhancement technique featuring inactive wall attachment to the heating surface. Techniques of this type are important for heated surfaces, such as that of a silicon chip, on which incorporation of an integrated structure is impossible.

In the present study, attention is given to the region near the contact line where the restricted geometry makes the thermal environment very different from the environment outside of the restricted region, and the effect of restricted liquid circulation and vapor movement on the heat transfer performance. Factors of influence were identified and explored. Results of this study provide information with regard to both the mechanism and the improvement of enhanced nucleate boiling heat transfer associated with existing structured surfaces that include internal angles. The results can also lead to the development of new and more effective enhancement techniques. The test results using a vertical heated wall and an inclined wall attachment will be published in a separate work.

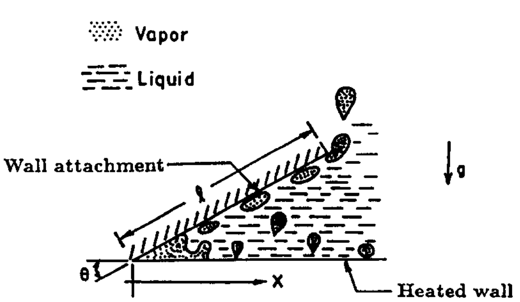

FIG. 4. Nucleate boiling within an angular geometry formed by a horizontal heated wall subjected to an inclined wall attachment.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

A pool boiling experiment was conducted to investigate nucleate boiling within an angular geometry formed by a heated wall and a wall attachment. Tests were run by boiling deionized water at atmospheric pressure using a facility as shown in Fig. 5. The facility included an electrically heated rectangular inconel foil (138 mm × 30 mm × 0.025 mm) serving as the heated wall, located at the bottom of a 21 cm cubical Lexan vessel. A Teflon sheet (3.2 mm thick) was sandwiched between the inconel and the Lexan to protect the latter from high temperature. A Lexan plate (70 mm × 60 mm × 12 mm or 70 mm × 12 mm × 12 mm) sitting on the heated foil served as the wall attachment. The plate had a thickness of 12 mm and an edge of 70 mm in contact with the foil, which was machined knife-sharp with a 45° wedge angle. This minimized any possible effect due to contact between the two walls. The surface roughness of the wall attachment was found influential in the present study. Unless otherwise specified, the surface of the plate was prepared with No. 50 abrasive cloth. The transparency of Lexan allowed visual observation of the vapor phase activity within the angular region. The tilt angle of the wall attachment was adjusted using a rod attached to an edge of the plate. A weight was placed on the attachment to ensure its good contact to the metal foil; otherwise, the attachment jumped up and down on the heated foil, particularly at high heat flux levels when boiling was violent.

Holes of 1 mm diameter were drilled 20 mm apart on the bottom plate of the test vessel, as well as on the sandwiched Teflon sheet, to accommodate three thermocouples attached to the back side along the center line of the heated foil (Fig. 6). The wall attachment was located so that the middle thermocouple was at the center of the contact line, and each thermocouple was at an equal distance x from the contact line of the attachment and the heated foil. The distance

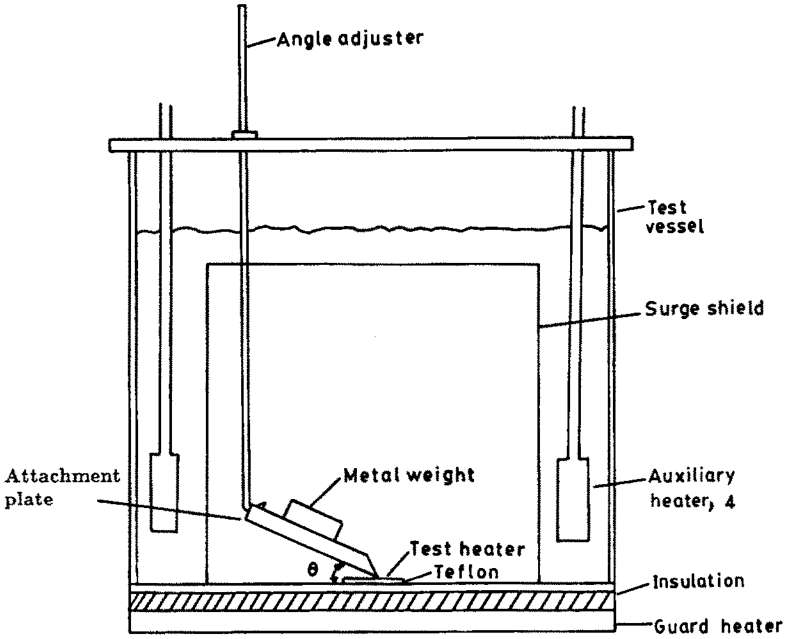

FIG. 5. Test facility.

x could be varied by moving the attachment. Unless otherwise specified, all temperature data presented in this paper were taken at $x = 5$ mm. Because heat conduction along the foil was negligible due to small thickness, each thermocouple basically read the local temperature of the foil. The thermocouple wires ran underneath the bottom wall of the vessel for a considerable distance (half of the vessel length); the conduction error associated with the foil temperature reading was thus negligible. The thermocouple located in the middle always read higher temperature than the other two because, due to symmetry, theoretically, there was no local velocity component in the direction of the contact line. Any boiling crisis was invariably first detected by this thermocouple. In order to restrict our attention to a two-dimensional phenomenon, only the temperature data taken in the middle of the contact line are presented in this paper.

