Pressure drop and heat transfer characteristics of boiling nitrogen in square pipe flow

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Abstract

Pressure drop and forced convection heat transfer were studied in the boiling nitrogen flow in a horizontal square pipe with a side of 12 mm at inlet pressure between 0.1 and 0.15 MPa with a mass flux between 70 and 2000 kg/m\textsuperscript{2}-s and with a heat flux of 5, 10 and 20 kW/m\textsuperscript{2}. Accordingly, the flow and heat transfer mechanisms specific to square pipe were elucidated, and the applicability to cryogenic fluids of pressure drop and heat transfer models originally proposed for room temperature fluids was clarified.

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Peer-review under responsibility of the organizing committee of ICEC 25–ICMC 2014

Keywords: pressure drop; heat transfer; vapor-liquid two-phase flow; liquid nitrogen; void meter; flow pattern; square pipe flow

1. Introduction

JAXA is moving forward with the development of the technology for hypersonic pre-cooled turbojet engines that will be fueled by liquid hydrogen. Because the liquid hydrogen is in a forced convection boiling heat transfer state as it undergoes heat transfer with high-temperature air in the air precooler during high-speed flight, pressure drop and heat transfer performance are important from a design standpoint. Pressure drop and heat transfer equations have

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been available using room temperature fluids, such as water, primarily for flows in circular pipe. However, empirical research on pressure drop and heat transfer in the cryogenic boiling flow is insufficient with regards to a) the applicability of the conventional equations to cryogenic fluids, and b) the applicability to pipe cross-sections that are not circular. In the present paper, boiling nitrogen two-phase flow patterns in a horizontal square pipe are observed in terms of both visualization and void fraction measurement. Based on the void fraction measurement results, comparison is undertaken between experimental pressure drop results and analytical (calculation) results using the conventionally proposed correlations between void fraction and thermal equilibrium quality (referred to hereafter as quality). Differences are also clarified between heat transfer coefficients stemming from differences in quality and flow pattern, as well as differences in heat transfer coefficients among the top, side and bottom of the heat transfer pipe. Furthermore, evaluation is conducted on the models for pressure drop and heat transfer.

2. Experimental apparatus and method

Fig. 1 presents an overview of the test apparatus, consisting mainly of a run tank for the holding of liquid nitrogen, a test section and a catch tank [1]. The test section is comprised of a horizontal square pipe, a visualization section made of lupilon and a set of void meters. The run tank is pressurized with helium gas, such that liquid nitrogen flows into the test section. A capacitance-type liquid level meter is installed in the run tank. The flow velocity in the test section is calculated by measuring the change in the liquid level during the flow test. Pressure drop (ΔP) measurement length (L) is 550 mm, located at 200 mm downstream from the initial heating point.

The heat transfer pipe (straight) is made of phosphorus deoxidized copper, with a side of 12 mm, a wall thickness of 1.5 mm and a heated length of 800 mm. Nichrome wire is wound around the outside of the pipe and affixed.

Six silicon diode temperature sensors (T1~T6) with an accuracy of ±0.022 K at 77 K are attached on the pipe outer wall surface to measure the heat transfer coefficient. The bulk temperature (Tbulk) and quality (ξ) were calculated from the temperature (Twp) measured at a point 125 mm upstream from the heat transfer section, the upstream pressure (Pwp) and the amount of supplied heat until the point of outer wall temperature measurement. The inner wall temperature (Twall) was evaluated analytically from the outer wall temperature. In the present paper, the local heat transfer coefficient (h) is evaluated for the pipe side (T5: 560 mm downstream from the initial heating point), top (T1: 580 mm) and bottom (T2: 580 mm). In the case of heat flux q, h = q / (Twall - Tbulk).

Flow patterns were observed using a high-speed video camera positioned in the visualization section downstream from the heat transfer pipe. In the segment where pressure drop and heat transfer coefficient were measured, flow patterns were found to contain smaller void fractions than in the visualization section. Two varieties of capacitance type void meters were used, one having its flat plate electrodes vertically opposed (vertical type) and the other having the electrodes horizontally opposed (horizontal type), together with an LCR meter for void fraction measurement. The electrode dimensions are 12×55 mm. Void meter calibration was performed in liquid nitrogen at having the electrodes horizontally opposed (horizontal type), together with an LCR meter for void fraction.
3. Experimental results and discussion

