OCCURRENCE OF CRITICAL HEAT FLUX

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by

Joseph Chung-Ming Leung

April 1976

NOTICE: This information document contains preliminary information prepared primarily for interim use in light water reactor programs in the U. S. Since it does not constitute a final report, it should be cited as a reference only in special circumstances, such as requirements for regulatory needs.

REACTOR ANALYSIS AND SAFETY DIVISION
ARGONNE NATIONAL LABORATORY
9700 South Cass Avenue
Argonne, Illinois 60439

MASTER

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DURING BLOWDOWN WITH FLOW REVERSAL

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OCCURRENCE OF CRITICAL HEAT FLUX
DURING BLOWDOWN WITH FLOW REVERSAL

A Thesis

Submitted to

The Graduate School
Northwestern University

In Partial Fulfillment
of the Requirements for the Degree
Master of Science in
Chemical Engineering

by

Joseph Chung-Ming Leung

April 1976

TABLE OF CONTENTS

	Page
Nomenclature.....	xi
Abstract.....	xvii
I. INTRODUCTION.....	1
1.1 Background and Definitions.....	2
1.1.1 Two-Phase Flow Regimes of Heat Transfer....	2
1.1.2 Boiling Crisis.....	4
1.2 Critical Heat Flux Models.....	5
1.2.1 Subcooled Critical Heat Flux.....	5
1.2.2 'Dryout' CHF in High Quality Flow Regimes..	7
1.3 LOCA and Associated Phenomena in a PWR.....	8
II. CRITICAL HEAT FLUX UNDER TRANSIENT CONDITIONS.....	12
2.1 Pool Boiling CHF under Transient Conditions.....	12
2.2 Flow Boiling CHF under Transient Conditions.....	13
III. EXPERIMENTAL INVESTIGATION.....	40
3.1 Outline of Experimental Technique.....	40
3.1.1 Choice of Working Fluid.....	40
3.1.2 Measurement of Flow During Transient.....	40
3.1.3 Measurement of Void During Transient.....	41
3.2 Description of Experimental Apparatus.....	42
3.2.1 Freon-11 Test Facility.....	42
3.2.2 Test Section.....	45
3.2.3 Instrumentation.....	51
3.3 Experimental Procedure.....	52
3.3.1 Calibration.....	52
3.3.2 Refilling Operation.....	52
3.3.3 Heatup Operation.....	54
3.3.4 Blowdown Operation.....	54
IV. EXPERIMENTAL RESULTS.....	56
4.1 Adiabatic Blowdown Tests.....	56
4.2 Preliminary Diabatic Blowdown Tests.....	56
4.3 Diabatic Blowdown with High Speed Photography.....	78

	Page
V. DISCUSSION.....	104
5.1 Adiabatic Blowdown.....	104
5.2 Effect of Dissolved Gas.....	107
5.3 Diabatic Blowdown.....	108
5.3.1 Preliminary Diabatic Tests.....	108
5.3.2 Effect of Bubble Growth in the Decompression Process.....	112
5.3.3 Diabatic Tests with High Speed Photography....	113
VI. SUMMARY AND CONCLUSIONS.....	121
VII. RECOMMENDATIONS FOR FUTURE STUDIES.....	123
APPENDICES	
A. Single-Phase Friction Pressure Drop Data.....	124
B. Single-Phase Liquid Heat Transfer Data.....	134
C. Evaluation of Outside Surface Temperature.....	139
C.1 Steady-State Consideration.....	139
C.2 Unsteady-State Consideration.....	140
D. Thermocouple Response Determination.....	142
D.1 Out-of-Loop Determination.....	142
D1.1 Theory.....	142
D1.2 Experiment.....	142
D.2 In-Loop Determination.....	143
E. Capacitance Void Measurement Calibration.....	150
E.1 Void Measurement Instrumentation.....	150
E.2 Interpretation of Void Measurement.....	150
E.3 Temperature Sensitivity in Void Measurement.....	163
F. Void Estimation During Adiabatic Blowdown.....	164
G. Void Estimation from Turbine and Weight Measurements.	167
H. Measurement of the Power Factor.....	168
I. Instrument Calibration.....	175
I.1 Weight Transducers.....	175
I.2 Turbine Flowmeters.....	175
I.3 Pressure Transducers.....	179
J. Freon-11 to Water Sealing.....	181
J.1 Zuber and Ishii.....	181
J.2 Tong et al.....	182
J.3 Ahmad.....	188

	Page
K. Critical Flowrate Calculation.....	190
L. Critical Heat Flux Calculation.....	194
ACKNOWLEDGMENTS.....	197
REFERENCES.....	198
VITA.....	209

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
1.1	Two-Phase Flow Regime (after Collier ⁸).....	3
1.2	Schematic of a PWR System (after Catton et al. ²).....	10
2.1	Dimensionless Density During Various Transients (after Premoli ⁵²).....	18
2.2	Effect of Power Excursion Time on CHF (after Hein et al. ⁵⁹).....	21
2.3	Time to CHF versus Excess Heat Flux (after Aoki et al. ⁶⁵).....	25
2.4	Measured and Predicted Wall Temperature for Semiscale Test 845 (after Morgan et al. ⁷⁴).....	29
2.5	Wall Temperature for Babcock and Wilcox Blowdown Test (after Morgan et al. ⁷⁴).....	31
2.6	Wall Temperature and Pressure for Holifield Blowdown Test (after Lawson ⁷⁶).....	32
2.7	Pressure History for Holifield Single-Rod Test (after ORNL Report ⁸⁰).....	34
2.8	MIT Double Flow Reversal Results (after Smith ⁸⁷).....	37
3.1	Freon-11 Blowdown Facility.....	43
3.2	Schematic Showing the Refrigeration Unit and the Receiving Vessel.....	46
3.3	Transient CHF Test Section Isometric.....	47
3.4	Thermocouple Locations.....	50
4.1a	Pressure and Void Measurement for Adiabatic Blowdown Run 4/4A.....	57
4.1b	Fluid Temperature During Blowdown Run 4/4-A.....	58
4.2a	Pressure and Weight Measurement for Adiabatic Blowdown Run 4/9B.....	59
4.2b	Void Fraction and Turbine Flowmeter Measurement for Adiabatic Blowdown, Run 4/9B.....	60
4.3a	Pressure History for Diabatic Blowdown, Run 4/22.....	63

No.	Title	Page
4.3b	Temperature History for Run 4/22.....	64
4.3c	Turbine Flow Measurement for Run 4/22.....	65
4.3d	Void Fraction Measurement for Run 4/22.....	66
4.4a	Pressure History for Diabatic Blowdown, Run 5/19A....	67
4.4b	Temperature History for Run 5/19A.....	68
4.4c	Void Fraction Measurement for Run 5/19A.....	69
4.4d	Time to CHF versus Axial Location for Run 5/19A.....	70
4.5a	Pressure History for Diabatic Blowdown, Run 5/29.....	71
4.5b	Temperature History for Run 5/29.....	72
4.5c	Void Fraction Measurement for Run 5/29.....	73
4.5d	Flow Measurement during Blowdown for Run 5/29.....	74
4.5e	Mass in Blowdown Vessel for Run 5/29.....	75
4.5f	Time to CHF versus Axial Location for Run 5/29.....	76
4.5g	Onset of Rewetting During Run 5/29.....	77
4.6a	Pressure History for Diabatic Blowdown, Run 6/4.....	79
4.6b	Temperature History for Run 6/4.....	80
4.6c	Void Fraction Measurement for Run 6/4.....	81
4.6d	Turbine Flow Measurement for Run 6/4.....	82
4.6e	Mass in Blowdown Vessel for Run 6/4.....	83
4.6f	Pressure Differential Measurement for Run 6/4.....	84
4.7a	Pressure History for Diabatic Blowdown, Run 6/9.....	85
4.7b	Temperature History for Run 6/9.....	86
4.7c	Void Fraction Measurement for Run 6/9.....	87
4.7d	Turbine Flow Measurement for Run 6/9.....	88
4.7e	Mass in Blowdown Vessel for Run 6/9.....	89
4.7f	Pressure Differential Measurement for Run 6/9.....	90

<u>No.</u>	<u>Title</u>	<u>Page</u>
4.7g	Axial Pressure Profile During Diabatic Blowdown, Run 6/9.....	91
4.8a	Pressure History for Run 6/12.....	92
4.8b	Temperature History for Run 6/12.....	93
4.8c	Void Fraction Measurement for Run 6/12.....	94
4.8d	Turbine Flow Measurement for Run 6/12.....	95
4.8e	Pressure Differential Measurement for Run 6/12.....	96
4.9a	Selected Prints for Run 6/4.....	98
4.9b	Selected Prints for Run 6/9.....	99
4.9c	Selected Prints for Run 6/12.....	100
4.10a	Velocity Observed in High Speed Photography, Run 6/4.	101
4.10b	Velocity Observed in High Speed Photography, Run 6/9.	102
4.10c	Velocity Observed in High Speed Photography, Run 6/12	103
5.1a	Pressure History for Adiabatic and Diabatic Blowdowns.....	114
5.1b	Void Fraction for Adiabatic and Diabatic Blowdowns...	115
A.1	Friction Factor Results Based on Hydraulic Equivalent Reynolds Number.....	127
A.2	Friction Factor Plot Based on Modified Reynolds Number.....	129
A.3	Isothermal Freon-11 Friction Data.....	130
B.1	Single Phase Heat Balance Results.....	135
B.2	Single Phase Heat Transfer Results.....	137
D.1	Temperature Response for Lower Half TS Thermocouples.	145
D.2	Temperature Response for Upper Half TS Thermocouples.	146
D.3	Temporal Voltage, Current, and Resistance Measurement During Rapid Joule Heating.....	148
E.1	Vertical Calibration Using Impedance Bridge.....	151

<u>No.</u>	<u>Title</u>	<u>Page</u>
E.2	Horizontal Calibration Using Impedance Bridge.....	152
E.3	Capacitance Bridge for Transient Void Measurement.....	153
E.4	Vertical Calibration Using Capacitance Bridge.....	154
E.5	Horizontal Calibration Using Capacitance Bridge.....	155
E.6	Output Voltage Characteristics of Capacitance Bridge..	156
E.7	Void Fraction Determination Using Capacitance Bridge..	159
E.8	Various Idealized Two Phase Flow Models.....	159
E.9	Theoretical Void versus Apparent Void.....	161
F.1	Generalized Temperature-Entropy Diagram.....	165
H.1	Determination of Phase Angle Using Lissajous Figure...	169
H.2	Measured Phase Angle versus TS Voltage.....	170
H.3	Amplitude of Ripple versus TS Voltage.....	172
H.4	Power Correction Factor versus TS Voltage.....	174
I.1	Supply Vessel Weight Transducer Calibration.....	176
I.2	Blowdown Vessel Weight Transducer Calibration.....	177
I.3	Agreement Between Two Types of Flow Measurements.....	178
I.4	Calibration of a 100 PSID Transducer.....	180
J.1	Density Ratio for Freon-11 and Water.....	183
J.2	Density Ratio versus Reduced Pressure.....	184

LIST OF TABLES

<u>No.</u>	<u>Title</u>	<u>Page</u>
2.1	Experimental Conditions for Transient Critical Heat Flux Studies.....	14
3.1	Dimensions of Test Section.....	48
3.2	Transient CHF Test Instrumentation.....	53
4.1	Initial Steady-State Conditions in Test Section Prior to Blowdown.....	61
4.2	Time to CHF (t_{CHF}) for Diabatic Blowdown.....	97
5.1	Effect of Prolonged Nitrogen Contact with Freon-11 (130°F) at Elevated Pressure (60 psig).....	109
A.1	Frictional Pressure Drop for Water in Test Section No. 1.....	131
A.2	Frictional Pressure Drop for Freon-11 in Test Section No. 2.....	133
D.1	Time Constant of Thermocouples.....	144
D.2	Overall Time Response of Test Section and Thermocouples	144
J.1	Thermodynamic and Transport Properties.....	185
J.2	Freon-11--Water Scaling Results.....	186
L.1	Summary of Critical Heat Flux Estimations.....	196

NOMENCLATURE

English Symbols

A	area
A_x	cross sectional flow area
C	capacitance
C_p	specific heat capacity
d	diameter
D	diameter
f	friction factor
Fr	Froude number
g	gravitational acceleration
G	mass flow rate
h	enthalpy
h_o	heat transfer coefficient
i	current
k	thermal conductivity
L	length
M	mass
N	dimensionless number used with subscripts
Nu	Nusselt number
p	instantaneous power
P	average power
P	perimeter
P	pressure
ΔP	pressure differential
Pr	Prandtl number
q	total heat rate

q'''	volumetric heat generation rate
Q	volumetric flow rate
r_1	inside radius of heated tube
r_2	outside radius of heated tube
R	electrical resistance
$\bar{R}$	bubble radius
Re	Reynolds number
Re^*	modified Reynolds number (Jones, ¹²⁷ Eq. A.9)
S	entropy
S	slip ratio
t	time, as defined locally
T	temperature
ΔT	temperature differential
U	velocity
v	specific volume
V	voltage
W	mass flow rate
W^*	reduced power number in Ahmad's model ¹⁴⁶
x	vapor quality
z	axial location in heated channel

Greek Symbols

α	void fraction
α_1	thermal diffusivity
Γ	rate of vapor generation per unit volume
ϵ	dielectric constant
η	critical pressure ratio
θ	absolute temperature

θ	phase angle
r	radius or diameter ratio
μ	viscosity
ν	kinematic viscosity
δ	tube thickness
ρ	density
ρ_r	electrical resistivity
σ	surface tension
τ	response time as defined locally
ϕ	surface heat flux
Φ	laminar coefficient defined in Eq. A.2
ψ	power factor
ω	angular rotation

Subscripts

a	apparent
ave	average value
b	bulk value
bd	bead of thermocouple
BD	blowdown
c	continuous phase
c	critical condition
CHF	critical heat flux condition
d	dispersed phase
e	equivalent
E	empty
f	liquid phase
F	full

fric	friction
fg	difference between vapor and liquid quantities
g	gaseous phase
h	heated
i	inlet, inside, interface
jn	junction
max	maximum
meas	measured value
M	momentum
O	outlet, outside, single phase
o	initial conditions
r	reduced property
s	steady state
sat	saturated value
SAT	saturation
sub	subcooled, subcooling
t	transient
T	total
TC	thermocouple
TS	test section
v	vaporization
w	wall
w	wetted
z	hydrostatic head
2ϕ	two phase

Abbreviations

ac	alternate current, also AC
----	----------------------------

CHF	critical heat flux
dc	direct current, also DC
DNB	departure from nucleate boiling
DP	differential pressure
DVM	digital voltmeter
fn	function of
Hg	mercury
ID	inside diameter
msec	millisecond
OD	outside diameter
s.g.	specific gravity
TC	thermocouple
TS	test section

Abbreviations Used in References

AEC	U. S. Atomic Energy Commission
AECL	Atomic Energy of Canada Limited
AEW	Atomic Energy Establishment, Winfrith, U.K.
AERE	Atomic Energy Research Establishment, U.K.
ANC	Aerojet Nuclear Company
ANL	Argonne National Laboratory
BAW	Babcock and Wilcox Company
CISE	Centro Informazioni Studi Esperienze, Milan, Italy
GEAP	General Electric Atomic Power Equipment Dept.
MIT	Massachusetts Institute of Technology
NASA	U.S. National Aeronautics and Space Administration
ORNL	Oak Ridge (now Holifield) National Laboratory
UCLA	University of California, Los Angeles

WAPD Westinghouse Electric Corp. Atomic Power Div.

WCAP Westinghouse Electric Corp. Commerical Atomic Power Activity

OCCURRENCE OF CRITICAL HEAT FLUX
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by

Joseph C. Leung

ABSTRACT

A small-scale experiment using Freon-11 at 130°F and 65 psia in a well-instrumented transparent annular test section was used to study the occurrence of critical heat flux (CHF) during blowdown with flow reversal. The inner stainless steel tube of the annulus was uniformly heated over its 2 ft length. Inlet and exit void fractions were measured by a capacitance technique. Flow regime transition was observed with high speed photography.

A 1-hr contact time between Freon-11 and nitrogen at 130°F and 60 psig was found to greatly affect the steady-state subcooled boiling initial conditions.

Delay in bubble growth was observed in adiabatic blowdown runs. This was caused by the thermodynamic nonequilibrium conditions required for the unstable bubble growth. For the diabatic runs, equilibrium was more closely approached in the test section during the early phase of blowdown.

Critical heat flux did not occur immediately during the flow decay in an approximately 60 msec reversal period. The first or early CHF which occurred at about 400 msec was independent of the blowdown volume and did not propagate upward. An annular flow pattern appeared at the onset of this CHF which occurred only at the lower 8 in. of the heated zone.

For some cases the early CHF was rewet and a second CHF was observed at near depletion of liquid in the heated channel. This CHF however propagated smoothly upward and the time scale for this event was dependent on the system blowdown volume.

CHAPTER 1

INTRODUCTION

Since both boiling water reactors (BWRs) and pressurized water reactors (PWRs) operate at elevated temperatures and pressures (1200 psia and 2250 psia, respectively),¹ if a major rupture of the primary coolant pipe occurred, commonly called a loss-of-coolant accident (LOCA)* the coolant would be lost from the core region.^{1,2} Current safety research is centered about specific phenomena relating to the two-phase behavior of the water following this transient. These experiments concentrate on areas such as critical flow, which determines the rate at which the coolant is discharged from the core, transient boiling heat transfer crisis or critical heat flux (CHF) which determines when the fuel rods will exceed the local critical values, and emergency core cooling system (ECCS) performance, which investigates the reentry of cooling water into the core.

In a PWR, a guillotine rupture of the inlet coolant pipe results in a fast depressurization, known as blowdown. A number of core flow directions may result depending on the relative thermodynamic conditions of the coolant inside the core and at the plenums, possible configurations being flow reversal, bidirectional flow, and counter-current flow. Current numerical analyses generally assume that steady-state critical heat flux correlations apply to this transient system. Consequently, as flow goes through stagnation, the onset of CHF is predicted immediately. Experiments indicate that CHF does not occur as the flow

*A loss-of-coolant accident is a postulated accident that results from the loss of reactor coolant at a rate in excess of the capability of the reactor coolant makeup system from breaks in the reactor coolant pressure boundary, up to and including a break equivalent in size to the double-ended rupture of the largest pipe of the reactor coolant system (AEC definition, Ref. 3).

reverses, but in fact is delayed by at least several hundred milliseconds. The peak fuel cladding temperature reached during the accident is strongly affected by the time delay to CHF, t_{CHF} .⁴ Accordingly, this CHF prediction has a very significant impact on reported reactor performance with respect to acceptance criteria on clad temperature and metal-to-water reactions among others.

Current water reactor safety research sponsored by the U. S. Nuclear Regulatory Commission consists of integral test programs such as Loss-of-Fluid Test (LOFT) and Semiscale at the Idaho National Engineering Laboratory, and Blowdown Heat Transfer (BDHT) program at Holifield (Oak Ridge) National Laboratory. In order to interpret these test results, one must be able to measure the instantaneous flow, void fraction, and heater surface temperature to understand the basic mechanisms involved. Such is the objective of the present study.

1.1 Background and Definitions

1.1.1 Two-Phase Flow Regimes of Heat Transfer

Consider a flow of subcooled liquid (e.g., water) into the bottom of a long uniformly heated vertical tube. Figure 1.1 shows, in diagrammatic form, the various flow patterns encountered over the length of the tube, together with the corresponding heat transfer regions. When the temperature remains below that necessary for nucleation, the mode of heat transfer is single-phase (liquid) convection (region A). Further downstream the superheated boundary layer is thick enough to support bubble growth at many nucleation sites. The mode of heat transfer in the presence of subcooled liquid (region B) is known as subcooled nucleate boiling. Saturated nucleate boiling region starts when the bulk liquid temperature reaches saturation. Slug flow (region D) results

Fig. 1.1. Two-Phase Flow Regime (after Collier^R).

when bubbles agglomerate and coalesce in the bulk stream. Annular flow occurs further down the channel (region E). In this region heat is mostly carried away from the wall by forced convection in the film to the liquid-vapor interface, where evaporation takes place. Complete evaporation of the liquid film results in 'dryout' and is accompanied by a rise in wall temperature. The liquid deficient region (region G) consists of high velocity vapor with entrained droplets, and single phase (vapor) heat transfer (region H) results when all liquid droplets disappear by evaporation.

