EXPERIMENTAL INVESTIGATION ON PARTIAL POOL BOILING HEAT TRANSFER IN PURE LIQUIDS

Article Highlights
- Boiling heat transfer increases with increasing surface roughness
- The bubble shape and oscillating characteristics determined the boiling heat transfer coefficient
- Eotvos and Roshko numbers are related to boiling heat transfer coefficient

Abstract
Saturated partial pool boiling heat transfer on a horizontal rod heater was investigated experimentally. The boiling liquids included water and ethanol. The heating section was made from various materials including SS316, copper, aluminum and brass. Experiments were performed at several degrees of surface roughness ranging between 30 and 360 µm average vertical deviation. The boiling heat transfer coefficient, bubble departing diameter and frequency, and nucleation site density were measured. The data have been compared to major existing correlations. It was found that experimental data do not match with major correlations in the entire range of experiments with acceptable accuracy. The boiling heat transfer area was divided in two complementary areas, the induced forced convection area and the boiling affected area. Based on two dimensionless groups, including Eötvös and Roshko numbers, a semi-empirical model is proposed for prediction of the boiling heat transfer coefficient. It is shown that the proposed model provides improved performance in prediction of the boiling heat transfer coefficient in comparison with existing correlations.

Keywords: induced force convection, pool boiling, surface roughness, heat transfer coefficient.

The nucleate pool boiling phenomenon is widely applied in many engineering processes. The heat transfer mechanism from the surface to the boiling fluid is known to be a very complicated phenomenon. Design, operation and optimization of the involved equipment require precise prediction of the boiling heat transfer coefficient. There has been a lot of research on pool boiling over the past few decades. However, the mechanism of pool boiling heat transfer is still not completely understood. This is because of the intense complexity of three interconnected heterogeneous parameters: 1) bubble departing diameter, 2) bubble departing frequency, and 3) nucleation site density. In addition, the structures of boiling heat transfer surface are usually very complex and contain nucleation cavities with various shapes and sizes. This information is not completely available for every given heating surface.

In this investigation, the experimental data covers a wide range of heating surfaces characteristics and liquids physical properties. Water and ethanol have been selected as the boiling liquids. The cylindrical heating surfaces were made by various metals including SS316, copper, aluminum and brass. Each surface has been sanded with several grades to provide various degrees of roughness. Note that the roughness is defined as the arithmetic average of the vertical deviations of the surface.

The experimental data have been compared to major existing correlations. It is shown that the existing correlations cannot predict the boiling heat transfer coefficient with a satisfactory accuracy. Some of
the existing correlations may agree well with present experimental data in some limited degrees of roughness; however the deviations between present experimental data and existing correlations exceed 50% absolute average error (A.A.E.) in some other degrees of roughness. In this investigation, a new semi-empirical model is presented to predict the boiling heat transfer coefficient with A.A.E. of 11% at full range of roughness degrees, which is much less than the A.A.E. of the existing correlations and is within maximum expected uncertainty of the experimental procedure.

McNelly [1] has proposed one of the first empirical correlations for prediction of pool boiling heat transfer coefficient. In this correlation, the physical characteristics of heating surface are not involved. Rohsenow [2] has proposed an empirical correlation based on the bubble agitation mechanism. In this correlation, the boiling fluid is assumed to be single phase. In the Rohsenow [2] correlation, the Nusselt number is empirically correlated to Prandtl and Reynolds numbers. Mostinski [3] has ignored the surface effects and applied the principle of corresponding states to pool boiling heat transfer. In this correlation, the experimental data are correlated to the reduced pressure and critical pressure of boiling liquid. In this correlation, many tuning parameters have been implemented and additionally the physical properties of heating surface are totally ignored. Stephan and Abdelsalam [4] proposed four specific correlations applying a statistical multiple regression technique to the following liquid classes: water, organics, refrigerants and cryogenics. In these correlations, the bubble diameter is estimated by Fritz [5] correlation. Cooper [6] proposed a new reduced pressure form of pool boiling heat transfer correlation including the roughness of the boiling surface. Gorenflo [7] has proposed an empirical correlation based on the reduced pressure of the boiling liquid. In this correlation, the surface roughness is also included. Application of the Gorenflo [7] correlation requires the specific reference heat flux, \( q_o \) and also reference boiling heat transfer coefficient, \( \alpha_b \). Vinayak and Balakrishnan [8] and also Alavi Fazel, Jamialahmadi and Safekordi [9] have also a wide-ranging survey on some other correlations. In Table 1, the major existing correlations have been summarized.

