2015

An Experimental Study of Subcooled Flow Boiling at Elevated Pressure in an Annular Flow Channel

Pervej Rahman
CUNY City College

How does access to this work benefit you? Let us know!
Follow this and additional works at: https://academicworks.cuny.edu/cc_etds_theses

Part of the Mechanical Engineering Commons

Recommended Citation
https://academicworks.cuny.edu/cc_etds_theses/700

This Thesis is brought to you for free and open access by the City College of New York at CUNY Academic Works. It has been accepted for inclusion in Master's Theses by an authorized administrator of CUNY Academic Works. For more information, please contact AcademicWorks@cuny.edu.
AN EXPERIMENTAL STUDY OF SUBCOOLED FLOW BOILING AT ELEVATED PRESSURE IN AN ANNULAR FLOW CHANNEL

THESIS
Submitted in partial fulfillment of the requirement for the degree
Master of Engineering (Mechanical)
at
The City College of New York
of the
City University of New York
by
Pervej Rahman
May, 2015
Approved:

______________________________
Professor Masahiro Kawaji, Thesis Advisor
Department of Mechanical Engineering

______________________________
Professor Feridun Delale, Chairman
Department of Mechanical Engineering
Abstract

An experimental setup was developed to study the region of subcooled flow boiling. Multiple studies were carried out to investigate the effects of liquid velocity, pressure, and temperature on the boiling heat transfer of subcooled fluid flowing through a heated annular channel. Water was used as the working fluid and principle of Ohmic heating was used to raise water temperature. The system pressure, heat flux, & mass flux ranged from 101 to 912 kPa, 19 to 155 kW/m², and 83 to 332 kg/m²-s, respectively. This report contains boiling curves, heat transfer coefficients of various studies and a description of the experimental setup.
Acknowledgement

First and foremost I would like to thank my advisor Professor Masahiro Kawaji of the Department of Mechanical Engineering for his guidance and patience. I would also like to thank Jorge Pulido who helped me with experimental setup and machining. Finally, I would like to give special thanks to Randy Samaroo who helped me with experiments and data processing.
## Table of Contents

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Abstract</td>
<td>1</td>
</tr>
<tr>
<td>Acknowledgement</td>
<td>2</td>
</tr>
<tr>
<td>List of figures</td>
<td>4</td>
</tr>
<tr>
<td>List of Tables</td>
<td>5</td>
</tr>
<tr>
<td>Nomenclature</td>
<td>6</td>
</tr>
<tr>
<td>1. Introduction</td>
<td>7</td>
</tr>
<tr>
<td>2. Review of subcooled flow boiling experiments</td>
<td>10</td>
</tr>
<tr>
<td>3. Experimental Setup</td>
<td>14</td>
</tr>
<tr>
<td>4. Experimental Procedure</td>
<td>24</td>
</tr>
<tr>
<td>5. Result &amp; Discussion</td>
<td>25</td>
</tr>
<tr>
<td>6. Conclusion</td>
<td>32</td>
</tr>
<tr>
<td>References</td>
<td>33</td>
</tr>
<tr>
<td>Appendix</td>
<td>34</td>
</tr>
</tbody>
</table>
# List of Figures

<table>
<thead>
<tr>
<th>Figure</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Typical Flow Boiling Curve</td>
<td>7</td>
</tr>
<tr>
<td>2</td>
<td>Enlarged sketch of a conventional machined metal surface</td>
<td>8</td>
</tr>
<tr>
<td>3</td>
<td>Bubble formation in partial nucleate boiling</td>
<td>8</td>
</tr>
<tr>
<td>4</td>
<td>Bubbles growing &amp; departing in fully developed nucleate boiling</td>
<td>9</td>
</tr>
<tr>
<td>5</td>
<td>Ohmic heating method</td>
<td>14</td>
</tr>
<tr>
<td>6</td>
<td>Design of Test Section</td>
<td>16</td>
</tr>
<tr>
<td>7</td>
<td>Spot welded thermocouples on the outside surface of the heater rod</td>
<td>17</td>
</tr>
<tr>
<td>8</td>
<td>Cut-away view of Chemsteel S935M gear pump</td>
<td>18</td>
</tr>
<tr>
<td>9</td>
<td>Oil-Free High-Vacuum Pump</td>
<td>19</td>
</tr>
<tr>
<td>10</td>
<td>SS Low-Pressure Proportional Relief Valve</td>
<td>20</td>
</tr>
<tr>
<td>11</td>
<td>Stainless Steel Heat Exchanger</td>
<td>20</td>
</tr>
<tr>
<td>12</td>
<td>DC power supply</td>
<td>21</td>
</tr>
<tr>
<td>13</td>
<td>Data acquisition &amp; Instrumentation components</td>
<td>22</td>
</tr>
<tr>
<td>14</td>
<td>Solid model design of frame</td>
<td>23</td>
</tr>
<tr>
<td>15</td>
<td>Schematic diagram of experimental apparatus</td>
<td>23</td>
</tr>
<tr>
<td>16</td>
<td>Conservation of energy for element dx</td>
<td>26</td>
</tr>
<tr>
<td>17</td>
<td>Comparison of subcooled flow boiling correlations with experimental data</td>
<td>28</td>
</tr>
<tr>
<td>18</td>
<td>Boiling curves for different pressure (0.5 gpm)</td>
<td>29</td>
</tr>
<tr>
<td>19</td>
<td>Comparison of heat transfer coefficients for various system pressure (0.5 gpm)</td>
<td>31</td>
</tr>
</tbody>
</table>
## List of Tables

<table>
<thead>
<tr>
<th>Table</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Some Important Correlations for Fully Developed Heat Transfer in Subcooled Flow Boiling.</td>
<td>13</td>
</tr>
<tr>
<td>2</td>
<td>304 stainless steel properties/geometry</td>
<td>15</td>
</tr>
<tr>
<td>3</td>
<td>Operating capacity of pump</td>
<td>19</td>
</tr>
<tr>
<td>4</td>
<td>Power supply specifics</td>
<td>26</td>
</tr>
</tbody>
</table>
## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Area (m$^2$)</td>
</tr>
<tr>
<td>$A_c$</td>
<td>Area of cross section (m$^2$)</td>
</tr>
<tr>
<td>$f$</td>
<td>Moody’s friction factor</td>
</tr>
<tr>
<td>$h$</td>
<td>Heat transfer coefficient (W/m$^2$-K)</td>
</tr>
<tr>
<td>$I_D$</td>
<td>Inner Diameter of heater rod (m)</td>
</tr>
<tr>
<td>$k$</td>
<td>Thermal conductivity (W/m-K)</td>
</tr>
<tr>
<td>$L$</td>
<td>Length of test section (m)</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>Mass flow rate of water (kg/s)</td>
</tr>
<tr>
<td>$Nu_D$</td>
<td>Nusselt Number</td>
</tr>
<tr>
<td>$O_D$</td>
<td>Outer Diameter of heater rod (m)</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>$p$</td>
<td>Pressure (kPa)</td>
</tr>
<tr>
<td>$q'$</td>
<td>Heat flux (W/m$^2$)</td>
</tr>
<tr>
<td>$R$</td>
<td>Resistance (ohm)</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$R_m$</td>
<td>Resistivity of material (ohm-m)</td>
</tr>
<tr>
<td>$T_b$</td>
<td>Bulk fluid temperature (°C)</td>
</tr>
<tr>
<td>$T_w$</td>
<td>Heater rod wall temperature (°C)</td>
</tr>
<tr>
<td>$T_{sat}$</td>
<td>Liquid saturation temperature (°C)</td>
</tr>
<tr>
<td>$\Delta T_{sat}$</td>
<td>Wall superheat ($T_w - T_{sat}$) (°C)</td>
</tr>
<tr>
<td>$v$</td>
<td>Liquid velocity (m/s)</td>
</tr>
<tr>
<td>$\dot{V}$</td>
<td>Volume flow rate of water (m$^3$/s)</td>
</tr>
</tbody>
</table>