The metal foil was directly heated by d.c. electricity. The electric power cables were first connected to the foil through bolts and nuts. However, chemical deposition was observed in the interstices between the connectors and the foil during the boiling tests. Because of the non-conductive nature of the deposit, the electric resistance became large and a substantial portion of the power was dissipated through heat generation at the connection. This problem was solved by soldering the cables to the foil. The difficulty of soldering a copper cable to the inconel foil was overcome by employing the following technique. A U-shaped notch was first cut at the edge of the foil to accommodate the cable. The cable was then sandwiched between two copper sheets slightly larger than the notch. During soldering, the molten solder was sucked into the crevice created by the two copper sheets, thus binding the cable and foil together (see Fig. 6).

Also included in the vessel were four auxiliary immersion heaters located at the corners for initial

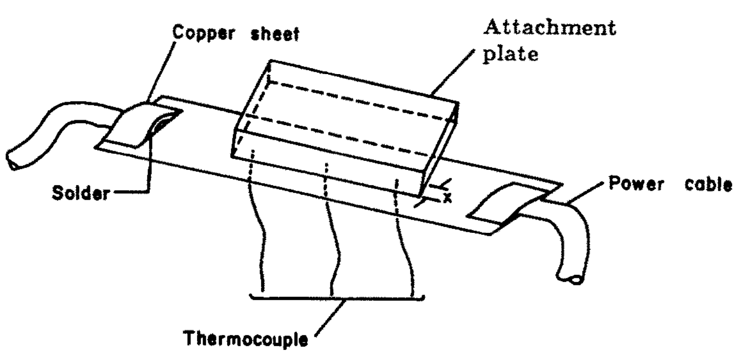

FIG. 6. Test heater and attachment plate.

heating and maintaining the pool temperature at saturation. The disturbance induced by boiling from the auxiliary heaters was shielded from the test heater by a four-walled structure of Lexan (see Fig. 5). This structure simply stood on the bottom of the vessel. Liquid exchange was allowed over the top of the structure. This arrangement also made it possible to maintain the pool temperature at saturation within the shield without heavy insulation on the vessel blocking the vision. The test vessel was insulated by a piece of fiberglass batt at the bottom. A plate guard heater was placed underneath the insulation to eliminate heat loss from the back side of the test heater. A thermocouple was installed between the vessel bottom plate and the insulation. Temperature measured by this thermocouple, along with that of the heated foil, was used to estimate the back heat loss. The guard heater on the other side of the insulation was adjusted to counterbalance the temperature gradient until the heat loss was negligible.

The electricity to the metal foil was provided by a d.c. power supply and was monitored by a voltmeter and an ammeter. All thermocouples were of chromel–constantan (E-type) and monitored by a digital voltmeter. The auxiliary heaters were controlled by a powerstat. A barometer measured the atmospheric pressure in order to determine the saturation temperature.

Boiling tests were conducted using pure water at atmospheric pressure. The test heater was cleaned with acetone before each test. Water was first brought to boiling by both the test heater and the auxiliary heaters for 3 h to drive away air content. During the tests, because vapor was let through a vent on the lid of the test vessel, water was replenished periodically. The replenishment water was heated to saturation and degassed in a separate beaker in advance. The variation in hydrostatic pressure due to change of pool level was taken into account in determining the saturation temperature. The pool temperature monitored by thermocouples placed randomly in the vessel was always maintained within 0.2°C from the calculated saturation temperature.

The following data were recorded during the test: the angle θ between the wall attachment and the heated wall as defined in Fig. 4, the distance x between the themocouples and the contact line, the heated wall temperature, the saturation temperature determined by the system pressure, and heat flux based on the power to the heated wall and the wall area. The error associated with the temperature data is estimated to be within ± 0.2°C, and that with heat flux is within ± 80 W m^{-2}. While the reproducibility varies with test conditions, most of the test data presented in this work were found to be reproducible within ± 0.7°C.

3. RESULTS AND DISCUSSION

The inclined wall attachment was observed to significantly affect the local nucleate boiling near the contact line of the heated wall and the wall attachment, where bubbling was remarkably more vigorous than elsewhere on the heated wall. Vapor bubbles generated move upward along the wall attachment as depicted in Fig. 4. The influence of the angle on the local boiling heat transfer performance is shown in Fig. 7, where boiling data taken at the location 5 mm away from the contact line under different angles are exhibited. At a 90° angle, the boiling curve coincides with that of the heated surface without attachment. Incipient boiling occurs at about 1000 W m^{-2}. The data then follow a curve of greater slope to the power limit. No discernible difference was observed on the data curve recorded during decreasing power.

In the 30° test, stagnant vapor phase was observed along the contact line starting from a very low heat flux (about 300 W m^{-2}). There was no bubble motion until heat flux reached about 1000 W m^{-2} where the boiling curve changed slope and established nucleate boiling set in. The slope was maintained and then decreased at a heat flux of about 50 000 W m^{-2}. The test was terminated when heat flux reached about 62 000 W m^{-2} and the thermocouple in the middle of the heated surface read a soaring temperature signifying burnout phenomenon. A similar curve pattern was followed by data taken at a 15° angle, with a difference of higher heat transfer coefficient and lower critical heat flux.