Modeling of problems with stochastic roughness is very difficult due to the complicated interactions between bubbles and surface. This problem has been reviewed by McHale and Garimella [10]. Also the problem of surface topography is investigated by

<table>
<thead>
<tr>
<th>Author</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>McNelly [1]</td>
<td>[ \alpha = 0.225 \left( \frac{\varphi P}{\rho c T} \right)^{0.69} \left( \frac{\varphi k}{\sigma} \right)^{0.31} \left( \frac{\rho c}{\rho c - 1} \right)^{0.33} ]</td>
</tr>
<tr>
<td>Rohsenow [2]</td>
<td>[ \frac{C_A T}{T_b} = C_b \left( \frac{q}{\mu \rho c} \right)^{0.5} \left( \frac{\sigma}{\varphi} \right)^{0.33} ]</td>
</tr>
<tr>
<td>Mostinski [3]</td>
<td>[ \alpha = b \left( \frac{q}{A} \right)^{0.7} \left( \frac{P}{P_c} \right)^{0.17} \left( \frac{\rho c}{\rho c - 1} \right)^{1.2} \left( \frac{\varphi}{\sigma} \right)^{0.371} \left( \frac{\varphi p c}{\sigma d} \right)^{0.35} \left( \frac{\rho c}{\rho} \right)^{-1.73} ]</td>
</tr>
<tr>
<td>Stephan and Abdelsalam [4]</td>
<td>[ \alpha = 0.23 \left( \frac{k}{\varphi} \right)^{0.674} \left( \frac{\rho c}{\rho c - 1} \right)^{-0.227} \left( \frac{h_{fg} d^2}{\sigma d} \right)^{0.371} \left( \frac{\varphi p c}{\sigma d} \right)^{0.35} \left( \frac{\rho c}{\rho} \right)^{-1.73} ]</td>
</tr>
<tr>
<td>Cooper [6]</td>
<td>[ \alpha = 55 \left( \frac{\rho c}{\rho c - 1} \right)^{0.12 - 0.4435 \phi} \left( - \log P_c \right)^{-0.55} ]</td>
</tr>
<tr>
<td>Gorenflo [7]</td>
<td>[ \frac{F_{wr}}{R_v} \left( \frac{\varphi p c}{\rho c} \right)^{3.158} = \left( \frac{q_o}{g (q/A)} \right)^{n} = 0.9 - 0.3 P_{wr}^{0.3} + 2.5 P_{wr} - \frac{P_{wr}}{1 - P_{wr}} ]</td>
</tr>
<tr>
<td>Nishikawa [24]</td>
<td>[ \alpha = \frac{31.4 P_{wr}^{0.2}}{M_{wr}^{0.7} \varphi \rho c} \left( \frac{8 \varphi p c}{(1 - 0.99 \rho c)} \right)^{0.2} \left( \frac{q_o}{g (q/A)} \right)^{0.8} ]</td>
</tr>
<tr>
<td>Boyko and Kruzhiline [25]</td>
<td>[ \alpha = 0.082 \left( \frac{h_{fg} d^2}{\sigma d} \right)^{0.371} \left( \frac{T_c \varphi p c}{\rho c} \right)^{0.33} \left( \frac{\varphi}{\rho c} \right)^{0.5} ]</td>
</tr>
</tbody>
</table>
Jabardo [11] and more recently by Moita, Teodori and Moreira [12].

EXPERIMENTAL

Apparatus

The boiling vessel contained 35 L of test liquid. This volume was sufficient to provide pool boiling conditions. The vessel was thermally insulated to minimize the heat loss. The temperature of the system was constantly monitored and regulated to saturation point. The vessel was equipped with a rod heater, which includes four thermocouples, embedded parallel to the heating surface. The input AC electrical power to the rod heater was adjustable by a variable electrical transformer. This transformer converts the input AC voltage of 220 V into any selectable voltage between 0-240 V. The electrical input power to the rod heater is calculated by the product of electrical voltage, current and cosine of the difference between electrical voltage and current. A schematic of the rod heater is shown in Figure 1.