1.1.2 Boiling Crisis

In forced convective boiling the boiling crisis is characterized by a sudden reduction in the heat transfer coefficient due to the change of heat transfer mechanisms as indicated by a temperature oscillation⁵ or excursion of the heated surface.^{6,7} The maximum heat flux just before the boiling crisis occurs is called the critical heat flux. At least two mechanisms of boiling crisis have been differentiated:

Departure from Nucleate Boiling (DNB) occurs in a bubbly flow pattern. When the bubble boundary layer becomes thick enough to prevent liquid from contacting the hot surface, boiling changes from nucleate boiling to film boiling resulting in an insulating vapor film covering the heating surface.⁶ Due to the low thermal conductivity of the vapor coupled with the relatively high heat flux values required to initiate film boiling, this process is characterized by an excursive temperature behavior.

'Dryout' occurs in an annular flow pattern. The liquid film which normally covers and cools the heating surface becomes too thin and dry patches form and spread as a result of breakdown of the film. Unlike

the conditions at DNB, the higher heat transfer coefficients associated with high velocity vapor flow and lower critical heat fluxes cause only modest temperature excursions at 'dryout.'⁷

1.2 Critical Heat Flux Models

Although many subcooled CHF models have been proposed recently, the mechanism is still not fully understood and these hypotheses still remain to be confirmed. Considerably more is known about the mechanism of 'dryout' in the higher quality regions. Various models proposed in the past are summarized below.

1.2.1 Subcooled Critical Heat Flux

Chang,⁹ formulating a model based on the hydrodynamic stability of bubbles, considered the bubbles to reach a critical velocity normal to the surface at CHF. This velocity is determined by taking force balance on the bubble and CHF is calculated based on enthalpy balance.

Bankoff¹⁰ presented the Sequential Rate Process model based on a turbulent bubble layer at the wall and a single-phase turbulent liquid core. CHF occurs when the core is unable to remove heat as fast as it is transmitted by the wall layer, resulting in a marked increase in bubble size and population, which in turn results in bubble coalescence and wall dryout.

Kirby et al.^{11a,b} suggested that CHF corresponds to the drying up of a liquid film under one of the large vapor bubbles. They later tested the validity of this mechanism by measuring the electrical conductance between the heating surface and the flowing liquid. The results were inconsistent with the proposed mechanism and CHF was found to occur in the absence of large vapor bubbles in the nucleate boiling

region. The spreading of a dry patch following microlayer evaporation under a nucleate boiling bubbles was suggested as a possible alternative mechanism.^{11c}

Fiori and Bergles¹² observed in low pressure water system that near CHF condition, the flow pattern is either slug or froth flow. Their model, being very similar to Kirby's, assumes that when a vapor clot passes near the wall, the nucleation bubbles disrupt the superheated liquid film and a dry spot appears on the wall. This dry spot is quickly quenched by the liquid following the bubble. CHF will occur when the temperature increase due to the dry spot is greater than the temperature drop resulting from quenching. It was suggested that when the Leidenfrost temperature is reached, a continuous vapor patch covers the surface locally, leading to CHF.

Another model, proposed by Kutateladze¹³ and modified by Tong¹⁴ considers the subcooled CHF to be the result of separation of the hydrodynamic boundary layer, analogous to that induced by the injection of gas into a flowing liquid through a porous wall. Dean et al.¹⁵ measured the CHF on a porous plate while various amounts of vapor were being injected into the flow stream. Their study showed that CHF was reduced with vapor injection which tends to support Kutateladze's and Tong's model.

The separation model was recently developed further by Purcupile and Gouse.¹⁶ They postulated that CHF occurs when the vapor injection into the flow stream reduces the velocity gradient near the wall and as the vapor bubbles become almost stagnant, a transition is made from nucleate to film boiling. By defining CHF to be the sum of the heat transferred by vapor generation and by liquid displacement at CHF point,

they were able to use the developed correlation to scale CHF between water and Freons.

1.2.2 'Dryout' CHF in High Quality Flow Regimes

A significant contribution to the understanding of the mechanism of dryout was the model proposed by Isbin and his coworkers.^{17,18} This model assumes an annular film thickness on the wall resulting from the processes of liquid entrainment, evaporation, and deposition. Dryout was assumed if the film thickness reduced to zero. Later Grace¹⁹ suggested that droplet deposition was negligible in most situations of practical interest.

Becker and Persson²⁰ further simplified Isbin's solution and showed that dryout was a local phenomenon independent of inlet subcooling and L/D ratio.

Goldmann et al.²¹ assumed a fog or mist flow pattern in which the existence of a liquid film can be neglected. The wall was assumed to be cooled by evaporation of droplets deposited, and the droplet diffusion through a vapor boundary layer was suggested as the controlling mechanism for dryout.

Stein²² considered a flowing liquid film from which entrainment is negligible. Thus dryout occurs when the rate of evaporation exceeds the rate of droplet supply to the film. He further simplified it to the case proposed by Goldmann et al. but proposed a different mathematical approach to the droplet transport problem.

Tippets²³ extended Isbin's model to include the effect of film stability. By assuming the film could be represented by a potential flow scheme, he determined from perturbation methods, when the interface would become unstable due to the forces resulting from entrainment and deposition, but not evaporation.

Chung²⁴ proposed a theoretical model for liquid mass transfer between the core of the annular flow and the film, and used it to predict CHF due to film depletion. Results indicated that the liquid entrainment was the controlling mechanism, and its local entrainment rate was governed by the Weber number associated with the upstream liquid film.

Hewitt *et al.*^{25,26} demonstrated that indeed CHF condition occurred when the flow rate of the liquid film on the heated wall fell to zero. They suggested that bubble nucleation observed within the film could enhance entrainment at high heat flux and cause film breakdown.

Lately the study of the stability of liquid film flow has received much attention. Not only is it related to the dryout process but also it is of great interest in the rewetting of a hot dry surface in emergency core cooling (ECC). The purely fluid-mechanical problem, in the absence of heat transfer, is well known, having been studied by Yih²⁷ and Benjamin.²⁸ Bankoff²⁹ extended their analyses to the case with heat addition and showed that evaporation from the surface of a falling liquid film has a destabilizing effect.

The effect of uneven heating and surface tension forces on film breakdown was reported by Norman and McIntyre,³⁰ Hallett,³¹ Simon and Hsu,³² and Dubrovskii *et al.*³³

Hence there appears to be at least two important analytical approaches to dryout: the film thickness and its stability, and the liquid mass transfer between the core and the film. It is uncertain at this time which model more accurately reflects the physical processes involved.

1.3 LOCA and Associated Phenomena in a PWR

One of the unlikely events considered in safety analysis of a PWR is a hypothetical rupture of the primary piping system producing a loss-

of-coolant accident (LOCA).¹ Preliminary results of an USAEC study, called the Rasmussen study, has shown that the probability of a large break LOCA is once in 10,000 years for a PWR² and this hypothetical LOCA generally becomes the design basis accident (DBA) because it presents the most extreme accident conditions. A schematic of a PWR system is shown in Fig. 1.2. The largest break is taken as though the pipe separated to permit flow out each end--a double-ended guillotine pipe break. The fluid initially undergoes decompression from a subcooled state, and reaches saturated or two-phase conditions rapidly. This decompression is controlled by the discharge of fluid from the system, which is, in turn, controlled by the compressibility of the flashing, two-phase mixture.

The cold leg or inlet break allows rapid decompression of the lower plenum in the reactor vessel because the fluid path from the break to the lower plenum is short and unobstructed in comparison to the path from the break to the upper plenum through the steam generator, see Fig. 1.2. As a result, flow is caused to change from upflow to stagnation, and possibly to downflow.

A major parameter in LOCA analyses is the time to critical heat flux, that is, the period of time from initiation of the break to the time CHF occurs in the core. Presently the steady state correlations accepted for use in LOCA transients³⁴ include the W-3,³⁵ B&W-2³⁶ and Macbeth³⁷ correlations. Slifer's³⁸ transient CHF correlation is also accepted. During the initial flow decay in the core, the CHF is assumed to be exceeded when the coolant flow rate becomes zero for a few msec or so. Further, it was stated by AEC (Atomic Energy Commission)³ that, "After CHF is first predicted at an axial fuel rod location during

Fig. 1.2. Schematic of a PWR System (after Catton *et al.*²). [Arrows indicate normal operating flow direction.]

blowdown, the calculation shall not use nucleate boiling heat transfer correlations at that location subsequently during the blowdown even if the calculated local fluid and surface conditions would apparently justify the reestablishment of nucleate boiling." Rewetting is only permitted to occur during the reflood period of a LOCA.

CHAPTER II

CRITICAL HEAT FLUX UNDER TRANSIENT CONDITIONS

A LITERATURE SURVEY

The present review covers the investigation of CHF under the following transient conditions:

- CHF in power transients.
- CHF in flow decay transients.
- CHF in pressure transients.
- CHF in reverse flow and stagnation.

Pool boiling experiments will be briefly summarized followed by an up-to-date review on flow boiling transients.

2.1 Pool Boiling CHF under Transient Conditions

Almost all pool boiling transients were tested under power excursion conditions. Cole,³⁹ 1956, and Graham,⁴⁰ 1964, studied stepwise power input with thin ribbon test sections. Using exponential power input, Rosenthal and Miller,⁴¹ 1957, observed temperature overshoot and delay time occurring in the process of incipient boiling. The effects of the excursion period and subcooling were studied and the authors concluded that transient CHF does not differ very much from the corresponding steady-state CHF. The same conclusion was made by Borishansky and Foxin,⁴² 1969.

More recently, Hall and Harrison,⁴³ 1966, have found that in a rapid exponential power transient with $\Delta t < 1$ msec, film boiling was always preceded by a short burst of nucleate boiling at heat fluxes about 5 to 10 times the steady-state values.

Tachibana et al.,⁴⁴ 1968, found that the transient CHF increases as the power impulse time decreases. They explained that higher CHF

may be due to delays in bubble growth and microlayer evaporation. The liquid layer adjacent to the heater is rapidly superheated, and nucleation occurs at many sites. The time period is such that all bubbles are in their early growth phase; rapid evaporation of a thin liquid film at the base of the bubble occurs, provoking very effective cooling.

Johnson,^{3,4} 1970, presented a summary of an extensive experimental investigation^{3,4} for transient boiling in water subject to exponential heat inputs. The transient CHF was as much as four times the steady-state CHF.

Sakurai et al.,⁵ 1970, suggested an expression relating transient CHF to steady-state CHF by including a term involving the period of the exponential heat input, p, i.e.,

$$q_{CHF}^* = q_{CHF} + kp^{-m} \quad (2.1)$$

where k and m are strong functions of the pool subcooling.

Howell and Bell,^{6,7} 1963, studied the effect of 5-10 psi/sec depressurization rates on CHF in the pool boiling of water. They found that the time to CHF increases with decreasing rate of depressurization and is longer than that predicted by steady-state correlations. This time delay was explained in terms of the diminishing thickness of the thermal boundary layer caused by violent convective currents in rapid bulk nucleation.

2.2 Flow Boiling CHF Under Transient Conditions

This section covers the important results in flow boiling transient CHF studies being conducted in many countries. The pertinent experimental conditions are listed in Table 2.1.

TABLE 2.1 Experimental Conditions for Transient Critical Heat Flux Studies

Reference	Test Section	Test Fluid	Pressure psia	Initial Mass Velocity 10^6 lbm/hr-ft^2	Initial Heat Flux 10^5 Btu/hr-ft^2	Type of Transient Operation
Schrock 1961-1966	0.004 x 0.25 in. ribbon heater in a rectangular channel	water	15-2000	0-3.0	0.15-0.20	Exponential power increase
Tong 1965	19-rod bundle 4.5 ft long	water	1500	2.0-4.0	5.0-10.0	Power ramps Flow coastdown
1967	19- and 21-rod bundles 5 ft long	water	1500	1.0-3.0	3.0-6.4	Power ramps at 13%/sec maximum Flow coastdown
Moxon 1967	1-tube and 37-rod bundle; both 12 ft long	water	1000	1.5-2.0	--	Power ramps at 150%/sec Flow coastdown
Redfield 1967	0.08 x 2.25 x 66 in. rectangular channel	water	1100	1.0	1.6	Inlet break Outlet break
Premoli 1969	1-tube 13 ft long	water	860	1.0-2.0	1.3-2.0	Inlet break Outlet break, Flow decay
Cermak 1970	21-rod bundle 5 ft long	water	750-1500	1.0-3.0	5.0-10.0	Outlet break
ORNL, FCTF 1970	1-rod in annular channel and 7-rod bundle, both 2 ft heated	water	1500	1.0	3.5	Outlet break
1974	1-rod in annular channel 12 ft heated	water	1500	1.0	3.7 ave chopped cosine	Inlet break
Kawamura 1970	1-annulus, inner tube heated 1.6 in. heated	water	15	0.35-1.4	6.3	Exponential power increase

TABLE 2.1 (contd.)

Reference	Test Section	Test Fluid	Pressure psia	Initial Mass Velocity 10^6 lbm/hr-ft ²	Initial Heat Flux 10^5 Btu/hr-ft ²	Type of Transient Operation
Slifer 1971	1-rod in annular channel 9 ft long	water	1000	0.25-1.0	2.0-5.0	Outlet break, and/or flow decay
Lahey 1971	1-rod in annulus 9 ft heated	water	1000	0.2-1.0 0.25-1.0	2.1-5.7 3.8-8.3	High-low-high flow Outlet break
	9-rod bundle 6 ft heated	water	1000	0.5-1.0 0.6-1.0 0.24-1.0	2.5-5.0 0.27-1.0 1.8-6.6	Flow decay ON/OFF/ON flow Outlet break
	16-rod bundle 12 ft heated	water	1000	0.6-1.0 0.6-1.0	1.7-3.1 ave 0.2-2.0 ave chopped cosine	Flow decay ON/OFF/ON flow
Farman 1972	1 tube 3 ft long	water	2250	1.0-2.5	6.3-11.0	Outlet break with and without flow decay
Morgan 1972	9-rod bundle 6 ft heated	water	2200	2.5	3.0	Outlet break and inlet break with programmed power
Smirnov 1972	1-tube	water	1400	0.74-2.2	0.9-1.07	Flow decay
Fritz 1972	1-tube 9.8 ft heated	water	1090- 2030	--	4.4	Outlet break
Hein 1972	1-tube, 1-annulus, 4- and 9-rod bundle	Freon-12	155	0.37-2.58	0.0-1.0	Flow and power transient
		water	1000,2000	~1.0	0.0-4.5	Flow and power transient
Hicken 1972	1-tube, 9.8 ft heated	water	1300	--	5.23	Inlet break, outlet break and double-ended break
Tokumitsu 1972	28-rod bundle in 1 pressure tube, 12 ft heated	water	1070	--	0.13	Downcomer rupture, and inlet break

TABLE 2.1 (Contd.)

Reference	Test Section	Test Fluid	Pressure psia	Initial Mass Velocity 10^6 lbm/hr-ft ²	Initial Heat Flux 10^4 Btu/hr-ft ²	Type of Transient Operation
ANC, Semiscale 1972	single-loop, 120-rod bundle 9 in. uniformly heated	water	2250	0.12	3.0-3.8	Outlet break Inlet break
Gaspari 1973,1974	1-annulus outer tube heated, 1-tube, both 13 ft, uniform heating	water	710	0.1-2.0 0.5-2.0 0.5-1.1	0.75-2.63 0.1-2.63 1.1-2.64	Flow stoppage Outlet break Exponential power increase
Zielke 1973	9-rod bundle, 6 ft heated length, radial and axial uniform heating	water	--	1.0-3.0	--	Flow decay, and power ramp at 52 in .2 sec
Aoki 1973	1-annulus, only inside rod heated	water	300-1010	up to 1.55	0.0	Rampwise and stepwise power to max 1.9×10^4 Btu/hr-ft ²
1974	1-annulus, only inner tube 3.6 in. heated, with outer pyrex tube	water	43	0.37-1.58	9.2	Outlet break
Griffith 1975	1 tube, 8 ft heated	Freon-113	200	2.50	0.24	Flow reversal, inlet flow stoppage, and combined at constant pressure
Banerjee 1973	1 tube, 6.6 ft heated	organic coolant	250	--	4.75	Inlet break
1975	1 tube, 13 ft uniformly heated	water	1450	1.13	2.54	Inlet break

UNITED KINGDOM

AEEW

Moxon and Edwards,⁵⁰ 1967, carried out quite an extensive study of flow and power transients on tubes with uniform and nonuniform heat flux profiles and with a 37-rod bundle with uniform heat flux. Fast transients were studied with the inlet flow halving in less than 500 msec and power rising to 130% of its initial value in 200 msec. Comparisons were made between the experimental and theoretical predictions of the time to CHF. The theoretical predictions were based on local condition hypothesis which states that CHF is a local phenomenon, governed by the local mass velocity, quality, and heat flux. The results of the comparison showed that in all cases the CHF occurred later than that predicted, by the SLIP code⁵¹ and the discrepancy varied between 0 and 400 msec.

ITALY

CISE

Premoli,⁵² 1969, studied the effect of blowdown following simulated ruptures upstream and downstream, and loss-of-flow at constant pressure. Transient mass hold-up in the test section as indicated by the ratio $\rho(t)/\rho(t = 0)$ was measured by quick acting closing valves. Therefore in order to perform a complete transient, several measurements were performed at different sampling time during repeated runs. Transients performed were less than 3 sec and occurrence of CHF was not mentioned. From the result shown in Fig. 2.1 Premoli concluded that voidings due to inlet break were most severe and density decrease was sharp due to the fact that flow reversal caused the inlet, high density, subcooled water to be lost from the test section sooner than under other break conditions.

Fig. 2.1. Dimensionless Density During Various Transients (after Premoli⁵²).

Gaspari et al.,^{53,54} 1973-1974, carried out loss of flow, power surge and outlet break blowdown studies with both annular and tubular test sections, related respectively to 1/14 and 1/40 scaled down configurations of the CIRENE reactor channel. Experimentally CHF occurred at the outlet during blowdown and propagated smoothly upstream. A code named TILT⁵¹ was used to analyze the transient behavior. Theoretical predictions were based on 'hydrodynamic condition hypothesis' which states that CHF is a hydrodynamic phenomenon governed by the hydrodynamic conditions existing in the saturated region of the channel. This and the 'local condition hypothesis' are related to each other through the heat balance equation. Comparison between these two hypotheses was done by Hassid et al.,⁵⁶ 1971, on channels with nonuniform axial heat flux distribution, and they found the hydrodynamic hypothesis to be better. Therefore Gaspari et al. assumed that both in steady-state and transient conditions, the hydrodynamic critical quantities, i.e., G , x , and P do not depend either on how they vary along the channel and with time, or how the heat flux varies along the length and with time, but depend on the boiling length travelled by the liquid which finally reaches CHF point. Time to CHF for flow stoppage and power surge tests was predicted within 150 msec (t_{CHF} was of the order of 0.5-3 sec) using TILT code together with steady-state CHF correlation. For blowdown transients, the computed t_{CHF} values were systematically lower than experimental values in the annular test section. The prediction accuracy was strongly affected by the range of validity of the steady-state CHF correlation and the accuracy of the blowdown code. They noted that significant differences exist between the phase distributions in steady-state and during blowdown where the liquid phase is subject to flashing and

violent acceleration. In particular liquid redistribution between the heated and unheated wall or between droplets and film may be strongly affected due to these effects. Time to CHF was found to decrease with increasing depressurization rate accomplished by variable break areas.

Euratom

Fritz and Riebold,⁵⁷ 1972, investigated the hot leg rupture of a PWR subchannel by diaphragms of 0.315, 0.236, and 0.157 in. diameter. Tests were conducted at initial pressure and water temperature of 2000 psia and 600°F. The onset of CHF occurred over the outlet portion of the channel at ~500 msec and was nearly independent of rupture size, and hence depressurization rate. However the upstream half of the test section did not experience CHF until 4 sec.

USSR

Moscow Power Institute

Smornov et al.,⁵⁸ 1972, conducted flow transient tests in a single tube with flow decreased by 40 to 70% of the steady-state value within 0 to 3 sec. Local mass velocity and quality at inception of CHF were determined by calculation of the transient process. Steady-state CHF correlation was obtained on the same test section. With a sharp decrease in mass velocity, the computed t_{CHF} values were lower than those experimentally observed.

GERMANY

Hein and Mayinger,⁵⁹ 1972, summarized the power and flow transient test results conducted at M.A.N. (Maschinenfabrik Augsburg Nürnberg) in single tube, annulus, and 4- and 9-rod bundles. Figure 2.2 shows an improvement in CHF with linear power excursion time of a few seconds while the equation for power transient proposed by Redfield,⁶⁰ 1965,

Fig. 2.2. Effect of Power Excursion Time on CHF (after Hein *et al.*⁵⁹).

underestimates CHF values. In some tests Freon-12 was used as the modeling fluid and it was claimed that Boure or Stevens' scaling laws were also valid for these slower transients. The results also indicated that improvement in CHF for tests with uniform heat flux was higher than nonuniform ones. The explanation for the improvement was obscure as they did not know whether CHF occurred in bubbly or annular flow. For flow transients good agreement was observed between transient and steady-state CHF. Unlike the power transients, nonuniform heat flux profile did not change this result. However at very low mass flow, improvement did occur and they attributed this to the finite time required for complete evaporation of liquid film on the heated wall.