3.1. Flow pattern and void fraction measurement

Six flow patterns were observed during experiments: bubbly, plug, slug, slug-annular, wavy-annular and wavy flows. Slug-annular flow, as suggested by the name, is a pattern incorporating both slug and annular aspects. Fig. 2 shows flow pattern images taken by the high-speed camera, together with time-wise changes in the measured capacitance ratio. The vertical axis indicates the ratio of capacitance during two-phase flow \( C_{TP} \) to capacitance in liquid nitrogen \( C_{L} \). For plug and slug flows, respectively, the time average void fraction \( \bar{\alpha} \) is 12\% and 42\%. Fig. 3 presents the relationships between flow pattern and superficial velocities for the vapor phase \( j_G \) and the liquid phase \( j_L \). In the region where \( j_L \) is large, it becomes difficult for bubbles to join with each other, instead flowing separately and producing bubbly flow. Where \( j_G \) is large, the vapor phase component flows more easily at the pipe center, and the flow shifts from plug or slug to annular. Note also that, in the region where \( j_L \) is small, the flow becomes wavy. It is thought that the effect of gravity is more pronounced in this region, inducing a separated flow in which the vapor phase is at the top of the pipe and the liquid phase at the bottom. While the boundary between bubbly flow and plug or slug flow is clear in the case of a square pipe, this interface becomes ambiguous in a circular pipe with a diameter of 15 mm [1]. The reason for this is considered to be the higher concentration of bubbles at the top of the pipe, such that bubbles tend to join up and become bigger.

The relationship between the void fraction (as measured by void meters) and quality is shown in Fig. 4. The solid lines correspond to the homogeneous flow model (slip ratio \( s = 1 \)), the separation flow models for slip ratios as proposed separately by Winterton [3] and Khalil [4], and the Butterworth’s model [5] with values calculated using Eq. (1) below. The portion in which quality is negative is where subcooled boiling occurs. Table 1 lists the relevant equations for slip ratio and void fraction model. Liquid density, vapor density and the Lockhart-Martinelli parameter are represented by \( \rho_L \), \( \rho_G \) and \( X_{tt} \), respectively. In the region of high quality, the separation flow model tends to show better agreement with the actual values than the homogeneous flow model.

\[
\alpha = \frac{x}{x + s \left( \frac{\rho_G}{\rho_L} \right) \left[ 1 - x \right]}
\]

(1)

Table 1. Slip ratio and void fraction model.

<table>
<thead>
<tr>
<th>Model</th>
<th>Equation</th>
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<tbody>
<tr>
<td>Winterton</td>
<td>( s = 0.93 \left( \frac{\rho_L}{\rho_G} \right)^{0.11} + 0.07 \left( \frac{\rho_L}{\rho_G} \right)^{0.561} )</td>
</tr>
<tr>
<td>Khalil</td>
<td>( s = 3 + 27.3 x )</td>
</tr>
<tr>
<td>Butterworth</td>
<td>( \alpha = \left( 1 + 0.28 X_{tt}^{0.71} \right)^{-1} )</td>
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Fig. 3. Superficial liquid velocity versus superficial gas velocity. Fig. 4. Quality versus void fraction measured by void meters.
results obtained for pressure drop at heat flux of 5, 10 and 20 kW/m². However, in the measurement interval calculation results for the Winterton’s and Khalil’s slip ratios, which are not shown in the present paper, also enabled values in terms of the absolute average of deviation, the Butterworth’s model delivered the best results. The for most of the experimental results in nearly all of the flow patterns. Considering the experimental and calculated separation flow model [1]. Here, \( \Delta P \) is a pipe friction factor of the Blasius equation, the subscripts \( \text{mid} \) and \( \text{down} \) are the values corresponding to pressure drop measurement section \( P_{\text{mid}} \) and \( P_{\text{down}} \), and \( \mu_L \) and \( \mu_G \) are the viscosity of liquid and vapor. Fig. 5 shows experimentally obtained pressure drop at heat flux of 10 kW/m², together with total pressure drop, acceleration loss and friction loss for the homogeneous flow and separation flow models, calculated at the representative experimental conditions indicated in the figure. Fig. 6 and Fig. 7 indicate, respectively, calculated results obtained using the homogeneous flow model and Butterworth’s model, together with experimental results obtained for pressure drop at heat flux of 5, 10 and 20 kW/m². However, in the measurement interval where \( x \leq 0 \), calculation is performed using the Blasius equation. Here, let us briefly compare the four types of models. While the homogeneous flow model can be used to evaluate experimental results for low void fractions (bubbly and plug), experimental values are overestimated by more than 30% when pressure drop is large (high void fraction). Because the flow velocities for the liquid and vapor phases are assumed to be the same in the homogeneous flow model, the liquid phase velocity tends to be overestimated, leading in turn to overestimation of pressure drop. The calculation results for the Butterworth’s model, indicated in Fig. 7, are within ±30% agreement for most of the experimental results in nearly all of the flow patterns. Considering the experimental and calculated values in terms of the absolute average of deviation, the Butterworth’s model delivered the best results. The calculation results for the Winterton’s and Khalil’s slip ratios, which are not shown in the present paper, also enabled evaluation within ±30% of the experimental values in most cases, with the separation flow model providing better agreement with the experimental results.