The heating surface temperature was calculated by the integrated form of Fourier’s conduction equation in cylindrical coordinates. In this investigation, several rod heaters have been produced from different metals, including stainless steel 316 (SS316), copper, aluminum and brass. These metals have been selected based on: 1) ease of metalworking, 2) providing a wide range of physical properties and 3) availability of the physical properties in the literature. Because the bubble dynamics and boiling heat transfer coefficient are strongly affected by surface roughness, the surfaces of the heaters were sanded to provide various degrees of roughness. Roughness is generally quantified by the vertical deviations of a real surface from its ideal form. There are many definitions for surface roughness. In this research, the roughness $R_a$, is defined as the arithmetic average of the vertical deviations. Figure 2 presents a typical measured value of surface roughness.

![Figure 1. A schematic of the rod heater.](image1)

![Figure 2. A typical value of surface roughness.](image2)
In this investigation, water and ethanol have been chosen as the boiling liquids, based on: 1) the availability of the physical properties, 2) covering a wide range of physical properties and 3) non-toxic properties.

Procedure

Initially, the entire system was cleaned by circulating and draining the boiling liquid through the vessel, after which the test solution was introduced. The pressure of the system was kept at about 10 kPa (abs.) for an hour by a vacuum pump to degas the boiling liquid. The temperature of the system was then raised to the saturation temperature, and the electrical voltage was supplied to the rod heater up to the maximum value. After reaching steady state, the surface temperature was recorded. Then, the electrical voltage was decreased in various intervals and the recordings were repeated in each interval after steady-state accomplishment. Note that the decreasing path of heat flux was to prevent a hysteresis effect. Each experiment took about five minutes to reach a steady state at any specific condition. The wall temperature was calculated based on the recorded temperatures of the thermocouples inside the rod heater. The distance between thermocouples location and surface was 0.5 mm, which was introduced in the integrated form of the Fourier conduction law in cylindrical coordinates. In addition, the thermal conductivities of individual heating materials were introduced in the Fourier conduction law. The arithmetic averages of four thermocouples were assigned to the actual wall temperature. Note that the measured temperatures from the four measuring points were approximately correspondent within ±0.2 K. Some runs were repeated twice to ensure the reproducibility of the experiments. The physical properties of liquid and heating surface were evaluated at bulk and wall temperature, respectively.

To measure the bubble diameter, photographs of the heating surface have been captured at high speed at each heat flux. The diameters of 20 bubbles were measured and the arithmetic averages were calculated. To measure the bubble departing frequency and nucleation site density, high speed video recording (1000 fps) was performed at each condition. The slow motion of the recordings was analyzed and the nucleation sites were counted and divided to project area. In addition, the bubble frequencies of the nucleation sites were counted and the arithmetic averages were calculated. A Casio EX-FH25 camera was used to record the visual information. The effective resolution of the mentioned camera is 10.1 megapixels and the shutter speed is 1/2000 s. These values were sufficient to provide a sharp and clear image from the heating surface.

A summary of surface characteristics and boiling fluid at various degrees of roughness are presented in Table 2. The measured values of the cosine of contact angle between boiling liquid and surface, which describes the wettability characteristics, are presented in Table 3. Note that the contact angle slightly varies upon variation of surface temperature; the mentioned table presents the average values. The static contact angles are measured when droplet is standing on the surface and the three-phase boundary is not moving. To measure the static contact angle, a droplet of specific liquid was placed on the particular metallic surface. Then, the image of the drop was captured using a digital camera, which was equipped with a micro-lens to magnify the subject. The experimental static contact angle was then defined by fitting the tangent line on the liquid-solid contact point. Note that because of the hysteresis effect, the static contact angle has a spectrum of contact angles ranging from advancing (maximal), to the receding (minimal) contact angle. The equilibrium contact angle is somewhere between those values, and was calculated by the Tadmor correlation [13].