At a small angle, vapor phase always formed at a very low superheat near the contact line of the two walls while there was no sign of nucleate boiling elsewhere on the heated wall. This clearly indicates an enhanced incipient boiling within the restricted region over that on an open heated wall. Based on data of 30° and 15° compared to that of an open surface (90°) in Fig. 7, nucleate boiling heat transfer coefficients are also higher within the restricted region than on an open surface. This special nucleate boiling behavior is attributed to the favorable local thermal environment within the angular geometry.

The extraordinary vapor generation within the restricted angular geometry was obviously not due to inadequate degassing because, first of all, the system was degassed for 3 h before each test, and secondly, excessive bubbling was observed only within the restricted geometry. If the system were not properly degassed, excessive vapor generation should have been observed all over the heated wall, not limited to a certain region. During the test, it was found that as soon as the wall attachment was removed from the heated wall, no excessive bubble generation was observed. This indicates that the extraordinary nucleate boiling is solely due to the restricted geometry created by the wall attachment. Nucleate boiling from the cavities on the heated wall within the restricted geometry is enhanced by a favorable thermal environment characterized by a special liquid temperature profile.

Nucleate boiling in restricted geometries has been studied in many works, including a narrow gap

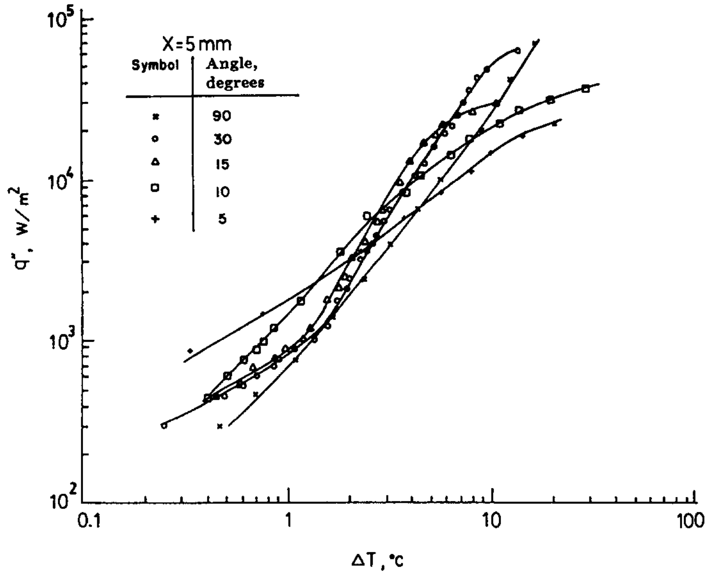

FIG. 7. Influence of angle on local boiling heat transfer performance.

between a horizontal disk and a parallel inactive disk [4–7], an open-ended vertical narrow annulus [8, 9], a vertical annulus with closed bottom [10, 11], a vertical rectangular channel [12], a vertical eccentric annulus [13–15], and a horizontal annular gap [16–18]. However, the attention has been on the fully developed nucleate boiling and boiling crisis. The incipient boiling behavior has been studied only for a narrow vertical rectangular channel [12]. In that study, the incipient boiling equation by Bergles and Rohsenow [19] was found to agree with the lower limit of the measured incipient boiling data.

A number of incipient nucleate boiling models have been proposed based upon different criteria. A recent summary including ten models can be found in ref. [20]. In all the models, a bubble is considered to exist when a bubble equilibrium requirement is met, which is determined based upon mechanical equilibrium of the bubble and the Clausius–Clapeyron equation, as qualitatively shown in Fig. 8(a). Incipient boiling takes place when the liquid temperature is tangent to the bubble equilibrium curve. A major difference among the models is in the bubble growth criterion. The bubble growth criterion used by Han and Griffith [21] is that, based on potential flow analysis, the liquid temperature should satisfy the bubble equilibrium requirement at $y = 1.5r$. Bergles and Rohsenow [19] considered that a bubble can grow if the liquid temperature at the bubble height, $y = r$, is greater than the bubble equilibrium temperature. The model proposed by Yin and Abdelmessih [22] took an empirical approach to treat the y–r relationship. Instead of proposing a new incipient nucleate boiling model, the objective of this part of the work is to demonstrate that the extraordinary nucleate boiling behavior in an angular geometry is supported by the existing theory.

Among all the incipient nucleate boiling models, Bergles and Rohsenow's work [19] has been most widely accepted. Its result was even employed in recent works [12, 20, 23]. According to their model, for an open surface, incipient boiling takes place when the liquid temperature profile is tangent to the bubble equilibrium curve. This happens when the liquid temperature profile I in Fig. 8(a) is reached. Bubbles will then be generated from cavities of size r_{I}, provided cavities of this size are available on the surface and residual vapor phase exists to function as nuclei in these cavities.