## Greek Letters

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\mu$</td>
<td>Dynamic viscosity of water (kg/m-s)</td>
</tr>
<tr>
<td>$\nu$</td>
<td>Kinematic viscosity of water (m$^2$/s)</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density of water (kg/m$^3$)</td>
</tr>
</tbody>
</table>
1. Introduction

Heat transfer from a heated metal surface to a flowing liquid has been the subject of many studies. The need for high rates of heat transfer in nuclear engineering, thermal engineering and refrigeration added to the appeal of fully understanding the mechanics of boiling heat transfer. Subcooled flow boiling process increases heat transfer rate significantly compared to single-phase convection. To understand the phenomena behind this process, we need to take a look at a typical flow boiling curve. In figure 1 below, the heat flux is plotted against the difference between the heated wall and the fluid saturation temperatures.

![Figure 1: Typical Flow Boiling Curve](image)

The boiling curve is divided into 5 regions (a-e). Each region has its own heat transfer mechanism. Only heat transfer regions (a-c) are discussed briefly below because other regions are not the focus of the experimental study.
**Region a:** The minimum condition for boiling requires the temperature of the heated wall surface to exceed the saturation temperature of the fluid. The wall temperature is below saturation temperature and cannot initiate bubble formation and development. Heat from the wall is removed by single-phase forced convection.

**Region b:** As the wall temperature increases, the first bubbles begin growing at isolated nucleation sites on the heated wall. Impurities on the wall surface such as pits and crevices allow non-dissolved gases and vapor to gather and form bubbles (figure 2). As the bubbles form and collapses on the heated wall they transfer latent heat, as well as create increased turbulence which increases the rate of heat transfer (figure 3). Heat transfer in this region consists of single-phase forced convection and nucleate boiling. This region is known as partial nucleate boiling region.

![Figure 2: Enlarged sketch of a conventional machined metal surface](image)

![Figure 3: Bubble formation in partial nucleate boiling](image)
**Region c:** After a further increase in wall temperature, bubbles form at such high rates and at such a large number. These bubbles depart from the heated wall surface with large amounts of latent heat and generate greater agitation and turbulence near the heated wall surface (figure 4). This region is known as a fully developed nucleate boiling region. A moderate increase in wall temperature significantly increases the rate of heat transfer in this region.

![Figure 4: Bubbles growing and departing in fully developed nucleate boiling](image)

The optimization of heat transfer rate in nuclear engineering, thermal engineering and refrigeration has a common goal of maximizing thermal efficiency, power densities and minimizing costs. To meet this goal, the operation of heat transfer equipment in subcooled boiling region is an important condition to be investigated. A significant increase in the heat transfer coefficient is obtained in subcooled boiling region when compared to the single-phase convection values. Increases in pressure drop and void fraction found in the fully developed boiling region are not present in subcooled boiling. Therefore, the proper knowledge of heat transfer mechanisms in the subcooled boiling region is very important in the enhancement of thermal systems and components used in nuclear, thermal and refrigeration fields.
2. Review of subcooled flow boiling experiments

In planning the experimental work, some of the past experiments on subcooled flow boiling have been reviewed. Although the number of papers reviewed is small, it is important to establish the parameters to be measured in the experiments. The papers listed in this section were surveyed to determine the parameters that have been measured by previous researchers.

Del Valle and Kenning [2] performed subcooled flow nucleate boiling experiments with water at atmospheric pressure. The test surfaces were stainless-steel plates, heated by direct current over an area of 150 x 10 mm, set into one side of a vertical flow channel of rectangular cross-section 12 x 5 mm. Bubble size, frequency, and the distribution of nucleation sites were measured at 1.7 m/s inlet velocity, 84 K subcooling, 0.08 mm wall thickness and heat fluxes corresponding to 70-95% of the critical heat flux (CHF). They found that the total population of nucleation sites increased with increasing wall superheat, the startup of new sites deactivated many of the sites active at lower superheats.

Hino and Ueda [3] performed subcooled flow boiling experiments with fluorocarbon R-113 liquid at 147 kPa. The test section was a vertically arranged concentric annulus with a stainless steel inner heater rod of 8 mm O.D., 0.5 mm wall thickness and 400 mm length. It was observed that the incipient boiling superheats measured were little affected by mass velocity and liquid subcooling. Increasing heat flux up to the critical heat flux increased, the bubble density on the heated rod and remarkably large coalescent bubbles appeared periodically near the heating section outlet.

Lin et al. [4] performed subcooled flow boiling experiments with water at 6.9-15.52 MPa pressure. The vertically oriented test section was directly heated by DC current and made of 15.88 mm O.D. and 1.24 mm thick wall Inconel 600 tubing with a uniformly heated length of 1.0
m. It was observed that the heat transfer mechanism has been influenced by a very thin liquid layer, which is trapped between the heated surface and the vapor blankets.

Basu et al. [5] conducted an experiment for vertical up flow using a copper plate and a Zircalloy-4 nine-rod bundle geometry with water as the test fluid and investigated the onset of nucleate boiling (ONB) point both by visual observations as well as temperature and heat flux data. Basu et al’s study showed that the nucleation site density depends only on contact angle and wall superheat. From the data obtained it was found that for the boiling inception, heat flux and wall superheat needed are dependent on flow rate, liquid subcooling and contact angle.

Mizutani et al. [6] conducted an experiment to observe the two phase flow patterns of vertical up flow of air-water mixture using four by four square lattice rod bundles made of an acrylic channel box of 68 mm width and transparent FEP (fluorinated ethylene propylene) rods of 12 mm in diameter. High speed video camera and a fiber scope were used to observe whole flow patterns in the bundle and local flow patterns in sub channels. The ranges of gas and liquid volume fluxes were 0.06 to 8.85 m/s for the gas and 0.1 to 2.0 m/s for the liquid, which covered typical two-phase flow patterns appearing in a fuel bundle of a boiling water reactor. The results indicated that the region of slug flow in the gas-liquid flow pattern diagram is so narrow that it can be regarded as a boundary between bubbly and churn flows.

