Experimental investigation of incipient boiling and incipient quenching of R-123.

Li. Liang

University of Windsor
INFORMATION TO USERS

This manuscript has been reproduced from the microfilm master. UMI films the text directly from the original or copy submitted. Thus, some thesis and dissertation copies are in typewriter face, while others may be from any type of computer printer.

The quality of this reproduction is dependent upon the quality of the copy submitted. Broken or indistinct print, colored or poor quality illustrations and photographs, print bleedthrough, substandard margins, and improper alignment can adversely affect reproduction.

In the unlikely event that the author did not send UMI a complete manuscript and there are missing pages, these will be noted. Also, if unauthorized copyright material had to be removed, a note will indicate the deletion.

Oversize materials (e.g., maps, drawings, charts) are reproduced by sectioning the original, beginning at the upper left-hand corner and continuing from left to right in equal sections with small overlaps. Each original is also photographed in one exposure and is included in reduced form at the back of the book.

Photographs included in the original manuscript have been reproduced xerographically in this copy. Higher quality 6" x 9" black and white photographic prints are available for any photographs or illustrations appearing in this copy for an additional charge. Contact UMI directly to order.

UMI
A Bell & Howell Information Company
300 North Zeeb Road, Ann Arbor MI 48106-1346 USA
313/761-4700 800/521-0600
NOTE TO USERS

The original manuscript received by UMI contains pages with indistinct and slanted print. Pages were microfilmed as received.

This reproduction is the best copy available

UMI
EXPERIMENTAL INVESTIGATION OF INCIPIENT BOILING
AND INCIPIENT QUENCHING OF R-123

by

Li Liang

A thesis
Submitted to the Faculty of Graduate Studies and Research
through the Department of Mechanical and Material Engineering
in Partial Fulfillment of the Requirements for
the Degree of Master of Applied Science
at the University of Windsor

Windsor, Ontario, Canada

1997

© L. Liang
The author has granted a non-exclusive licence allowing the National Library of Canada to reproduce, loan, distribute or sell copies of this thesis in microform, paper or electronic formats.

The author retains ownership of the copyright in this thesis. Neither the thesis nor substantial extracts from it may be printed or otherwise reproduced without the author's permission.

L’auteur a accordé une licence non exclusive permettant à la Bibliothèque nationale du Canada de reproduire, prêter, distribuer ou vendre des copies de cette thèse sous la forme de microfiche/film, de reproduction sur papier ou sur format électronique.

L’auteur conserve la propriété du droit d’auteur qui protège cette thèse. Ni la thèse ni des extraits substantiels de celle-ci ne doivent être imprimés ou autrement reproduits sans son autorisation.

0-612-30954-1
ABSTRACT

Experiments were carried out to investigate the factors which affect the incipient boiling wall superheat for a copper/refrigerant R-123 interface and to compare these results with the behaviour predicted by the Martin-Dominguez nucleation site boiling model. The factors which affect the boiling behaviour of individual nucleation sites as well as the actual surface incipient boiling characteristics such as fluid and system conditions were studied using a facility which produced vertical flow in an annulus between an electrically heated 22.2 mm (0.875 in) outside diameter copper tube and a 25.4 mm (1 in) internal diameter glass tube.

The experimental observations allowed the following conclusions to be drawn:

■ The boiling incipience wall superheat is dependent upon the past thermal history of the surface/fluid interface, decreases as the system pressure increases and is independent of the refrigerant inlet subcooling and its mass flow rate.

■ Each nucleation site is unique and has its own dormancy range between its boiling cessation and its quench wall superheats. A dormant site will resume boiling at the same wall superheat at which it ceased boiling. A quenched site can only reactivate at or above its cessation wall superheat when seeded by vapour from a nearby boiling site.

■ All of these conclusions are compatible with the bubble/cavity model of Martin-Dominguez, a model which provides a very useful tool to explain the often seemingly contradictory results found in the literature.
DEDICATIONS

Especially to my mother

Jianhua Liao

for her love, encouragement and efforts made for me throughout her life.

To my wife,

Dan Zhu

for her love and understanding.

To my father,

Chunfang Liang

for his love and confidence in me.

To my brothers and sister

Ke, Bo and Jin

for their understanding, encouragement and support.

v
ACKNOWLEDGEMENTS

The author wishes to express his sincere gratitude to Dr. Thomas W. Mcdonald for his guidance and supervision throughout the development of this thesis. His ever willingness to discuss either the academic research or the life is greatly appreciated.

Thanks are due to Dr. Chao Zhang for being a co-supervisor for this program.

The author wishes to extend his gratitude to Mr. W. Beck and R. Tattersall for their technical assistance in modifying the experimental facilities.

Thanks are also to Dr. Ignacio Ramiro Martin-Dominguez for his help in using the data acquisition program and for calculating thermodynamic properties of R-123.

Acknowledgement is made to Natural Sciences and Engineering Research Council of Canada for providing financial support through Grant-in-Aid A877.

The Department of Mechanical and Material Engineering and the Faculty of Graduate Studies and Research are thankfully acknowledged for their support through assistantships and scholarships.
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>ABSTRACT</td>
<td>iv</td>
</tr>
<tr>
<td>DEDICATIONS</td>
<td>v</td>
</tr>
<tr>
<td>ACKNOWLEDGEMENTS</td>
<td>vi</td>
</tr>
<tr>
<td>LIST OF FIGURES</td>
<td>x</td>
</tr>
<tr>
<td>LIST OF TABLES</td>
<td>xii</td>
</tr>
<tr>
<td>NOMENCLATURE</td>
<td>xiii</td>
</tr>
<tr>
<td>1. INTRODUCTION</td>
<td>1</td>
</tr>
<tr>
<td>1.1 The Phenomenon of Boiling</td>
<td>1</td>
</tr>
<tr>
<td>1.2 Applications of Boiling</td>
<td>3</td>
</tr>
<tr>
<td>1.3 Objective of the Present Work</td>
<td>4</td>
</tr>
<tr>
<td>2. BASIC KNOWLEDGE OF NUCLEATE BOILING</td>
<td>6</td>
</tr>
<tr>
<td>2.1 Flow Boiling Regime</td>
<td>6</td>
</tr>
<tr>
<td>2.2 Bubble Equilibrium</td>
<td>6</td>
</tr>
<tr>
<td>2.3 Contact Angle</td>
<td>11</td>
</tr>
<tr>
<td>2.4 Cavities</td>
<td>11</td>
</tr>
<tr>
<td>2.4.1 Re-Entrant Cavities</td>
<td>16</td>
</tr>
<tr>
<td>3. LITERATURE REVIEW</td>
<td>18</td>
</tr>
<tr>
<td>3.1 Solid Surface</td>
<td>18</td>
</tr>
<tr>
<td>3.2 Contact Angle</td>
<td>19</td>
</tr>
<tr>
<td>3.3 Mixture of Liquids</td>
<td>19</td>
</tr>
<tr>
<td>3.4 Operating Conditions</td>
<td>20</td>
</tr>
</tbody>
</table>
3.5 Hysteresis
3.6 Effects of Dissolved Gas
3.7 Models of Boiling Nucleation

4. EQUIPMENT AND INSTRUMENTATION

4.1 Description of the Loop
4.2 Equipment
   4.2.1 Circulating Pump
   4.2.2 Flow Meter
   4.2.3 Pre-heater
   4.2.4 Flow Control Valve
   4.2.5 Test Section
   4.2.6 Condenser and Water Bath
   4.2.7 System Accumulator
   4.2.8 Loop Fittings
   4.2.9 Video Camera and Display Screen
4.3 Instrumentation
   4.3.1 Temperature Measurement
   4.3.2 Pressure Measurement
   4.3.3 Data Acquisition System
4.4 Discharge and Charge Procedures
4.5 Leakage Check

5. TEST PROCEDURE
6. RESULTS AND DISCUSSIONS

6.1 Effect of Inlet Subcooling of the Working Fluid

6.2 Effect of Mass Flow Rate

6.3 Effect of Maximum Wall Subcooling by increasing pressure between Tests

6.4 Effect of Pressure

6.5 Effect of Minimum Wall Superheat between the Cooling and Reheating Procedures

6.6 Effect of Heater Position

6.7 General Visual Observations

6.8 Individual Site Boiling Behaviour

7. CONCLUSIONS AND RECOMMENDED FUTURE WORKS

7.1 Conclusions

7.2 Recommended Future Works

REFERENCES

VITA AUCTORIS

APPENDIX A. EXPERIMENTAL PROGRAM
# LIST OF FIGURES

1.1 Typical boiling curve ...................................................... 2
2.1 Flow regimes for boiling in an uniformly heated channel .......... 7
2.2 Forces acting on a bubble in a constant temperature fluid ........ 8
2.3 Contact angle at the triple interface ................................... 12
2.4.1 Effect of the liquid contact angle on the bubble radius of curvature for both non-and poor-wetting fluids ......................... 14
2.4.2 Effect of the cavity angle on the bubble curvature radius ........ 15
2.4.3a Conical re-entrant cavity shape ..................................... 17
2.4.3b Bubble shapes in a re-entrant cavity ................................ 17
3.5 Characteristic curves of nucleate pool boiling hysteresis ............ 23
3.7.1 Incipient boiling and incipient quenching curves for R-123 in a re-entrant cavity ....................................................... 28
3.7.2 Effect of pressure on the incipient boiling and incipient quenching for R-123 ................................................................. 29
4.1 Schematic of experimental test facility .................................... 33
4.2.3 Pre-heater ................................................................. 36
4.2.5.1 Photograph of main heater unit ....................................... 38
4.2.5.2 Photograph of main heater cartridge assembly and instrumentation ................................................................. 39
4.2.5.3 Drawing of the heater .................................................. 40
4.2.5.4 Test annulus cross section ........................................... 41
4.2.5.5 Eccentric of heater

