POST CRITICAL HEAT TRANSFER
TO FLOWING LIQUID IN
A VERTICAL TUBE
by
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Submitted to the Department of Mechanical Engineering on May 30, 1974 in partial fulfillment of the requirements for the degree of Doctor of Science

ABSTRACT

The heat transfer characteristics of a liquid in vertical upflow in a tube in which the critical heat flux has been exceeded is investigated. Using a novel transient experimental technique the entire forced convection boiling curve for liquid nitrogen was obtained for a given mass flux-quality combination from which parametric effects of heater material, surface roughness and oxide scale, and dryout length on the dry wall film boiling region were determined.

The results show that both increased roughness and oxide scale increase the post critical heat transfer. Increasing the dryout length decreases the heat transfer at a given mass flux-quality combination due to thermal nonequilibrium effects. No material effects were noted. Post critical heat transfer data is presented for a 0.4 inch I.D. tube at mass velocities of 30,000 to 220,000 lbm/hr-ft² for a quality range of 5 to 90 percent. Heat fluxes of 1,000 to 25,000 btu/hr-ft² at wall superheats, (T_wall - T_sat), from 50 to 550 °F were obtained.

A post critical heat transfer prediction scheme has been developed from the simplification of an existing dispersed flow film boiling model which predicts the transient nitrogen data within approximately 10%. The scheme gives the correct functional dependence of mass flux, dryout quality, dryout length, and wall superheat, (T_wall - T_sat), due to the implicit inclusion of thermal nonequilibrium effects.

A preliminary comparison of the post critical heat transfer prediction scheme with post dryout water and Freon 12 data indicates similar results.

Procedures are also presented that allow one to obtain the upper and lower bounds to the post critical heat transfer.

Thesis Supervisor:  Warren M. Rohsenow

Title:  Professor of Mechanical Engineering
BIOGRAPHICAL NOTE

The author was born in Machias, Maine on December 25, 1947. He was raised in the neighboring hamlet of Harrington where he attended primary and secondary schools. He entered Lowell Technological Institute in September 1966 and graduated with a B.S. in Mechanical Engineering, June 1970. He was also commissioned a Second Lieutenant in the U.S. Air Force Reserves. Through the USAF Educational Delay Program he was granted time to pursue studies leading to the degree of Doctor of Science at M.I.T. While at M.I.T., he was a National Science Foundation trainee and a Research Assistant in the Heat Transfer Laboratory. He has co-authored several technical reports in the area of post critical heat transfer. He will be assigned to the Air Force Systems Command at Kirtland Air Force Base, New Mexico.
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I am grateful, also, to Professor Borivoje Mikić for his solutions to the many problems I presented him.

I sincerely thank Mr. Onwuamaeze Iloje with whom I have been associated for the past four years for his support.

I wish to thank all the other members of the Heat Transfer Laboratory for their kind assistance in this investigation. I wish to thank, also, those people of the Mechanical Engineering Computer Facility who helped me formulate the data acquisition and processing procedures.

Support of my first year at M.I.T. by the National Science Foundation through its Traineeship program is gratefully acknowledged. Support for this investigation was obtained from a grant by the National Science Foundation.
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<td>P</td>
<td>pressure</td>
<td>psia</td>
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<tr>
<td>P₁, P₂</td>
<td>potentiometer settings on analog computer</td>
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Pr  Prandtl number $\mu C_p / k_p$

$q/A$  surface heat flux $\text{Btu/hr-ft}^2$

$q$  heat flow $\text{Btu/hr}$

$r, R$  radius $\text{ft}$

$R$  gas constant $\text{Btu/lbm}^0\text{R}$

$Re$  Reynolds number $\rho V D / \mu$

$S$  slip ratio $V_g / V_v$

$Sc$  Schmidt number --

c  thichness $\text{ft}$

$t$  time $\text{hr}$

$T$  temperature $^oF$ or $^oR$

$T_{cb}$  contact boundary temperature at the liquid-wall interface $^oF$

$T_{wo}$  initial wall temperature $^oF$

$T_{Lo}$  initial liquid temperature $^oF$

$V$  velocity $\text{ft/hr}$

$V$  volume $\text{ft}^3$

$V_g$  saturated and superheated vapor velocity $\text{ft/hr}$

$We$  Weber number $\rho (V_g - V_v)^2 D / \sigma$

$X$  quality

Greek

$\alpha$  void fraction --

$\gamma$  $C_p / C_v$ --

$\delta$  drop diameter $\text{ft}$

$\delta_{film}$  correlating constant for the Groeneveld wall-to-drop heat transfer coefficient (Equation (3.23)) --
\[\kappa\]  
slope of linear approximation to \(X_A\) vs \(X_E\) curve in post dryout (Equation A(4.1))

\[\rho\]  
density  \(\text{lbm/ft}^3\)

\[\mu\]  
dynamic viscosity  \(\text{lbm/hr-ft}\)

\[\sigma\]  
surface tension  \(\text{lbm/hr}^2\)

\[\Pi\]  
\(G\sqrt{D_T/\rho_v\sigma}(1 - X_{D0})^5\)  

\[\Delta T_s, \Delta T\]  
\((T_{\text{wall}} - T_{\text{sat}})\)  \(^\circ\text{F}\)

\[\Delta t\]  
incremental time  \(\text{hr}\)

\[\Delta X\]  
incremental length  \(\text{ft}\)

**Subscripts**

\(A\)  
actual

\(\text{crit}\)  
critical

\(d\)  
drag

\(E\)  
equilibrium

\(f\)  
film temperature  
\((T_v + T_w)/2 \quad \text{wall-to-vapor terms}\)

\((T_w + T_{\text{sat}})/2 \quad \text{wall-to-drop terms}\)

\(g\)  
gravity

\(g\)  
saturated vapor

\(\text{DO}\)  
dryout

\(\text{CHF}\)  
critical heat flux

\(\ell\)  
liquid

\(\text{min}\)  
minimum

\(o\)  
initial, at last shatter

\(\text{sat, s}\)  
saturated value

\(v\)  
vapor

\(\text{wall, w}\)  
heated wall
w,v  wall to saturated or superheated vapor
w,δ  wall to drop
δ, drop  refers to droplet
i  insulation
cu  copper
a  air
m  micarta
r  rubber O-ring
cold spot  defined in Fig A2
1. Introduction

1.1 General Description of Problem:

For centuries man has known that a liquid can be converted to vapor by a suitable application of heat and with his seemingly unlimited ingenuity has expanded this knowledge to the point where he is now capable of converting, with giant nuclear reactors, tremendous amounts of energy for performing useful work. But in sharp contrast to this high level of technical competence in energy conversion, the knowledge and understanding of its basic underlying principles, specifically that of boiling heat transfer and two phase flow phenomena, is still at an unacceptable low level.

Qualitatively, the physics of boiling heat transfer in a convective system is quite well understood. When a heated surface is in contact with a fluid, the general heat transfer behavior of the heater can be described on what is commonly referred to as a boiling curve. The boiling curve for a convective system which is similar to that given in a pool boiling situation is plotted in terms of the heater wall superheat defined as $(T_{\text{wall}} - T_{\text{sat}})$ on the abscissa and heat flux from the heater surface on the ordinate. Figure 1 gives a qualitative representation of a boiling curve in forced convection. As most forced convection boiling systems consist of a conduit (tube annulus,
enclosed rod bundle) of some length through which the fluid is passed, Figure 1 represents the boiling characteristics at a particular axial position in the conduit for a given mass flux.

The peculiar shape of the boiling curve is a result of a change in the relationship of the liquid and generated vapor to the heater surface as the surface heat flux is increased. The nucleate boiling region has the liquid completely attached to the heater surface with the vapor being generated from preferred cavities. This is a very effective heat transfer mode where tremendous amounts of heat can be transferred for low wall superheats. But if the surface heat flux exceeds the critical heat flux value given by point A, the wall becomes dry with the vapor insulating the liquid from the surface causing the wall temperature to rapidly increase to point B*. This region is called the drywall film boiling region or the post critical heat transfer region. The term, post critical heat transfer is actually used to define the entire boiling curve to the right of the maximum nucleate boiling temperature, thereby including the transition region, whereas the term, dry wall film boiling, only includes that part of the boiling curve starting from point D. The two terms

*Consideration is made of the possibility of direct liquid wall contact in film boiling which is discussed in section 2.7.4 as a possible explanation of the oxide effects observed in experimental phase of this work.
are used interchangeably for a heat flux controlled system in which the transition region can not be maintained. In this regime the heat transfer rates are two orders of magnitude lower than in the nucleate regime. The vapor is now generated at the vapor liquid interface and is capable of superheating. Further increase in the heat flux to the surface results in the surface temperature moving up along the film boiling curve. Once the heater has achieved a condition of stable film boiling it can return to the wet wall nucleate boiling condition in one of two paths. First it can return along path B-A once the heat flux is reduced to the critical heat flux value which previously forced that particular axial position into the film boiling mode or it can continue down further along the film boiling curve until the minimum heat flux is reached thereby transiting via path D-C. The surface can only reach this lower portion of the film boiling curve under certain conditions. This portion of the curve can be reached in a temperature controlled system such as a quench process, or it can be reached in a heat flux controlled system if there is no liquid reattached somewhere else in the heated conduit. It is the axial conduction inside the body of the heater from the dry area to the wet area of the heated surface that wipes out the hysteresis effect and results in the preferential path being B-A for the
transition back to complete wet wall conditions for the heat flux controlled case.

The region between the minimum film boiling point and maximum nucleate boiling point is the transition region characterized by intermittent liquid attachment to the heating surface and subsequent reevaporation. The frequency of liquid contact increases with decreasing wall superheat until liquid is completely reattached at the maximum nucleate boiling superheat. This region is difficult to define on the boiling curve as one instant, liquid is in contact with the surface providing good heat transfer and the next instant the surface is dry giving poor heat transfer. The transition line drawn on Figure 1 is, therefore, a time average of the two extreme conditions. Only a quench experiment or other types of temperature controlled systems can provide data in this negatively sloping region, and this data is only the average effect of the large temperature and or heat flux oscillations produced at the heating surface.

