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BOILING INITIATION AND HYSTERESIS -TEST FACILITY

BY

© DINKAR SHIVPRASAD

A thesis

submitted to the

Faculty of Graduate Studies and Research

through the Department of

Mechanical Engineering in partial fulfillment

of the requirements for the Degree of

Master of Applied Science at the

University of Windsor

Windsor, Ontario, CANADA

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ABSTRACT

A forced convection heat transfer loop was designed to study the wall superheat necessary to initiate nucleate boiling, the wall superheat and heat flux at which nucleate boiling is quenched, and the hysteresis associated in a plot of heat flux versus local wall superheat during cycles of increasing followed by decreasing heat flux, using R-11 and R-113 as working fluids.

The test section flow is upward through a vertically oriented, concentric annulus, formed by a 22.2 mm outside diameter copper pipe, within a 25.4 mm inside diameter glass tube, 0.4 m long. The copper pipe is heated internally by a 76 mm long instrumented cartridge heater, which could be positioned anywhere along the length of the annulus test section.

The following system parameters can be varied within the ranges shown below:

1) System pressure (2 to 7.9 bars),

2) Inlet temperature (20 degrees C to the lesser of the saturation temperature or 100 degrees C),

3) Reynolds number in the test section (2900 to 25000 for R-11 and 2500 to 22000 for R-113),

4) Heat flux at the test surface (0 to 60 kW/m2).

A computer assisted data acquisition program was written to enable fast collection of information and in situ data reduction.

Preliminary tests were conducted with R-11 at 1.5 and 1.9 bar system pressures, 3 and 9 degrees C subcooling respectively and

a Reynolds number of 3250.

It was observed from these preliminary runs that :

1) Boiling initiation required a finite, significant local wall superheat and the initiation was vigorous, all around the test surface. At higher flow rates and system pressures, the initiation becomes less vigorous.

2) Upon initiation, the wall superheat drops abruptly due to the higher heat transfer coefficient associated with boiling. For these tests, the wall superheat dropped to half of their previous value.

3) Increasing the wall heat flux with one or more sites already active results in a much different heat flux versus wall superheat relationship than for either cold startup or for cooling.

4) The disappearance of boiling at a site does not mean the site is quenched. Under some conditions, it will reactivate without the very large wall superheats required to initiate boiling from a quenched site.

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My final and most dearest expression of gratitude is to my Guru Sri Raghavendra Swami, my wife, parents, brother and sisters. Their inspirational encouragement and everlasting love and support made this endeavour possible.

Dedicated with love to

my Guru Jagadguru Sri Raghavendra,

my wife, son and my parents.

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NOMENCLATURE

A	Parameter defined in equations A9, A10
A_x	Cross sectional area (m^2)
B	Flange inner diameter (m)
C_p	Specific heat (kJ/kg C)
D	Diameter (m)
f	Darcy's friction coefficient
f	Tensile strength (N/m^2)
g	Acceleration due to gravity (m/s^2)
G	Mass flux ($kg/s\ m^2$)
h	Heat transfer coefficient (W/m^2 C)
h_{fg}	Latent heat of vaporization (kJ/kmol)
K	Thermal conductivity (W/m C)
l	Equivalent length (m)
L	Cooler tube length (m)
$\dot{m}$	Mass flow rate (kg/s)
p	Power (W)
P	Pressure (Bars)
Pr	Prandtl number
q	Heat flux (W/m^2)
Q	Volumetric flow rate (m^3/s)
r	Fouling coefficient
r, r_c	Cavity radius(m)
Re	Reynolds number
R	Universal gas constant (kJ/kmol K)

T	Temperature ($^{\circ}K$)
t	Temperature($^{\circ}C$)
u	Velocity (m/s)
U	Overall heat transfer coefficient (W/m^2 C)
y	Distance from wall (m)
Y	Parameter in Appendix-G

GREEK SYMBOLS:

β	Wedge angle (degrees)
δ	Thermal boundary layer distance (m)
Δ	Difference
ϵ	Heat exchanger effectiveness
λ	Parameter in equation A5
μ	Dynamic viscosity (kg/m s)
Ω	Ratio of flange diameters
θ	Contact angle between solid and bubble surface (degrees), measured through the liquid
ρ	Density (kg/m^3)
σ	Surface tension (N/m)
υ	Specific volume (m^3/kg)

Subscript:

ave	Average value
f,l,liq	Liquid
fr	Refrigerant
h, hyd	Hydraulic

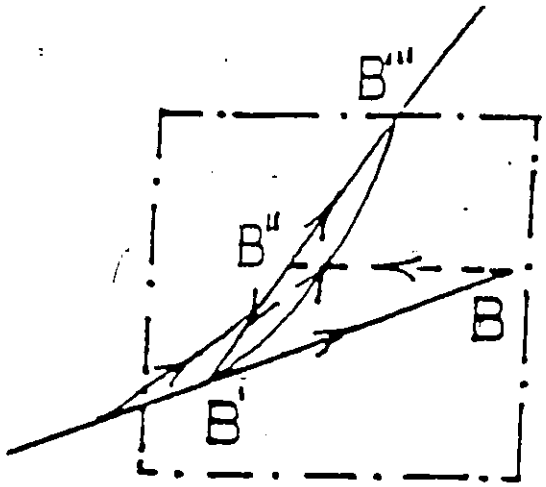
incp	Incipient
oo	Beyond the boundary layer
min	Minimum
s	Saturation
sub	Subcooling
1,ss	Shell side
2,ts	Tube side
g,v	Vapour
w	Wall, Water
1	Inlet
2	Outlet

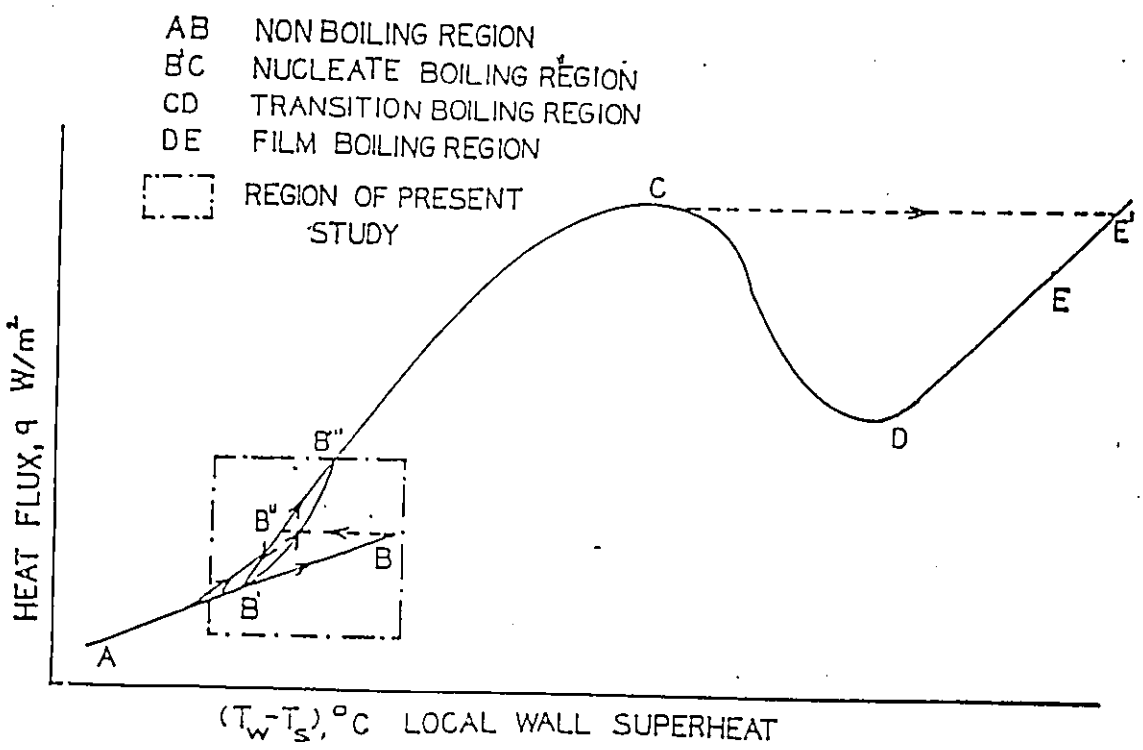
1. INTRODUCTION

The process of evaporation associated with vapour bubbles in a liquid is called Boiling.

Boiling heat transfer has been extensively studied in recent years because of many engineering applications which require high heat transfer rates under moderate temperature differences (viz.,nuclear reactors, rocket motors). The heat generated, in such applications must be removed to preserve the structural stability of the equipment. One method of removing this would be by single-phase forced convection flow. This, however, would require a large quantity of coolant and a large pump and motor. The same objective can be realized with a much lower flow rate if boiling is allowed (heat transfer coefficients typically ten times the single phase value). Also, uniform surface temperatures have been achieved in high speed computer components by their cooling them with boiling fluids.

One aspect of subcooled flow boiling which is not fully understood is the transition between convective heat transfer and nucleate boiling. When subcooled liquid is heated by an electrically heated wire or a flat plate, data for heat flux q versus local wall superheat ($T_w - T_s$), usually appear as shown in Figure 1.1. As the heat flux is increased, the initial forced convection regime A to B, changes to nucleate boiling regime. The appearance of the first bubble at B requires a significant finite local wall





1.1 TYPICAL BOILING CURVE

superheat (typically greater than 10 degrees C). The superheat necessary to initiate nucleate boiling is known as the Incipient Nucleate Boiling Wall Superheat. However, once boiling has been initiated, there is a significant and abrupt fall in the local wall superheat, (Figure 1.1, from B to B''). (Most textbooks present a simplified account of the boiling curve, showing a smooth transition from A to B'', ignoring the B to B'' transition). As $(T_w - T_s)$, or the heat flux, is increased, more nucleation sites become active producers of bubbles. A peak nucleate boiling heat flux is reached at point C. At this point, if any further increase in heat flux occurs, the bubbles stream forth from so many sites that the liquid is unable to flow to the heated surface and film boiling takes over at point E', which is at a much higher wall superheat. Operation in C-D-E region is achieved only if wall superheat is the independent variable.

If we decrease the heat flux from point C, it is observed that the curve C to B'' to B to A is not retraced. Instead, the path C to B'' to B' to A is followed. At B', the last nucleation site disappears. The superheat at which a nucleation site disappears is known as the site Nucleate Boiling Dormancy Wall Superheat. Nucleation sites which are dormant will reactivateup on a much smaller increase in wall superheat than that required to initiate a quenched site. When such sites reactivate, a smooth transition occurs from state B', to state B''', as the dormant sites initially reactivate then subsequently, the vapour streams from these sites reactivate others downstream. At B''' much of the

surface is still not boiling. If the liquid is further subcooled below B', all the nucleation sites may become quenched, depending upon the amount of subcooling. When this happens, a wall superheat, the same as that for "cold" startup will be required to reactivate the site. In this case, path B' to B to B'' will be retraced. The superheat at which a nucleation site is quenched is known as the site Nucleate Boiling Quench Wall Superheat. Between these two extremes of all nucleation sites being dormant or, all sites quenched, depending on the amount of subcooling after dormancy, we have cases of some sites being dormant and others quenched. Depending on their relative numbers, we have different boiling transition curves from non boiling to nucleate boiling region.

Thus we observe that the wall superheat necessary for boiling incipience is quite different from the superheat at which the site disappears, and the loop B' to B to B'' to B', is an example of a Boiling Curve Hysteresis.

McDonald et al. [14], in order to simulate the experimental results of their previous work [13], formulated a Quench/ Dormancy Model, in which they assumed that when the local wall superheat falls below some fraction, of the superheat at which a site becomes dormant, then the nucleation site is quenched, otherwise the nucleation site is dormant. A subsequent increase in wall superheat will cause a reactivation of dormant sites, but the quenched sites will require a local wall superheat

approximately equal to that required to initially activate them.

The objective of the present study is to design, test and commission a forced convection heat transfer loop which may be used to study individual nucleation sites for the following :

1) the incipient nucleate boiling wall superheat,

2) the nucleate boiling dormancy wall superheat, and nucleate boiling quench wall superheat,

3) the hysteresis associated with the above phenomena for cycles of increasing followed by decreasing heat flux,

4) the verification of the quench/ dormancy model,

so that a better understanding of these phenomena and their mechanisms is achieved. Once understood, one would be able to develop an engineering model which would allow these phenomena to be simulated on a computer.

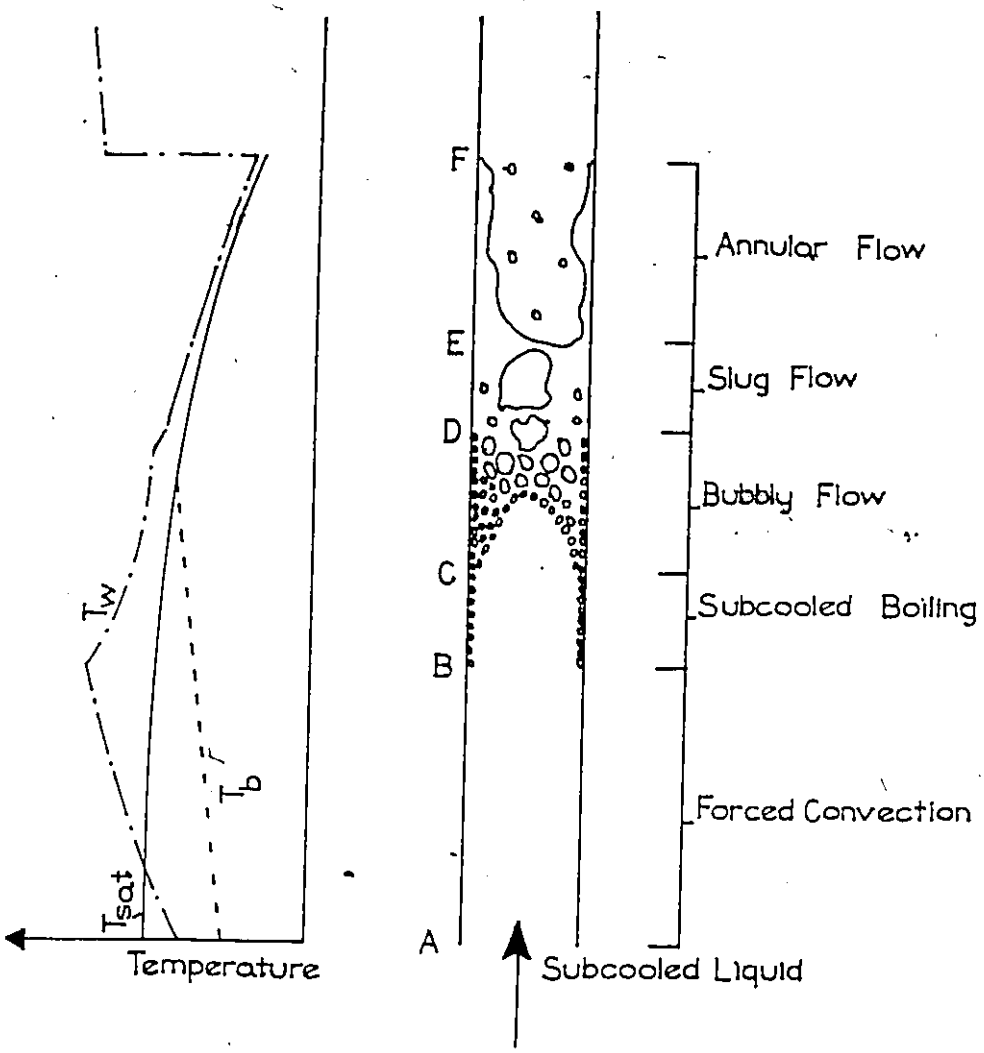
A review of literature in the areas of incipient boiling wall superheat, hysteresis in the transition region from forced convection heat transfer to nucleate boiling is presented first. In the following chapters, the heat transfer loop design and commissioning, the required instrumentation for the loop, the results obtained from preliminary runs, and conclusions from the present study and recommendations for further study, are discussed.

2. LITERATURE REVIEW

2.1 Preamble

2.1.1 Flow Regime Up To Incipient Boiling For A Subcooled Liquid In An Uniformly Heated Channel

In order to understand the behaviour of individual nucleation sites in flow boiling, it is necessary to know the flow regimes that occur for a subcooled liquid in an uniformly heated channel. In Figure 2.1, subcooled liquid enters the bottom of a uniformly heated vertical tube. From A to B, the heat transfer is entirely by single phase forced convection, since the wall temperature, though higher than saturation temperature, is still below the value necessary for bubble nucleation. At B, the wall temperature reaches a superheat high enough to initiate nucleation. There is a sharp drop in local wall temperature at each site. At B, the bulk liquid is still highly subcooled, so vapour bubbles are confined to the wall region, often sliding along the wall as they grow. Bubbles that detach recondense, as they enter the main stream. At C, the bulk temperature, although below T_{sat}, has increased sufficiently for bubbles to exist in the main flow. At this point there is a sharp increase in the void fraction. Between C and D, T_b reaches T_{sat}, and bubbles no longer condense as they leave the wall. Bubble coalescence leads to slug flow (D toE) followed by annular flow (E to F) in which a thin liquid film covers the tube wall, while vapour, liquid droplets flow up the centre at high velocity. The regime (A to C), is the region that exists in the



2.1 FLOW REGIMES FOR BOILING IN AN UNIFORMLY HEATED CHANNEL

test section to be discussed in the present thesis.

2.2 Review Of Previous Work

A large number of research workers have contributed to our understanding of various phenomena in the "knee" region of the boiling curve. The phenomena of nucleate boiling wall superheats have been studied in both pool boiling [4,7,9] , and flow boiling [1,2,10,15,16]. Also, studies have been carried out on hysteresis in pool boiling [7,12], flow boiling [11,17] and in thermosyphon loops [13,14].

The variables affecting incipient boiling wall superheat may be classified in the following manner:

2.2.1 Surface

2.2.1.1 Geometry

Incipient boiling is insensitive to system geometry, provided the surface dimensions are larger than typical bubble sizes (1 mm for water at atmospheric pressure) and the bubbles are able to escape from the heated surface [6].

2.2.1.2 Effect Of Surface Conditions

Surface finish has a pronounced effect on incipient boiling by altering the number of nucleation sites, but at present there is insufficient material in the literature to put it in a quantitative basis. Studies have been performed involving boiling enhancement surfaces[30] (surfaces which reduce the incipient

boiling wall superheat).

Corty and Foust[7] have reported the effects of surface roughness on the boiling curve for several fluid-surface combinations. In these experiments, saturated, pool boiling of diethyl ether, n-pentane, and R-113 were investigated from an upward facing, horizontal plate which had first been plated with either copper or nickel, then roughened by rubbing with various grades of emery paper. Their results indicated a major influence of microroughness on the wall superheat necessary to initiate nucleate boiling at any given heat flux. For an n-pentane/ nickel interface, with a 0.056 μm root mean square surface roughness, nucleate boiling started at a local wall superheat of 19.67 degrees C, while a surface with 0.58 μm root mean square roughness required a local wall superheat of 28.89 degrees C. They said, this was probably because the rougher surfaces contain larger cavities, and hence are easily filled with liquid and are therefore unable to trap vapour.

Howell and Siegel[4] studied the effect of the size and geometry of nucleation sites for a stainless steel AISI 410, water combination, in pool boiling. Two types of artificial sites were studied. The first was drilled by electron beam and the site diameter was 63.5 μm and the second type was 76.2 μm diameter produced by a mechanical drill. In most of the test strips, two sites of the same nominal diameter were drilled about 25.4 mm apart. Three strips were tested with different site diameters, to

observe the effect of site diameter under identical test conditions. Low heat fluxes were used near incipience so that it was possible to grow isolated bubbles. They concluded that larger cavities are filled with liquid before vapour replacement can take place, making the cavities inactive.

2.2.2 Role Of Surface Tension And Wettability

Bankoff [8] gives criteria to characterize conical or wedge type cavities (with β, conical or wedge angle as the case may be, refer to Figure 2.2) as to their abilities to trap vapour. The ability to trap vapour depends to a large extent on the contact angle θ, between the bubble interface and the solid wall measured through the liquid, and the wedge or cone angle as the case may be. Angle θ, is determined by the force balance between the surface tension forces of the vapour/ liquid/ solid interface.

The above mentioned cavities can be classified in one of four categories, depending on whether or not they obey the following inequalities:

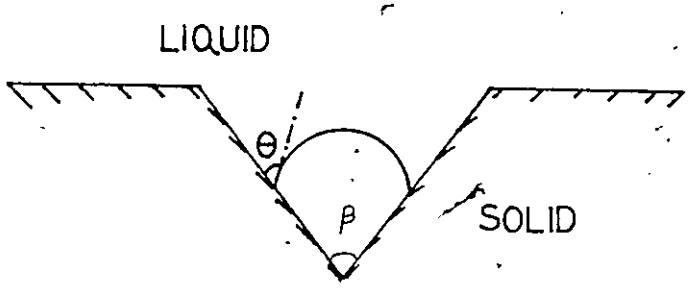
$$\theta > \beta$$

and

$$\beta > \theta - \pi$$

1) Those which obey the first inequality but not the second (Poorly wetted, shallow)-vapour entrapment not possible.

2) Those which obey the second inequality but not the first (Well wetted, steep slope)-possible to switch from liquid



2.2 WEDGE TYPE CAVITY

to gas filled cavity.

3) Those which obey both (Poorly wetted, steep slope)- complete displacement of liquid or gas, once filled is not possible.

4) Those which obey neither (Well wetted, shallow)- entrapment of vapour not possible.

2.2.3 Influence Of Pressure Temperature History

Sabersky and Gates[9] performed experiments with 2.54 mm diameter platinum and nichrome wires which were immersed in a vessel filled with water. The assembly was subjected to a pressure of 1035 bars for periods in excess of 15 minutes. The pressure was then reduced to atmospheric and the wire heated electrically. They found that the superheat for boiling incipience was 35 degrees C for a distilled water/nichrome wire combination and about 63 degrees C for a tap water/platinum wire combination. Further, they ran tests with unpressurized water and concluded that the pressurization treatment eliminated the more effective sites, through gas dissolution or vapour condensation. Once boiling has begun, and the surface has again been exposed to vapour masses, the larger cavities will reactivate and further boiling will occur at surface temperatures encountered before pressurization.

Cher. [10] performed experiments to determine the effect of subcooling and prepressurization on incipient nucleate boiling superheat. The measurements were made for potassium in forced

convection flow. He found that increasing subcooling or prepressurization causes higher subsequent superheats at boiling inception. In his experiments, the subcooling was between 0-220 degrees C, prepressurization between 0-275 bars and the measured superheats ranged between 9.5-65 degrees C.

2.2.4 Mathematical Models In Boiling Nucleation

Each of the above mentioned studies have contributed to the understanding of the parameters which influence bubble nucleation but a quantitative relationship between the basic surface parameters and the heat transfer was not found. In the work of Hsu[2], Bergles and Rohsenow[15], Davis and Anderson[16], and Sato and Matsumura[1], an attempt was made to establish a relationship between cavity radius, wall temperature, and heat transfer rate to determine the range of cavity sizes that qualify as active sites and predict the nucleate boiling dormancy. Readers may please note that, all the above mentioned authors [1,2,15,16] have erroneously referred to nucleate boiling dormancy as incipient nucleate boiling in their literature. All of them assume that a vapour bubble is already present, and give the criteria for bubble growth. Refer to Appendix-A for further details on the analytical model developed by Hsu, and Bergles and Rohsenow.

Hsu[2] postulated that the bubble nucleus grows only when the surrounding liquid is sufficiently superheated, so that there is a net heat flux transferred into the bubble to provide the heat of vaporization. He calculated the local liquid temperature by one

dimensional transient conduction in a thermal layer. He proposed nucleate boiling dormancy condition as:

$$q = \frac{k\,[\zeta + \xi + (2\zeta + \xi)\,\xi\,]}{\delta}$$

where, $\zeta = (T_s - T_b)$ and

$$\xi = \frac{4\,\sigma\,T_s(1 + \cos\theta)}{h_{fg}\,\rho_v\,\delta}$$

To determine the heat flux by his method, it is necessary to know the contact angle θ and thermal layer thickness δ, both of which are in general difficult to determine.

Bergles and Rohsenow [15] assumed a steady temperature gradient and adopted the procedure developed by Hsu to develop a criterion for nucleate boiling dormancy heat flux for surfaces with a wide range of cavity sizes.

From their experiments for incipient nucleate boiling they arrived at the following correlation for water over a pressure range of 15-2000 Psia.

$$q = \frac{15.6\,P^{1.156}\,(T_w - T_s)^{2.3}}{\sqrt{P^{0.0234}}}$$

where q, is in Btu/hr ft2; P is in Psia and T is in degrees F. They employed a graphical method from which they predicted incipient nucleate boiling dormancy superheat and the cavity size that is active, before nucleate boiling dormancy begins.

Kenning [6] suggests that using the procedure of Bergles and Rohsenow, we can predict incipient nucleate boiling superheat, if we assume 1 μm as optimum cavity size.

Sato and Matsumura[1] proposed an analytical formulation similar to Hsu's for prediction of nucleate boiling dormancy. They assumed:

1) That the temperature profile in the thermal layer is steady and linear,

2) The vapour bubble is hemispherical and its radius is half of the superheated layer thickness.

They obtained the following equation for nucleate boiling dormancy heat flux,

$$q = \frac{k\, h_{fg} (T_w - T_s)^2}{8\, \sigma\, T_s\, (v_v - v_l)}$$

An analysis of nucleate boiling dormancy was developed by Davis and Anderson[16] as a modification and extension of previous analyses. They assumed a truncated spherical bubble, and derived the following equation for the prediction of subcooled boiling of water in forced convection

$$q = \frac{k\, h_{fg}\, \rho_v\, (T_w - T_s)^2}{8\, \sigma\, T_s\, (1 + \cos\theta)}$$

where θ, is the contact angle between bubble and the surface.

They used the data from Sato and Matsumura[1], for comparison with their analysis and ,it may be concluded that their analysis predicts the local wall superheat required for nucleate boiling dormancy.

It may be concluded that, the thermodynamic requirements considered by Bergles and Rohsenow, Hsu, Davis and Anderson, Sato and Matsumura are capable of predicting nucleate boiling dormancy superheat and there are no models available in the literature for predicting incipient nucleate boiling wall superheat, and nucleate boiling quench superheat.

2.2.5 Hysteresis Effects

Corty and Foust[7] observed the difference in behaviour of nucleation centers under certain conditions, depending upon whether the heat flux was increasing or decreasing. While increasing heat flux, the wall superheat necessary for boiling inception was higher than that required to maintain boiling during decreasing heat flux.

Abdelmessih[11] et al. experimentally observed hysteresis in an electrically heated, horizontal, stainless steel AISI 304 tube test section using R-11 in forced flow. A wall superheat of 13.8 degrees C to 16.7 degrees C above the local saturation temperature was found necessary to initiate boiling in their test facility. Once local boiling was initiated, the wall superheat dropped substantially. This nucleate boiling could be sustained even when

the wall heat flux was reduced to the point where the tube wall temperature fell within a few degrees of the local saturation temperature.