4.2.5.6 Temperature difference between heater cartridge and heating surface

4.2.5.7 Uncertainty of temperature difference between heater cartridge and heating surface

4.2.5.8 Assembly of test section

4.2.6 Condenser

4.2.7 Photograph of accumulator

4.2.9.1 Schematic of the video camera assembly

4.2.9.2 Photograph of nucleation sites on the screen

4.3.2 Photograph of pressure calibration assembly

6.1 Effect of inlet subcooling of the working fluid

6.2 Effect of mass flow rate

6.3 Effect of maximum wall subcooling by increasing pressure between tests

6.4a Effect of pressure on wall superheat

6.4b Effect of pressure on wall temperature

6.5 Effect of minimum wall superheat between the cooling and reheating procedures

6.6 Effect of heater position

6.7a Heat flux vs wall superheat at heater for a pressure of 170 kPa

6.7b Heat flux vs wall superheat at heater for a pressure of 236 kPa
LIST OF TABLES

6.1 Cessation and quench wall superheat for six sites 77
NOMENCLATURE

\[ D = \text{Diameter} \quad \text{m} \]
\[ g = \text{Gravitational acceleration} \quad \text{m/s}^2 \]
\[ h = \text{Enthalpy} \quad \text{kJ/kg} \]
\[ M = \text{Mass flow rate} \quad \text{kg/s} \]
\[ P = \text{Pressure} \quad \text{Pa} \]
\[ q = \text{Heat flow rate} \quad \text{W} \]
\[ Q = \text{Volumetric flow rate} \quad \text{m}^3/\text{s} \]
\[ r = \text{Bubble radius of curvature} \quad \text{m} \]
\[ \text{Re} = \text{Reynolds number} \]
\[ T = \text{Temperature} \quad \circ\text{C} \]
\[ T_{\text{ws}} = \text{Wall superheat (Tw - Tsat)} \quad \text{K} \]
\[ T_{\text{is}} = \text{Inlet subcooling (Ti - Tsat)} \quad \text{K} \]
\[ \text{TOS} = \text{Temperature overshoot} \quad \text{K} \]
\[ Z = \text{Flow development length from annulus} \]
\[ \text{inlet to start of heater} \quad \text{m} \]

GREEK LETTERS

\[ \beta = \text{Contact angle (angle between surface and bubble on liquid side of bubble)} \quad \text{degrees} \]
\[ k = \text{Thermal conductivity} \quad \text{W/m-K} \]
\pi \quad = \quad \text{Ratio of the circumference of a circle to its diameter}

\rho \quad = \quad \text{Density} \quad \quad \quad \quad \quad \quad \quad \quad \text{kg/m}^3

\sigma \quad = \quad \text{Surface tension} \quad \quad \quad \quad \quad \text{N/m}

\theta \quad = \quad \text{Angle} \quad \quad \quad \quad \quad \quad \quad \quad \text{degrees}

\nu \quad = \quad \text{Specific volume} \quad \quad \quad \quad \quad \text{m}^3/\text{kg}

\text{SUBSCRIPTS}

f \quad = \quad \text{Saturated liquid state}

fg \quad = \quad \text{Difference between saturated liquid and saturated vapour}

g \quad = \quad \text{Saturated vapour state}

i \quad = \quad \text{Refrigerant state at inlet to the annulus}

is \quad = \quad \text{Inlet subcooling of refrigerant at inlet to the annulus}

s, sat \quad = \quad \text{Saturation state}

w \quad = \quad \text{Wall}

ws \quad = \quad \text{Wall superheat}

W \quad = \quad \text{Water}
1. **INTRODUCTION**

1.1 **The Phenomenon of Boiling**

The heat transfer from a hot surface to a cooler liquid with the generation of vapour bubbles is called boiling. Boiling is a very efficient heat transfer process with a very high heat transfer coefficient.

The boiling phenomenon has been studied for many years in a number of areas but has never been fully understood. The transition between convective heat transfer and subcooled flow nucleate boiling is even less well understood and is the topic of this study.

The typical boiling phenomenon curve, plotted as surface heat flux q versus local wall superheat (local wall temperature Tw - local saturation temperature Tsat), is shown in Figure 1.1. For the non-boiling forced convection region, AB, as the heat flux increases the wall superheat increases beyond the expected boiling curve to point B where the first bubbles appear. The wall superheat at B is called the Incipient Boiling Wall Superheat. Once boiling has been initiated, the heat transfer coefficient increases greatly so that the local superheat abruptly and significantly drops at a constant heat flux. This nucleate boiling transition is shown as B to B'' in Figure 1.1. The curve from B' to B to B'' is usually ignored by most text books which show a smooth transition from B' to B'' in a simplified boiling curve. If the wall heat flux increases further, more and more nucleate sites become active and wall superheat increases following the nucleate boiling curve B'' → B'''' → C the critical heat flux.

When the heat flux is decreased from point C, the path followed by the system is
Figure 1.1  Typical boiling curve
C \rightarrow B''' \rightarrow B'' \rightarrow B' \rightarrow A. At point B' the last nucleate boiling site disappears. The wall superheat at B' is called Nucleate Boiling Cessation Wall Superheat. Nucleate boiling incipience and cessation for a surface are not the same point on its boiling curve and the transition between convection and nucleate boiling, B' \rightarrow B \rightarrow B'' \rightarrow B', is a hysteretic process (Stauder et al. 1986) (Ayub et al. 1988) (You et al. 1990c) (Celeta et al. 1992) (Shi et al. 1993). The maximum wall superheat difference between the nucleate boiling incipience and cessation curves for a surface is called the Temperature Overshoot (TOS).

1.2 Applications of Boiling

Boiling occurs in many areas such as in the petroleum, pulp and paper and the refrigeration and air conditioning industries as well as steam power generation and distillation. Boiling heat transfer is a common and important process in the following industrial equipment: heat pipes, boilers, reboilers and evaporators. By making use of the very high boiling heat transfer coefficients, the size of many heat exchangers can be made much smaller.

In practice, a large variety of materials, water, liquid metals, fluid nitrogen, fluid oxygen, halocarbon and fluorinated refrigerant etc, can be used as the working fluid for different applications.

Two-phase thermosiphon loop heat exchangers rely on the nucleate boiling process. In two-phase thermosiphons, the available temperature difference coupled with gravity creates the motive force for the circulation of the working fluid. Only if the
temperature in the evaporator of two-phase thermosiphons is high enough to initiate the nucleate boiling will the system work. Two-phase thermosiphons can be used in diverse applications such as: for solar collectors; for pre-cool and reheat coils in dehumidifiers; for cooling of lap top microcomputer chips as well as for cooling electrical switch gear equipment. To properly design a system for such industry applications it is most important to fully understand the conditions required to initiate nucleate boiling.

1.3 Objective of the Present Work

In some situations the temperature overshoot can be very large. Experimental measurements of the temperature overshoot reported by previous investigators (You et al. 1990c) often have large variations for seemingly identical tests. Thus some applications of nucleate boiling are limited because of our inability to predict the temperature overshoot reliably.

The objective of this work is to experimentally investigate the nucleate boiling hysteresis phenomenon. These experimental studies have been carried out to investigate the validity of the Martin-Donninguez model nucleation site boiling predictions and to experimentally determine the heat flux vs wall superheat characteristics of R123 flowing over a heated copper surface.

Refrigerant, R-123, was selected because its saturation pressure-temperature characteristics are most suitable for use in the existing test facility and its properties are typical of most refrigerants.

More specifically, the following factors which might be expected to affect the
incipient boiling and incipient quenching of individual nucleation sites as well as the actual surface incipient boiling characteristics were studied:

1) Inlet subcooling of the working fluid.
2) Refrigerant mass flow rate
3) Past thermal history of system
4) System pressure
5) Different test surfaces of the same material
6) Individual site boiling behaviour
2. BASIC KNOWLEDGE OF NUCLEATE BOILING

2.1 Flow Boiling Regime

Subcooled liquid entering the bottom of a uniformly heated vertical tube is shown in Figure 2.1. The heat transfer is initially forced convection pattern and no bubbles exist from A to B. At B, the wall superheat Tw is high enough to initiate boiling. From B to C the bubbles exist and grow in the superheated liquid near the surface but collapse as they enter the main flow because the bulk liquid is still subcooled. As the bulk liquid temperature Tb continues to increase, C to D, eventually some bubbles are able to move into and exist in the main flow. Note also that as the fluid bulk temperature is increasing, the saturation temperature, Tsat, is decreasing as the pressure decreases due to both friction and gravity. When bulk liquid temperature are high enough (Tb ≥ Tsat) the bubbles grow and often coalesce. This tends to initially produce Slug Flow (D to E) and eventually Annular Flow (E to F). This thesis focuses on the conditions necessary to initiate nucleate boiling of a subcooled fluid flowing over a heated surface (B).