Experiments have also been performed that provide velocity fields under subcooled flow boiling conditions. Roy et al. [7] performed turbulent subcooled flow boiling experiments in a vertical annular channel of 11.37 mm gap width and heated inner wall using refrigerant R-113 as the working fluid. The experiments were performed at pressures of 219 kPa and 253 kPa, wall heat fluxes between 30-80 kW/m², wall temperatures of 70-95 °C, inlet fluid temperatures of 35-43 °C, and Reynolds numbers of 24,400 and 33,400. The velocity and temperature field
measurements were taken using a three sensor hot film anemometer and a thermocouple embedded in the flow. From the experiments, it was shown that the liquid turbulence structure is significantly affected in the vicinity of a bubbly sublayer.

Estrada-Perez et al. [8] conducted two-phase particle tracking velocimetry (PTV) experiments in a vertical rectangular channel with a heated wall using refrigerant HFE-301 as the working fluid. Reynolds numbers ranged between 3,309-16,549, wall heat flux ranged between 1-64 kW/m², and the channel cross-sectional area was 8.7x7.6 mm². The data obtained in this experiment detailed the effect of wall heat flux on various turbulence parameters.

Warrier et al. [9] conducted low pressure subcooled flow boiling experiments by heating upward flowing water in a rectangular channel. The test section was 1.83 m long, of which the heated section length was 0.30 m. The experiments were conducted at constant pressure (1.03 bar), mass flux ranged between 235-684 kg/m²s, heat flux ranged from 9.6-96.3 W/cm², inlet liquid subcooling ranged from 7-46.5 °C, and the channel cross-sectional area was 4.2x4.128 cm². A high speed CCD camera was used to record the bubble collapse in the bulk subcooled liquid. Based on the analyses of these digitized images, the bubble collapse rate and the associated heat transfer rate were determined. The experimental data were used to correlate the bubble collapse rates and interfacial heat transfer rate. These correlations are functions of bubble Reynolds number, liquid Prandtl number, Jacob number, and Fourier number.

Zeitoun & Shoukri [10] obtained axial void fraction profiles in subcooled flow boiling using gamma attenuation techniques. The test section was a vertically arranged concentric annulus. The outer tube was 25.4 mm I.D. made of plexi-glass tube and stainless steel inner heater tube of 12.7 mm O.D, 0.25 mm wall thickness and 30.6 cm length. The inner tube was connected to a 55 kW DC power supply. The experiments were conducted at different pressures.
(1.1-2.11 bar), mass flux ranged between 161.2-1115.5 kg/m$^2$s, heat flux ranged from 213.6-1164 kW/m$^2$, and inlet liquid subcooling ranged from 13.1-44.3 °C. A single beam gamma densitometer was used for void fraction measurements. The results indicated that the rate of increase of void fraction along the test section was very small in the upstream region resulting in almost flat void profile with void fraction typically in the range of 2-9%.

Subcooled flow boiling has been studied for water by earlier investigators in nuclear reactor applications. Table 1 provides summary of the important correlations developed by earlier investigators for heat transfer in subcooled flow boiling.

Table 1: Some Important Correlations for Fully Developed Heat Transfer in Subcooled Flow Boiling$^{[11]}$.

<table>
<thead>
<tr>
<th>Investigator/Year</th>
<th>Fluid</th>
<th>Correlation</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>McAdams et al. (1949)</td>
<td>Water</td>
<td>$\dot{q} = C(\Delta T_{sat})^{3.86}$</td>
<td>The constant ‘C’ depends on the dissolved air content.</td>
</tr>
<tr>
<td>Jen and Lottes (1951)</td>
<td>Water</td>
<td>$\dot{q} = \left(\frac{\Delta T_{sat}}{25e^{(p/62)}}\right)^{1/0.25}$</td>
<td>q”-MW/m$^2$ P-bar</td>
</tr>
<tr>
<td>Thom et al. (1965)</td>
<td>Water</td>
<td>$\dot{q} = \left(\frac{\Delta T_{sat}}{22.65e^{(p/87)}}\right)^{1/0.25}$</td>
<td>T-K</td>
</tr>
</tbody>
</table>

As seen in the literature reviews above, most experiments have been conducted with water using an electrically heated rod in the middle of an annular channel at atmospheric pressure. In the applications such as nuclear engineering, refrigeration and thermal engineering, depending on the operating conditions, system pressure is the frequently altered parameter to change the work conditions of those facilities. In our work, we have constructed a pressurized subcooled flow boiling loop to observe effects of system pressure on subcooled flow boiling.
3. Experimental setup

Since this was a temperature and pressure sensitive experiment, it was essential that all components coming in contact with the working fluid would be able to meet the desired operating temperature and pressure above 200 °C and 10 bar, respectively. Proper selection of each device was necessary to obtain results in an efficient manner.

Test Section:

The main focus of the apparatus was the annular test section region as shown in Fig. 5 which would undergo Ohmic heating by means of a power supply. In Ohmic heating, electrical current is passed through a material to generate heat. The amount of heat generated is governed by the resistivity of the material. Electrical insulation between the inner and outer tubes was included in the test section design to restrict the current flow to outer tube. An inner heater rod was made of a tube instead of pipe because tube walls have enough resistivity to be able to generate adequate heat for experiment.

Figure 5: Ohmic heating method
To have a satisfactory setup, the resistance of the metal tube must be high with a small wall thickness. The following formula for electrical resistance was used to select an appropriate material and size:

\[ R = \frac{R_m L}{A_c} \]  

(1)

where:

- \( R \) = electrical resistance (\( \Omega \))
- \( R_m \) = resistivity of material (\( \Omega \cdot \text{m} \))
- \( L \) = length of test section (m)
- \( A_c \) = cross-sectional area (m\(^2\))

To optimize the result for higher electrical resistance, it was necessary to utilize a material with high resistivity, long length, and small cross-sectional area. This material will heat up and transfer the thermal energy to the fluid uniformly. Choosing between 304 stainless steel and inconel 600 was simple, as the first was commercially available, cheaper, and in a small cross-sectional area. The physical properties and geometry of the 304 stainless steel pipe are shown in Table 2 and these physical properties were determined to be adequate for the experiment.