Joudi and James[12] carried out investigations on a flat horizontal stainless steel surface of known roughness. The system pressures for the tests were 0.25 bars, 0.5 bars, and 1.0 bar. At 1 bar system pressure, they observed that wall superheats of approximately 16 degrees C for R-113 and 19 degrees for methanol were required to initiate boiling.

Hino and Ueda [17] carried out the study on upward subcooled flow boiling in an annulus with a uniformly heated inner tube. The test fluid was R-113. The wall temperatures were measured and photographic observations carried out. They made the following observations:

1) The wall superheat at the position of incipient boiling is relatively high, and a sharp drop in wall temperature takes place after boiling initiation.
2) The wall superheats at the incipient boiling condition are virtually independent of the mass flux and the inlet subcooling.
3) Hysteresis occurs for complete cycle of increasing followed by decreasing heat flux.

Stauder and McDonald[13] studied the onset of nucleate boiling of R-11 for a thermosiphon heat exchanger. The experiments were

carried out on a 4 loop, 2 row of tubes per loop, prototype commercial air-to-air thermosiphon heat exchanger. They found that, for an R-11/ copper interface, a hot duct to cold duct temperature difference of 16 degrees C to 17 degrees C was required to initiate boiling for operating pressures between 1.2 bars and 1.6 bars. Once initiated, however, boiling was sustained until the air-to-air temperature difference decreased to 4 degrees C. This hysteresis behaviour was exhibited when operating with temperature differences below 50 degrees C. Upon reheating, in this region, a portion of the nucleate sites previously made inactive reactivated easily whereas others were completely quenched.

McDonald[14] et al. simulated the experimental results of their previous work[13], and using a quench/ dormancy model, found good agreement with the experimental results in the hysteresis region. In that model, they assumed that when the local wall superheat falls below some fraction, say P, of the minimum wall superheat necessary to maintain nucleate boiling, then the nucleation sites are quenched, otherwise the nucleation sites remain dormant. A subsequent increase in wall superheat will cause a reactivation of the dormant sites, but the quenched sites will require a local wall superheat approximately equal to that required to initially activate them.

In conclusion we might say that the subcooled flow boiling is a complicated nonequilibrium state involving the vapour bubbles and the subcooled liquid. Though each of the above mentioned studies

have contributed to our understanding of the various phenomena in the "knee" region of the boiling curve, subcooled boiling phenomena have not yet been understood completely in detail. None of the previous investigators have developed models for :

1) Incipient nucleate boiling wall superheat,

2) Nucleate boiling quench wall superheat,

3) Conditions under which a nucleation site is dormant, and the conditions under which it is quenched. The present study was undertaken to throw additional light on the behaviour of individual nucleation sites and the heat transfer in subcooled flow boiling, so that models can be developed which will allow these phenomena to be simulated on a computer.

3. EQUIPMENT DESIGN, INSTRUMENTATION AND COMMISSIONING

3.1 Design Objectives

A heat transfer loop was designed and commissioned to study the following phenomena:

1) Incipient boiling wall superheat,

2) Boiling dormancy and quench wall superheats,

3) The hysteresis associated with a plot of heat flux versus local wall superheat, for a complete cycle of increasing followed by a decreasing heat flux. The ultimate overall objective is to gain a better understanding of these phenomena so that a practical quench/ dormancy model may be established which may satisfactorily predict the hysteresis phenomena.

The loop was designed so that these phenomena could be studied as a function of the following parameters :

1) Heat flux at the test surface,

2) System pressure at the test section inlet ,

3) Subcooling at the test section inlet ,

4) Reynolds number in the test section ,

5) For both a developed or a developing flow field at the test section.

In order to observe the boiling, it was decided that the flow be annular between an inner heated surface and an outer glass pipe.

For data evaluation and analysis, it was decided that the

following parameters be controlled:

1) System pressure ,

2) Volumetric flow rate ,

3) Heat flux ,

4) Location of heater ,

and the following parameters be calculated or measured:

1) Inlet subcooling at the test section ,

2) Reynolds number in the test section ,

3) Incipient boiling wall superheat ,

4) Boiling quench wall superheat ,

5) Incipient boiling heat transfer coefficient ,

6) Boiling quench heat transfer coefficient ,

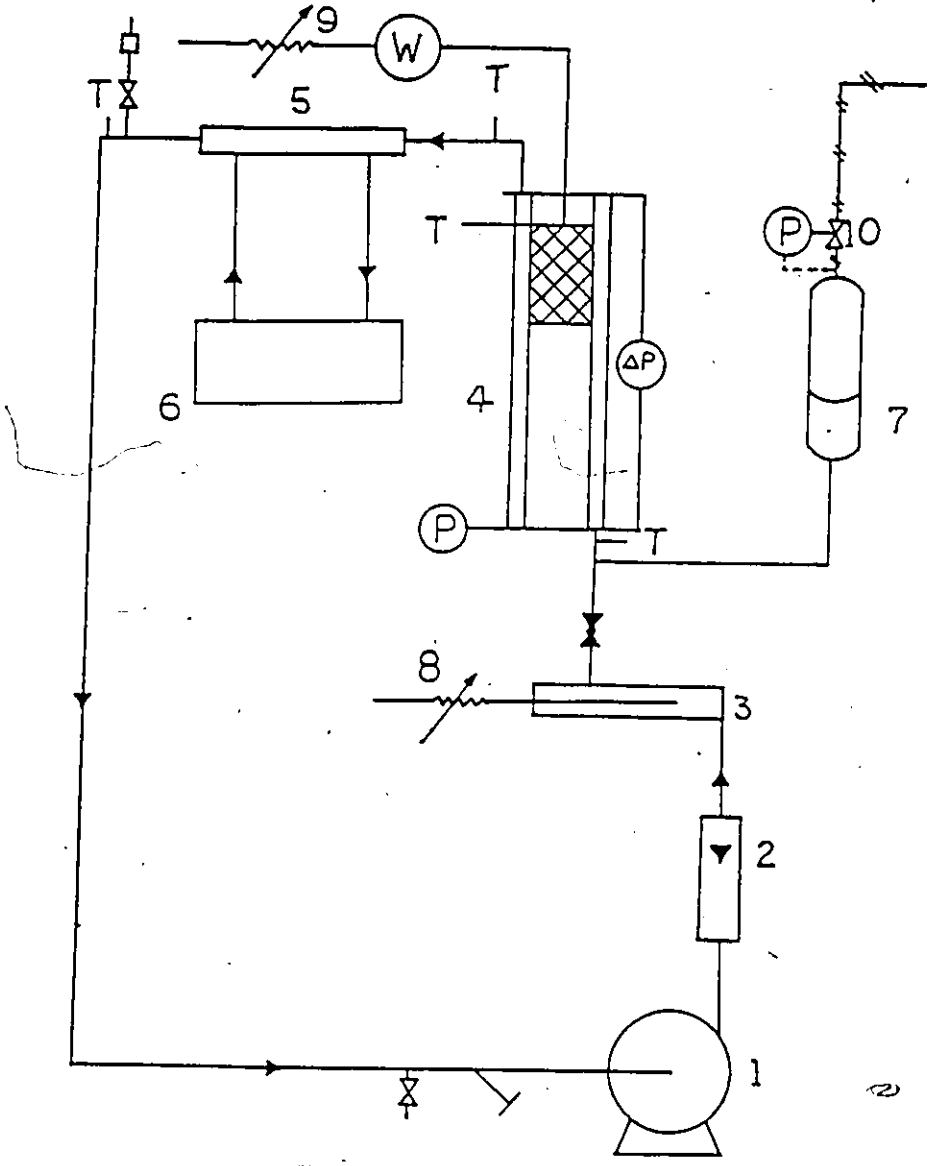
7) The location of both incipient boiling , and of boiling dormancy.

3.2 General Description Of The Loop

A schematic of the heat transfer loop is shown in Figure 3.1. The flow circuit is a closed circulation loop and is in one vertical plane. Figure 3.2 shows overall views of the test assembly.

The loop is designed to operate at a maximum working pressure of 7.9 bars (100 Psig) and a maximum working temperature of 120 degrees C. Refrigerant-11, with a latent heat of vaporization of 37.53 kJ/kg and a boiling point of 23.63 degrees C at 1 bar pressure was chosen as the working fluid.

The design details of each component is presented in the following



1 CIRCULATION PUMP
2 R-11 FLOW METER
3 ELECTRIC PREHEATER
4 ANNULAR TEST SECTION
5 WATER COOLED CONDENSER
6 T=C CIRCULATING WATER BATH
7 AIR MODULATED PRESSURIZER
8,9 VARIACS

10 AIR PRESSURE REGULATOR
T COPPER CONSTANTAN THERMOCOUPLE
P PRESSURE GAGE
ΔP DIFFERENTIAL PRESSURE GAGE
W WATTMETER

3.1 SCHEMATIC OF EXPERIMENTAL TEST FACILITY

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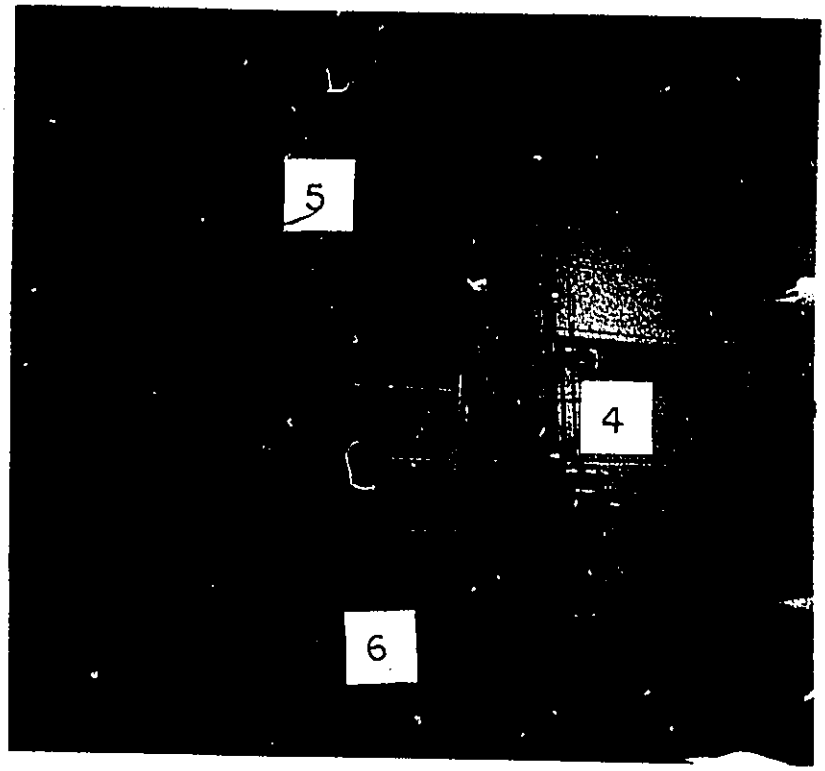
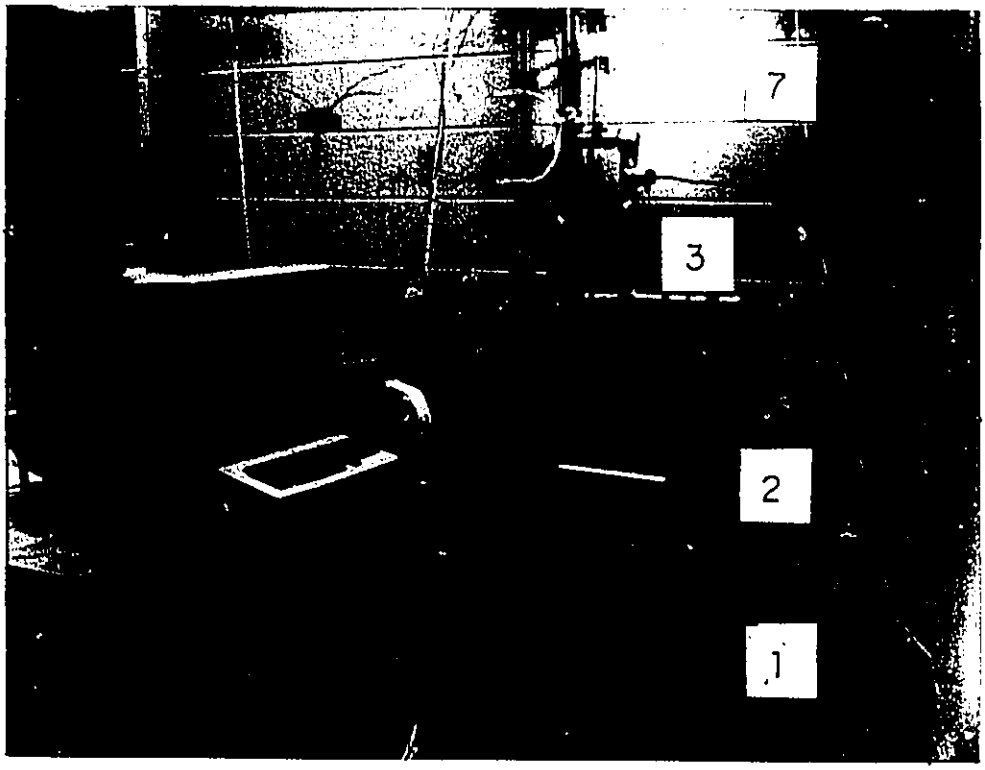


3.2 a



3.2 b

3.2 OVERALL VIEW OF THE EXPERIMENTAL SETUP

sections. Appendix-I lists the specifications of the equipment designed and used in the present study.

3.3 Selection Of Working Fluid

In order to observe the boiling phenomena it was decided that the test section be inside a glass pipe. With this in mind it was therefore necessary to operate the system close to room temperature to minimize heat losses so that no insulation is required in the test section. This in turn meant that a working fluid be chosen which has a low saturation pressure at room temperature, since low pressure operation was considered essential for reasons of safety. It is also desireable for the test fluid to have a low latent heat of vaporization so that the heater and condenser sizes may be small and easy to control.

R-11 and R-113 meet all the above criteria. They also have a low critical pressure, allowing us to study the boiling phenomena over a wide range of the two phase region, and they also have low toxicity. They are relatively cheap and easily available. Hence it was decided that the loop will be designed for R-11 and R-113.

3.4 Component Design

3.4.1 Test Section

As already mentioned in Section 3.1, in order to observe the boiling phenomena, it was decided that the flow be annular between an inner heated pipe and an outer glass pipe.

The following additional factors were considered in designing the

test section.

3.4.1.1 Test Section Heating Conditions

Electrical heating imposes a constant heat flux boundary condition, while fluid heating imposes constant temperature boundary conditions. If one type of boundary condition is imposed then the other parameter must be measured for data evaluation and analysis of results. It was found from [19] that the results from the two heating methods are not different from one another. Since it is physically easier to control and measure constant heat flux boundary conditions than constant temperature boundary conditions, it was decided to use electrical heating.

3.4.1.2 Type Of Electrical Heating

Direct electrical heating requires a low voltage high current supply (hundreds to thousands of amperes), especially for thick walled low electrical resistance channels. At present, the heat transfer research laboratory at the University of Windsor does not have such facilities. The option of using a high resistance, thin wire was eliminated because, as mentioned in Section 2.2.1.1, the nucleate boiling results are sensitive to system geometry especially if the surface dimensions are smaller or comparable with the bubble dimensions. Hence, it was decided to use indirect electrical heating.

3.4.1.3 Type Of Electrical Heater

Since it was decided that we should be in a position to study boiling in both developing and developed velocity profile regions,

it became necessary that we could position the heater anywhere along the length of the test section. Based on the considerations so far, it was decided to use a cartridge heater. These heaters allow a good thermal contact between the heater and the test pipe.

3.4.1.4 Size Of Glass Pipe

From the manufacturer's catalogue it was found that as the nominal size of the glass pipe increases, its maximum working pressure decreases. Thus a bigger diameter pipe reduces the test condition range. However, it should be big so that it can house test sections of different diameters.

At 7.91 bars (100 Psig), the saturation temperature of R-11 is 98.17 degrees C. Since water is being used as the coolant in the cooler(to be discussed in section 3.4.2) 98.17 degrees C is the maximum temperature and 7.91 bars is the maximum pressure under which we can run experiments. Hence, 25.4 mm (1 in) internal diameter, QVF (Quick Visible Flow) glass pipe supplied by Peagasus Industrial Specialities, Agincourt, Ontario with a maximum working pressure of 7.91 bars and maximum working temperature of 300 degrees C was selected.

3.4.1.5 Size Of Test Pipe

As mentioned in section 2.2.1.1, incipient boiling is insensitive to system geometry provided the surface dimensions are larger than typical bubble sizes(1 mm diameter). So it was decided to use pipe

of large diameter.

The outside diameter of the test pipe should be such that the resulting hydraulic diameter, hence the Reynolds number, should be in the range of interest (3000 to 30000, as this is the typical Reynolds number range existing in most industrial evaporators and condensers).

With a hydraulic diameter of 3.2 mm and the flow rate the pump is capable of delivering (refer to section 3.4.10), it was possible to achieve approximately the desired Reynolds number range ($2871 < Re < 28710$ for R-11 and $2475 < Re < 24750$ for R-113, at 20 degrees C).

The length of the annulus should be such that it allows for a smooth transition from plug flow to annular flow, and also allows the velocity profile to develop completely. From [21], the length for complete velocity profile development was found to be 20 hydraulic diameters.

Based on the above criteria, the test pipe was chosen from commercially available copper pipe, 22.225 mm (0.875 in) external diameter, 19 mm (0.75 in) and 0.762 m (30 in) long.Since the glass pipes are available in few selected lengths, a 0.4572 m (18 in) long pipe was selected. Please refer to Figure 3.5.

3.4.1.6 Size Of Main Heater

It should be capable of supplying a heat flux which will result in nucleate boiling -i.e., higher than 35kW/m2 [18].

It should supply less power to the working fluid, so that the bulk temperature of the fluid, evaluated over the volume, and the inlet subcooling, do not change appreciably as it passes through the test section. This is necessary because we use conditions at test section inlet as parameters, and do not want these parameters to change.

The length of the heater should be shorter than the length required to fully develop a temperature profile, so that the bulk temperature, evaluated over the cross section, is approximately the same as the one at inlet to the test section.

The thermal boundary layer thickness was determined for R-11, under minimum anticipated flow conditions, using the correlations from [19] for various heated surface lengths. It was found that this thickness is 1 mm at 63.5 mm (2.5 in) heated length. This therefore was decided as the upper limit for the cartridge heater length.

A boiling heat transfer coefficient of 1500 W/m2 C was assumed on the basis of results from [20], and a wall superheat of 18 degrees C, for boiling initiation on the basis of results from [11]. Thus a heat flux of 27000 W/m2 was found necessary at the boiling surface for boiling incipience.

For various cartridge heaters, the heat flux available at the test surface, and the bulk fluid temperature rise as it passes through the test section was determined.

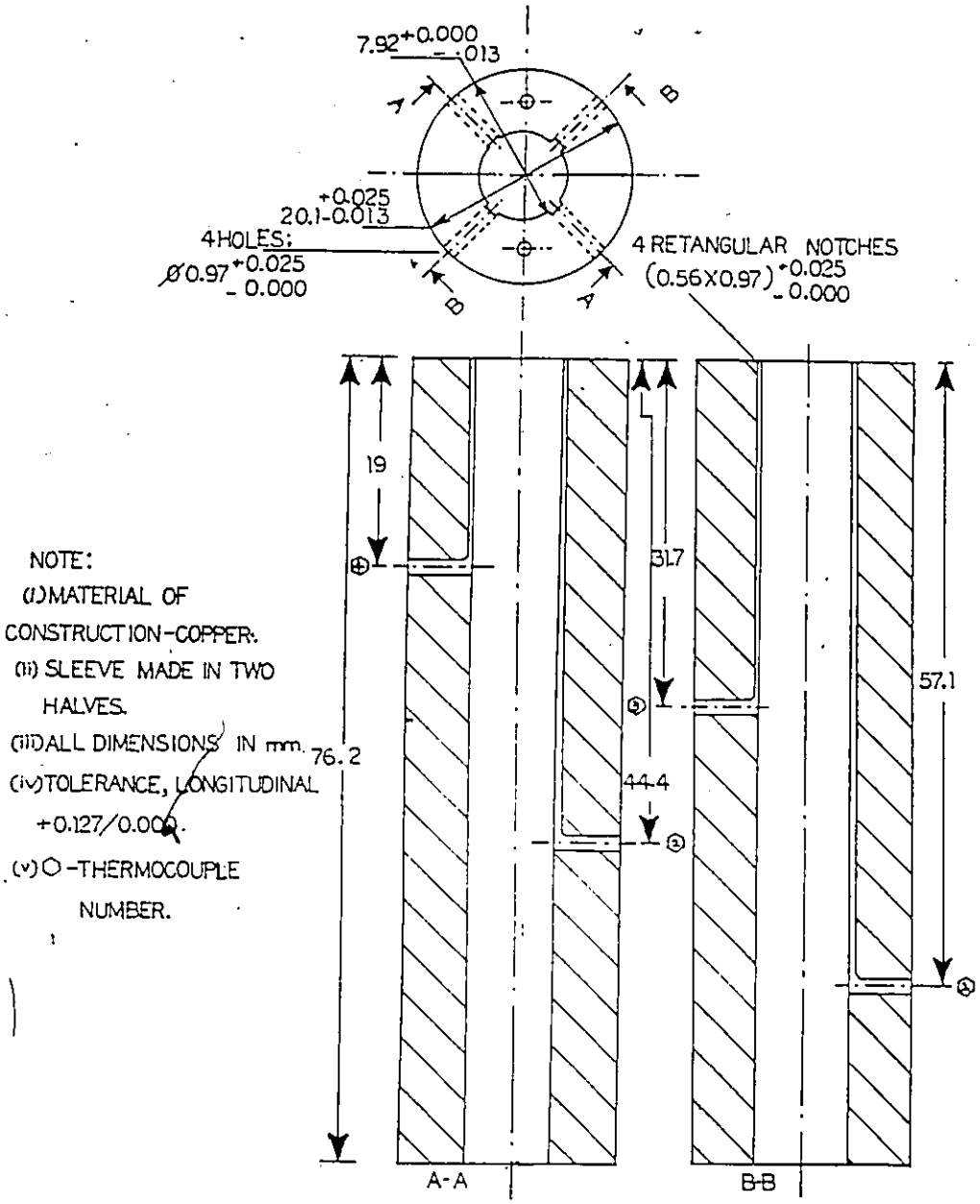
Finally, a H1-3, Fastheat, "Hi-Temp" cartridge heater with 7.874

mm (0.31 in) diameter, 76.2 mm (3 in) sheath length and 63.5 mm (2.5 in) heated length, capable of delivering 49972 W/m2 at the test surface 225 W rating at 120 V ,supplied by Acrolab Instruments, Windsor, Ontario was selected. At minimum flow, the flowmeter is capable of measuring (refer to section 3.4.5) and zero subcooling, the heat flux necessary for boiling initiation for R-11 was determined using Sato and Matsumura's [1] equation. At this heat flux, with this heater, the bulk temperature rise of the fluid was found to be 0.384 degrees C.

The power supply for the heater was through a Variac capable of delivering between 0-140 V. The power input to the heater was measured by a wattmeter capable of measuring 0-250 W with an accuracy of +/- 1.25 %.

3.4.1.7 Copper Sleeve

In order to house the instrumentation for measuring the heated surface temperature, a sleeve was designed. The sleeve shown in Figure 3.3 was made of copper so that the temperature drop across the sleeve would be small, and hence avoid excessively high temperatures in the heater element. Four thermocouples, one each in a slot, running along the length, each separated by 90 degrees, were installed at equal lengths along the sleeve to measure the temperature of its outside surface. This meant that to measure the wall temperature at a nucleation site accurately, the thermocouple orientation would have to be changed by the experimentor accordingly. The arrangement of thermocouples running parallel to the test section also reduces thermocouple conduction errors.



NOTE:
(i) MATERIAL OF CONSTRUCTION - COPPER.
(ii) SLEEVE MADE IN TWO HALVES.
(iii) ALL DIMENSIONS IN mm.
(iv) TOLERANCE, LONGITUDINAL +0.127/0.000.
(v) ⊙ - THERMOCOUPLE NUMBER.

3.3 MACHINED SLEEVE

These slots introduce discontinuity in the heat flux lines. To minimize this, thin wire thermocouples were selected. For the same reason it was decided to cut the slots at the inner surface of the sleeve , so that the heat flux lines are almost radial by the time they reach the sleeve outer surface and hence inner surface at the test section. Since thin slots are difficult to machine at the sleeve inner surface with accuracy if it is in one piece, it was decided to machine the sleeve in two equal halves and then cut the thermocouple slots.

One thermocouple was installed at the top of the sleeve, close to the sleeve inner surface. The temperature indicated by this thermocouple guides the experimentor if he can safely increase the heater power without damaging the heater element and also without damaging the soft solder which joins the heater/ sleeve assembly (to be discussed subsequently).

The external diameter of the sleeve was 20.066 mm (0.79 in) +0.000 mm/-0.1016 mm (0.004 in). This was the recommended tolerance [22] between mating parts for a locational clearance fit(a fit which allows good mating and easy positioning at the same time) for a 20.066 mm diameter shaft.

3.4.1.8 Thermocouples For Wall Temperature Measurement

The desirable properties of a thermocouple sensor are :

1) It must have a high Seeback coefficient,

2) It must have fast response to follow temperature changes accurately.

For a thermocouple, the response time is reduced only by reducing the wire diameter. However, as wire size is reduced to AWG (American Wire Gage) 36 and above, the junction becomes fragile and life shortened [23].

Type-T (Copper/Constantan-55% Copper and 45% Nickel) was selected because it has a moderately high Seeback coefficient of 38μV/C at 0 degrees C, with limits of error +/-0.5 C or 0.4% (whichever is greater). For the range of operation the error was therefore +/-0.5 degrees C. Teflon thermocouple insulation was chosen because it retains its dielectric properties up to 260 degrees C, and also has excellent resistance to solvent, acid, base, abrasion, water and is flexible. Thermocouple TT-T-30(type T, teflon insulation, 30 AWG size) supplied by Omega Engineering Company,Stamford, Connecticut was therefore selected.

For an accurate outside wall temperature measurement, the thermocouple sensing tip was soldered at the sleeve outside wall. Also, the two halves of the sleeve were soldered rather than bolted, to avoid any nonuniformity in the heat flux at the test surface.

To reduce any temperature drop due to air gap (if any) between the heater outer surface and sleeve inner surface and also between the sleeve outer surface and test inner surface, a coat of

Omegatherm-201 thermally conducting paste, supplied by Omega Engineering Inc.,Stamford,Connecticut was applied. This paste had a thermal conductivity of 2.31 W/m C , and was capable of continous operation up to 205 degrees C. The temperature drop across the gap while the heater delivers 225 W, was estimated at 1.03 degrees C using one dimensional steady state thermal conduction equations.