2.2 Bubble Equilibrium

Martin-Dominguez (1993) analyzed the bubble equilibrium in a uniform temperature fluid. Figure 2.2 shows that the forces acting on a bubble hemisphere consist of the internal and external pressures and the surface tension of the vapour-liquid interface. To maintain the size of a bubble the required equilibrium condition is:
Figure 2.1  Flow regimes for boiling in an uniformly heated channel
Figure 2.2  Forces acting on a bubble in a constant temperature fluid
\[ \pi r^2 P_g = \pi r^2 P_f + 2\pi r \sigma \]  \hspace{1cm} (2.1a)

or

\[ P_g = P_f + \frac{2\sigma}{r} \]  \hspace{1cm} (2.1b)

At the bubble interface the liquid and the vapour contact each other and both of them should be in a saturated state so that the temperature is the saturation temperature corresponding to the local pressure.

\[ T_g = T_s(P_g) \]  \hspace{1cm} (2.2)

\[ T_f = T_s(P_f) \]  \hspace{1cm} (2.3)

From equation (2.1b) it is concluded that

\[ P_g > P_f \]  \hspace{1cm} (2.4)

Hence:

\[ T_g > T_f \]  \hspace{1cm} (2.5)

From the Clapeyron equation
\[
\left( \frac{dT}{dt} \right)_{sat} = \frac{h_{tg}}{T_s v_{tg}}
\]  

(2.6)

If one assumes that the vapour behaves as a perfect gas and the enthalpy of vaporization is constant, then equation (2.6) can be integrated to form the Clausius-Clapeyron equation:

\[
T_s (P_g) - T_s (P_f) = \frac{RT_s (P_g) T_s (P_f)}{h_{tg}} \ln \left( 1 + \frac{2\sigma}{\zeta P_f} \right)
\]  

(2.7)

If one assumes that the saturation temperature is constant and for small pressure variations

\[
\frac{h_{tg}}{T_s v_{tg}} = \text{constant}
\]  

(2.8)

then equation (2.6) can be integrated to yield

\[
T_s (P_g) - T_s (P_f) = \frac{2\sigma T_{sat} v_{tg}}{h_{tg} \zeta}
\]  

(2.9)

Equations (2.7) and (2.9) show that there must be some superheat relative to bulk liquid inside a bubble to maintain thermal and mechanical equilibrium. The smaller the radius of the bubble the higher the superheat required. If no bubble is present, these equations indicate that it is impossible to generate a bubble by heating a liquid!

For fluid flow with liquid and solid interfaces the tiny bubbles are trapped in the
pores of the solid surface and form the nucleus of the vapour bubble which allows for boiling on the surface. But vapour bubble occurrence happens to also depend on factors such as the contact angle of the liquid and solid, and the cavity shape on the solid surface. For a bubble in contact with a solid surface, the mechanical equilibrium equation (2.1b) is also valid but here the radius is the curvature radius of a spherical segment of the vapour and liquid interface.

2.3 Contact Angle

Martin-Dominguez (1993) studied the nucleation at a solid surface. Figure 2.3 shows the contact angle at the vapour, liquid and solid interface for well wetting, poor wetting and non wetting substances.

For the flat surface with equilibrium vapour bubbles as shown in Figure 2.3, if the system is cooled, vapour starts to condense and the volume of the vapour decreases. As a result, the vapour pressure within the bubble decreases and radius of curvature decreases. From equation 2.1b we know that a smaller radius bubble requires a larger vapour pressure to maintain it. As a result, the vapour phase will collapse quickly. For the same reason if the liquid superheat increases the bubble will grow and depart the surface. As a result, there is no nucleus left for another bubble. So that it is noted that some other condition on the solid surface is required.

2.4 Cavities

If a bubble exists inside a cavity, the radius of curvature of the spherical segment
Figure 2.3  Contact angle at the triple interface
at the liquid and vapour interface varies with the contact angle, the bubble position and the cavity shape. Figure 2.4.1 shows the effect of contact angle on the curvature radius.

For the non-wetting fluid shown in Figure 2.4.1 the contact angle is large and the bubble has an inverse (negative) radius of curvature within a conical cavity. Equation 2.1b shows that the surface tension of the spherical bubble resists the push of the liquid phase on the vapour phase inside the cavity. As a bubble shrinks into the cavity, the inverse (negative) bubble curvature radius becomes smaller. This will result in a lower equilibrium vapour pressure which will prevent the collapse of the bubble under any liquid subcooling.

For cooling of poor and well-wetting fluids (with contact angles < 90°) the bubble curvature radius remains positive as the bubble slides into the cavity and the bubble curvature radius reduces. However, as the radius of curvature decreases, a larger liquid superheat (larger vapour pressure) is required for equilibrium. Since the liquid superheat is decreasing these bubbles will collapse and disappear.

Most common substances have contact angles which range from zero to 90° and yet they are able to initiate nucleate boiling at some sites even when no gases are present. Therefore, there must exist some other mechanism on the surface of the conical cavities to maintain some bubble nuclei after boiling has ceased.

Figure 2.4.2 shows the effect of cavity angle on the curvature radius for poor-wetting substances. From geometrical considerations it can be shown that curvature radius increases as the cavity angle gets smaller and becomes infinite when:

\[
\theta = 2(\beta - 90°)
\]  

(2.10)
Figure 2.4.1 Effect of the liquid contact angle on the bubble radius of curvature for both non-and poor-wetting fluids
Figure 2.4.2 Effect of the cavity angle on the bubble curvature radius
and becomes negative (convex towards the vapour) when:

\[ \theta < 2(\beta - 90^\circ) \]  \hspace{1cm} (2.11)

2.4.1 Re-Entrant Cavities

Figure 2.4.3a shows the conical re-entrant cavity shape. To prevent the collapse of a bubble inside a re-entrant cavity, the interior cavity angle must be large enough to permit a decrease in the inverse bubble curvature radius as the bubble shrinks into the cavity.

Figure 2.4.3b shows the bubble shapes in a re-entrant cavity. As the bubble shrinks into this cavity, the radius of curvature reduces until the bubble reaches the re-entrant neck where the bubble curvature radius changes from a small positive value to an infinite value, then to a large inverse value which then becomes smaller. During this transition at the cavity neck, the surface tension resists the liquid pressure to help the bubble survive. As long as the inverse radius of curvature continues to decrease, the bubble embryo can exist under its equilibrium subcooled conditions without collapsing. Once the inverse radius of curvature starts to increase (with decreasing surface temperature) the bubble embryo will collapse. The smaller the neck radius the lower will be the "quench " temperature for a given cavity geometry.
Figure 2.4.3a Conical re-entrant cavity shape

Figure 2.4.3b Bubble shapes in a re-entrant cavity
3. LITERATURE REVIEW

Many researchers have investigated incipient boiling phenomena. The following literature review focuses mainly on the recent findings and the factors affecting incipient boiling. These factors have been classified by topic.

3.1 Solid Surface

Anderson and Mudawwar (1988) presented an experimental investigation of boiling heat transfer from a simulated microelectronic component immersed in a stagnant pool of FC-72. They attached various enhancement surfaces to an electrically heated heat transfer surface area. They tested a number of enhancement schemes, employing various arrangements of fins, studs, grooves, and vapour-trapping cavities. They showed that:

a) Large artificially-created cavities on the order of 0.3 mm diameter are ineffective in lowering the incipience temperature.

b) Increasing roughness promoted earlier incipience and therefore reduced the probability of significant incipience excursion (TOS).

c) Low profile microfin, microstud and inclined microgroove structures significantly augment nucleate boiling heat transfer at the expense of increased probability of incipience excursion.

You and Bar-Cohen (1990a) conducted an experimental study of pool boiling using cylindrical heater surfaces of platinum, silicon dioxide, and aluminum oxide immersed in FC-72 or R-113, saturated at 1 atm pressure. They noted that: the difference
in incipience wall superheat values between those with FC-72 and R-113 was significant, but the surface material effect on boiling incipience was small. The surface material effect was more pronounced in the nucleate boiling regime than on the incipience process.

Mizukami et al. (1992) experimentally studied heated stainless steel surfaces of three different roughnesses in a pool of water and suggested that small cavities on a surface may have more varied shapes than large cavities.

### 3.2 Contact Angle

Tong et al. (1990) examined the influence of the transient solid/liquid contact angles and contact angle hysteresis on the incipience superheat. They suggest that variations in the contact angle induced by changes in the direction and magnitude of the liquid/vapour interface velocity can substantially affect the formation of bubble embryos and may well explain the wide experimental scatter in incipience superheat values reported for highly-wetting liquids.

### 3.3 Mixture of Liquids

Tuzla et al. (1995) experimentally investigated the inception of bubble nucleation in falling films of glycerol-water mixtures at falling film flow rates ranging from 0.2 to 1.3 kg/m s and system pressure ranging from 25 to 434 mmHg. They observed that wall superheat for boiling inception increased with: an increase of the less volatile (glycerol) component concentration; a decrease of the absolute pressure; and with a decrease of the flow rate.
3.4 Operating Conditions

Hino and Ueda (1985) measured wall temperatures and photographically studied upward, subcooled flow boiling of R 113 in an annulus with a uniformly heated inner tube. They noted that the incipient boiling corresponds to the case where the upper limit of available cavity sizes is restricted and the wall superheats at the incipient boiling position are practically independent of the mass velocity and the inlet subcooling. They considered that the boiling curve hysteresis is a result of the activation of relatively large cavities by the previous intensive boiling.

Bar-Cohen and Simon (1988) briefly reviewed the mechanisms that may be responsible for delayed nucleation and related these mechanisms to the observed values. They concluded that the superheat excursion or temperature overshoot diminishes with increasing liquid velocity, subcooling, and operating pressure.

Shivprasad (1989) performed an experimental study with a vertical subcooled flow of R-11. He observed that:

a) Large wall superheats could be required to initiate boiling.

b) Different boiling curves were observed depending upon the superheat to which the wall was subjected prior to reheating (e.g. if one or more sites were still active as compared to no sites).

c) Sites which disappeared during cooling may re-activate without a TOS provided the surface superheat was not reduced by more than a few degrees below the boiling cessation value.

You et al. (1990b) studied a highly - wetting fluid (FC-72) and found that:
a) The effect of subcooling on pool boiling incipience was apparently small.

b) Increased pressure was shown to reduce wall superheat at incipience (a 5°C reduction for a pressure increase of 56 kPa).

Carrica et al. (1995) studied the effect of an imposed electric field on boiling of R-113 on a heated wire. They observed that:

a) The wall temperature overshoot at boiling inception was reduced by the electric field.

b) Nucleate boiling was enhanced by the field at low wall heat fluxes ($q'' = 41 \times 10^3$ W/m$^2$), while it was degraded at higher fluxes.