### Table 2: 304 stainless steel properties/geometry

<table>
<thead>
<tr>
<th>Property/Geometry</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electrical resistivity</td>
<td>7.496*10^-7 (( \Omega \cdot \text{m} ))</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>15 (W/m( \cdot )( ^\circ )C)</td>
</tr>
<tr>
<td>Outside Radius</td>
<td>0.00635 (m)</td>
</tr>
<tr>
<td>Inside Radius</td>
<td>0.003302 (m)</td>
</tr>
<tr>
<td>Length</td>
<td>1.3716 (m)</td>
</tr>
</tbody>
</table>
Calculating the resistance of the tube we have:

\[ R = \frac{7.496 \times 10 - 7(ohm - m) \times 1.37(m)}{\pi(0.00635^2 - 0.00330^2)(m^2)} = 0.011\ (ohm) \]

The 304 stainless steel tube was purchased from McMaster-Carr in a 6 feet length and cut to size. The resistance was checked with a multimeter. At both ends of the tube, flanges are attached to enable connection of DC power cables as well as piping connections to feed water supply and tube wall temperature measurement system. The stainless steel heater rod of 12.7 mm O.D. is housed in a stainless steel tubing of 25.4 mm I.D. The test section is electrically insulated from the rest of the system with PTFE insulating gaskets between the flanges and fiberglass jacket around the bolts, as shown in figure 6. A fitted and sealed borosilicate sight glass was added to the test section for flow visualization experiment.

![Figure 6: Design of test section](image-url)
Additionally, two T-type thermocouples were spot welded on the outside surface of heater rod for wall temperature measurement. These thermocouples were placed 180° apart at a distance of 825 mm from the inlet and 190 mm below the outlet of the test section. See figure 7 below for spot welded thermocouples on heater rod outer surface. The average of temperatures measured by the two thermocouples was heater rod’s local wall temperature.

Figure 7: Spot welded thermocouples on the outside surface of the heater rod

Hydraulic Equipment

Most flows encountered in engineering practice are turbulent. This is particularly true for pipe flows, so it was essential to perform experiment in turbulent flow regime. The transition from laminar to turbulent flow occurs at a Reynolds number of 2,300 during internal flow which exists in this setup. As per reviewed literatures, most turbulent flow experiments were carried out above Reynolds number 20,000. Assuming a Reynolds number of 21,000 and the fluid water at 200°C, a velocity value to work around can be calculated using the formula for the Reynolds number shown below:

\[ Re = \frac{\rho v D}{\mu} \]  \hspace{1cm} (2)

Rearranging gives:

\[ v = \frac{\mu Re}{\rho D} = \frac{(21000)(0.1336 \times 10^{-3}\ \frac{kg}{s-m})}{(864.7 \frac{kg}{m^3})(0.0127 \ m)} = 0.255 \ m/s \]
For the pipe diameter given this velocity coincides with a flow rate of:

\[
\dot{V} = vL = \left(0.255 \frac{m}{s}\right)\left(\frac{\pi}{4} \frac{0.0127^2 \, m^2}{2}\right) = 3.24 \times 10^{-5} \, m^3/s
\]

Converting to gallons per minute:

\[
\dot{\nu} = \left(3.24 \times 10^{-5} \, m^3/s\right)\left(\frac{1 \, gpm}{6.309 \times 10^{-5} \, m^3/s}\right) = 0.51 \, gpm
\]

The next step was calculating the loop pressure drop to choose a pump. The Darcy-Weisbach equation was used to calculate the estimated pressure drop inside the flow boiling loop.

\[
\Delta p = f \frac{L \rho v^2}{2D} \quad (3)
\]

Appendix- B includes detailed pressure drop calculations of the loop. The total pressure drop of the loop was calculated to be 38.2 kPa. With this information the selection of the pump can be made. A gear pump shown in figure 8 with 23 GPM flow rate was chosen due to its tight seal to avoid leaks, maximum operating temperature of 230 °C, and cost. The gear pump was powered by a WEG Electric Corp. 3-phase, 3,600 rpm, 5-horsepower motor.

![Figure 8: Cut-away view of Chemsteel S935M gear pump](image)

To control the flow rate in the flow boiling loop, a variable frequency drive (VFD) was interlocked with the pump motor. VFD was used to vary speed of the pump motor. As per Pump affinity laws, flow or volume varies linearly with speed. If speed decreases by 50%, flow
decreases by 50%. Power or energy consumption varies as a cube of the speed. If speed decreases by 50%, power consumption decreases to 12.5%. VFD enabled control of the flow rate in a cost-effective and energy efficient manner.

Table 3: Operating capacity of pump

<table>
<thead>
<tr>
<th>Oberdorfer Chemsteel S935M Series</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Pressure [psig]</td>
</tr>
<tr>
<td>Maximum System Pressure [psig]</td>
</tr>
<tr>
<td>Maximum Fluid Temperature [°F]</td>
</tr>
<tr>
<td>Maximum Flow Rate [GPM]</td>
</tr>
</tbody>
</table>

Vacuum Pump

Prior to the start of an experiment, air had to be emptied out of flow boiling loop. A vacuum pump (figure 9) was used to extract air from the flow loop to create a vacuum. This practice guaranteed that loop was free from unwanted air and gas. It was also known from reference 1 that large amounts of air in the flow loop would influence boiling.

<table>
<thead>
<tr>
<th>Housing Material: Cast Iron</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Vacuum (in. Hg): 26</td>
</tr>
<tr>
<td>Free Air: 4.5 CFM</td>
</tr>
<tr>
<td>Motor power: ¼ hp</td>
</tr>
</tbody>
</table>

Figure 9: Oil-Free High-Vacuum Pump (McMaster Product ID: 9901K64)

Flow Loop Pressurizer

After the flow loop was completely filled with water prior to heat addition in the test section, the working fluid was pressurized by releasing nitrogen gas from a compressed gas tank.
into the flow loop. A dial pressure gauge was installed in the flow loop for pressure reading.

A Swagelok pressure relief valve (figure 10) rated at 150 psi and 200 °C was attached on top of the flow loop. This valve will relieve pressure from the flow loop by allowing nitrogen to flow through a secondary passage out of the system.

| Valve Material: 316 Stainless Steel
| Service Class: Low Pressure
| End Connection: ¼” Male NPT

Figure 10: SS Low-Pressure Proportional Relief Valve (Swagelok ID: SS-RL3M4-S4)

**Heat Exchanger**

A heat exchanger (figure 11) was used to achieve steady state conditions. Heat gained by water in the test section was removed inside a shell and tube heat exchanger. Hot water from the test section was passed through tube side and cold water from school plumbing system was passed through shell side of the heat exchanger. Cold water flow rate from the laboratory’s plumbing system was controlled by a gate valve.

| Capacity: 130,000 Btu/h
| Surface Area: 2.4 ft²
| Flow Capacity (gpm)
| Shell: 32
| Tube: 12

Figure 11: Stainless Steel Heat Exchanger (McMaster Product ID: 35185K42)
**Power Supply**

A DC power supply (figure 12) consists of a variable transformer connected to a 3-phase, 220 volt, 53 amps AC line. The output power can be continuously adjusted from 0 to 9 kW. The voltage across the test section and the current passing through it were continuously monitored and recorded.