3.4.1.9 Heater, Thermocouple, Sleeve Assembly

The assembly sequence for the heater, thermocouples, sleeve unit was as follows :

1) The thermocouple tips were soldered to the sleeve wall using silver solder (45 % silver, 15 % copper, 24% cadmium, 16% zinc) with a melting point of 604 degrees C,

2) A coat of thermally conducting paste was applied to the sleeve inner surface,

3) A thin coat of soft solder (62 % tin and 38 % lead) was applied on the mating faces of the sleeve halves,

4) The sleeves were joined and heated by a flame torch, melting the solder applied in step (3) in the process and thus completing the assembly,

5) Finally, the outer surface of the assembly was smoothed by a fine grained emery paper.

3.4.1.10 Heater Positioning Rod

For positioning the heater/ sleeve assembly at any desired location a positioning rod mechanism was designed. To reduce the heat lost along this rod due to fin effects, stainless steel

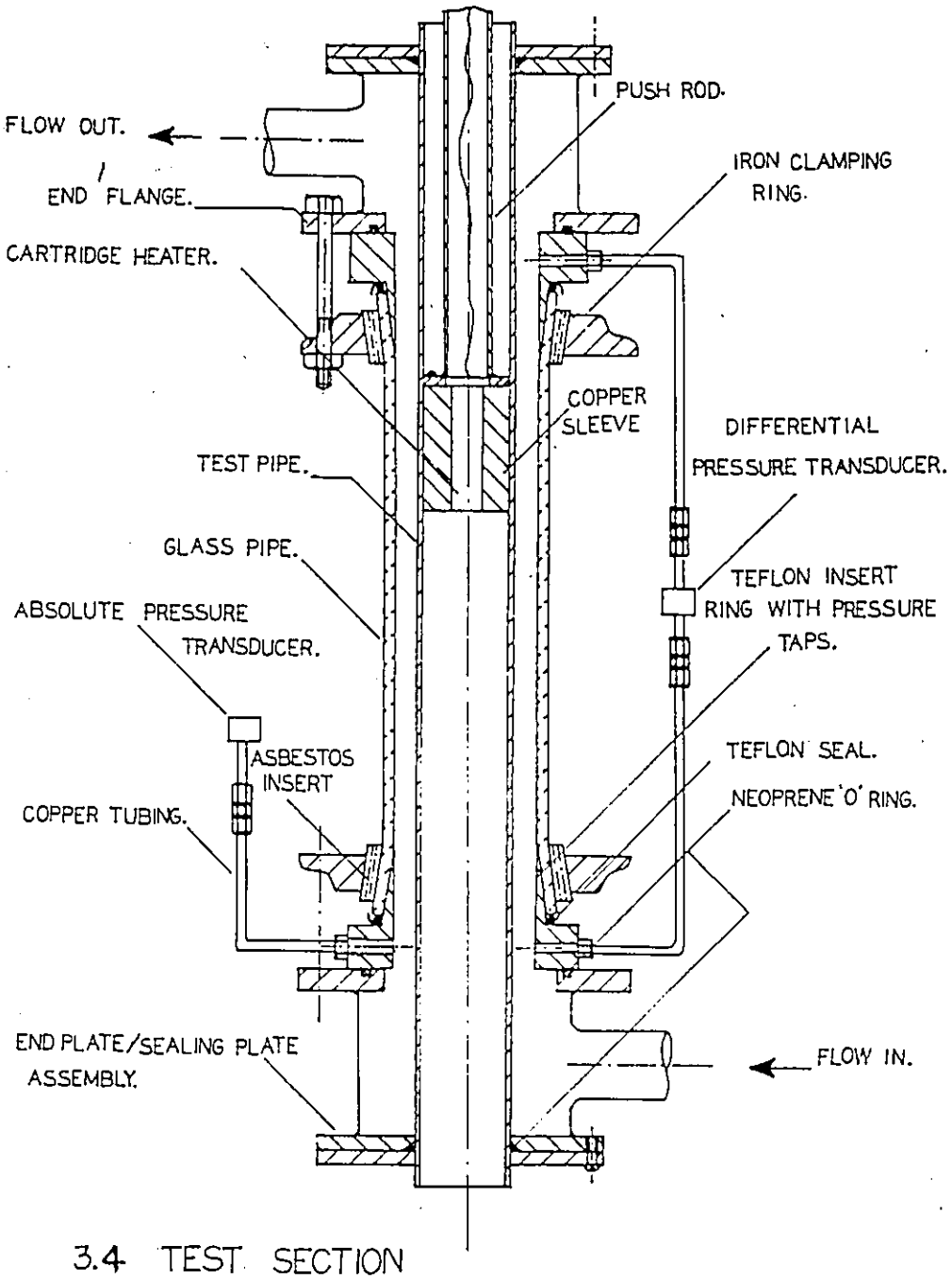
AISI(American Iron and Steel Institute) 316 (poor thermal conductivity) was selected. The end of the tube was welded to a flange, which in turn was bolted to the end of the sleeve.

The heat lost through the push rod was estimated as 2.5 W,(assuming the rod as an infinitely long fin, and the rod loses heat to air whose temperature is an arithmetic mean of the wall and room temperatures) and the total heat lost from the flat ends of the sleeve was estimated as 0.8 W (assuming a laminar free convection around the flat ends) i.e., a total heat loss of 1.5% of maximum heat input. Since this heat loss is negligible, it was decided not to fill the test pipe internals with insulation. For heat loss calculations please refer Appendix-B.

3.4.1.11 Test Section Sealing Assembly

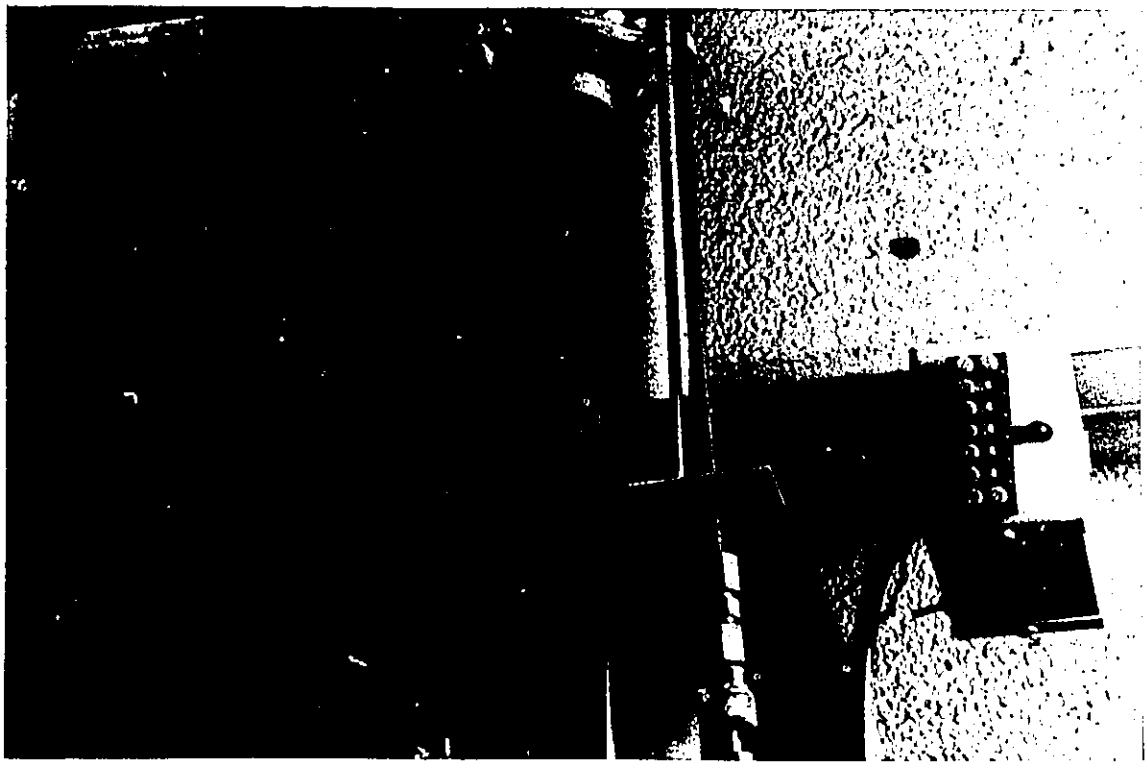
The test section is shown in Figure 3.4. The end caps used to seal each end of the glass annulus were constructed from an end flange made of free machining brass, an end plate/ sealing plate assembly made of free machining brass 11.11 mm (0.4375 in) thick, and a 38.1 mm (1.5 in) * 19.05 mm(0.75 in) unequal copper tee connecting them. The flow in/ out of the test section is through the tee branch. The end flange was 12 mm thick. Please refer to Appendix- C for flange design calculations.

The end flanges were bolted to the test section using a teflon



3.4 TEST SECTION

3.5 a



3.5b

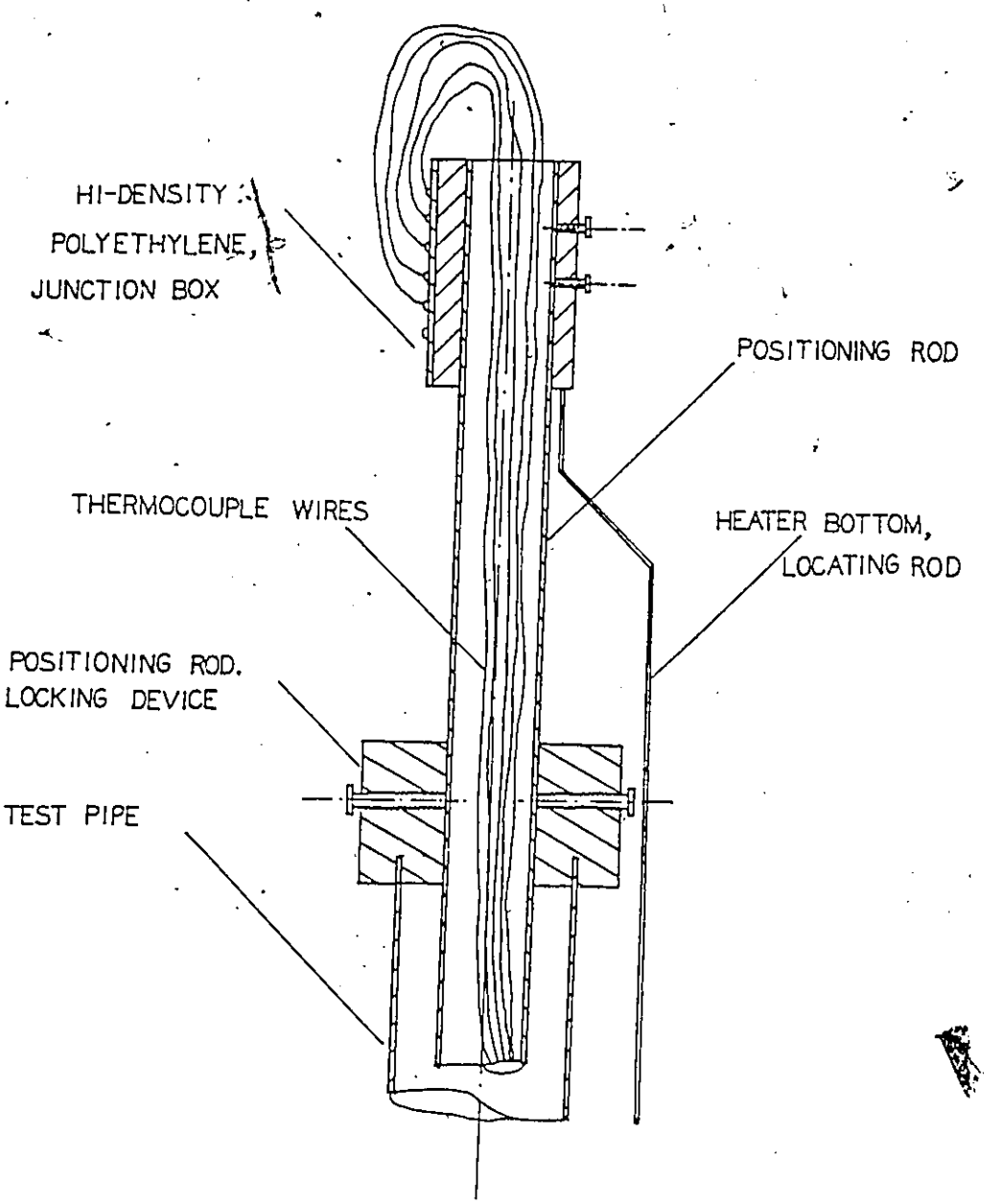
3.5 TEST SECTION ASSEMBLY

gasket ring and a teflon adapter piece to provide the glass to brass seal. As shown in Figure 3.4, the teflon adapter piece was profiled to be a companion for the flared ends in the glass section. Pressure measurement taps were machined in the shoulder of these adapters and are discussed in detail in Section 3.5.2. The teflon adapter piece/ end flange interface was sealed by a 41.275 mm (1.625 in) nominal diameter neoprene "O" ring which was seated in a sealing groove, in the end flange. The sealing groove was machined according to SAE J120a specifications[25] (Rubber Rings For Automotive Applications).

The test pipe was held in position with a 22.225 mm (0.875 in) nominal diameter neoprene "O" ring which was seated in a sealing groove cut into the end plate. The sealing groove was machined according to SAE J120a specifications [25]. As shown in Figure 3.4, the "O" ring is compressed against the end plate and the test pipe by a sealing plate which encircled the test pipe and is bolted to the end plate with four 3.175 mm(0.125 in) steel screws.

3.4.1.12 Locking Device For Positioning Rod

At the top of the test pipe is located the locking device, made from Hi-density polyethylene, for the heater/ sleeve assembly as shown in Figure 3.6. It has a 12.7 mm deep circumferential notch which fits snugly and is also glued to the test pipe using Omegabond 101 epoxy adhesive. The heater positioning tube can slide freely in the central hole. The positioning tube can be



3.6 LOCKING DEVICE FOR POSITIONING ROD

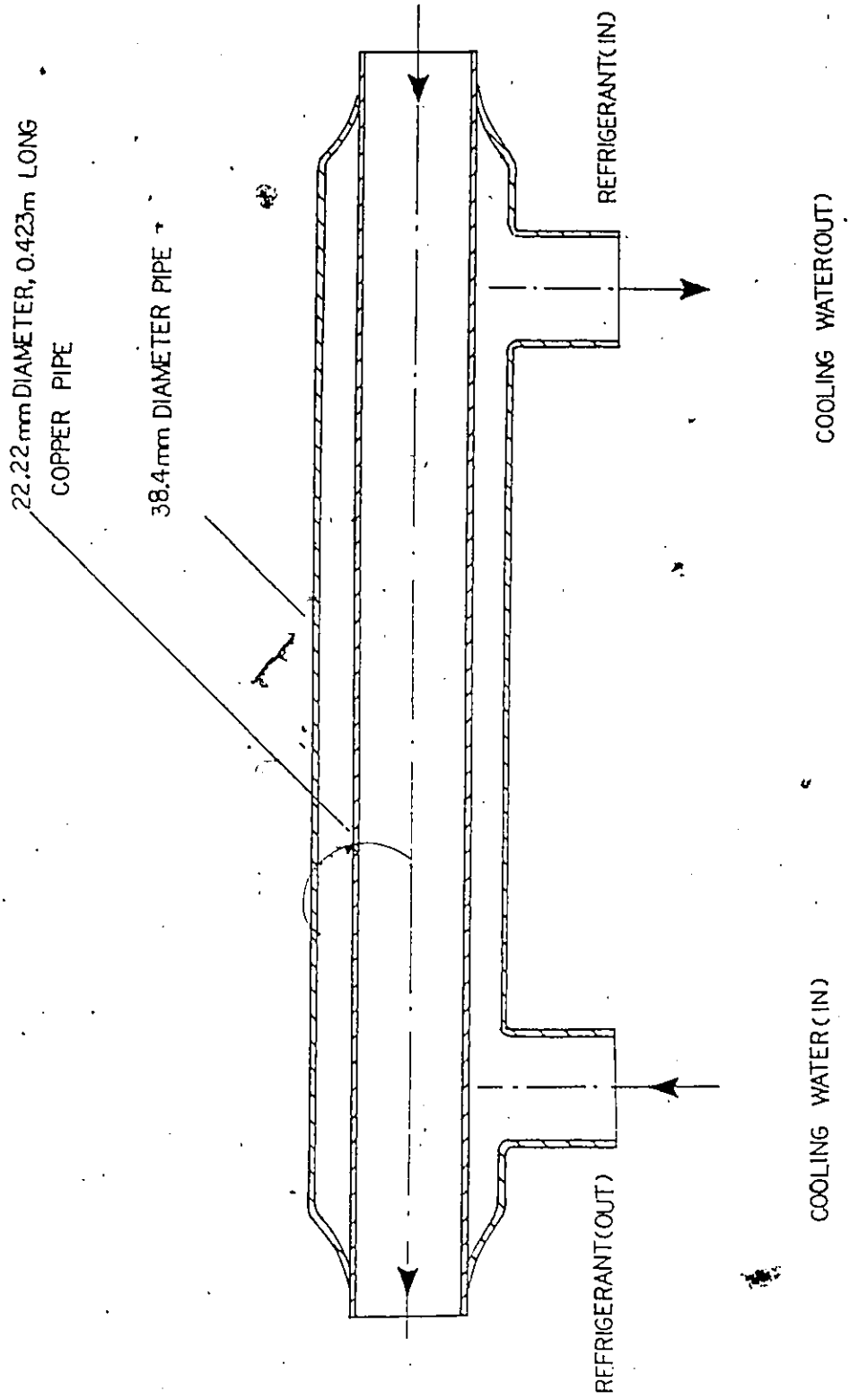
locked by two screws located diametrically opposite each other.

At the top of the positioning tube is a rectangular junction box, for the five heater thermocouples as shown in Figure 3.6. A 3.175 mm diameter rod, parallel to the positioning tube and supported at the top from the junction box was mounted, for locating the heater bottom. The rod moved in tandem with the heater assembly. The four wall temperature measuring thermocouples were oriented in directions perpendicular to the side faces of the rectangular block.

3.4.2 Cooler

A cooler was installed to remove the energy added in the test section. It was decided that the cooler should be as close to the test section exit as physically possible, so that two phase pressure drop (if any) is reduced as far as possible. To avoid possible cavitation at the pump, the condenser elevation was set 1.68m above the pump. For ease of construction, it was decided that the cooler would be a double pipe (two coaxial pipes), counter flow heat exchanger as shown in Figure 3.7.

Water was selected as the cooling medium, since it has a low thermal resistance. For establishing a stable cooling flow rate at a constant temperature, a circulating Lauda constant temperature bath, type K-40, 115 V, 800 W, Messerate-Werk Lauda, W.Germany, was used.


22.22mm DIAMETER, 0.423m LONG
COPPER PIPE
38.4mm DIAMETER PIPE
REFRIGERANT(IN)
COOLING WATER(OUT)
COOLING WATER(IN)
REFRIGERANT(OUT)

3.7 COOLER

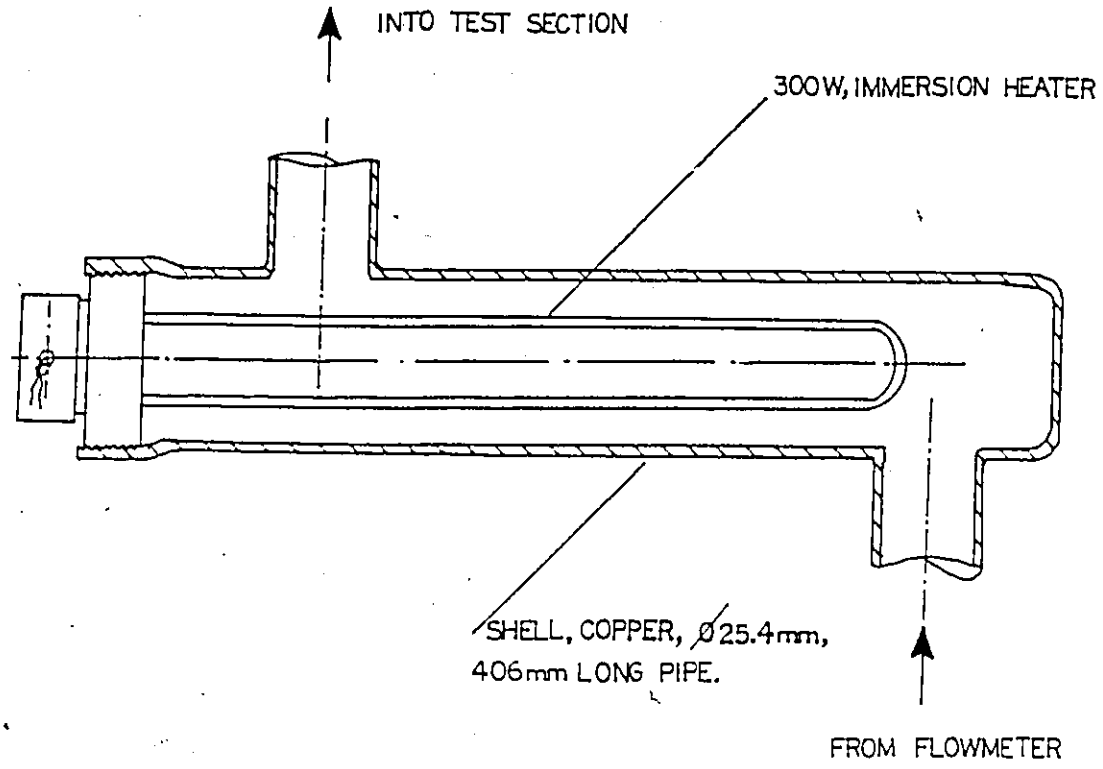
In sizing the cooler, the following extreme case was assumed with R-11 as the working fluid:

All of the fluid entering is saturated liquid and the maximum energy input in the main heater must be removed. The overall heat transfer coefficient , the effectiveness, flow ratio, NTU and finally, the heat transfer area were calculated. See Appendix-D, for details.the cooler sizing.

The required area was found to be 0.04 m2 (0.56 m long, 22.225 mm outer diameter tube). However, a 0.61 length (10% over area) was provided for reasons of safety.

3.4.3 Preheater

The preheater is used to control the inlet fluid temperature in the test section. This unit, together with the system pressurizer, and the cooler, allows the experimentor to control both the inlet subcooling and the system pressure as parameters independent of each other. The cooler alone is capable of controlling the inlet fluid temperature, but since the voltage to the heater is easier to control than the coolant flow to the cooler, the preheater is included in the loop. The pressure drop in the preheater should be as small as possible, so that any chances of flashing or two phase boiling in the preheater are reduced. So, a horizontal configuration for the preheater was decided. Also, the power density of the preheater should be lower than the power density which will result in boiling. From [18], this was found to be



3.8 PREHEATER

35 kW/m2.

The preheater (refer to Figure 3.8) selected was 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 and a power density 31000 W/m2.

At a flow rate of 0.000315 m^3/s (maximum flow the flowmeter is capable of measuring , refer to section 3.4.5), for R-11, the heater allowed the inlet temperature to be controlled within a range of 0 - 0.7°C, while at a minimum flow rate of 0.00315 m^3/s (refer to section 3.4.5), this range was 0- 7° C.

The preheater end was a 19 mm (0.75 in) NPT male connection which mated with its counterpart in the preheater housing, which was built from a 25.4 mm copper pipe. The preheater was connected to a variable transformer capable of delivering 0-140 V. Since the power supplied to the preheater is not required for data reduction, a wattmeter was not connected.

3.4.4 System Pressurizer

The functions of the pressurizer unit are:

1) To maintain the system pressure at the desired level,
2) To dampen any pulsations caused by the pump,
3) To act as reservoir for the test fluid.

A commercially available bladder type pneumatic-hydraulic accumulator was found to meet all the above requirements. In order

that the system pressure be maintained constant, it is necessary that the volume of the gas charged bladder be approximately constant during the experimental run. That is, the change in volume of the gas bladder, due to volumetric changes in the loop because of vapour generation must be a minimum. Since we have subcooled flow boiling in the test section, we can be reasonably safe in assuming that the volume of vapour downstream of the main heater will be negligible. Hence, we can assume that the maximum volume of the vapour that can be generated in the test section corresponds to the volume of the heater length in the test section annulus and this was estimated as 0.00000905 m3.

The synthetic rubber partition material should have:

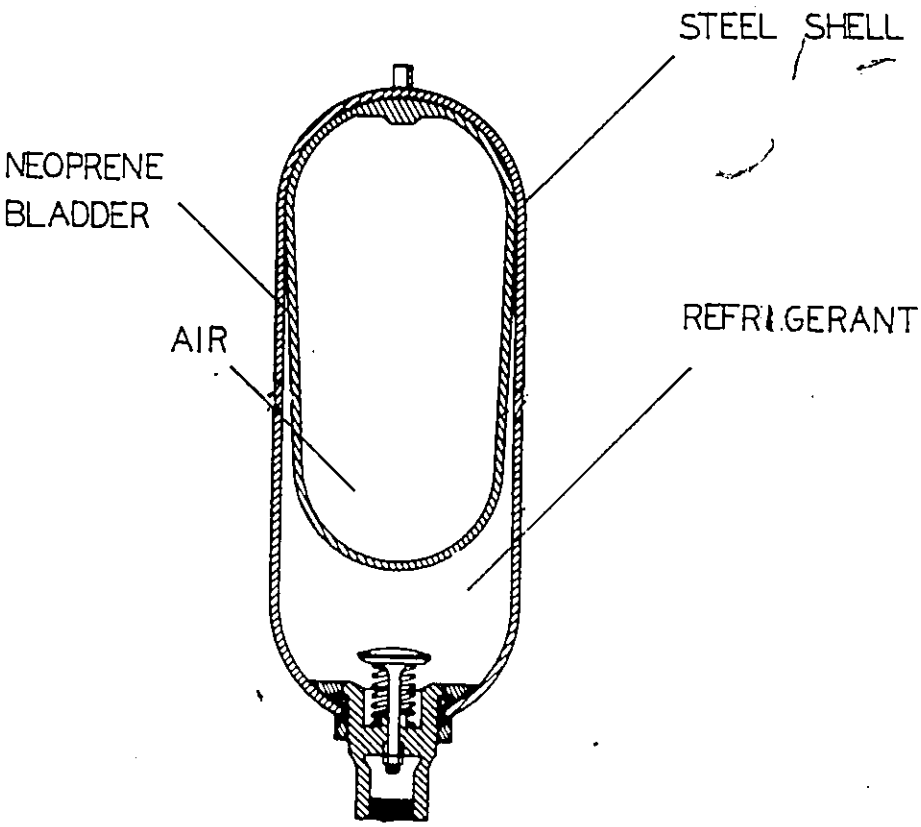
1) Tensile strength and heat resistance at the maximum design temperature,

2) Gas impermeability.