From this simplified description of the forced convection boiling picture one potentially dangerous aspect stands out for such systems as evaporator tubes in fossil fuel boilers or rod bundles in a water cooled nuclear reactor. That is the rapid temperature rise observed in going from A to B. The temperature at B,
as the critical heat flux is exceeded, could be sufficient to physically damage the heating surface. This transition into the critical heat transfer mode can happen in many types of boiling systems if the systems experience either a flow loss or a power transient. In order to know just what temperature the system will reach and thereby determining just how damaging the transition to post critical heat transfer can be, a quantitative description of the post critical heat transfer regime is necessary.

The nuclear reactor industry is currently placing considerable emphasis, through their safety analysis programs, on the problem of accurately predicting the post critical behavior in a reactor core. Their main concern is to prevent the rod bundle from reaching a temperature sufficient to melt the cladding material containing the fissile material if a post critical situation occurs. Even though there is an extremely low probability of this happening the reactor designer must prove that emergency core cooling systems are capable of cooling the reactor core if it is perturbed out of the design conditions. The worst accident postulated to perturb the reactor is that of a loss-of-coolant accident, Loca\textsuperscript{1*}, where one of the feed lines to the reactor core is postulated to break. The flow through the core slows down to a final value of zero and due to this flow stoppage the reactor core is tripped into

\* Superscripted numbers refer to references at the back of the thesis.
a post critical boiling situation. The situation is brought under control by the initiation of emergency core cooling systems consisting of flooding water from the bottom and spraying water from the top of the core. The conservatism used in estimating the heat transfer rates causes the calculated maximum cladding temperatures in the core to reach the melting point of the metal. This uncertainty factor can force the industry to operate the reactor at a reduced power level as a safety precaution.

1.2 Literature Survey:

Early attempts at understanding the film boiling phenomenon consisted of running experiments in order to observe its fundamental characteristics. The heat transfer data was used to develop empirical film boiling correlations applicable in the range for which the data was taken. The bulk of the film boiling data obtained used such test fluids as water, Freon 113, Freon 12, liquid nitrogen and hydrogen, and some hydrocarbons. The geometries consisted of tube, annular or multi rod, bundles.

Groeneveld in a recent publication\(^2\) does an excellent summary of the entire film boiling investigation of the past fifteen years. He not only lists 16 film boiling correlations developed by various researchers but also presents his own correlations based on the careful study of all available film boiling data. The general form of all the
correlations produced is the same and consists of a single phase forced convection heat transfer coefficient using the Reynolds and Prandtl numbers modified by a two phase flow multiplier to account for quality effects. The Groeneveld correlation for tubes and annuli is presented here for illustration.

\[
\text{Nu}_g = a \left[ \text{Re}_g (X + \frac{\rho_l}{\rho_g}(1-X)) \right]^b \text{Pr}_v^c y^d \left( \frac{q}{A} \right)^e
\]

\[ y = 1 - 0.1 \left( \frac{\rho_l}{\rho_g} - 1 \right)^4 (1-X)^4 \]

(1.1)

Where the constants are given for either heat flux or no heat flux dependancy as follows

<table>
<thead>
<tr>
<th>(a)</th>
<th>(b)</th>
<th>(c)</th>
<th>(d)</th>
<th>(e)</th>
<th>No. of Points</th>
<th>Rms Error</th>
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<tr>
<td>7.75 \times 10^{-4}</td>
<td>0.902</td>
<td>1.47</td>
<td>-1.54</td>
<td>0.112</td>
<td>704</td>
<td>11.6%</td>
</tr>
<tr>
<td>3.27 \times 10^{-3}</td>
<td>0.901</td>
<td>1.32</td>
<td>-1.50</td>
<td>0</td>
<td>704</td>
<td>12.4%</td>
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</tbody>
</table>

This equation with or without the heat flux dependancy correlates the effects of mass flux, quality and fluid properties on the post critical heat transfer coefficient.

As the mass flux increases for a constant quality, the heat transfer coefficient increases. As the quality increases for a constant mass flux, the heat transfer coefficient increases at a decreasing rate until it starts decreasing as the quality reaches 95-100%. The heat transfer coefficient is based on the saturated vapor temperatures.
and the quality variable is the equilibrium quality calculated from a thermodynamic heat balance. The limitation of the correlation to the data base is evident as both the vapor temperature and actual quality calculated by weight deviate from the equilibrium values in post dryout. The correlation was developed entirely from post critical water data for the geometry indicated. The mass flux range was \(0.5 - 3.8 \times 10^6\) lbm/ht-ft\(^2\) and the equilibrium quality ranged from 10-90 percent. By taking these ranges at face value one would assume that the correlation would predict the case where \(G = 0.5 \times 10^6\) and \(X = 10\%\). This is not necessarily true, however, as the experimental techniques from which the data for the correlation was obtained links all low quality data with the high mass flux runs. The converse is not always true but one can say that all low mass flux data points had high qualities associated with them. Therefore extrapolating a correlation which correlates a low mass flux, high quality data point within 10\% is almost surely going to lead to considerable uncertainties at low mass flux low quality range.

Recently Slaughterback\(^3\) conducted a parametric study and comparison analysis of Groeneveld's correlation for tubes only with four other film boiling correlations. The general conclusion from this study was that significant discrepancies
exist among the different correlations and between the correlations and experimental data. This conclusion led to statistical regression analysis\textsuperscript{4} of Groeneveld's collection of data which resulted in a modified form of the Groeneveld correlation including an empirical formulation of the uncertainty bounds. The uncertainty bounds are presented as upper and lower bounds on the heat transfer with a 95\% confidence limit. Again the same limitation as discussed for the Groeneveld correlation applies for this one.

Groeneveld as well as this author feels that the answer to the problem of predicting film boiling heat transfer does not lie in statistical manipulation of large amounts of data but instead in the complete understanding of the physical phenomenon involved. Considerable advancement was made in this direction when Forslund\textsuperscript{5} of MIT and Bennett\textsuperscript{6} of UKAEL independently developed governing differential equations for the so-called dispersed flow film boiling region. Both models have been subsequently revised, the Bennett model by Groeneveld\textsuperscript{7} and the Forslund model by Hynek\textsuperscript{8}. Both of these models will be extensively compared in Chapter II with the intention of taking the best points of each of the models to give a resulting hybrid model from which the generalized post critical heat transfer correlation presented in this thesis will be derived.
1.3 General Description of Post Dryout Heat Transfer and Flow Regimes:

Experimentally, the post critical heat transfer regime can be obtained in several ways which result in different two phase flow patterns depending on the technique used to generate the dryout condition. Most experimentors including all those using water develop the dry wall condition in the following manner. Flow of a particular mass flux and inlet quality is allowed into the uniformly heated tube whose power is either zero or at a very low value. The power is then increased incrementally until the dryout condition which starts from the exit moves into the tube to the desired position. Figure 2 gives a qualitative picture of this flow regime pattern and wall temperature profile. It consists of an annular liquid film attached to the heater surface with droplets entrained in vapor core upstream of dryout and a dry wall with droplets dispersed in a superheating vapor downstream. Characteristically this is a high quality, high void dryout phenomenon (above 50% void). This is of course dependent on a number of parameters such as inlet quality, heat flux and mass flux as well as the type of fluid used. (There is a possibility of an upstream dryout which will not be treated here but is discussed in some detail in Section 11.3.3 of Groeneveld7).
FIGURE 2  FLOW REGIME AND WALL TEMPERATURE PROFILE FOR HIGH VOID DRYOUT
The second technique used extensively by researchers employing liquid nitrogen [5,8,9] consists of raising the tube wall temperature at least above the minimum film boiling temperature as indicated on the boiling curve in Figure 1 before allowing the fluid to enter the tube. When the fluid is finally allowed into the tube, a dryout condition is set up throughout the entire heated tube length. Figure 3 gives a qualitative picture of this flow regime pattern. The flow pattern is initially that of a solid liquid core separated from the heater surface by a vapor film. The vapor-liquid interface has an unstable wavey nature. As the void fraction increases, the liquid core takes on a foamy frothy characteristic which finally transits into dispersed flow at some intermediate void. This transition point is on the order of 5-10% quality for nitrogen as its void fraction rises very rapidly with quality. For liquids whose ratio of vapor density to liquid density is much larger as with water and Freon 12 this transition point is generally higher due to a relatively slower increase of void with quality. To this author's knowledge no water data has been obtained using this technique supposedly due to the fear of melting the tube in the start-up procedure. There is an advantage though to using this technique. One is able to obtain film boiling data in the region between the minimum heat flux line and the
Figure 3: Flow regime and wall temperature profile for low void dryout with heat spike.

Axial position vs. wall temperature.
critical heat flux line on the boiling curve. That is to say that this flow regime does exhibit the hysteresis effect where upon a particular point in the tube can descend the film boiling curve below the critical heat flux line without traversing over into a wet wall condition until the minimum heat flux is reached.

This is an inherently unstable flow structure for a uniformly heated tube for if the inlet quality to the heated tube is below the quality at which dryout should occur for the given system mass flux and tube heat flux, then the liquid front will want to move into the heated tube to the position where the quality is equal to quality at which dryout was calculated to occur. The forcing function that prevents the liquid interface from moving into the tube for this experimental technique is the heat gain in through the electrical buss bar attached at the inlet end of the heated tube. In the nitrogen experiments of Forslund and Hynek the buss bar was a large copper electrode whose temperature was somewhat higher than the tube wall to which it was attached. This allowed a sufficient heat spike at the entrance to prevent the liquid interface from attaching itself to the heated tube. Hynek attached a cooling devise to the electrical buss bar which cooled the inlet region to such an extent that dryout could not be achieved directly at the inlet to the heated tube.
Further experimentation by Iloeje et al.\textsuperscript{10} using a buss bar which could control the amount of heat into the inlet region of the tube resulted in the determination of a threshold heat flux above which the minimum test section heat flux at which liquid reattaches the inlet region was unaffected by changes in heat flux in through the buss bar. Below this value the rewetting at the entrance occurred at higher values of test section heat flux for lower heat fluxes in through the buss bar.