3.5 Hysteresis

Stauder and McDonald (1986) investigated the onset of nucleate boiling of R-11 for an air-to-air thermosiphon heat exchanger. They found that a hot duct to cold duct air temperature difference of 16 to 17 °C was required to initiate system boiling at pressures of 1.2 bars and 1.6 bars. Once initiated, boiling could remain active until the temperature difference decreased to 4 °C. When the air to air temperature difference was increased from low (but still active) values the loop boiling behaviour fell between the cold start up and the vigorous boiling cool down curves.

Cooper (1990) described an experimental investigation into the effect of an electric field applied to pool boiling of R-114 on a finned tube and a theoretical model of electrically enhanced nucleate boiling applicable to a simple surface. He showed that boiling hysteresis is completely eliminated through the electrical activation of nucleation
sites on the heat transfer surface following a brief application of a modest electric field.

You et al. (1990c) observed wide variations in incipience superheat from case to case for nominally identical cases and a very significant influence of minimum wall superheat (at the end of the cool-down portion of one run and the beginning of the subsequent heat-up run) on the incipience superheat excursion.

Celata et al. (1992) experimentally investigated flow boiling incipience hysteresis in subcooled wetting fluids (R-12 and R-114) in a wide range of pressure, mass flux and heat flux using an electrically heated stainless steel tube. They observed that:

a) The maximum wall temperature at the boiling inception is a strong function of the pressure, while other parameters such as local liquid subcooling, mass flux and heat flux turn out to be negligible.

b) The TOS is a strong decreasing function of pressure and mass flux.

c) Under conditions of low pressure, mass flux and low heat flux, the maximum value of the hysteresis was obtained, tending to suppress the subcooled boiling region.

Shi et al. (1993) theoretically analyzed both the Temperature Overshoot (TOS) in boiling incipience and a new type of hysteresis which may occur during the boiling development process, termed Temperature Deviation (TD) (Figure 3.5). They believed that the vapour gathering, vapour propagation and covering are the main mechanism of TOS hysteresis, while TD hysteresis is caused by the further vapour propagation of nucleation bubbles. They concluded that: hysteresis curves varied considerably for pool boiling. The curves can be affected by surface aging, surface roughness, working liquid, experimental
Figure 3.5 Characteristic curves of nucleate pool boiling hysteresis

FDNPB - fully developed nucleate pool boiling
PNPB - partial nucleate pool boiling
DNC - departure from natural convection
NC - natural convection.
procedure, etc.; i.e. anything that affects the fluid/surface combination. Changing the fluid properties and the solid surface conditions, and the use of supplementary measures, could eliminate or reduce boiling hysteresis. Locally fluidized particles in close vicinity to the heated surface also have a significant effect on boiling hysteresis.

Ayub and Bergles (1988) performed nucleate pool boiling tests on a number of GEWA-T deformed low fin surfaces in saturated R-113. They observed that boiling curve hysteresis existed but the maximum thermal excursions were quite small compared to those observed with other enhanced surfaces. They thought that the small and multiple thermal excursions (TOS) could be due to the following effects:

a) The re-entrant helical channels promote enhanced free convection.

b) The channels also constrain the initial nucleate boiling so that boiling in one channel does not activate an adjacent channel.

3.6 Effects of Dissolved Gas

You et al. (1990b) found that dissolved gas affected boiling incipience only at high gas concentrations (> 0.005 moles/mole).

You et al. (1991) experimentally studied the reduced incipient superheat in boiling of fluids which hold dissolved gas. They tested two techniques which enhance the mixing of gassy working fluids, a magnetic stirring device and boiling from a heater which is below the test heater. In both cases, enhanced mixing may raise the gas partial pressure in the fluid near the heated surface resulting in slow condensation of vapour within stationary bubbles that reside in potential nucleation sites. This, in turn, increases the
incipient bubble size which reduces the value of incipient superheat. They noted that:

a) High incipient wall superheat, leading to temperature hysteresis at incipience, appeared to be a result of two major features: 1) vapour in stationary bubbles at the wall condensed immediately after the removal of wall heating. 2) the dissolved gas content of the boiling fluid may not be uniform: it may be degassed very near the wall by the boiling process of the previous run. Replenishment of that gas by diffusion is slow. Mixing with a stirring rod and mixing with rising bubbles might mitigate these features.

b) Mixing with a stirring rod, incipient superheat values were only a few degrees above the saturation temperature and nearly perfect agreement between increasing and decreasing heat flux boiling curves for all the runs.

c) When tests were run with an operating heater located below the test heater and this heater produced boiling in the gassy bulk fluid, a substantial reduction in incipient superheat values was observed.

3.7 Models in Boiling Nucleation

Ge et al. (1987) simulated previous experimental results (Stauder and McDonald 1986) using a dormancy and quench model. They assumed that if the system was cooled, nucleation sites became dormant when the local wall superheat fell below the minimum value necessary to maintain nucleate boiling, then quenched at a lower wall superheat. They postulated that increasing wall superheat would cause dormant sites to reactivate,
but quenched sites would require a high wall superheat to reactive (as from a cold startup).

Mizukami et al. (1990) proposed a mechanical model in which the contact angle hysteresis and re-entrant cavities were introduced. Their prediction for a combination of water and stainless steel surfaces agrees fairly well with Fabric’s (1964) experimental results.

Martin-Dominguez (1993) analyzed the conditions necessary for thermo-mechanical equilibrium of a vapour bubble in both the bulk fluid and in a cavity. He found that the only location where a bubble may achieve the conditions necessary to remain in a stable steady state condition is when it is in contact with a surface and even then, only in cavities where certain geometric conditions were satisfied. He showed that the presence of these embryonic, dormant cavity bubbles are the nucleus from which boiling starts. He also explained that, for a given cavity, since incipient boiling is a function only of the wall superheat surrounding the bubble embryo, it can be affected by fluid flow only if the bubble protrudes into a non isothermal boundary layer or is physically deformed by the flow or the cavity pressure is affected by the flow. This is unlikely to occur with well wetting liquids (refrigerants). Thus, fluid flow has a negligible effect on the incipient boiling wall superheat for most fluids but water is a notable exception because of its poor wetting characteristic.

Martin-Dominguez and McDonald (1997) reported a “geometric” study of incipient boiling superheat for a cavity (the maximum wall superheat that a bubble embryo can resist before growing and departing from the surface cavity) and its incipient quench
superheat (the minimum wall superheat, or maximum wall subcooling, that the bubble embryo can sustain before collapsing by condensation inside the cavity). They investigated these two wall superheat limits inside various re-entrant conical cavities for representative substances with large and small contact angles (water and R-11/ (later for R-12 (McDonald et al. 1996)) respectively) using the following main assumptions:

a) the contact angle is constant for any given vapour-liquid-solid interface.

b) the shape of the bubble in contact with the solid surface is always a spherical segment.

c) the surface tension is a function of temperature only.

Based on these assumptions, the bubble radius of curvature is a function of cavity size, contact angle and solid surface orientation at the line of contact between the liquid and the vapour.

From their studies (see for example Figure 3.7.1 and Figure 3.7.2 for R 123), they concluded that:

a) The larger the cavity neck radius, the smaller the wall superheat range over which an embryo can exist.

b) Upon cooling any surface, the boiling cavity with the smallest neck will have the highest boiling cessation wall superheat (it will be the first to stop boiling).

c) Each site which retains a (dormant) vapour embryo will resume boiling at the same wall superheat at which it ceased boiling.

d) Small cavities can have a shape such that their vapour embryos will always
Figure 3.7.1 Incipient boiling and incipient quenching curves for R-123 in a re-entrant cavity
(Positive cavity angles represent exterior angles
negative cavity angles represent interior angles)
Figure 3.7.2 Effect of pressure on the incipient boiling and incipient quenching for R-123 in a cavity where the interior cavity angle is 180 degrees for any exterior cavity angle
exist (can not be quenched), regardless of the wall temperature. Boiling can always be reinitiated from them.

e) Any wall subject to boiling has a memory! The greater the wall subcooling in the past, the smaller the surviving embryo’s and the greater will be the wall superheat required to initiate boiling. This can cause inconsistent incipient boiling test results.

f) The wall memory can be erased with sufficient wall superheat and the presence of vapour on the heated surface.

McDonald et al. (1996) discussed the nature and cause of the nucleate boiling hysteresis phenomenon based on previous analytical and experimental works for R-123, and provided recommendations to assist in the design of two-phase thermosiphon loop systems.

Existing models to predict incipient boiling wall superheats are based on cavity shape and size models which cannot be known for real surfaces. These models can only be used to provide rational explanations of phenomenon not predict what will happen on a real surface. In the following study, experiments are carried out to investigate the validity of the trends predicted by the Martin-Dominguez model (1993) (1997).
4. EQUIPMENT AND INSTRUMENTATION

The test loop system was designed and built by Shivprasad (1989) and subsequently modified by Martin-Dominguez (see Appendix A) in order to study the phenomena of incipient boiling wall superheat and the hysteresis associated with a plot of heat flux vs local wall superheat in a complete cycle of increasing followed by a decreasing heat flux.

As a result of work of Martin-Dominguez (1993), it became necessary to also attempt to study incipient dormancy wall superheats, incipient quench wall superheat and the factors which affect them. The test loop allows the following parameters to be controlled:

1) System pressure
2) Volumetric flow rate
3) Heat flux at the test surface
4) Location of main heater along the length of the test cylinder.
5) Inlet subcooling at the test section

The data acquisition system allows the following parameters to be measured or calculated

a) Local wall temperature and superheat on the heating surface
b) Heat transfer coefficient in the test section
c) Reynolds number in the test section
d) Fluid temperature at various locations around the loop and the room temperature.
In order to be able to observe the boiling behaviour, the test section was designed to provide an annular flow between an inner heated copper tube and an outer glass pipe. Refrigerant R-123 was selected as a suitable refrigerant to study based on its thermal properties and the limitations of the test loop. In order to make the test apparatus suitable for use with new refrigerant R-123, the pressure accumulator, some pressure tap tubing, the data acquisition program and the method of measuring the main heater power had to be modified.