Hicken et al.,⁶¹ 1972, studied the effect of break area, the ratio in upstream to downstream break area and the heat flux in a single tube at pressure of 1300 psia. Pressure decay was faster with larger breaks and different at various locations in the loop. For small breaks, a rise in system pressure was observed because the volumetric vapor generation rate in the heated section exceeded that which could be discharged at the break. The following generalization was made on t_{CHF} .

- t_{CHF} decreases with increasing ϕ .
- t_{CHF} decreases downstream for large ϕ but this dependency is small at low ϕ .
- with large break upstream t_{CHF} is large for low ϕ however quite short for large ϕ .
- for large upstream breaks, the upper region of test section did not experience CHF.
- t_{CHF} for double-ended breaks were intermediate between the times for single-ended breaks.

JAPAN

Kawamura et al.,⁶² 1970, studied pool and flow boilings of water under power surge conditions at atmospheric pressure. With motion pictures taken at 5,000-10,000 frames/sec, they observed the formation of vapor blanketing as a result of bubble coalescence at the CHF point in subcooled pool boiling. They also found that at the CHF point $N(\bar{R}_{\max})^2 \approx 1$ where N is the nucleation site density and $\bar{R}_{\max}$ is the maximum bubble radius which is proportional to the thermal boundary layer thickness. This postulate indicated that a high flux DNB was possible even with many nucleation sites if $\bar{R}_{\max}$ was kept small, i.e., thin thermal boundary layer. For pool boiling transients, they were able to show that transient CHF is reduced with increasing initial heat flux as is expected from above postulate.

Tokumitsu,⁶³ 1972, conducted LOCA studies on a full scale mockup of the FUGEN reactor, which is a heavy water moderated 165 MW BWR. However the power level was only maintained at 5% of full power in FUGEN, simulating the decay heat, and only three of 25 pressure tubes were electrically heated. In the downcomer rupture experiment, CHF was observed half a minute later at the top of the heated length. Single inlet pipe rupture tests were performed on one of the pressure tubes, each contained 28 heated rods. CHF was observed to start from top of the pin in nonruptured channels, and from bottom in ruptured ones. In general, t_{CHF} reported was rather long, corresponding to the complete depletion of coolant in the channels.

Tokyo Institute of Technology

Aoki and his coworkers,^{64,65} 1973-1974, made quite extensive studies on power excursion and blowdown with inlet quality, power ramp rate,

mass velocity, and heat flux as the parameters. For the case of transient heat input, the observed dependency of transient CHF on the parameters was explained qualitatively according to three regions of flow conditions. At high flow ($G \sim 1.5 \times 10^6$ lbm/hr-ft²) and large subcooling, ϕ_{CHF} seems independent of power ramp rate. This was explained by the fact that CHF mechanism by bubble coalescence is a very localized phenomenon, happening only in the vicinity of heater surface. The bulk conditions are therefore of secondary importance. In the second region where CHF occurred at 50-70% void, transient CHF was found to follow steady-state curve closely. Because of the moderate void fraction, the bulk conditions do affect the transport process at the surface. As for the third region where annular flow persists at high quality and void, CHF caused by 'dryout' occurs rather slowly, hence the term 'slow burnout' had been used by some researchers. Under this condition, ϕ_{CHF} appears to increase as power ramp rate.

The blowdown tests simulating hot leg break conducted by this group at water pressure of 65 psia indicated that CHF did not occur at the sudden vaporization just after blowdown and their observed CHF values were higher than steady-state CHF. In Fig. 2.3 the excess heat flux, i.e., $(\phi_{tCHF} - \phi_{sCHF})$ were plotted versus t_{CHF} . For large subcooling the experimental data was correlated by a curve for each flow velocity. Interesting results were observed in the high speed photographs taken with the transparent test section. To quote, "just after depressurization, (1) vigorous boiling-like sudden vaporization occurred, and then (2) there was a period in which the bubble disappeared or boiled only on the surface of the heater. After a while, (3) vigorous boiling including coalesced bubbles began again and then CHF occurred... The

Fig. 2.3. Time to CHF versus Excess Heat Flux (after Aoki *et al.*⁶⁵).

reason why CHF never occurred in region (1) seemed to be due to the large growth velocity of sudden vaporized bubble and also owing to that the stable liquid layer was kept on the heater surface through the bubble growth because steam was supplied to the bubble not only from the heater surface side but also from the superheated layer in bulk liquid." The collapse of bubbles in region (2) was apparently caused by subcooled water on the unheated wall (glass shroud) or from upstream. No flow reversal was studied.

CANADA

AECL, Pinawa

Banerjee et al.,⁶⁶ 1973, conducted blowdown studies of an organic coolant in a single heated tubular channel. Continuity, momentum, and energy equations were solved by the method of characteristics and the predictions of various parameters during transients were satisfactory. The report of the occurrence of CHF was not included.

More recently Banerjee et al.,⁶⁷ 1975, reported some initial diabatic blowdown tests conducted at the CISE high pressure water loop. For 100% inlet break area, CHF occurred at 3 and 7 sec, respectively, for inlet subcooling of 104 and 240°C whereas no CHF was observed for the 5% inlet break. From the coolant mass decay curve reported, onset of CHF corresponded to 10-15% of initial mass remaining.

USA

Westinghouse and Columbia University

Both power and flow transients were reported by Tong and his coworkers,^{68,69} 1965, 1967. For ramp power excursions, transient CHF levels were approximately equal to the corresponding steady-state values. In their simulated flow coastdown tests, no CHF was observed when a

2 sec delay in the effective control rod insertion was simulated by appropriate control of the power.

Cermak et al.,⁷⁶ 1970, studied the behavior of transient CHF during blowdown in rod bundles at high pressure. Steady-state CHF tests, conducted on the same bundle, were used to predict the transient CHF. The local conditions hypothesis was used with calculations based on the CHICKIN computer code⁶⁰ developed at Westinghouse. The time to CHF was reported to be affected by the rate of pressure decay, qualitatively in agreement with Gaspari's findings. The calculation using above procedure underestimated t_{CHF} by 0 to 4.2 sec for actual observed t_{CHF} of 1 to 7 sec. Hence, it was concluded that CHF during blowdown could be predicted on the basis of steady-state data but the prediction was a conservative one.

Farman and Cermak,⁷¹ 1972, studied the CHF and beyond CHF heat transfer during blowdown in a 3 ft tubular test section. Outlet pipe rupture was simulated by discharging the system coolant to atmosphere. However two-phase critical flow through the discharge valve was prevented by placing a tank of subcooled water in the line upstream of the valve to ensure single-phase exist at discharge location. Transient CHF occurred at the exit but no mention was made of its propagation upstream. Steady-state correlation developed by Tong³⁵ at Westinghouse was found to be conservative in predicting transient CHF results.

Bettis Atomic Power Laboratory

Redfield et al.,⁷² 1967, conducted blowdown studies for a rectangular heated channel simulating both cold and hot leg break of different sizes. The loop was not geometrically similar to a typical PWR, but it did contain the essential elements: a pressurizer, a canned rotor pump,

and a heat exchanger. The test results were compared with predictions using the FLASH code.⁷³ Though instantaneous flow reversal was predicted in the cold leg breaks, DP cells across the test section showed an oscillatory behavior about zero during the initial 1 sec. Temperature measurement indicated good heat transfer during the initial part of the transient as the wall temperature followed the falling saturation temperature of the coolant. Time to CHF was measured at 155, 182, and 280 sec for the three tests reported while FLASH predicted values were 108, 160, and 380 sec, respectively.

Babcock and Wilcox

Morgan et al.,⁷⁴ 1972, presented some cold leg break simulation tests using the Babcock and Wilcox Boiling Heat Transfer Loop. The test section consisted of a 9-rod, PWR bundle, 6 ft long. The Babcock and Wilcox evaluation model CRAFT⁷⁵ was used to calculate the fuel temperature by first assuming one of three modes of heat transfer exists at the surface, namely forced convection, nucleate boiling, or stable film boiling. When the computed heat flux exceeds the corresponding CHF determined by the W-3 correlation,³⁵ CRAFT switches to stable film boiling. By using criteria according to the USAEC's Interim Policy Statement,³ where this transition is assumed to occur almost instantaneously after the inlet break, the cladding temperature is calculated to rise sharply and continue to rise until power trips. The prediction by CRAFT was done on the Semiscale Test 845 and the result was highly conservative, see Fig. 2.4. However, by delaying CHF until dryout when quality in the core is close to 1.0, much better agreement was obtained (hence termed the CRAFT Best Estimate). The same conclusion was made in the ORNL blowdown data reported by Lawson,⁷⁶ i.e., the reported hot and

Fig. 2.4. Measured and Predicted Wall Temperature for Semiscale Test 845 (after Morgan *et al.*⁷⁴).

cold leg break simulations were again in reasonable agreement with the best estimate calculation, see Fig. 2.5. It was concluded that flow reversal occurring in the first few seconds after a large, cold leg break was not expected to give rise to a thermally identifiable flow stagnation. This tends to support the view that the early blowdown phase is dominated by a high degree of turbulence created by vigorous nucleation rather than by a flow boundary layer phenomenon as film boiling.

Flow and power transients were performed on a 6-ft long, 9-rod bundle by Zielke and Wilson,⁷⁷ 1973. Typical reactor coolant pump coastdown and 5% power ramp were simulated. Transient CHF values were compared with the Babcock and Wilcox-2³⁶ steady-state CHF correlation, and all data points except two had an experimental-to-predicted CHF ratio > 1.0.

Holifield National Laboratory, HNL (ORNL)

Lawson,^{76,78} 1970, had reported blowdown results for both single-rod and 7-rod bundle with an outlet break. Blowdown times ranging from 7 to 60 sec were achieved by varying the diameter of a critical flow orifice located near the rupture disks. One of the representative single-rod tests is shown in Fig. 2.6, with power reduced to 10% at 3 sec into blowdown. No CHF was observed until the loop was at, or near, atmospheric pressure when the coolant was almost entirely discharged. For the 15 sec 7-rod bundle test reported,⁷⁸ CHF occurred at ~9.5 sec when loop pressure was ~200 psig.

The effect of flow reversal resulting from an inlet pipe rupture was included in the more recent HNL Blowdown Heat Transfer program.^{79,80} The first is the FCTF (Forced Convection Test Facility) which consists

Fig. 2.5. Wall Temperature for Babcock and Wilcox Blowdown Test (after Morgan et al.⁷⁴).

Fig. 2.6. Wall Temperature and Pressure for Holifield Blowdown Test (after Lawson⁷⁶).

of a single 12-ft heated length rod in annular geometry, installed in the same loop as used in Lawson's tests. The heat generation profile was that of a chopped cosine with maximum heat flux of 610,000 Btu/hr-ft² in the middle section. The first six blowdown tests simulating single-ended inlet break showed that t_{CHF} varied from 4.0 to 9.5 sec as shown in Fig. 2.7. CHF was found to occur downstream of the highest heat flux zone and propagated along the heater in both directions. The THTF (Thermal Hydraulic Test Facility) will study blowdown with and without flow reversal in a 49-pin electrically heated PWR bundle.

Aerojet Nuclear Company (ANC)

The Semiscale Blowdown facility has a primary objective of providing an experimental data base to help assess the capability and adequacy of analytical models during a hypothetical LOCA. By 1970, single-loop semiscale testing with a 9-in. heated core was completed.⁸¹ One of the findings was that the coolant in the lower (inlet during steady state) and upper plenums did not simultaneously decompress to the saturation pressures corresponding to the individual plenum temperature following a hot leg break. Rather, the coolant decompressed to a pressure corresponding to the saturation temperature in the upper plenum within 0.2 sec. Coolant in the lower plenum remained subcooled for 2-3 sec. For the hot leg break test (Test 825)⁸¹ with full power on for 14 sec into blowdown, CHF occurred only at the top of the heated length at 12.5 sec. The depressurization rate during the saturation blowdown was slow, ~27 psia/sec. The heat transfer coefficient was calculated to rise up to 10,000 Btu/hr-ft²-°F within 100 msec after rupture.

The first test (Test 845) involving ECC injection was reported in 1972.⁸² Major observations for this cold leg rupture test were:

Fig. 2.7. Pressure History for Holifield Single-Rod Test (after ORNL Report⁸⁰).

- The reversed flow through the core carried a two-phase fluid at high enthalpy causing the lower plenum temperature to rise.
- Occurrence of CHF was delayed until 2.1 sec at the top location on central pin but the cladding was rewetted at 3.7 sec. The rewetting was attributed to increased downflow. CHF occurred subsequently at all monitored core locations between 5 and 7 sec.
- Following the transition from subcooled to saturated blowdown, the system decompressed at essentially fluid saturation temperature for the remainder of the transient.
- A mass balance indicated that ~25% of the fluid mass was ejected through the pump and cold leg side of the break and 75% was ejected through the vessel inlet and out the break.
- Vessel fluid qualities were >60% within 2 sec after rupture.

General Electric (GE)

Slifer,³⁸ 1971, reported on GE models for BWR performance in a LOCA. Employing the GE Heat Transfer Loop at San Jose, he studied the following transients using both single rod and 9-rod square-array geometry:

- Depressurization at rates of 50 to 100 psi/sec with constant flow showed that using the GE steady-state correlations (Janssen-Levy for single tube and Hensch-Levy for rod bundles) to predict the onset of CHF in transient was conservative.
- Flow decay tests also indicated the same conservatism.
- Combined depressurization and flow decay tests also showed that CHF did not occur earlier than steady-state prediction.

Lahey and his coworkers,^{83,84,85} 1971-1972, reported results from experimental transient CHF program in single-, 9-, and 16-rod test

sections. A simplified analysis based on the method of characteristics was developed which allowed prediction of the transient flow and quality in a heated channel. Computer code MAYU⁸⁴ was generated for transient CHF predictions. For flow decay transients, CHF occurred above the critical value corresponding to the quasi-steady-state predictions. Flow ON/OFF/ON transients were conducted such that the inlet flow was suddenly stalled for a time period and then turned back on. Time to CHF ranged from 2.3 sec for $\phi = 149,000$ Btu/hr-ft² in 16-rod bundle to 21 sec for $\phi = 51,000$ Btu/hr-ft². These tests indicated that pool boiling is an effective mode of heat transfer. In many cases the rods were found to be rewet with no apparent difficulties once the flow was re-established. For the depressurization tests with 50 to 100 psi/sec rate, no immediate onset of CHF took place. In general, the quasi-steady-state and MAYU method of prediction overestimated t_{CHF} , and in some cases, CHF was not experimentally observed.

Massachusetts Institute of Technology (MIT)

Using Freon-113 at 200 psia, Griffith and his students,^{86,87} 1975, studied transient CHF during flow reversal, inlet flow blockage, and a combination of the two. For the case of initial uniform heat flux and subcooled upflow, the transient consisted of two flow reversals bounding a period of downflow, τ_d . Their results showed that CHF did not occur at flow reversal but depended on the history of the peak enthalpy fluid. In Fig. 2.8, the heat flux marked the CHF boundary for the corresponding downflow duration, τ_d was plotted. For negligibly small τ_d (<100 msec) no CHF was observed until heat flux is at ϕ_{sCHF} level in which case the disturbance would trigger an incipient CHF. At τ_d of about one channel transport time (~1 sec in their runs), the ϕ_{CHF} value observed was

Fig. 2.8. MIT Double Flow Reversal Results (after Smith⁸⁷).

lowest. This result was explained by the fact that the enthalpy peak traversed three full passes through the heated section. Whereas transients with longer τ_d , the dependency of ϕ_{tCHF} on τ_d was small since they were essentially single reversals from initially steady downflow to upflow. They also postulated that if CHF did not occur during flow reversal, complete voiding due to incipient bulk boiling of Freon from the heated channel could occur in 1 to 2 sec under the test conditions. A small but versatile computer program BACTRAC3 was successfully developed⁸⁷ to predict the occurrence of CHF during rapid flow reversal transients.

To conclude then, CHF in a flow decay transient without reversal and in a power excursion can be predicted, but often conservatively, using steady-state CHF correlations with local parameters calculated by means of computer codes.^{50,53,59,64,77} Caution has to be exercised at low flow since only some correlations are suitable in this range. The concept of flooding heat flux has recently been introduced⁸⁸ and flooding phenomenon was experimentally observed in the low velocity, counter-current flow, CHF studies conducted at MIT.⁸⁹ Time to CHF for flow and power transients depends on a number of parameters, particularly on the critical heat flux ratio (CHFR)* at the steady-state condition. Flow stagnation experiments indicated that heat flux at low flow can be quite high as in pool boiling.⁸⁵ At this low flow CHF was found to depend on void fraction⁸⁹ (generally unknown in pool boiling experiments).

One of the major findings in blowdown studies is that time to CHF decreases as the rate of depressurization increases.^{54,70} It is interesting to note that the same conclusion was arrived at in pool boiling

*The critical heat flux ratio is defined as the ratio of the predicted critical heat flux to the initial local heat flux before the transient.

depressurization study.⁴⁹ High speed photos showed vigorous bulk nucleation, agreement with the findings in flow boiling depressurization.⁶⁵ Vigorous nucleation, acceleration,⁵³ turbulence,⁷⁴ and stability of the liquid film⁶⁵ on the heater wall during the initial phase of blowdown have all been proposed as plausible explanation for the time delay to CHF. But for time well into blowdown, annular flow with liquid film on the wall was suggested as the major mode of efficient heat transfer observed.⁷⁶ In general, CHF during blowdown occurred at a very late time, corresponding to near or complete loss of the coolant mass in the test section.^{67,74,76} At such a point where high quality flow was experienced, 'dryout' is probably the cause of the temperature excursion. For a hot leg break, CHF was usually reported to occur first at the exit and then propagate upstream,^{53,57,61,71,82} In the case of nonuniform heat flux, bidirectional propagation of the CHF front which occurred originally at the maximum heat flux location was observed.⁸⁰

CHAPTER III

EXPERIMENTAL INVESTIGATION

3.1 Outline of Experimental Technique

3.1.1 Choice of Working Fluid

In the study of steady-state CHF, success in scaling down of the water system has been achieved by a liquid having a lower latent heat of vaporization and operating at pressure having the same liquid/vapor density ratio as the water system. These liquids reported in the literature are Freon-11,^{90,91} Freon-12,^{92,93} Freon-113,⁹⁴ Freon-114,⁹⁵ Freon-22,⁹⁶ and CO₂.⁹⁷ Freons are usually chosen because of their very low latent heat, relatively low hazard and well established fluid properties. Freon-11 at 260 psia and 237°F has been used to scale CHF of steam-water mixtures at 1500 psia and 500°F.⁹⁸ At this lower pressure, visualization studies can be conducted. Furthermore heater failures which frequently result in post-CHF operation in water have not been reported in Freon studies. However, Freon-to-water scaling during pressure transient has not been explored and it is uncertain that scaling can be achieved. A discussion of scaling is included in Appendix J.

This experiment utilizes Freon-11 as the boiling fluid in a transparent annular test section so that direct photographic observations can be made of the fluid behavior prior to, during, and after local CHF resulting from a depressurization transient. Because of the transparent glass section, the experiment is conducted at 65 psia, corresponding to water equivalent pressure of 450 psia based on density ratio.

3.1.2 Measurement of Flow During Transient

Accurate flow measurements during two-phase transients are difficult because of the various flow configurations that can be

obtained. In single-phase flow, turbine flowmeter is commonly employed. Based on Grey's analysis⁹⁹ for the transient response of a turbine flowmeter, Popper¹⁰⁰ found that the turbine will respond very closely to the liquid velocity even in the presence of two-phase mixture, and the error is represented by the term

$$\frac{(1 - \alpha)\rho_f + \alpha\rho_g S^2}{(1 - \alpha)\rho_f + \alpha\rho_g S}$$

The error is small for low and intermediate void fractions as either slip ratio $S > 1$ or $\rho_g \ll \rho_f$. Later Rouhani¹⁰¹ cautioned that only for low quality that the response of the turbine will be proportional to liquid velocity alone.

Recently Hutcherson¹⁰² used weight transducer measurement to estimate mass flow rate during blowdown. The blowdown vessel in his study was suspended from a weight transducer, and the mass flow rate was determined by differentiating the mass decay signal. A similar arrangement is used in the present study to measure the mass flow for comparatively long transients. In addition, turbine meters are located in the inlet and outlet legs of the test section and employed mainly to indicate the flow direction and to detect significant changes in flow velocity.