This technique of initiating post critical conditions directly at the inlet to the heated tube is similar in purpose to the hot patch technique used by Groeneveld to obtain the lower portion of the film boiling curve. The hot patch provides the spike which initiates film boiling at a quality lower than the system mass flux and heat flux would dictate. The dryout spreads up the remainder of the heated tube as in the nitrogen experiments.

1.4 Scope of Research

An extensive experimental program was undertaken utilizing a unique transient technique to obtain the entire forced convection boiling curve for a vertical tube at one specified mass flux and equilibrium quality combination for nitrogen. The thermodynamic quality varied from 5% to 95% and the mass flux varied from 30,000 to 200,000 lbm/hr.$ft^{-2}$.
The effect of scaling roughness and heater material on the boiling curve were independently investigated.

A comprehensive comparison of Hymek's and Groeneveld's dispersed flow film boiling models with data available in the literature was carried out. The original Hymek computer code [8] was modified to include features of the Groeneveld model when those features were deemed better than the comparable feature in Hymek's code.

A generalized post critical heat flux correlation was developed after simplifying assumptions were applied to the modified dispersed flow mode. The correlation contains all the thermal variables (excluding any surface effects) known to affect the post critical heat transfer. This correlation can give upper and lower bounds for the heat transfer as well as predict the data. The correlation was not only compared against the transient nitrogen data obtained in this work with good success but also predicted steady state tube data for nitrogen, water and Freon 12 published in the literature with fair success.
II. Experimental Program

2.1 Concept of Forced Convection Transient Film Boiling Experiment

In the process of choosing an experimental technique for this work several important factors were considered. First, an experiment had to be designed that would allow for a detailed analysis of the minimum film boiling point and the surface rewet phenomenon. The main criterion here was that minimum film boiling data be obtained for particular mass flux-quality combinations that was free from axial conduction effects. That investigation was carried out concurrently with this work by O.C. Iloeje for his Ph.D thesis. Secondly, accurate data in the post critical heat flux regime needed to be obtained for particular mass flux-quality combinations in order to lend support to a post critical heat flux correlation. Also provisions had to be incorporated in the experiment to allow for the investigation of roughness, scale and material effects on the two quantities in question. The idea of a transient boiling experiment presented in this work very successfully satisfied these criteria.

The concept of the experimental technique was taken from the knowledge that a sufficiently hot body when immersed in a liquid will quench and in doing so passes through
all the regimes of boiling from dry wall film boiling through the transition region to nucleate boiling.* For this experiment the hot body consisted of a one inch long thick walled tube combined with all the necessary equipment for passing the test fluid of a particular inlet mass flux and quality through it and a system for initially heating the short tube into film boiling. The following sections describe the transient section in detail as well as the main loop related instrumentation.

2.2 Nitrogen Loop

The loop diagram for the experimental apparatus is given in Figure 4. It is a once through system employing liquid nitrogen as the test fluid. Aside from the adaption of the transient test section to the discharge of the main test section, the apparatus is essentially that used by Iloeje et al.\textsuperscript{10} and a detailed description of the apparatus can be found in that report.

The main test section, a uniformly heated 8 foot long Inconel 600 tube 0.5" O.D. by 0.4" I.D., operates as a preheater for the transient test section. In this manner a two phase flow mixture with a particular quality and mass flux can be supplied to the transient section. The preheater was operated in one of two states, either a wet wall condition or a drywall condition. The drywall length was

*See Section 2.71 for further discussion of this assumption
varied by constructing a movable electrode that could be bolted onto the preheater at any desired position. When using this electrode the electrical supply cable previously attached to the bottom buss bar was transferred to the new. Data was taken for two dryout lengths, four feet and eight feet.

2.3 Transient Test Section

A detailed drawing of the transient test section design is given in Figure 5. The test section consists of a one inch tube .4 I.D. by 1.0 inch O.D. supported and encased by a copper cup arrangement. This cup allows the specimen to be independently heated with steam supplied at a temperature of 220-250°F. The specimen is electrically and thermally insulated from the supporting structure by micarta insulators. (thermal conductivity of 0.2 BTU/hr-ft²) whose contact areas were purposefully reduced to a minimum.

A combination of rubber O-rings, silicon rubber sealant and compression of the cover assembly onto the specimen insure that the steam and flowing nitrogen are completely separated. The transient test section assembly was bolted onto the exit of the preheater via connection flanges. Glass wool insulation was wrapped around the transient section to reduce heat gains in through the sides of the section. The electrical supply cable for the top electrode to the preheater was bolted to the brass cover of the transient section.
FIGURE 5 TRANSIENT TEST SECTION (DESIGN 2)
The current was thus conducted through the copper casing to the electrode.

The transient test section was specifically designed to allow for the interchangeability of the test specimens. The specimens included smooth Inconel-600, aluminum 1100 and copper pieces; roughened Inconel-600; and oxidized Inconel-600. The smooth surfaces were rough drilled, bored to within .003" - .005" of final size and then finished with a Sunnen Products honing stone No. K12-395. Expected roughness for the Inconel piece is about 5 microinches (arithmetic average) and about 10 micro inches for the copper and aluminum pieces. The actual surface finishes were of this order of magnitude as determined by a profilometer. The roughened Inconel-600 piece was produced by boring and lapping as described for the smooth specimens but at a diameter .003" less than the final .40 inches. A series of left handed and right handed threads, at 20 threads per inch and a depth of about .002 inches were then scribed on the inside surface. The average roughness for this surface was on the order of 400 microinches.

The oxide coating of the Inconel-600 piece was achieved by baking it in an oven for 2 hours at 1000° and another hour at 1500°F. The hot piece was allowed to cool gradually in air at each step. The resulting oxide film was estimated to be .0001 inches or less.
Transient test section design 2 is a modification of an earlier design which was tested and found to have certain deficiencies. These deficiencies centered around design 1's inability to be completely insulated against extraneous heat additions. Figure 6 shows a blowup of the encircled area of Figure 5 where the two designs differ. The finned type arrangement that protrudes from the transient piece in design 1 is bad for two reasons. First it is capable of transferring considerable heat from the base into the transient piece as the small micarta sleeve does not supply sufficient insulation, and secondly, the liquid front attached to the preheater is capable of attaching itself to the fin quite easily thereby causing axial conduction effects. The small contact area and low conductivity of the material that does contact the test specimen in design 2 reduce the heat gains to a greater extent. The copper sleeve at the end of the preheater retards the liquid front in the preheater from coming close to the test specimen. This is because the copper will tend to have a more uniform temperature and any heat that is transferred to it from the brass base will be more uniformly distributed preventing the liquid from attaching it.

This design was so effective in reducing heat losses out the bottom that the small axial temperature gradient in the test piece which was observed to slope down during
FIGURE 6 COMPARISON OF DIFFERENCE BETWEEN DESIGN 1 AND DESIGN 2 TRANSIENT SECTIONS
a run with design 1 with the lowest temperature being recorded at the bottom thermocouple reversed itself for runs using design 2 thereby having the lowest temperature being recorded at the top thermocouple. This indicates that design 2 still has some problems with heat losses. Appendix A gives a total estimation of the heat losses from the test piece for the two designs.

2.4 Instrumentation and Data Acquisition

The instrumentation can be divided into two areas: that related to the monitoring of the loop operations and that related to the data acquisitions from the transient test section. In all cases copper-constantan thermocouples were utilized as the temperature sensing device.

2.4.1 Test loop instrumentation:

Seven thermocouples were placed on the preheater tube, three on the two inch length preceding the transient section (as shown in Figure 5) and four on the main preheater length. The purpose of these thermocouples was to detect the dryout position in the preheater tube. These thermocouples were measured on a Leeds & Northrup Precision Potentiometer. Other system temperatures such as inlet fluid temperature and exhaust gas temperature from the rotometers were also monitored on this instrument. Measurements of such system quantities as pressure, preheater power and mass flux were obtained as described in Reference (9).
tape before the start of each run. This reference signal, used in the data reduction program described in a further section, consisted of a zero reference obtained by placing the reference thermocouple in an ice bath and a maximum reference point obtained by placing the reference thermocouple in a dewar of liquid nitrogen. The temperatures of these two points were obtained by reading the reference thermocouple on the precision potentiometer.

The frequency response of the tape recorder was 650 cycles/second at a recording speed of 3 and 3/4 ips. The strip chart recorder was capable of reading a channel every 1.2 seconds. Being in the four channel model the recorder was able to read one particular thermocouple every 4.8 seconds. Both of these recording devices were able to record the temperature transient of the test specimen whose transient times averaged 30 minutes. (The extreme in total collapse times from $212^\circ F$ to $-320^\circ F$ depending on the mass flux and quality were 5 minutes to 90 minutes).

2.5 Experimental Procedure

Transient boiling curve data was obtained from the experimental apparatus with the preheater either in a completely wetted mode or in a low quality dryout mode with dryout lengths of 4 or 8 feet. The following sequence of operations were carried out for obtaining film boiling data with wet approach conditions. The steam supply to the
transient section was turned on which allowed the specimen to reach an initial temperature of 220-250°F. Liquid nitrogen subcooled 3-5°F was initiated into the preheater. When the preheater thermocouples registered a temperature near the saturation temperature of the liquid, power was applied to the preheater. The flow rate and power were adjusted to give the desired values of mass flux and exit quality to the transient section for that particular run. During the time needed for steady state to be achieved in the preheater the reference points were recorded on the tape recorder. With the tape recorder reading the data thermocouple the transient was initiated by closing off the steam to the transient section. When the transient was completed, the steam was reinitiated into the transient section, power was increased to produce a new quality for the same mass flux, and the transient procedure was repeated until all qualities for a particular mass flux setting was completed.

The procedures for obtaining film boiling data with dry approach differed only in the startup. The bottom electrode was positioned on the preheater for either a four foot or eight foot heated length. Power was applied to the preheater to raise the wall temperature to about 200°F. Flow was then allowed to enter the preheater resulting in a zero quality dryout starting at the position of the bottom
electrode. Flow rate and power were adjusted to give the desired mass flux and equilibrium quality at the exit of the preheater. The remaining procedures are the same as outlined for the wet wall approach. In some instances especially for the higher quality runs a dryout length of one or two inches was noted for the wet wall approach. This was unavoidable due to the physics of the situation.