### 3.2. Pressure drop

Pressure drop in a horizontal pipe is composed of acceleration loss (\( \Delta P_{\text{Acc}} \)) and friction loss (\( \Delta P_{\text{Fri}} \)). Using the homogeneous flow model (\( s = 1 \)), together with separation flow models that agree well with the void fraction obtained from measurement results (i.e., Winterton, Khalil and Butterworth) as shown in Fig. 4, pressure drop per unit length (\( \Delta P/L \)) was calculated. Table 2 presents the calculation equation used in the present paper for the separation flow model [1]. Here, \( \lambda_{\text{B}} \) is a pipe friction factor of the Blasius equation, the subscripts \( \text{mid} \) and \( \text{down} \) are the values corresponding to pressure drop measurement section \( P_{\text{mid}} \) and \( P_{\text{down}} \), and \( \mu_L \) and \( \mu_G \) are the viscosity of liquid and vapor. Fig. 5 shows experimentally obtained pressure drop at heat flux of 10 kW/m², together with total pressure drop, acceleration loss and friction loss for the homogeneous flow and separation flow models, calculated at the representative experimental conditions indicated in the figure. Fig. 6 and Fig. 7 indicate, respectively, calculated results obtained using the homogeneous flow model and Butterworth’s model, together with experimental results obtained for pressure drop at heat flux of 5, 10 and 20 kW/m². However, in the measurement interval where \( x \leq 0 \), calculation is performed using the Blasius equation. Here, let us briefly compare the four types of models. While the homogeneous flow model can be used to evaluate experimental results for low void fractions (bubbly and plug), experimental values are overestimated by more than 30% when pressure drop is large (high void fraction). Because the flow velocities for the liquid and vapor phases are assumed to be the same in the homogeneous flow model, the liquid phase velocity tends to be overestimated, leading in turn to overestimation of pressure drop. The calculation results for the Butterworth’s model, indicated in Fig. 7, are within ±30% agreement for most of the experimental results in nearly all of the flow patterns. Considering the experimental and calculated values in terms of the absolute average of deviation, the Butterworth’s model delivered the best results. The calculation results for the Winterton’s and Khalil’s slip ratios, which are not shown in the present paper, also enabled evaluation within ±30% of the experimental values in most cases, with the separation flow model providing better agreement with the experimental results.

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<td>Acceleration loss</td>
<td>( \frac{\Delta P_{\text{Acc}}}{L} = G^2 \left( \frac{1}{P_G} \left( \frac{x_{\text{mid}}}{x_{\text{down}}} \right)^2 \right) + \frac{1}{P_L} \left( \frac{1}{x_{\text{down}}} - \frac{1}{x_{\text{mid}}} \right) )</td>
</tr>
<tr>
<td>Friction loss</td>
<td>( \frac{\Delta P_{\text{Fri}}}{L} = \frac{\lambda_{\text{B}} G^2}{2 \mu_G} \left( \frac{x_{\text{mid}}}{x_{\text{down}}} - \frac{1}{X_a} \right) \int_{x_{\text{mid}}}^{x_{\text{down}}} \left( 1 - x \right)^3 \left( 1 + \frac{20}{X_a} + \frac{1}{X_a^2} \right) dx )</td>
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Fig. 5. Measured pressure drop versus mass flux at 10 kW/m². Fig. 6. Measured pressure drop versus calculated pressure drop.

With respect to the vertical type (V) and horizontal type (H) void meters, the latter appears to provide more accurate measurement of the actual void fraction. In the case of circular pipe as well, analytical results have been reported showing that a horizontal type void meter offers better measurement accuracy [6].
The heat transfer coefficients on the side of the square pipe for heat flux of $q = 5, 10$ and $20 \text{ kW/m}^2$ are presented in Fig. 8. In the region where mass flux is large, forced convection heat transfer in liquid phase is dominant; the heat transfer coefficient does not depend on heat flux, depending instead on the magnitude of mass flux. The high mass flux region was also characterized by almost no difference in terms of heat transfer coefficient among the top, side and bottom of the pipe. In the region of small mass flux, as designated by the arrows on the solid lines in the figure, boiling commenced and the heat transfer coefficient increased when the mass flux became less. When the mass flux decreased further, nucleate boiling heat transfer became dominant, and the heat transfer coefficient increased up to a certain point. The point of mass flux at which boiling commenced (the value of $q$ in the figure) become lower with reduced heat flux, while the amount of increase in the heat transfer coefficient rose with greater heat flux.

