Heat Transfer in Nucleate Boiling within a Vertical Narrow Space*

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Pool boiling heat transfer within a confined space was studied for the vertical narrow space between parallel heated and unheated planes. Experiments were performed for water under atmospheric pressure at a heat flux range up to the critical heat flux. The gap size was decreased from 5 mm to 0.15 mm in four steps and two heating surfaces of different heights were tested under the open or closed periphery condition of the narrow space. Based on observed boiling behavior, heat transfer mechanisms were deduced and the heat fluxes calculated by proposed models were compared with experimental data.

Key Words: Nucleate Boiling, Heat Transfer Enhancement, Confined Space

1. Introduction

It is well known that the heat transfer characteristics in pool boiling within a narrow space are quite different from those of the conventional unconfined pool boiling. There may arise in the narrow space two opposing effects, either to promote or to deteriorate heat transfer.

Katto et al.\textsuperscript{[11],[12]} investigated a horizontal circular heating surface covered by an unheated glass plate with a specified gap left between them. They indicated the evaporation of thin liquid film under elongated bubbles as the dominant mechanism for the boiling within the narrow gap. Ishibashi and Nishikawa\textsuperscript{[13]} identified an isolated and a coalesced bubble regime at low heat flux through experiments using vertical, open-ended annuli, and empirically correlated the enhanced heat transfer coefficient obtained in the latter regime. Yao and Chang\textsuperscript{[14]} used the vertical annuli with closed bottom and distinguished three boiling regimes on the map where the Bond number and the boiling number were chosen as the coordinates. Critical heat flux for the boiling within a narrow space was studied in Refs.\textsuperscript{[11]-[10]} for various space shapes and configurations.

Most of the investigations mentioned above were focused on a special configuration of a narrow space, and gave limited information on the important parameters which affect heat transfer, such as the size of the heating surface and its periphery condition. In the present study, systematic experiments were performed on the nucleate boiling heat transfer within a vertical narrow space. Two heating surfaces of different heights were set face to face with unheated glass plates to form a uniform gap ranging from 0.15 to 5 mm. Open and closed periphery conditions of the heating surface were tested. In the former case, all four sides of the narrow space were open to bulk liquid, while in the latter the bottom and both sides were closed and only the top was open to allow for the vapor removal from and liquid penetration to the narrow space. Heat transfer rate and critical heat flux were measured for distilled water under atmospheric pressure at the heat flux ranging from the inception of boiling up to the critical heat flux. Based on experi-
mental data and observed boiling behavior, heat transfer mechanisms peculiar to the narrow space were discussed and heat transfer characteristics were predicted from proposed models.

2. Experimental Apparatus and Procedure

An outline of the apparatus is shown in Fig. 1. A heating surface assembly (3) is supported in the middle of an inner boiling vessel (4) by horizontal tubes. Details of the heating surface assembly and the narrow space geometry are shown in Fig. 2. The heating surfaces are rectangular copper plates, 30 mm in width and 30 mm in height for Surface A and 30 mm in width and 120 mm in height for Surface B, their peripheries surrounded by thin fins to reduce heat loss. The fin and heating surface are cut out to form a continuous unit body to prevent preferential bubble generation from the connecting line. Heat flux is conducted to the heating surface via the copper block from the electric heater (6) which is inserted in the bottom slits of the block. Thermocouples are positioned at the center of Surface A, and at four locations along the vertical axis for Surface B; 15, 45, 75 and 105 mm from the lower edge. At each location, three thermocouples are embedded at depths of 3, 11 and 19 mm from the surface, respectively. Both the leads for the power supply and thermocouples are taken out from the heating surface assembly through the supporting tubes.

The vertical narrow space is formed between the heating surface (1) and the unheated glass plate (3) as shown in Fig. 2. Gap sizes of 5.0, 2.0, 0.6 and 0.15 mm are accurately set with the aid of space adjusters (5) of specified thickness. In the closed periphery condition, an enclosure spacer (4) is additionally attached to close the side and bottom edges of the heating surface.

Heat flux and surface temperature were calculated from the temperature gradient indicated by thermocouples under the assumption of one-dimensional heat conduction. In the case of Surface B, an average value for the four vertical locations was used as the representative.

Experiments were performed under the condition of saturated boiling of distilled water at atmospheric pressure. Prior to each test run, the heating surface was finished with No. 0/5 emery paper to ensure the same surface conditions for all test runs. Measurements except those of critical heat flux were carried out by reducing heat flux step by step in order to avoid boiling hysteresis.