2.6 Data Processing:

Figure 7 presents a complete flow diagram of the data from the thermocouple signal to the final boiling curve output. This section is concerned with the segments of the flow chart dealing with data processing. This part of the experimental program was carried out at the Joint Mechanical and Civil Engineering Computer facility using the analog-digital hybrid system as well as the INTERDATA Model 70 and Model 80 digital computer systems. The analog computer was an EAI 680 computer system.

A simple first order filter network, consisting of 3 inverters, 2 potentiometers, and one integrator, was patched into the analog computer to filter and amplify the analog signal from the tape recorder. The data-thermocouple leads acting as an antenna picked up strong 60 cycle noise which had to be removed from the analog signal before accurate digitizing could take place. The amplification was necessary to boost the ±1 volt signal to ±10 volts.
needed to give maximum sensitivity to the analog-to-digital conversion process. The analog part of Figure 8 can, therefore, be looked upon as a black box whose function is to amplify the oncoming signal by the ratio $P_1/P_2$ and filter all frequencies above that given by $10 \cdot N \cdot P_2$ as described in the figure.

The analog-to-digital package is a system that allows the digital computer to read the output of the analog circuit every time a timing pulse is sent to it by the analog computer. This timing pulse is termed the digitizing rate and can be set by the operator to any desired frequency. The output of the analog-to-digital block on Figure 7 is, therefore, an array of voltages whose elements are separated from one another by a constant unit of time specified by the digitizing rate. This array is stored in the digital computer for later processing.

The digital computation phase involves two basic processes: conversion of the voltage array to the actual temperature array and the use of this array to calculate the boiling surface heat flux and wall superheat. The conversion of the voltage array to corresponding temperature values is accomplished in two stages. First, the amplified voltage array is converted back to the original millivolt values using conversion factor generated from the two reference points which had been subjected to the same
\[ e_{out} = \frac{P_1}{P_2} e_{in} \]

\[ f = P_2 \times 10^N \]

where

\[ N = 1, 10, 10^2, 10^3 \]

**Figure 7** Data Flow Diagram
processes as the rest of the data. Then the millivolt array is converted to the actual temperature values using a function subroutine consisting of a series of fourth order polynomial curve fits for the copper-constantan thermocouple conversion table.

The main data processing code takes the temperature array for a particular data run and computes the surface heat flux and corresponding wall superheat. The code assumes that the test specimen exhibits no radial or axial temperature profiles allowing the heat flux to be calculated using a lumped heat capacity model. This is a good assumption and the internal temperature gradient will be within 5% of zero if the Biot number \( \left( \frac{h L}{K_s} \right) \), where \( L \) is the characteristic length of the body obtained by dividing the volume by the surface area, is less than .1 [11]. At the maximum post critical heat transfer coefficient of 50 BTU/hr - \( \text{ft}^2 \cdot ^\circ \text{F} \) obtained in the experiment the Biot number for the Inconel-600 piece was .22. The same value for the copper and aluminum 1100 pieces were .01 and .013 respectively. The average heat transfer coefficient was generally half the value quoted allowing one to calculate the heat flux from the following equation

\[
\frac{q}{A} = \rho \ C_p \ \frac{V}{A} \ \frac{dT}{dt}
\]  

(2.1)

The temperature dependence of \( C_p \) at the low temperature
obtained using liquid nitrogen was included in the calculation of heat flux. With this model the heat flux is directly proportional to the rate of change of the test specimen temperature with time. The code determines the first derivative in the following manner. The temperature array is divided into several segments, and for each segment a least square polynomial curve fit is applied up to order 6. The order of the curve fit is chosen to give the least RMS error between data and curve fit without introducing too strong a wavey character to the first derivative. A fourth order polynomial was generally used in processing the data. The first derivative is obtained directly from the polynomial representation of temperature-time data. This procedure calculates quite well the boiling curve from the film boiling region to the minimum point. The region of the calculated boiling curve to the left of the minimum film boiling wall temperature, including the transition and nucleate boiling regions, is not as well represented quantitatively. The shape is correct but due to the high head fluxes radial temperature gradients reduce the accuracy of the lumped heat transfer model causing the calculated heat fluxes and wall superheat temperatures to be more inaccurate in these regions.

A simple finite difference calculation technique was also performed as a check on the curve fitting method. This procedure consisted of averaging the slope over 4 to 8 time-
temperature increments and applying the calculated heat flux to the middle temperature value to generate the boiling curve from the temperature arrays. There appeared to be little discrepancy between the two methods.

2.7 Experimental Results:

The film boiling data for all the runs (Runs 101-159) using design 2 transient test pieces are tabulated in Appendix C. The results for a selected number of runs (Runs 90-100) employing design 1 transient test pieces are also tabulated. These runs include all those with the preheater in the dryout mode. Most of the data with the preheater in the wet mode is affected to some degree by heat losses as discussed in Appendix A. It is felt that the data obtained in the initial portions of the transient for each of these runs is affected to a much lesser degree by the heat losses than the portion of the transient where the test piece is considerably below the ambient room temperature. Therefore one should only consider the first portions of tabulated data for the runs with low $L_{DO}$'s. Appendix B gives the equations used in the data reduction process for the system variables and gives an estimated error for these quantities. The following sections discuss the results of a parametric study of heater material, roughness and oxide coating on the post critical heat transfer. A comparison is also made between the transient nitrogen
data taken in this program and some data obtained by Forslund in the steady state mode.

2.7.1 Transient vs. steady state data

There is some contention in the literature that a quench experiment will not reproduce a steady state boiling curve. Bergles and Thompson\textsuperscript{12} attempted to ascertain if there were any discrepancies between the steady state and transient boiling curves for the same fluid and heater geometry. Experimentally Bergles and Thompson found the steady state boiling curves differed in some respects from the transient boiling curves for the three fluids, water, Freon 113 and nitrogen, that he tested. The discrepancies for the water and Freon 113 steady state and transient boiling curves were easily explained to be caused by the oxide deposits deposited on the boiling surface during heat up. The combination of wettability and roughness of the oxide coating increased the minimum heat flux and wall temperatures, increased the film boiling heat transfer, increased the critical wall temperature and decreased the critical heat flux. The discrepancies between the transient and steady state boiling curves for nitrogen were localized around the minimum point. The transition point was reduced to a much lower heat flux and wall temperature for the transient experiment than exhibited by the steady state experiment. This phenomenon could not be explained by oxide
scaling as the quench piece was only heated to room temperature. Bergles and Thompson took the position that the steady state boiling curve represented the correct physical phenomenon and tried to explain why the transition point in the transient case was delayed to lower a wall temperature and heat flux than that resulted in the steady state case. It is our contention that the transient case is the more accurate representation of the boiling characteristic and that the steady state experiment conducted by Bergles and Thompson was forced to transit earlier due to axial conduction effects brought on by the power lead as well as the possibility of liquid reattachment to the unheated areas of the test cylinder.

Figure 8 presents a comparison of several of the transient data runs obtained in this program with some steady state nitrogen data taken by Forslund. The transient data appears to be slightly lower than each of the corresponding steady state data points, even though the mass flux and quality differences between the two would cause one to expect slightly higher values. This is attributed to experimental uncertainties in the transient data rather than any transient effects. As the transient times for the tests averaged over thirty minutes per run the transient experiments can be considered nearly quasisteady.
FIGURE 8 TRANSIENT VS STEADY STATE FILM BOILING HEAT TRANSFER
2.7.2 Effect of heater material on post critical heat transfer.

Figure 9 presents a comparison of the three different smooth specimens tested for a mass flux of 60,000 lbm/hr-ft$^2$ and 210,000 lbm/hr-ft$^2$. The results for each of the materials produce a fairly large band at each of the two mass flux cases. For the 60,000 lbm/hr-ft$^2$ case copper gives the highest heat flux for a given wall superheat, followed by aluminum 1100 and then Inconel-600. For the high mass flux case the order from highest to lowest is copper, Inconel-600 and aluminum 1100. There appears to be no consistent material effect in the data, and the banding is considered to be the result of experimental error (Appendix A & B).

Bergles and Thompson$^{12}$ in the process of determining the relationships between steady state and transient boiling curves presented some data for copper and inconel. Taking the view that the two processes used gave the same boiling curve, no strong material effects were noted in the film boiling data for the two materials where the surfaces were considered to be free of oxide scale effects.

2.7.3 Effect of roughness on the post critical heat transfer.

The roughness effect on the post critical heat transfer should follow the same trends as observed in single phase heat transfer. By applying the Colburn analogy$^{13}$ between
Figure 9 Effect of Material on Dry Wall Film Boiling Heat Transfer

- RUN 110 (Inconel 600)
- RUN 133 (Aluminum 1100)
- RUN 148 (Copper)

G ≈ 60,000 lbm/hr-ft²
X_E = 50%
L_DO ≈ 2 inches

- RUN 126 (Inconel 600)
- RUN 93 (Copper)
- RUN 73 (Aluminum 1100)

G ≈ 210,000 lbm/hr-ft²
X_E = 10%
L_DO ≈ 2 inches

Transient Nitrogen Data
P = 20 psia
heat transfer and fluid friction and considering the heat transfer in post dryout to be entirely due to the wall to vapor component where the vapor mass flux is defined as

\[ \frac{G_v}{\alpha} = \frac{G_X}{\alpha} \]  \hspace{1cm} (2.2)

one can obtain a functional dependance of roughness on the post critical heat flux in an approximate manner to be

\[ \frac{q}{A} = \frac{f}{2} C_p \frac{G_X}{\alpha} Pr^{-2/3} (T_w - T_v) \]  \hspace{1cm} (2.3)

With all other variables held constant the heat flux will increase with increasing friction factor, \( f \). \( f \) is directly related to roughness heights defined by the ratio, \( e/D \), and Reynolds number as given by Moody\(^1^4\).