In the case of an angular geometry, the liquid temperature profile near the heated wall is quite different from that of an open wall. Near the apex of the restricted region, because of limited liquid circulation, conduction is the dominant mode of heat transfer. Consider a liquid temperature profile in the space of height δ between the heated wall and the wall attachment at a particular location x, as depicted in Fig. 8(b). Typical liquid temperature profiles in such a space are shown as profiles II and III in Fig. 8(a). In these profiles, the temperature gradient at $y = 0$, and thus the wall heat flux, is taken to be the same as that of profile I of an open surface for the purpose of comparison. At $y = \delta$, the liquid temperature gradient is taken as zero, based on an assumed negligible heat transfer through the wall attachment, which is a Lexan plate in the present case. Qualitatively, these temperature profiles satisfy the bubble equilibrium requirement at a lower superheat or heat flux than the profiles on an open surface. For instance, when temperature profile II is reached, which has an intersection with the bubble equilibrium requirement curve at $y = \delta$, bubbles will be generated from surface cavities of size δ and greater, provided residual vapor phase is available in these cavities. The fact that the wall superheat of profile II, ΔT_{II}, is smaller than that of profile I, ΔT_{I}, indicates that at

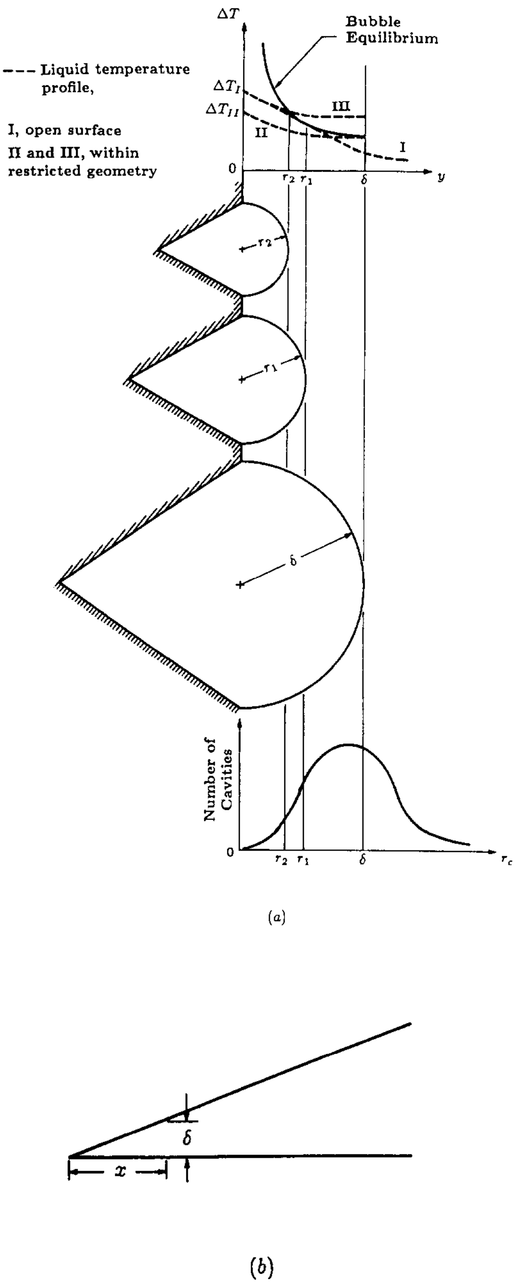

Fig. 8. Nucleate boiling in a restricted angular geometry.

the wall heat flux that incipient boiling takes place on an open heated wall, nucleate boiling can take place at a lower wall superheat within an angular geometry.

Also consider the liquid temperature profile within the restricted region when the heated wall is at the superheat ΔT_{I}, given as profile III in Fig. 8(a). It is shown that profile III intersects the equilibrium profile at $y = r_2$. Hence surface cavities of size r_2 and greater could generate bubbles under this condition. This indicates that at the wall superheat that incipient boiling takes place on an open heated wall, cavities of a range of sizes could generate bubbles within an angular geometry.

Since the restricted space has one wall generating heat and the other insulated, the liquid temperature can increase indefinitely, no matter how small the wall heat flux is, if heat transfer in the direction normal to y is negligible. Incipient boiling will thus take place when the liquid temperature meets the bubble equilibrium requirement. However, the growth of bubbles at low heat flux levels could be limited by the cold liquid outside the restricted region, and bubbles could maintain their sizes without departure. Equilibrium is reached between evaporation at the bubble base and condensation where the bubble meets the cold liquid. This explains the existence of stationary bubbles observed at very low heat flux levels near the contact line in the present experiment.

The above was a qualitative model of incipient boiling. The major difficulty associated with a quantitative analysis will be in the natural convective liquid temperature profile within the angular geometry. Of particular interest is the size of the region, in terms of x, where heat is transferred basically through conduction and the liquid within the angular geometry is superheated. A relevant work was done on natural convection within a corner formed by a vertical wall and an inclined wall [24]. A recent work studied a corner formed by two vertical walls [25]. However, none of these results can be directly applied to the present geometry. Even if the liquid temperature profile is known for the present geometry, for an angular geometry in one of the structured boiling surfaces such as those shown in Figs. 1–3, the liquid temperature profile will be different. The temperature profile within the angular geometry will be dependent upon the geometry and dimensions of the entire surface structure. However, the present qualitative analysis has provided support to the observed enhanced nucleate boiling in an angular geometry.

The data of 5° angle in Fig. 7 demonstrate the smallest slope among all the curves, and the curve is nearly a straight line without significant change of slope signifying departure from natural convection as observed at 30° and 15° angles. This behavior is considered to be related to the vapor strip on the contact line at small angles. At a small angle, bubbles generated on the contact line coalesce into a vapor strip starting from a very low heat flux. Even though the boundary of the vapor strip fluctuated all the time, the width of the strip generally increased with heat flux and decreased with the angle. A similar phenomenon was reported by Katto and Yokoya [4] when the gap between a horizontal heated wall and a parallel interference plate is decreased below a certain width. The small gap is mostly filled with vapor and the heated wall is dry. Liquid penetrates into the gap and vigorous nucleate boiling takes place near the entrance. In the present angular geometry, the gap width δ between the heated wall and the interference wall (wall attachment) decreases toward the contact line (see Fig. 8(b)); therefore, vapor fills the gap starting from the contact line where the gap width is zero, up to a certain location where the gap width is so large that vapor does not fill the gap. This forms a strip of vapor on the contact line, as observed in the present experiment. Katto and Yokoya also observed that vapor can fill a larger gap at a higher heat flux. This agrees with the present observation in an angular geometry that the width of the vapor strip increases with heat flux since, for a fixed angle, a larger δ corresponds to a larger x. Also, for a fixed gap width, δ, the dimension x is smaller if the angle is larger (Fig. 8(b)). Therefore, the vapor strip becomes narrower with a larger angle.

