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**OCCURRENCE OF CRITICAL HEAT FLUX
DURING BLOWDOWN WITH FLOW REVERSAL**

by

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MASTER

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Reactor Analysis and Safety Division

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NOMENCLATURE

A	Area	r	Radius or diameter ratio
A_g	Cross-sectional flow area	μ	Viscosity
C	Capacitance	ν	Kinematic viscosity
c_p	Specific heat capacity	δ	Tube thickness
d	Diameter	ρ	Density
D	Diameter	ρ_f	Electrical resistivity
f	Friction factor	σ	Surface tension
Fr	Froude number	τ	Response time as defined locally
g	Gravitational acceleration	q	Surface heat flux
G	Mass flow rate	λ	Lamellar coefficient defined in Eq. A.2
h	Enthalpy	γ	Power factor
h	Heat-transfer coefficient	ω	Angular rotation
I	Current	Subscripts	
k	Thermal conductivity	a	Apparent
L	Length	ave	Average value
M	Mass	b	Bulk value
N	Dimensionless number used with subscripts	bd	Break of thermocouple
Nu	Nusselt number	BD	Blowdown
p	Instantaneous power	c	Continuous phase - critical condition
P	Average power, perimeter	CHF	Critical-heat-flux condition
P	Pressure	d	Dispersed phase
ΔP	Pressure differential	e	Equivalent
Pr	Prandtl number	f	Empty
q	Total heat rate	f	Liquid phase
q''	Volumetric heat-generation rate	f	Fall
Q	Volumetric flow rate	fr	Friction
r_1	Inside radius of heated tube	fg	Difference between vapor and liquid quantities
r_2	Outside radius of heated tube	g	Gaseous phase
R	Electrical resistance	h	Heated
R	Bubble radius	i	Inlet, inside, interface
Re	Reynolds number	in	Injection
Re^*	Modified Reynolds number	max	Maximum
S	Entropy	$meas$	Measured value
S	Slip ratio	M	Momentum
t	Time as defined locally	o	Outlet, outside, single phase
T	Temperature		Initial conditions
ΔT	Temperature differential	r	Reduced property
U	Velocity	s	Steady state
v	Specific volume	sat	Saturated value
V	Voltage	SAT	Saturation
W	Mass flow rate	sub	Subcooled, subcooling
W_o	Reduced power number in Ahmad's model	t	Transient
x	Vapor quality	T	Total
z	Axial location in heated channel	TC	Thermocouple
α	Void fraction	TS	Test section
α_t	Thermal diffusivity	v	Vaporization
$\dot{\alpha}$	Rate of vapor generation per unit volume	w	Wall
ϵ	Dielectric constant	w	Wetted
ϵ^*	Critical pressure ratio	z	Hydrostatic head
ϵ	Absolute temperature	ϕ	Two phase
θ	Phase angle		

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ABSTRACT

A small-scale experiment using Freon-11 at 130°F (54.4°C) and 65 psia (0.45 MPa) 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 61-cm 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 (54.4°C) and 60 psig (0.517 MPa) 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 conditions of thermodynamic nonequilibrium 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-ms reversal period. The first or early CHF, which occurred at about 400 ms, 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. (20.3 cm) 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.

1. INTRODUCTION

Since both boiling-water reactors (BWR's) and pressurized-water reactors (PWR's) operate at elevated temperatures and pressures [1200 and 2250 psia (8.273 and 15.51 MPa) 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 are flow reversal, bidirectional flow, and countercurrent flow. Current numerical analyses generally assume that steady-state CHF correlations apply to this transient system. Consequently, as flow goes through stagnation, the onset of CHF³ is predicted immediately. Experiments indicate that CHF does not occur as the flow 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.⁴ Accordingly, this CHF prediction has a significant impact on reported reactor performance with respect to acceptance criteria on clad temperature and metal-to-water reactions among others.

Current research on water-reactor safety 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 Oak Ridge National Laboratory. 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.

A. Background and Definitions

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 shows, in diagrammatic form, the various flow patterns encountered over the length of the tube, together with

*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).

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

Fig. 1 Two-phase Flow Regime⁵
ANL Neg. No. 900-75-14.

of heat transfer in the presence of subcooled liquid (region B) is known as subcooled nucleate boiling. The region of saturated nucleate boiling starts when the bulk liquid temperature reaches saturation. Slug flow (region D) results 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 a single-phase (vapor) heat transfer (region H) results when all liquid droplets disappear by evaporation.

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 oscillation⁶ or excursion of the heated surface.^{7,8} 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.

a. Departure from Nucleate Boiling (DNB) occurs in a bubbly flow pattern. When the bubbly 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.⁷ Due to the low thermal conductivity of the vapor coupled with the relatively high values of heat flux required to initiate film boiling, this process is characterized by an excursive temperature behavior.

b. "Dryout" occurs in an annular-flow pattern. The liquid film that 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 excursion at "dryout."⁸

B. Critical-heat-flux Models

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

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 a force balance on the bubble, and the CHF is calculated based on an 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.^{11,12} 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 bubble was suggested as a possible alternative mechanism.¹³

Fiori and Bergles¹⁴ observed in a low-pressure water system that near the CHF condition the flow pattern is either slug or froth flow. Their model, 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. The authors suggested that when the Leidenfrost temperature is reached, a continuous vapor patch covers the surface locally, leading to CHF.

Another model, proposed by Kutateladze¹⁵ and modified by Tong,¹⁶ 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.¹⁷ 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 developed further by Purcupile and Gouse.¹⁸ They postulated that CHF occurs when the vapor injection into the flow stream reduces the velocity gradient near the wall; 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.

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 co-workers.^{19,20} 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 Grace²¹ suggested that droplet deposition was negligible in most situations of practical interest.

Becker and Persson²² further simplified Isbin's solution and showed that dryout was a local phenomenon independent of inlet subcooling and L/D ratio.

Goldmann et al.²³ 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.

Stein²⁴ 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.

Tippets²⁵ 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 from evaporation.

Chung²⁶ 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 the local entrainment rate was governed by the Weber number associated with the upstream liquid film.

Hewitt et al.^{27,28} demonstrated that indeed the 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.

The study of the stability of liquid film flow has recently received much attention. Not only is the stability related to the dryout process, but it is also 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 Yih²⁹ and Benjamin.³⁰ Bankoff³¹ 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 effects of uneven heating and surface tension forces on film breakdown were reported by Norman and McIntyre,³² Hallett,³³ Simon and Hsu,³⁴ and Dubrovskii et al.³⁵

Hence there appear 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.

C. 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 AEC study (the Rasmussen study) have shown that the probability of a large-break LOCA is once in

Fig. 2. Schematic of a PWR System.² Arrows indicate normal operating flow direction. ANL Neg. No. 900-75-364.

10,000 yr for a PWR,² 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. 2. The largest break is taken as though the pipe separated to permit flow out of 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. 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 time from initiation of the break to the time CHF occurs in the core. Presently the steady-state correlations accepted for use in LOCA transients³⁶ include the W-3,³⁷ B&W-2,³⁸ and Macheth³⁹ correlations. Slifer's⁴⁰ 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 milliseconds. Further, it was stated by the AEC³ that "After CHF is first predicted at an axial fuel rod location during 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.

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.

A. Pool-boiling CHF under Transient Conditions

Almost all pool-boiling transients were tested under power excursion conditions. Cole⁴¹ and Graham⁴² studied stepwise power input with thin-ribbon test sections. Using an exponential power input, Rosenthal and Miller⁴³ observed temperature overshoot and delay time occurring during incipient boiling. The effects of the excursion period and subcooling were studied, and the authors concluded that transient CHF does not differ much from the corresponding steady-state CHF. The same conclusion was reached by Borishansky and Fokin.⁴⁴

Hall and Harrison⁴⁵ found that, in a rapid exponential power transient with $\Delta t < 1$ ms, film boiling was always preceded by a short burst of nucleate boiling at heat fluxes about 5-10 times the steady-state values.

Tachibana et al.⁴⁶ 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 effective cooling.

Johnson⁴⁷ summarized an extensive experimental investigation^{48,49} 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.⁵⁰ 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.,

$$\phi_t \text{CHF} = \phi_s \text{CHF} + kp^{-m},$$

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

Howell and Bell⁵¹ studied the effect of depressurization rates of 5-10 psi/s (35-70 kPa/s) 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.

B. 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 I.

1. United Kingdom: AEEW

Moxon and Edwards⁵⁴ studied flow and power transients for 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 ms and power rising to 130% of its initial value in 200 ms. The experimental and theoretical predictions of the time to CHF were compared. The theoretical predictions were based on a 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 code,⁸³ and the discrepancy varied between 0 and 400 ms.

TABLE I. Experimental Conditions for Studies of Transient Critical Heat Flux^a

Reference	Test Section	Test Fluid	Pressure, psia	Initial Mass Velocity, $10^6 \text{ lb}_m/\text{hr}\cdot\text{ft}^2$	Initial Heat Flux, $10^5 \text{ Btu}/\text{hr}\cdot\text{ft}^2$	Type of Transient Operation
47-49	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
52	19-rod bundle, 4.5 ft long	Water	1500	2.0-4.0	5.0-10.0	Power ramps; flow coastdown
53	19- and 21-rod bundles, 5 ft long	Water	1500	1.0-3.0	3.0-6.4	Power ramps at 13%/s maximum; flow coastdown
54	One tube and 37-rod bundle; both 12 ft long	Water	1000	1.5-2.0	-	Power ramps at 150%/s; flow coastdown
55	0.08 x 2.25 x 66-in. rectangular channel	Water	1100	1.0	1.6	Inlet break; outlet break
56	One-tube, 13 ft long	Water	860	1.0-2.0	1.3-2.0	Inlet break; outlet break; flow decay
57	21-rod bundle, 5 ft long	Water	750-1500	1.0-3.0	5.0-10.0	Outlet break
58, 59	One rod in annular channel and 7-rod bundle, both 2 ft and heated	Water	1500	1.0	3.5	Outlet break
60, 61	One rod in annular channel and 12 ft heated	Water	1500	1.0	3.7 ave chopped cosine	Inlet break
62	One annulus, inner tube heated	Water	15	0.35-1.4	6.3	Exponential power increase
40	One rod in annular channel, 9 ft long	Water	1000	0.25-1.0	2.0-5.0	Outlet break and/or flow decay
63	One 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
64	Nine-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
65	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
66	One tube, 3 ft long	Water	2250	1.0-2.5	6.3-11.0	Outlet break with and without flow decay
67	Nine-rod bundle, 6 ft heated	Water	2200	2.5	3.0	Outlet break and inlet break with programmed power
68	One tube	Water	1400	0.74-2.2	0.9-1.07	Flow decay

TABLE I (Contd.)

Reference	Test Section	Test Fluid	Pressure, psia	Initial Mass Velocity, $10^6 \text{ lb}_m/\text{hr}\cdot\text{ft}^2$	Initial Heat Flux, $10^5 \text{ Btu/hr}\cdot\text{ft}^2$	Type of Transient Operation
69	One tube 9.8 ft heated	Water	1090-2030	-	4.4	Outlet break
70	One tube, 1 annulus, 4- and 9-rod bundles	Freon-12, water	155 1000, 2000	0.37-2.58 ~1.0	0.0-1.0 0.0-4.5	Flow and power transient Flow and power transient
71	One tube, 9.8 ft heated	Water	1300	-	5.23	Inlet break, outlet break, and double-ended break
72	28-rod bundle in one pressure tube, 12 ft heated	Water	1070	-	0.13	Downcomer rupture and inlet break
73	Single-loop, 120-rod bundle, 9 in., uniformly heated	Water	2250	0.12	3.0-3.8	Outlet break; inlet break
74, 75	One-annulus outer tube heated, One tube, both 13 ft, with 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
76	Nine-rod bundle, 6-ft heated length, radial and axial uniform heating	Water	-	1.0-3.0	-	Flow decay and power ramp at 5% in 2 s
77	One annulus, only inside rod heated	Water	300-1010	Up to 1.55	0.0	Rampwise and stepwise power to max $\sim 1.9 \times 10^6 \text{ Btu/hr}\cdot\text{ft}^2$
78	One annulus, only inner tube, 3.6 in. heated, with outer Pyrex tube	Water	43	0.37-1.58	9.2	Outlet break
79, 80	One tube, 8 ft heated	Freon-113	200	2.50	0.24	Flow reversal, inlet-flow stoppage, and combined at constant pressure
81	One tube, 6.6 ft heated	Organic coolant	250	-	4.75	Inlet break
82	One tube, 13 ft uniformly heated	Water	1450	1.13	2.54	Inlet break

^aConversion factors: 1 psia = 6.895 kPa, $10^6 \text{ lb}_m/\text{hr}\cdot\text{ft}^2 = 1356 \text{ kg}\cdot\text{m}^2/\text{s}$; $1 \text{ Btu/hr}\cdot\text{ft}^2 = 3.155 \text{ W/m}^2$, 1 in. = 2.54 cm, 1 ft = 0.305 m.