The roughness effect on the transient data was not very strong for the 30,000 lbm/hr-ft\(^2\) mass flux tested as seen in Figure 10. Both the low and high quality cases showed no pronounced effect. This is because at the low vapor Reynolds numbers, the difference between the rough and smooth friction factors is small. There are indications though that the roughened surface gives slightly higher heat transfer for the high quality case if one looks at the actual mass fluxes for the two sets of data being compared. The roughened specimen had a mass flux 16% lower than the smooth but had the same heat flux for a given wall superheat.
FIGURE 10 EFFECT OF ROUGHENING ON FILM BOILING HEAT TRANSFER

\[ Q = 28,000 \text{ Btu/hr-ft}^2 \]
\[ \dot{m} = 75\% \]
\[ \theta = \frac{3 \text{ inches}}{\text{inches}} \]
\[ T_{\text{sat}} = -320 \degree F \]

- Run 105 - smooth
- Run 156 - artificially roughened

FIGURE 10 EFFECT OF ROUGHENING ON FILM BOILING HEAT TRANSFER

\[ Q = 30,000 \text{ Btu/hr-ft}^2 \]
\[ \dot{m} = 10\% \]
\[ \theta = \frac{2 \text{ inches}}{\text{inches}} \]
\[ T_{\text{sat}} = -320 \degree F \]

- Run 101 - smooth
- Run 159 - artificially roughened
2.7.4 Effect of oxide scale on the post critical heat transfer.

There is considerable evidence in the literature [12, 15, 16, 19] that oxide coatings on the heating surface have large effect on the boiling curve. Figure 11 presents the effects of oxides on the film boiling heat transfer found in this program for several mass fluxes and qualities which is consistant with that observed by other experimenters. For both mass fluxes at each of the different qualities the oxide coated specimen has higher heat fluxes than the smooth specimen. The low qualities gave more pronounced effects than the higher qualities. It is felt that the heat losses or gains, if present as discussed in Appendix A, will affect all the data used in the comparison and will not influence the differences observed between the smooth and oxide coating runs. The following is postulated to be the reason for the observed oxide effect.

It is postulated that there is sporadic liquid contact in the dry wall film boiling region. (Visual observations have been observed and the heat transfer effects have been measured for liquid-metal contact of single droplets on horizontal heated plates [17, 18, 19]). The contact time may be of an infinitesimal ly small duration but is sufficient for the liquid to sense the wall. If the contact temperature which can be estimated to be
FIGURE 11 EFFECT OF OXIDE SCALE ON FILM BOILING HEAT TRANSFER
\[
\frac{T_{cb} - T_{Lo}}{T_{wo} - T_{cb}} = \frac{(k \rho c)_w}{(k \rho c)_L}
\]

at the instant of contact from the solution of two semi infinite bodies of different initial temperatures is greater than the maximum liquid temperature*, the liquid will not remain in contact but immediately be expelled from the surface. However, if the surface is covered by an oxide coating, several effects will allow the liquid to remain for a longer period of time. First the contact temperature will be initially lower due to the lower \((k \rho c)\) ratio of the oxide as evidenced by equation (2.4) and if the base temperature is higher than the maximum liquid superheat a certain amount of time will be necessary to heat the surface temperature of the oxide to the maximum liquid temperature. During this heat up time the liquid is evaporating producing good heat transfer. Second, if the oxide is porous or highly wetting the liquid will spread out covering a larger portion of the heater and adhere more strongly to the heater than if the surface was clean. Both these processes will increase the heat transfer. The quality effect can be linked by the fact that the probability of the liquid contact will increase as the void fraction decreases. The momentum of the small drops, characteristic of high
The maximum liquid temperature is defined by Groeneveld [7] as a thermodynamic liquid temperature for a given pressure above which the liquid state can not be maintained. It can be obtained from homogeneous nucleation theory [29].
void film boiling, towards the heated surface is too small to resist the repelling force due to the liquid evaporating from the surface of the drop.

The explanation of the oxide effect presented here is somewhat contradictory to the idea that the minimum film boiling temperature is identical to the maximum liquid temperature (Groeneveld 7) and as such no liquid contact can result in film boiling. Instead of the minimum film boiling temperature being thermally controlled, a theory has been developed (Iloeje 28 ) that indicates that the minimum film boiling point is the result of the changing importances of three heat transfer mechanisms: the mechanism controlling heat transfer to liquid in direct contact with the heater, the mechanism controlling heat transfer to liquid that comes near the heater surface without touching (dry collisions) and the mechanism of forced convection heat transfer to vapor. The first term, starting from the maximum nucleate boiling temperature where total liquid contact is assumed, decreases with increasing wall superheat. The last two terms increase with increasing wall superheat. The addition of the three terms produces a minimum in the boiling curve. Anything that affects each of the separate terms will influence the minimum point. Iloeje considers that any droplet can contact the surface regardless of the wall temperature if it has sufficient radial momentum towards
the wall. His model can not predict the effect of wet collisions with the wall whose temperature is above the maximum liquid superheat as the vigorous evaporative heat transfer mechanism is not understood.

It is felt that this concept of the three step process is a major advancement towards the full knowledge of the boiling curve and helps one to understand in the light of surface effects such data as McDonough, Milich, and King's 27 and Plummer and Iloeje's 16 where the transition boiling data was considerably above the maximum liquid superheat. Assuming the minimum film boiling temperature to be determined by the maximum liquid superheat would discount this data as suspect when in actuality it is real data which is probably influenced by surface effects. (In the data analyzed by Plummer and Iloeje actual physical evidence of oxide was found upon examination of the test section).
III. Two-Step Dispersed Flow Film Boiling Heat Transfer Model

3.1 General Properties of Model

The dispersed flow model [5,6,7,8] assumes that the two phase mixture beyond the dryout-point is composed of spherical drops uniformly distributed in the vapor phase. In simple terms the model allows heat to be transferred from the wall to the bulk flow in two steps; first from the wall to the vapor phase and from the vapor to the entrained droplets. The model also allows for direct heat transfer to the drop via direct collisions with the heating surface. The model begins at the point of dryout where equilibrium conditions are assumed. There the vapor temperature is at the saturation temperature, and the quality is that given by thermodynamic conditions. At this point initial drop size, vapor and liquid velocities are calculated from continuity, momentum and critical Weber number equations. The Weber number is a ratio of inertia to surface tension forces and in essence restricts the diameter of the drop given the droplet and vapor velocities. To move the solution downstream the gradients of vapor temperature, droplet diameter, actual quality and liquid velocity are derived. These quantities are derived from energy, momentum and continuity considerations. The
empirical nature of the model comes in through the use of friction factor and heat transfer coefficient for a droplet moving in a super-heated vapor, the single phase transfer coefficient from heater surface to the super-heated vapor and heat transfer coefficient used to define behavior of direct wall to droplet heat transfer.

The Hynek and Groeneveld treatments of the basic quantities of the model differ in some instances. The purpose of this chapter is to present the fundamental equations for the two step model as used by Hynek and Groeneveld and to compare each calculation procedure with intention of ascertaining their strong and weak points. The two models are compared with data and a hybrid model is presented which is generally as consistent or somewhat better in predicting the data than either the Hynek or Groeneveld model.

3.2 Conditions at Dryout

3.2.1 Groeneveld Technique:

In calculating the dryout conditions Groeneveld employs the void fraction definition and a slip correlation developed for predryout annular flow by Ahmad\textsuperscript{20}. Groeneveld modified the slip predicted by Ahmad's correlation to be halfway between \( S_{\text{Ahmad}} \) and \( S = 1 \) to account for the discrepancy in flow regimes. The void fraction at dryout
is therefore, given by

\[
\alpha_{DO} = \frac{X_{DO}}{X_{DO} + \frac{\rho_v}{\rho_\ell} (1 - X_{DO}) S}
\]  

(3.1)

where \( S \) is determined from

\[
S = \frac{1}{2} \left[ \left( \frac{\rho_\ell}{\rho_v} \right)^{205} / \left( \frac{GD_T}{\mu_\ell} \right)^{0.016} \right] - 1 + 1
\]  

(3.2)

The average vapor and liquid velocities are found from

\[
(V_{\ell_{DO}}) = \frac{(1 - X_{DO}) G}{\rho_\ell (1 - \alpha_{DO})}
\]  

(3.3)

\[
(V_{g_{DO}}) = \frac{G X_{DO}}{\rho_v \alpha_{DO}}
\]  

(3.4)

Groeneveld assumes the droplet diameter to be critical at the point of dryout, and furthermore he assumes that the diameter can be predicted by a critical Weber number criterion given by

\[
(W_e)_{crit} = \frac{\rho_v (V_g - V_{\ell}) \delta}{\sigma}
\]  

(3.5)

He selects a value of 6.5 for the critical Weber number based on Isshiki's\textsuperscript{21} water data. Substituting in the values of \((W_e)_{crit}\), \((V_{\ell_{DO}})\) and \((V_{g_{DO}})\) gives \(\delta_{DO}\)
3.2.2 Hynek Technique:

Hynek solves the momentum, continuity and critical Weber number equations along with the assumption that the liquid acceleration is related to the vapor acceleration by

\[
\frac{dV_p}{dZ} = \frac{V_g}{dZ}
\]

and the assumption that the vapor acceleration is related to the heat flux under isothermal conditions with \( A_v = A \) by

\[
\frac{dV_g}{dZ} = \frac{(q/A)}{\rho_v h_{fg} D_T}
\]

to obtain two equations relating \( V_{gD0} \) to \( V_{vD0} \):

\[
(V_g)_{D0} = (V_v)_{D0} + \sqrt{\frac{4}{g(1 - \frac{\rho_v}{\rho_v}) + \frac{4(q/A)X_{D0} (V_v)_{D0}}{\rho_v h_{fg} D_T}}} 
\]

\[
\frac{\rho_v (W_e)_{crit\sigma}}{.75C_D \rho_v^2}
\]

and

\[
(V_g)_{D0} = \frac{G X_{D0}}{\rho_v [1 - \frac{G(1-X_{D0})}{\rho_v (V_v)_{D0}}]}
\]
These two equations require an iterative solution. A fortran subroutine, developed by Hynek is given in Appendix D which gives \( V_{g, DO} \) and \( V_{\phi, DO} \) for any desired set of conditions. Hynek chose a value of 7.5 for the critical Weber number based also on the Isshiki data. Knowing the liquid and vapor velocities at dryout the droplet diameter is obtained from the critical Weber number criterion given by equation (3.5).