3. Experimental Results

Figure 3 shows boiling curves for Surface B, where $q$ is the heat flux, $\Delta T_{sat}$ the degree of superheat of the heating surface, and $s$ the gap size. Measured critical heat fluxes are also shown by arrows in the figure. Typical boiling behavior for each gap size is shown in Fig. 4. The following general features are
noted in Fig. 3. Decreasing the gap size to some extent shifts the boiling curves to the left, while further decrease moves them back toward higher wall superheat. This tendency was also recognized in previous results quoted in the literatures. Another observed feature is the slope change in the boiling curves, depending on the gap size and the level of heat flux. This may be associated with the change in the heat transfer mechanisms, which are inferred from boiling behavior as discussed in the following.

3.1 Large gap at $s=5\text{ mm}$

In this case, an isolated bubble is smaller than the gap size and bubble behavior, and heat transfer characteristics are almost similar to those in the unconfined pool boiling except that higher heat transfer rate is achieved due to the additional two-phase forced convection within the space. The enhancement in heat transfer for the closed periphery condition is more pronounced due to the countercurrent of bubble outflow and liquid inflow. Similar situations hold true for Surface A.

3.2 Moderate gaps at $s=2$ and $0.6\text{ mm}$

At low heat flux, an isolated bubble is repeatedly squeezed and flattened. As the results, the bubble grows rapidly and which causes the rapid growth of the bubble and covers the whole width of the heating surface. The resulting large enhancement in heat transfer is attributable to the evaporation of thin liquid film formed between flattened bubbles and the heating surface. A more pronounced enhancement with smaller gap size is considered to be caused by the formation of a thinner liquid film, because vapor velocity is higher in a smaller gap even at the same heat flux. A larger enhancement in heat transfer under the closed periphery condition results from the stability of the flattened bubble in the absence of liquid inflow from the side and bottom edges.

At high heat flux, flattened bubbles cover most of the heating surface and prevent the penetration of liquid to the surface, which reduce the degree of enhanced heat transfer. This situation becomes more intense at higher heat flux and results in the surface reaching the critical condition due to dry out, as observed at $s=0.6\text{ mm}$ under the closed periphery condition. Similar behaviors was also observed in the experiment by the use of Surface A.

3.3 Small gap at $s=0.15\text{ mm}$

As seen in Fig. 4, most of the heating surface is always covered with vapor, even at low heat flux, and wavy liquid–vapor interfaces are observed only in the vicinity of the open edges. The part covered by vapor is supposed to be dry because small droplets which occasionally disperse on it are seen to immediately disappear. Hence, the heat transfer may be controlled mainly by the evaporation at the liquid–vapor interface near the open edges. Since the wetted area is very localized compared to the total area, the heat transfer becomes lower than that for the unconfined pool boiling. The deterioration is more pronounced in the case of a closed periphery where liquid is allowed to wet only the surface close to the top edge.

![Fig. 4 Boiling behavior](image-url)
3.4 Effect of height of heating surface

Boiling curves for Surface A with height of 30 mm and for Surface B with height of 120 mm are compared in Fig. 5. The height of the heating surface does not have a marked effect on the boiling curves under the open periphery condition. However, in the case of a closed periphery, the higher surface results in a lower heat transfer rate at a high heat flux for the gap size of 0.6 mm and over the entire heat flux range for the gap size of 0.15 mm. This is due to the vapor blanketing which takes place over the heating surface as a result of the higher heat flux, smaller gap and longer surface which cause the higher vapor velocity and hence prevention of liquid penetration from the open top edge.


Predictive methods for heat transfer are presented for cases of moderate and small gaps where boiling behavior and heat transfer characteristics are quite different from those for unconfined pool boiling.

4.1 Moderate gaps at $s=2$ and 0.6 mm

Boiling behavior in which flattened bubbles rise between the surfaces is similar to that observed in the boiling from an inclined surface facing downwards\textsuperscript{119}. Hence, it appears that the two transport mechanisms of sensible heat and latent heat are also effective in the present case.

The sensible heat is removed from the surface when a rising flattened bubble carries away the superheated liquid layer in front of the bubble. The heat flux is calculated in terms of the transient heat conduction from the heating surface to the liquid during the liquid period $t_l$, defined as the time interval during which a point in question on the heating surface is in contact with the liquid filling the gap. In this calculation, the existence of the opposite, unheated cover plate was neglected, even in the case when a thermal boundary layer within the liquid slug developed and the temperature front reached the unheated plate. This estimation, however, gives a good approximation because the thermal conductivities of the glass material and water are roughly regarded as being the same. Thus, the time-averaged heat flux due to sensible heat transport during the liquid period is

$$q_l = \frac{2h_0dT_{sat}}{\sqrt{\pi a_l t_l}}$$

where $h_0$, $a_l$ are the thermal conductivity and thermal diffusivity of the liquid, respectively.

Immediately after the liquid period, the next rising bubble reaches the point in question on the surface and covers it. During this vapor period $t_v$, heat is transported by the evaporation of the thin liquid film between the surface and the flattened bubble. The heat flux due to this latent heat transport is estimated by the heat conduction across the liquid film;

$$q_v = \frac{h_v}{\delta} \Delta T_{sat}$$

where $\delta$ is the average thickness of the liquid film.