2. Italy

a. CISE. Premoli⁵⁶ studied the effect of blowdown following simulated ruptures upstream and downstream, and loss of flow at constant pressure. Transient mass holdup in the test section, as indicated by the ratio $\rho(t)/\rho(t = 0)$, was measured by quick-acting closing valves. Therefore, to perform a complete transient, several measurements were performed at different sampling times during repeated runs. Transients performed were less than 3 s, and occurrence of CHF was not mentioned. From the result shown in Fig. 3, Premoli concluded that voidings due to inlet break were most severe and that density decrease was sharp because flow reversal caused the inlet, high-density, subcooled water to be lost from the test section sooner than under other break conditions.

Fig. 3

Dimensionless Density during Various Transients.⁵⁶ Conversion factors:
 1 psia = 6.895 kPa; 1 lb_m = 0.4536 kg;
 10⁶ lb_m/hr·ft² = 1356 kg/m²·s. ANL
 Neg. No. 900-75-782.

Gaspari et al.^{74,75} carried out loss-of-flow, power-surge, and outlet-break blowdown studies with both annular and tubular test sections, related to 1/14th and 1/40th scaled-down configurations of the CIRENE reactor channel. Experimentally, CHF occurred at the outlet during blowdown and propagated smoothly upstream. The code TILT⁸⁴ was used to analyze the transient behavior. Theoretical predictions were based on a "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.

These two hypotheses were compared by Hassid and Rychlicki⁸⁵ for channels with nonuniform axial heat-flux distribution; they found the hydrodynamic hypothesis to be better. Therefore Gaspari et al. assumed that, in both steady-state and transient conditions, the hydrodynamic critical quantities, i.e., G , x , and P , do not depend on how they vary along the channel and with time, or on how the heat flux varies along the length and with time, but depend on the boiling length traveled by the liquid that finally reaches the CHF point. Times to CHF for flow stoppage and power-surge tests were predicted within 150 ms (t_{CHF} was 0.5-3 s) using TILT 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 accuracy of prediction was strongly affected by the range of validity of the steady-state CHF correlation and the accuracy of the blowdown code. Gaspari et al. 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 decreased with the increasing depressurization rate accomplished by variable break areas.

b. Euratom. Fritz and Riebold⁶⁹ investigated the hot-leg rupture of a PWR subchannel by diaphragms of 0.315-, 0.236-, and 0.157-in. (0.800-, 0.599-, and 0.399-cm) diameter. Tests were conducted at initial pressure and water temperature of 2000 psia (14 MPa) and 600°F (316°C). The onset of CHF occurred over the outlet portion of the channel at ~500 ms and was nearly independent of rupture size, and hence of depressurization rate. However, the upstream half of the test section did not experience CHF until 4 s.

3. USSR: Moscow Power Institute

Smornov et al.⁶⁸ conducted flow-transient tests in a single tube with flow decreased by 40-70% of the steady-state value within 0-3 s. Local mass velocity and quality at inception of CHF were determined by calculation of the transient process. Steady-state CHF correlation was obtained with the same test section. With a sharp decrease in mass velocity, the computed t_{CHF} values were lower than those observed experimentally.

Fig. 4. Effect of Power-excursion Time on CHF.⁷⁰ ANL Neg. No. 900-75-781 Rev. 3.

4. Germany

Hein and Mayinger⁷⁰ summarized the power- and flow-transient test results conducted at M.A.N. (Maschinenfabrik Augsburg Nürnberg) in a single tube, an annulus, and four- and nine-rod bundles. Figure 4 shows an improvement in CHF with linear power-excursion time of a few seconds. The equation for power transients proposed by Redfield⁵⁵ underestimates CHF values. In some tests, Freon-12 was used

as the modeling fluid, and it was claimed that Boure or Steven's 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 for nonuniform ones.

The explanation for the improvement was obscure as Hein and Marjinger 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, a 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.⁷¹ studied the effect of break area, the ratio in upstream to downstream break area, and the heat flux in a single tube at 1300 psia (9 MPa). 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 generalizations were made for 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 upstream breaks, t_{CHF} is large for low ϕ but short for large ϕ .

With large upstream breaks, the upper region of test section did not experience CHF.

t_{CHF} for double-ended breaks was intermediate between the times for single-ended breaks.

5. Japan

a. General. Kawamura et al.⁶² studied pool and flow boilings of water under power-surge conditions at atmospheric pressure. With motion pictures taken at 5,000-10,000 frames/s, 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 thickness of the thermal-boundary layer. 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 showed that the transient CHF is reduced with increasing initial heat flux as is expected from above postulate.

Tokumitsu⁷² 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, the CHF was observed half a minute later at the top of the heated length. Single-inlet pipe-rupture tests were performed with one of the pressure tubes, each of which contained 28 heated rods. CHF was observed to start from the top of the pin in non-ruptured channels and from the bottom in ruptured ones. In general, t_{CHF} reported was rather long, corresponding to the complete depletion of coolant in the channels.

b. Tokyo Institute of Technology. Aoki and his co-workers,^{77,78} made extensive studies of power excursion and blowdown with inlet quality, power ramp rate, mass velocity, and heat flux as the parameters. For 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 [$G \approx 1.5 \times 10^6 \text{ lb}_m/\text{hr}\cdot\text{ft}^2$ ($2.5 \times 10^3 \text{ kg}/\text{m}^2\cdot\text{s}$)] and large subcooling, ϕ_{CHF} seems independent of power ramp rate. This was explained by the fact that the CHF mechanism by bubble coalescence is a very localized phenomenon, happening only near the heater surface. The bulk conditions are therefore of secondary importance.

In the second region, where CHF occurred at 50-70% void, the transient CHF followed the 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 a water pressure of 65 psia (448 kPa) indicated that the CHF did

Fig. 5. Time to CHF vs Excess Heat Flux.⁷⁸
Conversion factors: 1 kcal/hr-m²
1.163 W/m²; 1 ft/sec = 0.305 m/s;
1 Btu/hr-ft² = 3.155 W/m², ANL.
Neg. No. 900-75-784 Rev. 1.

not occur at the sudden vaporization just after blowdown, and their observed CHF values were higher than steady-state CHF. In Fig. 5, the excess heat flux, i.e., $\phi_{CHF} - \phi_{sCHF}$, was plotted against t_{CHF} . For large subcooling, the experimental data were 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 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.

6. Canada: AECL, Pinawa

Banerjee et al.⁸¹ 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.⁸² 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 s, respectively, for inlet subcooling of 104 and 240°C, whereas no CHF was observed for the 5% inlet break. From the mass decay curve of the coolant reported, onset of CHF corresponded to 10-15% of initial mass remaining.

7. USA

a. Westinghouse and Columbia University. Both power and flow transients were reported by Tong and his co-workers.^{52,53} 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-s delay in the effective control-rod insertion was simulated by appropriate control of the power.

Cermak et al.⁵⁷ studied the behavior of transient CHF during blowdown in rod bundles at high pressure. Steady-state CHF tests, conducted with the same bundle, were used to predict the transient CHF. The local-conditions hypothesis was used with calculations based on the CHICKIN computer code⁵⁵ 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 the above procedure underestimated t_{CHF} by 0-4.2 s for actual observed t_{CHF} of 1-7 s. Hence, CHF during blowdown could be predicted on the basis of steady-state data, but the prediction was a conservative one.

Farman and Cermak⁶⁶ studied the CHF and beyond-CHF heat transfer during blowdown in a 3-ft (0.9-m) 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 exit at the discharge location. Transient CHF occurred at the exit, but no mention was made of its propagation upstream. Steady-state correlation developed by Tong³⁷ at Westinghouse was conservative in predicting transient CHF results.

b. Bettis Atomic Power Laboratory. Redfield et al.⁸⁶ conducted blowdown studies for a rectangular heated channel, simulating both cold- and hot-leg breaks 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.⁸⁷ Although 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 s. Temperature measurement indicated good heat transfer during the initial part of the transient, for the wall temperature followed the falling saturation temperature of the coolant. Time to CHF was measured at 155, 182, and 280 s for the three tests reported, the corresponding FLASH-predicted values being 108, 160, and 380 s.

c. Babcock and Wilcox. Morgan et al.⁶⁷ presented some simulation tests for cold-leg break using the Babcock and Wilcox Boiling Heat Transfer Loop. The test section consisted of a nine-rod PWR bundle, 6 ft (1.8 m) long. The Babcock and Wilcox evaluation model CRAFT⁸⁸ was used to calculate the fuel temperature by first assuming one of three modes of heat transfer (forced convection, nucleate boiling, or stable film boiling) exists at the surface. When the computed heat flux exceeds the corresponding CHF determined by the W-3 correlation,³⁷ CRAFT switches to stable film boiling. With 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. 6). However, delaying CHF until dryout, when quality in the core is close to 1.0, resulted in much better agreement (hence termed the CRAFT Best Estimate). The same conclusion was made in the ORNL blowdown data reported by Lawson,⁸⁸ i.e., the reported hot- and cold-leg break simulations were again in reasonable agreement with the best-estimate calculation (see Fig. 7). 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 such as film boiling.

Fig. 6

Measured and Predicted Wall Temperature
for Semiscale Test 845.67 Conversion
factor: $t(^{\circ}\text{C}) = [t(^{\circ}\text{F}) - 32]/1.8$ ANL
Neg. No. 900-75-780 Rev. 2.

Fig. 7

Wall Temperature for Babcock and
Wilcox Blowdown Test.⁶⁷ Conversion
factor: $t(^{\circ}\text{C}) = [t(^{\circ}\text{F}) - 32]/1.8$ ANL
Neg. No. 900-75-785 Rev. 1.

Flow and power transients were performed on a 6-ft (1.8-m)-long, nine-rod bundle by Zielke and Wilson.⁷⁶ Typical reactor coolant-pump coastdown and 5% power ramp were simulated. Transient CHF values were compared with the Babcock and Wilcox-2 steady-state CHF correlation,¹⁸ and all data points except two had an experimental-to-predicted CHF ratio > 1.0 .

d. Oak Ridge National Laboratory. Lawson^{38,39} had reported blowdown results for both single- and seven-rod bundles with an outlet break. Blowdown times of 7-60 s were achieved by varying the diameter of a critical flow orifice located near the rupture disks. Results from one of the representative single-rod tests are shown in Fig. 8, with power reduced to 10% at 3 s

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-s seven-rod-bundle test reported,⁵⁹ CHF occurred at ~9.5 s, when loop pressure was ~215 psia (1.5 MPa).

Fig. 8

Wall Temperature and Pressure for ORNL Blowdown Test.⁵⁸ Conversion factors:
 1 psia 6.895 kPa; $t (^{\circ}\text{C}) [t (^{\circ}\text{F}) - 32]/$
 1.8; 1 kW/ft 3.281 kW/m. ANL
 Neg. No. 900-75-779 Rev. 2.

The effect of flow reversal resulting from an inlet-pipe rupture was included in the more recent ORNL Blowdown Heat Transfer program.^{60,61} The first test was in the FCTF (Forced Convection Test Facility), which consisted of a single heated 12-ft (3.66-m)-long rod in annular geometry, installed in the same loop as that 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 (1.9×10^6 W/m²) in the middle section. The first six blowdown tests simulating a single-ended inlet break showed that t_{CHF} varied from 4.0 to 9.5 s, as shown in Fig. 9. CHF occurred downstream of the highest heat-flux zone and propagated along the heater in both directions. The Thermal Hydraulic Test Facility (THTF) will study blowdown with and without flow reversal in a 49-pin, electrically heated PWR bundle.

e. 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. (22.86-cm) heated core was completed.⁷³ 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 s. Coolant in the lower plenum remained subcooled for 2-3 s. For the hot-leg-break test (Test 825)⁷³ with full power on for 14 s into blowdown, CHF occurred only at the top of the heated length at 12.5 s. The depressurization rate during the saturation blowdown was slow, ~27 psia/s. The heat-transfer coefficient was calculated to rise up to 10,000 Btu/hr·ft²·°F (57 kW/m²·°C) within 100 ms after rupture.