3.2.3 Discussion:

Tabulations of dryout slip and void fraction calculated, by each model are presented in Table 1 for a selected set of mass fluxes and heat fluxes with dryout quality ranging from 10-90% for liquid nitrogen. The Hynek slip is strongly influenced by dryout quality and mass flux. The Groeneveld slip has no dryout quality variable and the mass flux variable is quite small being raised to the 0.016 power. Groeneveld has no heat flux variable in the slip ratio, and while the Hynek model does include heat flux, an increase of heat flux by a factor of 4 for the same mass flux and quality changed the resulting slip by less than 1 percent. For Nitrogen the maximum slip the Groeneveld method was 1.7 compared to a value of 41. for Hynek's at the same conditions of \( X_{DO} = 10\% \) and \( G = 30,000 \) lbm/hr-ft\(^2\). This resulted in a dryout void of 89% for the Groeneveld method as compared
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(D_T = .4 inches)

Representative Values of S_{DO} and \alpha_{DO} for the Hynek and Groeneveld Initialization Techniques

**TABLE 1**
to 26% void for the Hynek technique. While it is possible that the Hynek model fails at this set of conditions due to the fact that the flow pattern might not be dispersed flow, the value of 26% for the void seems intuitively more correct than the 89% value. As the mass flux and dryout quality increase to the point where the dryout void is high (greater than 85-90%), both techniques give reasonably close values. The more physical nature of the Hynek model as well as its ability to give more reasonable results outweigh its awkwardness.

3.3 Gradients in Post Dryout

Groeneveld revised the Bennett post dryout gradients to include pressure drop effects and flashing effects. Groeneveld found that these terms can be neglected except in the case of Freon 12 at high heat flux-mass flux conditions. Therefore, the simplified post dryout gradients of Groeneveld which are identical to those used by Hynek are presented here.

Liquid droplet velocity gradient:

$$\frac{dV}{dZ} = \frac{3 C_D^{\nu} (V_G - V_L)^2}{4 \delta \rho_L V_L} - \left[ 1 - \frac{\rho_v}{\rho_L} \right] \frac{g}{V_L}$$  (3.10)

Droplet diameter gradient:

$$\frac{d\delta}{dZ} = \frac{-2(q/A)_{\text{vapor to drop}}}{h_{fg} \rho_L V_L} \quad \frac{4 \delta (q/A)_{\text{wall to all drops}}}{3(1-X_A)D_T G} h_{fg}$$  (3.11)
Actual quality gradient

\[ \frac{dX_A}{dZ} = \frac{-3(1-X_o)\delta^2}{\delta_o^3} \frac{d\delta}{dZ} \]  \hspace{1cm} (3.12)

where \( X_o = X_{D0} \) and \( \delta_o = \delta_{D0} \) to start.

Equilibrium quality gradient (thermodynamic)

\[ \frac{dX_e}{dZ} = \frac{4g/A}{G h_{fg} D_T} \]  \hspace{1cm} (3.13)

Vapor temperature gradient:

\[ \frac{dT_v}{dZ} = h_{fg} \frac{dX_e}{dZ} - \left[ h_{fg} + C_{pv}(T_v - T_{sat}) \right] \frac{dX_A}{dZ} \]  \hspace{1cm} (3.14)

3.4 Droplet Breakup

The point at which droplet breakup occurs is determined by the critical Weber number for both the Hynek and the Groeneveld model. Hynek used 7.5 whereas Groeneveld used 6.5.

When the critical Weber number is reached in the Groeneveld model the new droplet diameter is set equal to the critical droplet size given by

\[ \delta_{crit} = \frac{(W_e)_{crit} \sigma}{\rho_v (V_g - V_e)^2} \]  \hspace{1cm} (3.15)
The values, $\delta_o$ and $X_o$, in the actual quality gradient are updated to $\delta_{crit}$ and $X_A$ at shatter. The droplet flux [droplets/ft$^2$-fr.] is increased to a new value given by

$$N_d = \frac{6 \frac{G}{\pi \delta^3_{crit} \rho_l}}{1 - (X_A)_{at\ shatter}}$$

(3.16)

A new velocity gradient is calculated from which new values of $V_g$ and $V_l$ are determined. The Weber number is rechecked. If the Weber number is still critical the cycle is repeated until the Weber number is just subcritical.

The Hynek model assumes the droplet to shatter in two as the critical Weber number is reached. This results in the doubling of the droplet population and a reduction of each drop diameter by $1/\sqrt[3]{2}$. The values of $X_o$ and $\delta_o$ are updated. The same procedure as Groeneveld used is applied to ascertain if the new Weber number after shattering is less than the critical value.

Unlike the large discontinuity in drop diameter resulting from shattering in the Hynek model the trend of the droplet diameter change for the Groeneveld model is more gradual. The Weber number in the post dryout calculations tends also to remain near the critical value for the Groeneveld method as the drop diameter doesn't change
much after shattering.*

3.5 Heat Transfer Correlations:

The dispersed flow model is by reason of the extremely complicated flow structure a semi-theoretical model which depends on empirical correlations to describe the heat transfer behavior of the component parts.

3.5.1 Vapor to droplet:

There is general agreement between the two models that the analogy between heat and mass transfer modified by the Froessling ventilation factor will predict the vapor to drop heat transfer. While Groeneveld derives a simple technique for determining the diffusive resistance associated with this heat transfer mechanism, he subsequently neglects it as did Hynek citing that high turbulence levels are sufficient to wipe out this resistance. The vapor to drop heat transfer is given by

\[
\frac{Q}{A_{\text{vapor}}^{\text{vapor}}} = \frac{2(T_v - T_{\text{sat}})k_v}{\delta} \left[1 + 0.276 Re_d^{1/2} Sc^{1/3}\right] (3.17)
\]

to drop

where Hynek assumes \( Sc = Pr_{\text{vapor}} \)

and Groeneveld assumes \( Sc = \frac{\mu g^R}{k_g (\gamma-1)} \)

from Kinetic theory of gases

* A provision was made in the dispersed flow film boiling code to reduce the diameter by 10% if after four cycles of the shattering process at a tube position the Weber number was still critical.
3.5.2 Wall to vapor:

There are a number of single phase vapor heat transfer coefficients applicable for the wall-to-vapor heat transfer term. These correlations are differentiated by the fluid on which they were based. Two such correlations tested in this work were that developed by Forslund for nitrogen and that developed by Heineman\textsuperscript{22} for superheat steam. Groeneveld modified a generalized heat transfer coefficient developed by McAdams\textsuperscript{23}. These three equations are presented here.

\[ h_{w,v} = \frac{k_v}{D_T} \cdot 0.035 \cdot Re_v^{0.743} \cdot Pr_v^{0.4} \] (3.18)

\[ h_{w,v} = \frac{k_f}{D_T} \cdot \left( \frac{L_{DO}}{D_T} \right) \cdot 0.0157 \cdot \Re_f^{0.84} \cdot Pr_f \cdot \left( \frac{L_{DO}}{D_T} \right) \] (6 < \frac{L_{DO}}{D_T} < 60) (3.19)

\[ h_{w,v} = \frac{k_f}{D_T} \cdot 0.0133 \cdot \Re_f^{0.84} \cdot Pr_f^{1/3} \] (\frac{L_{DO}}{D_T} > 60) (3.20)

Modified McAdams:

\[ h_{w,v} = \frac{k_v}{D_T} \cdot 0.023 \cdot Re_v^{0.8} \cdot Pr_v^{1/3} \cdot \left( \frac{\mu_v}{\mu_w} \right)^{0.14} \cdot \left[ 1 + 0.3 \left( \frac{D_T}{L_{DO} + 0.01 D_T} \right)^{0.7} \right] \] (3.21)

3.5.3 Wall to drop:

At low qualities and correspondingly low void fractions Forslund found that large discrepancies were present between the experimental data and his core flow analysis using only
a wall-to-vapor heat transfer term. He postulated that
direct wall-to-droplet term made up the difference and
derived a heat transfer coefficient based on Baumeister's work related to sessile drops on horizontal heated plates.
A correlating constant was employed to take into account the droplet velocity and concentration and evaluated from data comparisons. His term is given as

\[
h_{w,\delta} = K_{1}K_{2}(\pi/4) (6/\pi)^{2/3}(1-\alpha)^{-2/3} \left[\frac{k_{f}h_{fg}^{*}\rho_{f}\rho_{l}}{(T_{w}-T_{\text{sat}})\nu_{f}^{3/4}l_{\delta}}\right]^{1/4} \tag{3.22}
\]

where \((1-\alpha) = \frac{G(1-X_{A})}{\rho_{l}V_{l}}\)

and \(h_{f}^{*} = h_{fg}[1+\frac{7}{20}\frac{C_{p}(T_{w}-T_{\text{sat})}}{h_{fg}}]^{-2}\)

Groeneveld questions the validity of extrapolating the use of the heat transfer coefficient for sessile drops to predict the wall-drop interaction in dispersed flow. He feels both the droplet velocity and its rotation invalidates the solution for a sessile drop. The proper evaluation of the wall-to-droplet term requires the knowledge of the drop dynamics as it is projected towards the wall as well as the droplet-wall interaction ear the heating surface. Groeneveld estimates the wall to droplet term by a simple heat conduction term assuming a linear temperature profile between the wall and drop.
\[ h_{w,\delta} = \frac{k_{v,f}(1-\alpha)}{\delta_{\text{film}}} \exp\left[\frac{-2D_T}{L_{DO}}\right] \]  \hspace{1cm} (3.23)

where

(1-\alpha) is that fraction of wall facing the liquid droplets.

\[ \exp[-2D_T/L_{DO}] \] is estimated to account for the reduced wall-droplet interaction just beyond dryout.

\[ \delta_{\text{film}} \] is the average distance of droplets above the heated surfaces.