4.1 Description of the Loop

A schematic of the apparatus is shown in Figure 4.1. The equipment is essentially an enclosed loop. Refrigerant is circulated by a pump through a flow meter, pre-heater and flow control valve, then is heated in an annular test section, cooled in a condenser, and finally returned to the pump.

The temperature at the inlet of the test section can be controlled by adjusting the cooling water temperature of the condenser and the power input to the pre-heater. The pressure within the loop is controlled by setting the air pressure in the accumulator. The refrigerant flow rate is controlled by adjusting the flow control valve.

The loop has a pressure limit of 7.9 bars (790 kPa) and a maximum heater temperature of 120 °C.
Figure 4.1 Schematic of experimental test facility

1. Circulation pump
2. Flow meter
3. Electric pre-heater
4. Annular test section
5. Condenser
6. Circulating water bath
7. Accumulator
8, 9. Variacs
10. Air pressure regulator
11. Flow valve
12. Sight glass
13. Main heater
14. Wattmeter

T. Copper constantan thermocouple
P. Pressure gage
ΔP. Differential pressure gage
4.2 Equipment

4.2.1 Circulating Pump

A centrifugal pump with a steep Head-Flow characteristic was used as the circulating pump of the loop. It did not introduce large pulsations which can be detrimental to studies on individual nucleation sites. The pump inlet and outlet were connected to the loop through 12.7 mm (0.5 in) NPT female connectors. A 0.04 hp, 3450 rpm, 115 V a.c motor was coupled to the pump through a sealless magnetic drive.

4.2.2 Flow Meter

A rotameter supplied by Omega Engineering Company, Stamford, Connecticut, (model # FL-1504-A), was used to measure the refrigerant flow rate. It could be operated at pressures up to 7.9 bars (790 kPa) and 121 °C with an accuracy of ± 2 % and repeatability of ± 0.5 %.

The manufacturer calibrated the flowmeter by using water. The calibration curve for another fluid can be determined from the manufacturer supplied curve for water with the aid of the following formula:

\[
Q_{\text{refrigerant}} = \frac{Q_{\text{water}}}{\left( \frac{\rho_{\text{float}} - \rho_{w}}{\rho_{\text{refrigerant}}} \right)^{1/2} \left( \frac{\rho_{\text{float}} - \rho_{\text{refrigerant}}}{\rho_{w}} \right)}
\]

where, \( \rho_{\text{float}} = 8238 \text{ kg/m}^3 \)

This formula was used in the data acquisition program. Shivprasad experimentally
checked the validity of using this equation for R-11.

4.2.3 Pre-heater

The inlet fluid temperature was controlled by the pre-heater power input and the condenser cooling water temperature. The condenser alone is capable of controlling the inlet temperature, but the pre-heater provides a finer adjustment. The pre-heater used was a steel sheathed TEMRO # 220 1968, 282.6 mm (11.125 in) long oil immersion heater with a rated output of 300 W at 120 V. The pre-heater assembly is shown in Figure 4.2.3. A Variable transformer, capable of delivering 0 - 140 V, was used to control the power supplied to the pre-heater.

At an R-123 mass flow rate of 0.046 kg/s the pre-heater allowed the inlet temperature to be increased from 0 to 6.6 °C.

4.2.4 Flow Control Valve

A 19 mm (0.75 in) diameter hand operated ball valve was used to adjust the R-123 flow rate up to 0.382 kg/s. It is recommended that the valve be relocated to a position up stream of the pre-heater whenever the system is modified in the future. This will help avoid flashing (cavitation) when small inlet subcooling is desired.

4.2.5 Test Section

The test section was designed originally by Dinkar Shivprasad. Martin-Dominquez modified the main heater and sleeve so that it was possible to measure 16 temperatures
Figure 4.2.3 Pre-heater
along the heater cartridge sheath surface by thermocouples.

The following factors were considered when the test section was designed.

1) Electrical heating was used to provide a constant heat flux boundary condition because it was physically easier to control and to measure the heat flux than try to provide a constant temperature boundary condition.

2) The heater could be positioned anywhere along the length of the test section to study boiling in both the developing and the developed velocity profile regions.

3) The annulus dimensions were selected to provide annulus Reynolds numbers in the range existing in most industrial heat processors (3,000 to 30,000).

Figures 4.2.5.1 and 4.2.5.2 show photographs of the heater and sleeve assembly. Figure 4.2.5.3 shows a drawing of the heater. Sets of four thermocouples were installed at 12.7 mm (0.5 in) intervals along the length of sleeve. At each level the four thermocouples were arranged circumferentially at 90 degree intervals. The test annulus cross section is shown in Figure 4.2.5.4. Each thermocouple sensing tip was soldered flush with the sleeve outside wall and then the two halves of the sleeve were soldered together. To reduce possible temperature drops, in the gaps between the heater and inner side of the sleeve and between outside of the sleeve and inside of the test pipe, Omega therm-201 high thermal conductivity paste was used.

The copper tube outer surface temperature corresponding to a thermocouple position on the heater cartridge (Figure 4.2.5.5) was estimated from the measured thermocouple temperature (Martin-Dominguez 1993). Figure 4.2.5.6 shows the estimated temperature difference between the heater assembly surface temperature and the outside
Figure 4.2.5.1 Photograph of main heater unit
Figure 4.2.5.2 Photograph of main heater cartridge assembly and instrumentation
Figure 4.2.5.3  Drawing of the heater
Figure 4.2.5.4  Test annulus cross section
Figure 4.2.5.5  Eccentric of heater
Figure 4.2.5.6 Temperature difference between heater cartridge assembly surface (T3) and heating surface (T5)
surface of the copper tube for various gaps. These differences increase as the gap increases due to eccentric installation. For q=140 W (usually boiling initiation heat flux) maximum eccentric value is about 5 °C higher than aligned value. Figure 4.2.5.7 shows the uncertainty of this temperature difference. For q=450 W (highest heat flux in the tests) uncertainty is about ± 2.3 °C. In practice the readings were the average of four positions at each level. To minimize the effect of an eccentric installation, the heater cartridge was aligned carefully and if the TCs indicated uneven heating, the heater position would be adjusted by either rotating it within the tube, re-greasing the heater, or by changing its inclination.

A rod mechanism was designed to position the heater at any desired location along the length of the heater tube. At the point of boiling inception, the heat loss through this rod was estimated to be 1.5 % of the heater power input. The test section assembly is shown in Figure 4.2.5.8. A junction box (not shown in Figure 4.2.5.8) was mounted on the top of the positioning rod to provide connector terminals for the heater thermocouples.

4.2.6 Condenser and Water Bath

The condenser of the loop, which removes the energy input to the loop by the preheater and the main heater, was installed as close to the test section as physically possible to avoid a large vapour volume build up. The elevation of condenser was set 1.68 m above pump to prevent pump cavitation. The condenser, shown in Figure 4.2.6, consists of two coaxial copper pipes 0.423 m long to form a counter flow heat exchanger. Water from a controlled temperature circulating bath unit was used as the condenser coolant.
Figure 4.2.5.7 Uncertainty of temperature difference between heater cartridge and heating surface
Figure 4.2.5.8  Assembly of test section
Figure 4.2.6  Condenser
This made it easy to control the R-123 outlet temperature.

4.2.7 System Accumulator

An air pressure controlled accumulator was incorporated in the loop in order to:

1) Maintain the system at a desired pressure,

2) Act as a reservoir for the working fluid when part of the loop fills with vapour.

The Buna-N rubber bladder of the original pressurizer failed shortly after R-123 was charged in the loop. Pieces of Buna-N rubber from a broken bladder were immersed in R-123 for several days to test for a possible chemical reaction. These tests showed that Buna-N rubber swelled and weakened (became very easy to tear). The pressure accumulator was then redesigned to consist of a glass vertical tube with its one lower end linked to the loop and the upper end connected to a pressure controlled air source as shown in Figure 4.2.7 (photograph). The refrigerant and the air are in contact with each other at their interface. Although air will dissolve in the refrigerant with time, dissolved gases were purged from the loop during a pretest where the system was run boiling for about 30 min. During boiling, the non-condensible gases came out of the fluid with the vapour and were collected in a small accumulator at the top of the loop. The collected gas was periodically bled from this accumulator.

4.2.8 Loop Fittings

Copper, brass and glass were used as the material of the various components of the loop with plastic tubing for the various pressure and accumulator lines. A
Figure 4.2.7  Photograph of accumulator
liquid/vapour indicator was installed at the highest elevation in the loop. A release valve was installed above the indicator. An accumulator was soldered above this release valve to allow non-condensible gases to collect and periodically be bled off through a downstream female Schrader core valve located at the mouth of the release outlet. On the suction side of the pump a filter drier was used to remove suspended particles in the loop. A discharge and charge line was connected to the loop through a valve.

4.2.9 Video Camera and Display Screen

It was very difficult for the unaided eye to observe low intensity boiling behaviour of individual sites near their cessation wall superheat because the bubbles were very small. A video camera and a display screen were used to magnify and view nucleation sites more clearly. A schematic of the setup is shown in Figure 4.2.9.1. The presence of boiling as seen on the video screen is shown in Figure 4.2.9.2.

4.3 Instrumentation

4.3.1 Temperature Measurement

The test surface temperatures were estimated by the data acquisition system based on the measured heater cartridge sheath surface temperatures and assuming one dimensional radial heat transfer through the grease filled clearance annulus and through the copper tubing.