Fig. 9. Pressure History for ORNL Single-rod Test.⁶¹ Conversion factors;
 1 ft³ = 0.02837 m³; 1 in. = 2.54 cm; 1 psi = 6.895 kPa. ANL
 Neg. No. 900-75-576

The first test (Test 845) involving ECC injection was reported in 1972.⁸⁹ Major observations for this cold-leg-rupture test were:

- (1) The reversed flow through the core carried a two-phase fluid at high enthalpy, causing the lower-plenum temperature to rise.
- (2) Occurrence of CHF was delayed until 2.1 s at the top location on the central pin, but the cladding was rewetted at 3.7 s. The rewetting was attributed to increased downflow. Subsequently, CHF occurred at all monitored core locations between 5 and 7 s.
- (3) Following the transition from subcooled to saturated blowdown, the system decompressed at essentially fluid-saturation temperature for the rest of the transient.
- (4) 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 of the break.
- (5) Vessel fluid qualities were > 60% within 2 s after rupture.

f. General Electric (GE). Slifer⁴⁰ reported on GE models for BWR performance in a LOCA. Using the GE Heat Transfer Loop at San Jose, he studied the following transients using both single- and nine-rod square-array geometry:

(1) Depressurization at 50-100 psi/s (344.7-689.5 kPa/s) with constant flow showed that use of the GE steady-state correlations (Janssen-Levy for single tube and Hench-Levy for rod bundles) to predict the onset of CHF in transient was conservative.

(2) Flow-decay tests also indicated the same conservatism.

(3) Combined depressurization and flow-decay tests also showed that CHF did not occur earlier than steady-state prediction.

Lahey and his co-workers⁶³⁻⁶⁵ reported results from an experimental transient CHF program in single-, 9-, and 16-rod test sections. A simplified analysis based on the method of characteristics was developed that allowed prediction of the transient flow and quality in a heated channel. Computer code MAYU⁶⁴ 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 and then turned back on. Time to CHF ranged from 2.3 s for $\phi = 149,000$ Btu/hr·ft² (3×10^6 W/m²) in a 16-rod bundle to 21 s for $\phi = 51,000$ Btu/hr·ft² (1.6×10^5 W/m²).

These tests indicated that pool boiling is an effective mode of heat transfer. In many cases, the rods were rewet with no apparent difficulties once the flow was reestablished. For the depressurization tests with rates of 50-100 psi/s (344.7-689.5 kPa/s) 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.

g. Massachusetts Institute of Technology (MIT). Using Freon-113 at 200 psia, Griffith and his students^{79,80} studied transient CHF during flow reversal, inlet-flow blockage, and a combination of the two. For initial uniform heat flux and subcooled upflow, the transient consisted of two flow reversals bounding a period of downflow, τ_d . The results showed that CHF did not occur at flow reversal, but depended on the history of the peak enthalpy fluid. In Fig. 10, the heat flux marking the CHF boundary for the corresponding downflow duration τ_d is plotted. For negligibly small τ_d (<100 ms), no CHF was observed until the heat flux was at the ϕ_{sCHF} level, in which case the disturbance would trigger an incipient CHF. At a τ_d of about one channel transport time (~1 s in their runs), the ϕ_{CHF} value observed was lowest.

Fig. 10

MIT Results for Double Flow Reversal.⁸⁰
 Conversion factors: 1 Btu/hr-ft² =
 3.155 W/m²; 1 psia = 6.895 kPa.
 ANL Neg. No. 900-75-783 Rev. 2.

This result was explained by the fact that the enthalpy peak traversed three full passes through the heated section. For transients with longer τ_d , the dependency of ϕ_{CHF} on τ_d was small, since these transients 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 1-2 s under the test conditions. A small but versatile computer program BACTRAC3 was successfully developed⁸⁰ to predict the occurrence of CHF during rapid flow-reversal transients.

8. Summary

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.^{54,70,74,76,77} 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 introduced,⁹⁰ and flooding phenomenon was experimentally observed in the low-velocity, countercurrent-flow, CHF studies conducted at MIT.⁹¹ 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. (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.) Flow-stagnation experiments indicated that heat flux at low flow can be quite high, as in pool boiling.⁶⁵ At this low flow, CHF was found to depend on void fraction⁹¹ (generally unknown in pool-boiling experiments).

A major finding in blowdown studies is that time to CHF decreases as the rate of depressurization increases.^{57,75} The same conclusion was arrived at in a pool-boiling depressurization study.⁵¹ High-speed photos showed vigorous bulk nucleation, in agreement with the findings in flow-boiling

depressurization.⁷⁸ Vigorous nucleation, acceleration,⁷⁴ turbulence,⁶⁷ and stability of the liquid film⁷⁸ on the heater wall during the initial phase of blowdown have all been proposed as plausible explanations 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.⁵⁸ 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.^{58,67,82} 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,^{66,69,71,74,89} for nonuniform heat flux, bidirectional propagation of the CHF front which occurred originally at the maximum heat-flux location was observed.⁶¹

III. EXPERIMENTAL INVESTIGATION

A. Outline of Experimental Technique

1. Choice of Working Fluid

In the study of steady-state CHF, the water system has been scaled down successfully by using a liquid having a lower latent heat of vaporization and operating at pressure having the same liquid/vapor density ratio as the water system. Such liquids reported in the literature are Freon-11,^{92,93} Freon-12^{94,95} Freon-113,⁹⁶ Freon-114,⁹⁷ Freon-22,⁹⁸ and CO₂.⁹⁹ Freons are usually chosen because of their very low latent heat, relatively low hazard, and well-established fluid properties. Freon-11 at 260 psia (1.8 MPa) and 237°F (114°C) has been used to scale CHF of steam-water mixtures at 1500 psia (10.3 MPa) and 500°F (260°C).¹⁰⁰ 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 a pressure transient has not been explored, and it is uncertain that scaling can be achieved. A discussion of scaling appears in Appendix J.

The experiment described here used Freon-11 as the boiling fluid in a transparent annular test section so that direct photographic observations could be made of the fluid behavior before, during, and after local CHF resulting from a depressurization transient. Because of the transparent glass section, the experiment was conducted at 65 psia (0.45 MPa), corresponding to a water-equivalent pressure of 450 psia (3.1 MPa) based on density ratio.

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. A turbine flowmeter is commonly used in single-phase flow. Based on Grey's analysis¹⁰¹ for the transient response of a turbine flowmeter, Popper¹⁰² 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

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

The error is small for low and intermediate void fractions as either slip ratio $\mathcal{S} \rightarrow 1$ or $\rho_g \ll \rho_f$. Later Rouhani¹⁰³ cautioned that only for low quality will the response of the turbine be proportional to liquid velocity alone.

Hutcherson¹⁰⁴ 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 was used in the present study to measure the mass flow for comparatively long transients. In addition, turbine meters were located in the inlet and outlet legs of the test section, and used mainly to detect significant changes in flow velocity.

3. Measurement of Void during Transient

Most of the void-measuring techniques are based on the attenuation of a radiation beam such as gamma and X rays. Among the nonnuclear methods of void measurement, one of the most important is the impedance method,^{105,106} which is based on the difference between liquid and vapor electrical conductivity and dielectric constant. The impedance method again can be divided into conductivity¹⁰⁷ and capacitance¹⁰⁸ measurements.

Dukler¹⁰⁹ 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.^{64,110,111} 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.¹¹² In the present study, a similar void probe was used because of its simplicity in design and low cost. Appendix E contains information on the electronic circuit for dynamic void measurement as well as the calibration test.

B. Description of Experimental Apparatus

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, shown in Fig. 11, was a once-through loop and consisted of three main vessels: a supply vessel, a variable-volume blowdown vessel, and a receiving vessel. The vessels are described below.

a. Supply Vessel. A 5.5-kW, 14-in. (35.56-cm) Watlow immersion heater was used for heating ~600 lb of Freon to the desired operating temperature inside this 8-ft³ (2.4-m³) vessel. Four immersed thermocouples (Type K) were installed at four vertical locations, and the temperatures were monitored on a multichannel recorder during heating. The close agreement among the four thermocouples indicated that no significant stratification occurred.

Fig. 11. Freon-11 Blowdown Facility. ANL Neg. No. 900-4557 Rev. 2.

The vessel and its contents were suspended from a weight transducer that was a Satham, UC3, unbonded-type strain gauge, Universal Transducing Cell. The cell was mounted in a Satham, UL4, 1000-lb_f (4448-N) load-cell accessory head, which permitted the cell to detect a maximum of 1250 lb_f (5560 N) 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.

b. Blowdown Vessel. The blowdown vessel was a 30-in. (76.2-cm)-long Type 304 stainless steel cylinder with an inside diameter of 4 in. (10 cm) and a wall 0.237 in. (0.6 cm) thick. It had a dome end cap on the bottom, and a flange connection on the top. Both ends had 1-in. (2.54-cm) 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 about 1 in. (2.54 cm) of Fiberglas insulation. Four immersed thermocouples (Type K) were installed at four vertical locations, 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 of weight transducer. A 500-lb_f (2224-N) load-cell accessory head was used in this case, with the calibration curve shown in Appendix I.

Fig. 12. Schematic Showing Refrigeration Unit and Receiving Vessel. Conversion factor: 1 ft³ = 0.02837 m³. ANL Neg. No. 900-75-13 Rev. 2.

c. Receiving Vessel. This large vessel of ~45 ft³ (13.7 m³) with cooling capability supplied from a refrigeration unit, is shown in Fig. 12. The refrigeration unit consisted of a tank containing a 50% water-50% ethylene glycol mixture to which 100-150 lb (45-68 kg) 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 (373-W) pump inside this coil. The 1000 lb of Freon-11

in the vessel was typically maintained at 10°F; a vacuum pump maintained the pressure inside at 10 psia (68.95 kPa) 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 a 2-in. (5.08-cm) inner layer of asbestos and 8 in. of Fiberglas material.

2. Test Section

The test section, shown isometrically in Fig. 13, consisted of a 24-in. (86-cm)-long, 0.752-in. (1.9-cm)-ID, 0.805-in. (2.045-cm)-OD, directly heated, Type 304 stainless steel tube, silver-brazed at both ends to two

Fig. 13. Isometric of Transient CHF Test Section. Conversion factor: 1 in. = 2.54 cm. ANL Neg. No. 900-5597.

14-in. (35.56-cm)-long copper leads of low electrical resistance. The circumferential variation of the Type 304 stainless steel tube thickness, measured at both ends using a tubing micrometer, was ± 0.001 in. (0.0254 mm). Hence, radial variation in heat flux was minimal. The resistance of this 0.0265-in. (0.67-mm) wall tube was calculated to be 0.0103Ω , which agreed closely with Kelvin-bridge measurement of 0.0104Ω . Dimensional information is listed in Table II.

The outer 3-ft (0.9-m) transparent Pyrex pipe had five 1/8-in. (0.3175-cm) holes drilled at 6-in. (15-cm) intervals for measurements of differential pressure. 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 was cushioned from the glass by a molded asbestos insert. The flat Type 304 stainless steel surface of the plenum had been tap drilled at the proper locations to take three 5/16-in. (0.79-cm) flange bolts. A Teflon gasket was gripped between the grooved pipe end and the plenum surface when the flange bolts were tightened.

To ensure that the heated tube was properly centered with respect to the Pyrex pipe, the following assembling procedure was used. 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 could be pushed into or pulled out of the channel. Flow straighteners, as shown in Fig. 13, were made to center the heated tube with respect to the plenums. First, the two collars were pushed into the annular gap at both ends, followed by the flow straighteners. This resulted in proper centering of the Pyrex pipe 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.

TABLE II. Dimensions of Test Section

Stainless steel-heater OD	0.805 in. (2.0447 cm)
Stainless steel-heater ID	0.752 in. (1.9101 cm)
Copper-tube OD	0.805 in. (2.0447 cm)
Copper-tube ID	0.625 in. (1.5875 cm)
Outer-wall Pyrex OD	1.344 in. (3.4138 cm)
Outer-wall Pyrex ID	1.022 in. (2.5959 cm)
Heated length	23.5 in. (59.69 cm)
Heated-length L/D ratio	108
Calming length	6.0 in. (15.24 cm)
Calming-length L/D ratio	28
Hydraulic diameter	0.217 in. (0.5512 cm)
Heated equivalent diameter	0.492 in. (1.2497 cm)
Annular flow area	0.311 in. ² (2.006 cm ²)
Annular-section volume	11.210 in. ³ (183.699 cm ³)
Upper-plenum volume	29.360 in. ³ (74.574 cm ³)
Lower-plenum volume	29.360 in. ³ (74.574 cm ³)

Twenty-two 0.032-in. (0.762-mm)-OD, Type-K (Chromel-Alumel), ungrounded, sheathed thermocouples were spot-welded onto the inside wall of the stainless steel tube (see Fig. 14). Nine of these were used for CHF detection by means of temperature controllers. The measurements from the rest of the thermocouples were recorded as primary data. Connections were made to flexible leads where the sheathed thermocouples emerged from the ends of the copper leads. Two 0.0625-in. (1.5875-mm)-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 was 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 a 0.001-in. (0.025-mm)-thick, 1-in. (2.54-cm)-wide stainless steel foil, which surrounded the tube completely. There were two such electrodes located 3 in. upstream and downstream of the heated region, as shown in Fig. 13. 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.

Fig. 14. Thermocouple Locations. Conversion factor:
1 in. = 2.54 cm. ANL Neg. No. 900-75-387.