A theoretical analysis of droplet trajectories in post dryout dispersed flow is currently being investigated by O.C. Iloeje which indicates that \( \delta_{\text{film}} \) is strong function of wall temperature. Therefore the choice of one value for \( \delta_{\text{film}} \) for predicting wall-to-droplet effects in a uniformly heated tube will of necessity be in error somewhere in the tube. An optimum value of \( \delta_{\text{film}} \) can be chosen, though, from comparison of model with post dryout data.

Figure 12 gives a quantitative comparison of the two wall-to-droplet terms for a mass flux of 130,000 lbm/hr-ft\(^2\).
NITROGEN

\[ G = 130,000 \text{ lbm/hr-ft}^2 \]
\[ q/A = 10,000 \text{ btu/hr-ft}^2 \]
\[ X_{DO} = 5\% \]
\[ D_T = .323 \text{ inches} \]

\[ \delta_{\text{film}} = 1 \times 10^{-4} \text{ feet} \]
\[ K_{1,2} = .5 \]

Used by Hynek

**FIGURE 12** COMPARISON OF FORSLUND'S AND GROENEVELD'S WALL-TO-DROP HEAT FLUX TERMS FOR A SELECTED SET OF CONDITIONS
using liquid nitrogen as the test fluid. $\delta_{\text{film}}$ was determined to be $1 \times 10^{-4}$ ft from comparison of model with experimental data for this mass flux. Using the same technique a value of 0.5 was chosen for $K_1 K_2$. As can be seen from the figure the Groeneveld term is zero near the dryout point (as determined by the exponential term) whereas the Forslund term is finite. The two terms coincide as the solution is moved away from the dryout point. This happens at about 40 diameters away from dryout which is the point where the exponential term in Groeneveld's heat transfer coefficient has a 5% effect. As the void fraction tends to one, both droplets terms reduce to zero.

Due to the 2/3 power on the liquid fraction term, $(1-\alpha)$, in the Forslund term instead of the one power as used by Groeneveld, Forslund's wall to drop term tends toward zero at a slower rate.

From observations of temperature length data, low quality dryouts exhibit a low heat transfer coefficient after dryout which builds to a maximum and decreases again as the quality increases from the dryout value. The Groeneveld wall-to-drop term is consistent with this trend. The Groeneveld term was adopted for use in the generalized post critical correlation developed in Chapter 4 for this reason as well as for its more intuitively
justifiable basis.

3.6 Total Heat Transfer in Disperse Flow Film Boiling

The total heat transfer calculated by the model is the sum of the wall-to-vapor heat flux and the wall-to-droplet heat flux given by

\[
\frac{q}{A}_{heater} = h_{w,v}(T_w - T_v) + h_w \delta (T_w - T_{sat})
\]  \hspace{1cm} (3.24)

A finite difference computer code, FILMBOIL, was developed using a combination of techniques used by Groeneveld and Hynek. This is given in Appendix D. The code uses the Hynek initialization procedure, the Groeneveld droplet breakup and the Groeneveld wall-to-droplet heat transfer coefficient. Provisions were made to select either the modified McAdams equation or Heineman equation for the wall-to-vapor heat transfer coefficient depending on what fluid was being tested. This code was compared against Forslund's nitrogen data, Bennett's water data and Groeneveld's Freon 12 data, to ascertain the variations of \( \delta_{film} \) need to predict the respective data.

3.6.1 Comparisons of model with nitrogen data

Figure 13 represents the resulting comparisons of FILMBOIL with Forslund's nitrogen data. This data was taken in zero quality dryout mode and is a good example of the low void dryout explained in the introduction. The
FIGURE 13 COMPARISON OF FILMBOIL WITH FORSLUND'S STEady STATE NITROGEN DATA
inlet qualities for the four runs presented here are about -2% indicated a slight inlet subcooling. The calculation procedure was started at a quality of 5% which is a small distance away from the dryout point. The code cannot be started at a zero quality value as that is a singular point for some of the equations in the code. The modified McAdam's single phase vapor equation was used in the code for this comparison analysis. \( \delta_{\text{film}} = 1 \times 10^{-4} \) feet predicts the nitrogen data quite well and shows no heat flux or mass flux dependence for the three mass fluxes and the two corresponding heat fluxes presented. The model diverges somewhat in the low quality region as can be expected. The flow regime in all probability is not that of dispersed flow but more like a froth flow. The prediction converges to the data as the equilibrium quality increases to a value of 10-30%.

3.6.2 Comparison of model with water data

The water data taken by Bennett is all high void dryout data. Figure 14 gives the resulting comparisons of this data with the dispersed flow model. The Heinemen vapor heat transfer coefficient was used for this set of comparisons. The predictions for these runs were very insensitive to the value of \( \delta_{\text{film}} \) chosen. A range of \( 0.5 \times 10^{-4} \) to \( 5 \times 10^{-4} \) ft did not appreciable affect the results given in the figure. For \( \delta_{\text{film}} = 4 \times 10^{-4} \) feet the
Figure 14: Comparison of Filmboil with Bennett's Steady State Water Data

\[ \delta_{\text{film}} = 4 \times 10^{-4} \text{ feet} \]

Bennett (AEKE-85373)

- \( G = 290,000 \text{ lbm/hr-ft}^2 \) (Run 5391)
- \( G = 1 \times 10^6 \text{ lbm/hr-ft}^2 \) (Run 5251)
- \( G = 3.8 \times 10^6 \text{ lbm/hr-ft}^2 \) (Run 5397)

\( q/A = 172,000 \text{ btu/hr-ft}^2 \)
\( q/A = 306,900 \text{ btu/hr-ft}^2 \)
\( q/A = 582,000 \text{ btu/hr-ft}^2 \)

X\(_{\text{DO}} = 27\%\)
X\(_{\text{DO}} = 51.5\%\)
X\(_{\text{DO}} = 95\%\)
two low mass flux runs (Runs 5359 and 5251) are predicted quite well, but there is a hundred degree overprediction in the case of the $3.8 \times 10^6$ lbm/hr-ft$^2$ run (Run 5397). As the void fraction is too high for $\delta_{\text{film}}$ to have much effect, the reason for the discrepancy must lie in the vapor heat transfer coefficient. A check on the range of variables used in the correlation revealed first that the property data used in the correlation was within 2% of the values used in this work.* This discounts one possible source of error. The Reynolds numbers encompassed in the correlating procedure were in the range of 20,000 - 370,000. However, the saturated vapor Reynolds number for Run 5397 is $3.5 \times 10^6$ or an order of magnitude higher than the maximum value correlated in the Heineman equation. It is possible that the discrepancy arises from extrapolating the correlation too far. Using the McAdams equation reduces somewhat the discrepancy between the prediction and data as shown in Figure 10 for the $3.8 \times 10^6$ lbm/hr-ft$^2$ case. The McAdams equation was found to underpredict the other two mass fluxes. Groeneveld compared a $3 \times 10^6$ lbm/hr-ft$^2$ mass flux run of Bennett's using an optimized vapor heat transfer

*See Appendix F for a listing of all property data used in this thesis. Included in Appendix F are polynomial curve fit equations for temperature dependence on the transport properties.
coefficient developed by Hadoller\textsuperscript{24} with very good results. As the correlation is not published in the literature it could not be used here. This problem of uncertainties in the vapor heat transfer coefficient coupled with the fact that the data is in the high void region makes the choice of $\xi_{\text{film}}$ somewhat arbitrary for the case of water. This in some measure explains why Bennett did not use a direct wall to droplet term in his model.

3.6.3 Comparison of model with Freon 12 data.

The ability of the model to predict the Freon 12 data presented by Groeneveld is somewhat poorer than in the cases of the other two fluids as indicated in Figure 15. Using the modified McAdams equation and a $\delta_{\text{film}} = 1 \times 10^{-4}$ ft gives a fairly accurate prediction of the high quality-low mass flux case given by Run 8602.12, but this combination does not predict well at all the rapidly decreasing temperature-length profile exhibited after dryout for the low quality-high mass flux case given by Run 8620.14. Two values of $\delta_{\text{film}}$, $2 \times 10^{-4}$ and $0.75 \times 10^{-4}$ ft, are presented to indicate the effect of its decrease on the resulting temperature profile. As is expected the decreasing of $\delta_{\text{film}}$ affects the first half of the profile much more than the second half due to the generally lower void fraction. Therefore, increasing $\delta_{\text{film}}$ to affect a better prediction of the experimental profile will only succeed in giving an in-
FREON 12 (155 psia)
Groeneveld

$\delta_{\text{film}} = 2 \times 10^{-4}$ ft.
$\delta_{\text{film}} = 1 \times 10^{-4}$ ft.

Hynek Prediction with $K_1K_2 = 0.5$

$X_{D0} = 39\%$

$X_{D0} = 64\%$

$G = 2 \times 10^6$ lbm/hr-ft$^2$
$q/A = 92,000$ btu/hr-ft$^2$
Run 8620.14

$q/A = 39,000$ btu/hr-ft$^2$
$G = 490,000$ lbm/hr-ft$^2$
Run 8602.12

FIGURE 15 COMPARISON OF FILMBOIL WITH GROENEVELD'S STEADY STATE FREON 12 DATA
correct trend of increasing wall temperature with length. Groeneveld observed this same problem of trying to predict the Freon 12 profiles and postulated that the inaccuracy in the Freon 12 properties could be responsible as well as the use of a generalized correlation for the vapor heat transfer to Freon 12. Another possible explanation for the models' overprediction of the low quality run could be related to the oxide effects and increased wettability discussed in section 2.7.4. If in some manner the tube was made more wettable the low void runs such as Run 8620.14 would display a higher heat transfer than the model predicts where as the high void runs such as 8602.12 would not due to the liquid droplets being too small to approach the wall. DuPont bulletin B2 indicates that a one percent decomposition rate per year was found for Freon 12 when subjected to steel at 400°F. This figure resulted from exposures for a six day period where an initial higher rate of decomposition was disregarded. Groeneveld indicated that Freon 12 was susceptible to decomposition at temperatures above 600°F in the presence of moisture. This temperature was not reached in his experimental program. In a private communication Groeneveld indicated that the test section appeared clean after the experimental program was completed.