As the surface is exposed alternately to a liquid slug and a vapor bubble, both heat fluxes $q_l$ and $q_v$ are weighted by the void fraction $\frac{t_v}{t_l + t_v}$, defined as the time ratio of the vapor period to the total period, to give a time-averaged heat flux;

$$q = \frac{q_l + q_v}{t_l + t_v} = (1 - f) q_l + f q_v$$

Periods $t_l$ and $t_v$ were measured for Surface B at a distance 0.5 mm horizontally apart from the heating surface at three vertical locations of 20 mm (L-posi-
tion), 60 mm (M-position) and 100 mm (H-position) from the bottom edge, using electric probes which detect liquid and vapor phases. Figures 6 and 7 show measured periods and void fraction. Because of difficulties in measuring the actual film thickness during the boiling process, the thickness of liquid film formed between an unheated surface and an injected air bubble rising within the narrow gap was used for the present calculation. Film thickness measured at M-position for Surface B is \( \delta = 34 \mu m \) for \( s = 2 \) mm and \( \delta = 9 \mu m \) for \( s = 0.6 \) mm, independent of the volume and injection frequency of an air bubble.

Predicted heat flux as an average of local heat fluxes at three positions is compared with the measured heat flux in Fig. 8. Good coincidence between the prediction and the measurement is confirmed for a gap of \( s = 2 \) mm, while the predicted heat flux exceeds the measured heat flux for a gap of \( 0.6 \) mm. This deviation increases with heat flux, and implies that the liquid film is completely evaporated at some locations before the end of the vapor period, which was confirmed by the emergence of dried areas in the observation of the boiling behavior.

### 4.2 Small gap at \( s = 0.15 \) mm

As the surface is wetted only near the open edges, the heating surface is divided into two parts: a wetted area and the remaining dry area. For both periphery conditions, the wetted area was assumed to have a uniform width \( W \) along the open edge where the

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**Fig. 6** Liquid and vapor periods

**Fig. 7** Void fraction

**Fig. 8** Boiling curves for moderate gaps \( s = 2 \) and \( 0.6 \) mm
evaporation of a thin liquid wedge or of splashed fine droplets became a dominant mechanism for heat transfer. At the dry area, on the other hand, the convection of vapor filling the gap was assumed to control the heat transfer. Uniform heat transfer coefficients \( \alpha_w \) and \( \alpha_d \) were assumed for the wetted and dry areas respectively, and the latter was approximately estimated by \( \lambda_v/s \) where \( \lambda_v \) was the thermal conductivity of the vapor. The validity of this approximation was confirmed by the insensitivity of the calculated heat flux to \( \alpha_d \).

Temperature distribution within the heating block was obtained by the numerical solution of three dimensional heat conduction under boundary conditions depicted in Fig. 9. Predicted boiling curves for several combinations of \( \alpha_w \) and \( W \) are compared with the measured data in Fig. 10. The predicted boiling curves shift with the assumed values of \( \alpha_w \) and \( W \), while the slopes of curves reproduce well the trend of the experimental data. If the width of the wetted area is taken as \( W = 2.5 \text{ mm} \) based on the observation of the boiling behavior, \( \alpha_w = 15 \text{ 000 W/m}^2\text{K} \) gives the best agreement with the experimental data under the open periphery condition.

On the other hand, if the same value of \( \alpha_w \) is assumed for the closed periphery condition, the best agreement requires that \( W = 20 \text{ mm} \), which is larger than the observed width oscillating between 5 and 15 mm. Another discrepancy between the predicted and the measured values can be observed in the vertical distribution of heating surface temperature as shown in Fig. 11. While good agreement is obtained for the open periphery condition, the nearly uniform distribution of measured temperatures is contrary to the prediction under the closed periphery condition. These two discrepancies may indicate that the liquid flows filmy down on the opposite unheated plate and wets a bottom part of the heating surface.

![Fig. 9 Boundary conditions for heat conduction in a copper block (Four sides of a block are treated as adiabatic walls)](image)

![Fig. 10 Boiling curves for small gap \( s = 0.15 \text{ mm} \)](image)

![Fig. 11 Vertical temperature distribution of heating surface](image)
5. Concluding Remarks

Nucleate boiling heat transfer within a vertical narrow space of various gap size was studied using heating surfaces of different height and various periphery condition at the surface edges. The heat transfer coefficient increases up to a certain maximum value with decreasing the gap size, especially at a moderate heat flux. However, a further decrease of gap size induces the deterioration of heat transfer over the entire range of heat flux. From observations of boiling behavior, relevant characteristics of heat transfer were classified into three groups in terms of the gap size. Boiling curves for the moderate and small gaps, which are distinctively different from those for unconfined pool boiling, were well reproduced by the predictive methods based on the proposed mechanisms in heat transfer.

References