<table>
<thead>
<tr>
<th>Features</th>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max. output power</td>
<td>15 kW</td>
</tr>
<tr>
<td>Constant DC voltage or current output</td>
<td></td>
</tr>
<tr>
<td>Efficiency</td>
<td>80-93%</td>
</tr>
<tr>
<td>Aircooled</td>
<td></td>
</tr>
<tr>
<td>Maximum Voltage</td>
<td>10 V DC</td>
</tr>
<tr>
<td>Maximum Current</td>
<td>1,500 Amp</td>
</tr>
<tr>
<td>Operating temperature</td>
<td>-55 °C to 85 °C</td>
</tr>
</tbody>
</table>

*Figure 12: DC power supply (LAMBDA EMI Product ID: ESS 10-1500-2-D)*

**Instrumentation & Data acquisition system**

K-type thermocouples were used to measure working fluid bulk temperature at inlet and outlet of the test section & heat exchanger. To measure heater rod wall temperature for heat transfer analysis, two T-type thermocouples were inserted into 0.51 mm holes drilled through the stainless steel tube wall and embedded by spot welding. All thermocouples were connected to a National Instrument Data Acquisition Switch Unit (NI 9213) with a scanning speed of 1200 samples per second. LabVIEW software data acquisition software was used collect, format and save data. Analog inline flow meter (OMEGA Engineering, FLMW-1020SS-HT), with a 0.5-20 GPM flow range and ±2.0% was used to measure flow rate of working fluid. A dial pressure gauge (McMaster, 9891T23), with a 0-150 psi pressure range and ±0.5% accuracy was used to track flow loop pressure.
Insulation

To prevent heat loss to the environment from the outer surface of the test section & flow loop piping during uniform heat flux experiment, 1” thick fiberglass insulation with thermal conductivity of 0.0389 W/m-K was fitted around piping and fittings.

After the major components of the experiment were chosen, a frame setup was necessary to assemble flow loop. Figure 14 shows the solid model of frame. Aluminum T-slotted framing material was used to construct the frame. Quick fasteners were used to connect framing pieces and loop equipment.
The major components of the flow loop are the gear pump, heat exchanger, flow meter, power supply for ohmic heating, vacuum pump, thermocouples, DAQ system for data logging. Figure 15 shows the schematic diagram of the experimental apparatus.

![Schematic diagram of experimental apparatus](image)

Photographs of experimental apparatus and component systems are shown in Appendix C.

**Experimental procedure**

Prior to the initiation of tests, the system is prepared according to the following procedure.

1. Set all the valves to closed position except for vacuum pump disconnect valve.
2. Vacuum pump is turned on and draws out unwanted air in the flow loop.
3. Vacuum pressure is monitored via compound pressure gauge. After obtaining -25 in. Hg of vacuum inside the loop, the vacuum pump is turned off and vacuum disconnect valve is closed.
4. Flow loop is filled with water from bottom drain connection. Water level is monitored through transparent tube in pressurizer section.
5. After the flow loop has been completely filled with water prior to heat addition in the test
section, the flow loop can be pressurized by releasing nitrogen gas from the compressed
gas tank and closing the shut-off valve when the flow loop has reached desired pressure.

6. Water pump is turned on and pressurized water is circulated through the flow loop and
the desired flow rate is established by adjusting VFD.

7. Coolant water is supplied to heat exchanger shell side by school plumbing system and
desired flow rate is established by adjusting globe valve.

8. A power supply to heater rod is turned on. DC output voltage of power supply is adjusted
for desired temperature of test section.

9. After the test section is heated to an equilibrium temperature, the thermocouples readings
are recorded for heat transfer analysis.

10. After carrying out experiment, nitrogen gas is released from flow loop by activating
toggle valve & water is drained to reduce fouling and corrosion inside flow loop.

5. Results & Discussion

After the experimental setup was complete, it was time to run experiments with water as
the working fluid. To obtain results, the loop was filled with water, set at desired system
pressure, pump was turned on and the desired flow rate was established by adjusting the variable
frequency drive. Once a fixed flow rate was reached, it was then time to turn on the power
supply to the desired power output. This was accomplished by adjusting DC output voltage of
power supply. As expected, increasing the power increases the surface heat flux over the test
section which leads to an increasing temperature change from inlet to outlet of test section. The
experiment was run by gradually increasing the power to the test section and recording the
thermocouple readings. The governing equation for determining heat transfer rate between a
surface and an adjacent fluid in motion is based on Newton’s law of cooling.
\[ q'' = h(T_w(x) - T_b(x)) \]  \hspace{1cm} (4)

Where:

\( q'' = \text{heat flux (W/m}^2\text{)} = \text{Power input/Surface area of heater rod} \)

\( T_b(x) = \text{bulk fluid temperature at location x from inlet of test section (°C)} \)

\( T_w(x) = \text{surface temperature of heater rod at location x from inlet of test section (°C)} \)

\( h(x) = \text{local heat transfer coefficient at distance x from inlet of test section (W/m}^2\cdot{\degree}\text{C)} \)

In this investigation, the local heat transfer coefficient was calculated for section where wall thermocouples of heater rod were fitted. Under the condition of the present experiments, the exit temperature of water from test section was generally quite close to saturation. Hence, any non-equilibrium effects were quite small.

The power supply required adjustment of the DC output voltage which in effect changed the current to give a specific power output. The following current and voltage inputs in table 4 resulted in specific power and surface heat flux output over the test section:

Table 4: Power supply specifics

<table>
<thead>
<tr>
<th>Voltage (V)</th>
<th>Current (A)</th>
<th>Power (W)</th>
<th>Surface Heat Flux (W/m(^2))</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>273</td>
<td>818</td>
<td>19,206</td>
</tr>
<tr>
<td>4</td>
<td>364</td>
<td>1455</td>
<td>34,144</td>
</tr>
<tr>
<td>5</td>
<td>455</td>
<td>2273</td>
<td>53,350</td>
</tr>
<tr>
<td>6</td>
<td>545</td>
<td>3273</td>
<td>76,824</td>
</tr>
<tr>
<td>6.5</td>
<td>591</td>
<td>3841</td>
<td>90,162</td>
</tr>
<tr>
<td>7.5</td>
<td>682</td>
<td>5114</td>
<td>120,038</td>
</tr>
<tr>
<td>8.5</td>
<td>773</td>
<td>6568</td>
<td>154,182</td>
</tr>
</tbody>
</table>
The wall temperature of heater rod was obtained from thermocouples readings.

Satisfactory evaluation of \( h \) by Newton’s law of cooling depends on the appropriate determination of temperature difference between heater rod wall temperature and bulk fluid temperature. Since bulk fluid temperature was not directly measured an appropriate calculation scheme is required. To determine the bulk fluid temperature an energy balance is made for an element \( dx \) of the test section flow channel shown in figure 16. Neglecting changes in kinetic and potential energy and assuming steady state and constant \( c_p \), conservation of energy for the element is given by

\[
q^* P dx + \dot{m} c_p T_m = \dot{m} c_p \left[ T_m + \frac{dT_m}{dx} dx \right] \tag{5}
\]

Figure 16: Conservation of energy for element \( dx \).

where \( P \) is channel perimeter. Simplifying gives

\[
\frac{dT_m}{dx} = \frac{q^* P}{\dot{m} c_p} = \text{constant} \tag{6}
\]

This result shows that the axial gradient of the mean temperature is constant along the channel.