3. Instrumentation

Power to the test section was supplied by a 20-V, 1500-A, 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, 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 about 300 ms. If the heated tube was suddenly insulated, an average temperature rise of 90°F/s (~32°C/s) at a heat flux of 45,000 Btu/hr-ft² would be experienced. Thus it was believed 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 A.

Table III contains a summary list of instruments and the corresponding recording devices.

TABLE III. Transient CHF Test Instrumentation

Parameter	Instrument	Recorder
Wall temperature	Thirteen Chromel-Alumel thermocouples	Visicorder Honeywell Model 1508
Fluid temperature	Four Chromel-Alumel thermocouples	Honeywell strip chart
Test-section pressure	Two strain-gauge Statham transducers	Visicorder
Blowdown-vessel pressure	Statham transducer	Honeywell strip chart
Receiving-vessel pressure	Statham transducer	Visicorder
Test-section pressure drop	Six strain-gauge Statham and CFC DP cells with 5-15 psid (34-103 kPa)	Visicorder
Test-section flow	Inlet and outlet Cox turbine flowmeters (bidirectional), 2.5-60 gpm (1.6×10^{-4} - 3.8×10^{-3} m ³ /s) used with two Cox flow converters	Honeywell strip chart
	Weight transducers on supply vessel and blow-down vessel, both Statham unbonded-type strain gauges	Hewlett-Packard strip chart
Void fraction	Capacitance void probes with detection circuit	Honeywell strip chart
Test-section power	50-mV, 2000-A shunt	Honeywell strip chart
Photographic measurement	Hycam camera	800-1000-fps 1244-305-mm sl Kodak Ektachrom film

C. Experimental Procedure

1. Calibration

Pressure transducers, weight transducers, and turbine flowmeters were calibrated 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.

Amplifiers were used to raise transducer and thermocouple signals to the required input level for visicorder galvanometers, but the drifts in these amplifiers gain were found to be significant. To minimize the effect of such drift, the amplifiers were calibrated by inputting a known millivolt signal from a standard dc voltage supply just before each test.

2. Refilling Operation

The supply vessel was filled 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. 11 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.¹¹

3. Heatup Operation

The temperature controller for the immersion heater in the 85% full-supply tank was first set at 100°F (38°C) with the heater on. When the controller started to operate, the vent was opened slightly to bleed off any dissolved gas that 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,^{111,114} and the common degassing procedure used here was to boil the system at its saturation temperature.^{110,111} After degassing in a two-phase state, the vent was closed, and the controller was set at the higher operating temperature [135°F (57°C)]. A four-fifths-full vessel of Freon-11 would attain stable and uniform temperature as indicated by the four immersed thermocouples after 1½ 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.

4. Blowdown Operation

The following checkout procedure was performed before each run:

- a. Air at 45 psig was supplied to the three air-operated solenoid valves SW1, SW2, and SW3 shown in Fig. 11. Valves SW1 and SW3 were opened by a manually operated switch, while SW2 remained closed.
- b. The loop was purged with hot Freon, and the pressure transducers were bled for trapped gases.
- c. Various readout devices were checked for proper recording conditions, and photographic equipment was made ready.
- d. The supply vessel was pressurized by nitrogen, resulting in subcooled Freon.
- e. Single-phase upflow in the test section was established by opening the flow-control valves.
- f. Desired power level to the test section was turned on, and the flow and inlet pressure were set to the desired values by adjusting the flow-control valves and the throttle-line valve located just upstream of valve SW1.
- g. After steady-state upflow with subcooled boiling was established and maintained for 1 min to ensure steady loop-component temperatures, all recording and photographic systems were actuated.
- h. The manually operated switch in step a 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, 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.

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 IV for the diabatic tests. The original analog data signals, however, are not presented, due to the enormous number of traces involved.

TABLE IV. Initial Steady-state Conditions in Test Section before Blowdown^a

Run No	Pressure P_0 , psia	Temperature T_0 , °F	Subcooling ΔT_{sub} , °F	Velocity in TS, U_0 , ft/s	Mass Velocity G , lb _m /hr-ft ²	Heat Flux q , Btu/hr-ft ²	Blowdown Volume ^b V_{BD} , ft ³
4/22	67.5	127.1	40.8	5.28	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	7.84	2.37×10^6	45,520	0.318
6/4	64.8	132.8	32.7	7.84	2.37×10^6	45,460	0.318
6/9	64.5	133.9	30.7	7.84	2.37×10^6	46,400	0.201
6/12	65.1	133.1	32.0	7.84	2.37×10^6	46,570	0.072

^aConversion factors: 1 psia = 6.895 kPa, 1 °C = $\frac{5}{9}$ (°F - 32), 1 ft/s = 0.305 m/s, 10^6 lb_m/hr-ft² = 1356 kg/m²s, 1 Btu/hr-ft² = 3.155 W/m², 1 ft³ = 0.0283 m³.

^bBlowdown volume is defined to be the system volume between the lower plenum and the SW3 valve.

A. Adiabatic Blowdown Tests

A number of adiabatic blowdown tests were conducted with Freon at nominal conditions of 65 psia (448 kPa) and 145 °F (63°C). The tests were performed by establishing a steady upflow and then initiating a decompression that produced flow reversal within the system. Results for two typical blowdown transients are presented in Figs. 15-18. The shortest transient was obtained with the blowdown vessel isolated from the loop; the system required about 15 s 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 (59°C)] than that of the flowing liquid in the turbine legs and test section [151°F (66°C)].

B. 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, the corresponding initial conditions being summarized in Table IV.

1. Run 4/22

For Run 4/22, the transient was initiated with solenoid valves SW1 and SW3 closed about 1 s before SW2 opened. As 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. 19, until decompression

Fig. 15

Pressure and Void Measurement
 for Adiabatic Blowdown Run 4/4A.
 Blowdown volume $\approx 0.08 \text{ ft}^3$.
 Conversion factors: 1 psia =
 0.895 kPa; $t (^{\circ}\text{C}) = [t (^{\circ}\text{F}) - 32]/$
 1.8; 1 ft/sec = 0.305 m/s. ANL
 Neg. No. 900-75-431 Rev. 1.

Fig. 16
 Fluid Temperature during Blow-
 down Run 4/4-A. Conversion
 factor: $t (^{\circ}\text{C}) = [t (^{\circ}\text{F}) - 32]/$
 1.8. ANL Neg. No. 900-75-847.

Fig. 17. Pressure and Weight Measurement for Adiabatic Blowdown Run 4/9B. Blowdown volume ≈ 0.582 ft³. Conversion factors: 1 psia = 6.895 kPa; 1 lb_m = 0.4536 kg; $t(^{\circ}\text{C}) = [t(^{\circ}\text{F}) - 32]/1.8$; 1 ft/sec = 0.305 m/s. ANL Neg. No. 900-75-430.

Fig. 18. Void Fraction and Turbine Flowmeter Measurement for Adiabatic Blowdown Run 4/9B. Conversion factor: 1 ft/sec = 0.305 m/s. ANL Neg. No. 900-75-429.

Fig. 19

Pressure History for Diabatic Blowdown
Run 4/22. Blowdown volume $\approx 0.08 \text{ ft}^3$.
Conversion factors: $1 \text{ psia} = 6.895 \text{ kPa}$;
 $t (^{\circ}\text{C}) = [t (^{\circ}\text{F}) - 32]/1.8$; $1 \text{ ft/sec} =$
 0.305 m/s ; $10^6 \text{ lb}_m/\text{hr}\cdot\text{ft}^2 = 1356 \text{ kg}/$
 $\text{m}^2\cdot\text{s}$; $1 \text{ Btu/hr}\cdot\text{ft}^2 = 3.155 \text{ W/m}^2$.
ANL Neg. No. 900-75-342.

occurred on opening valve SW2. Figure 20 shows the wall temperature and the fluid temperature. Figures 21 and 22 show the flow and void-fraction measurements, respectively. Time zero is taken to be the start of depressurization. The time at power trip is indicated on the figures.

2. Run 5/19-A

Like Run 4/22, Run 5/19-A was performed with the blowdown vessel removed from the loop and replaced by a flexible stainless steel tube as shown in Fig. 11. Subcooled flow boiling was visually observed all along the heated length during steady-state operation. Time zero for the transient was again taken to be the instant at which the plenum pressures started to drop (see Fig. 23). Temperature and void-fraction measurements are shown in Figs. 24 and 25. Figure 26 shows the time to CHF, t_{CHF} , at various locations along the heated section. The inception of CHF is defined as

$$t_{\text{CHF}} = t_{\text{TC}} - \tau_{\text{TS}} \quad (1)$$

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 a separate test, as described in Appendix D.

3. Run 5/29

Run 5/29 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. 27-33. Heater-wall temperatures on the bottom 8 in. rose without power trip, the most severe rise occurring 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 at $\sim 40 \text{ s}$ as the whole test section went into CHF condition. Figure 31 shows that the blowdown vessel was essentially empty at 40 s.

Fig. 20. Temperature History for Run 4/22. Conversion factor:
 $t (^{\circ}\text{C}) = [t (^{\circ}\text{F}) - 32]/1.8$. ANL Neg. No. 900-75-346.

Fig. 21. Turbine Flow Measurement for Run 4/22.
 Conversion factor: 1 ft/sec = 0.305 m/s.
 ANL Neg. No. 900-75-339.

Fig. 22

Void-fraction Measurement for Run 4/22. ANL
Neg. No. 900-75-341.

Fig. 23

Pressure History for Diabatic Blowdown Run 5/19A. Blowdown volume $\approx 0.072 \text{ ft}^3$. Conversion factors: $1 \text{ psia} = 6.895 \text{ kPa}$; $t (^{\circ}\text{C}) = [t (^{\circ}\text{F}) - 32]/1.8$; $1 \text{ ft/sec} = 0.305 \text{ m/s}$; $10^6 \text{ lb}_m/\text{hr}\cdot\text{ft}^2 = 1356 \text{ kg/m}^2\cdot\text{s}$; $1 \text{ Btu/hr}\cdot\text{ft}^2 = 3.155 \text{ W/m}^2$. ANL Neg. No. 900-75-389.

Fig. 24. Temperature History for Run 5/19A. Conversion factor:
 $t (^{\circ}\text{C}) = [t (^{\circ}\text{F}) - 32]/1.8$. ANL Neg. No. 900-75-801.

Fig. 25

Void-fraction Measurement for Run 5/19A.
ANL Neg. No. 900-75-388 Rev. 1.

Fig. 26

Time to CHF vs Axial Location for Run 5/19A.
Conversion factor: 1 in. = 2.54 cm. ANL
Neg. No. 900-75-394.

Fig. 27

Pressure History for Diabatic Blowdown
Run 5/29. Blowdown volume ≈ 0.318 ft³.
Conversion factors: 1 psia = 6.895 kPa;
 t (°C) = $[t$ (°F) - 32]/1.8; 1 ft/sec =
0.305 m/s; 10^6 lbm/hr-ft² = 1356 kg/
m²·s; 1 Btu/hr-ft² = 3.155 W/m².
ANL Neg. No. 900-75-417.

Fig. 28. Temperature History for Run 5/29. Conversion factor:
 $t (^{\circ}\text{C}) = [t (^{\circ}\text{F}) - 32]/1.8$. ANL Neg. No. 900-75-412.

Fig. 29

Void-fraction Measurement for Run 5/29.
ANL Neg. No. 900-75-418.

Fig. 30

Flow Measurement during Blowdown for
Run 5/29. Conversion factor: 1 ft/sec =
0.305 m/s. ANL Neg. No. 900-75-420.

Fig. 31

Mass in Blowdown Vessel for Run 5/29.
Conversion factor: 1 lb_m = 0.4536 kg.
ANL Neg. No. 900-75-414.

Fig. 32

Time to CHF vs Axial Location for
Run 5/29. Conversion factor: 1 in. =
2.54 cm. ANL Neg. No. 900-75-413.

Fig. 33

Onset of Rewetting during Run 5/29.
Conversion factor: 1 in. = 2.54 cm.
ANL Neg. No. 900-75-419.

C. Diabatic Blowdown with High-speed Photography: Runs 6/4, 6/9, and 6/12

The following tests were performed after the following minor change. 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 airflow 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. (15 cm) 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 photographs. 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 IV. The results are shown in Figs. 34-51.

The pressure profile obtained from DP cell measurement for Run 6/9 is shown in Fig. 45. Some pressure traces were lost in the other two runs; the signals did not appear on the visicorder paper. Results of high-speed photography are also shown as selected prints at various times into blowdown in Figs. 52-54. The transient coolant velocities, obtained with a motion-picture analyzer, are presented in Figs. 55-57. The velocity shown is more indicative of the liquid phase when significant slip occurs as in annular flow.