Table 2. The experimented degrees of roughness at various boiling fluids and surfaces

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Aluminum</th>
<th>Brass</th>
<th>Copper</th>
<th>Stainless steel 316</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>1.9×10^-4 m</td>
<td>3.0×10^-5 m</td>
<td>3.0×10^-5 m</td>
<td>2.5×10^-6 m</td>
</tr>
<tr>
<td></td>
<td>3.5×10^-4 m</td>
<td>1.4×10^-4 m</td>
<td>3.6×10^-4 m</td>
<td></td>
</tr>
<tr>
<td>Ethanol</td>
<td>3.0×10^-5 m</td>
<td>3.0×10^-5 m</td>
<td>3.0×10^-5 m</td>
<td>Not tested</td>
</tr>
<tr>
<td></td>
<td>1.9×10^-4 m</td>
<td>1.4×10^-4 m</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>3.5×10^-4 m</td>
<td>3.6×10^-4 m</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3. The measured values of the cosine of contact angle between boiling liquid and surface

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Aluminum</th>
<th>Brass</th>
<th>Copper</th>
<th>Stainless steel 316</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>0.738</td>
<td>0.623</td>
<td>0.435</td>
<td>0.813</td>
</tr>
<tr>
<td>Ethanol</td>
<td>0.902</td>
<td>0.901</td>
<td>0.922</td>
<td>Not tested</td>
</tr>
</tbody>
</table>

Experimental uncertainty

The resolution of the voltmeter, ammeter and the millivolt meter used in the present study was ±1 V, ±0.1 A and ±0.01 mV respectively. The uncertainty in the measurement of temperature is ±0.2 K. The propagation error of the four parallel thermocouples, which are provided to measure the surface temperature, was estimated to (4×0.2)/4 = 0.2 K, which is equal to 0.2/100 = 0.002 or 0.2% at boiling point of
pure water. The maximum propagation error of heat flux in terms of fractional uncertainty is estimated to:

$$\frac{\Delta \left( \frac{q}{A} \right)}{\Delta \left( \frac{q}{A} \right)_{\text{best value}}} = \left( \frac{\Delta f}{f} \right) \left( \frac{\Delta V}{V} \right) = \left( \frac{0.1}{1.1} \right) \left( \frac{1}{50} \right) = 0.11$$

Note that the heat flux is calculated by the products of electrical voltage and current divided to the heating area of the rod heater. It is considered that the heating area was accurate enough to ignore from error analysis. The standard deviation of measured bubble diameters and bubble frequencies for water at 25 kW/m² was typically equal to 0.00009 mm and 2 Hz, respectively. This means that the uncertainty for measured bubble diameter and bubble frequency would be about 0.00009/0.002 = 4.5% for a bubble with 2 mm in diameter and 2/100 = 2% for bubble frequency of 100 Hz. The nucleation sites were visually counted by the slow motion playback of the recorded videos without any significant degree of uncertainty.

**Experimental results**

The raw numerical values of heat flux versus degree of superheat are presented in Figure 3. It can be inferred that the boiling heat transfer coefficient increases with increasing the degree of superheat at any constant condition. In addition, the boiling heat transfer coefficient increases with increasing the surface roughness at any constant condition. This is because of the enhancement in nucleation site density.

The performances of major existing correlation are presented in Figure 4. The numerical comparisons show that for water/SS316, water/brass and ethanol/SS316, the Mostinski [3] correlation has the best performance with 11, 24 and 11% absolute average error (A.A.E.), respectively, while the mentioned correlation has about 56% A.A.E. for Water/Cu and Ethanol/Cu boiling systems. For water/Cu, ethanol/brass, ethanol/Al and ethanol/Cu, the correlation proposed by Stephan and Abdelsalam [4] has the best agreement with experimental data with 44, 16, 14 and 34% absolute average error respectively, while the mention correlation has more than 60% A.A.E. for water/SS316 boiling system at the average experimented roughness and heat fluxes. These deviations are large because of the experimental basis of the existing correlations.