At 10° angle, the heated wall near the contact line was better wetted because the vapor strip was narrower. Therefore, there was more evaporation/boiling at 10°, and the data exhibited a curve of greater slope than that of 5°, with a slight change of slope in the high heat flux range. However, the slope of the curve is still smaller and the change of slope is not as significant compared with 30° and 15° tests. It is interesting to observe the gradual change of the curve pattern from a typical boiling curve at a large angle to a straight line with a relatively small slope at a small angle.

In the high heat flux range, it is noticed that the heat transfer coefficient first increases, then decreases as the angle becomes smaller. This behavior agrees with that reported by Katto *et al.* [7] for nucleate boiling of water on a horizontal surface subjected to a parallel interfering disk above. In the relatively high heat flux range covered in their experiment, the heat transfer coefficient is invariable while the interfering disk is placed at a distance greater than 2.0 mm away from the boiling surface. Within 2.0 mm, the coefficient first increases, then decreases as the interfering disk moves toward the boiling surface. In the present experiment, decrease in the angle results in a smaller gap width between the boiling surface and the interfering wall attachment. Therefore, the change of heat transfer coefficient with the angle qualitatively agrees with the change in gap width between two facing disks.

Based on the data of 30° and 15° angles exhibited in Fig. 7, the critical heat flux at which burnout occurs is lower for a smaller angle. This again qualitatively agrees with the result reported by Katto *et al.* [7], which showed that critical heat flux increases with gap width between the heated wall and the parallel interfering wall. At 10° and 5°, no significant temperature jump, indicating critical heat flux, was observed during the tests. This phenomenon was also observed by Katto *et al.* [7] for a very small gap.

Data presented in Fig. 7 are all taken at 5 mm from the contact line. In fact, at a fixed heat flux, the wall superheat varies with the distance from the contact line. This is shown by the data exhibited in Fig. 9, which were obtained by moving the contact edge of the wall attachment away from the thermocouples. The data in Fig. 9 demonstrate that at a particular angle (10°) and a fixed heat flux (1.37×10^4 W m^{-2}),

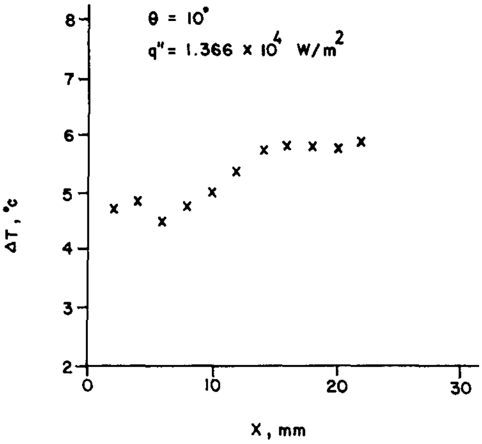

FIG. 9. Variation of wall temperature within the angular geometry.

the heated wall temperature first decreases slightly, then increases as the thermocouple moves away from the contact line. The temperature is relatively low near the contact line because of enhanced nucleate boiling under the favorable thermal environment provided by the restricted geometry. The temperature drops toward a minimum at about 6 mm from the contact line, and then increases toward a constant value as the thermocouple moves toward the edge of the wall attachment at $x = 12$ mm. This value agrees with that recorded without the wall attachment as shown in Fig. 7 with $\theta = 90°$.

The effect of boiling surface roughness on heat transfer has been intensively studied in the past. In the present study, the influence of wall attachment roughness is studied for the first time. Boiling surface roughness is known to affect bubble formation, and wall attachment roughness is found to affect bubble motion in the present study. As depicted in Fig. 4, in an angular geometry, the large amount of bubbles generated near the contact line have to slide along the inclined wall attachment. Bubbles were observed to move faster along a roughened wall attachment than along a smooth one. This characteristic affects both the boiling heat transfer coefficient and the critical heat flux.

As exhibited by the boiling data in Fig. 10, for a fixed angle of 10°, the curve with a wall attachment prepared with a coarse abrasive cloth (No. 36) shows no sign of burnout within the test range. The test curve with a surface prepared with a less coarse abrasive cloth (No. 50), which is in fact the 10° curve in Fig. 7, shows lower heat transfer coefficient in the high heat flux range. A mirror-smooth surface demonstrates the lowest coefficient and burnout at a heat flux as low as about 7×10^3 W m^{-2}. The above data suggest that a rougher wall attachment yields higher boiling heat transfer coefficient and higher critical heat flux.