Fig. 34

Pressure History for Diabatic Blowdown Run 6/4.
 Blowdown volume = 0.318 ft³. Conversion factors:
 1 psia = 0.885 kPa; $t \text{ (} ^\circ\text{C)} = [t \text{ (} ^\circ\text{F)} - 32]/1.8$;
 1 ft/sec = 0.305 m/s; $10^6 \text{ lbm/hr-ft}^2 = 1356 \text{ kg/m}^2\text{-s}$;
 1 Btu/hr-ft² = 3.155 W/m². ANL Neg. No. 900-75-488.

Fig. 35. Temperature History for Run 6/4. Conversion factor:
 $t (^{\circ}\text{C}) = [t (^{\circ}\text{F}) - 32]/1.8$. ANL Neg No. 900-75-804.

Fig. 36. Void-fraction Measurement for Run 6/4. ANL Neg. No. 900-75-501.

Fig. 37. Turbine Flow Measurement for Run 6/4. Conversion factor: 1 ft/sec = 0.305 m/s. ANL Neg. No. 900-75-500.

Fig. 38. Mass in Blowdown Vessel for Run 6/4. Conversion factor: 1 lb_m = 0.4536 kg. ANL Neg. No. 900-75-497.

Fig. 39. Pressure-differential Measurement for Run 6/4. Conversion factors: 1 psia = 6.896 kPa; 1 in. = 2.54 cm. ANL Neg. No. 900-75-921.

Fig. 40. Pressure History for Diabatic Blowdown Run 6/9. Blowdown Volume = 0.201 ft³. Conversion factors: 1 psia = 6.895 kPa; $t(^{\circ}\text{C}) = [t(^{\circ}\text{F}) - 32]/1.8$; 1 ft/sec = 0.305 m/s; $10^6 \text{ lb}_m/\text{hr} \cdot \text{ft}^2 = 1356 \text{ kg}/\text{m}^2 \cdot \text{s}$; 1 Btu/hr-ft² = 3.155 W/m². ANL Neg. No. 900-75-495.

Fig. 41. Temperature History for Run 6/9. Conversion factor: $t(^{\circ}\text{C}) = [t(^{\circ}\text{F}) - 32]/1.8$. ANL Neg. No. 900-75-803.

Fig. 42. Void-fraction Measurement for Run 6/9. ANL Neg. No. 900-75-502.

Fig. 43. Turbine Flow Measurement for Run 6/9. Conversion factor: 1 ft/sec = 0.305 m/s. ANL Neg. No. 900-75-499.

Fig. 44. Mass in Blowdown Vessel for Run 6/9. Conversion factor: 1 lbm = 0.4536 kg. ANL Neg. No. 900-75-490.

Fig. 45. Pressure-differential Measurement for Run 6/9. Conversion factors: 1 in. = 2.54 cm; 1 psi = 6.895 kPa. ANL Neg. No. 900-75-920.

Fig. 46

Axial Pressure Profile during Diabatic Blowdown Run 6/9. Conversion factors: 1 psi = 6.895 kPa; 1 in. = 2.54 cm. ANL Neg. No. 900-75-503.

Fig. 47

Pressure History for Run 6/12. Blowdown volume $\approx 0.072 \text{ ft}^3$. Conversion factors: 1 psia = 6.895 kPa; $t (^{\circ}\text{C}) = [t (^{\circ}\text{F}) - 32]/1.8$; 1 ft/sec = 0.305 m/s; $10^6 \text{ lbm/hr}\cdot\text{ft}^2 = 1356 \text{ kg/m}^2\cdot\text{s}$; 1 Btu/hr $\cdot\text{ft}^2 = 3.155 \text{ W/m}^2$. ANL Neg. No. 900-75-416.

Fig. 48. Temperature History for Run 6/12. Conversion factors:
 $t (^{\circ}\text{C}) = [t (^{\circ}\text{F}) - 32]/1.8$. ANL Neg. No. 900-75-802.

Fig. 49
Void-fraction Measurement for
Run 6/12. ANL Neg. No. 900-75-415.

Fig. 50. Turbine Flow Measurement for
Run 6/12. Conversion factor:
1 ft/sec = 0.305 m/s. ANL Neg.
No. 900-75-494.

Fig. 51. Pressure-differential Measurement
for Run 6/12. Conversion factors:
1 psi = 6.895 kPa; 1 in. = 2.54 cm.
ANL Neg. No. 900-75-922.

Fig. 52. Selected Prints for Run 6/4. ANL Neg. No. 900-75-326.

Fig. 53. Selected Prints for Run 6/9. ANL Neg. No. 900-75-325.

Fig. 54. Selected Prints for Run 6/12. A.NL Neg. No. 900-75-327 Rev. 1.

Fig. 55. Velocity Observed in High-speed Photography during Run 6/4. Conversion factor: 1 ft/sec = 0.305 m/s. ANL Neg. No. 900-75-988.

Fig. 56. Velocity Observed in High-speed Photography during Run 6/9. Conversion factor: 1 ft/sec = 0.305 m/s. ANL Neg. No. 900-75-944.

Fig. 57

Velocity Observed in High-speed Photography during Run 6/12. Conversion factor: 1 ft/sec = 0.305 m/s. ANL Neg. No. 900-75-987.

For all these runs, time to CHF was short (~400 ms), as shown in Table V. Similar to Run 5/29, temperature excursion was observed only at the bottom 8-10 in. (20-25 cm).

TABLE V. Time to CHF^a (t_{CHF}) for Diabatic Blowdown

Run No.:	Thermocouple Location, in. (cm)						
	1	2	4	6	8	10	12
Time to CHF, s							
6/4	0.4 (1.0)	0.5 (1.3)	0.4 (1.0)	0.5 (1.3)	0.4 (1.0)	0.5	b
6/9	0.3 (0.8)	0.5 (1.3)	0.4 (1.0)	0.5 (1.3)	0.4 (1.0)	b	b
6/12	0.3 (0.8)	0.4 (1.0)	0.4 (1.0)	0.2 (0.5)	0.2 (0.5)	0.3 (0.8)	b

^a t_{CHF} is defined by Eq. 1.

^bNo CHF.

V. DISCUSSION

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

A. Adiabatic Blowdown

1. Run 4/4-A

Figure 15 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 considered 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 measured experimentally.

During the initial stage of the unstable growth of a bubble, the inertia of the surrounding liquid and the surface tension control the process,⁸ however, this period is 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.¹⁰⁴

The measured and calculated void in the lower test section reached 50% at ~2.5 s, at which time the pressure started to drop and the bulk fluid temperature agreed with the saturation temperature predicted from pressure measurement (see Fig. 16). 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 that closely approached thermodynamic equilibrium. In other words, the liquid mass or droplet surface temperature was 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 (see Fig. E.9 in Appendix E). However, the calculated void is also overestimated, because the actual slip ratio is greater than unity in dispersed flow.

Figure 16 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. The recovery rate was determined by the relative rate of vapor generation and discharge choked flow. A bubble-growth model was suggested to describe this period.¹⁰⁴ The saturation blowdown that followed lasted until 8 s, at which time the fluid temperature seemed to indicate significant subcooling. Physically this is not possible. Since the flow was adiabatic, the generation of a two-phase mixture at a pressure greater than the saturation pressure could not be due to subcooled boiling.¹¹⁵ Rather, nitrogen gas coming out of solution probably occurred. The continued lowering in system pressure facilitated the diffusion of the dissolved gas towards the liquid-vapor interface and its subsequent evaporation. As noted in Sec. V.B below, the contact time between Freon and nitrogen inside the supply vessel at 130 F (54°C) and 75 psia (0.5 MPa) had to be limited to a very short period (~10 min) 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).

2. Run 4/9-B

For Run 4/9-B, the initial peak in void indicated by the equilibrium calculation was again not measured by the void probe (see Fig. 18), and the same reasoning applies here.

As mentioned in Sec. IV.A, the blowdown fluid was initially at two different temperatures. In this earlier version of the test-loop design,¹¹⁶ the blowdown vessel had only one flexible tube joining the flowing loop; 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. 11, 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 (66°C)] in the test section and upper turbine leg. Figure 18 shows that this rise was well predicted by the equilibrium calculation. The void fraction dropped when the colder fluid [138°F (59°C)] 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 initial temperature of the fluid in the blowdown vessel.

Measurements at the lower turbine meter (see Fig. 18) showed a trend similar to that of 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 17 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 more than 75% during the period from 5 to 20 s. 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.

B. 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.^{111,117} Dissolved gases affect two-phase-flow heat transfer.^{114,118,119} The presence of gas increases the heat transfer in subcooled boiling.⁸ Gas bubbles can form at the surface and agitate the liquid in the same manner that vapor bubbles do. The first evolution of bubbles occurred at a much lower wall superheat in the presence of dissolved gas for a water-air system.¹²⁰ 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. Table VI shows the effect of a 1-hr contact interval between nitrogen and Freon in the supply vessel at 130°F (54°C) and 75 psia (0.5 MPa). 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.

TABLE VI. Effect of Prolonged Nitrogen Contact with Freon-11
[130°F (54°C)] at Elevated Pressure [75 psia (0.5 MPa)]^a

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, Hz	188.0	190.0	188.5	193.0	182.5	183.0	187.5	200.0
Volumetric flow, cfm	0.981	0.992	0.984	1.067	0.953	0.955	0.979	1.514
Mass flow, lbm/s	1.435	1.451	1.439	1.445	1.399	1.403	1.437	2.181
Agreement in mass flow, %	1.1		0.4		0.3		34	
Measured Void ^b	-	-	-	0.16	-	-	-	0.45
Inferred Void ^c	-	-	-	0.0	-	-	-	0.34

^aConversion factors: 1 cfm = 4.719×10^{-4} m³/s, 1 lbm/s = 0.4536 kg/s, 1 Btu/hr-ft² = 3.155 W/m².

^bCapacitance void measurement made 3 in. (7.6 cm) downstream of heated section.

^cCalculated void from turbine measurement according to $\epsilon_0 = 1 - (\rho_l Q_l) / (\rho_0 Q_0)$, where subscripts l and o stand for inlet and outlet locations.

First, the Freon was stored in the receiving vessel under saturation conditions at 10 psia (68.95 kPa) so that the Freon was not in contact with air during the storage period. Second, 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.^{110,111} Third, 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 VI, demonstrates the success of these measures.

C. Diabatic Blowdown

1. Preliminary Diabatic Tests

a. Run 4/22. Run 4/22 was conducted with an almost 1-s 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 the liquid phase only, the calculated temperature rise of the average bulk fluid was about 40°F/s (22°C/s), and if any part of the heater surface was experiencing CHF, a temperature excursion of 80°F/s (44°C) would result. The measured temperature rise of 7°F (3.9°C) indicates that CHF did not occur and the fluid did not remain in an all-liquid state. But part of the heat was used 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 about 8°F (4.4°C) in the saturation temperature, which lends support to the above argument. Vapor generation was evident from the upper void probe, which indicated a 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 the heat flux at low flow can be quite high without experiencing CHF as in pool boiling.^{65,80} For a horizontal flat plate in an unconfined pool boiling, Zuber¹²¹ suggested the following critical-heat-flux correlation:

$$\phi_{CHF} = \frac{\pi}{24} h_f g \rho_g^{1/2} [\sigma g (\rho_f - \rho_g)]^{1/4} \left(\frac{\rho_f}{\rho_f + \rho_g} \right)^{1/2}, \quad (2)$$

where for ϕ_{CHF} is 1.2×10^5 Btu/hr·ft² (3.8×10^5 W/m²) for Freon-11 at 65 psia (0.45 MPa) and 1.2×10^6 Btu/hr·ft² (3.8×10^6 W/m²) for water at 1500 psia (10 MPa).

As shown above, this heat flux is at least three times as large as the operating heat flux. Even though the above equation is not applicable to

the present confined geometry. Walkush¹²² and Avedisian et al.⁹¹ 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. 20 indicates that CHF occurred at about 2 s into blowdown, at which time the upper void measurement had a sharp rise to 100%, indicative of a fluid-depletion condition. Many wall thermocouples were defective due to improper installation and were therefore replaced. The most severe temperature rises were at the lower end, where CHF seemed to occur first, although an equally early CHF onset was found at TC 18 [6 in. (15 cm) from the top], but this was rewet after ~1 s.

b. Runs 5/19A and 5/29. For Runs 5/19A, 5/29, 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. In the three runs taken with high-speed photography, this reversal period was observed to last 50-60 ms. 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 (see Fig. 25). A void fraction of over 90% indicated that the channel was nearly depleted of liquid coolant, and as a result, "dryout" occurred. The propagation of the dryout front for Run 5/19A was quite smooth, as shown in Fig. 26.