Integrating above equation gives

\[
T_m(x) = \frac{q^* P}{\dot{m} c_p} x + C_1 \tag{7}
\]
where $C_1$ is a constant of integration which is determined from inlet condition at $x = 0$.

\[ T_m(0) = T_{mi} \quad (8) \]

Hence, $C_1 = T_{mi}$. Bulk fluid temperature is also the mean fluid temperature.

\[ T_b(x) = \frac{q^p}{mc_p} x + T_{bi} \quad (9) \]

In applying the equations and models in subcooled flow boiling, it was essential to determine the local subcooling at the location where wall temperature was measured. In many data sets reported in literature, including Del Valle and Kenning [2], the subcooling at the inlet to the test section was reported. As the liquid flows through the test section, it gets heated, and the local subcooling decreases in the flow direction. This effect is quite significant in long test sections under high heat flux conditions.

Figure 17 shows the flow boiling curve, indicating the dependence of the wall heat flux $q''$ on wall superheat, $\Delta T$, which is defined as the difference between the wall temperature and the saturation temperature of the liquid at the system pressure. Subcooled water ($\Delta T_{sub}$ = 15°C), at a velocity of 0.249 m/s and 8.53-3.52 °C subcooling (where wall temperature was measured), was circulated at 1 atm pressure. It can be seen that substantial increases in heat flux result in only minor increases in the temperature of heater wall. Figure 17 also shows a comparison of some of the subcooled boiling correlations listed in Table 1 with the experimental data. The McAdams et al. (1949) correlation considerably over predicts the wall superheat. The Jens and Lottes (1951) correlation also over predicts wall superheat. The Thom et al. (1965) correlation agrees closely with experimental data. All the correlations display the same trend as the experimental data listed in appendix A.
The relation between wall heat flux and wall superheat can be written as

\[ q'' = (\Delta T_{sat})^a \]  \hspace{1cm} (10)

For the data plotted in figure 17, the exponent \( a \) is found to be approximately 3.41. The dependence of \( q'' \) on \( \Delta T_{sat} \) is expressed through an exponent of 3.86 by McAdams et al. (1949), whereas it is 4.0 in the equations by Jens and Lottes (1951), and Thom et al. (1965). The value of exponent \( a \) is affected by the orientation of the heater surface, geometry, surface roughness, and system pressure [11]. If we compare figure 17 with figures A7, A8 & A9 in appendix A, we can see that at constant pressure with increases in flow rate and heat flux, the Jens and Lottes and Thom et al. correlations over-predict wall superheat. These correlations do not account for liquid flow rate. The exponent \( a \) is found to lie approximately between 3.15 and 3.50 for all experiments.
In figure 18, the heat fluxes are plotted against wall superheat for various pressures. During these experiments, the flow rate of water was held constant at 0.5 gpm. After investigation of figure 18, the effect of increased system pressure is to shift the boiling curve to the left. This, in turn, corresponds to a higher heat flux at a given wall superheat for a higher system pressure. McAdams et al.’s (1949) correlation did not account for the effect of system pressure on subcooled flow boiling. The correlations of Jens and Lottes (1951), and Thom et al. (1965) account for system pressure but the wall superheat for the different pressures calculated from correlations fall approximately along the same straight line and not distinguishable. Therefore, plots of correlations for various system pressures are not presented. In appendix A, figures A1-A3 represents boiling curves at various system pressures for flow rate (1-2.0 gpm), system pressure (1-9 atm), and wall heat flux (19-154 kW/m$^2$).
It has been shown that the subcooled boiling heat transfer is influenced by the amount of dissolved gases or impurities in the liquid. When a degassed liquid is used as the working fluid, the heater rod wall temperature may exceed the boiling point by as much as 20 °C before bubbles begin to form. On the other hand, if a large amount of gas is dissolved in the working fluid, bubble formation may occur at heater rod wall temperatures below the saturation temperature [5]. Tap water was used in the current experiments. Vacuum was applied to draw out unwanted air and gas from flow boiling loop prior to filling the loop with water. Results on figure 18 indicate that tap water is full of dissolved air and gases. For this reason, the curves of figure 18 are extended to wall superheat temperatures of less than 10 °C. The dissolved gas in coolant not only promotes bubble nucleation by providing pre-existing interfaces, but it also reduces the superheat required for nucleation.

Equation (1) was used to determine bulk liquid temperature at heater rod thermocouple location. Then Newton’s law of cooling was used to determine heat transfer coefficient. In figure 19, the heat transfer coefficients are plotted against heat fluxes for various pressures. During these experiments, the flow rate of water was held constant at 0.5 gpm. After investigation of figure 19, it was determined that in subcooled boiling region the temperature difference between the heater wall and the bulk fluid decreases with increasing heat flux. The subcooled boiling heat transfer coefficient is insensitive to an increase in system pressure at a given flow rate and heat flux. In subcooled flow boiling region, the total heat transfer coefficient is a combination of convective heat transfer coefficient and nucleate boiling coefficient. For a given fluid at a given pressure, the convective heat transfer coefficient depends on local mass flux and local vapor quality. The convective heat transfer coefficient is not, therefore, strongly influenced by the thermal boundary conditions. However, since the nucleate boiling coefficient is a function of
local wall superheat, or local heat flux, it can be influenced considerably by the thermal boundary conditions [4]. In appendix A, figures A4-A6 represent heat transfer coefficients for various system pressures (1-9 atm), flow rates (1-2.0 gpm), and wall heat fluxes (19-154 kW/m$^2$).

Figure 19: Comparison of heat transfer coefficients for various system pressure (0.5 gpm)
6. Conclusion

Experimental data have been obtained on heat transfer from a stainless-steel tube to water in the heat flux range from 19 to 155 kW/m$^2$ over a pressure range from 101 to 912 kPa (1-9 atm) for flow rates from 2.27 to 9.09 LPM (0.5-2 gpm). The results of these tests (i.e. summarized in figure 18) permit the determination of the surface temperature necessary for the removal of a given heat flux in the subcooled boiling regime. The Thom et al. (1965) correlation agrees closely with experimental data.

The temperature of the heat-transfer surface (when it exceeds the boiling temperature of the liquid) is insensitive to changes in heat flux. In the subcooled boiling regime at constant liquid flow rate and heat flux, the pressure of the liquid determines the surface temperature of the heater rod.

The effect of increased system pressure is to shift the boiling curve to the left. This, in turn, corresponds to a higher heat flux at a given wall superheat for a higher system pressure. The maximum wall superheat was less than 10 °F for all experiments.