To cross-check the validity of the derived model, an independent dataset was collected [14]. The mentioned dataset consists of water, acetone, ethyl acetate, 2-propanol, methanol and ethanol at the boiling liquid. The heating section was pure copper with smooth texture. To quantify the impact of various physical properties on boiling heat transfer coefficient, the sensitivity analysis was performed by arranging the following equation:

$$\alpha = \rho_l^{g_0} \rho_v^{g_1} \sigma_l^{g_2} C_l^{g_3} \mu_l^{g_4} \sigma_l^{g_5} k_l^{g_6} \rho_v^{g_7} C_v^{g_8} k_v^{g_9}$$

By using the genetic algorithm, the vector $G = [g_0, g_1, \ldots, g_9]$, which represents the exponents of Eq. (1), is found equal to:

$$G = [0.46, -0.86, -0.39, 0.78, -0.16, 0.24, -0.32, -0.26, -0.10, 0.37, -0.10, 0.38]$$

![Figure 3. The raw numerical values of heat flux versus degree of superheat.](image)
Figure 3. The raw numerical values of heat flux versus degree of superheat.

**Modeling**

According to Newton's cooling law, the heat transfer is proportional to the area and the thermal driving force, i.e.:

$$qA = \alpha A \Delta T$$  \hspace{1cm} (3)

In the presence of bubbles on the heating surface, the heating area can be divided by two complementary zones: 1) $A_b$, the area that is affected by bubbles and 2) $A_c$, the convective heat transfer area, as designated in Figure 5. Each zone has the individual magnitude of heat transfer, i.e.:

$$q_c A_c = \alpha_c A_c \Delta T$$  \hspace{1cm} (4)

and:

$$q_b A_b = \alpha_b A_b \Delta T$$  \hspace{1cm} (5)

where the subscripts “c” and “b” stand for “convection” and “bubble affected” areas, respectively. Openly, the following equation is already established:

$$\frac{A}{A} = \frac{A_c}{A} + \frac{A_b}{A} = 1$$  \hspace{1cm} (6)

Figure 5. Dividing the boiling heat transfer area in the modeling.

Summing up Eqs. (4) and (5) yields:

$$Q = qA = q_c A_c + q_b A_b$$  \hspace{1cm} (7)

By combining Eqs. (4), (5) and (7) the heat flow rate can be calculated as:

$$Q = qA = \alpha_c A_c \Delta T + \alpha_b A_b \Delta T$$  \hspace{1cm} (8)

Assuming the affected areas by spherical bubbles are equal to the projected area of the bubbles, $A_b$ can be calculated by:

$$\frac{A_b}{A} = \frac{N}{A} \beta \frac{\pi \rho g \Delta \rho}{4}$$  \hspace{1cm} (9)

where $N/A$ is the nucleation site density. In the aforementioned equation, $\beta$ is the ratio of area of influence to projected area of bubble at departure. Judd and Hwang [15] have matched their predicted heat fluxes with experimental data and reported that $\beta = 1.8$. Some other investigators, such as Han and Griffith [16], postulated that $\beta = 4$. In this investigation, it is found that the parameter $\beta$ depends on the shape and the oscillating behavior of the departing bubble. Clift, Grace and Weber [17] proposed that the bubble shapes can be describes by the dimensionless Eötvös number. In addition, because the dimensionless Roshko number describes the oscillating nature of the rising bubbles, here it is postulated that the parameter $\beta$ should be a function of both $Ro$ and $E\ddot{o}$. By regression analysis it is found that:

$$\beta = 22 \sqrt{Ro.E\ddot{o}}$$  \hspace{1cm} (10)

where:

$$E\ddot{o} = \frac{(\rho_l - \rho_v) \rho g \Delta \rho^2}{\sigma}$$  \hspace{1cm} (11)

and:

$$Ro = \frac{k \Delta \rho}{\mu \rho_v}$$  \hspace{1cm} (12)

Figure 6 describes the relation between the aforementioned dimensionless groups. Combining Eqs. (8) and (9) yields:

$$\alpha = \alpha_c + (\alpha_b - \alpha_c) \beta \frac{N}{A} \frac{n \rho g \Delta \rho^2}{4}$$  \hspace{1cm} (13)

where $\alpha$ is the total heat transfer coefficient, $\alpha_c$ is the convective heat transfer coefficient and $\alpha_b$ is the heat transfer coefficient in the bubble affected area (all in W m$^{-2}$ K$^{-1}$).