During the tests with the smooth wall attachment,

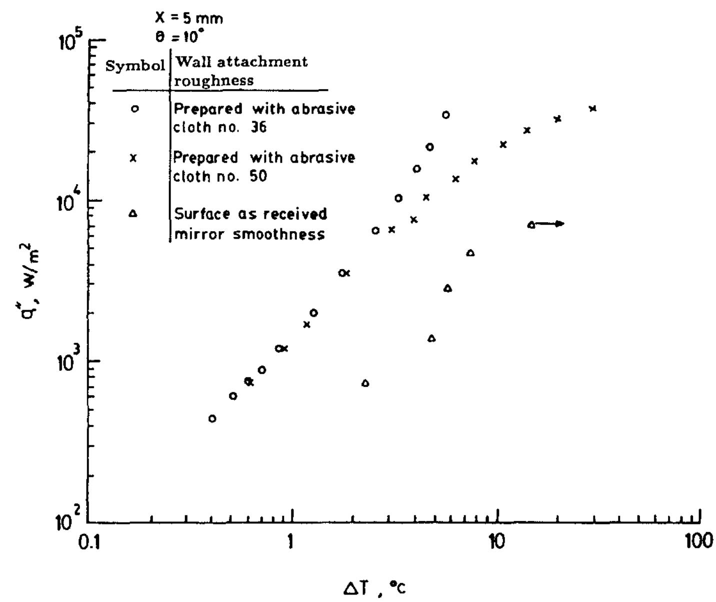

FIG. 10. Influence of wall attachment roughness on boiling heat transfer performance.

stagnant vapor mass was observed to be always in existence between the two walls on the contact line starting from a low heat flux. Portions of this vapor mass began to move along the wall attachment in the form of bubbles at a higher heat flux, but at a much lower velocity, compared with that along a rough wall attachment. The slow vapor mass movement is believed to be responsible for the low heat transfer coefficient and the low critical heat flux with a smooth wall attachment.

The motion of an adiabatic bubble sliding along a vertical wall against gravity has been studied by Cornwell and Schueller [26] and around the circumference of a horizontal tube by Cornwell *et al.* [27]. Bubble motion along an inclined wall can be analysed following a similar procedure as presented below. In this analysis both the bubble and liquid velocities are taken to be constant. A balance is reached among the surface tension, the buoyancy, and the drag force acting on the bubble ascending along the inclined wall (Fig. 11); i.e.

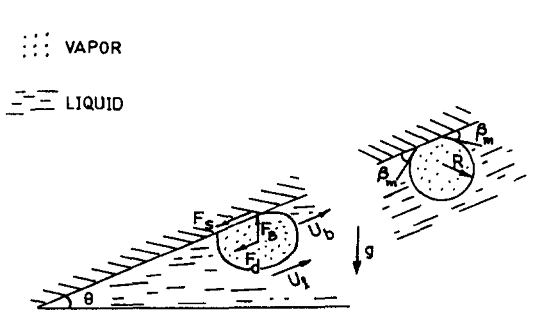

FIG. 11. Forces acting on a bubble moving along an inclined wall.

$$F_s + F_d = F_B \sin\theta. \qquad (1)$$

The net surface tension force acting against bubble movement is due to the change in the liquid contact angle from a maximum at the trailing edge to a minimum at the leading edge of the vapor–solid contact area. By assuming a sinusoidal variation of the contact angle along the tri-phase contact circumference, the net surface tension force obtained through integration is given as [26]

$$F_s = \pi\sigma R \sin\beta_m(1-\cos\beta_m) \qquad (2)$$

where β_m is the mean liquid contact angle around the contact circumference.

The drag force can be written as

$$F_d = \frac{C_d\rho_l}{2} R^2(U_b - U_l)^2(\pi-\beta_m+\sin\beta_m\cos\beta_m) \qquad (3)$$

with C_d the drag coefficient of a sphere and U_l the liquid velocity component along the wall, which is smaller than U_b, the bubble velocity, in the present case.

The buoyancy force can be written as

$$F_B = \frac{\pi R^3}{3}(1+\cos\beta_m)^2(2-\cos\beta_m)\rho_l g. \qquad (4)$$

By substituting the three force expressions into equation (1), it follows that

$$U_b - U_l = \left\{\frac{2\pi}{3C_d}\frac{1}{\pi-\beta_m+\sin\beta_m\cos\beta_m}\left[Rg\sin\theta(1+\cos\beta_m)^2(2-\cos\beta_m) - \frac{3\sigma}{\rho_l R}\sin\beta_m(1-\cos\beta_m)\right]\right\}^{1/2}. \qquad (5)$$

The above equation enables calculation of bubble sliding velocity for a particular liquid contact angle. The contact angles of water on the wall attachments of different roughness employed in the present study were compared by measuring the receding liquid contact angle as the wall was slowly pulled upward from a pool of water at the same state as that in the boiling tests. It was determined that the contact angle is smaller on a rough surface than on a smooth one. By invoking equation (5), calculation shows that smaller contact angle β_m yields higher bubble sliding velocity U_b. This agrees with the present experimental observation that vapor bubbles move faster along a rough wall attachment than along a smooth one. A high bubble velocity can increase the heat transfer coefficient near the contact line because the surface is better wetted as the vapor phase is less likely to accumulate. The critical heat flux is improved for the same reason. Heat transfer coefficient can also be increased by the greater liquid circulation and agitation. It is also likely that the wetting condition is improved by the rough wall feeding liquid toward the corner through surface scratches due to capillary effect.