In subcooled flow boiling region, the total heat transfer coefficient is a combination of convective heat transfer coefficient and nucleate boiling coefficient. The subcooled boiling heat transfer coefficient is insensitive to increases in system pressure at a given flow rate and heat flux.
References


Appendix-A

Figure A1: Boiling curves for different pressures (1 gpm).
Figure A2: Boiling curves for different pressures (1.5 gpm) (q*:9.02-120.1 kW/m²).

Figure A3: Boiling curves for different pressures (2 gpm).
Figure A4: Comparison of heat transfer coefficients for various system pressure (1 gpm)

Figure A5: Comparison of heat transfer coefficients for various system pressure (1.5 gpm)
Figure A6: Comparison of heat transfer coefficients for various system pressure (2 gpm)

$\Delta T_{\text{sub inlet}} \approx 15 \, ^\circ\text{C}$

$P_{\text{system}} = 1 \, \text{atm}$

$\dot{V} = 1 \, \text{gpm}$

$q'' = 53.4-76.9 \, \text{kW/m}^2$

$a = 3.41$

Figure A7: Comparison of subcooled flow boiling correlations with experimental data
\[ \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ C \]

\[ P_{\text{system}} = 1 \, \text{atm} \]

\[ \dot{V} = 1.5 \, \text{gpm} \]

\[ q'' = 90.24 - 120.14 \, \text{kW/m}^2 \]

\[ a = 3.26 \]

Figure A8: Comparison of subcooled flow boiling correlations with experimental data

\[ \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ C \]

\[ P_{\text{system}} = 1 \, \text{atm} \]

\[ \dot{V} = 2 \, \text{gpm} \]

\[ q'' = 120.14 - 154.32 \, \text{kW/m}^2 \]

\[ a = 3.42 \]

Figure A9: Comparison of subcooled flow boiling correlations with experimental data
\[ \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ\text{C} \]

\[ P_{\text{system}} = 2 \, \text{atm} \]

\[ \dot{V} = 0.5 \, \text{gpm} \]

\[ q'' = 19.22-34.17 \, \text{kW/m}^2 \]

\[ a = 3.36 \]

Figure A10: Comparison of subcooled flow boiling correlations with experimental data

\[ \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ\text{C} \]

\[ P_{\text{system}} = 2 \, \text{atm} \]

\[ \dot{V} = 1 \, \text{gpm} \]

\[ q'' = 53.4-76.9 \, \text{kW/m}^2 \]

\[ a = 3.428 \]

Figure A11: Comparison of subcooled flow boiling correlations with experimental data
\[ \Delta T_{\text{sub inlet}} \approx 15^\circ \text{C} \]

\[ P_{\text{system}} = 2 \text{ atm} \]

\[ \dot{V} = 1.5 \text{ gpm} \]

\[ q'' = 90.24-120.14 \text{ kW/m}^2 \]

\[ a = 3.38 \]

Figure A12: Comparison of subcooled flow boiling correlations with experimental data

\[ \Delta T_{\text{sub inlet}} \approx 15^\circ \text{C} \]

\[ P_{\text{system}} = 2 \text{ atm} \]

\[ \dot{V} = 2 \text{ gpm} \]

\[ q'' = 120.14-154.32 \text{ kW/m}^2 \]

\[ a = 3.29 \]

Figure A13: Comparison of subcooled flow boiling correlations with experimental data
\( \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ\text{C} \)

\( P_{\text{system}} = 3 \, \text{atm} \)

\( \dot{V} = 0.5 \, \text{gpm} \)

\( q'' = 19.22-34.17 \, \text{kW/m}^2 \)

\( a = 3.327 \)

\( \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ\text{C} \)

\( P_{\text{system}} = 3 \, \text{atm} \)

\( \dot{V} = 1 \, \text{gpm} \)

\( q'' = 53.4-76.9 \, \text{kW/m}^2 \)

\( a = 3.39 \)

Figure A14: Comparison of subcooled flow boiling correlations with experimental data

Figure A15: Comparison of subcooled flow boiling correlations with experimental data
ΔT_{sub inlet} ≈ 15 °C

P_{system} = 3 atm

\dot{V} = 1.5 \, \text{gpm}

q'' = 90.24 - 120.14 \, \text{kW/m}^2

a = 3.414

Figure A16: Comparison of subcooled flow boiling correlations with experimental data

ΔT_{sub inlet} ≈ 15 °C

P_{system} = 3 atm

\dot{V} = 2 \, \text{gpm}

q'' = 120.14 - 154.32 \, \text{kW/m}^2

a = 3.37

Figure A17: Comparison of subcooled flow boiling correlations with experimental data
\[ \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ\text{C} \]

\[ P_{\text{system}} = 4 \, \text{atm} \]

\[ \dot{V} = 0.5 \, \text{gpm} \]

\[ q'' = 19.22-34.17 \, \text{kW/m}^2 \]

\[ a = 3.28 \]

Figure A18: Comparison of subcooled flow boiling correlations with experimental data

\[ \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ\text{C} \]

\[ P_{\text{system}} = 4 \, \text{atm} \]

\[ \dot{V} = 1 \, \text{gpm} \]

\[ q'' = 53.4-76.9 \, \text{kW/m}^2 \]

\[ a = 3.27 \]

Figure A19: Comparison of subcooled flow boiling correlations with experimental data
\[ \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ\text{C} \]

\[ P_{\text{system}} = 4 \, \text{atm} \]

\[ \dot{V} = 1.5 \, \text{gpm} \]

\[ q'' = 90.24\text{-}120.14 \, \text{kW/m}^2 \]

\[ a = 3.50 \]

**Figure A20:** Comparison of subcooled flow boiling correlations with experimental data

\[ \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ\text{C} \]

\[ P_{\text{system}} = 4 \, \text{atm} \]

\[ \dot{V} = 2 \, \text{gpm} \]

\[ q'' = 120.14\text{-}154.32 \, \text{kW/m}^2 \]

\[ a = 3.15 \]

**Figure A21:** Comparison of subcooled flow boiling correlations with experimental data
**Figure A22:** Comparison of subcooled flow boiling correlations with experimental data

\[ \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ\text{C} \]

\[ P_{\text{system}} = 6 \, \text{atm} \]

\[ \dot{V} = 0.5 \, \text{gpm} \]

\[ q'' = 19.22-34.17 \, \text{kW/m}^2 \]

\[ a = 3.26 \]

**Figure A23:** Comparison of subcooled flow boiling correlations with experimental data

\[ \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ\text{C} \]

\[ P_{\text{system}} = 6 \, \text{atm} \]

\[ \dot{V} = 1 \, \text{gpm} \]

\[ q'' = 53.4-76.9 \, \text{kW/m}^2 \]

\[ a = 3.32 \]
ΔT_{sub inlet} ≈ 15 °C

P_{system} = 6 atm

\dot{V} = 1.5 \text{ gpm}

q'' = 90.24 - 120.14 \text{ kW/m}^2

a = 3.34

Figure A24: Comparison of subcooled flow boiling correlations with experimental data