In Run 5/19A, a very small temperature excursion appeared at TC 1 [1 in. (2.54 cm) from the lower end] shortly after flow reversal, but rewetting occurred in ~1 s. In Run 5/29, inception of CHF occurred twice (see Fig. 28). The early temperature excursion was more pronounced as temperatures above 400°F (204°C) were measured. However, only the bottom 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 ms for Run 5/29), as shown in Fig. 32. 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 (see Fig. 30), indicating flow of high-quality vapor. Further, the mass remaining in the blowdown vessel, as sensed by the weight transducer, was nearly zero at this time. This "dryout" front followed the same kind of locus as in Run 5/19A (see Fig. 32), but the rate of propagation was faster. Unlike the second CHF, the first CHF requires further discussion (see Sec. V.C.3 below).

2. Effect of Bubble Growth in the Decompression Process

As mentioned in Sec. V.A. the bubble-growth rate affected the initial decompression characteristic. This is understandable, since the system pressure would drop to the ambient pressure if it were not for the inception of vaporization of the hot liquid. For 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 growth as a result of the thermodynamic nonequilibrium state of the fluid has been reported in other adiabatic blowdown experiments.^{82,104} 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 Figs. 58 and 59. For the diabatic case, the system pressure dropped to the saturation pressure corresponding to the exit fluid temperature in the upper plenum, whereas 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, since a slug-flow model had been used in the interpretation of the measurement.

Fig. 58. Pressure Histories for Adiabatic and Diabatic Blowdowns. Conversion factors: 1 psi = 0.895 kPa; $t(^{\circ}\text{C}) = [t(^{\circ}\text{F}) - 32]/1.8$; 1 ft/sec = 0.305 m/s; 1 Btu/hr-ft² = 3.155 W/m². ANL Neg. No. 900-75-423 Rev. 1.

Fig. 59. Void Fractions for Adiabatic and Diabatic Blowdowns. ANL Neg. No. 900-75-421.

The success of this prediction is further demonstrated in the diabatic blowdown Run 6/12 (see Fig. 49), 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. (7.6 cm) 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.

3. Diabatic Tests with High-speed Photography

The results of the three diabatic blowdown tests were presented in Figs. 34-51. Selected prints showing the flow regime during the early blowdown phase are presented in Figs. 52-54. 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 ms. The upflow decayed as soon as depressurization started. The selected prints show that, as the flow slowed down, bubble density on the heating surface increased rapidly. Vigorous vaporization occurred during downflow, as seen from prints at 80 ms, and the high-void mixture was swept out of the heated channel at an accelerating speed, up to 23 ft/s (7 m/s) (see Figs. 55-57). The lower-void probe showed an immediate jump to high void fraction, qualitatively in agreement with the results of high-speed photography. However, this high-velocity flow dropped off at around 250 ms, leading to the development of an annular flow pattern, which, judging from the pictures, lasted until 500 ms (see Figs. 52-54).

During this period, considerable slip between liquid and vapor phases was observed; roll waves frequently passed along, while a few smaller droplets traveled down the core at a much higher velocity. Wavy liquid films adhered to the outer unheated wall, although it is impossible to judge 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 experienced at around 300 ms (see Table V) coincided with the time of appearance of annular flow. Thus, the heater wall where pictures were taken was inferred to be dry, and hence the temperature excursion. Considering the uncertainty in determining the time to CHF, we can conclude 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 ms was uncertain when judged 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 probably 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 conduction¹²³ occurred during the preliminary test, Run 5/29, as the liquid rewetting front that existed above the 8-in. (20-cm) location propagated downward (see Fig. 33). Rewetting also took place at the 8- and 6-in. (20- and 15-cm) locations in Run 6/9 and at the 10-in. (25-cm) location in Run 6/12 before power trip (see Figs. 41 and 48).

At about 100 ms, the upper-plenum pressures all showed spikes. This was observed after the valving adjustment, as mentioned in Sec. IV.C. 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:⁸

$$\Delta P_T = \Delta P_M + \Delta P_{fric} + \Delta P_z. \quad (3)$$

Near the lower end of the annular channel, the momentum pressure drop dominates, as shown in Fig. 46. For low-quality flows ($x \approx 0$), the momentum pressure drop in a constant-area channel may be approximated¹¹⁵ by

$$\Delta P_M = \frac{G^2}{\rho_f} \left(\frac{1}{1 - \alpha_2} - \frac{1}{1 - \alpha_1} \right), \quad (4)$$

where α_2 and α_1 are the void fractions 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 (see Fig. 46), indicative of acceleration. Exception was found during the interval 300-500 ms, when a more linear profile was observed. This coincides with the observance of the annular flow of low velocity (see Figs. 55-57).

Because the steepness of the pressure profiles is characteristic of the region upstream from a choking location,¹¹⁵ the possibility of choking cannot be overlooked. The results of the critical-flow-rate calculation are presented in Appendix K. The calculation was made at 200 ms, since the velocity then was observed to be highest. The flow decay promoted at about 300 ms 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 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 1.98 cm^2 , followed by an abrupt expansion to a 2.22-cm-ID tube. The homogeneous, nonequilibrium, compressible-flow equation can be approximated for saturated liquid in terms of the critical pressure ratio η as¹²⁴

$$G = [2\rho_f \mathcal{P}(1 - \eta)]^{1/2}, \quad (5)$$

where $\mathcal{P}$ 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.¹²⁴ With Eq. 5, a volumetric discharge rate of $0.069 \text{ ft}^3/\text{s}$ ($0.021 \text{ m}^3/\text{s}$) was calculated at the throat, i.e., the cross. This may be compared with the vapor generation rate in the heated channel as given by

$$Q_v = \frac{\varphi \pi D L h}{h_{fg}} (v_g - v_f). \quad (6)$$

A value of $0.066 \text{ ft}^3/\text{s}$ ($0.020 \text{ m}^3/\text{s}$) was calculated alone using Eq. 6, neglecting the effect of flashing caused by pressure drop and possible condensation in the lower plenum. The actual rate of vapor generation 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.⁷¹ in the small-break blowdowns. The rise in lower-plenum pressure then caused the flow retardation observed at 300 ms. 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 rate of vapor generation was lowered. Subsequently the velocity picked up, as observed in the high-speed photography, because of the reduced vapor production and the increasing volumetric flow rate 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 rate of mass decay 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 s 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 used 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 XIII (in Appendix L) 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 ms was at a CHF of 0.91, 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 s in comparison with Runs 6/4 and 6/9, for which no complete rewetting occurred (case 2 versus case 5). The corresponding CHF's 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 CHF of 0.91 at 20 s (case 3), but was not observed. Most probably the inaccuracy of the correlation increases with decrease in pressure and the density ratio decreases sharply with pressure below 50 psia (345 kPa), as shown in Fig. J.1. The late (second) CHF occurred at a CHF of 0.79 only (case 4). Hence, because of the aforementioned difficulties, we cannot conclude from the above discussion that steady-state CHF correlation is conservative when used in a transient situation.

VI. SUMMARY AND CONCLUSIONS

Experimental data have been presented for the blowdown of a small-scale facility using Freon-11 at 130°F (54°C) and 65 psia (448 kPa). 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 (54°C) and 75 psia (0.5 MPa) greatly affected the steady-state subcooled-boiling initial-upflow conditions. Nitrogen gas was exiting from solution in the heated zone. Special precaution had to be taken to ensure that 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-ms reversal period.
4. The first or early CHF, which occurred at about 400 ms, 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 third 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.

VII. RECOMMENDATIONS FOR FUTURE STUDIES

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

1. Due to the slow-response nature of the strip-chart recorders, the present recording system for void and turbine meter is not adequate. Visi-corder or magnetic-tape recording units will be used to record all transient signals.

2. The solenoid valves (SW1, SW2, and SW3) will have to be synchronized to avoid nonreproducible results. Rupture-disk assemblies may offer good alternatives.

3. Pressure measurement should be made downstream of the test section to determine the exact choking plane during blowdown.

4. The difference between inlet and outlet breaks 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 should be investigated.

6. Other test-section geometries can be studied to determine the effect on flow regime and CHF.

7. A test section capable of operating at higher pressures and temperatures can be installed.

APPENDIX A

Data for Single-phase Friction Pressure Drop

Pressure drop and fluid friction associated with the isothermal flow of water at 70-190°F (21-88°C) 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 a pressure drop down to 0.0004 psi (0.003 kPa). Two 60-in. (152-cm) U-tube manometers using 1.75 and 2.95 M.F. were mounted vertically to measure pressure differential across the 24-in. (61-cm) heated length. Mass flow rate was determined by the weighing-tank method.

The inside diameter of the commercially available 1-in. (2.54-cm) 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 was 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

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

where

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

ΔV = volume of water added, in.³,

and

Δh = rise in water level, in.

The following results were obtained:

	<u>d_o, in. (cm)</u>	<u>Standard Deviation, in. (cm)</u>
TS No. 1, first pipe	1.023 (2.598)	0.007 (0.018)
TS No. 2, second pipe	1.022 (2.596)	0.007 (0.018)

The laminar-flow velocity profile in annuli is well known.¹²⁵ The relationship between the Moody friction factor and Reynolds number is given by

*The first Pyrex pipe used in obtaining the 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.

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

where

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

and

$$\kappa = d_i/d_o, \text{ diameter ratio.}$$

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

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

and

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

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 $\kappa = 1$, i.e., parallel plates of infinite width.

Experimentally, the Moody friction factor is evaluated according to¹²⁶

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

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

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

the experimental laminar data yielded an average $f \cdot Re_h$ value of 95.93. The equation obtained by Nikuradse,¹²⁷ applicable over the entire turbulent Reynolds number range for smooth tubes, is

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

which is plotted in Fig. A.1. The experimental data clearly show that, for Reynolds number based on the hydraulic diameter, the friction data lie considerably above the smooth tube, the difference being 10-20%. Similar results were obtained in a recent survey of past literature,¹²⁸ and it was concluded that the hydraulic diameter is not the proper length dimension to use for the Reynolds number to ensure both geometric and hydrodynamic similarity between circular tubes and annular ducts. Jones¹²⁹ proposed a modified Reynolds number, Re^* , which yields a friction-factor relation as identical with that obtained for a round tube in laminar flow:

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

This Reynolds number also provides good similarity in fully developed turbulent flow for smooth rectangular ducts, as demonstrated by Jones. If Eqs. A.8 and A.2 are compared, 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 $Re > 10^4$ yield 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 VII.

Fig. A.1. Friction-factor Results Based on Hydraulic-equivalent Reynolds Number. ANL Neg. No. 900-75-806 Rev.1.

Fig. A.2. Friction-factor Plot Based on Modified Reynolds Number. ANL Neg. No. 900-4837 Rev. 1.

TABLE VII. Frictional Pressure Drop for Water in Test Section No. 1^a

Run	Temp. °F	Mass Flow, lb _m /s	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	0.5456	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	1.6131	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

TABLE VII (Contd.)

Run	Temp. °F	Mass Flow, lb _m /s	Pressure Drop (24 in.), psi	Re _h	Re*	f
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	36150	0.02249
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	158.0	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

*Conversion factor: $t(^{\circ}\text{C}) = [t(^{\circ}\text{F}) - 32]/1.8$; $1 \text{ lb}_m/\text{s} = 0.4536 \text{ kg/s}$; $1 \text{ in.} = 2.54 \text{ cm}$; $1 \text{ psi} = 6.895 \text{ kPa}$.

Limited isothermal friction data were 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 VIII. Agreement with the circular-tube equation based on the modified Reynolds number is good, as shown in Fig. A.3.

TABLE VIII. Frictional Pressure Drop for Freon-11 in Test Section No. 2^a

Run	Temp, °F	Mass Flow, lb _m /s	Pressure Drop (24 in.), psi	Re _h	Re*	f
1	68.7	0.385	0.144	10920	7290	0.0353
2	69.2	0.910	0.612	26050	17380	0.0270
3	69.6	1.601	1.676	45960	30670	0.0238
4	69.2	1.912	2.402	55010	36710	0.0239
5	67.3	2.278	3.092	65400	43640	0.0217

^aConversion factors: $t (^{\circ}\text{C}) = [t (^{\circ}\text{F}) - 32]/1.8$; $1 \text{ lb}_m/\text{s} = 0.4536 \text{ kg/s}$; $1 \text{ in.} = 2.54 \text{ cm}$; $1 \text{ psi} = 6.895 \text{ kPa}$.

Fig. A.3. Isothermal Freon-11 Friction Data.
ANL Neg. No. 900-75-805 Rev. 1.

APPENDIX B

Data for Single-phase Liquid Heat Transfer

A simple macroscopic heat balance on the heated test section (length = L_h) results in

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

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

Fig. B.1. Results for Single-phase Heat Balance. Conversion factor:
1 Btu/hr = 0.293 J/s. ANL Neg. No. 900-75-385 Rev. 1.

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

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

where

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

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

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

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.