3.1.3 Measurement of Void During Transient

Most of the void measuring techniques are based on the attenuation of a radiation beam such as gamma attenuation and x-ray attenuation. Among the nonnuclear methods of void measurement, one of the most important is the impedance method,^{103,104} which is based on the difference between liquid and vapor electrical conductivity and dielectric

constant. The impedance method again can be divided into conductivity¹⁰⁵ and capacitance¹⁰⁶ measurements. Dukler¹⁰⁷ in 1952 reported using a capacitance probe for measurement of falling film thickness. A commercially available capacitance void meter, manufactured by Ikor, Inc. (Burlington, Mass.), has been successfully tested for Freon and water at three different laboratories^{84,108,109} in this country. However, this commercial void meter is tubular and therefore only suitable for mounting in a circular channel. The first attempt to measure void in an annulus by a capacitance probe used a brass electrode surrounding the outside of the glass tube and measured the capacitance between the inside heater tube and the electrode.¹¹⁰ In the present study, a similar void probe is used because of its simplicity in design and low cost. The electronic circuit for dynamic void measurement as well as the calibration test are reported in Appendix E.

3.2 Description of Experimental Apparatus

3.2.1 Freon-11 Test Facility

This experiment was conducted in the Freon Test Loop of the Reactor Analysis and Safety Division, Argonne National Laboratory. The apparatus, which is shown in Fig. 3.1, was a single-pass loop and consisted of three main vessels; a supply vessel, a variable volume blow-down vessel, and a receiving vessel. A description of each vessel is as follows:

Supply Vessel

A 5.5 kw 14 in. Watlow immersion heater was used for heating ~600 lb of Freon to the desired operating temperature inside this 8 ft³ vessel. Four immersed thermocouples (Type K) were installed at four vertical locations and these temperatures were monitored on a multichannel

recorder during heating. The close agreement among the four thermocouples indicates that no significant stratification occurs.

The entire weight of the vessel and its contents were suspended from a weight transducer which was a Statham, UC3, unbonded-type strain gauge, Universal Transducing Cell. The cell was mounted in a Statham, UL4, 1000 lbf load cell accessory head which permitted the cell to detect a maximum of 1250 lbf of total suspended weight. The weight transducer was calibrated with dead weights and the results are shown in Appendix I.

As a safety precaution against overpressurization of the supply vessel, a pressure switch was installed to cut off the heater power and open a solenoid valve located at the top of the vessel.

During the test, the supply vessel was maintained at a constant pressure above the liquid saturation value by three nitrogen cylinders to ensure that Freon flowing in the loop was subcooled.

Blowdown Vessel

The blowdown vessel was a 30-in. long, SS-304 cylinder with an inside diameter of 4 in., and a wall 0.237 in. thick. It had a dome end cap on the bottom, and a flange connection on the top. Both ends had 1 in. pipe connections. The volume was altered as desired by inserting a cylindrical displacement vessel through the flanged end.

The vessel was wrapped with a 30 ft shielded nichrome heater wire on the outside and covered with approximately 1-in. of fiberglass insulation. Four immersed thermocouples (Type K) were installed at four vertical locations and they were continuously monitored on a multichannel recorder. The empty vessel was brought to the operating temperature before the test so that the heat loss to the vessel wall was minimal.

Similar to the supply vessel, the blowdown vessel was suspended from an identical type weight transducer. A 500 lbf load cell accessory head was used in this case, with the calibration curve shown in Appendix I.

Receiving Vessel

This large vessel of $\sim 45 \text{ ft}^3$ with cooling capability supplied from a refrigeration unit, is shown in Fig. 3.2. The refrigeration unit consisted of a tank containing a 50% water-50% ethylene glycol mixture to which 100 to 150 lb of dry ice was added to increase the cooling capacity. A closed-circuit cooling coil was wrapped around the receiving vessel and formed a helix inside the cooling tank. Ethylene glycol solution was continuously circulated by a 1/2-hp pump inside this coil. The 1000 lb Freon-11 in the vessel was typically maintained at 10°F where a vacuum pump was used to maintain the pressure inside at $\sim 20 \text{ in. Hg}$ for a considerable interval before the test. Thus the Freon was kept cold and prevented from prolonged contact with air or nitrogen. The vessel was well insulated by 2 in. inner layer of asbestos and 8 in. fiberglass material.

3.2.2 Test Section

The test section, shown isometrically in Fig. 3.3, consisted of a 24 in. 0.752 in. ID, 0.805 in. OD, directly heated, Type 304 stainless steel tube, silver brazed at both ends to two 14 in. long copper leads of low electrical resistance. The circumferential variation of the SS-304 tube thickness measured at both ends using a tubing micrometer is $\pm 0.001 \text{ in.}$ Hence radial variation in heat flux was minimal. The resistance of this 0.0265 in. wall tube was calculated to be 0.0103 ohm which agreed closely with Kelvin bridge measurement of 0.0104 ohm. Dimensional information is listed in Table 3.1.

Fig. 3.2. Schematic Showing the Refrigeration Unit and the Receiving Vessel.

Fig. 3.3. Transient CHF Test Section
Isometric.

TABLE 3.1 Dimensions of Test Section

Stainless-steel heater OD	0.805 in.
Stainless-steel heater ID	0.752 in.
Copper tube OD	0.805 in.
Copper tube ID	0.625 in.
Outer-wall pyrex OD	1.344 in.
Outer-wall pyrex ID	1.022 in.
Heated length	23.5 in.
Heated length L/D ratio	108
Calming length	6.0 in.
Calming length L/D ratio	28
Hydraulic diameter	0.217 in.
Heated equivalent diameter	0.492 in.
Annular flow area	0.311 in. ²
Annular section volume	11.210 in. ³
Upper plenum volume	29.360 in. ³
Lower plenum volume	29.360 in. ³

The outer 3 ft transparent pyrex pipe had five 1/8-in. holes drilled at 6-in. intervals for differential pressure measurements. Copper clamps with viton gaskets tightly fitting over the pyrex pipe were used to secure the pressure taps and prevent leakage of fluid at the test pressure. The pyrex pipe was made with flared conical ends to take a specially designed metal flange, which is cushioned from the glass by a molded asbestos insert. The flat SS-304 surface of the plenum had been tap drilled at the proper locations to take three 5/16-in. flange bolts. A teflon gasket is gripped between the grooved pipe end and the plenum surface when the flange bolts are tightened.

To insure the heated tube was properly centered with respect to the pyrex pipe, the following assembling procedure is employed. Two carefully machined brass collars were made to slip fit in the annular channel at both ends. Two rods with diameters smaller than the annular gap were attached to each collar so that the collar can be pushed into or pulled out of the channel. Flow straighteners, as shown in Fig. 3.3, were made to center the heated tube with respect to the plenums. First, the two collars are pushed into the annular gap at both ends followed by the flow straighteners. This resulted in the pyrex pipe being properly centered with respect to the two plenums and the inner tube so that the flange bolts at both ends could be tightened. The flow straighteners and the collars were slid out of the assembly and the former put back in the original position inside the plenums.

Twenty-two 0.032-in. OD ungrounded sheathed thermocouples Type K (chromel-alumel) were spot-welded onto the inside wall of the SS tube, see Fig. 3.4. Nine of these were used for CHF detection by means of temperature controllers. The rest were recorded as primary data.

Fig. 3.4. Thermocouple Locations.

Connections are made to flexible leads where the sheathed thermocouples emerge from the ends of the copper leads. Two 0.0625-in. OD, ungrounded sheathed, Type-K thermocouples were inserted through the plenums into each end of the annular channel to measure bulk fluid temperatures.

Void fraction in the annular channel is evaluated by measuring the electrical capacitance between the heater tube and an electrode wrapped around the outer surface of the pyrex tube. The electrode was an 0.001-in. thick, stainless steel foil, 1-in. wide, which surrounded the tube completely. There are two such electrodes located at 3-in. upstream and downstream of the heated region as shown in Fig. 3.3. The electronic circuit and calibration technique are presented in Appendix E.

To maintain the lowest possible resistance at bus bar connections, the contact surfaces were coated with a layer of silver by electroplating. The test section was isolated electrically from the rest of the piping and tubing by flanges with viton gaskets wherever appropriate.

3.2.3 Instrumentation

Power to the test section was supplied by a 20 V, 1500 amp, DC power supply (Model 815 rectifier manufactured by Crown Rheostat). To protect the test section from physical burnout and to avoid chemical decomposition of Freon-11, all the nine temperature controllers were set to trip the test section power at a heated wall temperature of 300°F. The response time of the controller (manufactured by Weather Measure, TPC-1 Type K) as measured by the time lag between the input and output signals, was found to be about 300 msec. If the heated tube was suddenly insulated, an average temperature rise of $\sim 90^\circ\text{F}/\text{sec}$ at a heat flux of $45,000\text{ Btu/hr-ft}^2$ would be experienced. Thus it was felt quite safe to run power tests without causing physical failure of the test section.

The test section power was obtained from two measurements: voltage drop across the test section and voltage drop across an on-line shunt rated at 50 mv/2000 amp.

A summary list of instruments is shown in Table 3.2 together with the corresponding recording devices.

3.3 Experimental Procedure

3.3.1 Calibration

Calibration of pressure transducers, weight transducers, and turbine flowmeters was performed at an early stage before any tests were conducted in the loop. Later calibration showed that the drift was insignificant. The calibration procedure together with the calibration curves is presented in Appendix I.

The drift in the amplifier's gain however was found to be significant. These were used to amplify transducer and thermocouple signals to the required input level for visicorder galvanometers. To minimize the effect of such drift, they were calibrated by inputting a known millivolt signal from a standard DC voltage supply just prior to each test.

3.3.2 Refinishing Operation

This operation refers to the filling up of the supply vessel with Freon-11 from the receiving vessel. First a vacuum was pulled on the supply vessel through the line at its top while the vent on the receiving vessel was opened to atmosphere. The two valves on the bottom line of the loop in Fig. 3.1 were opened to allow Freon flow into the supply vessel. At the same time the Freon was allowed to pass through a coarse copper gauge filter and then a filter dryer assembly, similar to the one used in the AECL Chalk River Freon Loop.¹¹¹

TABLE 3.2 Transient CHF Test Instrumentation

<u>Parameter</u>	<u>Instrument</u>	<u>Recorder</u>
Wall temperature	13 chromel-alumel thermocouples	Visicorder Honeywell, Model 1508
Fluid temperature	4 chromel-alumel thermocouples	Honeywell strip chart
Test section pressure	2 strain gauge Statham transducers	Visicorder
Blowdown vessel pressure	Statham transducer	Honeywell strip chart
Receiving vessel pressure	Statham transducer	Visicorder
Test section pressure drop	6 strain gauge Statham and CEC DP cells with 5-15 psid	Visicorder
Test section flow	Inlet and outlet Cox turbine flowmeters (bidirectional) 2.5-60 gpm used with 2 Cox flow converters	Honeywell strip chart
	Weight transducers on supply vessel and blowdown vessel; both Statham unbonded type strain gauge	Hewlett-Packard strip chart
Void fraction	Capacitance void probes with detection circuit	Honeywell strip chart
Test section power	50 mv/2000 amp shunt	Honeywell strip chart
Photographic measurement	Hycam camera	800-1000 fps Kodak Ektachrome film

3.3.3 Heatup Operation

The temperature controller for the immersion heater in the ~85% full supply tank was first set at 100°F with the heater on. When the controller started to operate, the vent was opened slightly to bleed off any dissolved gas which had come out of the bulk solution for 1 hr. Dissolved gases had been found to play a significant role in two-phase flow heat transfer^{109,112} and the common degassing procedure used herein was to boil the system at its saturation temperature.^{108,109} After degassing in a two-phase state, the vent was closed, and the controller was set at the higher operating temperature (~135°F). A 4/5 full vessel of Freon-11 would attain stable and uniform temperature as indicated by the four immersed thermocouples after 1-1/2 hr.

AC power to the blowdown vessel heating wire was turned on to bring the vessel wall up to operating temperature while the vessel was still empty. Nichrome heater wires were also used for heating the loop piping to the desired temperature.

3.3.4 Blowdown Operation

The following checkout procedure was performed prior to each run:

1. Air at 45 psig was supplied to the three air-operated solenoid valves, SW1, SW2, and SW3 shown in Fig. 3.1. Valves SW1 and SW3 were opened by a manually operated switch, while SW2 remained closed.
2. The loop was purged with hot Freon and the pressure transducers were bled for trapped gases.
3. Various readout devices were checked for proper recording conditions and photographic equipment was made ready.

4. The supply vessel was pressurized by nitrogen, resulting in subcooled Freon.
5. Single-phase upflow in the test section was established by opening the flow control valves.
6. Desired power level to the test section was turned on and the flow and inlet pressure were set to the desired value by adjusting the flow control valves and the throttle-line valve located just upstream of valve SW1.
7. After establishing steady-state upflow with subcooled boiling, and maintaining these conditions for ~1 min to insure steady loop component temperatures, all recording and photographic systems were actuated.
8. The manually operated switch in (1) was then quickly turned to the opposite position to initiate the blowdown transient. This caused solenoid valves SW1 and SW3 to close and SW2 to open, thus closing the path from the top of the test section to the receiving vessel, shutting off the supply vessel from the test section and opening a path to the receiving vessel.

CHAPTER IV

EXPERIMENTAL RESULTS

The data from two adiabatic and six diabatic (i.e., with power input to the test section) blowdown tests are reported in this study. The initial steady-state conditions prior to blowdown are indicated in the figures showing the pressure decay curves, and also in Table 4.1 for the diabatic tests. The original analog data signals however are not presented due to the enormous number of traces involved.

4.1 Adiabatic Blowdown Tests

A number of adiabatic blowdown tests were conducted with Freon at nominal conditions of 65 psia and 145°F. They were performed by establishing a steady upflow and then initiating a decompression which produced flow reversal within the system. Two typical blowdown transients are presented in Figs. 4.1 and 4.2. The shortest transient was obtained with the blowdown vessel isolated from the loop and the system required about 15 sec to reach ambient pressure. For the long transient, Run 4/9-B, the fluid for blowdown was at two different initial temperatures. The stagnant fluid stored in the blowdown vessel was at a lower temperature (138°F) than the flowing liquid in the turbine legs and test section (151°F).

4.2 Preliminary Diabatic Blowdown Tests

Three diabatic blowdown runs are presented here. They were performed in order to check system behavior under various conditions. The main features of each run are described below with the corresponding initial conditions summarized in Table 4.1.

Run 4/22--For this particular run, the transient was initiated with solenoid valves SW1 and SW3 closed about 1 sec before SW2 opened. As

Fig. 4.1a. Pressure and Void Measurement for Adiabatic Blowdown Run 4/4A.

(Blowdown Volume ≈ 0.08 ft³.)

Fig. 4.1b. Fluid Temperature During Blowdown Run 4/4-A.

Fig. 4.2a. Pressure and Weight Measurement for Adiabatic Blowdown Run 4/9B.

(Blowdown Volume $\approx 0.582 \text{ ft}^3$.)

Fig. 4.2b. Void Fraction and Turbine Flow-meter Measurement for Adiabatic Blowdown, Run 4/9B.

TABLE 4.1 Initial Steady-State Conditions in Test Section Prior to Blowdown

Run No.	Pressure P_o (psia)	Temperature T_o (°F)	Subcooling ΔT_{sub} (°F)	Velocity in TS U_o (ft/sec)	Mass Velocity G (lbm/hr-ft ²)	Heat Flux ϕ (Btu/hr-ft ²)	Blowdown* Volume V_{BD} (ft ³)
4/22	67.5	127.1	40.8	3.28	1.04×10^6	38,500	0.080
5/19-A	65.7	132.8	33.2	8.05	2.53×10^6	46,340	0.072
5/29	66.0	132.8	33.5	7.54	2.37×10^6	45,520	0.318
6/4	64.8	132.8	32.2	7.54	2.37×10^6	45,460	0.318
6/9	64.5	133.9	30.7	7.54	2.37×10^6	46,300	0.201
6/12	65.1	133.3	32.0	7.54	2.37×10^6	46,570	0.072

*Blowdown volume is defined to be the system volume between lower plenum and SW3 valve.

a result, the fluid in the test section was stagnant and because of the continuous energy input, the pressure in the test section began to rise, as shown in Fig. 4.3a, until decompression occurred on opening valve SW2. Figure 4.3b shows the wall temperature as well as fluid temperature. Time zero is taken to be the start of depressurization. The time at power trip is also indicated on the figures.

Run 5/19-A--Like Run 4/22, this run was performed with the blowdown vessel removed from the loop, and replaced by a flexible SS tube as shown in Fig. 3.1. Subcooled flow boiling was visually observed all along the heated length during steady-state operation. Time zero for the transient is again taken to be the instant at which the plenum pressures start to drop. Temperature and void measurements are shown in Figs. 4.4b and 4.4c. Figure 4.4d shows the time to CHF, t_{CHF} at various locations along the heated section. The inception of CHF is defined as

$$t_{CHF} = t_{TC} - \tau_{TS} \quad (4.1)$$

where t_{TC} is the point at which the thermocouple begins to rise and τ_{TS} is the overall time response of the heated tube and thermocouples. The latter is determined by separate test as described in Appendix D.

Run 5/29--This run was performed with the full volume of the blowdown vessel, which produced a transient lasting almost 1 min. The experimental results are shown in Figs. 4.5a-g. Heater wall temperatures on the bottom 8 in. rose without power trip with the most severe rise at the extreme end. Rewetting of this dried-out region was observed as the wall temperature at this location dropped before any physical damage was done on the heater surface. Power trip subsequently occurred

Fig. 4.3a. Pressure History for Diabatic Blowdown, Run 4/22.

(Blowdown Volume ≈ 0.08 ft³.)

Fig. 4.3b. Temperature History for Run 4/22.

Fig. 4.3c. Turbine Flow Measurement for Run 4/22.

Fig. 4.3d. Void Fraction Measurement for Run 4/22.

Fig. 4.4a. Pressure History for Diabatic Blowdown, Run 5/19A (Blowdown Volume ≈ 0.072 ft³).

Fig. 4.4b. Temperature History for Run 5/19A.

Fig. 4.4c. Void Fraction Measurement for Run 5/19A.

Fig. 4.43. Time to CHF versus Axial Location for Run 5/19A.

Fig. 4.5b. Temperature History for Run 5/29.

Fig. 4.5c. Void Fraction Measurement for Run 5/29.

Fig. 4.5d. Flow Measurement during Blowdown
for Run 5/29.

Fig. 4.5e. Mass in Blowdown Vessel for
Run 5/29.

Fig. 4.5f. Time to CHF versus Axial Location for Run 5/29.

Fig. 4.5g. Onset of Rewetting During Run 5/29.

at ~40 sec as the whole test section went into CHF condition. Figure 4.5e shows that the blowdown vessel was essentially empty at 40 sec.

4.3 Diabatic Blowdown with High Speed Photography

The following tests were performed after the following minor change. It was noticed that during the preliminary diabatic blowdown tests (Runs 4/22, 5/19-A, and 5/29), the three air-operated solenoid valves did not close or open at the same time. To minimize this effect, small globe valves were used to meter the air flow to individual solenoid valves. This allowed a finer tuning of the system response and produced repeatable and satisfactory operation.

Three tests were conducted with high speed photography (800-1000 frames/sec) taken at the bottom 6 in. of the heated section. A neon lamp was used as a coincidence mark on both the pictures and the visicorder chart so that correlation in time could be made between recording traces and the high speed photos. This series of tests (Runs 6/4, 6/9, and 6/12) was aimed at studying the effect of blowdown volume on time to CHF. Hence the initial steady-state conditions were maintained unchanged as shown in Table 4.1. The results are shown in Figs. 4.6, 4.7, and 4.8. The pressure profile obtained from DP cell measurement for Run 6/9 is shown in Fig. 4.7f. Some pressure traces were lost in the other two runs; the signals did not appear on the visicorder paper. High speed photographic results are also shown as selected prints at various time into blowdown in Figs. 4.9a-c. The transient coolant velocities, obtained with a motion picture analyzer, are presented in Figs. 4.10a-c. The velocity shown is more indicative of the liquid phase when significant slip occurs as in annular flow. For all these runs, time to CHF was short, ~400 msec as shown in Table 4.2. Similar to Run 5/29, temperature excursion was observed only at the bottom 8-10 in.

2

Fig. 4.6a. Pressure History for Diabatic Blowdown, Run 6/4.

(Blowdown Volume $\approx 0.318 \text{ ft}^3$.)

Fig. 4.6b. Temperature History for Run 6/4.

Fig. 4.6c. Void Fraction Measurement for Run 6/4.

Fig. 6.6d. Turbine Flow Measurement for
Run 6/6.

Fig. 4.6c. Mass in Blowdown Vessel for Run 6/4.

Fig. 4.6f. Pressure Differential Measurement for Run 6/4.

Fig. 4.7a. Pressure History for Diatomic Blowdown, Run 6/9.
 (Blowdown Volume $\approx 0.201 \text{ ft}^3$)

Fig. 4.7b. Temperature History for Run 6/9.