Pressurizer model # AB01B3T1A1K, refer to Figure 3.9, 0.000946 m3 size, 206.9 bars(3000 Psia) and Buna-N rubber bladder, supplied by Parker Fluid Power, Des Plaines, Illinois was found suitable and selected. The pneumatic pressure in the bladder in turn was controlled by a pressure regulating valve with a pressure indicator 0-12.05 bars (0-160 Psig) size.

3.4.5 Flowmeter

It was decided to use a variable area flowmeter for steady state flow measurements, because it was inexpensive, durable, stable,



3.9 SYSTEM PRESSURIZER

very reliable and does not require periodic calibration. The flowmeter should have the following characteristics:

1) It should be capable of withstanding the loop design pressure and temperature ,

2) It should be capable of providing the measurements accurately,

3) The pressure drop encountered across the flowmeter should be small.

A rotameter supplied by Omega Engineering Company, Stamford, Connecticut, model # FL-1504- A with borosilicate glass metering tube, SS 316 float, and Viton-A "O" ring , all compatible with R-11 and R-113, was selected. It could be operated at pressures up to 7.9 bars (100 Psig) and 121 degrees C (250 degrees F) with an accuracy of +/- 2 % and repeatibility of +/- 0.5 %. Its float had a design which made it virtually immune to viscosity variations. It was capable of measuring a flow range of 0.0000315 m3/s to 0.0.000315 m3/s for R-11 and 0.0000302 m3/s to 0.000302 m3/s for R-113 (corresponding to Reynolds number range of 2871 to 28710 for R-11 and 2475 to 24750 for R-113, in the test section evaluated at 20 degrees C). The manufacturer estimated that maximum pressure drop across the flow meter was 0.3302 m of water (13 in) (or 0.223 m of R-11 or 0.2189 m of R-113 , evaluated at 20 degrees C). The flow meter inlet and outlet were connected to the loop through 25.4 mm (1 in) NPT male connections. The manufacturer's calibration was based on water. The flowmeter was calibrated for R-11, by the following procedure:

The float position, the mass transferred and the time taken for this transfer was noted. The curve for R-11 was directly converted

from the manufacturer supplied curve for water, with the aid of the following formula:

$$Q_{fr} = \frac{Q_w}{\left[\frac{(\rho_{float} - \rho_w)\ \rho_{fr}}{(\rho_{float} - \rho_{fr})\ \rho_w}\right]}$$

The calibration results are shown in Appendix-E and they matched within 6.4 % with the formula and hence it was decided to use the formula in the data acquisition program.

3.4.6 Loop Fittings And Piping

The various heat transfer and hydraulic components are made primarily of copper or brass. Loop piping is 19 mm (0.75 in) internal diameter , copper pipe and its corresponding fittings. The flow in the loop is controlled by a 19 mm (0.75 in) diameter ball valve. The loop was also provided with a hand operated, 9.525 mm (0.375 in) diaphram drain valve. A Schrader core valve 6.35 mm (0.25 in) was soldered downstream of this valve.

At the top-most point in the loop, a 6.35 mm liquid/ moisture indicator was installed. A hand operated 9.525 mm diaphram valve was soldered at the indicator downstream.

A filter drier, model ALCO ASD 45S6-VV was incorporated on the suction side of the pump to remove moisture and suspended particles (if any) in the loop.

A 0.000019 m^3 volume, non condensable accumulator was soldered at the valve downstream. For sizing the accumulator, the loop volume was estimated. The pressure due to noncondensables at maximum achievable vacuum conditions was estimated. The amount of noncondensibles released is directly proportional to their gage pressure immediately before release into the atmosphere. Their final pressure was set at 5 bars gage (arbitarily) and using Boyles law, the volume of the accumulator was calculated. A Schrader core valve was soldered downstream of this vessel. The vent and drain Schrader valves were capped using a standard SAE threaded flare cap.

The section of the loop between the pump outlet and test section inlet was thermally insulated using a 12.5 mm thick , Armstrong, armflex insulation to minimize the heat losses to the ambient air. This is also a factor in obtaining thermal stabilization of the running system and will result in shorter waiting time for reaching steady state conditions.

3.4.7 Centrifugal Pump

The circulating pump is an important part of the forced convection loop and should have a steep Head-Flow characteristic, so that a small change in head does not produce flow oscillations. Positive displacement pumps have a flow proportional to speed and are almost independent of pressure differential. But they introduce large pulsations which can be detrimental to our studies on individual nucleation sites. So, it was decided to use a centrifugal pump with a steep Head-Flow characteristic.

For sizing the pump, a tentative configuration was assumed for the loop and its various components and the pressure drop in the loop was calculated for a flow rate of 0.0003119 m^3/s , (Re = 28360 for R-11 and 24800 for R-113 evaluated at 20° C, in the test section). Please refer to Appendix-F for pressure drop calculations. In making these calculations, it was assumed that the two phase accelerational pressure drop component is negligible. This was justified because the two phase flow length was kept no longer than necessary and also because we have subcooled flow boiling, the void fraction is very small. The estimated pressure drop was 2.7 m and 2.8 m for R-11 and R-113 respectively at 20 degrees C.

If the pressure drop in the length between the cooler outlet and the pump inlet is less than the height of the downcomer pipe, we can be sure that the pump will not cavitate. The pressure drop was estimated as 0.371 m for R-11 and 0.385 m for R-113. Since the height of the downcomer is 1.828 m, we have a minimum Net Positive Suction Head available at the pump inlet of 1.458 m. Centrifugal pump model #. J-7004-54 from Cole-Parmer Instrument Company, Chicago,Illinois was found to meet the requirements and was selected. For the characteristic curve of the pump, please refer to Appendix-G. The pump had a flow rate of 0.0003028 m3/s for R-11 and 0.0002996 m3/s for R-113 at the above estimated differential heads. This was as close to the desired flow rate as we could get. The pump required a Net Positive Suction Head of 0.1524 m (6in).The pump housing was Ryton, the spindle was ceramic and the

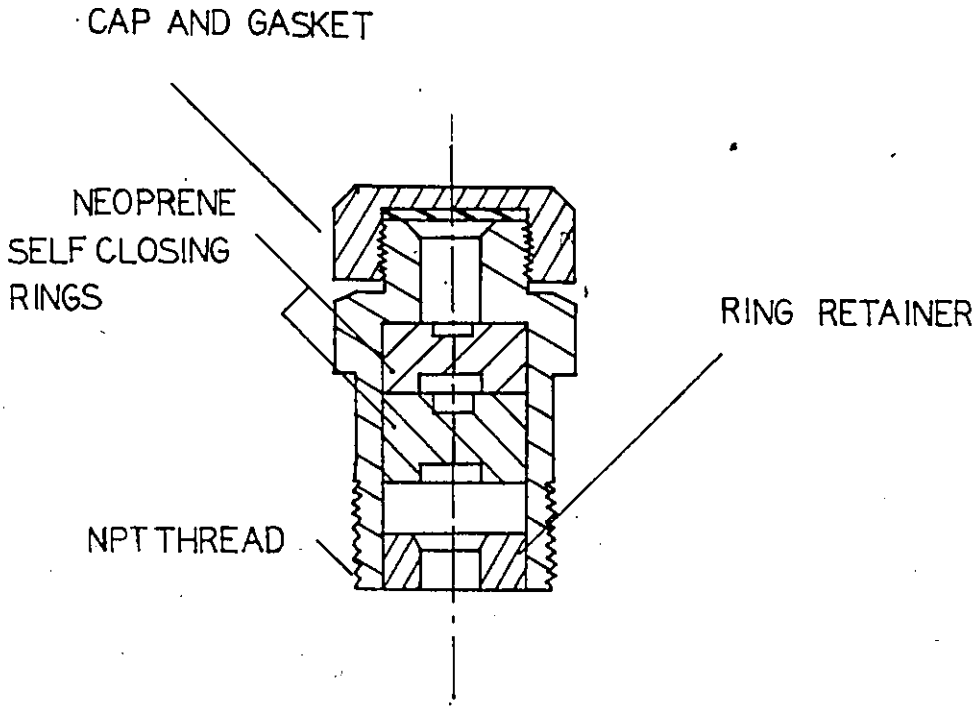
sealing- "O" ring was Viton, all of which are compatible with both R-11 and R-113. The pump was coupled to a 0.04 HP, 3450 RPM, 115 V a.c motor through a sealless magnetic drive. The pump inlet and outlet were connected to the loop through 12.7 mm (0.5 in) NPT female connections.

3.5 Instrumentation

3.5.1 Temperature Measurement

The wall temperature thermocouples have already been discussed in section 3.4.1.8. The refrigerant bulk temperatures at the inlet and the outlet to the test section were measured to control the heater settings; the outlet from cooler, and the cooling water bath temperatures were measured to control the cooling water flow rate; and the ambient temperatures was measured by 24 gage copper-constantan thermocouples with teflon insulation.

The thermocouples which measure the refrigerant bulk temperatures at the test section inlet and outlet, and the outlet from the cooler, were positioned with their tips held at the fluid stream centerline. The fittings holding these thermocouples is a self-sealing temperature test plugs model # OPNE-14 (refer to Figure 3.10) with two neoprene self-sealing rings, supplied by Omega Engineering Inc. The surface of the thermocouple is not cylindrical and there are possibilities of a leak. So, Superpoxee from Watsco Inc., Hialeah, Florida was applied over the thermocouple insulation to achieve an uniformly cylindrical surface. The sealing rings were punctured and the thermocouple



3.10 THERMOCOUPLE PLUGS

wire was threaded. The output from the thermocouples were fed into the scanner board of a 2240C Fluke data logger.

3.5.2 Pressure Measurement

To calculate the wall superheat, it is necessary to know the wall temperature and the local saturation temperature. Section 3.4.1.8 dealt with the instrumentation for wall temperature measurement. The inlet saturation temperature was calculated by measuring system pressure and using thermodynamic correlations. The local saturation temperatures in the test section, were calculated by measuring the pressure drop in the test section and inlet pressure and calculated using thermodynamic correlations. The data acquisition software (see Section 3.5.3) performed these calculations.

The system pressure and pressure drop in the test section was measured by piezoresistive pressure transducers. Model# PX 82-100 GV, 111.3 mV span, 7.9 bars (100 Psig) transducer was used for absolute pressure measurement and model# PX 83-005 DV, 143.9 mV span, 0.345 bars differential (5 Psid) transducer was used for differential pressure measurement. Both the transducers had an accuracy of +/- 0.25% of the full scale reading.

The power supply to these units was supplied by a constant current supply unit model# PS-KIT-1 which supplied 1.5 mA. This unit in turn received power through a 120 V/12 V stepdown transformer. The transducers and constant current supply unit were supplied by

Omega Engineering Inc.

Two 3.175 mm (0.125 in) female NPT taps were machined in the shoulder of the bottom teflon adapter of the test section. One 4.76 mm (0.1875 in) female NPT tap was machined in the shoulder of the top teflon adapter piece. Care was taken to maintain the centre lines of the pressure taps perpendicular to the axis of the test section. All burrs around the holes were removed to eliminate any eddies which might cause a dynamic pressure component in the measured static pressure. The absolute pressure transducer is connected to one bottom tap through 3.175 mm copper tubing and brass compression fitting. The differential pressure transducer is connected to the bottom tap through 3.175 mm copper tubing and brass compression fitting, and to the top through a 4.76 mm copper tubing and compression fitting.

The outputs from the pressure transducers was fed into the scanner board of a 2240C Fluke data logger. Corrections to the indicated pressures, due to elevation differences between the pressure tap and the transducer sensor was included in the data acquisition software.

3.5.3 Data Acquisition System

A fast and convenient on-line data analysis was developed using a 2240 C Fluke data logger which was interfaced with a 64K Apple computer through a RS 232C serial card. The computer controlled

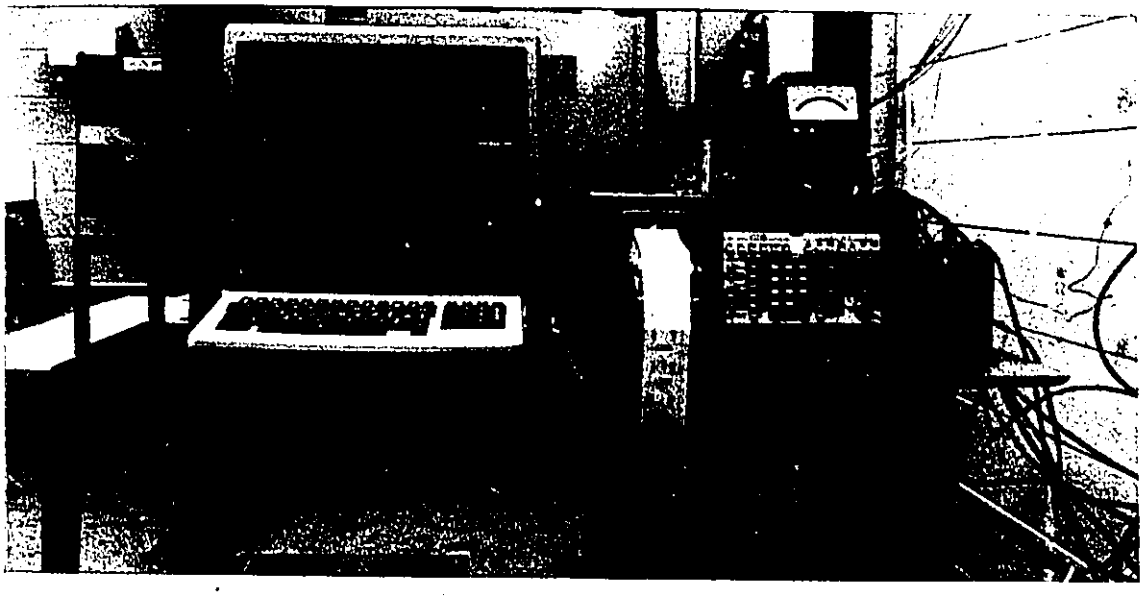
the data acquisition system. Please refer to Figure 3.11.

The computer data acquisition software "BOIL7" was written to enable fast collection of information on wall superheats, and heat transfer coefficients. The program was written in Apple Basic and was designed to ask clear, consise questions about the input data required for calculations. It is very user friendly and no prior knowledge of the calculation methodology is required to run the program.

The sub-program titled "FLUKES.BJO" is a machine language program which enabled the Fluke's data transmission to be decoded and stored at specific memory addresses within the Apple computer. This program is automatically retrieved from disk storage to initialize the Fluke data logger whenever the data acquisition program "BOIL7" is run. Special thanks to Dr.N.W. Wilson for writing "FLUKES.BJO" software.

The computer was programmed to scan any number of times as directed by the experimentor. The data acquisition software used these scanned data, together with manually inputted data for atmospheric pressure, heater location, nucleation site location, flow meter reading, heater power, to calculate the wall superheats and heat transfer coefficients. The software allowed the experimentor to save the data results in a disk if he so desired.

Appendix- H describes the flow chart, the variables nomenclature, the program listing and a sample output. Note that in the flow



3.11 DATA ACQUISITION SYSTEM FOR THE EXPERIMENTAL SETUP

chart, unless otherwise indicated, a horizontal line indicates a flow from left to right, and a vertical line indicates a flow from top to bottom.

3.6 Loop Commissioning

3.6.1 Leakage Test

Leakage tests are necessary to prevent:

1) Refrigerant loss to atmosphere (as in case of R-11),

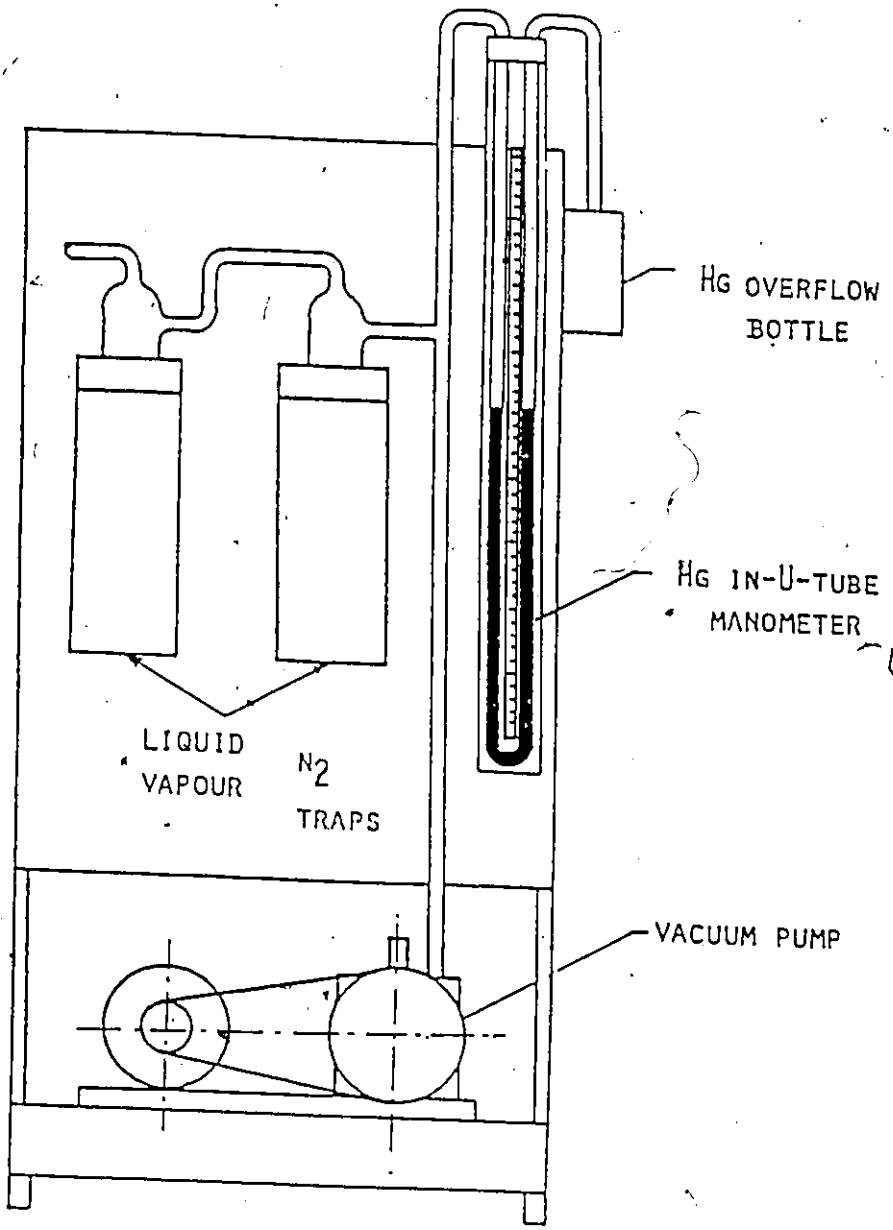
2) Moisture ingress into the system (as in the case of R-113), which will affect the results.

Three methods of leak detection were used as mentioned below:

1) The entire system was charged with air, to a pressure of 5.15 bars (60 Psig), and the system was isolated and the loop pressure was observed. A drop in indicated pressure indicates a leak, and the locations of the leaks was detected using a soap solution.

For ease of assembly, the loop was fabricated in sections, and was assembled through union joints. Numerous leaks were detected, especially in these unions and all attempts to tighten them were unsuccessful. These unions were replaced by brass compression fittings. Other leaks, depending on the nature of joints, were fixed using silver solder(for soldered joints) or Loctite sealant(threaded joints).

2) The system was evacuated to 744 mm of mercury using a vacuum pump system, shown in Figure 3.12. This unit was equipped with liquid-gas separators in series, and was immersed in liquid



3.12 SCHEMATIC OF LEAKAGE TEST UNIT

nitrogen. This arrangement condensed the vapours in the loop. The vacuum was held by the system for thirty hours.

3) After the system was charged (as described in Section 3.6.2), the system was checked by a halogen electronic detector which is capable of detecting leaks of the order of 0.5 oz per year. The device emits a "beep" which increases in frequency when higher concentrations of halogens are sensed.

No leaks were found.

3.6.2 Charging Procedure

The following charging procedure was followed (refer to Figure 3.13):

1) The Schrader valve in the loop vent was connected to the vacuum pump through a testing manifold provided with a gage.

2) The refrigerant charging cylinder was connected to the loop through a charging valve close to the cylinder, a charging hose and a Schrader valve in the filter drier. The cylinder was placed at an elevation of 1.5 m above the loop top most point to provide an additional hydrostatic head.

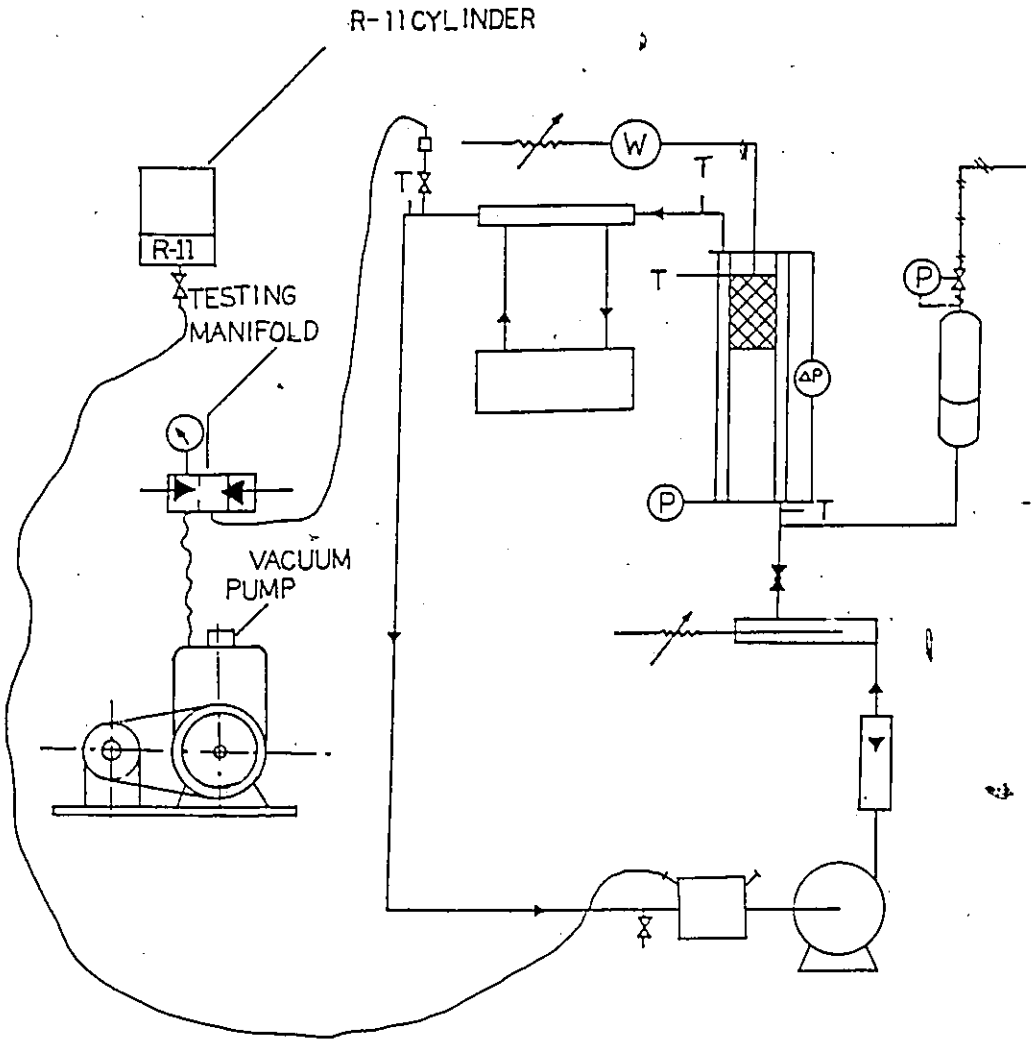
3) The system was charged by alternatively

(3.1) Opening the valve in the vacuum line with the charging valve closed, and pulling the desired vacuum.

(3.2) Closing the valve in the vacuum line with the charging valve open, and charging the loop.

4) When the loop was completely filled as observed through the site glass in the top, the charging valve was closed.

5) The vent Schrader valve was depressed, until just a little



3.13 SCHEMATIC OF LOOP CHARGING FACILITY

liquid refrigerant from the system escaped. This way, we were sure that most of the noncondensibles in the loop were evacuated.

3.6.3 Results Of System Commissioning

The heat transfer loop was commissioned and the results are given herewith :

Parameter	Targeted Range	Achieved Range
System pressure	1- 7.9 bars	2- 7.9 bars
Inlet subcooling	0- 40° C	0-75° C for R-11 and 0- 105° C for R-113
Reynolds number	3000-30000	2900-25000 for R-11 and 2500-22000 for R-113, at 20° C
Heat flux at test surface	0-40000 W/m^2	0-60000 W/m^2
Heater location	Anywhere along the test section	Anywhere along the test section
Bubble nucleation visualization	At all flow rates	Difficult to discern individual sites beyond Reynolds number 22000, for R-11

4. RESULTS, DISCUSSION AND CONCLUSIONS

4.1 Experimental Procedure

The following procedure was followed during the course of preliminary experimental runs:

1) The main heater was positioned at the desired location,

2) The loop was pressurized to test conditions,

3) The instrumentation was turned on, and the Fluke data logger was turned on and initialized. The computer was loaded with the data acquisition program "BOIL7",

4) The pump was turned on, and the throttling valve manipulated to achieve the required flow rate,

5) The preheater was adjusted to achieve the desired inlet subcooling,

6) The main heater was turned on, and the input voltage was raised in small steps by manual control of the variac. The fluid exit temperature and test section wall temperatures were monitored,

7) Due to buoyancy effects, the flow rate changed, and it was adjusted if necessary,

8) The constant temperature, water circulation was turned on(if necessary) and the cooling water flow rate adjusted to maintain the subcooling at the test section inlet. The loop was now operating at a quasi-equilibrium condition that was suitable for data acquisition. The computer scans the temperatures and pressures and displays the wall temperatures, wall superheats and heat transfer coefficients. For comparison of results, the

nucleate boiling dormancy superheat predicted from Sato and Matsumura's correlation was also displayed. Appendix-H gives a sample output of a scan.

9) Save the results if so desired, else, repeat the scan under the same conditions, or scan under new set of conditions,

10) The same procedure was followed for a decreasing heat flux.

The waiting time for the system to reach steady state conditions from "cold" startup was about 0.25 hours. Subsequent runs attained steady state in 0.15 hours. Twenty to thirty sets of readings were usually required for a complete cycle of increasing- and-decreasing heat flux.

Preliminary experiments were carried out for various heater power inputs, system pressures, inlet subcooling and fluid flow rates. These runs were trial runs to check the loop design and debug the data acquisition program.