In Fig. 8, the nucleate boiling heat transfer coefficient for the pipe side at heat flux of $20 \text{ kW/m}^2$ is shown as a solid line, while the coefficients for the top and bottom are indicated as broken lines. At the pipe bottom, following the rise in the heat transfer coefficient to a certain level accompanying the reduction in mass flux, this is maintained through the low mass flux region. Nucleate boiling heat transfer becomes dominant; it is not dependent on the magnitude of mass flux, but depends instead on the magnitude of heat flux, with the heat transfer coefficient becoming constant. At the top of the pipe, while the rise in the heat transfer coefficient accompanying the reduction in mass flux exceeds that of the sides or bottom, this falls off when the mass flux continues to decline. That is, because the bubbles formed due to boiling are concentrated in the flow at the top of the pipe, heat transfer is thus promoted and the heat transfer coefficient rises. When mass flux falls further, the wall surface is subject to dry-out, and the heat transfer coefficient is reduced. The side exhibits similar behavior to the bottom, but the heat transfer coefficient starts to decline when dry-out occurs. Compared to the top, the liquid phase is present even in the low mass flux region. Because nucleate boiling heat transfer is maintained, the mass flux point where heat transfer coefficient decline starts to occur is low, and the rate of decline is therefore gentle.

The heat transfer coefficients for the pipe top, side and bottom were evaluated using the heat transfer correlations proposed for room temperature fluids. The Gungor-Winterton and Liu-Winterton equations for vapor-liquid two-phase heat transfer are shown in Table 3. Here, $h_{fp}$ is the two-phase heat transfer coefficient, $h_{con}$ is the forced convection heat transfer coefficient (Dittus-Boelter equation), $h_{nb}$ is the nucleate boiling heat transfer coefficient (Cooper equation), $Bo$ is the boiling number, $Pr$ is the liquid phase Prandtl number and $Re_L$ is the liquid phase Reynolds number. In the measurement point where $x \leq 0$, the flow patterns observed were liquid single-phase, bubbly and plug, and evaluation was possible to about $\pm 20\%$ using the Dittus-Boelter equation, regardless of the circumferential location of measurement. However, in the case of $x > 0$, the flow patterns observed with slug, annular and wavy, and results were obtained as follows. Comparison with the Gungor-Winterton equation is presented in Fig. 9. At quality of $x > 0.006$, evaluation was possible within $-20$ to $+30\%$ for most of the experimental values, regardless of the circumferential location of measurement. While the trend of the Lie-Winterton equation, shown in the Fig. 10, is to underestimate the experimental values as compared with the Gungor-Winterton equation, the difference is small even in the region where quality is low, such that evaluation was possible within $\pm 30\%$ for most of the experimental values.
Fig. 9. Quality versus heat transfer coefficient ratio (G-W eq.). Fig. 10. Quality versus heat transfer coefficient ratio (L-W eq.).

Table 3. Heat transfer correlation equation.

<table>
<thead>
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<tbody>
<tr>
<td>Gungor-Winterton</td>
<td>$h_p = E_{GW} \cdot h_{con}$, $E_{GW} = 1 + 3000B_0^{0.36} \times 1.12 \left(\frac{x}{1-x}\right)^{0.75} \left(\frac{f_L}{\rho_G}\right)^{0.41}$</td>
</tr>
<tr>
<td>Liu-Winterton</td>
<td>$h_p = \sqrt{(F_{LW} \cdot h_{con})^2 + (S_{LW} \cdot h_{boil})^2}$, $F_{LW} = 1 + x \cdot \left(\frac{\rho_L}{\rho_G} - 1\right)^{0.15}$, $S_{LW} = \frac{1}{1 + 0.055 \cdot F_{LW} \cdot 0.10 \cdot Re_G^{0.10}}$</td>
</tr>
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values. In terms of the absolute average of deviation between the experimental and calculated values, the Gungor-Winterton equation provided the best agreement with the experimental results.

4. Conclusion

Flow pattern visualization and void fraction measurement were undertaken for boiling nitrogen flow in a horizontal square pipe, and evaluation was conducted with respect to the Winterton’s and Khalil’s slip ratios, and the Butterworth’s model showing the relationship between void fraction and quality. Comparison of experimental results with calculated results for pressure drop using homogeneous flow and separation flow models indicated good agreement (within about ±30%) on the part of the Butterworth’s model. Also differences in heat transfer coefficients due to quality and flow pattern were elucidated, while the differences in heat transfer mechanisms and heat transfer coefficients at the top, side and bottom of the square pipe were clarified. Heat transfer coefficient comparison with the Liu-Winterton equation showed that the Gungor-Winterton equation provided good agreement (around -20 ~ +30%) with the experimental results.

References