The refrigerant bulk temperatures at the inlet and the outlet of the test section and after the condenser, and the cooling water bath temperature and the ambient temperature
Figure 4.2.9.1  Schematic of the video camera assembly
Figure 4.2.9.2 Photograph of nucleation sites on the screen
were measured by type-T (Copper/Constantan 55% Copper and 45% Nickel) thermocouples. Over the range of operation, between 10 °C and 120 °C, the error induced by a possible non-constant Seebeck coefficient was ± 0.5 °C. Teflon thermocouple insulation was chosen because it is flexible, retains its dielectric properties up to 260 °C and resists solvents, acids, bases, abrasion and water.

4.3.2 Pressure Measurement

To estimate the wall superheat, the wall temperature and the local saturation temperature must be known. The inlet saturation temperature was calculated by measuring the pressure in the inlet section with a pressure transducer. The frictional and hydrostatic pressure drop from this inlet location to any downstream location was calculated from the flow rate, the elevation change and the use of an empirically developed equation for the system based on single phase flow. The local pressure was then calculated from the inlet pressure and the calculated pressure drop. The local saturation temperature was calculated from the local pressure using saturation property tables. The data acquisition program performed the above calculations from inputs of the heater cartridge location, the volume flow rate and the inlet pressure.

The system pressure was measured by an Omega piezoresistive pressure transducer, Model# PX 82-100 GV, 7.9 bars (790 kPa). The transducer had an accuracy of ± 0.25 % of the full scale reading, 790 kPa. The pressure transducer was calibrated against a dead weight tester (a photograph of the setup is shown in Figure 4.3.2). The resulting calibration formula was incorporated into the data acquisition program.
Figure 4.3.2 Photograph of pressure calibration assembly
The diaphragm of a differential pressure transducer installed in the system between the inlet and the outlet of the annular test section failed part way through the test sequences. This allowed an unexpected circulation through the pressure tap tubing. The differential transducer and tubing/fittings were removed from the system for the remainder of the tests.

4.3.3 Data Acquisition System

The on-line computer controlled data acquisition and data analysis system was developed originally by Martin-Dominquez utilizing an 8082A (Sciemetric) data logger interfaced with a computer. He wrote a Pro-Basic program "BOILNEW" which allows the operator to view any of the scanned inputs from the thermocouples and the pressure transducer as well as any of the calculated results. This data may be saved at any time either by manual decision or by periodically scheduling it.

The Martin-Dominguez program was further modified for these tests in the following aspects:

1) The computer data acquisition program was edited to incorporate a new pressure calibration.

2) It was found that temperature readings were affected when the power to the main (test section) heater was changed. It was concluded that the larger voltage power channel in the data acquisition system was electrically affecting the weak \(10^{-6} \text{v}\) signal of the thermocouple channels inside the 8082A A/D converter. To avoid this problem the power input channel was disconnected and the program was modified
to accept a manual power input from our wattmeter.

3) The program was edited to print an error message identifying which channel failed in order to troubleshoot the system more easily.

4) Physical data for R-123 supplied by Martin-Dominguez was incorporated.

4.4 Discharge and Charge Procedures

The procedure used for the removal of R-11 from the loop was as follows:

1) The waste storage tank was connected to the discharge/charge outlet/inlet at the bottom of the loop. The discharge/charge valve was opened. The liquid refrigerant was drained out into the waste tank. The release valve on the top of the loop was occasionally opened to avoid a vacuum buildup inside the loop as the liquid flowed out of the loop.

2) The pump was disconnected and the liquid inside the inlet and outlet pipes to the pump was drained into the waste tank.

3) The pump within the loop was reconnected.

4) The valve on the bottom of the loop was closed.

5) The remaining refrigerant vapour and non condensible gases were removed by evacuating the loop through the outlet on the top of the loop. To avoid harmful refrigerant going into the ambient air, the vapour coming out of the loop was condensed in an ice bath condenser. This condensate was later dumped into the waste tank which was then sealed.

6) The exhaust valve was closed when the pressure reached a vacuum of 25 in Hg
(635 mm Hg).

After the loop was discharged and the remaining vapour/gas removed, the loop was charged with R-123 using the following procedure:

1) R-123 storage tank was put at a high elevation and connected by tubing to the charge/discharge inlet/outlet at bottom of the loop.

2) The vacuum system was connected by tubing to the outlet at the top of the loop and a vacuum pulled on the loop. The valve at the top of the loop was then closed.

3) The charge valve was opened. If the loop did not fill completely, the top valve was opened with the vacuum pump running in order to suck non-condensibles and vapour out of the loop until it was filled with liquid R-123 and the accumulator half filled with liquid R-123.

4) The valve at top of the loop was closed.

5) The charge valve at bottom of the loop was closed.

6) The vacuum discharge line liquid trap was disconnected and any condensed R-123 was poured into the waste storage tank.

4.5 Leakage Check

Leakage of the loop was checked after both procedures of discharge and charge as follows:

1) After the loop was discharged and a vacuum pulled on the system the following indicated that there were leaks somewhere in the loop:
a) The loop pressure could not reach a low value with the vacuum pump operating and no liquid refrigerant in the loop.

b) The loop lost its vacuum with time after a number of hours or days.

2) After the loop was charged, the system was pressurized above atmosphere and leakage around the loop was checked by using a Halogen Leak Monitor.
5. TEST PROCEDURE

Tests were carried out by maintaining the system pressure, refrigerant mass flow rate and inlet subcooling constant while changing the test section heat input. The following is a more detailed description of how such tests were carried out.

1) The main heater was positioned at the desired location along the length of the test cylinder.

2) The pressure of the loop was set by adjusting the air pressure control valve.

3) The data acquisition system was loaded in the computer and started. For details see Appendix A.

4) The pump was turned on, and the refrigerant control valve adjusted to achieve the desired flow rate.

5) The condenser cooling water was supplied by a constant temperature water circulation bath. Its temperature was controlled by circulating cold water through the bath’s coils while simultaneously providing a thermostatically controlled electrical heat input to the bath.

6) Fine control of the refrigerant inlet subcooling (temperature) was achieved by adjusting the power input to the pre-heater.

7) The test section power input was controlled by adjusting the supply voltage across the test section heater. The loop was allowed to reach a steady state before any data was recorded.

8) When the system reached each desired steady state condition, all the desired data
and visual observations (heat flux, temperatures, pressures, superheats etc) were recorded.

9) During tests with low level boiling, some bubble sites were viewed with the assistance of a video camera/display monitor. This made it easier to note when a site became inactive and when it reactivated.

To investigate the effect of pressure, refrigerant mass flow rate or inlet subcooling on the boiling performance, a series of tests were carried out with different settings of one parameter for each test while keeping the same settings for the other parameters.

Because the boiling history of the system prior to a test can affect the boiling performance of a surface, prior to each test the surface of test section was heated until the wall superheat exceeded the incipient boiling value. This condition was then sustained for about 30 minutes, then the system was allowed to cool down to the desired initial temperature. This procedure was followed in order to ensure that the surface would have the same boiling history for every test.

The waiting time for the system to reach steady state conditions after a heat flux change was about 5 minutes. When vigorous boiling was present this time interval become shorter. Twenty to forty sets of data are required for a typical cycle of heating, boiling and cooling.
6. RESULTS AND DISCUSSIONS

Typical test cycles consisted of increasing the power input to the heater until boiling erupted and continuing until the limiting heater temperature of 120 °C was reached, cooling to any desired wall superheat (subcooling), then possibly reheating the system to investigate the quenching or dormancy of the last boiling sites.

Experimental results are presented in figures of Wall Superheat ($T_{ws} = T_{wall} - T_{sat}$) versus Surface Heat Flux ($q$) at a given system pressure ($P$), inlet subcooling ($T_{is}$) and mass flow ($M_f$) of refrigerant R-123. These figures show the hysteresis behaviour of the system and clearly define the boiling initiation point. In keeping with common practice the surface heat flux has been plotted on the ordinate axis even though it was the experimentally controlled (independent) variable.

The wall superheat values presented were an average of the topmost thermocouple elevation (level 4) of the cartridge heater except where otherwise noted.

The heater was positioned at $Z = 0.24$ m except for a few tests where $Z = 0.17$ m. Here, $Z$ represents the flow development length from the annulus inlet to the start of the heated surface.

6.1 Effect of Inlet Subcooling of the Working Fluid

Figure 6.1 shows the results of tests run with an inlet subcooling of 10 K and 4 K after each was previously cooled to a wall superheat of -10 K. The forced convective heat transfer portion of the curves were parallel to each other (displaced horizontally by
Figure 6.1  Effect of inlet subcooling of the working fluid
6 K for the same heat flux). Neither the boiling initiation wall superheat nor the remainder of the boiling curve were affected by the inlet subcooling. This agreed with the Martin-Dominguez model.

6.2 Effect of Mass Flow Rate

Figure 6.2 shows the heat flux vs wall superheat plots for mass flow rates of 0 kg/s, 0.046 kg/s and 0.092 kg/s with Z=0.17 m. The incipient boiling wall superheat was unaffected by the fluid mass flow rate. This is in agreement with the Martin-Dominguez model. As the flow velocity increased the heat flux required to initiate boiling increased because the convective heat transfer coefficient increased. During the cooling, as the flow velocity increased the wall superheat decreased for a given heat flux because the boiling heat transfer coefficient increased.

6.3 Effect of Maximum Wall Subcooling by increasing pressure between Tests

To investigate the effect of prior subcooling on the incipient boiling wall superheat, a low wall superheat was produced before each test by increasing the system pressure to either 170, 308, or 515 kPa at room temperature. The minimum wall superheats corresponding to these pressures were respectively - 16 K, - 35 K and -54 K. Each subsequent test was run with a heater position Z=0.24 m, at a pressure of 170 kPa, a refrigerant mass flow rate of 0.046 kg/s and an inlet subcooling of 13 K. The heat flux vs the wall superheat curves presented in Figure 6.3 show that, over the range of pretest minimum wall superheats studied, there was no observed effect on the boiling behaviour.
Figure 6.2  Effect of mass flow rate
Figure 6.3 Effect of maximum wall subcooling by increasing pressure between tests
This seeming contradiction of the Martin-Dominguez model may be explained by the following fact:

During the pressure gage calibration prior to all of these tests the surface was subjected to a pressure of 655 kPa which resulting in a wall subcooling of 65 K, substantially above the three values (16, 35, and 54 K) used in this study. According to the Martin-Dominguez model there should have been no effect and none was detected.