The quantitative result regarding the bubble velocity can be obtained based on equation (5) and the contact angle measured. The result can be compared in turn with measured bubble velocity using a high-speed camera. However, this exercise would not help us in the quantitative prediction of boiling heat transfer performance of a structured boiling surface. The relationship between the bubble velocity and the boiling heat transfer performance is difficult to determine, particularly when the bubbles move inside tiny passages in a structured surface, after being generated from an angular geometry in the structure. Empirical treatment for individual surface geometry may be necessary. The present work simply points out the importance of the surface roughness of bubble escaping passages, which ought to be considered in the design and modeling of a structured boiling surface.

Tests were also conducted using a wall attachment of shorter length. Compared to the data taken with a longer wall attachment at the same angle, a shorter attachment yields higher heat transfer coefficients and a higher critical heat flux. This is shown in Fig. 12(a) where data taken with a short wall attachment (12 mm) are compared with those taken with a long attachment (60 mm) at an angle of 30°. Both attachments have the same length (70 mm) of contact line with the heated wall. A similar comparison is made in Fig. 12(b) for 5° angle. The data with the long wall attachment exhibited in both Figs. 12(a) and (b) are from Fig. 7, which show heat transfer coefficients always lower than the data for a short attachment. Also, in Fig. 12(a), there is no sign of burnout in the test using a short wall attachment at the critical heat flux of the long attachment test. With a short attachment, it was observed that bubbles readily escaped from the restricted geometry between the heated wall and the wall attachment, and there was less vapor mass accumulation within the restricted geometry. This is because the distance a bubble has to travel along the wall attachment before escaping to the open space is shorter. The heat transfer coefficient and critical heat flux are, therefore, higher with a shorter wall attachment.

The present results reveal the importance of angular geometry to the enhanced nucleate boiling by structured surfaces including such geometry. The contribution of the angular geometry is in the incipient boiling at a low wall superheat, and the high vapor generation rate. Even though the present results were based on an angular test heater with only one wall heated and the other inactive, greater enhancement in nucleate boiling is anticipated with both walls heated, whether the heating is symmetrical or not. This can be easily seen from Fig. 8(a). When both walls are heated, the liquid temperature profile would have a higher value and display an upswing at $y = \delta$, which makes the profile intersect the bubble equilibrium curve at a lower wall superheat, and makes possible more bubble generation at a particular superheat, compared with the case with only one wall heated.

With regard to the design of structured surfaces, the present results suggest inclusion of a large quantity of angular geometries in the surface structure. However, the critical heat flux may be reduced by an excessive quantity of angles because vapor tends to accumulate in the angular geometry. The proper quantity of angles should be determined based on actual testing of the surface structure as well as the consideration of enhanced nucleate boiling desired and possible reduction in the critical heat flux by such geometry. The present results also suggest a rough internal surface of the structure on which vapor bubbles are to travel. The height (or thickness) of the structure should be small to facilitate bubble escape. However, there ought to be a lower limit regarding the height in order to warrant a restricted geometry.

4. CONCLUSION

Nucleate pool boiling within an angular geometry found in structured boiling surfaces has been studied using a horizontal heated wall and an inclined wall attachment making a line contact. It is concluded that nucleate boiling is significantly enhanced within the angular geometry in terms of lower wall superheat for incipient boiling and higher vapor generation rate than an open surface. This is attributed to a favorable thermal environment within the angular geometry where liquid circulation is restricted. The enhanced nucleate boiling is supported by well accepted nucleate boiling theory. On the other hand, boiling crisis occurs at a lower heat flux within an angular geometry than on an open surface because of impeded upward movement of vapor bubbles and consequent accumulation of vapor mass within the restricted geometry.