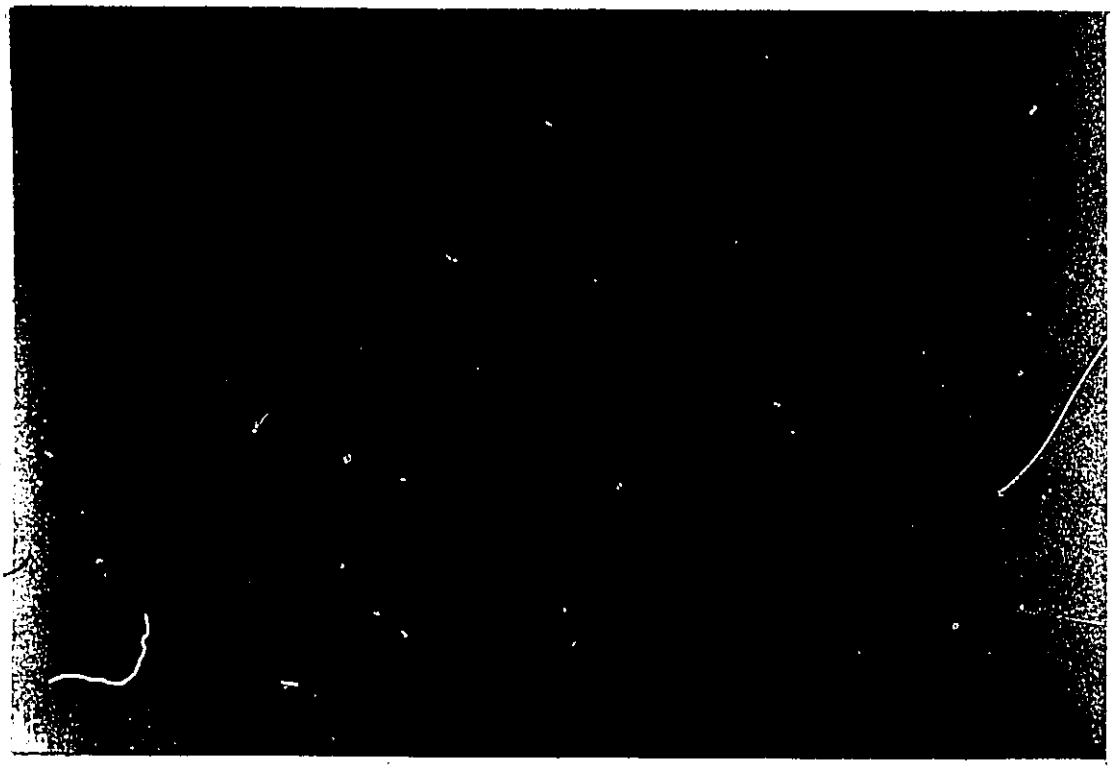
In a typical run, the inlet subcooling, system pressure and volumetric flow rate were held constant and the heater power was increased in steps, until nucleate boiling occured and was then decreased in steps, until the sites disappeared. Only the results, in which individual bubbles on the surface could be discerned were recorded. However, when the bubble population was large, the values were not used as incipience values. For observing these sites, a desk lamp was mounted and care was taken so that the heat from the lamp did not affect the results. The nucleation

sites were viewed through a convex magnifying lens. Figures 4.1 through 4.4 show the photographs of nucleation sites as captured by a low and high speed camera. They show the site behaviour at different boiling stages.

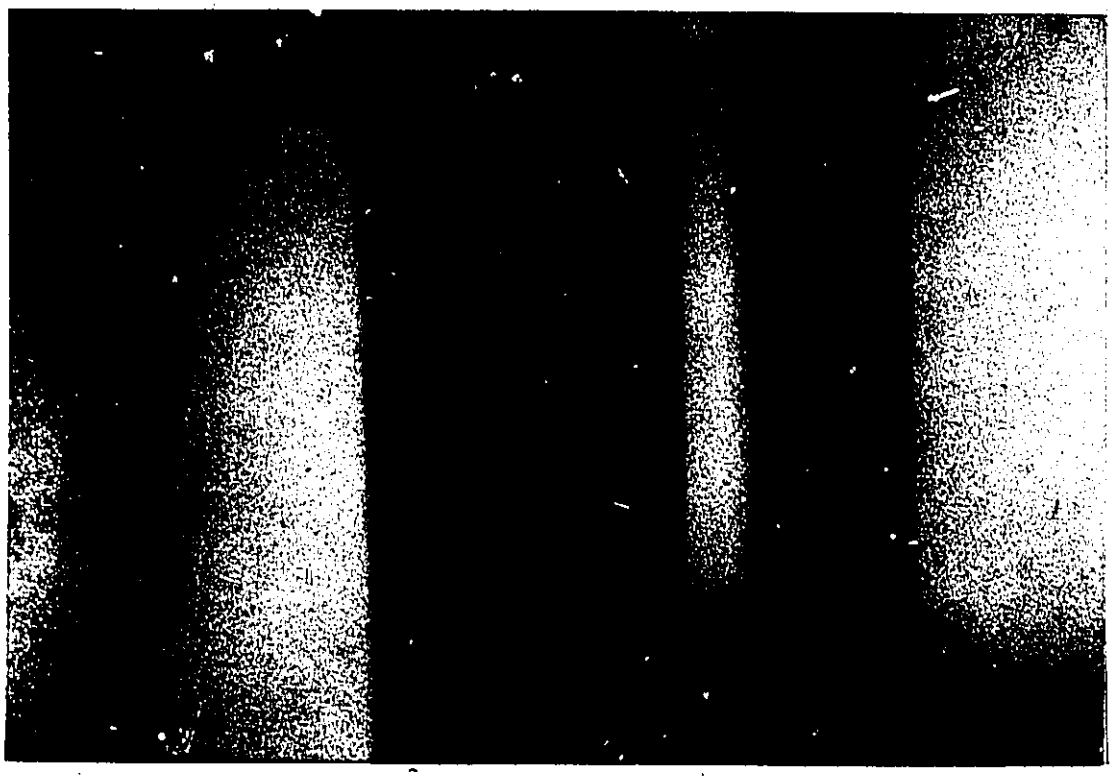
4.2 Results And Discussion

The experimental results are shown in Figures 4.5 through 4.7. These results are from [31]. The results presented are the local wall superheat at the topmost thermocouple elevation. In these figures, the surface heat flux has been shown as the dependent variable, though the wall heat flux is the experimentally controlled variable. This was done so that the results presented are consistent with typical boiling studies done in the past.

From Figures 4.5 and 4.6, we see that there is an abrupt drop in temperature after boiling inception. This temperature drop was about 25 degrees C, and a superheat of about 45 degrees C is required to initiate nucleation, under the test conditions. At about 48 degrees C, vigorous boiling erupted all over the test surface. This observed temperature drop of 25 degrees C, is higher than the results obtained by other researchers [11,12,13]. Also, note the hysteresis effects of the heat transfer coefficients. This hysteresis was obtained by cooling all sites, until they were quenched or made dormant. This is similar to the results observed in pool boiling by Corty and Foust [7] and Sabersky and Gates [9]. The sites were reheated, and we can observe the substantial difference in the behaviour of sites between the "cold" startup

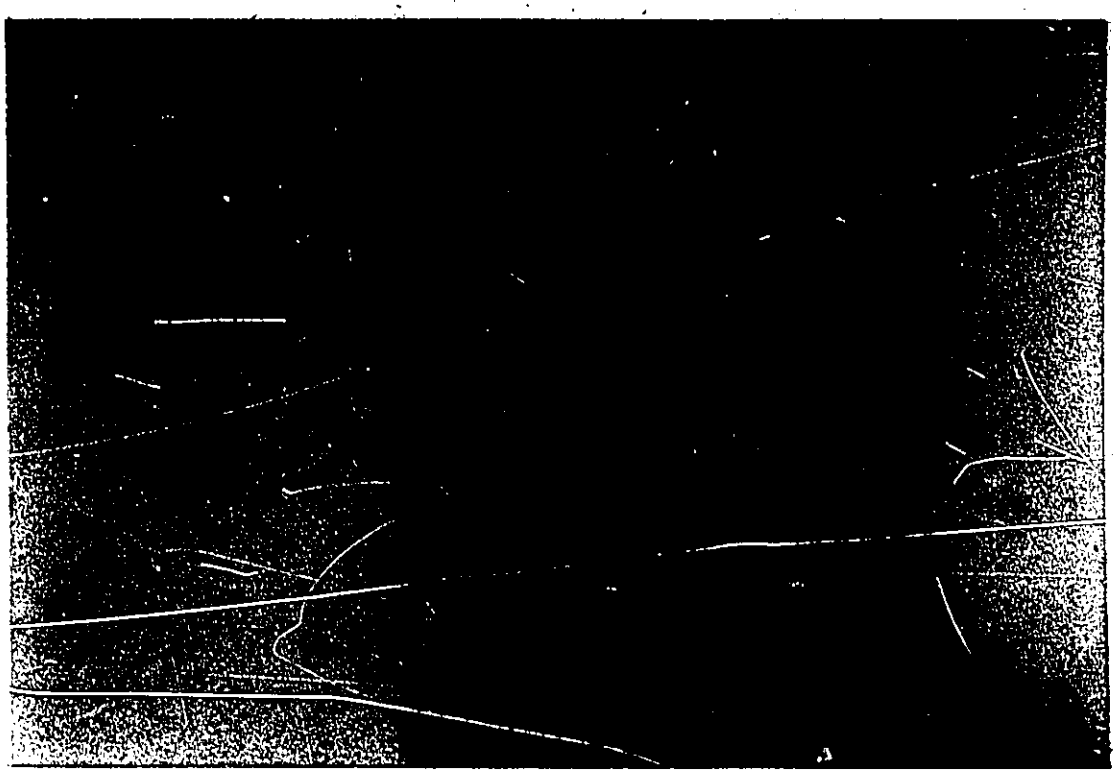


4.1 a

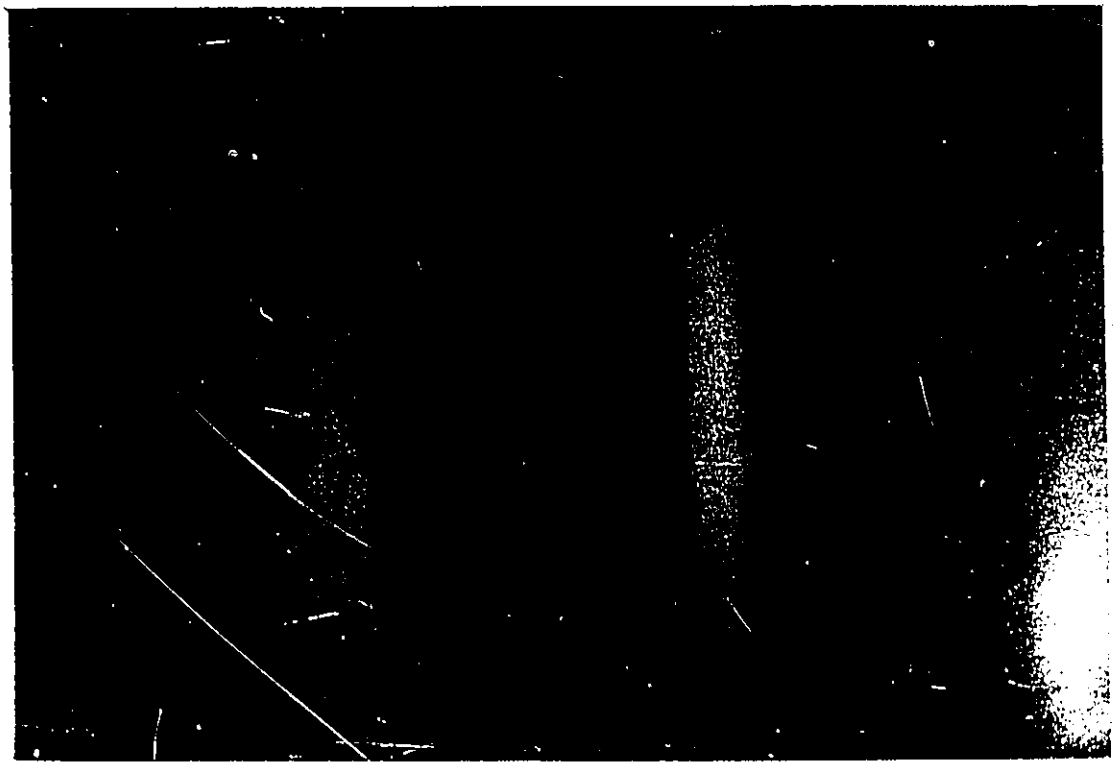


4.1 b

4.1 PHOTOGRAPH OF THE FLOW IN THE TEST SECTION AFTER BUBBLE INITIATION

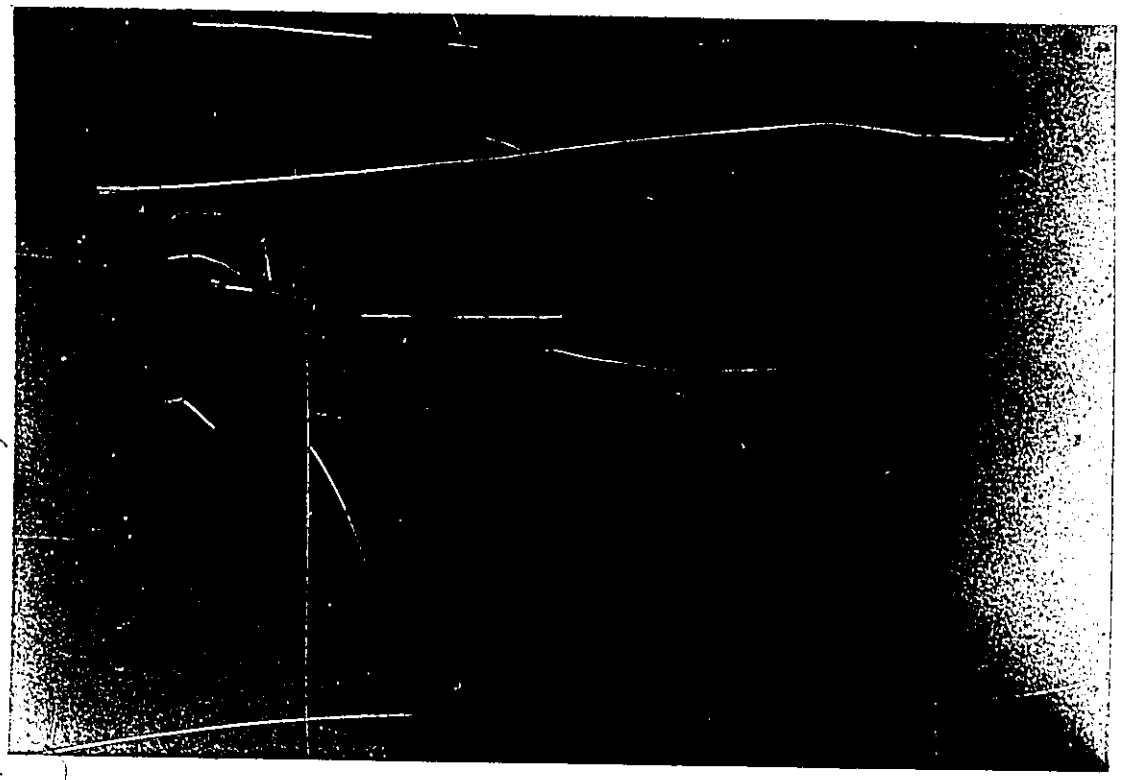


4.2 a

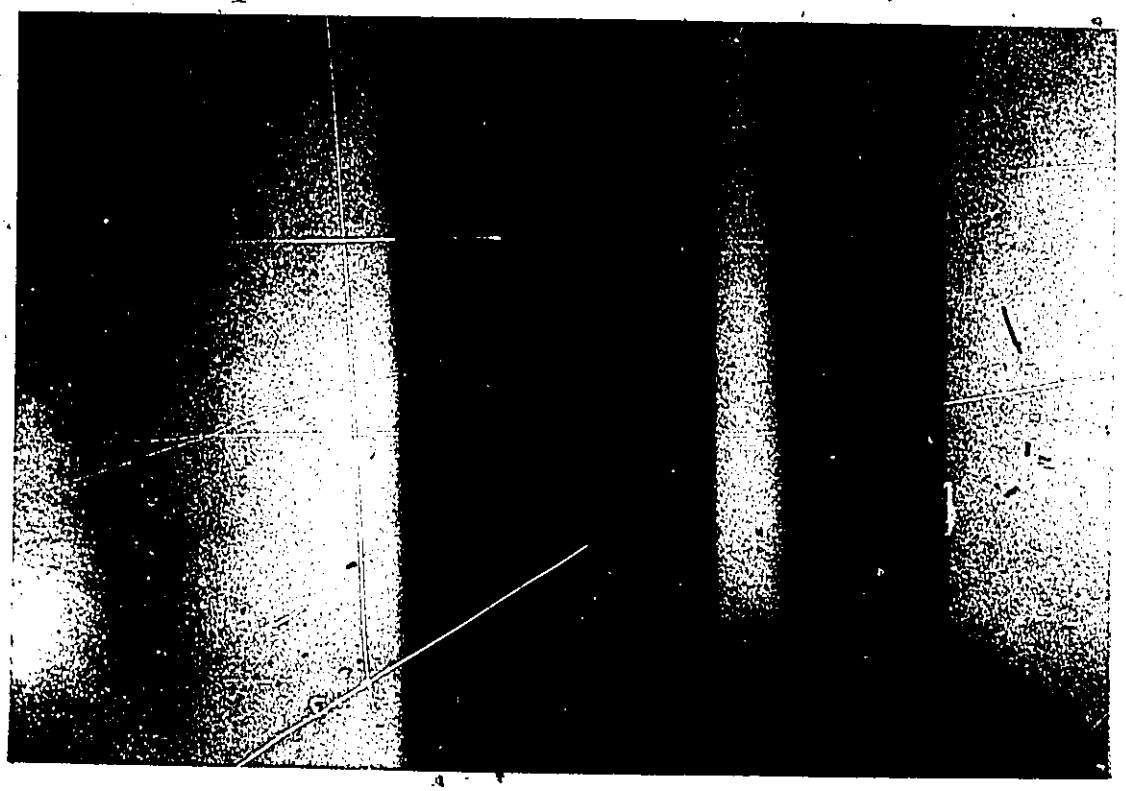


4.2 b

4.2 PHOTOGRAPH OF THE FLOW IN THE TEST SECTION SHOWING VIGOROUS SPONTANEOUS BOILING

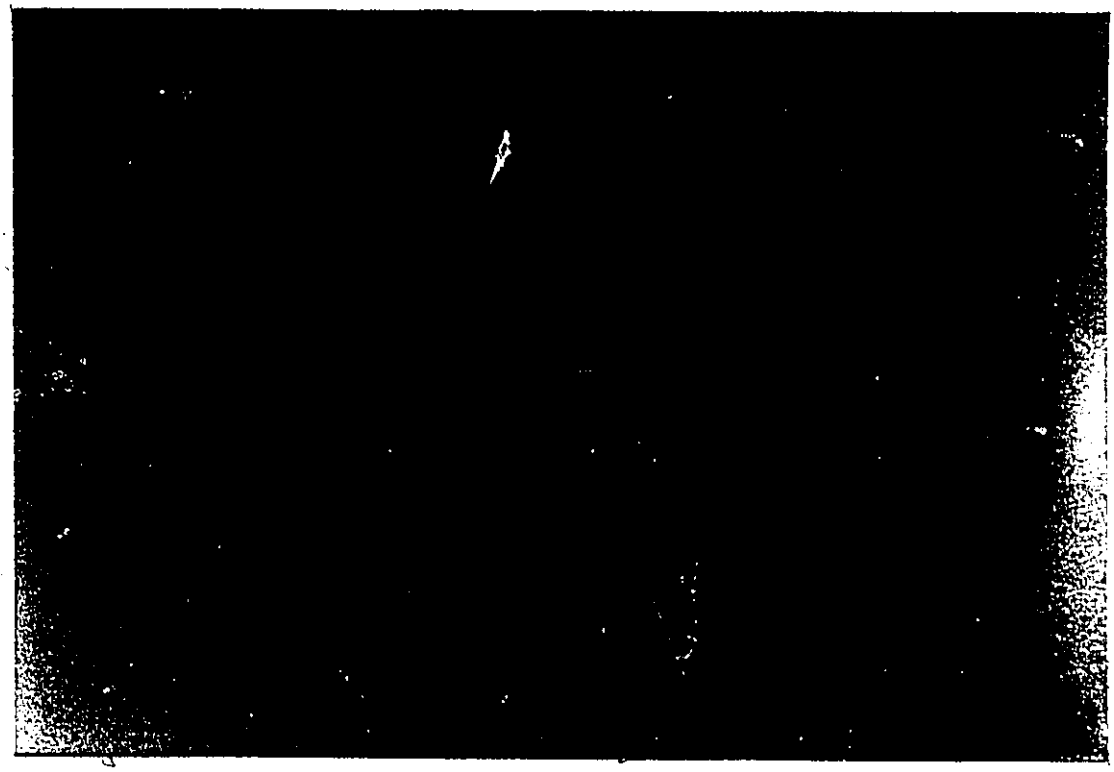


4.3a

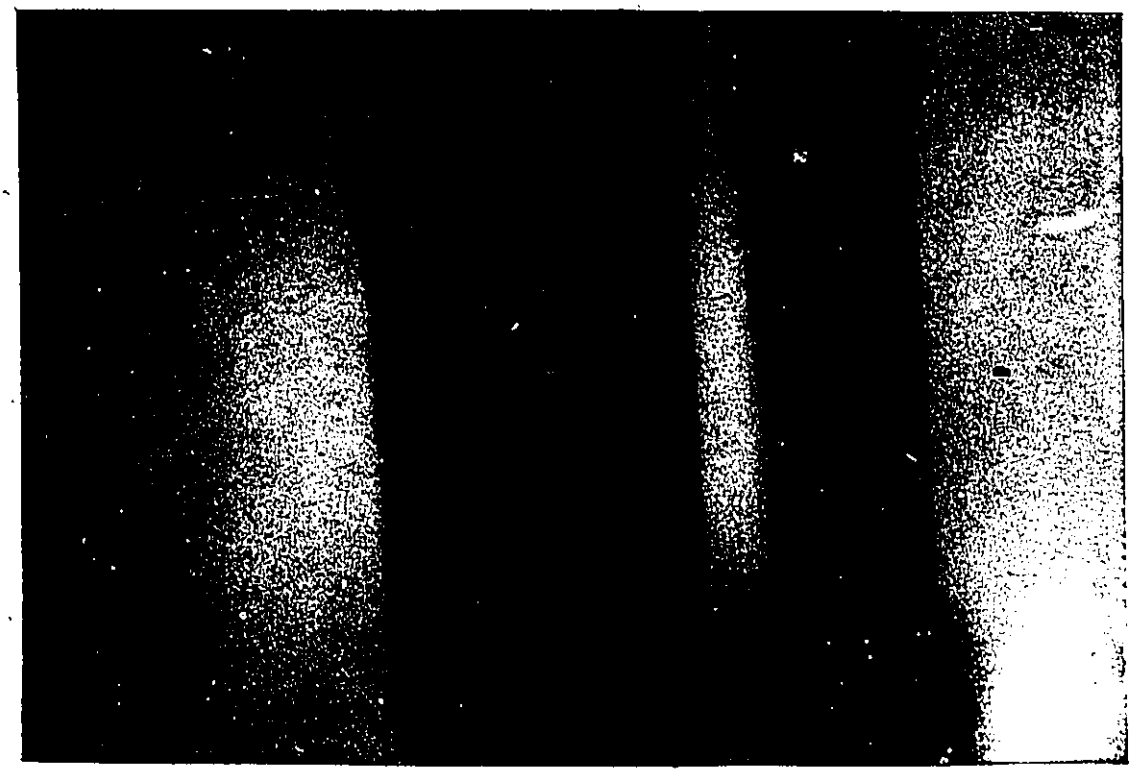


4.3b

4.3 PHOTOGRAPH OF THE FLOW IN TEST SECTION BEFORE DORMANCY

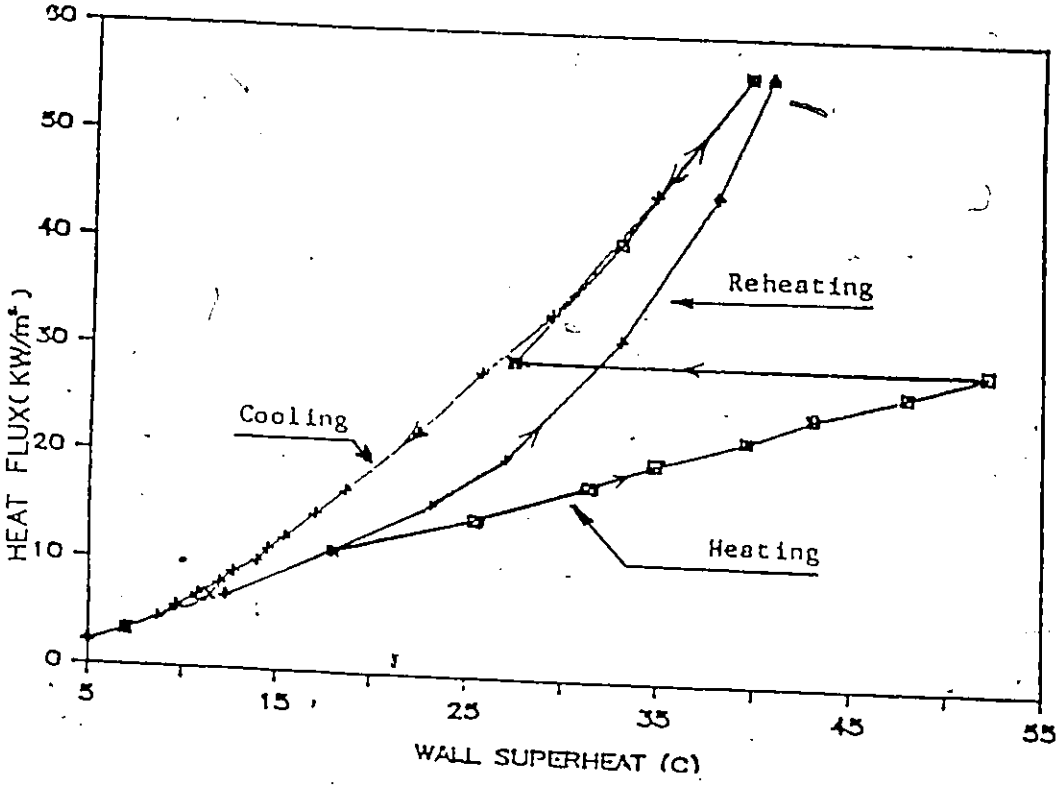


4.4 a

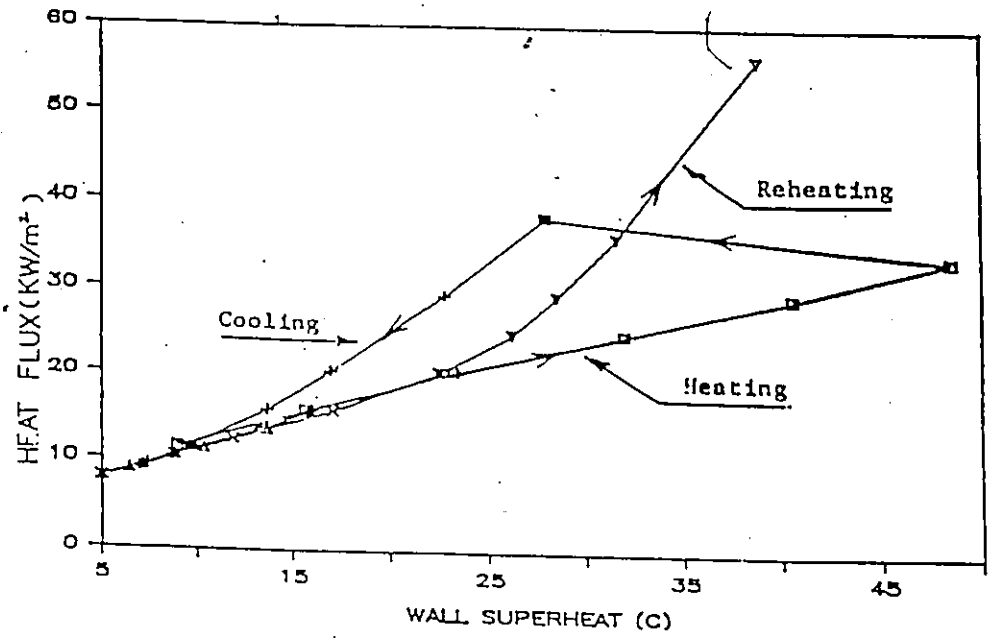


4.4 b

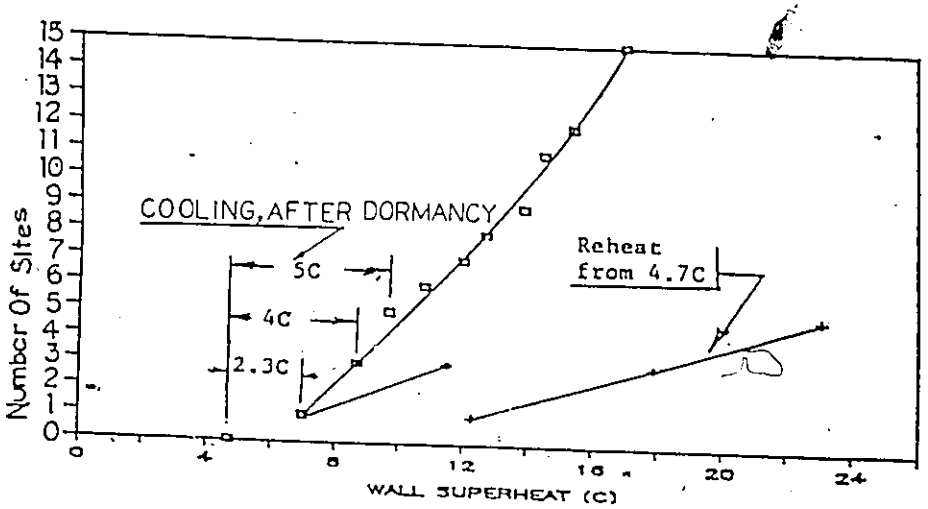
4.4 PHOTOGRAPH OF THE FLOW IN TEST SECTION SHOWING SMOOTHER TRANSITION TO VIGOROUS BOILING



4.5 Heat Flux vs Wall Superheat for a pressure of 1.5 b, inlet Sub cooling of 3 C, and mass flux of 390 kg/sm^2.



4.6 Heat Flux vs Wall Superheat for a pressure of 1.9 b, inlet sub-cooling of 11 C, and mass flux of 390 kg/sm^2.



4.7 Number of Nucleation Sites vs Wall Superheat for a pressure 15 bar, mass flux of 390 kg/s m^2, and subcooling of 3 C.

and the reheating process. In the second case, much higher heat transfer coefficients were achieved along with a smoother transition to vigorous boiling.