6.4 Effect of Pressure

Tests were run at Mf=0.046 kg/s, Z=0.24 m and constant inlet fluid temperature of 28 °C for system pressures of 170 kPa, 236 kPa and 305 kPa. As the system pressure increased the pretest wall superheat decreased and the inlet subcooling increased from 13 K, 25 K and 35 K respectively. However, from the results of section 6.1, the inlet subcooling effect on the boiling curve is to shift the non-boiling portion of the curves horizontally and does not affect the incipient boiling wall superheat. The results were plotted as q vs Tw in Figure 6.4a and as q vs the wall temperature Tw in Figure 6.4b. Figure 6.4a shows that the wall superheat required to initiate system boiling decreased with system pressure from approx 42 K to 37 K. This agreed with the Martin-Dominguez model. Figure 6.4b shows that higher pressures require higher wall temperatures to initiate boiling.

For the higher pressures the boiling was weaker, there were fewer active sites and the boiling did not extend over the entire surface. With each successive increase in the power input, the number of sites and boiling vigour increased slowly. Vigorous boiling
R123, \( M_f = 0.046 \text{ kg/s}, Z = 0.24 \text{ m} \)

![Graph showing the effect of pressure on wall superheat.](image)

Figure 6.4a Effect of pressure on wall superheat

---

- P = 305 kPa
- P = 236 kPa
- P = 170 kPa
- \( T_{is} = 35 \text{ K} \)
- \( T_{is} = 25 \text{ K} \)
- \( T_{is} = 13 \text{ K} \)
Figure 6.4b  Effect of pressure on wall temperature
over the entire surface was not achieved.

6.5 Effect of Minimum Wall Superheat between the Cooling and Reheating Procedures

After boiling initiation, the heat flux was incrementally increased until the heater temperature exceeded 100 °C then decreased until the last nucleate boiling site ceased then increased again. The results of such a series of tests at a pressure of 238 kPa for Mf=0.046 kg/s and Tis=24 K are shown in Figure 6.5. It was found that:

a) when the boiling surface was reheated after it was cooled to a wall superheats of Tws = 7.3 K (the site cessation wall superheat) and Tws = 4.9 K, boiling at the last site initiated at its boiling cessation wall superheat and the reheat flux vs wall superheat curves were the same as that generated by decreasing the heat flux. As the heat flux was increased, many of the same sites that were last active reactivated then more and more sites activated downstream of the reactivated sites.

b) when the system was reheated after the system was cooled to a wall superheat of Tws = 2.9 K or 2 K the last active sites did not reactivate. A much large wall superheat was required to initiate other sites. As a result, the reheat flux vs wall superheat curves were lower than the cooling curve but higher than the initial heating convective heat transfer curve. This would seem to imply that some sites that ceased boiling at a higher wall superheat than the surface boiling cessation wall superheat were not quenched and hence reactivated at their boiling cessation wall superheat. Apparently all of these sites were quenched when the wall superheat dropped to -0.6 K since for this case the reheat curve was identical to the original "cold start" convective heating curve.

69
Figure 6.5  Effect of minimum wall superheat between the cooling and reheating procedures
6.6 Effect of Heater Position (New Boiling Surface)

After all the tests at \( Z = 0.24 \) m were completed, the heater was moved to \( Z = 0.17 \) m. For both of these locations, the flow is in the fully developed turbulent flow regime (see Shivprasad 1989 and Kays 1966) and hence any differences should be due to the fresh surface being studied.

Figure 6.6 shows the results for \( Z = 0.24 \) m and 0.17 m. It should be noted that the heat flux vs wall superheat curves for the whole cycle are similar in shape and the initiation of boiling occurs at essentially the same wall superheat and heat flux.

For a given heat flux, after the system boiled the boiling was stronger and the wall superheat was lower for the fresh (\( Z = 0.17 \) m) surface.

The reason for this difference must be due to the different sites on the surface which had not previously boiled. Another possible reason is the effect of long term boiling, i.e. before charging the loop the surface was dry and all cavities on it were filled with gas. Once charged this gas would saturate with vapour and all cavities would be potential nucleation sites. During boiling this gas is slowly purged from the boiling cavities. The new surface has probably not been subject to vigorous boiling long enough to eliminate all the gas from cavities. This can explain the higher boiling performance on the new surface.

6.7 General Visual Observations

Based on visual observations during the experiments (Section 6.1 through section 6.6) the following general behaviour was observed after boiling incipience:
R123, $P = 170$ kPa, $T_{is} = 13$ K, $M_f = 0.046$ kg/s

Heat Flux $q$ (kW/m²) vs. Wall Superheat $T_{ws}$ (K)

- $Z = 0.17$ m
- $Z = 0.24$ m

Figure 6.6  Effect of heater position
For a pressure of 170 kPa, very strong boiling suddenly took place over the entire heated surface. The time interval for this to take place was so short that it could not be measured. Once the boiling began, the local wall superheat quickly dropped because of the increased heat transfer coefficient due to boiling. As the wall superheat decreased the vigour of the boiling decreased and many of the sites disappeared. This superheat drop was about 25 °C for these constant heat flux tests. Figure 6.7a shows that wall superheats at each level are essentially the same.

For the higher pressures of 236 kPa and 305 kPa, boiling initiated on only a few sites on the heating surface and, as a result, the wall superheat drop was smaller. Figure 6.7b shows that after boiling the wall superheats increase with distance downstream along the heater.

For all these tests, as the incipient boiling wall superheat was approached, smaller power input increments were used and a shorter scan interval was selected so that the state at which boiling initiated could be obtained by manually recording from the computer screen the wall superheat at all four thermocouple levels just prior to that moment.

6.8 Individual Site Boiling Behaviour

The boiling cessation and incipient quench wall superheats for several boiling sites were studied as follows: After the system boiled, the surface was cooled incrementally until the bubbles at a site disappeared. This boiling cessation wall superheat was then recorded. Any further decrease in the wall superheat was noted. The surface was then
Figure 6.7a  Heat flux vs wall superheat at heater
for a pressure of 170 kPa
Figure 6.7b  Heat flux vs wall superheat at heater for a pressure of 236 kPa
reheated incrementally to determine if the site would reactivate. If so, the incipient boiling wall superheat was recorded again. It was noted that the incipient boiling and boiling cessation wall superheats for a given site were identical. This procedure was repeated with even more cooling until the site would no longer reactivate.

The recorded wall superheat below which boiling would not reactivate was recorded as the quench superheat/subcooled for the site. In these tests each site ceased boiling at its own unique wall superheat. Those that reactivated at the boiling cessation wall superheat for the site when reheated were considered to be dormant over that range of wall superheat between the cessation and the quench values.

The individual behaviour for each of several nucleation sites was studied by observing their reactivation, cessation and finally quench wall superheats. Once these last sites were quenched, they did not reactivate on their own during reheating.

The last boiling sites studied were closest to thermocouple B at level 2. Incipient boiling (reactivation)/cessation and incipient quench wall superheats at this position for sites number 2 through 7 are summarized in Table 6.1.
Table 6.1  Cessation and Quench Wall Superheat $T_{ws} \ (K)$ for six sites

$M_f=0.046 \ \text{kg/s, } P=170 \ \text{kPa, } T_{is}=14 \ \text{K}$

<table>
<thead>
<tr>
<th>Site Number</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
</tr>
</thead>
<tbody>
<tr>
<td>Incipient Boiling Wall Superheat</td>
<td>2.9</td>
<td>8.6</td>
<td>2.7</td>
<td>6.3</td>
<td>3.7</td>
<td>5.5</td>
</tr>
<tr>
<td>Boiling Cessation Wall Superheat</td>
<td>2.9</td>
<td>8.5</td>
<td>2.6</td>
<td>6.3</td>
<td>3.7</td>
<td>5.5</td>
</tr>
<tr>
<td>Incipient Quench Wall Superheat</td>
<td>0.6</td>
<td>4.5</td>
<td>0.6</td>
<td>1.6</td>
<td>0.6</td>
<td>4.0</td>
</tr>
</tbody>
</table>

As shown in Table 6.1, each nucleation site has its own unique boiling behaviour.

The geometric shapes of real sites are unknown and are most likely all unique. It is thus impossible to directly compare the results from an experimental test, such as those in Table 6.1, with the predicted values for conical re-entrant cavities of the I. Martíndominguez theoretical model. It does however verify that (a) for a site the incipient boiling and boiling cessation wall superheats are identical. (b) a site has its own quench value. Once the wall superheat is below this value, the site will not reactivate by itself. (c) sites which boil at low wall superheats (eg. all sites listed in table 6.1) quench easily
when cooled.

After all of above sites quenched, the system behaves essentially as it did from the initial cold startup (ie. high wall superheat is required to initiate boiling in the system). Upon cooling the re-boiled system and noting the last active sites it is apparent that they are usually different from the ones studied in Table 6.1. This implies that raising the temperature well above the incipient boiling wall superheat for a site does not ensure its reactivation. Apparently another mechanism is required to activate quenched sites.