ΔT_{sub inlet} ≈ 15 °C

P_{system} = 6 atm

\dot{V} = 2 \text{ gpm}

q'' = 120.14 - 154.32 \text{ kW/m}^2

a = 3.41

Figure A25: Comparison of subcooled flow boiling correlations with experimental data
ΔT_{sub \text{ inlet}} \approx 15^\circ C

P_{system} = 8 \text{ atm}

\dot{V} = 0.5 \text{ gpm}

q'' = 19.22-34.17 \text{ kW/m}^2

a = 3.23

Figure A26: Comparison of subcooled flow boiling correlations with experimental data

ΔT_{sub \text{ inlet}} \approx 15^\circ C

P_{system} = 8 \text{ atm}

\dot{V} = 1 \text{ gpm}

q'' = 53.4-76.9 \text{ kW/m}^2

a = 3.29

Figure A27: Comparison of subcooled flow boiling correlations with experimental data
ΔT_{sub \ inlet} \approx 15 \ ^\circ C

P_{system} = 8 \ atm

\dot{V} = 1.5 \ gpm

q'' = 90.24-120.14 \ kW/m^2

a = 3.276

Figure A28: Comparison of subcooled flow boiling correlations with experimental data

ΔT_{sub \ inlet} \approx 15 \ ^\circ C

P_{system} = 8 \ atm

\dot{V} = 2 \ gpm

q'' = 120.14-154.32 \ kW/m^2

a = 3.17

Figure A29: Comparison of subcooled flow boiling correlations with experimental data
\( \Delta T_{\text{sub inlet}} \approx 15^\circ \text{C} \)

\( P_{\text{system}} = 9 \text{ atm} \)

\( \dot{V} = 0.5 \text{ gpm} \)

\( q'' = 19.22-34.17 \text{ kW/m}^2 \)

\( a = 3.20 \)

Figure A30: Comparison of subcooled flow boiling correlations with experimental data

\( \Delta T_{\text{sub inlet}} \approx 15^\circ \text{C} \)

\( P_{\text{system}} = 9 \text{ atm} \)

\( \dot{V} = 1 \text{ gpm} \)

\( q'' = 53.4-76.9 \text{ kW/m}^2 \)

\( a = 3.18 \)

Figure A31: Comparison of subcooled flow boiling correlations with experimental data
\[ \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ\text{C} \]

\[ P_{\text{system}} = 9 \, \text{atm} \]

\[ \dot{V} = 1.5 \, \text{gpm} \]

\[ q'' = 90.24-120.14 \, \text{kW/m}^2 \]

\[ a = 3.29 \]

Figure A32: Comparison of subcooled flow boiling correlations with experimental data

\[ \Delta T_{\text{sub inlet}} \approx 15 \, ^\circ\text{C} \]

\[ P_{\text{system}} = 9 \, \text{atm} \]

\[ \dot{V} = 2 \, \text{gpm} \]

\[ q'' = 120.14-154.32 \, \text{kW/m}^2 \]

\[ a = 3.21 \]

Figure A33: Comparison of subcooled flow boiling correlations with experimental data
Appendix-B

Flow boiling loop pressure drop calculations:

**Pressure Drop for Heat Exchanger:**

- **Length of each tube** = 0.1615 m
- **Diameter, D** = 0.25 \times 0.0254 = 0.0063 m
- **Temperature, T_m** = 170°C
- **Flow area, A** = \( \frac{\pi \times D^2}{4} \) = 3.16 \times 10^{-4} m^2
- **Perimeter** = 26 \times \pi \times D = 0.199
- **Hydraulic Diameter, D_h** = \frac{Flow area, A}{Perimeter} = 0.016 m
- **Density, \rho** = 864 kg/m³
- **\( \mu \)** = \( 1.336 \times 10^{-2} \frac{kg}{s \cdot m} \)
- **Velocity, u** = \frac{mass flow rate}{26 \times Flow area \times density} = 1.06 m/s

- **Reynolds number, Re** = \( \frac{density \times velocity \times D}{\mu} \) = 3.65 \times 10^5
- **\( e = absolute\ roughness \)** = 0.15 \times 10^{-2}
- **Friction** = 0.036

**Pressure drop for heat exchanger, \Delta P_1** = \frac{friction \times L \times \rho \times u^2}{2 \times D_h} \times 7 = 5.46 kPa

**Pressure Drop for minor components (Bending Pipe 90 degree):**

- **Gravitational acceleration, g** = 9.8 m/s²
  - **Velocity, u** = 1.6646 m²/s

- **Head loss, h** = \( 1.2 \times \frac{velocity^2}{2 \times g} \) = 0.17 m

  **Pressure drop** = \( p \times g \times h \) = 1.45 Kpa

  **Total pressure drop for pipe (90°C), \Delta P_2** = 11.5 kpa

**Pressure drop between heat exchanger and test section:**

- **Diameter of the pipe, D** = 1 \times 0.0254 = 0.0254 m
- **Length of the pipe, L** = 48 \times 0.0254 = 1.21 m

  **Pressure drop, \Delta P_2** = \frac{friction \times L \times \rho \times u^2}{2 \times D} = 135.73 pa
Pressure drop between bend pipe and heat exchanger:

\[ \text{Length of the pipe, } L_2 = 36 \times 0.0254 = 1.21 \text{ m} \]
\[ \Delta P_3 = \frac{\text{friction} \times L \times \rho \times u^2}{2 \times D} = 135.73 \text{ pa} \]

Pressure drop in vertical pipe between bend pipe and pump:

\[ \text{Length of the pipe, } L_2 = 48 \times 0.0254 = 1.21 \text{ m} \]
\[ \text{Diameter of the pipe, } D = 1 \times 0.0254 = 0.0254 \]
\[ \Delta P = \frac{\text{friction} \times L_2 \times \rho \times u^2}{2 \times D} = 135.73 \text{ pa} \]
\[ \Delta P_5 = \Delta P + \rho \times g \times L_2 = 10.4 \text{ kpa} \]

Pressure drop between test section and pump:

\[ \text{Length of the pipe, } L_3 = 36 \times 0.0254 = 1.22 \text{ m} \]
\[ \text{Diameter of the pipe, } D = 1 \times 0.0254 = 0.0254 \]
\[ \Delta P_6 = \frac{\text{friction} \times L_3 \times \rho \times u^2}{2 \times D} = 135.73 \text{ pa} \]

Pressure drop in test section:

\[ \text{Length of the pipe, } L_2 = 48 \times 0.0254 = 1.22 \text{ m} \]
\[ \text{Diameter of the pipe, } D_h = \]  
\[ \Delta P = \frac{\text{friction} \times L_2 \times \rho \times u^2}{2 \times D_h} = 1 \text{ kpa} \]
\[ \Delta P_7 = \Delta P + \rho \times g \times L_2 = 10.04 \text{ kpa} \]

\text{TOTAL PRESSURE FOR THE LOOP} = 38.23 \text{ KPA}
Figure C1: Assembly of flow boiling loop and frame