Fig. 4.7c. Void Fraction Measurement for Run 6/9.

Fig. 4.7d. Turbine Flow Measurement for Run 6/9.

Fig. 4.7e. Mass in Blowdown Vessel for Run 6/9.

Fig. 4.7f. Pressure Differential Measurement for Run 6/9.

Fig. 4.7g. Axial Pressure Profile During Diabatic Blowdown, Run 6/9.

Fig. 4.8a. Pressure History for Run 6/12.
(Blowdown Volume ≈ 0.072 ft³)

Fig. 4.8b. Temperature History for Run 6/12.

Fig. 4.8c. Void Fraction Measurement for
Run 6/12.

FIG. 4.8d. Turbine Flow Measurement for Run 6/12.

Fig. 4.8e. Pressure Differential Measurement for Run 6/12.

TABLE 4.2 Time to CHF^a (t_{CHF}) for Diabatic BlowdownTime to CHF (sec)

Run No.	Thermocouple Location (in.)					
	1	2	4	6	8	10
6/4	0.4	0.5	0.4	0.5	0.4	0.5
6/9	0.3	0.5	0.4	0.5	0.4	--
6/12	0.3	0.4	0.4	0.2	0.2	0.3

a t_{CHF} is defined by Eq. 4.1

b -- indicates no CHF

SELECTED PRINTS FOR RUN 6/4

Fig. 4.9.1.

SELECTED PRINTS FOR RUN 6/9

Fig. 4.9b.

SELECTED PRINTS FOR RUN 6/12

Fig. 4.9c.

FIG. 4.10a. Velocity observed in High Speed
Photography, Run 6/4.

Fig. 4.10b. Velocity Observed in High Speed Photography, Run 6/9.

Fig. 4.10c. Velocity Observed in High Speed Photography, Run 6/12.

CHAPTER V

DISCUSSION

The experimental results presented in Chapter IV are discussed in the present chapter. The critical flow and critical heat flux calculations accompanying the discussion are detailed in Appendices K and L. The discussion is divided into the adiabatic blowdowns, the effect of dissolved gas, and the diabatic blowdowns.

5.1 Adiabatic Blowdown

Run 4/4-A—Figure 4.1a shows the inlet and exit void as measured by the capacitance probes for this short transient. The present interpretation of void fraction from capacitance measurement is simply based on the slug flow model (see Appendix E). A more advanced calibration will be undertaken to determine the effect of flow patterns on void measurement but will not be a consideration in this report. The calculated void, assuming an isentropic expansion based on the measured pressure history with no slip between phases (see Appendix F) is also plotted for comparison. Interestingly enough, the initial peak indicated by the equilibrium calculation was not experimentally measured.

During the initial stage of the unstable growth of a bubble, the inertia of the surrounding liquid and the surface tension control the process,⁷ however, this period is very short compared to the thermally controlled stage for the Freon-11 system under investigation. The latter is limited by the rate of thermal conduction to the bubble liquid-vapor interface. The above observed discrepancy is attributed to the nonequilibrium conditions required for unstable bubble growth.¹⁰²

The measured and calculated void in the lower test section reached 50% void at ~2.5 sec, at which time the pressure started to drop and

the bulk fluid temperature agreed with the saturation temperature predicted from pressure measurement (Fig. 4.1b). At such a void fraction, transition to slug-annular flow was evident from the high speed movies, resulting in a large increase in the liquid-vapor interfacial area. Thus the liquid configuration changed from vapor bubbles growing in a superheated liquid toward a dispersed liquid-vapor mixture which closely approached thermodynamic equilibrium. In other words, the liquid mass or droplet surface temperature were essentially in equilibrium with the vapor because of the large interfacial area available for interphase mass transfer. Agreement between measured and calculated void fraction is encouraging in this region. Because of the present interpretation, the measured void can be overestimated by as much as 15% in the annular flow (high void fraction) regime (Appendix E, Fig. E.9). However, the calculated void also overestimates because the actual slip ratio is greater than unity in dispersed flow.

Figure 4.1b shows that after the very short period of subcooled decompression, the fluid was superheated during the pressure recovery period. That is, the system pressure had dropped below the local saturation pressure; and the recovery rate is determined by the relative rate of vapor generation and discharge choked flow. A bubble growth model was suggested to describe this period very well.¹⁰² The saturation blowdown which follows lasted until 8 sec at which time the fluid temperature seems to indicate significant subcooling. Physically this is not possible. Since the flow is adiabatic, the generation of a two-phase mixture, at a pressure greater than the saturation pressure, cannot be due to subcooled boiling.¹¹³ Rather nitrogen gas coming out of solution is believed to occur. The continued lowering in system pressure

facilitates the diffusion of the dissolved gas towards the liquid-vapor interface and its subsequent evaporation. As will be mentioned in Sect. 5.2, the contact time between Freon and nitrogen inside the supply vessel at 130°F and 60 psig has to be limited to a very short period (~ 10 min) in order to eliminate appreciable dissolution of nitrogen. However this precaution was not observed for the adiabatic and the first two reported diabatic blowdown tests (Runs 4/22 and 5/19A).

Run 4/9-B--For this longer transient, the initial peak in void indicated by the equilibrium calculation was again not measured by the void probe (Fig. 4.2b), and the same reasoning applies here.

As mentioned in Sect. 4.1, the blowdown fluid was initially at two different temperatures. In this earlier version of the test loop design,¹¹⁴ the blowdown vessel had only one flexible tube joining the flowing loop and as a result, the fluid had to be stored in a stagnant condition. Attempting to maintain the same fluid temperature inside the blowdown vessel and the rest of the flowing loop was difficult. This version of the facility was subsequently modified to the present design, as shown in Fig. 3.1 after this test. Continuous fluid flow inside the blowdown vessel was allowed during the initial steady-state operation.

The early rise in the measured void resulted from the flashing of hotter fluid (151°F) in the test section and turbine legs. Figure 4.2b shows that this rise was well predicted by the equilibrium calculation. The void fraction dropped when the colder fluid (138°F) from the blowdown vessel entered the test section. The later rise in void fraction development was again well predicted by an equilibrium calculation, based on the temperature of the fluid in the blowdown vessel.

The lower turbine meter in Fig. 4.2b showed a trend similar to the lower void probe. If the meter is indicative of the liquid velocity for a two-phase mixture, then a knowledge of the mass flow rate during blowdown enables one to calculate the local void fraction, as described in Appendix G. Figure 4.2a shows that the mass flow can be found by differentiating the mass decay curve. The void fraction at the turbine estimated using the above mentioned approach was over 75% during the period 5-20 sec. This is considerably higher than the measured values within the test section because choking is thought to have existed at the piping immediately downstream of the lower plenum. The associated pressure drop resulted in additional flashing before the two-phase fluid reached the turbine meter.

5.2 Effect of Dissolved Gas

With subcooled boiling before blowdown, the vapor will condense as it leaves the heated surface and comes into contact with the bulk fluid. But the presence of a noncondensable gas has been demonstrated to prolong, and in the present case eliminate, the collapse of subcooled bubbles.^{109,115} Dissolved gases have been found to affect two-phase flow heat transfer.^{112,116,117} The presence of gas increases the heat transfer in subcooled boiling.⁷ Gas bubbles can form at the surface and agitate the liquid in the same manner that vapor bubbles do. The first evolution of bubbles was seen to occur at a much lower wall superheat in the presence of dissolved gas for a water-air system.¹¹⁸ In fact, bubble evolution had been observed during steady adiabatic flow in the present system while the test loop pressure was still above the saturation value. This phenomenon stimulated the following investigation.

During each run the Freon in the supply vessel was pressurized by nitrogen, thus the rate of nitrogen going into solution had to be examined. The effect of 1 hr contact interval between nitrogen and Freon in the supply vessel at 130°F and 60 psig is shown in Table 5.1. It is apparent that nitrogen was exiting from solution in the heated zone since the exit turbine meter showed a much higher flow than the inlet one under all liquid equilibrium conditions, indicating the presence of an uncondensed two-phase mixture. Therefore the following precaution was taken to ensure the dissolved gas was kept to a minimum. First the Freon was stored in the receiving vessel under saturation conditions at ~20 in. Hg so that the Freon was not in contact with air during the storage period. Secondly the Freon was boiled at or above ambient pressure in the supply vessel for 1/2 to 1 hr while constantly vented to atmosphere, thus bleeding off any noncondensable gas coming out of solution before each run.^{108,109} Thirdly, the contact time with nitrogen at operating pressure was kept to a minimum, normally 10 min before each run. The agreement between the inlet and exit flowmeter measurements for this case as illustrated in Table 5.1 demonstrates the success of these measures.

5.3 Diabatic Blowdown

5.3.1 Preliminary Diabatic Tests

Run 4/22--This test was conducted with an almost 1 sec inlet flow stoppage before blowdown initiation. However, efficient cooling was maintained all during this very low flow period, as indicated by the heater wall temperature. During the inlet flow stoppage, if the fluid in the heated section remained close to saturation in liquid phase only, the calculated temperature rise of the average bulk fluid is about

TABLE 5.1. Effect of Prolonged Nitrogen Contact with
Freon-11 (130 F) at Elevated Pressure (60 psig)

Heat Flux, Btu/hr-ft ²	Contact Time 10 min				Contact Time 1 hr			
	0		45,600		0		45,600	
	Inlet	Exit	Inlet	Exit	Inlet	Exit	Inlet	Exit
Turbine rotation speed, cps	188.0	190.0	188.8	193.0	182.8	183.0	187.8	190.0
Volumetric flow, cfm	0.981	0.992	0.984	1.007	0.983	0.988	0.979	1.014
Mass flow, lbm/sec	1.435	1.451	1.439	1.445	1.399	1.403	1.437	1.481
Agreement in mass flow, %	1.1		0.4		0.5		34	
Measured Void ^a	-	-	-	0.16	-	-	-	0.48
Inferred Void ^b	-	-	-	0.0	-	-	-	0.34

^aCapacitance void measurement located 3 in. downstream of heated section

^bCalculated void from turbine measurement according to $\alpha_o = 1 - (\rho_l Q_l) / (\rho_o Q_o)$, where subscripts *l* and *o* stand for inlet and outlet locations.

40°F/sec, and if any part of the heater surface was experiencing CHF, a temperature excursion of 80°F/sec would result. The measured temperature rise of 7°F indicates that CHF did not occur and the fluid did not remain in an all liquid state. But part of the heat was utilized in raising the bulk temperature of the fluid and the rest as latent heat of vaporization. The observed pressure rise in the test section resulted in an increase of saturation temperature of about 8°F, which lends support to the above argument. Vapor generation was evident from the upper void probe which indicated continuous rise to almost 100% void just before depressurization. The existence of countercurrent flow during the low flow period was inferred.

The above observation tends to support the fact that heat flux at low flow can be quite high without experiencing CHF as in pool boiling.^{85,87} For a horizontal flat plate in an unconfined pool boiling, Zuber¹¹⁹ suggested the following critical heat flux correlation,

$$\phi_{CHF} = \frac{\pi}{24} h_{fg} \rho_f^{1/2} \left[\sigma g_c (\rho_f - \rho_g) \right]^{1/4} \left(\frac{\rho_f}{\rho_f + \rho_g} \right)^{1/2} \quad (5.1)$$

where for

Freon-11 at 65 psia

Water at 1500 psia

$$\phi_{CHF} = 1.2 \times 10^5 \text{ Btu/hr-ft}^2$$

$$\phi_{CHF} = 1.2 \times 10^6 \text{ Btu/hr-ft}^2$$

As shown above this heat flux is at least three times larger than the operating heat flux. Even though the above equation is not applicable to the present confined geometry, Walkush¹²⁰ and Avedisian et al.⁸⁹ have shown that a value of 85-90% of Zuber's prediction can be expected at low flow and low void fraction in an internally heated annulus with small L/D.

The heater wall temperature shown in Fig. 4.3b indicates that CHF occurred at about 2 sec into blowdown, at which time the upper void measurement had a sharp rise to 100%, indicative of fluid depletion condition. Many wall thermocouples were found to be defective due to improper installation and were therefore replaced. The most severe temperature rises were found at the lower end where CHF seemed to occur first although an equally early CHF onset was found at TC 18 (6 in. from top) but this was rewet after ~ 1 sec.

Runs 5/19A and 5/29--For these runs and all others, no CHF was observed upon flow reversal. The flow reversal period is defined as the time from the initiation of decompression until the start of downflow. This reversal period was observed to last 50-60 msec in the three runs taken with high speed photography. Efficient cooling was maintained during this time as the heater wall thermocouples showed a slight drop in temperature.

Similar to Run 4/22, the start of temperature excursion in Run 5/19A coincided with the sharp rise in upper channel void development, Fig. 4.4c. Void fraction of over 90% indicated that the channel was nearly depleted of liquid coolant and as a result 'dryout' occurred. The propagation of dryout front for Run 5/19A is quite smooth as shown in Fig. 4.4d.

In Run 5/19A, there appeared a very small temperature excursion at TC 1 (1 in. from the lower end) shortly after flow reversal but was rewet in ~ 1 sec. In the case of Run 5/29, inception of CHF occurred twice, Fig. 4.5b. The early temperature excursion was more pronounced as temperatures above 400°F were measured. However only the bottom one-third of the heated zone experienced CHF, and rewetting quickly took place at the 8 in. location and propagated downstream. The time to first CHF was

about the same at all the CHF locations, 400 msec for Run 5/29, as shown in Fig. 4.5f. The occurrence of the second CHF again coincided with the upper void probe measuring close to 100% void. The upper turbine meter also showed a rapid rise, Fig. 4.5d, indicating high quality vapor flow. Further, the mass remaining in the blowdown vessel, as sensed by the weight transducer, was nearly zero at this time. This 'dryout' front was observed to follow the same kind of locus as in Run 5/19A, Fig. 4.5f, but the rate of propagation was faster. Unlike the second CHF, the first CHF requires further discussion under Sect. 5.3.3.

5.3.2 Effect of Bubble Growth in the Decompression Process

The bubble growth rate has been found to affect the initial decompression characteristic as mentioned in Sect. 5.1. This is understandable in view of the fact that the system pressure would drop all the way to the ambient pressure if it were not for the inception of vaporization of the hot liquid. In the case of adiabatic blowdown, nucleation sites in the form of surface cavities are activated as a result of decompression. With Freon-11 at operating temperatures and pressures, the bubble growth rate is essentially limited by the heat conduction to the liquid-vapor interface, consequently the actual pressure must drop below the P_{SAT} value, causing some superheat to occur. Hence the initial rise in calculated equilibrium void fraction at the lower channel was not experimentally observed. This phenomenon of bubble as a result of the thermodynamic nonequilibrium state of the fluid has been reported in other adiabatic blowdown experiments.^{67,102} But for the diabatic blowdowns, where subcooled bubbles were present initially, thermodynamic equilibrium was more closely approached during the early phase of the blowdown. The different characteristics

associated with adiabatic and diabatic blowdown are best illustrated by showing the system pressure and void fraction as in Fig. 5.1a and b. For the diabatic case, the system pressure dropped to the saturation pressure corresponding to the exit fluid temperature in the upper plenum, while for the adiabatic case, an undershoot in pressure was observed. The immediate jump to 90% void in the lower channel was characteristic of the diabatic case. This was due to the large vapor generation during flow reversal accompanying the depressurization. Thermodynamic equilibrium void was estimated at the upper channel for the diabatic blowdown. The isentropic expansion was thought to hold since the fluid underwent decompression and flow reversal under adiabatic conditions because the upper turbine leg and upper plenum were well insulated. The prediction was encouraging in view of the fact that a slug flow model had been used in the interpretation of the measurement. The success of this prediction is further demonstrated in the diabatic blowdown, Run 6/12 (Fig. 4.8c) where the very early void development was well followed by the calculation. This was expected if the system approached thermodynamic equilibrium. The upper void probe (located 3 in. above the heated zone) indicated the presence of subcooled bubbles in the channel exit during steady-state operation. However these bubbles were soon condensed before reaching the upper turbine location, most probably in the upper plenum where more mixing occurred. The existing subcooled bubbles in the upper test section channel therefore acted as nucleation sites, where interphase mass transfer took place during the early phase of blowdown.

5.3.3 Diabatic Tests with High Speed Photography

The results of these three blowdown tests are presented in Figs. 4.6, 4.7, and 4.8. Selected prints showing the flow regime

Fig. 5.1a. Pressure History for Adiabatic and Diabatic Blowdowns.

Fig. 5.1b, Void Fraction for Adiabatic and Diabatic Blowdowns.

during the early blowdown phase are presented in Fig. 4.9. This series of tests (Runs 6/4, 6/9, and 6/12) was aimed at studying the effect of blowdown volume on time to CHF.

With the help of high speed photography, the flow reversal period was observed to last 50-60 msec. The upflow decayed as soon as depressurization started, and referring to the selected prints, as the flow slowed down, bubble density on the heating surface increased rapidly. Vigorous vaporization occurred during downflow, as seen from prints at 80 msec and this high void mixture was swept out of the heated channel at an accelerating speed, up to 23 ft/sec, Fig. 4.10. The lower void probe showed an immediate jump to high void fraction, qualitatively in agreement with the high speed photographic results. However this high velocity flow was observed to drop off at around 250 msec, leading to the development of an annular flow pattern, which judging from the pictures, lasted until 500 msec, Fig. 4.9a-c. During this period, considerable slip between liquid and vapor phases were observed; roll waves frequently passed along while a few smaller droplets were seen traveling down the core at a much higher velocity. Wavy liquid films were observed adhering to the outer unheated wall although it is impossible judging from the motion pictures whether a liquid film was also present on the heated wall. According to the thermocouples at the heater wall location, critical heat flux conditions were experienced at around 300 msec (Table 4.2) which coincided with the time of appearance of annular flow. Thus the heater wall where pictures were taken was inferred to be dry, hence the temperature excursion. Considering the uncertainty in determining the time to CHF, it could be concluded that no difference was observed among these three runs with different blowdown volumes.

The temperature excursion in these runs continued until the power tripped. The flow pattern after 500 msec was uncertain when judging from the pictures due to the presence of liquid film on the inner wall of the transparent shroud. With the measured high void (>80%) at the lower end of the channel, dispersed or mist flow is thought to be predominant. The entrained liquid droplets on impinging onto the dry wall could be vaporized but the heat removal is apparently insufficient to cool the heated surface. Rewetting by axial conduction¹²¹ was observed during the preliminary test, Run 3/29, as the liquid rewetting front which existed above the 8-in. location propagated downward (Fig. 4.5g). Rewetting also took place at the 8- and 5-in. locations in Run 6/9 and at the 10-in. location in Run 6/12 before power trip (Figs. 4.7b and 4.8b).

At around 100 msec, the upper plenum pressures all showed spikes. This was observed after the valving adjustment as mentioned in Sect. 4.3. These pressure spikes are perhaps due to the acoustic phenomena associated with the closing of the downstream valve SW3. The rise is due to the compression wave traveling down the upper turbine leg into the test section where a rarefaction wave was sent back causing the pressure to drop and the flow to reverse.

The total pressure drop is the summation of the momentum, frictional, and hydrostatic pressure drop.⁷

$$\Delta P_T = \Delta P_M + \Delta P_{fric} + \Delta P_z \quad (5.2)$$

Near to the lower end of the annular channel the momentum pressure drop dominates as shown in Fig. 4.7g. For low quality flows ($1 - x \ll 1$) the momentum pressure drop in a constant area channel may be approximated¹¹³

$$\Delta P_M = \frac{Q^2}{\rho_f g_c} \left(\frac{1}{1 - \alpha_2} - \frac{1}{1 - \alpha_1} \right) \quad (5.3)$$

where α_2 and α_1 are the void fraction at axial locations 2 and 1. In the present condition prior to CHF, vigorous vaporization is taking place during the decompression, as indicated by the significant increase in void fraction measured at the lower void probe. As a result the momentum pressure drop increases as the two-phase mixture moves downstream. Hence the pressure profile exhibits an increasing negative slope (Fig. 4.7g), indicative of acceleration. Exception was found during the time interval 300-500 msec when a more linear profile was observed. This coincides with the observance of the annular flow of low velocity, Figs. 4.10a-c.