For the test at 1.9 bars, Figure 4.6, when the wall superheat was decreased to 6.8 degrees C all sites disappeared, and a number of

sites reappeared when the wall superheat was increased. Subsequently, when the wall superheat was reduced to 5 degrees C, one site reactivated, from its dormant state. When the wall superheat was reduced to 2 degrees C, the sites reactivated, however , they behaved as though they were quenched, and required the same large wall superheat to initiate boiling from the "cold" startup. From Figures 4.5 and 4.6 we also infer that the inlet subcooling and system pressure affects the incipient wall superheat. This is similar to the results of Chen [10]. Also , we note that the temperature drop subsequent to boiling incipience is a function of subcooling and system pressure.

As shown in Figure 4.7, at a superheat of 9.6 degrees C, 5 sites were active, at 8.6 degrees C superheat, 3 sites were active and at 6.9 degrees C superheat, 1 site was active. When the superheat was reduced to 4.6 degrees C, none of the sites were quenched, but only made dormant. These same sites reactivated at wall superheats of 12, 18 and 23 degrees C as against 53 degrees C wall superheat required to initiate from "cold" startup.

It was also observed that at maximum flow rates, it was not possible to discern individual nucleation sites.

4.3 Conclusion And Recommended Future Work

The design and commisioning of a forced convection heat transfer loop is reported for the study of:

1) the incipient boiling wall superheat,

2) the nucleate boiling quench wall superheat, and nucleate boiling dormancy superheat,

3) the hysteresis associated with the above phenomena for cycles of increasing followed by decreasing heat flux,4) the verification of the quench/ dormancy model.

Though there is a large volume of literature available, none has attempted to establish a quench/ dormancy model of nucleation sites in the incipient boiling region to explain the hysteresis effect. Hence, it was decided to build an experimental apparatus to gain a better understanding of above mentioned phenomena, so that mathematical models may be developed, which would aid us in predicting these phenomena.

The existing literature and mathematical models were reviewed in Chapter 2. The design, instrumentation and commissioning of the forced convection heat transfer loop was described in Chapter 3. Preliminary runs were carried out to check the loop design and debug the data acquisition program. It was found that the loop and

its associated instrumentation is capable of studying the physical phenomena for which it was designed. Incipient boiling, quench superheats were measured, and their respective heat transfer coefficients evaluated. The hysteresis effect is very significant in this region. However, a major limitation of the present design is that it is not possible to observe individual sites at high flow rates.

Based on these results it is recommended that in the future, the following research be done:

1) Investigations should be carried out to study the phenomena for which the loop has been designed and the results compared with the existing correlations in the literature,

2) The results obtained should be used to develop a practical quench/ dormancy model to simulate hysteresis phenomena on a computer,

3) A comprehensive study of the pressure and temperature effects on boiling incipience needs to be performed,

4) Extend the applicability of experimental results, to a larger range of test section/ test fluid combinations. This might necessitate the upgrading of the main heater capacity.

An optical device should be used in conjunction with the photographic recording device to capture the first nucleation and quench event.

5) Finally, a time dependent model for the prediction of boiling

initiation and quench (given a certain pressure temperature history) be developed.

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APPENDIX- A

Mathematical Models In Boiling Dormancy

semi-sphere, we have from Helmholtz's equation,

$$P_v - P_f = \frac{2\sigma}{r_c} \quad \text{---(A4)}$$

For thermal equilibrium,

$$T_v = T_f \quad \text{at } y = \lambda r_c \quad \text{---(A5)}$$

Bergles and Rohsenow assume $\lambda=1$; Hsu recognises it as a function of contact angle and cavity geometry, but suggests a typical value of 2.

Since $P_v > P_f$, and the vapour is at saturation temperature corresponding to its pressure, the surrounding liquid must be superheated with respect to its pressure. This superheat is related to its pressure by the Clausius-Clapeyron equation

$$\frac{dP}{dT} = \frac{h_{fg}}{\upsilon T} \quad \text{---(A6)}$$

Considering the vapour as a perfect gas, and eliminating υ, we have,

$$\frac{dP}{dT} = \frac{h_{fg} P}{R T^2} \quad \text{---(A7)}$$

Assuming there is no variation in h_{fg} within the temperature limits in which we intend integrating, and integrating equation (A7) between limits T_s, P_s and T_v, P_v, we have,

$$T_v - T_s = \frac{R T_s T_v \ln (1 + 2\sigma / r_c P_s)}{h_{fg}} \quad \text{---(A8)}$$

Since, $2\sigma/r_c P_s \ll 1$, $\ln (1 + 2\sigma/r_c P_s) \doteq 2\sigma/r_c P_s$.

Equation (A8) can be put in general as,

$$T_v = T_s (1 + A/r_c) \quad \text{---(A9)}$$

Equations (A2) and (A9) are shown in Figure A.1.

semi-sphere, we have from Helmholtz's equation,

$$P_v - P_f = \frac{2\sigma}{r_c} \text{---(A4)}$$

For thermal equilibrium,

$$T_v = T_f \quad \text{at } y = \lambda r_c \text{---(A5)}$$

Bergles and Rohsenow assume λ=1; Hsu recognises it as a function of contact angle and cavity geometry, but suggests a typical value of 2.

Since $P_v > P_f$, and the vapour is at saturation temperature corresponding to its pressure, the surrounding liquid must be superheated with respect to its pressure. This superheat is related to its pressure by the Clausius-Clapeyron equation

$$\frac{dP}{dT} = \frac{h_{fg}}{vT} \text{---(A6)}$$

Considering the vapour as a perfect gas, and eliminating v, we have,

$$\frac{dP}{dT} = \frac{h_{fg} P}{R T^2} \text{---(A7)}$$

Assuming there is no variation in h_{fg} within the temperature limits in which we intend integrating, and integrating equation (A7) between limits T_s, P_s and T_v, P_v, we have,

$$T_v - T_s = \frac{RT_s T_v \ln (1 + 2\sigma / r_c P_s)}{h_{fg}} \text{---(A8)}$$

Since, $2\sigma/r_c P_s \ll 1$, $\ln (1 + 2\sigma/r_c P_s) \doteq 2\sigma/r_c P_s$.

Equation (A8) can be put in general as,

$$T_v = T_s (1 + A/r_c) \text{---(A9)}$$

Equations (A2) and (A9) are shown in Figure A.1.

For a given fluid, given flow rate, bulk fluid temperature, heat transfer coefficient are fixed. This in turn fixes the intercept of fluid temperature profile with $T = T_{bulk}$ line.

The variables in equations (A2) and (A9) are T_w, q and r_c. The fluid temperature profile is fixed by either fixing T_w or q.

When equation (A2) is tangent to equation (A9), the conditions correspond to nucleate boiling For $q > q_{dormancy}$, two values of r_c are obtained. Cavities lying outside the range of those two solutions are ineffective dormancy sites, while those lying in the range are effective, but not necessarily active sites.

For incipient nucleate boiling dormancy,

$$T_s(1+A/r_c) = T_b + \frac{q}{h} - \frac{\lambda q r_c}{k} \text{---(A10)}$$

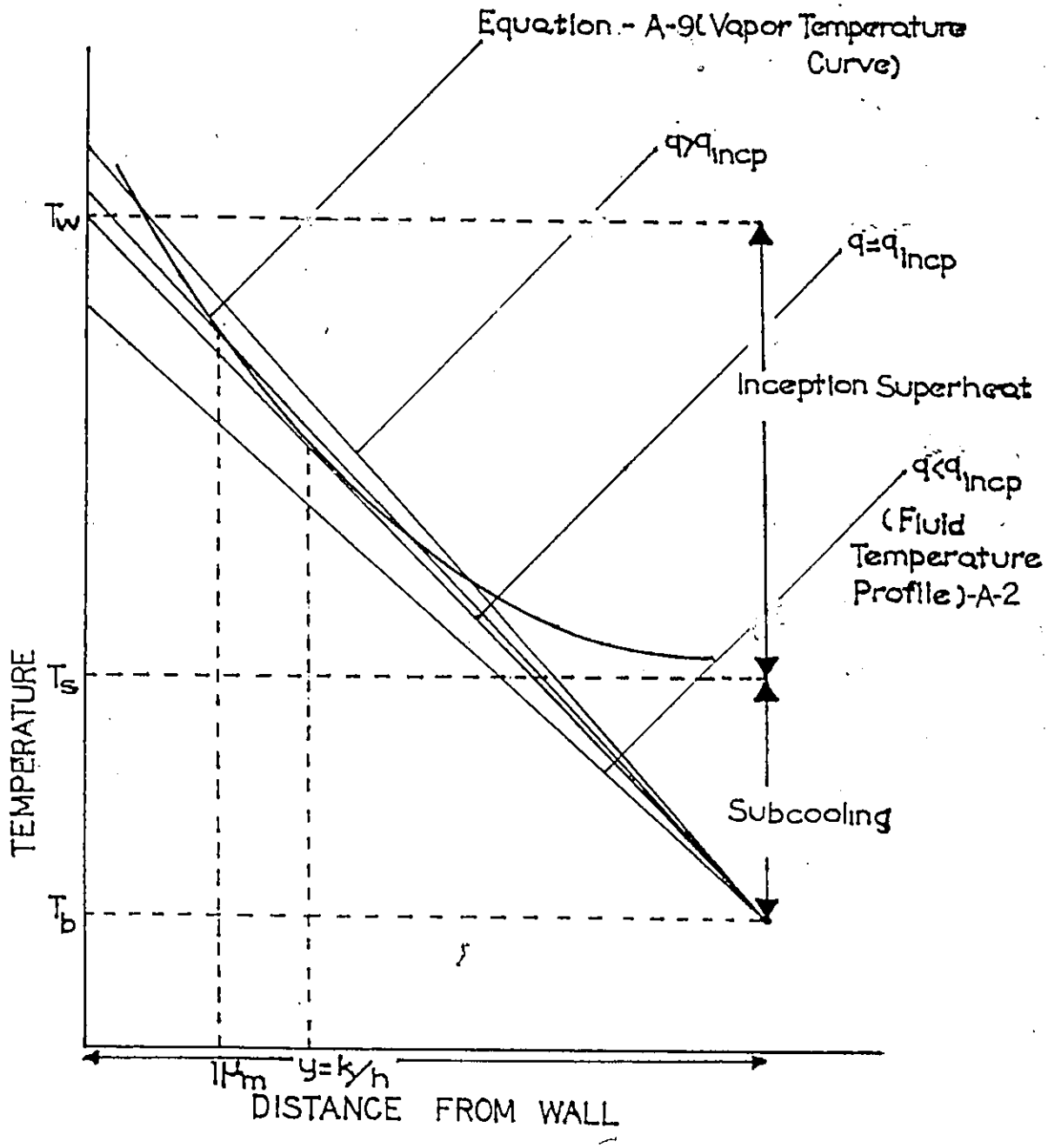
$$\text{or } \frac{q \lambda r_c^2}{k} + r_c(\Delta T_{sub} - q/h) + AT_s = 0 \text{---(A11)}$$

Equation (A-11) is a quadratic in r_c, and for incipient dormancy, we must have equal roots.

$$\therefore (\Delta T_{sub} - q/h)^2 = \frac{4 q \lambda A T_s}{k} \text{---(A12)}$$

Equation (A-12) is a quadratic in q, which results in two roots. The root with the negative sign corresponds to the solution in the third quadrant, where $T_w < T_b$, and therefore has no physical significance.

$$q = h\left[\Delta T_{sub} + \frac{2\lambda T_s hA}{k} + \left[\left[\Delta T_{sub} + \frac{\lambda hA T_s}{k}\right]\left[\frac{4\lambda T_s hA}{k}\right]\right]^{0.5}\right] \text{-(A 13)}$$



A.1 SCHEMATIC OF HSU, BERGLES AND ROHSENOW'S MODEL

The sum of second and third terms in equation (A 13) gives the value of $(T_w - T_b)$ necessary for incipient nucleate boiling dormancy, where

$\lambda = 1$; $A = \dfrac{2\sigma}{1.25\rho_g h_{fg}}$ (Bergles and Rohsenow's model)

and $\lambda = 2$; $A = \dfrac{2\sigma}{\rho_g h_{fg}}$ (Hsu's model)

APPENDIX - B

Heat Losses Through The Push Rod and Sleeve End Faces Calculations

APPENDIX-B

The heat lost from the push rod and the flat ends of the sleeve is presented in this section. These calculations were done to make a decision, if the inside of the test pipe should be filled with insulation to minimize the heat losses.

B.1 Heat Lost Through The Push Rod

Assumptions:

1. The rod is infinitely long.

2. The temperature of air is an arithmetic mean of heated wall and room temperature.

Temperature of wall during inception $\doteq 115^{\circ}C + 18^{\circ}C = 133^{\circ}C$.

Room temperature $\doteq 21^{\circ}C$.

$\therefore$ Mean temperature $= (133 + 21)/2 = 77^{\circ}C$.

Air properties at $77^{\circ}C$;

$\rho = 0.9950\ kg/m^3$

$\mu = 208.2 * 10^{-7}\ N\text{-}s/m^2$

$\beta = 1/350\ {}^{\circ}K^{-1}$

length of the push rod = 0.60975 m.-1

Temperature difference between the heated wall temperature and ambient air = $\Delta T = 112^{\circ}C$.

Pr = 0.70

$$Gr = \frac{l^3 \rho^2 g \beta \Delta T}{\mu^2} = 1625380655$$

$\therefore$ (Gr)(Pr) = 1137766459

$\therefore$ $Nu = 0.129(1137766459)^{0.33} = 134.67$

$\therefore$ $h_{air} = \frac{Nu\ k_{air}}{D_{o\text{-}rod}} = \frac{(134.67)(30*10^{-3})}{0.0127} = 318.118 W/m^2\ C.$

$\therefore$ $m^2 = \frac{h_{air}\ P}{k_{air}\ A_x} = 20038.93$

$\therefore$ m = 141.56

Since m l > 0.99, we can assume the rod as an infinitely long fin.

$\therefore$ $q_{\text{lost from fin}} = -(h\ P\ k\ A)^{0.5}(\Delta T)$

$= (318.118*\Pi*0.003175*20*\Pi*0.25*0.003175*0.003175)^{0.5}112$

= 2.5 W.

B.2 Heat Lost From The Flat Ends Of The Sleeve[24]

Assumptions:

1. The convection around the sleeve top and bottom is laminar and free convection.

D_{hyd} of the sleeve top surface = 0.0121412 m.

$\therefore$ characteristic dimension = 0.9 D_{hyd} = 0.01092708 m.-1

Using the same properties as in the previous calculation and the same temperature difference, we get,

Gr = 9349.344 and (Gr) (Pr) = 6544.54.

Hence, our assumption of laminar range is correct.

$\therefore$ $h_{air} = 0.27\ (\Delta t/D)^{0.25}$ (all in British units)

$\Delta t = 234^{\circ}F$ and D= 0.0398 ft.

$\therefore h_{air} = 2.36$ Btu/hr $ft^{2o}F = 13.397$ W/m^2 C.

$\therefore q_{lost\ from\ 1\ flat\ side} = h_{air} A_x \Delta T$

$= 13.397 * \frac{\Pi}{4} * (0.000402644 - 0.000062802)^{0.5} * 112$

$= 0.4$ W.

$\therefore q_{lost\ from\ 2\ flat\ sides} = 0.8$ W.

$\therefore$ Total heat lost from the sleeve and the push rod = 3.3 W.

= 1.5 % of maximum heat input.

Since this is negligible, it was decided not to fill the inside of the test pipe with insulation.

APPENDIX - C

End Flange Design Calculations

APPENDIX-C

This section presents the design of the end flange of the test section as in [29]. For ease of welding the test section inlet and outlet copper tees, and also for ease of machining, a brass flange was chosen. Since brass flanges are not off the shelf items, they had to be designed specifically for our requirements. Since the units in graphs in [29] are British, the same has been retained here.

1.Since the operating pressure < 300 Psi, and operating temperature < 700°F, a ring type flange (loose flange) was chosen.

2. Bolt material :BS 3692 , tensile strength = 65250 Psi.

3. Allowable bolt stress = 65250/4 = 16312.5 Psi.

4. Bolt area(M8 bolt) = $\frac{\Pi(8/25.4)^2}{4}$ = 0.2474 in^2

5. Maximum bolt load = (16312.5)(0.2474) = 4035.7125 lb_f

6. Bolt circle diameter = 2.8 in_f

7. Gasket centre circle diameter = $\frac{(\text{gasket O.D} + \text{gasket I.D})}{2}$

= (1.062+1.65)/2=1.356 in.

8. Moment on flange under no load conditions=

$$\frac{(4035.7125)(2.8-1.356)}{2} = 2913.78\ lb_f\text{-in.}$$

9. f for brass = 81000/4 = 20250 Psi.

10. Outside diameter of flange = 2.8 in.

11. Inside diameter of flange = 1.375 in. Therefore, the ratio of

diameters = Ω = 2.8/1.375 = 2.036.

12. Y (factor involving Ω) = $\dfrac{0.66845 + \dfrac{5.7169(\log_{10}\Omega)\Omega^2}{\Omega^2-1}}{\Omega} = 2.34$

13. Flange thickness = $\left[\dfrac{Y \text{ (Moment maximum)}}{f\ B}\right]$ = 0.4797 in.

= 12.18 mm.

APPENDIX - D

COOLER DESIGN CALCULATIONS

APPENDIX-D

COOLER DESIGN [28]

Assuming That All The Fluid Coming In Is Liquid

1. Heat to be removed = 225 = p

2. Recommended water velocity = 0.610- 0.915 m/s and 0.610 m/s was chosen.

3. Inlet water temperature = 15°C.

4. ΔT for water in the cooler = 0.2° C. (arbitarily chosen value)

5. Mass flow rate of water in the cooler

$\dot{m}_w = p/C_p\ \Delta T = \frac{225}{4186(0.2)} = 0.250 kg/s.$

$\therefore$ Volumetric flow rate of water = Q = 0.00025 m^3/s.

6. $\therefore$ crossectional area of the water flow area = Q/u

$0.000250/0.610 = 0.0004098\ m^2 = 4.10\ cm^2$.

i.e., $4.1 = \frac{\Pi\ (D_o^2 - 2.2225^2)}{4}$. Hence, a 31.75mm inside 38.1 mm outside diameter pipe was chosen.

7. Calculation of overall heat transfer coefficient

$Nu_{\text{Refrigerant side}} = 0.0296\ Re^{0.8}\ Pr^{0.333}$

$Pr = 4.2147$

$Re_{\text{minimum}} = 7100$

$\therefore Nu = 57.62 = \frac{h_{\text{inner side}} \cdot D_{\text{inner side}}}{k}$

$\therefore h_{\text{inner side}} = (57.62)(0.07793)/0.0201 = 223.38\ W/m^2\ C$

Similarly, Re_{water} = 5570 and Pr_{water} = 5.83

$\therefore$ Nu = 52.87

$\therefore$ $h_{outer\ side}$ = (52.87)(.613)/0.022225 = 1458.23 W/m^2 C

K_{copper} = 401 W/m K

Thickness of copper pipe = 0.002125 m

$\therefore$ $U_{overall}$ = 193.36 W/ m^2 C

8. Calculation of effectiveness

Temperature of refrigerant (saturation at 2 bars)= 45° C

$\therefore$ $p_{maximum}$ = (0.0003119)(1480)(0.907)(45-15) = 12600 W

$\therefore$ ϵ = 225/12600 = 0.0179

9. Calculation of R

	R-11	R-113
Maximum	$\dot{m}C_p$ = 0.418 kJ/s C	0.497 kJ/s C
Minimum	$\dot{m}C_p$ = 0.0418 kJ/s C	0.0497 kJ/s C

$\dot{m}C_p$ for water = 1.0465 kJ/s C.

Since $(\dot{m}C_p)_{hot} < (\dot{m}C_p)_{cold}$

$$R = \frac{(\dot{m}C_p)_{hot}}{(\dot{m}C_p)_{cold}} = 0.04$$

10 Calculation of NTU

$$NTU = -\frac{1}{R-1} \ln\left[\frac{\epsilon - 1}{\epsilon R - 1}\right]$$

= 0.018

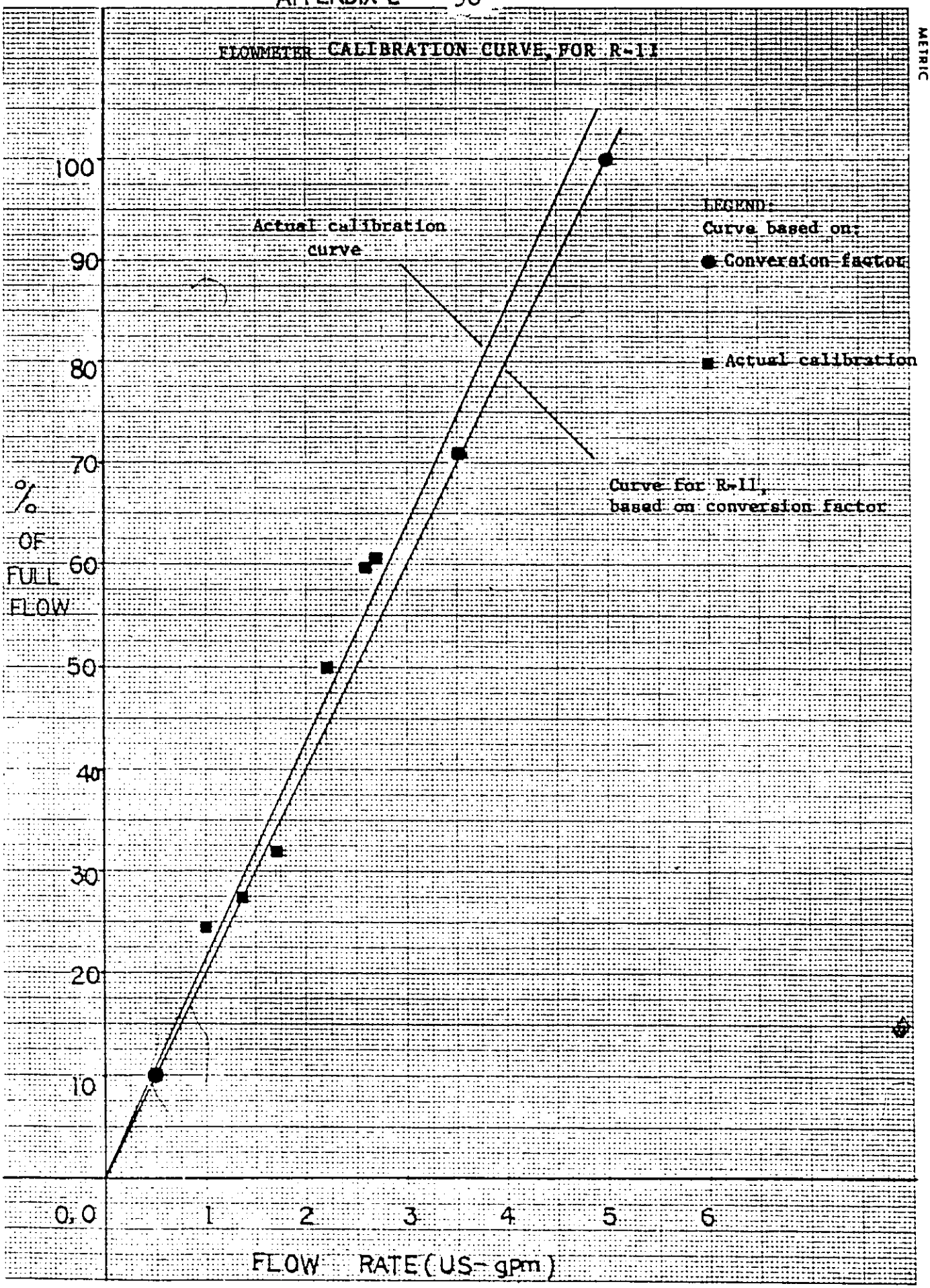
$\therefore$ $$L = \frac{NTU(\dot{m}C_p)_{minimum}}{U \Pi D_o}$$

= 0.56 m

However, 0.61 m length (10% over area) was provided for reasons of safety.