When a bubble is develops on a heating surface, the heat transfer coefficient through the bubble stem, $\alpha_b$ can be predicted by the correlation proposed by Mikic and Rohsenow [18]. The heat transfer
mechanism is substantiated to be transient conduction around nucleation sites. The heat transfer coefficient is calculated by:

\[
\alpha_c = 2\pi k \rho C_p f
\]  

(14)

**Figure 6. The ratio of area of influence to projected area of bubble at departure as a function of the products of Roshko number and Eötvös number.**

The convective heat transfer coefficient, \(\alpha_c\), can be calculated by the equation proposed by Churchill and Bernstein [19], which is applicable to forced convection around the horizontal cylinders:

\[
\overline{\text{Nu}}_c = 0.3 + 0.62 \frac{Re_c^{1/2} Pr^{1/3}}{1 + (0.4 \, Pr)^{2/3} \left[ 1 + \frac{Re_c}{282000} \right]^{5/8} \left[ 1 - \frac{Re_c}{282000} \right]^{4/5}}
\]  

(15)

where the dimensionless Reynolds number is calculated based on the upward terminal velocity of bubbles, \(u_t\). The upward terminal velocity can be calculated by:

\[
u_t = \frac{4}{3} g \frac{\bar{d} \rho_l - \rho}{C_s \rho}
\]  

(16)

and the drag coefficient, \(C_d\), is already calculated by Ishii and Zuber [20]:

\[
C_d = \frac{24}{Re_d} \left( 1 + 0.1 Re_d^{0.75} \right)
\]  

(17)

Note that to find the terminal velocity, Eqs. (16) and (17) should be calculated iteratively, because the Reynolds number is already a function of the terminal velocity.

To predict the bubble departing diameter, many correlations have been compared to experimental data. It is found that the Stephan correlation [21] has the best agreement with experimental data with average absolute error of 14% for the entire systems. The Stephan correlation [21] is an experimental correlation with the following mathematical form:

\[
\bar{d} = 0.25 \left[ 1 + \left( \frac{Ja}{Pr} \right)^2 \frac{100000}{Ar} \right]^{-0.5} \sqrt{\frac{2 \sigma}{g (\rho - \rho)}}
\]  

(18)

To predict the nucleation site density, the correlation proposed by Xiao, Jiang, Zheng, Chen and Liu [22] is recommended:

\[
N = 7.8125e-29 (1 - \cos \varphi) R_{c,min}^{-6}
\]  

(19)

In the aforementioned correlation, the minimum cavity radius is calculated by:

\[
R_{c,min} = \frac{\delta}{c_1} \left[ 1 - \frac{\theta_c}{\theta_w} - \sqrt{\left( 1 - \frac{\theta_c}{\theta_w} \right)^2 - \frac{4 \zeta c_2}{\delta \theta_w}} \right]
\]  

(20)

where:

\[
\zeta = \frac{2 \sigma T_{sat}}{\rho_v h_v}
\]  

(21)

\[
c_1 = \frac{1 + \cos \varphi}{\sin \varphi}
\]  

(22)

\[
c_2 = 1 + \cos \varphi
\]  

(23)

where \(\varphi\) is the contact angle of the fluid and the heater material. The boundary layer thickness can be calculated by dividing the liquid thermal conductivity to the natural convection heat transfer coefficient:

\[
\delta = \frac{k_l}{\alpha_{NC}}
\]  

(24)

To calculate the bubble departing frequency, the experimental correlation proposed by Zuber [23] is recommended:

\[
\bar{f} = 0.59 \sqrt{g (\rho - \rho) \left( \frac{1}{\rho} \right)^{0.25}}
\]  

(25)

Note that in the modeling of the present data, the experimental values of nucleation site density, bubble departing frequency and diameter are used.