Because of the steepness of the pressure profiles is quite characteristic of the upstream of a choking location,¹¹³ the possibility of choking cannot be overlooked. The results of the critical flowrate calculation are presented in Appendix K. The calculation was made at 200 msec since the velocity was observed to be highest. The flow decay promoted at about 300 msec was first thought to be caused by choking at the exit of the test section. But the calculation in Appendix K indicates that the flow was quite far away from choking at the lower test section. The choked location was then assumed to be at the lower stainless steel cross where the three outlet tubes from the lower plenum merged. It had a cross-sectional flow area of 0.0256 in.², ~2% smaller than the annular area of the test section. The homogeneous, nonequilibrium compressible flow equation can be approximated for saturated liquid in terms of the critical pressure ratio η as¹²²

$$G = \left[2\rho_f P(1 - \eta) \right]^{1/2} \quad (5.4)$$

where P refers to the stagnation pressure. The fluid in the lower plenum was discharged first at subcooled condition during blowdown. The critical pressure ratio is about 0.8 at zero quality.¹²² Employing Eq. 5.4, a volumetric discharge rate of 0.069 ft³/sec was calculated at the throat, i.e., the cross. This is compared with the vapor generation rate in the heated channel as given by

$$Q_v = \frac{\phi \pi D L_h}{h_{fg}} (v_g - v_f) \quad (5.5)$$

A value of 0.066 ft³/sec was calculated alone using Eq. 5.5, neglecting the effect of flashing caused by pressure drop and possible condensation in the lower plenum. The actual vapor generation rate was higher and in fact exceeded the discharge rate at the throat. Consequently the system pressure was observed to rise. This has also been reported by Hicken et al.⁶¹ in the small break blowdowns. The rise in lower plenum pressure then caused flow retardation observed at 300 msec. The accompanying velocity decay was thought to cause the CHF onset in the lower end of the heated section. The local occurrence of CHF then cut off the heat transfer to the fluid and hence the total vapor generation rate was lowered. Subsequently the velocity picked up, as observed in the high speed photography, because of the reduced vapor production and the increasing volumetric flowrate at the limiting area.

During the initial steady-state upflow, the enthalpy profile would increase linearly from bottom ($h = h_i$) to top ($h = h_i + \Delta h$).

The enthalpy at the top would be slightly higher during the upflow decay period after blowdown initiation. Assuming a linear mass decay rate during flow reversal, the exit enthalpy at stagnation would only be $\Delta h/6$ higher, about 2% of the local enthalpy. This enthalpy peak is calculated to reach the lower heated zone in about 1 sec based on the integrated velocity curve and measured void, but CHF had occurred much earlier in time.

Estimation of critical heat fluxes is presented in Appendix L. The results however have to be regarded with caution since the CHF correlation employed was originally developed at a higher pressure range (in terms of density ratio). Further the correlation is used in terms of inlet subcooling or quality and not of the local parameters. Table L.1 shows a number of cases. The critical heat flux ratio (CHFR, also known as DNBR) was 1.75 during steady-state operation (case 1). The occurrence of the early CHF at ~300 msec was at a CHFR of 0.91, which is well within the $\pm 15\%$ accuracy of the correlation (case 6). In Run 5/29 where the early CHF was rewet, a higher mass velocity was measured by the weight transducer at 3 sec in comparison with Runs 6/4 and 6/9 where no complete rewetting occurred (case 2 versus case 5). The corresponding CHFRs were 1.01 and 0.94 respectively. A higher mass flow caused the rewetting in Run 5/29. After rewetting, CHF was predicted with a CHFR of 0.91 at 20 sec (case 3) but was not observed. Most probably the inaccuracy of the correlation increases with decrease in pressure and that the density ratio decreases sharply with pressure below 50 psia as shown in Fig. J.1. The late (second) CHF occurred at a CHFR of 0.79 only (Case 4). Hence it is unfair to conclude from the above discussion that steady-state CHF correlation is conservative when employed in transient situation because of the aforementioned difficulties.

CHAPTER VI

SUMMARY AND CONCLUSIONS

Experimental data are presented for the blowdown of a small-scale facility using Freon-11 at 130°F and 65 psia. A transparent annular test section with its inner tube heated was used to study the onset of critical heat flux during blowdown with flow reversal. Flow regime transition was observed with high speed photography.

An analysis of the experimental data leads to the following conclusions.

1. A 1-hr contact time between Freon-11 and nitrogen at 130°F and 60 psig was found to greatly affect the steady-state subcooled boiling initial upflow conditions. Nitrogen gas was found exiting from solution in the heated zone. Special precaution had to be taken to ensure the dissolved gas was kept to a minimum.
2. Delay in bubble growth was observed in the adiabatic blowdown. This was caused by the nonequilibrium conditions required for unstable bubble growth. For the diabatic runs, thermodynamic equilibrium was more closely approached in the test section during the early phase of blowdown.
3. CHF did not occur immediately during the flow decay in an approximately 60 msec reversal period.
4. The first or early CHF which occurred at about 400 msec was not affected by the blowdown volume. An annular flow pattern (with vapor adjacent to the heated wall and liquid at the unheated wall) appeared at the onset of CHF as best judged by

the photographic results. There was no sign of CHF propagation and its occurrence was restricted to the bottom one-third portion of the heated channel.

5. For some cases the early CHF was rewet and a second CHF was observed at near depletion of liquid in the channel. This strongly suggests a dryout mechanism and the dryout front was found to propagate smoothly upward.

CHAPTER VII

RECOMMENDATIONS FOR FUTURE STUDIES

The following is recommended for further studies in the present facility:

1. The present recording system for void and turbine meter is not adequate due to the slow response nature of the strip chart recorders. Visicorder or magnetic tape recording units will be employed to record all transient signals.
2. The solenoid valves (SW1, SW2, and SW3) will have to be synchronized to avoid nonreproducible results. Rupture disks assembly may offer a good alternative.
3. Pressure measurement downstream of the test section to determine the exact choking plane during blowdown.
4. Difference between inlet and outlet break can be examined. In particular a simultaneous inlet and outlet break (double-ended break) could result in a flow stagnation located in the heated channel.
5. The effect of break area on time to CHF.
6. Other test section geometries can be studied to determine their effect on flow regime and CHF.
7. Test section capable of operating at higher pressures and temperatures can be installed.

APPENDIX A

SINGLE-PHASE FRICTION PRESSURE DROP DATA

Pressure drop and fluid friction associated with the isothermal flow of water at 70°F-190°F through the present concentric annular test section* with radius ratio of 0.787 was obtained by measuring the static pressure drop. Data are presented for laminar, transition, and turbulent flow covering a Reynolds number (based on hydraulic or equivalent diameter) range from 20 to 60,000. An inclined manometer using 1.75 Meriam fluid (M.F.) was used to give pressure drop down to 0.0004 psi. Two 60 in. U-tube manometers employing 1.75 and 2.95 M.F. were mounted vertically to measure pressure differential across the 24 in. heated length. Mass flow rate was determined by the weighing tank method.

The inside diameter d_o of the commercially available 1 in. pyrex pipe was determined more precisely as described below. The pipe was mounted vertically with a stopcock fitted at the bottom end. A piece of copper tubing was made to go through the stopcock at one end and connected to a vertically mounted graduated manometer tube at the other end. Exactly 25 ml of water were delivered into the pipe by a burette and the water level in the manometer was recorded each time. The inside diameter of the pipe was calculated according to

$$d_o = \left(\frac{4}{\pi} \frac{\Delta V}{\Delta h} - d_m^2 \right)^{1/2} \quad (A.1)$$

where

*The first pyrex pipe used in obtaining single phase pressure drop broke during the installation in the loop later. Hence a second pipe was used in obtaining single phase heat transfer and subsequent blowdown tests.

where

d_m = ID of manometer tube (predetermined by volumetric method)

ΔV = volume of water added, in.³

Δh = rise in water level, in.

The following results were obtained:

	d_o (in.)	Standard Deviation (in.)
TS No. 1, first pipe	1.023	0.007
TS No. 2, second pipe	1.022	0.007

Laminar flow velocity profile in annuli has been well known.¹²³

The relationship between the Moody friction factor and Reynolds number is given by

$$f = \frac{64}{Re_h} \quad (A.2)$$

where

$$\phi = \frac{(1 - \kappa)^2}{1 + \kappa^2 + \frac{1 - \kappa^2}{\ln \kappa}}$$

and

$\kappa = d_i/d_o$, diameter or radius ratio

Reynolds number is defined on the basis of hydraulic or equivalent diameter:

$$Re_h \equiv \frac{\rho D_e U}{\mu} \quad (A.3)$$

and

$$D_e = \frac{4 A_x}{P_w} = \frac{4 \frac{\pi}{4} (d_o^2 - d_i^2)}{\pi (d_o + d_i)} = d_o - d_i \quad (A.4)$$

where P_w is the wetted perimeter.

The function Φ has a value of 1.0 at $\kappa = 0$ for a circular tube and a value of 1.5 at $\kappa = 1$, e.g., a parallel plates of infinite width.

Experimentally, the Moody friction factor is evaluated according to¹²⁴

$$f = \frac{2 D_e}{\rho U^2} \left(- \frac{dP}{dz} \right) \quad (A.5)$$

A plot of f versus Re_h is shown in Fig. A.1; the agreement with theory in laminar region is excellent. For the present geometry, Eq. A.2 becomes

$$f = \frac{95.91}{Re_h} \quad (A.6)$$

while the experimental laminar data yielded an average $f.Re_h$ value of 95.93. The equation obtained by Nikuradse¹²⁵ applicable over the entire turbulent Reynolds number range for smooth tubes is also shown,

$$\frac{1}{\sqrt{f}} = 2.0 \log_{10} (Re \sqrt{f}) - 0.8 \quad (A.7)$$

The experimental data clearly shows that for Reynolds number based on hydraulic diameter, the friction data lie considerably above the smooth tube, between 10 to 20%. Similar results were obtained in a recent survey of past literature¹²⁶ and it was concluded that the hydraulic diameter is not the proper length dimension to use in the Reynolds

Fig. A.1. Friction Factor Results Based on Hydraulic Equivalent Reynolds Number.

number to insure both geometric and hydrodynamic similarity between circular tubes and annular ducts. Most recently Jones¹²⁷ proposed a modified Reynolds number, Re^* , which yields the identical friction factor relation as would be obtained for a round tube in laminar flow. That is

$$f = \frac{64}{Re^*} \quad (A.8)$$

This Reynolds number also provides good similarity in fully-developed turbulent flow for smooth rectangular ducts, as demonstrated by Jones. Comparing Eq. A.8 with A.2, the modified Reynolds number is given by

$$Re^* = \frac{Re_h}{\phi} \quad (A.9)$$

A plot of f versus Re^* is shown in Fig. A.2; the success of this method is readily demonstrated. The turbulent data for $Re > 10^4$ yields a mean deviation of 0.9% and a standard deviation of 2.7% about the smooth tube line. All pertinent results are tabulated in Table A.1.

Limited isothermal friction data was obtained for Freon-11 in the second pyrex pipe. Mass flow rate was determined by turbine flowmeters and pressure loss was measured by DP cells. Results are tabulated in Table A.2. Agreement with the circular tube equation based on modified Reynolds number is good, as shown in Fig. A.3.

Fig. A.2. Friction Factor Plot Based on Modified Reynolds Number.

Fig. A.3. Isothermal Freon-11 Friction Data.

TABLE A.1. Frictional Pressure Drop for Water in Test Section No. 1

Run	Temp °F	Mass Flow lbm/sec	Pressure Drop (24 in.) psi	Re _h	Re*	f
1	75.2	0.2269	0.1001	3070	2049	0.04813
2	75.2	0.3002	0.1224	4062	2711	0.03362
3	75.2	0.3210	0.1414	4344	2899	0.03397
4	75.2	0.3386	0.1648	4582	3058	0.03558
5	75.2	0.3716	0.1969	5029	3356	0.03529
6	75.2	0.3900	0.2284	5278	3522	0.03717
7	75.2	0.4226	0.2665	5719	3817	0.03694
8	75.2	0.4522	0.3018	6119	4084	0.03605
9	75.2	0.4702	0.3317	6363	4246	0.03714
10	75.2	0.5082	0.3698	6877	4589	0.03544
11	75.2	0.4083	0.2863	5525	3687	0.04251
12	75.2	0.5605	0.4826	7585	5062	0.03802
13	75.2	0.6450	0.6335	8728	5825	0.03769
14	75.2	0.7682	0.8647	10395	6937	0.03627
15	75.2	0.8868	1.0985	12000	8008	0.03457
16	75.2	0.9671	1.3413	13087	8734	0.03550
17	75.2	1.0550	1.5131	14276	9527	0.03365
18	75.2	1.1040	1.5879	14939	9970	0.03225
19	72.5	0.6160	0.5856	7962	5313	0.03822
20	72.5	0.3712	0.2310	4798	3202	0.04152
21	72.5	0.5076	0.4117	6562	4379	0.03957
22	72.5	0.6925	0.7157	8952	5974	0.03696
23	72.5	0.8065	0.9266	10425	6957	0.03528
24	72.5	0.8850	1.0899	11440	7635	0.03446
25	72.5	0.9542	1.2780	12334	8231	0.03476
26	72.5	1.0225	1.4654	13218	8821	0.03471
27	72.5	1.0846	1.6141	14020	9356	0.03398
28	72.5	1.1737	1.6227	15172	10125	0.02917
29	72.5	1.2020	1.8273	15537	10369	0.03132
30	72.5	1.2987	2.0000	16788	11204	0.02937
31	72.5	1.4084	2.2091	18206	12150	0.02758
32	72.5	1.4368	2.4182	18573	12395	0.02901
33	72.0	0.00177	0.000566	23	15	4.476
34	72.0	0.00453	0.001274	59	39	1.538
35	72.0	0.01146	0.003161	148	99	0.5961
36	72.0	0.02130	0.005898	275	187	0.3220
37	72.0	0.00146	0.000425	19	13	4.939
38	72.0	0.01898	0.005946	245	164	0.4087
39	72.0	0.03438	0.01048	444	297	0.2195
40	72.0	0.05529	0.01628	715	477	0.1319
41	72.0	0.07083	0.02213	916	611	0.1092
42	72.0	0.08594	0.02661	1111	741	0.08924
43	72.0	0.10357	0.03241	1339	893	0.07484
44	72.0	0.11771	0.03681	1522	1015	0.06578
45	72.0	0.13333	0.03962	1724	1150	0.05520
46	72.0	0.15179	0.04456	1962	1309	0.04788
47	82.0	1.4451	1.9767	50960	34010	0.02281
48	82.0	1.5480	2.2355	54190	36160	0.02249

TABLE A.1.(contd.)

Run	Temp °F	Mass Flow lbm/sec	Pressure Drop (24 in.) psi	Re _h	Re*	f
49	82.0	1.1390	1.2711	39300	26230	0.02364
50	82.0	1.2755	1.5663	43700	29160	0.02324
51	82.0	1.4205	1.7933	48320	32250	0.02146
52	82.0	1.5291	2.1612	51650	34470	0.02233
53	82.0	0.9025	0.8380	30270	20200	0.02486
54	82.0	1.5337	2.2468	51090	34090	0.02309
55	185.0	1.5723	2.2328	58190	38840	0.02171
56	184.0	1.5385	2.1658	56550	37740	0.02201
57	183.0	1.4706	2.0349	53690	35830	0.02264
58	182.0	1.4124	1.9216	51220	34190	0.02318
59	181.0	1.3263	1.7378	47780	31890	0.02378
60	180.0	1.2500	1.6105	44730	29850	0.02482
61	179.0	1.2225	1.5043	43430	28980	0.02425
62	178.0	1.1765	1.4193	41480	27690	0.02454
63	177.0	1.1494	1.3505	40240	26850	0.02465
64	176.0	1.1111	1.2634	38620	25770	0.02468
65	175.0	1.0638	1.1572	36710	24500	0.02467
66	174.0	0.9901	1.0581	33920	22640	0.02605
67	173.0	0.9653	1.0154	32830	21910	0.02631
68	172.0	0.9259	0.9290	31280	20870	0.02617
69	171.0	0.8977	0.8666	30110	20090	0.02598
70	170.0	0.8460	0.7957	28180	18810	0.02687
71	169.0	0.8170	0.7481	27010	18030	0.02710
72	168.0	0.7987	0.7034	26220	17500	0.02667
73	167.0	0.7576	0.6678	24680	16470	0.02815
74	166.0	0.7386	0.6358	23890	15940	0.02821
75	165.0	0.7321	0.6179	23510	15690	0.02791
76	164.0	0.6812	0.5414	21720	14500	0.02826
77	163.0	0.6112	0.4550	19350	12920	0.02951
78	162.0	0.5834	0.4159	18340	12240	0.02961
79	161.0	0.5192	0.3415	16210	10820	0.03071
80	160.0	0.4501	0.2777	13950	9310	0.03324
81	159.0	0.4587	0.2755	14110	9420	0.03176
82	153.0	0.4112	0.2447	12550	8473	0.03512
83	155.0	0.5247	0.3528	15640	10440	0.03113
84	154.0	0.5814	0.4198	17200	11480	0.03018
85	154.0	0.6188	0.4791	18310	12220	0.03040

TABLE A.2. Frictional Pressure Drop for Freon-11 in Test Section No. 2

Run	Temp °F	Mass Flow lbm/sec	Pressure Drop psi/ft	Re _h	Re*	f
1	68.7	0.385	0.072	10920	7290	0.0353
2	69.2	0.910	0.306	26050	17380	0.0270
3	69.6	1.601	0.838	45960	30670	0.0238
4	69.2	1.912	1.201	55010	36710	0.0239
5	67.3	2.278	1.546	65400	43640	0.0217

APPENDIX B

SINGLE-PHASE LIQUID HEAT TRANSFER DATA

A simple macroscopic heat balance on the heated test section (Length = L_h) results in the following equation,

$$\phi \pi D_i L_h = W_f C_{pf} [T_f(L_h) - T_{fi}] \quad (B.1)$$

The term on the left is the total heat input q supplied by the dc rectifier, provided heat loss to surrounding is negligible. Figure B.1 shows the heat balance results which were performed with Freon-11 at close to room temperature. In most cases, agreement is within 5%, giving good confidence in the instrumentation involved.

The heat transfer coefficient to single-phase liquid under forced convection is given by

$$h_{fo} = \phi / \Delta T_f \quad (B.2)$$

and

$$\Delta T_f = T_{wo} - T_f(z) \quad (B.3)$$

where ΔT_f is the temperature difference between the heated tube outside surface (annular geometry) and the bulk liquid temperature at a length z from the inlet. Similar to Eq. B.1, $T_f(z)$ is given by

$$T_f(z) = T_{fi} + \frac{\pi D_i z}{W_f C_{pf}} \phi \quad (B.4)$$

The temperature difference between the inside and outside of the heated tube ($T_{wi} - T_{wo}$), is derived in Appendix C and is given by Eq. C.5.

Fig. B.1. Single Phase Heat Balance Results.

The laws governing laminar or turbulent convective heat transfer are well established. For turbulent flow the well-known Dittus-Boelter equation has been found to be satisfactory.

$$\left[\frac{h_{fo} D}{k_f} \right] = 0.023 \left[\frac{\rho U D}{\mu_f} \right]^{0.8} \left[\frac{C_p \mu_f}{k_f} \right]^{0.4} \quad (B.5)$$

However it was mainly developed for tube geometry.

For an annulus, an additional term is included to account for the geometry of the system. Monrad and Pelton¹²⁸ recommended the following equation:

$$\left[\frac{h_{fo} D_e}{k_f} \right] = 0.020 \left[\frac{\rho U D_e}{\mu_f} \right]^{0.8} \left[\frac{C_p \mu_f}{k_f} \right]^{0.33} \left[\frac{D_o}{D_i} \right]^{0.53} \quad (B.6)$$

and is valid for $Re_h > 10^4$.

Wiegand¹²⁹ also suggested a correlation which gives results very close to Eq. B.6. It is interesting to note that for the present geometry, the product $0.020 (D_o/D_i)^{0.53}$ comes out to be 0.023 which is the same constant as in the Dittus-Boelter equation.

The plot of $Nu/Pr^{0.33}$ versus Re_h is shown in Fig. B.2 where agreement is good, although data tends to show discrepancy at higher Reynolds number. The variation of h_{fo} along the test section is thought to be mainly due to inaccurate thermocouple temperature measurement. Possible errors are mentioned at the end of this section.

Finally some isothermal upward flow of Freon at 135°F were performed to get an estimate of the heat loss through the outer pyrex pipe. Results obtained showed a mean value of 125 Btu/hr and an overall heat

Fig. B.2. Single Phase Heat Transfer Results.

transfer coefficient of about 2 Btu/hr-ft² °F which compares favorably with the calculated value 0.93 Btu/hr-ft² °F based on free convection from the pipe. The presence of copper clamps at each pressure tap locations may conduct some heat away, thus enhancing the heat loss.

Temperatures along the heated test section were measured by 0.031 in. OD stainless steel ungrounded, sheathed, chromel-alumel (type K) thermocouples. Some possible errors in temperatures are:

- a. Inhomogeneity of thermocouple wires.
- b. Stray emf pickup in the leads.
- c. Drift of the reference junction units.
- d. Inaccuracy of amplifier units, such as calibration drift and nonlinearity output.
- e. Inaccuracy in measurement of trace on visicorder paper.

A maximum deviation of 3°F in wall thermocouple readings during isothermal flow of Freon has been observed.