APPENDIX - E

Flowmeter Model # FL-1504-A, Calibration Curve, with R-11


FLOWMETER CALIBRATION CURVE, FOR R-11
METRIC
LEGEND:
Curve based on:
Conversion factor
Actual calibration
Actual calibration curve
Curve for R-11, based on conversion factor
% OF FULL FLOW
100
90
80
70
60
50
40
30
20
10
0.0
1
2
3
4
5
6
FLOW RATE (US-gpm)

APPENDIX-F

Loop Pressure Drop Calculations

APPENDIX - F

F.1 Pressure Drop In The Test Loop Calculations

Note that in these calculations, all the fluid properties are evaluated at 20° C.

F.1.1 Equivalent Length Calculations.(19 mm diameter pipe, equivalent lengths are calculated according to [27].)

DESCRIPTION	EQUIVALENT LENGTH (m)
1. Straight pipe (outlet from pump)	0.4572
2. Standard tee (inlet to preheater)	1.3106
3. Straight pipe (preheater shell)	0.2286
4. Standard tee (outlet from preheater)	1.3106
5. Globe valve	5.4864
6. Standard 90° elbow	0.5334
7. Straight pipe	0.2286
8. Standard tee (inlet to test section)	1.3106
9. Standard tee (outlet from test section)	1.3106
10 Straight pipe (heat exchanger)	1.4478
11.Standard 90° elbow	0.5334
12. Straight pipe (down comer)	1.8228
13. Standard 90° elbow	0.5334
14. Straight pipe (leading to pump suction)	1.7526
15. Standard 90° elbow	0.5334
16. Standard 90° elbow	0.5334
17. Sudden expansion (strainer inlet d/D = 1/4)	0.5334

18. Suden contraction (strainer outlet d/D =1/4) 0.2591

$\sum$ Equivalent length (m)(excluding the test section)= 19.909

F.1.2 Pressure Drop In The Pipe (Excluding the test section)

Area of cross section of pipe = $\frac{\Pi\ (0.01905)^2}{4}$ =0.0002851 m^2

Flow rate (minimum) = 0.4933 gpm (US)=0.000031119 m^3/s

Flow rate (maximum) =4.933 gpm (US)= 0.00031119 m^3/s

$\therefore$ G (minimum, R-11) = $\frac{Q_{min}\rho}{A_x}$ = $\frac{(0.000031119)(1480)}{0.0002851}$

= 161.5507 kg/s m^2

and G (maximum, R-11) = 1615.507 kg/s m^2

Similarly, for G (minimum, R-113) = $\frac{(0.000031119)(1480)}{0.0002851}$

= 164.6102 kg/s m^2

$\therefore$ Reynolds number (minimum, R-11) = $\frac{G\ D}{\mu}$

= $\frac{(161.5507)\ (0.01905)}{(\frac{1.488}{3600})(1.0502)}$ =7090

and Reynolds number (maximum, R-11) = 70900

Similarly, for Reynolds number (minimum, R-113)= 6198

and Reynolds number (maximum, R-113) = 61980

Using Blasius correlation for turbulent flow, in smooth tubes and for Reynolds number $\leq$ 100000,

Darcy's friction factor = $\frac{0.316}{Re^{0.25}}$ = $f=\frac{(\Delta P/L)2gD}{V^2}$

$\therefore$ f(minimumflow,R-11)=0.034437 and

for maximum flow, = 0.019365

and f (minimum flow, R-113) = 0.03561 and,

for maximum flow =0.020027

$\therefore (\Delta P/L)_{maximum}$ for R-11 =0.06178 m/m

and $(\Delta P/L)_{maximum}$ for R-113 = 0.064167 m/m

$\therefore (\Delta P)_{maximum}$ = 1.2437 m of R-11 and, is = 1.2918 m of R-113.

F.1.3 Pressure Drop In The Flow Meter

From manufacturer's catalog,

$(\Delta P)_{flowmeter}$, maximum = 0.3302 m of water column.

$(\Delta P)_{flowmeter}$, maximum= 0.223 m of R-11

and $(\Delta P)_{flowmeter}$= 0.219 m of R-113

F.1.4 Pressure Drop In The Test Annulus

Cross sectional area of the test annulus =

$$\frac{\Pi[(0.0254)^2 - (0.022225)^2]}{4} = 0.000118775 \text{ m}^2.$$

Hydraulic diameter = 0.003175 m.

For R-11, G(minimum)=388 and G(maximum)= 3880.

and for R-113, G(minimum)= 395 and G(maximum)=3950.

For R-11, Reynolds number (minimum)= 2836 and

Reynolds number (maximum) =28361

Similarly, for R-113, Reynolds number (minimum)=2479 and

Reynolds number (maximum) = 24794.

For R-11, Darcy's friction factor (minimum)= 0.0433 and

Darcy's friction factor (maximum) = 0.02435.

For R-113, Darcy's friction factor (minimum)=0.04478 and

Darcy's friction factor (maximum)= 0.02518.

$\therefore$ the $(\Delta P)_{maximum}$ for R-11 in the test section=1.2437 m.

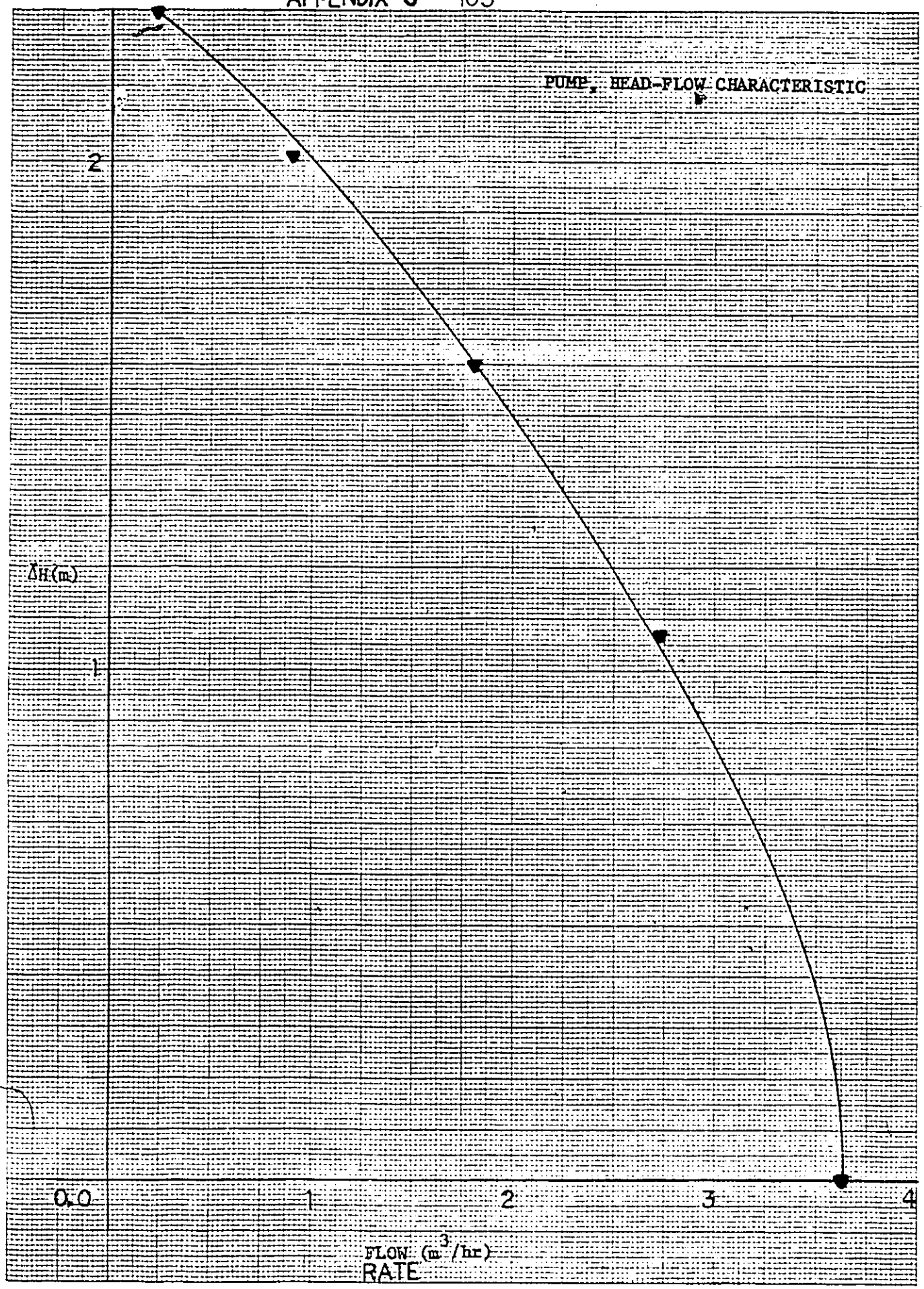
and $(\Delta P)_{maximum}$ for R-113 in the test section = 1.2918 m.

$\therefore$, the maximum $(\Delta P)_{loop}$ for R-11 = 2.7 m of R-11 and

the maximum $(\Delta P)_{loop}$ for R-113 = 2.8 m of R-113.

APPENDIX - G

Pump Model # J-7004-54, Head-Flow Characteristic Curve


PUMP, HEAD-FLOW CHARACTERISTIC
ΔH (m)
2
1
0.0
1
2
3
4
FLOW RATE (m³/hr)
METRIC

APPENDIX - H

"BOIL7" Data Acquisition Program

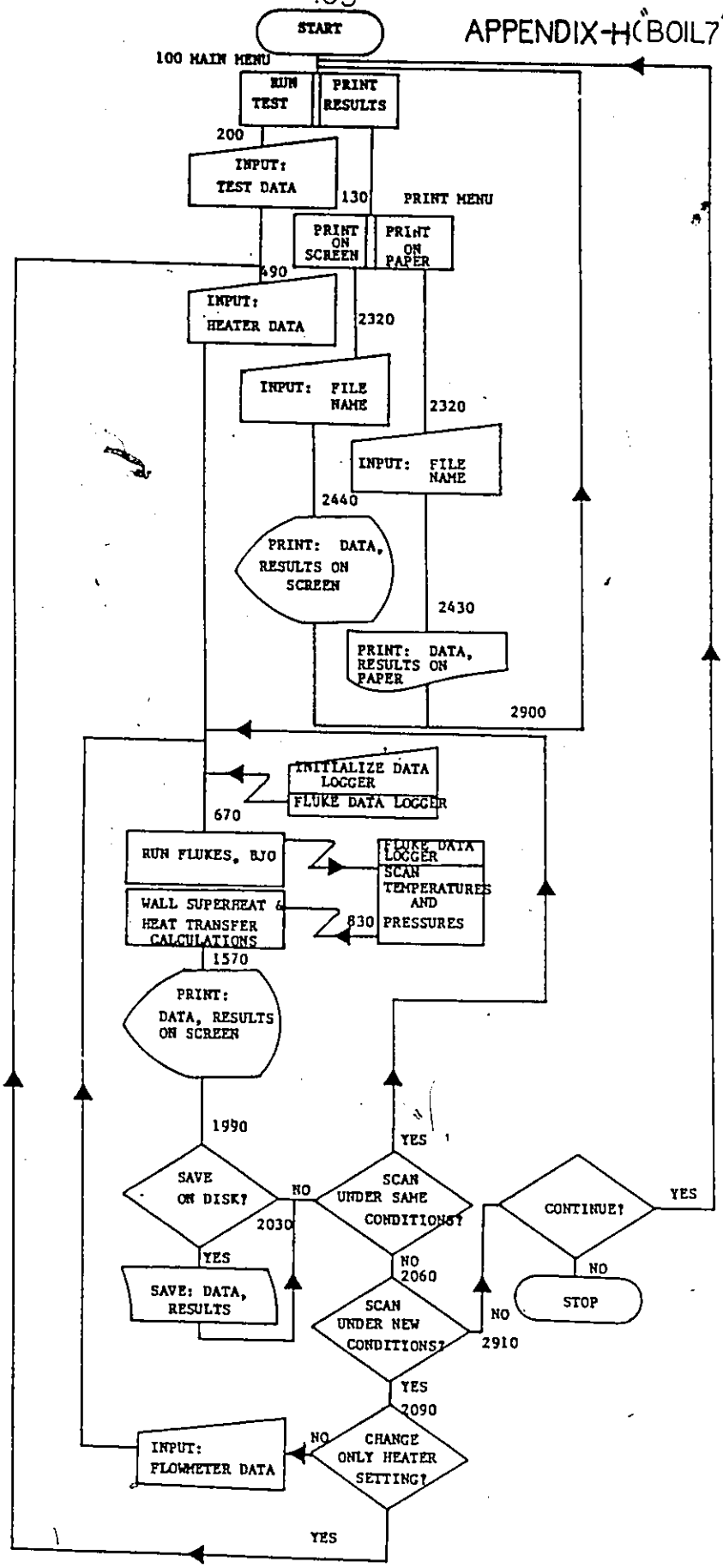
H.1 "BOIL7" Flowchart

H.2 "BOIL7" Variables Nomenclature

H.3 "BOIL7" Program Listing

H.4 "BOIL7" Sample Output

APPENDIX-H("BOIL7" FLOWCHART)



APPENDIX- H.2

COMPUTER PROGRAM "BOIL7" VARIABLES NOMENCLATURE

A) NUMERIC VARIABLES

Note: In the nomenclature below, the bottommost thermocouple is number one and the top most is number four.

VARIABLE DESCRIPTION

1) AP Atmospheric pressure (cm of mercury).

2) B1, B2, B3, B4 Boiling heat transfer coefficient at the first, second, third, fourth thermocouple.

3) B5, B6, B7, B8 Convective heat transfer coefficient at first, second, third, fourth thermocouple(W/m2 C)

4) BM Bench mark (=0, if it is the first sequence in the run, else =1).

5) BP Two phase pressure drop in the test section (Pa).

6) BU Two phase pressure drop per unit length (Pa/ cm).

7) C1, C2 Intermediate variables for calculating RC.

8) CF Correction factor for flow rate.

9) CM Heater bottom location with respect to inlet of the glass section (cm).

10) D1, D2 Intermediate variables for calculating fluid density.

11) DD Date of experiment.

12) DE Fluid density at room temperature (kg/m3).

13) DS Fluid density at flow meter inlet (kg/m3).

14) DV Differential pressure transducer (V).

15) FL Loop flow meter reading (%).

16) FR Volumetric flow rate (m3/s).

17) HV Heater power (W).

18) LA Latent heat of vaporization (kJ/kg).

19) LK Liquid thermal conductivity (W/m C).

20) LX Heat flux at the test surface (W/m2).

21) M% Month during which the test is being run.

22) MF Mass flow rate (kg/s).

23) MR Maximum radius of cavity in the test section(m).

24) MX Mass flux at the test section annulus (kg/s m2) .

25) P1, P2, P3, P4 Pressure at the first, second, third and fourth thermocouples respectively (Pa)

26) P5 Intermediate variable for calculating system pressure

27) PD Total pressure drop in the test section (Pa).

28) PI Pressure at test section inlet (Pa).

29) PU Single phase pressure drop per unit length (Pa/cm).

30) PV Preheater voltage (V).

31) RC Thermodynamic wall superheat required for bubble growth based on Bergles and Rosehnow's equation (C)

32) RE Reynolds number (#).

33) RS Thermodynamic wall superheat required for bubble growth based on Sato and Matsumura's equation (C).

34) S1, S2, S3, S4 Local wall superheat at first, second, third, fourth thermocouple locations (C).

35) SE Test sequence #.

36) SH Single phase pressure drop in the test section (Pa)

37) ST Surface tension (N/m).

38) T1, T2, T3, T4 Wall temperatures at the first, second, third and fourth thermocouple locations (C).

39) T5, T6, T7, T8 Bulk fluid temperature at the first, second, third and fourth thermocouple elevations (C)

40) TA, TB, TC, TD Fluid saturation temperature at the first, second, third and fourth thermocouple elevations (C)

41) TE Fluid temperature at test section exit (C).

42) TF, TR Non dimensional temperatures.

43) TI Fluid temperature at test section inlet (C).

44) TM Room temperature (C).

45) TN% Test #.

46) TS Saturation temperature at test section inlet (C).

47) TT Cooling water bath temperature (C).

48) TU Fluid temperature at cooler outlet (C).

49) VD Specific volume of vapor (m3/Kg).

50) VI System pressure output transducer (mV).

51) VL Intermediate variable for calculating dynamic viscosity

52) VQ Liquid dynamic viscosity (N s/m).

53) WA Heater wall temperature (C)

B) STRING VARIABLES

1) B$ Strings used to store comments about the run.

2) BF$ Filename (Saving file to disk).

3) C$ If "Y", then continue scan under the same experimental conditions.

4) D$ CHR$(4).

5) F$ If "Y", then print data, results from the disk.

6) H$ If "S", then print on screen and if "P", then print on paper, the data and results from the disk.

7) HR$ If "H", then only heater setting is to be changed, and if "F", then the flow rate setting is to be changed.

8) HS$ Heater setup time (Hours: Minutes).

9) K1$ If "Y", then the month input data is correct.

10) K2$ If "Y", then the test # input data is correct.

11) K4$ If "Y", then the atmospheric pressure, input data is correct.

12) K5$ If "Y", then the heater bottom location, input data is correct.

13) K8$ If "Y", then the loop flow meter reading,input data is correct.

14) K9$ If "Y", then the preheater voltage input data is correct.

15) L1$ If"Y", then the heater power input data is correct.

16) L2$ If"Y", the heater set up time input data is correct.

17) N$ If "Y", then scan under new experimental conditions.

18) R1$, R2$, R3$, R4$, R5$ Strings used to name the file (saving data, results to disk).

19) ST$ If "Y", store data, results on the disk.

20) T$ If "Y", run test.

21) VV$ Filename (retrieving file from data).

22) AK$, CN$, RC$, Z$ Control string variables for continuing further with the program.