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:

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

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 Pelton¹³⁰ recommended

$$\frac{h_{fo} D_e}{k_f} = 0.020 \left(\frac{\epsilon U D_e}{\mu_f} \right)^{0.8} \left(\frac{C_p \mu_f}{k_f} \right)^{0.43} \left(\frac{D_o}{D_i} \right)^{0.53} \quad (B.6)$$

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

Wiegand¹³¹ also suggested a correlation that gives results close to Eq. B.6. For the present geometry, the product $0.020 (D_o/D_i)^{0.53}$ is 0.023, which is the same constant as in the Dittus-Boelter equation.

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

Fig. B.2
Results for Single-phase Heat Transfer
ANL Rep. No. 44-75-32 Rev. 1

Finally, some isothermal upward flows of Freon at 135°F (57°C) 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 (37 J/s) and an overall heat-transfer coefficient of about 2 Btu/hr·ft²·°F (11 W/m²·°C), which compares favorably with the calculated value 0.93 Btu/hr·ft²·°F (5.3 W/m²·°C) based on free convection from the pipe. The presence of copper clamps at each pressure-tap location may conduct some heat away, thus enhancing the heat loss.

Temperatures along the heated test section were measured by 0.031-in. (0.079-cm)-OD stainless steel, ungrounded, sheathed, Chromel-Alumel (Type K) thermocouples. Some possible errors in temperatures are:

1. Inhomogeneity of thermocouple wires.
2. Stray emf pickup in the leads.
3. Drift of the reference-junction units.
4. Inaccuracy of amplifier units, such as calibration drift and nonlinearity output.
5. 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

1. Steady-state Consideration

In the absence of axial heat conduction, the temperature difference between the inside and outside of the heated tube can be calculated quite straightforwardly. 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

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

and

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

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 q

$$T_{wo} = T_{wi} - \frac{qr_1}{k} \left(\frac{1}{2} - \frac{r_2^2}{r_1^2 - r_2^2} \ln \frac{r_1}{r_2} \right). \quad (C.5)$$

At a heat flux of 50,000 Btu/hr-ft² (157.75 kW/m²), ΔT_w is ~6°F (~3.3°C).

2. Unsteady-state Consideration

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

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

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

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

and

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

and the initial condition is

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

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

The problem above has been referred to as the "inverse-heat-conduction" problem in the literature.¹³²⁻¹³⁴ 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 have been categorized into four methods:

1. Integral-equation approach.¹³⁵
2. Laplace or Fourier transform.¹³⁶
3. Infinite-series technique.¹³⁴
4. Nonlinear estimation technique.¹³³

The above equations cannot be viewed as a good description of the real situation at a 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 is significant at dryout location.¹¹³

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

Determination of Thermocouple Response1. Out-of-loop Determinationa. 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 by¹³⁷

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

Equation D.1 holds when the Biot number is small ($h_0 d/k < 0.1$);¹³⁸ 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 (TC), 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:

$$\tau = \frac{(\rho V C_p)_{jn}}{h_0 A_{bd}}, \quad (D.2)$$

where the subscripts jn and bd represent the junction and bead, respectively. Therefore, for a given thermocouple, $\tau \propto 1/h_0$.

b. Experiment

The test was conducted using a beaker of boiling water at 212°F (100°C). Heating was supplied by a hot plate, and good agitation was ensured by the boiling process. The test thermocouple was allowed to stabilize at room temperature and then 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 IX. Because the time constant varied from one thermocouple to another, the approximate range is given.

TABLE IX. Time Constant of Thermocouples^a

Type	Application	τ , ms
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

^aConversion factor: 1 in. = 2.54 cm.

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 the inception of CHF decreases by at least two orders of magnitude. Therefore one way to determine the overall time response of the heated tube and thermocouples was as follows.

With no coolant on the heat-transfer side, so as to simulate "dryout" or vapor-blanketing condition, the test section was suddenly subjected to full-power load as used in the transient. The time response τ_{TS} was then taken to be the time delay until the thermocouple temperature began to rise. To avoid physical damage, power to the test section was tripped in less than 2 s. Results are shown in Figs. D.1 and D.2, and the corresponding time response τ_{TS} is given in Table X. The estimated temperature excursion was also plotted based on both the temporal-resistance measurement and an adiabatic-heatup calculation.

Fig. D.1. Temperature Response for Lower-half Test-section Thermocouples. Conversion factor: $t(^{\circ}\text{C}) = [t(^{\circ}\text{F}) - 32]/1.8$. ANL Neg. No. 900-75-365 Rev. 2.

Fig. D.2. Temperature Response for Upper-half Test-section Thermocouples. Conversion factor: $t(^{\circ}\text{C}) = [T(^{\circ}\text{F}) - 32]/1.8$. ANL Neg. No. 900-75-366 Rev. 1.

TABLE N Overall Time Response of Test Section
and Thermocouples

[Heat flux = 45 500 Btu/hr ft² (143 55 kW/m²)

TC No.	τ_{TS} , ^a s	TC No.	τ_{TS} , ^a s
1	0.2	14	0.8
2	0.3	16 ^b	-
4	0.9	18	0.2
6	0.3	20	0.2
8	0.2	22	0.2
10	0.3	23	0.3
12	0.7		

^aDue to uncertainty, τ_{TS} values rounded off to first-place decimal.

^bTC 16 was detached from the wall.

The resistivity of Type 304 stainless steel, ρ_r , is a function of temperature, and the following equation was obtained based on the values given in Ref. 139:

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

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/s (47°C/s).

Fig. D.3. Temporal Voltage, Current, and Resistance Measurement during Rapid Joule Heating. ANL Neg. No. 900-75-367.

If no axial conduction and no radial temperature gradient are assumed, the adiabatic wall analysis yields

$$\frac{\Delta T}{\Delta t} = \frac{q}{\rho C_p \xi}, \quad (\text{D.5})$$

where ξ is the thickness of the tube. This value was calculated to be 96°F/s (53°C/s).

As evidenced from Figs. D.1 and D.2, not all the thermocouples followed the estimated temperature rise. In fact, those with long τ_{TS} 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. In addition, the exact location of the bead from the thermocouple sheathed end was not known. As a result, some beads were placed further away from the heated wall, giving rise to slower responses due to increased depth and lower temperature measurement due to excessive conduction loss. Hence for transient temperature measurement, the exact location of the thermocouple bead is important, and this fact is more obvious in the discussion of "inverse heat conduction" in Appendix C.

APPENDIX E

Calibration of Capacitance-void Meter1. Void-measurement Instrumentation

The geometry and dimension of the void probe is outlined in Sec. III. B.2. A bench setup 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; in this case, at both empty and full positions distortion was caused by a fringe effect of the outside electrode.

Fig. E.1. Vertical Calibration Using Impedance Bridge. Conversion factor: 1 in. = 2.54 cm. ANL Neg. No. 900-75-510.

Fig. E.2. Horizontal Calibration Using Impedance Bridge. ANL Neg. No. 900-75-517 Rev. 1.

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 checking for linearity in the dc output. The result, shown in Fig. E.6, indicates that the bridge did provide linear response as desired in the range of interest.

2. Interpretation of Void Measurement

If the dc outputs or capacitance measurements at both empty and full positions are known, and if the calibration is linear as in Fig. E.5, the apparent void fraction is found from

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

which is a direct result of Eq. E.4 below 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.

Fig. E.3

Capacitance Bridge for Transient Void Measurement. ANL Neg. No. 900-75-514 Rev. 1.

Fig. E.4
Vertical Calibration Using Capacitance Bridge. Conversion factor: 1 in. = 2.54 cm. ANL Neg. No. 900-75-512 Rev. 1.

Fig. E.5

Horizontal Calibration Using Capacitance Bridge. ANL Neg. No. 900-75-511 Rev. 1.

Fig. E.6. Output-voltage Characteristic of Capacitance Bridge. ANL Neg. No. 900-75-516.

Fig. E.7. Void-fraction Determination Using Capacitance Bridge. ANL Neg. No. 900-75-509 Rev. 1.

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 are made in the rest of this appendix.

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

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

where

A = total effective area of electrode

and

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.¹⁴⁰

a. Slug-flow Model (see Fig. E.8)

The two phases are in parallel between the electrodes; therefore

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

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 $\epsilon_{2\phi}$, we obtain

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

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

Fig. E.8. Various Idealized Two-phase-flow Models. ANL Neg. No. 900-75-506.

b. Bubbly-flow Model

To find the total dielectric constant of a two-phase mixture with one phase dispersed as small spheres, Maxwell¹⁴¹ derived the approximate formula

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

Fig. E.9. Theoretical Void vs Apparent Void. ANL Neg. No. 900-75-340 Rev. 1.

where

β = volume fraction of dispersed phase,

ϵ_c = dielectric constant of continuous phase,

and

ϵ_d = dielectric constant of dispersed phase.

Equation E.7 holds if the dispersed spheres are separated by distances large in comparison with their diameters.

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 is overestimated in comparison with the theoretical void, since the quantity in parentheses is greater than one.

c. Dispersed-flow Model

From the discussion in Sec. b, Eq. E.7 can be written as

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

or

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

which implies that $(1 - \alpha_a) < (1 - \alpha)$ or $\alpha_a > \alpha$. Again the apparent void is overestimated in comparison with the theoretical void.

d. Film-flow Model

By considering the liquid and vapor phases as capacitances in series, we obtain

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

and

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

which says that the apparent void is overestimated in comparison with 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 in Fig. E.9. Dielectric constants at 70°F (21°C) for Freon-11 were used in the calculation. Also, the effect of the glass shroud has been included.

3. 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 my knowledge, no complete information has been obtained for Freon-11, although Du Pont¹⁴² suggested an empirical way of estimating the ϵ_f and ϵ_g values at various temperature. Hence, at present, 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;¹²⁴ i.e., the system has an isentropic process for which

$$dS_0 = d[(1+x)S_f + xS_g] = 0. \quad (F.1)$$

Fig. F.1. Generalized Temperature-Entropy Diagram. ANL Neg. No. 900-75-923.

The blowdown from a subcooled state follows the constant-entropy path as shown in Fig. F.1. As blowdown proceeds, the temperature continues 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_0 , as shown in Fig. F.1. Therefore, at any time into blowdown, the following equation holds, assuming thermodynamic equilibrium between phases:

$$S_0 = (1-x)S_f + xS_g \quad (F.2)$$

or

$$S_0 = S_f + xS_{fg}. \quad (F.3)$$

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

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

where

$$\mathcal{S} = \frac{u_g}{u_f}; \text{ slip ratio} = 1.0 \text{ for homogeneous flow model.}$$

The 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 Sec. III.A.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 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 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

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

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

$$\alpha_2 = 1 - \left(\frac{W_f}{\rho_f AU_f} \right)_2. \quad (G.4)$$

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

APPENDIX II

Measurement of the Power Factor

The power dissipated by an electric heater is given by

$$P = VI \cos \Theta, \quad (\text{H.1})$$

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 is the voltage drop across the shunt) to the Y input. Properly adjusting the gain to give equal amplitude input generated a Lissajous figure, 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 (10.08-cm) Type 304 stainless steel tube are shown in Fig. H.2. The average value of $\cos \Theta$ is 0.993.

Fig. H.1. Determination of Phase Angle Using Lissajous Figure. ANL Neg. No. 900-75-508.

Fig. H.2. Measured Phase Angle vs Test-section Voltage. ANL Neg. No. 900-75-515.

The instantaneous power for ac heating is

$$p = I_m V_m \sin(\omega t + \Theta) \sin(\omega t). \quad (\text{H.2})$$

where I_m and V_m are the amplitudes of current and voltage, respectively. The average power can be written as

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

Hence, with Eq. H.1, root-mean-square (rms) values are to be used. However, measurement by digital voltmeter and Honeywell strip-chart recorder gives the mean or average value.

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

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

and

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

If the shapes of the ripples and the dc component are known, the average power is given by

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

where, for $\theta < 10^\circ$,

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

In terms of measured values (average),

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

Define a power factor ψ such that

$$P = V_{\text{meas}} I_{\text{meas}} \psi. \quad (H.8)$$

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

Fig. H.3. Amplitude of Ripple vs Test-section Voltage. ANL Neg. No. 900-75-513

Fig. H.4
Power Correction Factor vs Test-section Voltage. ANL Neg. No. 900-75-507.

APPENDIX I

Instrument Calibration

Calibrations of the weight transducers, turbine flowmeters, and pressure transducers are discussed in this appendix.

1. Weight Transducers

The two Statham UC3 unbonded-type strain-gauge, Universal Transducing cells were calibrated with dead weights. The millivolt output was read on an HP DVM Model No 2401C. Hysteresis was less than 0.5%. Calibration results are shown in Figs. I.1 and I.2.

Fig. I.1. Calibration of Supply-vessel Weight Transducer. Conversion factor: $1 \text{ lb}_m = 0.4536 \text{ kg}$. ANL Neg. No. 900-75-571.