APPENDIX C

EVALUATION OF OUTSIDE SURFACE TEMPERATURE

C.1 Steady-State Consideration

In the absence of axial heat conduction the temperature difference between inside and outside of the heated tube can be calculated quite straightforward. The general differential equation for an infinite heat generating cylinder is

$$\frac{d^2\theta}{dr^2} + \frac{1}{r} \frac{d\theta}{dr} + \frac{q'''}{k} = 0 \quad (C.1)$$

where $\theta = \theta(r)$ only. The boundary conditions are

$$\theta(r_2) = T_{wi} \quad (C.2)$$

$$\left. \frac{d\theta}{dr} \right|_{r_2} = 0 \quad (C.3)$$

where T_{wi} values are known from thermocouples readings. The second condition is valid in view of the fact that the inside tube was packed with magnesium oxide which is a good insulator.

The solution in terms of outside wall temperature T_{wo} as a function of inside wall temperature T_{wi} , inside radius r_2 , outside radius r_1 , and heat generation rate q''' is

$$T_{wo} = T_{wi} - \frac{q'''}{2k} \left[\frac{r_1^2 - r_2^2}{2} - r_2^2 \ln \frac{r_1}{r_2} \right] \quad (C.4)$$

and in terms of the outside surface heat flux ϕ ,

$$T_{wo} = T_{wi} - \frac{\phi r_1}{k} \left[\frac{1}{2} - \frac{r_1^2}{r_1^2 - r_2^2} \ln \frac{r_1}{r_2} \right] \quad (C.5)$$

At a heat flux of 50,000 Btu/hr-ft², ΔT_w is ~6°F.

C.2 Unsteady-State Consideration

For the time dependent case, Eq. C.1 has to be written as

$$\frac{\partial^2 \theta}{\partial r^2} + \frac{1}{r} \frac{\partial \theta}{\partial r} + \frac{q'''}{k} = \frac{1}{\alpha_1} \frac{\partial \theta}{\partial t} \quad (C.6)$$

where $\theta = \theta(r, t)$. The boundary conditions being

$$\theta(r_2, t) = T_{wi}(t) \quad (C.7)$$

$$\left. \frac{\partial \theta}{\partial r} \right|_{r_2} = 0 \quad (C.8)$$

with initial condition as

$$\theta(r_2, 0) = T_{wi}(0) \quad (C.9)$$

where $T_{wi}(0)$ and $T_{wi}(t)$ are known.

The above posed problem has been referred to as the "inverse heat conduction" problem in the literature.^{130,131,132} Any numerical technique that seeks to solve the above equations "exactly" is inherently unstable as small changes in the input data can cause large changes in the solution. The approaches to the numerical solution has been categorized into four methods:

1. Integral equation approach¹³³
2. Laplace or Fourier transform¹³⁴
3. Infinite series technique¹³²
4. Nonlinear estimation technique¹³¹

The above equations cannot be viewed as a good description of the real situation at CHF location since many parameters are functions of temperature, e.g., k , α_1 , and q''' (heat generation varies because of changes in the electrical resistivity of the material due to temperature). Further axial conduction cannot be neglected because it was found to be significant at dryout location.¹¹¹

Therefore the determination of the heater surface temperature during transients was a difficult task. Furthermore, the measurement of the inside wall temperature T_{wi} during transients involves some uncertainty as the wall thermocouple beads might not be right next to the wall. This is certainly the case for all ungrounded sheathed thermocouples. As a result (see Appendix D), the surface temperature and the corresponding heat transfer coefficient calculated could be so uncertain as to make any further analysis or conclusion unreasonable.

APPENDIX D

THERMOCOUPLE RESPONSE DETERMINATION

D.1 Out-of-Loop Determination

D1.1 Theory

If a thermometric element is immersed in a constant temperature (θ_b) bath, the temperature of the thermocouple $\theta_{TC}(t)$ is related to the temperature of immersion by¹³⁵

$$\theta_{TC}(t) - \theta_b = [\theta_{TC}(0) - \theta_b] \exp^{-(1/\tau)t} \quad (D.1)$$

Equation D.1 holds for cases where the Biot number is small ($h_o d/k < 0.1$),¹³⁶ i.e., internal-conduction resistance is negligible in comparison with surface convection resistance.

The time constant τ depends not only on the physical size and shape of the thermocouple, but also on the flow conditions in which it is immersed.

For a step change in temperature, the time constant is given by the ratio of the thermal capacitance of the TC to the thermal conductance of the environmental film,

$$\tau = \frac{(\rho V C_p)_{jn}}{h_o A_{bd}} \quad (D.2)$$

where subscripts jn and bd represent the junction and bead respectively.

Therefore, for a given thermocouple $\tau \propto 1/h_o$.

D1.2 Experiment

The test was conducted using a beaker of boiling water at 212°F. Heating was supplied by a hot plate and good agitation was

ensured by the boiling process. The test thermocouple was first allowed to stabilize at room temperature and then it was plunged into the boiling water. Zero time occurred when the thermocouple first entered the fluid, as shown by a steep rise in temperature displayed on the visicorder chart. The time constant was found, knowing the speed of the chart and using Eq. D.1,

$$\frac{\theta_{TC}(\tau) - \theta_b}{\theta_{TC}(0) - \theta_b} = \frac{1}{e} = 0.368 \quad (D.3)$$

Results are shown in Table D.1. Because the time constant varied from one to another thermocouple, the approximate range is given.

D.2 In-Loop Determination

At the occurrence of boiling crisis, the cooling surface temperature rises rapidly due to the presence of vapor film or dry spots. The heat transfer coefficient at inception of CHF decreases at least by two orders of magnitude. Therefore one possible way to determine the overall time response of the heated tube and thermocouples was done as follows. With no coolant on the heat transfer side so as to simulate 'dryout' or vapor blanketing condition, the test section was suddenly subject to full power load, as used in the transient. The time response τ_{TS} is then taken to be the time delay until the thermocouple temperature begins to rise. Power to the test section was tripped in less than 2 sec to avoid physical damage. Results are shown in Figs. D.1 and D.2, and the corresponding time response τ_{TS} is given in Table D.2. The estimated temperature excursion was also plotted based on both the temporal resistance measurement and an adiabatic heatup calculation.

TABLE D.1 Time Constant of Thermocouples

Type	Application	τ (msec)
0.032 in. OD ungrounded, sheathed	spot-welded to inside wall of TS	50-80
0.0625 in. OD ungrounded, sheathed	placed in annular flow channel	150-250

TABLE D.2 Overall Time Response of Test Section and ThermocouplesHeat Flux = 45,500 Btu/hr-ft²

TC No.	τ_{TS}^* (sec)
1	0.2
2	0.3
4	0.9
6	0.3
8	0.2
10	0.3
12	0.7
14	0.8
16**	-
18	0.2
20	0.2
22	0.2
23	0.3

* τ_{TS} values rounded off to first place decimal due to uncertainty.

**TC 16 was detached from the wall.

Fig. D.1. Temperature Response for Lower Half TS Thermocouples.

Fig. D.2. Temperature Response for Upper Half TS Thermocouples.

The resistivity of SS-304, ρ_r , is a function of temperature and the following equation was obtained based on the values given in Ref. 137.

$$\rho_r \text{ (ohm - ft)} = 2.2755 \times 10^{-6} + 0.00128 \times 10^{-6} T(^{\circ}\text{F}) \quad (\text{D.4})$$

Therefore from the measured rate of change of resistance in the test section as shown in Fig. D.3, the rate of temperature rise was found to be 84°F/sec .

Assuming no axial conduction and no radial temperature gradient, the adiabatic wall analysis yields

$$\frac{\Delta\theta}{\Delta t} = \frac{\phi}{\rho C_p \xi} \quad (\text{D.5})$$

where ξ is the thickness of the tube. This value was calculated to be 96°F/sec .

As evidenced from Figs. D.1 and D.2, not all the thermocouples followed the estimated temperature rise. Those with long τ_{TS} in fact rarely follow at all. This large discrepancy observed among the behavior of the thermocouples during rapid temperature excursion is most likely due to the variation in the radial distance of the thermocouple beads from the wall. This variation was mainly caused by the spot-welding technique which did not attempt to place the bead right next to the welding spot, and further the exact location of the bead from the thermocouple sheath-end is not known. As a result, some beads were placed further away from the heated wall, thus giving rise to slower response due to increased depth and lower temperature measurement due to excessive conduction loss. Hence for transient temperature measurement, the exact location

Fig. D.3. Temporal Voltage, Current and Resistance Measurement During Rapid Joule Heating.

of the thermocouple bead is very important and this fact is more obvious in the discussion of 'inverse heat conduction' problem in Appendix C.

APPENDIX E

CAPACITANCE VOID MEASUREMENT CALIBRATION

E1. Void Measurement Instrumentation

The geometry and dimension of the void probe has been outlined in Sect. 3.2.2. A bench set up was conducted first to check the linearity of the calibration curve using a mockup test section with similar electrodes. Vertical slug (plug) and horizontal stratified flow patterns were created simply by filling the test section vertically and horizontally respectively with Freon. Figures E.1 and E.2 show the results of the bench test using a commercial impedance bridge for capacitance measurement. Measured capacitance is seen to vary linearly with void except for the vertical slug pattern where at both empty and full positions, distortion was caused by fringe effect of the outside electrode.

The dynamic measurement circuit is shown in Fig. E.3. With the test section empty, the variable capacitor C_k was adjusted so that a minimum DC output was obtained. Figures E.4 and E.5 show the results of the bench test using the devised circuit. The same linearity was achieved. Next the bridge was calibrated by providing a known capacitance (using a General Radio Precision Capacitor, Type No. 722-MEQ) across the unknown and check for linearity in DC output. The result as shown in Fig. E.6 indicates that the bridge did provide linear response as desired in the range of interest.

E2. Interpretation of Void Measurement

Knowing the DC output or capacitance measurement at both empty and full positions, and if the calibration is linear as in Fig. E.5, the apparent void fraction is found by

Fig. E.1. Vertical Calibration Using Impedance Bridge.

Fig. E.2. Horizontal Calibration Using Impedance Bridge.

Fig. E.3. Capacitance Bridge for Transient Void Measurement.

Fig. E.4. Vertical Calibration Using Capacitance Bridge.

Fig. E.5. Horizontal Calibration Using Capacitance Bridge.

Fig. E.6. Output Voltage Characteristic of Capacitance Bridge

$$\alpha_a = \frac{C_F - C_{2\phi}}{C_F - C_E} = \frac{V_F - V_{2\phi}}{V_F - V_E} \quad (E.1)$$

which is a direct result of Eq. E.4 by assuming a slug model as shown later. The resulting plot of theoretical void α versus the apparent void α_a is shown in Fig. E.7.

But is the apparent void given by Eq. E.1 sensitive to flow patterns? Although this was not experimentally tested in this investigation, some estimates of this dependence can be made in the remaining part of this section.

The capacitance C is directly related to the dielectric constant ϵ of the material between the electrodes according to

$$C = \epsilon \frac{A}{d} \quad (E.2)$$

where

A = total effective area of electrode

d = spacing between electrode plates

Hence depending on the flow patterns, the effective dielectric constant of a two-phase mixture can be calculated as a function of the dielectric constants of the two media by the following idealized considerations.¹³⁸

(a) Slug Flow Model (see Fig. E.8)

The two phases are in parallel between the electrodes; therefore

$$C_{2\phi} = C_f + C_g \quad (E.3)$$

Fig. E.7. Void Fraction Determination Using Capacitance Bridge.

(a) SLUG MODEL **(b) BUBBLE MODEL** **(c) DROPLET MODEL** **(d) FILM MODEL**

L = LIQUID
V = VAPOR

Fig. E.8. Various Idealized Two Phase Flow Models

which together with Eq. E.2 gives

$$\epsilon_{2\phi} = (1 - \alpha)\epsilon_f + \alpha\epsilon_g \quad (\text{E.4})$$

According to Eq. E.1,

$$\alpha_a = \frac{\epsilon_f - \epsilon_{2\phi}}{\epsilon_f - \epsilon_g} \quad (\text{E.5})$$

which can be regarded as a change in the dielectric of the medium. After substituting the value of $\epsilon_{2\phi}$, the following results are obtained

$$\alpha_a = \alpha \quad (\text{E.6})$$

which indicates that for the slug flow model the apparent void is identical to the theoretical void, as shown in Fig. E.9. In the present study, this slug flow model was employed in the interpretation of void fraction.

(b) Bubbly Flow Model

To find the total dielectric constant of two phase mixture with one-phase dispersed as small spheres, Maxwell¹³⁹ derived the approximate formula:

$$\frac{\epsilon_{2\phi} - \epsilon_c}{\epsilon_{2\phi} + 2\epsilon_c} = \beta \frac{\epsilon_d - \epsilon_c}{\epsilon_d + 2\epsilon_c} \quad (\text{E.7})$$

where β = volume fraction of dispersed phase

ϵ_c = dielectric constant of continuous phase

ϵ_d = dielectric constant of dispersed phase

The above equation holds if the dispersed spheres are separated by distances large in comparison with their diameters.

Fig. E.9. Theoretical Void versus Apparent Void.

In the bubbly flow model, Eq. E.7 can be written as

$$\frac{\epsilon_{2\phi} - \epsilon_f}{\epsilon_{2\phi} + 2\epsilon_f} = \alpha \frac{\epsilon_g - \epsilon_f}{\epsilon_g + 2\epsilon_f} \quad (\text{E.8})$$

or

$$\alpha_a = \alpha \left[\frac{\epsilon_{2\phi} + 2\epsilon_f}{\epsilon_g + 2\epsilon_f} \right] \quad (\text{E.9})$$

which indicates that the apparent void overestimates the theoretical void since the quantity in parenthesis is greater than one.

(c) Dispersed Flow Model

Following from the discussion in (b), Eq. E.7 can be written as

$$\frac{\epsilon_{2\phi} - \epsilon_g}{\epsilon_f - \epsilon_g} = (1 - \alpha) \frac{\epsilon_{2\phi} + 2\epsilon_g}{\epsilon_f + 2\epsilon_g} \quad (\text{E.10})$$

and

$$(1 - \alpha_a) = (1 - \alpha) \left[\frac{\epsilon_{2\phi} + 2\epsilon_g}{\epsilon_f + 2\epsilon_g} \right] \quad (\text{E.11})$$

which implies that $(1 - \alpha_a) < (1 - \alpha)$ or $\alpha_a > \alpha$

Again the apparent void overestimates the theoretical void.

(d) Film Flow Model

By considering the liquid and vapor phases as capacitances in series, the following equations were obtained,

$$\epsilon_{2\phi} = \frac{\epsilon_f \epsilon_g}{(1 - \alpha)\epsilon_g + \alpha\epsilon_f} \quad (\text{E.12})$$

and

$$\alpha_a = \alpha \frac{\epsilon_{2\phi}}{\epsilon_g} \quad (E.13)$$

which says that the apparent void overestimates the theoretical void.

With a concentric electrode configuration, the capacitance measurement is also found to be sensitive to the distribution of the film in the annular channel. Due to the lengthy derivation which is not repeated here, the final results are shown in the plot of α versus α_a , Fig. E.9. Dielectric constants at 70°F for Freon-11 were used in the calculation. Also, the effect of the glass shroud has been included.

E3. Temperature Sensitivity in Void Measurement

Since the dielectric constant is a function of temperature, temperature correction has to be considered in the void measurement. To the author's knowledge, no complete information has been obtained for Freon-11, although DuPont¹⁴⁰ suggested an empirical way of estimating the ϵ_f and ϵ_g values at various temperature. Hence at the present stage, no attempt was made to correct this effect.

APPENDIX F

VOID ESTIMATION DURING ADIABATIC BLOWDOWN

The mass quality and void fraction during an adiabatic blowdown can be estimated as below. When wall shear, heat exchange with the environment, and interfacial viscous terms are neglected, the system entropy during the blowdown expansion can be assumed constant,¹² i.e., an isentropic process

$$dS_o = d[(1+x)S_f + xS_g] = 0 \quad (F.1)$$

The blowdown from a subcooled state is going to follow the constant entropy path as shown in Fig. F.1. As blowdown proceeds, the temperature will continue to drop as quality increases. Since liquid is essentially incompressible, the initial subcooled liquid has almost the same entropy as that at the saturation value, S_o , as shown in Fig. F.1. Therefore at any time into blowdown, the following equation holds assuming thermodynamic equilibrium between phases,

$$S_o = (1-x)S_f + xS_g \quad (F.2)$$

or

$$S_o = S_f + x S_{fg} \quad (F.3)$$

Hence the quality x can be estimated if the pressure at the location of interest is known. Then the void fraction can be calculated according to

$$\alpha = \frac{1}{1 + \frac{\rho_g}{\rho_f} \left(\frac{1-x}{x} \right) S} \quad (F.4)$$

Fig. F.1. Generalized Temperature-Entropy Diagram

where

$$S = \frac{u_g}{u_f} ; \text{slip ratio}$$

= 1.0 for homogeneous flow model.

Homogeneous flow model has been assumed for the above calculation in the present study.

APPENDIX G

VOID ESTIMATION FROM TURBINE AND WEIGHT MEASUREMENTS

Flow measurement technique from turbine and weight transducers was discussed in Sect. 3.1.2. The continuity equation can be written as

$$W_1 = W_2 = (W_f + W_g)_2 \quad (G.1)$$

where 1 and 2 may be at the test section exit and turbine location, respectively. For no storage effect present in the loop, the mass flow W can be found from the mass decay signal of the weight transducer. At the turbine location, for quality x being small compared to 1,

$$(W_f)_2 = (1 - x)(W)_2 \approx (W)_2 \quad (G.2)$$

so that in conjunction with the definition for liquid mass flow

$$(W_f)_2 = [\rho_f(1 - \alpha)A U_f]_2 \quad (G.3)$$

where U_f is indicated by the turbine meter, the void at the turbine location can be found after rearranging,

$$\alpha_2 = 1 - \left[\frac{W_f}{\rho_f A U_f} \right]_2 \quad (G.3)$$

Therefore the void can be estimated as above if the mass flow and liquid velocity are known.

APPENDIX H

MEASUREMENT OF THE POWER FACTOR

The power dissipated by an electric heater is given by

$$P = VI \cos \theta \quad (H.1)$$

where θ is the phase angle between voltage and current. This phase angle comes about because a hollow tube is not a purely resistive load but has some inductance.

To measure the power factor, the test section voltage was connected to the X input of an oscilloscope and the current (which in this case the voltage drop across the shunt) to the Y input. By properly adjusting the gain to give equal amplitude input, a Lissajous figure was generated, as shown in Fig. H.1. The angle θ may be calculated from

$$\sin \theta = \frac{\text{Horizontal intercept}}{\text{Maximum horizontal deflection}} = \frac{b}{a}$$

The results obtained for the present 2 ft SS-304 tube are shown in Fig.

H.2. The average value of $\cos \theta$ is 0.993.

The instantaneous power for ac heating is given by

$$p = I_m V_m \sin(\omega t + \theta) \sin(\omega t) \quad (H.2)$$

where I_m and V_m are the amplitudes of current and voltage respectively.

The average power can be written as

$$P = \frac{1}{\pi} \int_k^{k+\pi} p(\omega t) d(\omega t) = \frac{I_m V_m}{2} \cos \theta \quad (H.3)$$

Fig. H.1. Determination of Phase Angle Using Lissajous Figure.

Fig. H.2. Measured Phase Angle versus TS Voltage.

Hence in using Eq. H.1, root mean square (rms) values are to be used. However measurement by DVM and Honeywell strip chart recorder is the mean or average value.

Since the dc rectifier puts out rippling dc, oscilloscope pictures were taken at various power level in order to access the amount of ripple present. The result is shown in Fig. H.3. Assuming the ripples are rectified sine waves, then

$$V_{\text{meas}} = V_{\text{ave}} = V_b + \frac{2}{\pi} V_{\text{max}} \quad (\text{H.4a})$$

$$I_{\text{meas}} = I_{\text{ave}} = I_b + \frac{2}{\pi} I_{\text{max}} \quad (\text{H.4b})$$

Knowing the shape of the ripples and the dc component, the average power is given by

$$P = V_b I_b + \frac{V_{\text{max}} I_{\text{max}}}{2} \left[\frac{2 \sin \theta + \pi \cos \theta - 2 \theta \cos \theta}{\pi} \right] + \frac{2}{\pi} V_{\text{max}} I_b + I_{\text{max}} V_b \quad (\text{H.5})$$

where for $\theta < 10^\circ$ in the present case

$$\left[\frac{2 \sin \theta + \pi \cos \theta - 2 \theta \cos \theta}{\pi} \right] = \cos \theta = 0.993 \quad (\text{H.6})$$

In terms of measured values (average),

$$P = V_{\text{meas}} I_{\text{meas}} + (\cos \theta - 4/\pi^2) V_{\text{max}} I_{\text{max}} \quad (\text{H.7})$$

Define a power factor ψ such that

Fig. H.3. Amplitude of Ripple versus TS Voltage.

$$P = V_{\text{meas}} I_{\text{meas}} \psi$$

(H.8)

A plot of ψ versus V_{meas} is shown in Fig. H.4.