APPENDIX-H.3(PROGRAM LISTING)

```
10  HOME
20  DIM CH(15)
30 D$ =  CHR$ (4)
40  PRINT  CHR$ (12)
50  PRINT  TAB( 10)"FORCED CONVECTION HEAT TRANSFER LOOP"
60  PRINT  TAB( 10)"DATA ACQUISITION PROGRAM"
70  PRINT  TAB( 10)"BY DINKAR SHIVPRASAD"
80  PRINT "**IF IT WORKS ---IT IS A FLUKE**"
90  HOME
100  INPUT "FOR RUNNING TEST,TYPE Y/N";T$: PRINT
110  IF T$ = "Y" THEN  PRINT "IF THIS IS THE FIRST RUN IN THE SEQUENCE,THEN IT SHOULD BE SING
     LE PHASE FLOW ONLY": PRINT : GOTO 230
120  IF T$ < > "N" THEN  GOTO 100
130  INPUT "FOR PRINTING DATA,RESULTS,TYPE Y/N";F$: PRINT
140  IF F$ = "N" THEN  GOTO 2910
150  IF F$ < > "Y" AND F$ < > "N" THEN  GOTO 130
160  INPUT "FOR PRINTING ON SCREEN TYPE S,ON PRINTER TYPE P";H$: PRINT
170  IF H$ = "S" THEN  GOSUB 2320: GOTO 100
180  IF H$ = "P" THEN  GOSUB 2320: PR# 0: GOTO 100
190  IF H$ < > "P" AND H$ < > "S" THEN  GOTO 160
200  REM  INPUT FOR TEST STARTS
210  REM  TEST SECTION MAXM. CAVITY RADIUS IS INPUT IN THE NEXT LINE
220 MR = 0.28 / 1000000
230  INPUT "MONTH=";MZ: PRINT
240  PRINT "MONTH IS ";MZ: PRINT
250  INPUT "MONTH CORRECT?,TYPE Y/N";K1$
260  IF K1$ < > "Y" THEN  GOTO 230
270  INPUT "TEST #";TNZ: PRINT
280  PRINT "TEST # IS";TNZ: PRINT
290  INPUT "TEST # CORRECT?,TYPE Y/N";K2$
300  IF K2$ = "Y" THEN SE = TNZ: GOTO 320
310  IF K2$ < > "Y" THEN  GOTO 270
320  INPUT "ATMOSPHERIC PRESSURE(CM OF HG)";AP: PRINT
330  PRINT "ATM. PRESSURE(CM OF HG)=";AP: PRINT
340  INPUT "ATM. PRESSURE CORRECT?,TYPE Y/N";K4$
350  IF K4$ < > "Y" THEN  GOTO 320
360  GOTO 370
370  INPUT "HEATER BOTTOM LOCATION WRT INLET TO TEST SECTION(CM)";CM: PRINT
380  PRINT "HEATER BOTTOM LOCATION WRT INLET TO TEST SECTION(CM)=";CM: PRINT
390  INPUT "HEATER LOCATION CORRECT?,TYPE Y/N";K5$
400  IF K5$ < > "Y" THEN  GOTO 370
410  INPUT "LOOP FLOW METER READING(Z)";FL: PRINT
420  PRINT "LOOP FLOW METER READING(Z)=";FL: PRINT
430  INPUT "FLOW METER READING CORRECT?,TYPE Y/N";K8$
440  IF K8$ < > "Y" THEN  GOTO 410
450  INPUT "PREHEATER VOLTAGE(V)";PV: PRINT
460  PRINT "PREHEATER VOLTAGE(V)=";PV: PRINT
470  INPUT "PREHEATER VOLTAGE CORRECT?,TYPE Y/N";K9$
480  IF K9$ < > "Y" THEN  GOTO 450
490  INPUT "HEATER POWER(W)";HV: PRINT
500  PRINT "HEATER POWER(W)=";HV: PRINT
510  INPUT "HEATER POWER CORRECT?,TYPE Y/N";L1$
520  IF L1$ < > "Y" THEN  GOTO 490
530  INPUT "HEATER SETUP TIME(HH MM)";HS$: PRINT
540  INPUT "HEATER SETUP TIME CORRECT?,TYPE Y/N";L2$
550  IF L2$ < > "Y" THEN  GOTO 530
560  HOME
570 D$ =  CHR$ (4)
580  REM  FLUKE DATA ACQUISITION AND APPLE COMPUTER INTERFACE PROGRAM
590  PRINT "SET FLUKE TO SCAN CHANNELS 0 TO 11,AT 20 SECOND INTERVALS"
600  PRINT : PRINT
610  PRINT "SET EXTERNAL ENABLE ON ALL DATA": PRINT
620  PRINT : PRINT "TURN OFF ALL SCAN CONTROL BUTTONS"
630  PRINT : INPUT "PRESS RETURN TO CONTINUE";Z$
640  PRINT : PRINT "PRESS INTERVAL ON SCAN CONTROL TO START"
650  PRINT : PRINT : PRINT : PRINT "INITIALIZING DATA LOGGER"
660  PRINT : PRINT "PLEASE WAIT"
670  HOME
680  PRINT D$;"BLOAD FLUKES.BJO,D1"
690  PRINT D$;"IN#2": INPUT " ";A$: PRINT D$;"IN#0"
700 C$ = "01234567890123456789": POKE 254, PEEK (131): POKE 255, PEEK (132):KZ =  PEEK (255)
     * 256 +  PEEK (254) + 1:K1Z = KZ + 1
710  CALL 768
720 KK = 32768
730 A =  PEEK (KK): IF A = 65 THEN 710
740  IF A < > 89 THEN KK = KK + 1: GOTO 730
```

```
750 KHZ = KK / 256: POKE K1Z,KHZ: POKE KZ,KK - KHZ * 256:DD =  VAL ( MID$ (C$,2,3)):HH =  VAL
    ( MID$ (C$,6,2)):MM =  VAL ( MID$ (C$,9,2)):SS =  VAL ( MID$ (C$,12,2))
760  PRINT DD;"-";HH;" ";MM;" ";SS;" "
770 KK = KK + 22
780 KK = KK + 1: IF  PEEK (KK) <  > 65 GOTO 780
790 KHZ = KK / 256: POKE K1Z,KHZ: POKE KZ,KK - KHZ * 256:CNZ =  VAL ( MID$ (C$,2,3)):CH(CNZ) =
    VAL ( MID$ (C$,6,7)):KK = KK + 18: IF  PEEK (KK) = 65 GOTO 790
800  IF  PEEK (KK) <  > 20 THEN  PRINT "ERROR UNPACKING,WILL SCAN AGAIN": GOTO 710
810  PRINT : PRINT "COMPLETED SCAN SUCCESSFULLY!"
820  FOR I = 1 TO 10: CALL  - 198: NEXT I
830  REM  DATA TRANSFER BEGINS
840 DV = CH(0)
850 VI = CH(1)
860 TU = CH(2)
870 TE = CH(3)
880 TI = CH(4)
890 T1 = CH(5)
900 T2 = CH(6)
910 T3 = CH(7)
920 T4 = CH(8)
930 WA = CH(9)
940 TT = CH(10)
950 TM = CH(11)
960  IF WA >  = 120 THEN  PRINT "HEATER TEMPERATURE GREATER THAN OR EQUAL TO 120 DEGREES C **
    * WARNING***"
970  REM  CALCULATION BEGINS
980 P5 = (VI * 100 * 6896.5517) / 111.3 + 1315.7895 * AP
990 TR = (198 - TM) / 471.15
1000 D1 = 4.031191 + 6.721152 * (TR ^ 0.3333) + 5.08793 * (TR ^ 0.6667) - 4.993635 * TR + 4.2
    80343 * (TR ^ 1.3333)
1010 DE = 137.38 * D1
1020 P1 = P5 + (DE * 9.81 * 9.5 * .0254)
1030  IF BM = 0 THEN SH = ((6.285 + DV) * 5 * 1000 * 6896.5517) / 143.9 + DE * 18 * .0254 * 9
    .81:BP = 0:PU = SH / (18 * 2.54)
1040  IF BM = 1 THEN PD = ((6.285 + DV) * 5 * 1000 * 6896.5517) / 143.9 + DE * 18 * .0254 * 9
    .81:BP = PD - PU * (CM + .635)
1050 BU = BP / (18 * 2.54 - (CM + 0.635))
1060 P1 = PI - (PU * (CM + 0.635) + BU * 1.2217)
1070 P2 = PI - (PU * (CM + 0.635) + BU * 2.4917)
1080 P3 = PI - (PU * (CM + 0.635) + BU * 3.7617)
1090 P4 = PI - (PU * (CM + 0.635) + BU * 5.0317)
1100 TS =  - 2568.333 / ( LOG (P1) - 21.04427681) - 245.92511
1110 TA =  - 2568.333 / ( LOG (P1) - 21.04427681) - 245.92511
1120 TB =  - 2568.333 / ( LOG (P2) - 21.04427681) - 245.92511
1130 TC =  - 2568.333 / ( LOG (P3) - 21.04427681) - 245.92511
1140 TD =  - 2568.333 / ( LOG (P4) - 21.04427681) - 245.92511
1150 LX = (HV * 7 * 100 * 100) / (22 * 2.5 * 2.54 * 0.875 * 2.54)
1160  IF LX <  = 0 THEN  GOTO 1330
1170 S1 = T1 - ( INT (TA * 10) / 10)
1180 T5 = T1 - (TI + .2 * (TE - TI))
1190 B1 =  INT (LX / S1)
1200 B5 =  INT (LX / T5)
1210 S2 = T2 - ( INT (TB * 10) / 10)
1220 T6 = T2 - (T1 + .4 * (TE - TI))
1230 B2 =  INT (LX / S2)
1240 B6 =  INT (LX / T6)
1250 S3 = T3 - ( INT (TC * 10) / 10)
1260 T7 = T3 - (T1 + .6 * (TE - TI))
1270 B3 =  INT (LX / S3)
1280 B7 =  INT (LX / T7)
1290 S4 = T4 - ( INT (TD * 10) / 10)
1300 T8 = T4 - (T1 + .8 * (TE - TI))
1310 B4 =  INT (LX / S4)
1320 B8 =  INT (LX / T8)
1330 TB = ( INT (TS * 10) / 10) - TI
1340 TF = (198 - TU) / 471.5
1350 D2 = 4.031191 + 6.721152 * (TF ^ 0.3333) + 5.08793 * (TF ^ 0.6667) - 4.993635 * TF + 4.2
    80343 * (TF ^ 1.3333)
1360 DS = 137.38 * D2
1370 CF =  SQR ((7.238 * DS) / (8238 - DS))
1380 FR = (6.28 * 3.785 * FL) / (60 * CF * 1000 * 100)
1390 MF = DS * FR
1400 MX = MF * 10000 / 1.187823278
1410 MX =  INT (MX * 10) / 10
1420 MF =  INT (MF * 1000) / 1000
1430 VL =  - 3.9026924 + (1640.191052323 / (TI * 1.8 + 32 + 460))
1440 VQ =  EXP (VL) * 0.000672 * 5357
1450 RE = (MX * 3600 * 0.125 * 2.54) / (VQ * 100)
1460 RE =  INT (RE)
```

```
1470 LX =  INT (LX * 10) / 10
1480 P1 =  INT (P1)
1490 TA =  INT (TA * 10) / 10
1500 PI =  INT (PI)
1510 VD = .01742 * P1 / (TA + 273.2)
1520 LK = .08702 - .000278 * (TA - 27)
1530 ST = (.3873 - .000729 * TA) ^ 4
1540 LA = 173.19 - .4154 * (TA - 45)
1550 RS = (8 * ST * (TA + 273.16) * LX * 0.001 / (LA * VD * LK)) ^ .5
1555 C1 = LX * MR / LK
1556 C2 = (2 * ST * (TA + 273.16) * 1000000) / (LA * VD * .28 * 1000)
1560 RC = C1 + C2
1570  HOME
1580  PRINT  TAB( 05)"RESULTS OF SCAN": PRINT : PRINT
1590  VTAB 3: PRINT  TAB( 05)"SYSTEM PRESSURE(Pa)=";PI: PRINT
1600  PRINT  TAB( 05)"BULK SUBCOOLING AT INLET TO TEST SEC(C)=";TB: PRINT
1610  PRINT  TAB( 05)"TEST SECTION TI(C)=";TI;" AND TE (C)=";TE: PRINT
1620  PRINT  TAB( 05)"MASS FLUX (KG/M2.S)=";MX: PRINT
1630  PRINT  TAB( 05)"MASS FLOW RATE(KG/S)=";MF: PRINT
1640  PRINT  TAB( 05)"VOL.FLOW RATE(M3/S)=";FR: PRINT
1650  PRINT  TAB( 05)"REYNOLD'S # AT TEST SECTION=";RE: PRINT
1660  PRINT  TAB( 05)"PRESSURE AT FIRST THERMOCOUPLE LOCATION(Pa)=";P1: PRINT
1670  INPUT "PRESS RETURN FOR FURTHER DISPLAY";AK$: HOME
1680  PRINT  TAB( 03);"THERMOCOUPLE TEMP(C)"; TAB( 24);"WALL S'HEAT(C)": PRINT : PRINT
1690  PRINT  TAB( 07);"T1=";T1; TAB( 28);"S1=";S1: PRINT
1700  PRINT  TAB( 07);"T2=";T2; TAB( 28);"S2=";S2: PRINT
1710  PRINT  TAB( 07);"T3=";T3; TAB( 28);"S3=";S3: PRINT
1720  PRINT  TAB( 07);"T4=";T4; TAB( 28);"S4=";S4: PRINT : PRINT
1730  PRINT  TAB( 08);"WALL TEMP. DIFF.(C)": PRINT : PRINT
1740  PRINT  TAB( 07);"DT1=";T5; TAB( 28);"RS1(C)= ";RS
1750  PRINT  TAB( 20);"DT2="; TAB( 28);T6: PRINT
1760  PRINT  TAB( 20);"DT3="; TAB( 28);T7: PRINT
1770  PRINT  TAB( 20);"DT4="; TAB( 28);T8: PRINT
1780  INPUT "PRESS RETURN FOR FURTHER DISPLAY";AK$: HOME
1790  PRINT  TAB( 05)"HEAT FLUX (W/M2)=";LX: PRINT
1800  PRINT  TAB( 04);"BOILING HT.TR COEF"; TAB( 25);"CONV HT.TR COEF": PRINT : PRINT
1810  PRINT  TAB( 08);"AT TC#1(W/M2.C)=";B1; TAB( 32);B5: PRINT
1820  PRINT  TAB( 08);"AT TC#2(W/M2.C)=";B3; TAB( 32);B6: PRINT
1830  PRINT  TAB( 08);"AT TC#3(W/M2.C)=";B3; TAB( 32);B7: PRINT
1840  PRINT  TAB( 08);"AT TC#4(W/M2.C)=";B4; TAB( 32);B8: PRINT
1850  INPUT "PRESS RETURN FOR FURTHER DISPLAY";AK$: HOME
1860  PRINT  TAB( 05)"COOLER OUTLET TEMP(C)=";TU: PRINT
1870  PRINT  TAB( 05)"CW BATH TEMP(C)=";TT: PRINT
1880  PRINT  TAB( 05)"HEATER TEMP(C)=";WA: PRINT
1890  PRINT  TAB( 05)"ROOM TEMP(C)=";TM: PRINT
1900 RS =  INT (RS * 10) / 10
1910 RC =  INT (RC * 10) / 10
1920  PRINT  TAB( 5)"SUPHT REQD @ TC1(C)=";"TD";RS;"  ";"CAV";RC: PRINT
1930 PU =  INT (PU * 100) / 100
1940  PRINT  TAB( 5)"SINGLE PHASE DP/DZ (Pa/M) = ";PU * 100: PRINT
1950 BU =  INT (BU * 100) / 100
1960  PRINT  TAB( 5)"TWO PHASE DP/DZ (Pa/M) = ";BU * 100: PRINT
1970  PRINT "VD= "; INT (VD * 100) / 100;" ST= "; INT (ST * 10000) / 10000;" DE= "; INT (DE);
     " PRESS.DROP="; INT (PD);"  "; INT (SH)
1980  INPUT "PRESS RETURN TO CONTINUE";RC$
1990  HOME
2000  INPUT "TYPE Y/N FOR STORING DATA,RESULTS ON DISK";ST$: PRINT
2010  IF ST$ = "Y" THEN  INPUT "COMMENTS ABOUT THE RUN?";B$: GOSUB 2140: GOTO 2030
2020  IF ST$ <  > "Y" AND ST$ <  > "N" THEN  GOTO 2000
2030  HOME : INPUT "TYPE Y/N FOR CONTINUING SCAN UNDER SAME EXPT CONDITIONS";CS$: PRINT
2040  IF CS$ = "Y" THEN SE = SE + 0.01:BM = 1:PU = PU: GOTO 650
2050  IF CS$ <  > "Y" AND CS$ <  > "N" THEN  GOTO 2030
2060  INPUT "TYPE Y/N FOR SCAN UNDER NEW EXPT. CONDITIONS";NS$: PRINT
2070  IF NS$ = "N" THEN  GOTO 2910
2080  IF NS$ <  > "Y" AND NS$ <  > "N" THEN  GOTO 2060
2090  INPUT "TO CHANGE ONLY THE HEATER POWER TYPE H. THE FLOW RATE TYPE F ";HR$
2100  IF HR$ = "H" THEN TNZ = TNZ + 1:SE = TNZ: HOME : GOTO 490
2110  IF HR$ = "F" THEN TNZ = TNZ + 1:SE = TNZ: HOME : PRINT "THE FIRST RUN MUST BE SINGLE PH
     ASE FLOW":BM = 0: PRINT : INPUT "FLOW METER %= ";FL: PRINT : INPUT "RESET TIME = ";HS$: GOTO
     560
2120 TNZ = TNZ + 1:SE = TNZ: HOME : PRINT "IF THIS IS THE FIRST RUN IN THIS SEQUENCE,THEN IT
     SHOULD BE SINGLE PHASE FLOW":BM = 0: GOTO 360
2130  IF HR$ <  > "Y" AND HR$ <  > "N" THEN  GOTO 2090
2140  REM  SUBROUTINE FOR STORING DATA,RESULTS
2150 D$ =  CHR$ (4)
2160  PRINT D$;"CATALOG,D2"
2170  PRINT "FILENAME IS B.";DD;".";SE
2180 R1$ = "B"
2190 R2$ = "."
```

```
2200 R4$ =  STR$ (DD)
2210 R5$ =  STR$ (SE)
2220 BF$ = R1$ + R2$ + R4$ + R2$ + R5$
2230  PRINT D$;"OPEN";BF$
2240  PRINT D$;"WRITE";BF$
2250  PRINT MZ: PRINT DD: PRINT HH: PRINT MM: PRINT B$: PRINT AP: PRINT TM: PRINT PV: PRINT H
     V: PRINT CM: PRINT FL: PRINT TA: PRINT TI: PRINT TE: PRINT TU: PRINT TT: PRINT HS$
2260  PRINT PI: PRINT P1: PRINT FR: PRINT MF: PRINT S1: PRINT S2: PRINT S3: PRINT S4: PRINT T
     B: PRINT MX: PRINT RE: PRINT LX: PRINT B1: PRINT B2: PRINT B3: PRINT B4
2270  PRINT T1: PRINT T2: PRINT T3: PRINT T4: PRINT T5: PRINT T6: PRINT T7: PRINT T8: PRINT B
     5: PRINT B6: PRINT B7: PRINT B8: PRINT RS: PRINT PU * 100: PRINT BU * 100
2280  PRINT RC
2290  PRINT D$;"CLOSE"
2300  RETURN
2310  REM  SUBROUTINE FOR PRINTING FILE
2320 D$ =  CHR$ (4)
2330  PRINT D$;"CATALOG,D2"
2340  PRINT : INPUT "FILENAME:";VV$
2350  HOME
2360  PRINT D$;"OPEN";VV$
2370  PRINT D$;"READ";VV$
2380  INPUT MZ: INPUT DD: INPUT HH: INPUT MM: INPUT B$: INPUT AP: INPUT TM: INPUT PV: INPUT H
     V: INPUT CM: INPUT FL: INPUT TA: INPUT TI: INPUT TE: INPUT TU: INPUT TT: INPUT HS$
2390  INPUT PI: INPUT P1: INPUT FR: INPUT MF: INPUT S1: INPUT S2: INPUT S3: INPUT S4: INPUT T
     B: INPUT MX: INPUT RE: INPUT LX: INPUT B1: INPUT B2: INPUT B3: INPUT B4
2400  INPUT T1: INPUT T2: INPUT T3: INPUT T4: INPUT T5: INPUT T6: INPUT T7: INPUT T8: INPUT B
     5: INPUT B6: INPUT B7: INPUT B8: INPUT RS: INPUT PU: INPUT BU
2410  INPUT RC
2420  PRINT D$;"CLOSE"
2430  IF H$ = "P" THEN  PRINT "POSITION PAPER IN PRINTER": INPUT "PRESS RETURN TO CONTINUE";C
     R$: PR# 1
2440  PRINT "***FILENAME IS=  ";VV$
2450  PRINT "SCAN MONTH= ";MZ;"  ";
2460  PRINT "SCAN DATE=";DD;"   SCAN TIME=";HH;":";MM
2470  PRINT "COMMENTS? ";B$
2480  PRINT "ATM.PRESSURE(CM OF HG)=";AP;"  ";
2490  PRINT "ROOM TEMP(C)=";TM
2500  IF H$ = "S" THEN  INPUT "PRESS RETURN FOR FURTHER DISPLAY";AK$: HOME
2510  PRINT "SYSTEM PRESSURE(Pa)=";PI;"  ";
2520  PRINT "INLET SUBCOOLING(C)=";TB
2530  PRINT "FLUID TEMP AT TEST SEC INLET(C)=";TI;"  ";
2540  PRINT "AND AT OUTLET(C)=";TE
2550  PRINT "MASS FLUX IN TEST SEC(KG/M2.S)=";MX;"  ";
2560  PRINT "REYNOLD'S # AT TEST SEC=";RE
2570  PRINT "MASS FLOW RATE(KG/S)=";MF;"  ";
2580  PRINT "VOL.FLOW RATE(M3/S)=";FR
2590  PRINT "PREHEATER VOLTAGE(V)=";PV;"  ";
2600  PRINT "HEATER POWER(W)=";HV;"  SET TIME=  ";HS$
2610  PRINT "HEATER BOTTOM LOCATION WRT TEST SEC INLET(CM)=";CM
2620  PRINT "PRESSURE AT FIRST THERMOCOUPLE ELEVATION(Pa)=";P1
2630  PRINT "SATN. TEMP AT FIRST THERMOCOUPLE ELEVATION(C)=";TA
2640  IF H$ = "S" THEN  INPUT "PRESS RETURN FOR FURTHER DISPLAY";AK$: HOME
2650  PRINT "TC#1 TEMP(C)=";T1;"  ";
2660  PRINT "WALL1 SUPHT TEMP(C)=";S1
2670  PRINT "REQD SUPHT(C)=";"THERMODYNAMIC  ";RS;"CAVITY RAD ";RC
2680  PRINT "TEMP DIFF#1(C) = "T5
2690  PRINT "TC#2 TEMP(C)=";T2;"  ";
2700  PRINT "WALL2 SUPHT TEMP(C)=";S2
2710  PRINT "TEMP DIFF#2 = ";T6
2720  PRINT "TC#3 TEMP(C)=";T3;"  ";
2730  PRINT "WALL3 SUPHT TEMP(C)=";S3
2740  PRINT "TEMP DIFF#3 = ";T7
2750  PRINT "TC#4 TEMP(C)=";T4;"  ";
2760  PRINT "WALL4 SUPHT TEMP(C)=";S4
2770  PRINT "TEMP DIFF#4 = ";T8
2780  IF H$ = "S" THEN  INPUT "PRESS RETURN FOR FURTHER DISPLAY";AK$: HOME
2790  PRINT "HEAT FLUX (W/M2)= ";LX;" (DP/DZ)1P = ";PU;" (DP/DZ)2P = ";BU
2800  PRINT "BOILING HEAT TRANSFER COEFF AT TC#1 LOC.(W/M2.C)=";B1
2810  PRINT "CONV.HEAT TRANSFER COEFF AT TC#1 LOC.(W/M2.C)=";B5
2820  PRINT "BOILING HEAT TRANSFER COEFF AT TC#2 LOC.(W/M2.C)=";B2
2830  PRINT "CONV.HEAT TRANSFER COEFF AT TC#2 LOC.(W/M2.C)=";B6
2840  PRINT "BOILING HEAT TRANSFER COEFF AT TC#3 LOC.(W/M2.C)=";B3
2850  PRINT "CONV.HEAT TRANSFER COEFF AT TC#3 LOC.(W/M2.C)=";B7
2860  PRINT "BOILING HEAT TRANSFER COEFF AT TC#4 LOC.(W/M2.C)=";B4
2870  PRINT "CONV.HEAT TRANSFER COEFF AT TC#4 LOC.(W/M2.C)=";B8
2880  PRINT "FLUID TEMP AT COOLER OUTLET(C)=";TU;"  ";
2890  PRINT "CW BATH TEMP(C)=";TT: PRINT
2900  RETURN
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2910  INPUT "TYPE Y/N TO CONTINUE";CN$
2920  IF CN$ = "Y" THEN  GOTO 100
2930  IF CN$ <  > "Y" AND CN$ <  > "N" THEN  GOTO 2910
2940  END
```

APPENDIX - H.4 (SAMPLE OUTPUT)

```
 ERR#0***FILENAME IS=  B.9.35.01
 SCAN MONTH= 1  SCAN DATE=9   SCAN TIME=0:26
 COMMENTS? SITES JUST QUENCHED.
 ATM.PRESSURE(CM OF HG)=76  ROOM TEMP(C)=25.8
 SYSTEM PRESSURE(Pa)=306234  INLET SUBCOOLING(C)=22.4
 FLUID TEMP AT TEST SEC INLET(C)=36.9  AND AT OUTLET(C)=38
 MASS FLUX IN TEST SEC(KG/M2.S)=388.6  REYNOLD'S # AT TEST SEC=3240
 MASS FLOW RATE(KG/S)=.046  VOL.FLOW RATE(M3/S)=3.18970317E-05
 PREHEATER VOLTAGE(V)=0  HEATER POWER(W)=70  SET TIME=  00 23
 HEATER BOTTOM LOCATION WRT TEST SEC INLET(CM)=29
 PRESSURE AT FIRST THERMOCOUPLE ELEVATION(Pa)=300587
 SATN. TEMP AT FIRST THERMOCOUPLE ELEVATION(C)=58.7
 TC#1 TEMP(C)=66.5  WALL1 SUPHT TEMP(C)=7.80000001
 REQD SUPHT(C)=THERMODYNAMIC  1.6CAVITY RAD 44.2
 TEMP DIFF#1(C) = 29.38
 TC#2 TEMP(C)=67.5  WALL2 SUPHT TEMP(C)=8.90000001
 TEMP DIFF#2 = 30.16
 TC#3 TEMP(C)=66  WALL3 SUPHT TEMP(C)=7.40000001
 TEMP DIFF#3 = 28.44
 TC#4 TEMP(C)=65.8  WALL4 SUPHT TEMP(C)=7.20000002
 TEMP DIFF#4 = 28.02
 HEAT FLUX (W/M2)= 15781.8 (DP/DZ)1P = 18125 (DP/DZ)2P = 22611
 BOILING HEAT TRANSFER COEFF AT TC#1 LOC.(W/M2.C)=2023
 CONV.HEAT TRANSFER COEFF AT TC#1 LOC.(W/M2.C)=537
 BOILING HEAT TRANSFER COEFF AT TC#2 LOC.(W/M2.C)=1773
 CONV.HEAT TRANSFER COEFF AT TC#2 LOC.(W/M2.C)=523
 BOILING HEAT TRANSFER COEFF AT TC#3 LOC.(W/M2.C)=2132
 CONV.HEAT TRANSFER COEFF AT TC#3 LOC.(W/M2.C)=554
 BOILING HEAT TRANSFER COEFF AT TC#4 LOC.(W/M2.C)=2191
 CONV.HEAT TRANSFER COEFF AT TC#4 LOC.(W/M2.C)=563
 FLUID TEMP AT COOLER OUTLET(C)=36.8  CW BATH TEMP(C)=34.1

 ***FILENAME IS=  B.8.26.04
 SCAN MONTH= 1  SCAN DATE=8   SCAN TIME=22:54
 COMMENTS? ALL SITES QUENCHED.
 ATM.PRESSURE(CM OF HG)=76  ROOM TEMP(C)=25.9
 SYSTEM PRESSURE(Pa)=368073  INLET SUBCOOLING(C)=28.5
 FLUID TEMP AT TEST SEC INLET(C)=37.7  AND AT OUTLET(C)=39.1
 MASS FLUX IN TEST SEC(KG/M2.S)=388.1  REYNOLD'S # AT TEST SEC=3260
 MASS FLOW RATE(KG/S)=.046  VOL.FLOW RATE(M3/S)=3.19530821E-05
 PREHEATER VOLTAGE(V)=0  HEATER POWER(W)=93  SET TIME=  22 39
 HEATER BOTTOM LOCATION WRT TEST SEC INLET(CM)=29.9
 PRESSURE AT FIRST THERMOCOUPLE ELEVATION(Pa)=360970
 SATN. TEMP AT FIRST THERMOCOUPLE ELEVATION(C)=65.4
 TC#1 TEMP(C)=74.8  WALL1 SUPHT TEMP(C)=9.40000002
 REQD SUPHT(C)=THERMODYNAMIC  1.7CAVITY RAD 36.8
 TEMP DIFF#1(C) = 36.82
 TC#2 TEMP(C)=74.3  WALL2 SUPHT TEMP(C)=8.90000002
 TEMP DIFF#2 = 36.04
 TC#3 TEMP(C)=75.8  WALL3 SUPHT TEMP(C)=10.4
 TEMP DIFF#3 = 37.26
 TC#4 TEMP(C)=70.8  WALL4 SUPHT TEMP(C)=5.40000002
 TEMP DIFF#4 = 31.98
 HEAT FLUX (W/M2)= 20967.3 (DP/DZ)1P = 22855 (DP/DZ)2P = 10202
 BOILING HEAT TRANSFER COEFF AT TC#1 LOC.(W/M2.C)=2230
 CONV.HEAT TRANSFER COEFF AT TC#1 LOC.(W/M2.C)=569
 BOILING HEAT TRANSFER COEFF AT TC#2 LOC.(W/M2.C)=2355
 CONV.HEAT TRANSFER COEFF AT TC#2 LOC.(W/M2.C)=581
 BOILING HEAT TRANSFER COEFF AT TC#3 LOC.(W/M2.C)=2016
 CONV.HEAT TRANSFER COEFF AT TC#3 LOC.(W/M2.C)=562
 BOILING HEAT TRANSFER COEFF AT TC#4 LOC.(W/M2.C)=3882
 CONV.HEAT TRANSFER COEFF AT TC#4 LOC.(W/M2.C)=655
 FLUID TEMP AT COOLER OUTLET(C)=38.5  CW BATH TEMP(C)=28
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APPENDIX - I

List Of Equipment and Instrumentation

APPENDIX-I

LIST OF EQUIPMENT AND INSTRUMENTATION

1.Cartridge Heater, Model # H1-3, Fast heat, "Hi-Temp", 7.874 mm diameter, 63.5 mm heated length, 76.2 mm sheath length, 141.05 KW/m2 heat flux, 225 W, at 120 V. Acrolab Instruments, Windsor, Ontario.

2.Computer, APPLE II Plus with RS 232 interface card, Program language APPLE BASIC, Apple Computer, Inc., 10260 Bandley Drive, Cupertino, California.

3.Constant Current Supply, Model # PS-KIT-1, input 12V, output 1.5 mA, Omega Engineering Company, Stamford, Connecticut.

4.Constant Temperature Bath, Model # K-40, 115 V, 800 W, Messerate-Werk Lauda, W. Germany.

5.Data Logger, Fluke Data Logger, Model # 2240 C, John Fluke Manufacturing Co., Everett, Washington.

6.Dial Pressure Gage, Model # R07-200-RNKA, 0-11 bar(gage). C.A Norgren Co Inc., Littleton, Colarado.

7.Differential Pressure Transducer, Model # PX83-005 DV, 0.345 bars differential, span 143.9 mV at 1.5 mA, Omega Engineering Company, Stamford, Connecticut.

8.Glass Pipe, Borosilicate glass pipe, PS1/ 18, I.D. 25.4 mm, maximum working temperature 300 degrees C, maximum working pressure 7.91 bars, Pegasus Industrial Specialities Ltd, Agincourt, Ontario.

9.Filter Drier, Model # ALCO ASD 45S6-VV, Mueller Brass Co., Port

Huron, Michigan.

10.Flowmeter, Model # FL-1504 A, with flow range 0.628 US gpm-6.28 US gpm for water and with graduations at an increment of 0.1256 gpm. Omega Engineering Company, Stamford, Connecticut.

11.Preheater, Model # 220 1968, 282.6 mm long, oil immersion heater, 300 W, 120 V, Temro Products Inc., Winnipeg, Manitoba.

12. Pump, Model # J-7004-54, maximum flow rate 7.5 US gpm, maximum head 5.18 m of water, Net Positive Suction Head (required) 0.1524 m, Cole Parmer Instrument Company, Chicago, Illinois.

13.Pressure Transducer, Model # PX 82-100 GV, maximum pressure 7.9 bars, 111.3 mV span at 1.5 mA, Omega Engineering Company, Stamford, Connecticut.

14.Printer, Model # 26-1168, TRS-80 Line printer VIII, Tandy Electronics Ltd., Barrie, Ontario.

15.System Pressurizer, Model # AB01B3T1A1K, 0.000946 m3 size, 206.9 bars, Parker Fluid Power, Des Plaines, Illinois.

16.Thermocouple Plugs, Model # OPNE-14, with 2 self sealing neoprene rings, Omega Engineering Company, Stamford, Connecticut.

17.Vacuum Pump, Model # 1399, The Welch Scientific Company, Skokie, Illinois.

18.Valve, 19mm, Model # 901 A, ball valve, Jenkins Company, USA.

19.Variac, Model # W5MT3, 0-140 V, General Radio Company, Concord, Massachussets.

20.Wattmeter, Model # 880, 0-250 W, Simpson Electric Company, Chicago, Illinois.

VITA AUCTORIS

1958 Born in Tuticorin, India on November 25.

1975 Completed the Secondary School Certificate from Maharashtra Board, through S.I.W.S High School, Bombay, India.

1977 Completed the Intermediate Science examination from Bombay University, through S.I.E.S College of Arts, Science and Commerce, Bombay, India.

1981 Completed the Bachelor of Engineering from Bombay University, through Victoria Jubilee Technical Institute, Bombay, India.

1981 Joined National Organic Chemical Industries Limited, Bombay, India as Trainee Engineer, and held positions as Assistant Maintenance Engineer, Project Engineer.

1989 Currently a candidate for the degree of Master of Applied Science at University Of Windsor.