Immediately after boiling starts, the vigour of the boiling decreases, the wall superheat decreases, and many of the sites disappear. Since the initiating sites are now subject to a lower wall superheat they should stop boiling. Since boiling does not cease other nearby sites must have been initiated by seeding from the initial active sites. This would be expected to occur at sites located down stream of active sites. Consequently, individual site behaviour is not independent of the presence of other sites. Our efforts to determine the characteristics of an individual site were limited to only those sites that can operate at low wall superheat. The Martin-Dominguez theoretical model can be used to explain the observed phenomenon in the real test quite well. Predictions explain that: Once nucleate boiling is initiated at one site, its bubbles can seed other adjacent quenched sites with which they come in contact. This expansion (to sites that require less superheat to maintain boiling), causes the wall superheat to decrease. This decrease causes the initial boiling sites (and others that require larger wall superheats) to either become dormant or quench (Martin-Dominguez et al. 1997).
7. CONCLUSIONS AND RECOMMENDED FUTURE WORK

7.1 Conclusions

1. The inlet subcooling of R-123 has no effect on the boiling incipience wall superheat of a surface.

2. The mass flow rate of R-123 has no effect on the boiling incipience wall superheat of a surface.

3. Increasing the system pressure causes the boiling incipience wall superheat of a surface to decrease.

4. Each nucleation site has its own unique boiling behaviour.

5. Upon cooling, a nucleation site becomes dormant at its cessation wall superheat and may quench at a lower wall superheat. A dormant site is easy to reactivate whereas one which quenches requires seeding by vapour from other boiling sites and a wall superheat above its incipient boiling value in order to start boiling.

6. A dormant site will resume boiling at the same wall superheat at which it ceased boiling (i.e., incipient boiling wall superheat and incipient cessation of boiling wall superheat are identical for a site which becomes and remains dormant).

7. The boiling activity of a site can and usually does affect the boiling behaviour of nearby sites. The vapour of an active bubble can initiate (or seed) other sites with which it comes in contact.
8. The past thermal history of a surface/fluid interface determines the subsequent incipient boiling wall superheat of a surface (i.e., the effect of the minimum wall superheat (maximum wall subcooling) prior to heating the surface).

9. Although the prediction of boiling initiation on real surfaces is much easier to investigate based on statistics rather than a study of individual sites, studies of individual site behaviour provide a very useful insight into the nature of surface boiling initiation and the factors which affect it.

10. These test results are compatible with the bubble/cavity model of Martin-Dominguez (1993).

7.2 Recommended Future Work

1. Study the effect on the system boiling performance of dissolved air in the R-123.

2. Investigate various methods of erasing the thermal memory of a heating surface. (e.g., by raising the wall superheat to a very high value or by drying out the surface by temporarily lowering the liquid level in the test facility.)

3. Relocate the flow control valve to a site between the flow meter and the pre-heater.
REFERENCES


Martin-Dominguez, I.R. Onset and cessation of nucleate boiling. PhD Dissertation,


You, S.M.; Simon, T.W. and Bar-Cohen, A. *Experiments on boiling incipience with a


VITA AUCTORIS

Name: Li Liang

Place of birth: Jilin city, Jilin province, P.R. China

Date of birth: January 29 of 1960

Education:
1978-1983 Bachelor of Engineering, Department of Automotive and Thermal Engineering, Tsinghua University, Beijing, China.

1983-1986 Master of Environmental Engineering, Chinese Academy of Preventive Medicine, Beijing, China.

1993-1997 A Candidate for the Degree of Master of Applied Science (Mechanical Engineering) at University of Windsor, Windsor, Ontario, Canada.

Professional Experience:
1986-1992 Mechanical Engineer, Institute of Environmental Health and Engineering, Beijing, China.

1992- Present Research and Teaching Assistant, Department of Mechanical and Material Engineering, University of Windsor, Windsor, Ontario, Canada.
APPENDIX A. EXPERIMENTAL PROGRAM

This appendix is a report by Martin-Dominguez (93) on experimental work done but not reported in his theses (93). It is reported here because of its relevance to this work and to ensure it is readily available to other researchers.
A. Introduction.

An experimental program has been implemented to study the onset of nucleate boiling in subcooled vertical annular flow, using refrigerant R-11 as a working fluid. The purpose of the experiments is to validate some aspects of the model to be developed. The behaviour of the ONB, to be studied during the experiments, is:

- Is the wall superheat at the ONB independent of flow velocity and subcooling for a well wetting fluid?
- The survival of nucleation sites should depend on the degree of (sub) cooling imposed on the heat transfer surface during non-boiling periods.

The test surface to be used is a cold rolled copper tube, initially cleaned with acetone. The test section is an annular flow section formed by the copper tube and an external glass tube, located in a forced flow closed loop. The heat source is an electrical heater cartridge assembly, that could be located anywhere along the length of the inner surface of the test tube.

The variables to be controlled are:

- Heat flux
- Mass flux
- Subcooling
- Pressure

The monitoring and measurement of temperatures and pressures is done using a computer-driven data acquisition system. A video system is used to visually observe the phenomena.

B. Experimental apparatus.

An existing flow loop was modified to serve for the purposes of this work. The main components
of this apparatus are described in the following paragraphs.

B.1. Subcooled flow boiling loop.

The constructive characteristics of the flow loop are shown schematically in Figure VI.1. An exterior glass tube forms an annular flow test section that permits the boiling phenomenon in the surface of a copper tube to be observed. The fluid is forced, by a vane pump, to flow through the rotameter, the flow control valve, the preheater and then through the vertical test section. After the test section the fluid passes through the condenser then returns to the pump.

The system pressure is adjustable by means of a bladder accumulator, where the balance pressure is controlled by an air pressure regulator valve.

B.2. Test section and Instrumentation.

The working fluid enters and leaves the test section laterally, perpendicular to the axis of the test tube, thus permitting the test specimen to have both ends open to the atmosphere, as can be observed in Figure VI.1 and Figure VI.2.
The ends of the glass test section tube are attached to brass "T" sections by means of tapered Teflon fittings and held in place by bolted cast iron clamps, supplied for use with the QSF glass tube. The test tube is held in position by a copper end cap on the "T" section which was bolted in place. The clearance between the tube and the end cap were sealed with Buna-N "O" rings.

An electric heater, assembled inside a copper cartridge as shown in Figure VI.3, is used to heat the test section. The assembly is firmly attached to the tip of a push rod and fits tightly inside the test tube. It can be positioned anywhere along the length of the test specimen as is shown in Figure VI.2.

The system pressure is measured using a gage pressure transducer and two bourdon gages, connected to the test section inlet. The readings from the pressure transducer are directed to the data logger.

The pressure drop in the test section is monitored with a U-manometer, using mercury at an inclination of 30 degrees with respect to the horizontal. The pressure differential between the inlet and the outlet sections of the test annulus was measured for several flow runs, where the values of pressure drop and flow rate were carefully recorded. A correlation was then obtained using a linear regression method and incorporated into the data reduction program to provide a fast method for reporting the pressure drop.
B.3. Heater assembly and Instrumentation.

The bulk temperature of the working fluid is measured by three type "T" thermocouples, located at the inlet and outlet of the test section and after the condenser. The junctions of these thermocouples are in physical contact with the flow, the thermocouple wires enter the flow loop through threaded caps where an epoxy cement is used to seal and keep them in place.

The heater cartridge sheath surface temperature is measured by 16 thermocouples, mounted at 90° intervals around the heating cartridge assembly as shown in Figure VI.3.

The cross section of the test annulus is shown in Figure VI.4, where the location of the thermocouples, the heater assembly and the glass tube can be appreciated.

B.4. Video monitoring.

A video camera with microscope lens was installed to observe the heat transfer surface and monitor the evolution of the boiling sites. Figure VI.5 Shows the video camera and mounting frame.

B.5. Data acquisition system.

All the thermocouples and the pressure transducer are connected to a electronic data acquisition system, that permits the pressure and temperatures to be monitored in real time.
The system consist of a computer-controlled multichannel electronic data logger connected to an IBM compatible micro computer, which also performs simultaneous data reduction and shows the results on the screen in real time.

The system is composed of the following major parts.
B.5.a. Data logger.

The data logger is a "Sciemetric Instruments" Electronic Measurement System, model 8082A, which is basically a 64 channel voltameter that converts the analog signal from the measuring transducers into a digital signal, readable by the computer.

The system is especially suitable for temperature measurements using thermocouples since the block of connectors is maintained at a fairly constant temperature (room temperature), and by installing a thermistor in one of them the block temperature can be directly obtained. This temperature is the "Cold Junction" or reference temperature and the arrangement eliminates the requirement of using the customary "Cold Bath", normally water in an ice bath.

The software utilized to communicate with the logger and monitor the experiment consists of two main parts, described in the next paragraphs.

B.5.b. Data acquisition routines.

The programming code required to communicate between the micro-computer and the data logger was provided by the logger's manufacturer in the form of an initialization subroutine and a set of sensor-specific subroutines, written in BASIC language. It reads the desired values from memory locations and converts them to voltages and/or temperatures.

B.5.c. Monitoring program.

A menu-driven computer program was developed that performs the following operations:

- Controls the timing of the readings.
- Performs the data reduction.
- Permits monitoring of several sets of important variables on the screen, both raw measurements and calculated values.
- Permits storage on disk of selective data and results, attaching a time stamp and a short commentary to each set.
- Provides a means to quickly change values for any of the system parameters that must be set using the keyboard.
> Provides the user with an on-line calculation of saturated thermodynamic and transport properties for the working fluid.

The program is written using the BASIC programming language. The data files produced can be directly imported into any electronic spreadsheet for graphical analysis.

C. Thermodynamic and transport property correlations.

For the data reduction process it is necessary to have accurate correlations for all the thermodynamic and transport properties involved in the calculations.

The availability of such equations was found to be restricted to few properties and then often with poor agreement with published numerical values.

For those reasons it was decided to correlate the numerical values published by ASHRAE (1989) for the required properties.

The thermodynamic and transport properties correlations for R-11 used in this work were already published elsewhere, Martín-Domínguez and McDonald (1992).