Fig. H.4. Power Correction Factor versus TS Voltage.

APPENDIX I

INSTRUMENT CALIBRATION

Calibration of the weight transducers, turbine flowmeters, and pressure transducers will be mentioned in this section.

11. Weight Transducers

The two Statham, UC3, unbonded type strain gauge, Universal Transducing cells were calibrated with dead weights. The millivolt output was read on a HP DVM Model #2401C. Hysteresis was found to be less than 0.5%. Calibration results are shown in Figs. I.1 and I.2.

12. Turbine Flowmeters

COX Model AN 16 flowmeters were individually calibrated at 12 flow rates using the standard reference fuel, MIL-F-7024A Type 2 at 80°F by the manufacturer before shipment. From the data so obtained, the mean cycles/gallon 'K' factor was calculated. A quick check on this K factor was performed by measuring the flow rate of water using a weight tank. The rotational speed of the turbine was read on a HP Frequency Counter, Model #5202L. Agreement in K values was within 4%, mainly due to inaccurate experimental measurement. The K factor found in the case of backflow in these bidirectional turbine meters agreed to within 2% of the value for the designated flow direction.

These flowmeters were also calibrated in the loop with Freon-11. The flow was determined from the mass decay curve given by the weight transducer on the supply vessel. 'K' factors supplied from Cox were found to show better agreement with the weight measurement and the result is shown in Fig. I.3. Thus it gave a good and confident check on the weight and flow measurement.

Fig. I.1. Supply Vessel Weight Transducer Calibration.

Fig. 1.2. Blowdown Vessel Weight Transducer Calibration.

Fig. 1.3. Agreement Between Two Types of Flow Measurements.

13. Pressure Transducers

Absolute pressure transducers were calibrated using pressurized nitrogen or air, and the pressure was read with a 100 in. well-type mercury manometer (i.e., able to read below 750 psig). Hysteresis was less than 0.1% full scale for the 1000 psig transducer used at the upper plenum.

Differential pressure transducers were calibrated more or less the same way on both high and low pressure sides. A 60 in. U-tube manometer with 2.95 s.g. Meriam fluid was used to read any pressure under 5 psi. High and low pressure side calibrations of each DP cell agreed to within 1% and hysteresis in general was found to be small, see Fig. 1.4 for one calibration. Drift in the calibrations was also negligible during a test period of 4 months.

Fig. I.4. Calibration of a 100 PSID Transducer.

APPENDIX J

FREON-11 TO WATER SCALING

As already mentioned in Sect. 3.1.1, the use of Freon-water scaling for steady-state CHF has met with numerous success. Though it is doubtful how such modeling techniques can be employed for a rapid depressurizing system as in the present study, it is quite educational to look into some existing scaling criteria.

A good review of the state-of-the-art with an extensive bibliography can be found in Refs. 141 and 142. An attempt is made here to compare three existing scaling laws.

J1. Zuber and Ishii^{143,144}--In their method, the continuity, momentum, and energy equations were linearized; these together with seven constitutive equations were integrated and the steady-state values were subtracted to give a perturbed characteristic equation. Assuming thermodynamic equilibrium flow, the following dimensionless groups were obtained:

ρ_f/ρ_g	liquid to vapor density ratio.	
L_h/D_e	heated length to diameter ratio	
U_{gj}/U_{fi}	ratio of vapor drift velocity to liquid inlet velocity	

$$Re = \frac{\rho_f U_{fi} D_e}{\mu_f} \quad \text{Reynolds number} \quad (J.1)$$

$$Fr = \frac{g L_h}{U_{fi}^2} \quad \text{Froude number} \quad (J.2)$$

$$N_{SUB} = \frac{\Delta \rho}{\rho_g} \frac{\Delta h_i}{h_{fg}} \quad \text{Subcooling number} \quad (J.3)$$

$$N_{PCh} = \frac{\Gamma_g v_{fg} L}{U_{fi}} \quad \text{Phase change number} \quad (J.4)$$

where

$$\begin{aligned} \Gamma_g &= \frac{\text{Total heat generated at surface}}{\text{Latent heat of vap.} \times \text{total volume}} \\ &= \text{Rate of vapor generation/unit volume} \\ &= \frac{\phi}{h_{fg}} \frac{4D_i}{(D_o^2 - D_i^2)} \quad \text{for annular channel} \quad (J.5) \end{aligned}$$

Pressure is scaled via the density ratio, see Figs. J.1 and J.2; hence, the corresponding thermodynamic and transport properties are established as shown in Table J.1. Freon conditions during the tests at steady-state before initiation of blowdown were used.

Rearrangement of the groups L_h/D_e , Re , and Fr leads to gD_e^3/ν_l^2 . Since the kinematic viscosity for Freon-11 is 40% greater than that for water, the equivalent diameter in the water system is smaller, 25% in this case. For the sake of comparison, the geometry is made the same. Further it would be inconvenient to change model geometry for each pressure level. But this leads to a problem where both Re and Fr cannot be satisfied simultaneously. In the present approximate case, the Froude number rather than the Reynolds number has been maintained, since wide variations in Re have little effect on the friction factor.¹⁴⁵ The results are shown in Table J.2.

J2. Tong et al.¹⁴²--Tong et al. suggested the dynamic scaling expressions in subcooled and low quality flow. The physical model pictured the existence of a bubble boundary layer sliding between the subcooled core and the heating surface. First the ratio of the vapor mass velocity of vaporization and the liquid mass velocity G_f in the subcooled core is defined as Boiling number,

Fig. J.1. Density Ratio for Freon-11 and Water.

Fig. J.2. Density Ratio versus Reduced Pressure.

TABLE J.1. Thermodynamic and Transport Properties

<u>Parameter</u>	<u>Freon-11</u>	<u>Water</u>
P [psia]	65	430
T_{SAT} [°F]	165	452
ρ_f [LBM/FT ³]	84.37	51.39
ρ_g [LBM/FT ³]	1.485	0.93
ρ_f/ρ_g	56.0	55.5
μ_f [LBM/HR-FT]	0.68	0.29
μ_g [LBM/HR-FT]	0.0311	0.0394
h_{SAT} [BTU/LBM]	42.73	432.2
h_{fg} [BTU/LBM]	68.68	772.6
γ [FT ² /LBF]	6.25×10^{-3}	1.00×10^{-3}
C [HR-FT ^{5/3} /LBM]	9.40×10^{-5}	6.99×10^{-5}

TABLE J.2. Freon-11--Water Scaling Results

<u>Freon-11 (present study)</u>		<u>Water Equivalent Conditions</u>		<u>Reactor Condition (PWR)</u>
		<u>Zuber's Criteria</u>	<u>Ahmad's Criteria</u>	
P (psia)	65	430	430	2250
D_e (in.)	0.22	0.22	0.22	0.53
L_h (ft)	2	2	2	12
T_i (°F)	130	370	370	550
Δh_i (Btu/lbm)	7.71	89.2	86.5	151.0
U_{fi} (ft/sec)	7.5	7.5	17.3	15.3
G (lbm/hr-ft ²)	2.37×10^6	1.39×10^6	3.20×10^6	2.5×10^6 (ave)
ϕ (Btu/hr-ft ²)	45,000	316,000	684,000	200,000 (ave) 580,000 (max)
Reynolds number	63,900	87,700	202,300	502,000
Froude number	1.14	1.14	0.21	1.69

$$N_{Bo} = \frac{\phi}{h_{fg} \rho_f U_f} = \frac{\phi}{h_{fg} G_f} \quad (J.6)$$

A hydrodynamic similitude requires a similar flow pattern and similar velocity and shear distribution in the channel. The latter is a function of Reynolds number.

$$Re = \frac{\rho_f U_f D_e}{\mu_f} \quad (J.7)$$

Based on Tong's two-phase flow friction factor¹⁴

$$f_{TP} \propto Re^{-0.6} \quad (J.8)$$

and his boundary layer analysis at CHF (DNB) the following expression was arrived;

$$(N_{Bo})_{CHF} = C Re^{-0.6} \quad (J.9a)$$

or

$$\phi_{CHF} = C h_{fg} G_f^{0.4} \left(\frac{\mu_f}{D_e} \right)^{0.6} \quad (J.9b)$$

where C is a function of reduced pressure and the properties of the sub-cooled core. It is to be empirically determined.

Similar flow patterns can be ensured by having the same Froude number, void fraction, and density ratio. This was achieved by requiring geometric similar channels and equivalent test pressure giving the same ρ_f/ρ_g value.

The scaling factor determined by an empirical graphical method for Freon-11 was reported only at water equivalent pressure of about 1400 psia.

J3. Ahmad¹⁴¹—In Ahmad's scaling, the following parameters were first identified,

$$\text{fn}\left(\phi, G, \Delta h_i, L_h, D_e, g, h_{fg}, \rho_f, \rho_g, \mu_f, \mu_g, C_{pf}, C_{pg}, k_f, k_g, \sigma, \left|\frac{\partial}{\partial p} \left(\rho_f/\rho_g\right)\right|, \left|\frac{\partial T}{\partial p}\right|\right) = 0. \quad (\text{J.10})$$

By applying the classical dimensional analysis and Buckingham's Pi theorem, one possible set of 12 dimensionless groups was obtained.

Obviously it is impractical to satisfy all 12 conditions and so some have to be distorted. The "Compensated Distortion Model" introduced a CHF modeling parameter ψ :

$$\psi = CG(D_e)^{1/3} \quad (\text{J.11})$$

$$\text{where } C = \left(\frac{\gamma}{\rho_f \mu_f}\right)^{1/3} \left(\frac{\mu_f}{\mu_g}\right)^{1/8} \quad (\text{J.12})$$

$$\text{and } \gamma = \left|\frac{\partial}{\partial p} \left(\rho_f/\rho_g\right)\right| \quad (\text{J.13})$$

This model then showed that if ρ_f/ρ_g , $\Delta h_i/h_{fg}$ and L_h/D_e are made the same for model and prototype, both sets of CHF data drawn on ϕ_{CHF}/Gh_{fg} vs ψ_{CHF} plot were on the same curve. It was found to be successful with Freon-12, -113, -11, -21, -114, CO₂, and potassium for circular, annular, and rod bundle geometries. The result using above scaling is shown in Table J.2.

Conclusion

It is noted that Zuber's U_{gj}/U_{fj} group does not enter into the scaling procedure. This group is characteristic of the flow regime and it varies along the length. The expression for vapor drift velocity is still being developed. Zuber's criteria had met with success in hydrodynamic stability studies in two-phase flow^{1,4,145} but had not been attempted to scale steady-state CHF data. The power reduction factor in Zuber and Ahmad's scaling is 7 and 15, respectively. The discrepancy is certainly due to different approach used in scaling. However Ahmad's model should be used for CHF scaling.

The effects of flashing and vapor expansion were brought to attention by Crowley and Bergles¹⁴⁵ in a low pressure system, e.g., Freons. At low pressure where the pressure drop across the test section is large, and hence $\Delta P_{TS}/P$ is significant, the flashing and vapor expansion increase the velocity gradient, which is dynamically quite different from that caused by heat flux. The problem of high air solubility, extremely good wetting characteristics and low conductivity found in Freons has not been included in scaling approach.

APPENDIX K

CRITICAL FLOWRATE CALCULATION

. As a first approximation for low quality critical flow, the following relation is used.¹¹³

$$G_c^2 = -g_c \left[x \left(\frac{dv_g}{dP} \right) + v_g \left(\frac{dx}{dP} \right) \right]^{-1} \quad (K.1)$$

where for $x = x(P(z))$, then the derivative (dx/dP) may be expressed as the quotient $(dx/dz)/(dP/dz)$ and approximated by $\frac{\Delta x}{\Delta z} / \frac{\Delta P}{\Delta z}$. The derivative (dv_g/dP) may be evaluated using thermodynamic data.

The data of Run 6/9 at 200 msec is taken to make the following calculation since both pressure profile (Fig. 4.7g) and velocity (Fig. 4.10b) were available.

Time: 200 msec

Pressure at upper channel, $P_i = 53.0$ psia

Pressure at $z = 6$ in. $P = 50.5$ psia

Pressure at lower channel, $P_o = 48.0$ psia

Void at upper channel, $\alpha_i \simeq 0.10$

Void at lower channel, $\alpha_o \simeq 0.80$

Velocity at lower channel, $U_o \simeq 22$ ft/sec

Uniform heat flux, $\phi = 46,300$ Btu/hr-ft²

A. First, the inlet quality x_i at the upper channel can be estimated in two ways:

(i) Assuming homogeneous flow model and $\alpha_i \simeq 0.10$,

$$x_i = \frac{\alpha_i}{(1 - \alpha_i) \frac{\rho_f}{\rho_g} + \alpha_i} \quad (K.2)$$

$$x_i = 0.0015$$

(ii) assuming isentropic expansion as outlined in Appendix F,

$$S_o = S_f + x S_{fg} \quad (K.3)$$

where

$$S_o (151^\circ\text{F}) = 0.07723 \text{ Btu/lbm}^\circ\text{R}$$

$$S_f (53 \text{ psia}) = 0.07717 \text{ Btu/lbm}^\circ\text{R}$$

$$S_{fg} (53 \text{ psia}) = 0.11512 \text{ Btu/lbm}^\circ\text{R}$$

and

$$x_i = 0.0005$$

Therefore enthalpy at inlet

$$\begin{aligned} h_i &= h_f + x h_{fg} \\ &\approx h_f \\ &= 39.50 \text{ Btu/lbm} \end{aligned}$$

B. Assuming quasi-steady state condition, the heat balance between the inlet at the upper channel and the $z = 6$ in. location can be written as

$$h_i + \frac{\pi D_i \phi L}{W} = h_f + x h_{fg}, \quad (K.4)$$

where the terms on the right hand side are evaluated at the pressure corresponding to $z = 6$ in. location ($P = 50.5$ psia).

Mass flow and velocity are estimated as follows,

$$W = A_x G \quad (K.5)$$

$$G = \rho_{2\phi} U \quad (K.6)$$

and $\rho_{2\phi}$ is the two-phase density, given by

$$\rho_{2\phi} = \alpha \rho_g + (1 - \alpha) \rho_f \quad (K.7)$$

From the lower void probe measurement

$$\rho_{2\phi} = 18.16 \text{ lbm/ft}^3$$

thus

$$G = 1.44 \times 10^6 \text{ lbm/hr-ft}^2,$$

and

$$W = 3128 \text{ lbm/hr}$$

Equation K.4 then yields the quality at $z = 6 \text{ in.}$

$$x(z = 6 \text{ in.}) = 0.077$$

Using the same heat balance, the quality at $z = 0$ location is estimated to be

$$x(z = 0 \text{ in.}) = 0.113$$

The derivatives in Eq. K.1 can be approximated now;

$$\frac{\Delta x / \Delta z}{\Delta P / \Delta z} = \frac{0.077 - 0.113}{50.5 - 48.0} = -0.0144$$

where Δz is taken at the bottom 6 in. of the heated section, and

$$\frac{\Delta v_g}{\Delta P} = \frac{0.856 - 0.898}{50.5 - 48.0} = -0.0172$$

Thus the critical mass velocity at $z = 0 \text{ in.}$

$$G_c = \left\{ \frac{-32.2 \frac{\text{lb}_m \text{ft}}{\text{lb}_f \text{sec}^2} \quad 144 \frac{\text{in.}^2}{\text{ft}^2}}{\left[0.113(-0.0172) + 0.898(-0.0144) \right] \frac{\text{ft}^3 \text{in.}^2}{\text{lb}_m \text{lb}_f}} \right\}^{1/2}$$

$$= 2.71 \times 10^6 \text{ lbm/hr-ft}^2$$

$$G/G_c = 0.72$$

The values for α_o and U_o were varied by $\pm 10\%$ to study the sensitivity on the ratio G/G_c . In no case was this ratio calculated to be > 0.8 .

A similar calculation was performed at the bottom 6 in. unheated length. The quality at the exiting end was evaluated using isentropic calculation and the derivatives were estimated in the same manner. The critical mass velocity at $z = -6$ in. is

$$G_c = 3.1 \times 10^6 \text{ lbm/hr-ft}^2$$

$$G/G_c = 0.46$$

APPENDIX L

CRITICAL HEAT FLUX CALCULATION

Two CHF modeling approaches, namely that of Tong's¹⁴² and Ahmad's¹⁴¹ have been outlined in Appendix J. Attempting to use the former approach was met with difficulty because the empirical scaling factor for Freon-11 was given at pressure of above 250 psia,¹⁴² which are much higher than the present investigation. The result in using the latter approach was summarized in Table J.2. Knowing the water equivalent conditions, the operating CHF level can be estimated using existing correlations developed for water. Unfortunately no water CHF data for annuli have been reported to the author's knowledge at around 400 psia. However a CHF correlation for annular geometries which is independent of fluid types was developed from Freon-12 data.¹⁴⁶ On the basis of Ahmad's "Compensated Distortion Model," the following prediction equation was proposed:

$$W^* = (1 - e^{-\rho^* E^*}) \quad (L.1)$$

where W^* , the reduced power number, is defined as

$$W^* = \frac{4L\phi/G \, h_{fg} \, D_h}{1 - x_i} \quad (L.2)$$

$$\rho^* = \text{fn}(\rho_f/\rho_g) \quad (L.3)$$

and

$$E^* = \text{fn}(\psi_{CHF}, L/D_e, x_i) \quad (L.4)$$

The resulting correlation which was obtained with Freon-12 data between 88 to 244 psia ($0.15 < P_r < 0.41$) is as follows:

$$W^* = 1 - e^{-\left[0.025(\rho_f/\rho_g - 1)^{0.266} E(1-x_i)^{3.9/\sqrt{E}}\right]} \quad (L.5)$$

where

$$E = \left(\frac{L}{D_e} \right)^{0.715} / \psi_{CHF}^{0.866} \quad (L.6)$$

The above equation was used to calculate the critical heat flux during steady-state and transient as summarized in Table L.1. The results however have to be regarded with caution since the correlation was developed at a higher pressure range and at relatively constant pressure along the test section. During the blowdown, the fluid properties change more drastically along the test section due to large pressure drop and hence the correlation which is in terms of the inlet subcooling or quality, is bound to be less accurate.

TABLE L.1. Summary of Critical Heat Flux Estimations

Case No.	1	2	3	4	5	6
Run No.	5/29, 6/4, 6/9, and 6/12	5/29	5/29	5/29	6/4 & 6/9	5/29, 6/4, 6/9, and 6/12
Time (sec)	<0	3.0	20.0	35.0	3.0	~0.3
P (psia) ^a	65	48	43	40	48	50
T _{SAT} (°F)	155	144	137	132	144	147
ρ_f (lbm/ft ³)	84.37	86.32	86.96	87.41	86.32	86.05
ρ_f/ρ_g	56.0	77.63	86.48	93.55	77.63	74.18
μ_f (lbm/hr-ft)	0.68	0.743	0.765	0.782	0.743	0.730
μ_g (lbm/hr-ft)	0.0311	0.0305	0.0301	0.0298	0.0305	0.0305
h_{fg} (Btu/lbm)	68.68	70.98	71.69	72.19	70.98	70.68
γ (ft ² /lbf)	0.0063	0.0117	0.0145	0.0167	0.0117	0.0105
C (hr-ft ^{5/3} /lbm)	9.4×10^{-5}	1.14×10^{-4}	1.21×10^{-4}	1.26×10^{-4}	1.14×10^{-4}	1.10×10^{-4}
G (lbm/hr-ft ²)	2.4×10^6	1.55×10^{6b}	1.06×10^{6b}	0.55×10^{6b}	1.36×10^{6b}	1.0×10^{6e}
ψ_{CHF}	59.3	46.54	33.67	18.1	42.08	29.0
x_i	-0.10	~0.0 ^c	0.01 ^c	0.02 ^c	~0.0 ^c	~0.0
W^*	0.087	0.081	0.106	0.179	0.087	0.115
q_{CHF} (Btu/hr-ft ²)	80,500	45,820	41,400	36,000	43,450	41,760
CHFR ^d	1.75	1.01	0.91	0.79	0.94	0.91
Observation	no CHF	no CHF	no CHF	CHF	CHF	CHF

^aPressure is taken at the flow exit of the heated zone.

^bThe mass velocity is based on the weight transducer signal of the blowdown vessel.

^cThe inlet quality is calculated based on isentropic expansion.

^dCalculated CHF/operating heat flux, CHF occurs when CHFR < 1.0.

^eBecause sharp flow decay occurred around 300 msec, as shown in Fig. 4.10a-c, a mean value of $U = 15$ ft/sec at 80% void was used to calculate the mass velocity.

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