Fig. I.2. Calibration of Blowdown-vessel Weight Transducer. Conversion factor: $1 \text{ lb}_m = 0.4536 \text{ kg}$. ANL Neg. No. 900-75-575.

2. 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 (27°C) by the manufacturer before shipment. From the data so obtained, the mean cycles/gallon "K" factor was calculated. A quick check of 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 an HP Frequency Counter, Model No. 5202L. Agreement in K values was within 4%, mainly due to inaccurate experimental measurement. The K factor for 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 by Cox showed better agreement with the weight measurement; the result is shown in Fig. I.3. Thus the result gave a good and confident check on the weight and flow measurements.

3. Pressure Transducers

Absolute pressure transducers were calibrated using pressurized nitrogen or air, and the pressure was read with a 100-in. (2.54-m) well-type mercury manometer [i.e., able to read below 65 psia (0.45 MPa)]. Hysteresis was less than 0.1% full scale for 1000-psig (6.9 MPa) transducer used at the upper plenum.

Differential pressure transducers were calibrated in more or less the same way on both high- and low-pressure sides. A 60-in. (152-cm) U-tube manometer with a 2.95 Meriam fluid was used to read any pressure less than 5 psi (34 kPa). Calibrations of the high- and low-pressure sides of each DP cell agreed to within 1%, and hysteresis in general was small (see Fig. I.4) for one calibration. Drift in the calibrations was also negligible during a test period of 4 months.

Fig. I.3. Agreement between Two Types of Flow Measurements. Conversion factor: $1 \text{ cfm} = 4.710 \times 10^{-4} \text{ m}^3/\text{s}$. ANL Neg. No. 900-75-573.

Fig. I.4. Calibration of a 100-psid (689.5-kPa) Transducer. Conversion factor: $1 \text{ in.} = 2.54 \text{ cm}$. ANL Neg. No. 900-75-572 Rev. 1.

APPENDIX J

Freon-11-to-Water Scaling

As mentioned in Sec. III.A.1, the use of Freon-water scaling for steady-state CHF has met with numerous success. Although such modeling techniques can be used to only a limited extent 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 appears in Refs. 143 and 144. Here three existing scaling laws are compared.

1. Zuber and Ishii

In the method of Zuber and Ishii, 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. If thermodynamic-equilibrium flow is assumed, the following dimensionless groups were obtained:

$$\rho_f/\rho_g = \text{ratio of liquid to vapor density,}$$

$$L_h/D_e = \text{ratio of heated length to diameter,}$$

$$U_{gj}/U_{fi} = \text{ratio of vapor drift velocity to liquid inlet velocity;}$$

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

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

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

$$N_{PCH} = \frac{\Gamma_g v_{fg} L}{U_{fi}} = \text{phase-change number,} \quad (J.4)$$

where

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

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 XI. Freon conditions during the tests at steady state before initiation of blowdown were used.

Fig. J.1. Density Ratio for Freon-11 and Water. Conversion factor: 1 psia = 6.895 kPa. ANL Neg. No. 900-75-368.

Rearrangement of the groups L_h/D_e , Re , and Fr leads to gD_e^3/ν_f^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 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 in which 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.¹⁴⁷ The results are shown in Table XII.

2. Tong et al.¹⁴⁴

Tong et al. suggested some dynamic scaling expressions in subcooled and low-quality flow. The physical model pictured

Fig. J.2
Density Ratio vs Reduced Pressure Conversion factor: 1 psia = 6.895 kPa. ANL Neg. No. 900-75-914.

TABLE XI. Thermodynamic and Transport Properties^a

Parameter	Freon-11	Water
P , psia	65	430
T_{SAT} , °F	165	452
ρ_f , lb _m /ft ³	84.37	51.39
ρ_g , lb _m /ft ³	1.485	0.93
ρ_f/ρ_g	56.0	55.5
μ_f , lb _m /hr·ft	0.68	0.29
μ_g , lb _m /hr·ft	0.0311	0.0394
h_{SAT} , Btu/lb _m	42.73	432.2
h_{fg} , Btu/lb _m	68.68	772.6
ν , ft ² /lb _f	6.25×10^{-3}	1.00×10^{-3}
C , hr·ft ^{5/3} /lb _m	9.40×10^{-5}	6.99×10^{-5}

^aConversion factors: 1 psia = 6.895 kPa; t (°C) = [t (°F) - 32]/1.8; 1 lb_m/ft³ = 16.0 kg/m³; 1 lb_m/hr·ft = 4.134×10^{-4} N·s/m²; 1 Btu/lb_m = 2326 J/kg; 1 ft²/lb_f = 0.021 m²/N; 1 hr·ft^{5/3}/lb_m = 1.10 s·m^{5/3}/g.

TABLE XII. Results of Freon-11 Water Scaling^a

	Freon-11 (present study)	Water Equivalent Conditions		
		Zuber's Criteria	Ahmad's Criteria	Reactor Condition (PWK)
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_j , °F	130	370	370	550
Δh_f , Btu/lb _m	7.71	89.2	86.5	151.0
U_{ff} , ft/sec	7.5	7.5	17.3	15.0
G , lb _m /hr·ft ²	2.37×10^6	1.39×10^6	3.20×10^6	2.5×10^6 (avg)
q , Btu/hr·ft ²	45,000	316,000	684,000	200,000 (avg) 580,000 (max)
Reynolds number	63,900	87,700	202,300	502,000
Froude number	1.14	1.14	0.21	1.69

^aConversion factors: 1 psia = 6.895 kPa; 1 in. = 2.54 cm; 1 ft = 30.48 cm; t (°C) = [t (°F) - 32]/1.8; 1 Btu/lb_m = 2326 J/kg; 1 ft/s = 0.305 m/s; 10^6 lb_m/hr·ft² = 1356 kg/m²·s; 1 Btu/hr·ft² = 3.155 W/m².

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 the boiling number:

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

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

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

Based on Tong's two-phase-flow friction factor¹⁶

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

and his boundary-layer analysis at CHF (DNB), the following expression was derived:

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

or

$$\dot{q}_{CHF} = Ch_{fg} G_f^{0.4} \left(\frac{\mu_f}{D_e} \right)^{0.6} \quad (J.9b)$$

where C is a function of reduced pressure and the properties of the subcooled 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 geometrically 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 a water-equivalent pressure of about 1400 psia.

3. Ahmad¹⁴³

In Ahmad's scaling, the following parameters were first identified

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

Applying the classical dimensional analysis and Buckingham's Pi theorem produced one possible set of 12 dimensionless groups.

Obviously all 12 conditions cannot be satisfied; therefore, some have to be distorted. The "Compensated Distortion Model" introduced a CHF modeling parameter ψ :

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

where

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

and

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

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 a plot of ϕ_{CHF} / Gh_{fg} versus ψ_{CHF} were on the same curve. It was found to be successful with Freon-12, -113, -11, -21, and -114, CO_2 , and potassium for circular, annular, and rod-bundle geometries. The result using Ahmad's scaling is shown in Table XII.

4. Conclusion

Zuber's U_{gj} / U_{fi} group does not enter into the scaling procedure. This group is characteristic of the flow regime and 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,^{146,147} but had not been attempted to scale steady-state CHF data. The power-reduction factors in Zuber and Ahmad's scaling are 7 and 15, respectively. The discrepancy is certainly due to different approaches 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 Bergles¹⁴⁷ in a low-pressure system, e.g., Freons. At low pressure, where the pressure drop across the test section is large, and hence $\Delta \mathcal{P}_{TS} / \mathcal{P}$ is significant, the flashing and vapor expansion increase the velocity gradient, which is dynamically 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 the scaling approach.

APPENDIX K

Calculation of Critical Flow Rate

As a first approximation for low-quality critical flow, the following relation is used:¹¹⁵

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

where if we set $x = x(P(z))$ the derivative dx/dP may be expressed as the quotient $(dx/dz)/(dP/dz)$ and approximated by $(\Delta x/\Delta z)/(\Delta P/\Delta z)$. The derivative dv_g/dP may be evaluated using thermodynamic data.

The data of Run 6/9 at 200 ms were taken to make the following calculation since both pressure profile (see Fig. 46) and velocity (see Fig. 56) were available.

Time: 200 ms

Pressure at upper channel: $P_i = 53.0$ psia (365 kPa)

Pressure at $z = 6$ in. (15 cm): $P = 50.5$ psia (348 kPa)

Pressure at lower channel: $P_o = 48.0$ psia (331 kPa)

Void at upper channel: $\alpha_i \approx 0.10$

Void at lower channel: $\alpha_o \approx 0.80$

Velocity at lower channel: $U_o \approx 22$ ft/s (6.7 m/s)

Uniform heat flux: $\phi = 46,300$ Btu/hr·ft² (146 kW/m²).

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

a. If the homogeneous-flow model and $\alpha_i \approx 0.10$ are assumed,

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

A similar calculation was performed at the bottom 6-in. (15-cm) unheated length. The quality of the exiting end was evaluated using an isentropic calculation, and the derivatives were estimated in the same manner. The critical mass velocity at $z = -6$ in. (-15 cm) is

$$G_c = 3.1 \times 10^6 \text{ lb}_m/\text{hr}\cdot\text{ft}^2 \text{ (4203.6 kg/m}^2\cdot\text{s)};$$

$$G/G_c = 0.46.$$

APPENDIX L

Calculation of Critical Heat Flux

Two CHF modeling approaches, namely, those of Tong¹⁴⁴ and Ahmad¹⁴³ have been outlined in Appendix J. An attempt to use the former approach met with difficulty because the empirical scaling factor for Freon-11 was given for pressures of greater than 250 psia (1.72 MPa),¹⁴⁴ which are much higher than those involved in the present investigation. The result of the latter approach was summarized in Table XII. By knowing the water-equivalent conditions, we can estimate the operating CHF level, using existing correlations developed for water. Unfortunately, to my knowledge, no water CHF data for annuli have been reported at around 400 psia (2.76 MPa). However, a CHF correlation for annular geometries that is independent of fluid types was developed from Freon-12 data.¹⁴⁸ 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

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

$$\rho^* = \ln(\rho_f / \rho_g), \quad (L.3)$$

and

$$E^* = \ln(\psi_{CHF} L / D_c x_i). \quad (L.4)$$

The resulting correlation obtained with Freon-12 data between 88 and 244 psia (0.61 and 1.68 MPa) ($0.15 < P_r < 0.41$), is

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

where

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

Equation L.6 was used to calculate the critical heat flux during steady state and transient as summarized in Table XIII. 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; hence, the correlation, which is in terms of the inlet subcooling or quality, is bound to be less accurate.

TABLE XIII. Summary of Estimations of Critical Heat Flux^a

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 and 6/9	5/29, 6/4, 6/9, and 6/12
Time, s	<0	3.0	20.0	35.0	3.0	-0.3
P , psia ^b	65	48	43	40	48	50
TSAT, °F	165	144	137	132	144	147
ρ_f , lb _m /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
u_f , lb _m /hr·ft	0.68	0.743	0.765	0.782	0.743	0.730
u_g , lb _m /hr·ft	0.0311	0.0305	0.0301	0.0298	0.0305	0.0305
h_{fg} , Btu/lb _m	68.68	70.98	71.69	72.19	70.98	70.68
v , ft ² /lb _f	0.0063	0.0117	0.0145	0.0167	0.0117	0.0105
C , hr·ft ^{5/3} /lb _m	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 , lb _m /hr·ft ²	2.4×10^6	1.55×10^{6c}	1.06×10^{6c}	0.55×10^{6c}	1.36×10^{6c}	1.0×10^{6d}
$\dagger$ CHF	59.3	46.54	33.67	18.1	42.08	29.0
x_i	-0.10	-0.0 ^e	0.01 ^e	0.02 ^e	-0.0 ^e	-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 ^f	1.75	1.01	0.91	0.79	0.94	0.91
Observation	No CHF	No CHF	No CHF	CHF	CHF	CHF

^aConversion factors: 1 psia = 6.895 kPa; $t(^{\circ}\text{C}) = [t(^{\circ}\text{F}) - 32]/1.8$; 1 lb_m/ft³ = 16 kg/m³; 1 lb_m/hr·ft = 4.134×10^{-4} N·s/m²; 1 Btu/lb_m = 2326 J/kg; 1 ft²/lb_f = 0.021 m²/N; 1 hr·ft^{5/3}/lb_m = 1.10×10^{-5} s^{5/3}/g; 10^6 lb_m/hr·ft² = 1356 kg/m²·s.

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

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

^dBecause sharp flow decay occurred about 300 ms, as shown in Figs. 55-57, a mean value of $U = 15$ ft/s at 80% void was used to calculate the mass velocity.

^eThe inlet quality is calculated based on isentropic expansion.

^fCalculated CHF, operating heat flux; CHF occurs when CHFR < 1.0.

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