**MODEL VALIDATION**

The performance of the new model is compared to experimental data, as presented in Figure 7. It is found that 95% of the data points are matching within ±11% absolute average error with experimental data. Note that the value of ±11% is calculated as the max-
imum expected uncertainty. To revalidate the new model, an independent dataset [14] have been compared to the new model as presented in Figure 8. It is shown that about 90% of the data points are matching within ±20% absolute average error with experimental data. It is important to note that the new model is very sensitive to three key parameter including nucleation site density, bubble departing frequency and diameter. In evaluating the performance of the new model by the independent dataset [14], the three aforementioned parameters were estimated by existing correlation introduced by Eqs. (18), (19) and (25). In the mentioned reference [14], these values are not reported.

Figure 7. The predicted values of heat flux versus the experimental values of present study.

Figure 8. The predicted values of heat flux versus the experimental values from independent investigation.

CONCLUSION

In this investigation, saturated nucleate pool boiling heat transfer was studied experimentally for two boiling liquids - water and ethanol. Several heating elements were made and tested with different metals including copper, aluminum, brass and SS316 with various degrees of roughness. The measured data includes boiling heat transfer coefficient, nucleation site density, bubble departing diameter, as well as bubble departing frequency. The experimental heat fluxes were limited to 100 kW m$^{-2}$ to be able to measure the visual information of the boiling phenomenon. It was found that bubble departure frequency, diameter and nucleation site density are three key-parameters in determining the boiling heat transfer coefficient. Furthermore, the ratio of area of influence to projected area of bubble at departure can be correlated to the products of two dimensionless groups, Roshko and Eötvös numbers. The Roshko number describes the oscillating nature of bubble dynamics, while the Eötvös number characterizes the shape of bubbles.

It was shown that by dividing the heating surface to two complementary areas, one is directly influenced by bubbles and the other is free from bubbles effects; the boiling heat transfer is predictable. A new semi-empirical model was proposed to predict the boiling heat transfer coefficient.

Acknowledgment

The author is thankful to department of chemical engineering, college of chemistry and chemical engineering, Mahshahr Branch, Islamic Azad University, Mahshahr, Iran for financial support of research project entitled “Pool boiling heat transfer in pure liquids”.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$A$</td>
<td>area, m$^2$</td>
</tr>
<tr>
<td>$Ar$</td>
<td>Archimedes number</td>
</tr>
<tr>
<td>$b$</td>
<td>constant (see Mostinski correlation)</td>
</tr>
<tr>
<td>$c_1$</td>
<td>constant, see Eq.(22)</td>
</tr>
<tr>
<td>$c_2$</td>
<td>constant, see Eq.(23)</td>
</tr>
<tr>
<td>$C_d$</td>
<td>drag coefficient</td>
</tr>
<tr>
<td>$C$</td>
<td>heat capacity, J kg$^{-1}$ K$^{-1}$</td>
</tr>
<tr>
<td>$C_m$</td>
<td>constant, see Rohsenow correlation</td>
</tr>
<tr>
<td>$d$</td>
<td>bubble diameter, m</td>
</tr>
<tr>
<td>$Eö$</td>
<td>Eötvös number</td>
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<tr>
<td>$f$</td>
<td>bubble departing frequency, Hz</td>
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<td>$F_p$</td>
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<td>see Gorenflo correlation</td>
</tr>
<tr>
<td>$g$</td>
<td>acceleration of gravity, N kg$^{-1}$</td>
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</table>
**Greek symbols**

- $\alpha$: heat transfer coefficient, W m$^{-2}$ K$^{-1}$
- $\alpha_t$: thermal diffusivity, m$^2$ s$^{-1}$
- $\beta$: the ratio of area of influence to projected area of bubble at departure
- $\delta$: boundary layer thickness, m
- $\zeta$: see Eq. (21)
- $\theta_c$: D-value of $T_c$-$T_a$
- $\theta_w$: D-value of $T_w$-$T_a$
- $\rho$: density, kg m$^{-3}$
- $\sigma$: surface tension, N m$^{-1}$
- $\phi$: contact angle - see Eq. (19), (22) and (23)

**REFERENCES**

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3. I.L. Mostinski, Teploenergetika 4 (1963) 66

Ključne reči: indukovana prinudna konvekcija, ključanje zasičene tečnosti, hrapavost površine, koeficijent prenosa toplote.