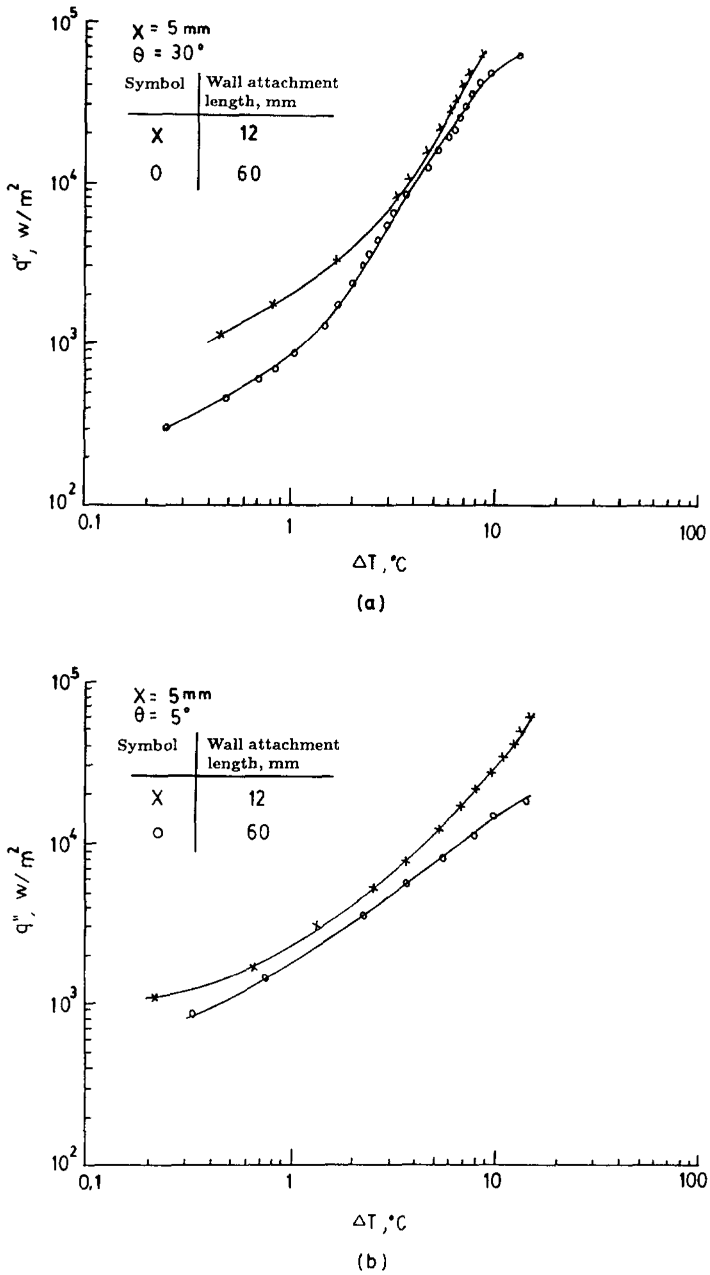

FIG. 12. Effect of wall attachment length.

The roughness of the wall attachment has an influence on the heat transfer performance in that it affects the movement of vapor bubbles along the attachment. Vapor bubbles were observed to move faster on a roughened attachment than on a smooth one. This agrees with the result of an analysis considering bubble movement on an inclined wall under the balance of surface tension, buoyancy, and drag force. Both the heat transfer coefficient and the critical heat flux are higher with a roughened wall attachment than with a smooth one. It is also found that a short wall attachment yields both higher heat transfer coefficient and higher critical heat flux than a long one by reducing the distance that escaping bubbles need to travel on the attachment.

Based upon the results of the present study, it is recommended that a boiling-enhancing surface structure should include a large quantity of internal angles because of their superior incipient boiling and bubble generation capability. It should be noticed that the critical heat flux could be reduced as the result of vapor accumulation within the structure. The present results also suggest that a rough internal surface of the structure, or a short vapor escape passage, can facilitate vapor phase movement and increase the critical heat flux.

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AMELIORATION DE L'EBULLITION NUCLEEE PAR UNE GEOMETRIE ANGULEUSE DANS LA STRUCTURE DE LA SURFACE

Résumé—On étudie l'ébullition nucléée dans une géométrie angulaire, une géométrie de base que l'on trouve dans les surfaces pour ébullition. Des essais d'ébullition en réservoir sont conduits avec de l'eau à pression atmosphérique et une paroi horizontale chaude avec un pan incliné qui forme une géométrie anguleuse sur une ligne de contact. On trouve que l'ébullition nucléée est fortement augmentée par un environnement thermique favorable dans la géométrie anguleuse. Les effets de l'angle entre les deux parois, la rugosité et la longueur de la ligne de contact sont étudiés. Les résultats révèlent l'importance de la géométrie sur l'augmentation de transfert thermique par ébullition nucléée et ils fournissent une information pour le développement de surfaces plus efficaces.

INTENSIVIERTES BLASENSIEDEN IN DER WINKLIGEN GEOMETRIE STRUKTURIERTER OBERFLÄCHEN

Zusammenfassung—Das Blasensieden in der winkligen Geometrie, welche als Grundform bei vielen strukturierten Siedeoberflächen vorhanden ist, wird untersucht. Es werden Messungen beim Behältersieden mit Wasser unter Atmosphärendruck durchgeführt. Die winklige Geometrie wird dadurch hergestellt, daß eine waagerechte Heizfläche mit einer geneigten Wand in linienförmigen Kontakt gebracht wird. Es zeigt sich, daß das Blasensieden an der waagerechten Heizfläche stark durch eine thermisch günstige Umgebung in der winkligen Geometrie verbessert wird. Die Einflüsse des Winkels zwischen den beiden Flächen, der Rauhigkeit und der Länge der Berührungslinie werden untersucht. Die Ergebnisse bestätigen die große Bedeutung der winkligen Geometrie für die Erhöhung des Wärmeübergangs beim Blasensieden an strukturierten Oberflächen und geben Informationen für die Entwicklung noch effizienterer Oberflächen.

ИНТЕНСИВНОЕ ПУЗЫРЬКОВОЕ КИПЕНИЕ В ОБЛАСТЯХ С УГЛОВОЙ ГЕОМЕТРИЕЙ, ВСТРЕЧАЮЩИХСЯ В СТРУКТУРИРОВАННЫХ ПОВЕРХНОСТЯХ

Аннотация—Исследуется пузырьковое кипение в областях с угловой геометрией, преобладающей во многих структурированных поверхностях теплопереноса при кипении. Проводятся опыты по кипению в большом объеме с использованием воды при атмосферном давлении и горизонтальной нагреваемой стенки с присоединенной стенкой, являющейся линией контакта в образуемой области с угловой геометрией. Найдено, что пузырьковое кипение на горизонтальной нагреваемой стенке существенно интенсифицируется в силу благоприятного теплового окружения в угловой геометрии. Изучаются эффекты величины угла между двумя стенками, а также степени шероховатости и длины присоединенной стенки. Полученные результаты иллюстрируют влияние области с угловой геометрией на усиление теплопереноса на структурированных поверхностях при пузырьковом кипении и предоставляют информацию для разработки более эффективных поверхностей.