3.7 Discussion of Generalized Dispersed Flow Model
While the dispersed flow film boiling model adapted from the Hynek and Groeneveld models and used in this work appears not to greatly improve the accuracy of predicting post critical heat transfer, it is fundamentally more self consistent in its basic components than either of the parent models as previously discussed. Strictly speaking the model is valid only in the dispersed flow regime which is generally found at void fractions exceeding 80%. But by using the Hynek initialization method the froth regime encountered at void fractions above 10% can be transformed to a hypothetical dispersed flow regime whose initial drop sizes and velocities are given by the Hynek method. From the comparison with the nitrogen data which, as Forslund visually determined, was in the froth regime this extrapolation technique appears to work quite well. It is postulated, therefore, that the dispersed flow model can be used in any post critical flow regime with the accuracy increasing as the actual flow regime more closely resembles that of dispersed flow. There is a limiting mass flux, however, below which the model will not work, and that is the value necessary to give a sufficient vapor velocity to carry the liquid out the tube. This critical mass flux is derived in Appendix E assuming that the liquid fraction can be represented as droplets for any quality dryout. Using the Groeneveld wall-to-droplet term
allows for a more physical understanding of the correlating term, $\delta_{\text{film}}$, than $K_2K_2$ used by Forslund to extrapolate Baumeister's theory for heat transfer to a drop sitting on a flat plate. While it is physically reasoned that the average height that the droplet population remains away from the heated surface is a function of the surface temperature, choosing an average value of $1 \times 10^{-4}$ feet tends to predict the heat transfer data for the three fluids tested. $\delta_{\text{film}}$ is also capable to a certain extent of correcting for the errors associated with the vapor heat transfer coefficient by making up in the droplet term the underprediction of the vapor correlation. The modified McAdams equation for single phase vapor heat transfer gives reasonable results for the three fluids tests. It is felt that a greater degree of accuracy can be obtained by using a vapor correlation tailored for the particular fluid and operating conditions desired. The final variable which is very sensitive in the model is the position and quality of the dryout point. The technique of predicting post dryout wall temperatures used in this work is simplified because the dryout points were known in advance. (For zero quality dryouts a small positive value was used, and the calculation procedure was started there). For cases where the dryout point is not specified CHF correlations have to be relied on. This could introduce considerable uncertainties in the
predictions. The sensitivity on the quality variable is reduced for those dryout cases where the wall temperature profiles decrease after dryout with quality as is the case for low quality-high mass flux dryouts.

The general use of the dispersed flow film boiling code to predict the behavior of systems in post dryout is relatively combersome. The integration procedure is sensitive to the step size chosen. A small step size is needed just after dryout where the wall temperature profile is rapidly changing. This can subsequently be gradually increased to reduce computation costs. In the next chapter procedures are presented which reduce the dispersed flow film boiling model to an algebraic equation in terms of system variables which is capable of predicting post critical heat transfer equally well.
IV Generalized Post Critical Heat Transfer Correlation

4.1 Concept of Correlation

The complexity of predicting the post critical heat transfer is a result of the departure from the equilibrium state after dryout. This nonequilibrium is manifested in the superheating of the vapor phase with the result that the mass fraction of vapor, termed in this work the actual quality, is always less than that value calculated by an equilibrium energy balance. The degree to which the vapor is superheated and the actual quality deviates from the equilibrium at any point downstream of the dryout for a particular fluid is related to the distance from dryout, the dryout quality, the mass flux and to some extent the heat flux as can be deduced from the analysis of the dispersed flow model presented in Chapter 3. Figure 16 reproduced from Forslund's thesis demonstrates quite well the nonequilibrium of post critical heat transfer. The mass flux and equilibrium quality effects on the actual quality are indicated. As the mass flux increases the nonequilibrium at any distance from dryout decreases. This figure also indicates a small diameter effect on the nonequilibrium.

The premise of the post critical heat transfer prediction scheme presented in this chapter is that the thermal nonequilibrium in post dryout indicated by Figure 16 can be approximated
FIGURE 16 CALCULATED $X_A$ VS $X_E$ FOR NITROGEN AT SEVERAL MASS FLUXES (FROM FORSLUND REF. 5)
with the following linear relationship between \( X_A \) and \( X_E \)

\[
(X_A - X_{DO}) = \kappa (X_E - X_{DO})
\]  

(4.1)

where \( \kappa \) is viewed as a correlating constant. A reexamination of the two-step dispersed flow model of Chapter 3 in light of Equation (4.1) resulted in the development of the generalized post critical heat transfer correlation.

The general form of the correlation is taken directly from the two-step heat transfer process.

\[
\frac{q}{A} = h_{w,v} (T_w - T_v) + h_{w,\delta} (T_w - T_{sat})
\]  

(4.2)

where \( h_{w,v} \) as well as \( h_{w,\delta} \) can be defined in terms of void fraction and actual quality to be

\[
h_{w,v} = 0.023 \frac{k_v}{D_T} \left( \frac{G X_A \Delta T}{\mu \alpha} \right)^{0.8} \left( \frac{Pr_v}{Pr_w} \right)^{1/3} \left( \frac{\mu_v}{\mu_w} \right)^{0.14} \left[ 1 + 0.3 \left( \frac{D_T}{L_{DO} + 0.01 D_T} \right) \right]^{0.7}
\]  

(4.3)

and

\[
h_{w,\delta} = \frac{k_v \delta_{film}}{\delta_{film}} (1-\alpha) \exp \left[ -\frac{2D_T}{L_{DO}} \right]
\]  

(4.4)

The quantities needed to determine the heat flux in the post critical region are the vapor temperature, \( T_v \), the void fraction, \( \alpha \), \( \delta_{film} \) and \( \kappa \) as defined in equation (4.1).

The following sections describe the formulations for \( T_v \) and \( \alpha \) and the procedures for obtaining values of \( \delta_{film} \) and \( \kappa \) to predict post critical heat transfer data. The resulting correlation procedure is compared with the transient
nitrogen data taken in this investigation as well as with steady state nitrogen data in the literature.

Procedures for obtaining a bounded solution for the heat transfer in post dryout are presented and a preliminary check of the correlating model is made with water and Freon 12.
4.2 Determination of Vapor Temperature in Post Dryout

With the approximation to the actual quality curve as given in equation (4.1) the vapor temperature can be obtained immediately upon writing the energy balance between the evaporating liquid and superheating vapor. This equation is simply

\[ X_E h_{fg} = X_A [h_{fg} + C_{pv} (T_v - T_{sat})] \]  

(4.5)

Substituting equation (4.1) into (4.5) and rearranging gives the vapor temperature as

\[ (T_v - T_{sat}) = \frac{h_{fg} (1 - \kappa)}{C_{pv}} \frac{X_E - X_{DO}}{X_{DO} + \kappa (X_E - X_{DO})} \]  

(4.6)

For liquid nitrogen, \( C_{pv} \) is not a function of \( T_v \) which allows one to obtain \( T_v \) directly in terms of \( X_E \), \( X_{DO} \) and \( \kappa \). For water at 1000 psia and to some extent Freon 12 at 155 psia \( C_{pv} \) is a function of \( T_v \). This necessitates iterating equation (4.6) with \( C_{pv} \) vs \( T_v \) data.
4.3 Determination of Void Fraction After Dryout

The slip ratio and void fraction can be determined at dryout from Hynek's initialization procedure. Determining the void fraction, hence, the vapor velocity after dryout requires the simultaneous solution of the drop diameter gradient, equation (3.12), the droplet velocity gradient, equation (3.10), continuity equation, similar to equation (3.9) together with the droplet breakup criterion, equations (3.15 and 3.16).

The drop diameter gradient can be integrated directly when equation (4.1) is differentiated and substituted into equation (3.12). The resulting drop diameter gradient becomes

\[ d\delta = -\frac{\kappa \delta_D^3 dX_E}{3(1-X_D^0)\delta^2} \]  \hspace{1cm} (4.7)

Integrating (4.7) from \( \delta = \delta_D^0 \) at \( X_E = X_D^0 \) to \( \delta = \delta' \) at \( X_E = X_E' \) gives

\[ \delta'^3 - \delta_D^3 = -\kappa \delta_D^3 \frac{(X_E' - X_D^0)}{(1 - X_D^0)} \]  \hspace{1cm} (4.8)

rearranging (4.8) gives the drop diameter as a function of \( X_D^0, X_E \) and \( \kappa \) to be

\[ \delta = \delta_D^0 \left[ 1 - \frac{\kappa(X_E - X_D^0)}{(1 - X_D^0)} \right]^{1/3} \]  \hspace{1cm} (4.9)

The droplet velocity gradient becomes
\[
\frac{dV_L}{dz} = \frac{3C_D \rho_V (V_g - V_L)^2}{4V_L \rho_L \delta_{DO}} \left[ 1 - \frac{\kappa (X_E - X_{DO})}{(1 - X_{DO})} \right] - \frac{\delta_{DO}}{V_L} \left[ 1 - \frac{\rho_V}{\rho_L} \right] \tag{4.10}
\]

The vapor velocity can be eliminated using the continuity equation

\[
V_g = \frac{G X_A}{\rho_V \left[ 1 - \frac{G(1-X_A)}{\rho_L V_L} \right]} \tag{4.11}
\]

where \(X_A\) is given by equation (4.1). Equation (4.10) and (4.11) must be solved together with the critical Weber number criterion, equations (3.15) and (3.16), to obtain the values of liquid and vapor velocities necessary to determine the slip ratio and void fraction. The solution of these two quantities will be in terms of system variables and \(\kappa\). The critical Weber number is utilized in the same manner as discussed in Section 3.4 for the updating of \(\delta_{DO}\) and \(X_{DO}\) in equation (4.9) after each shatter. It is obvious from the fact that equation (4.10) is a non linear differential equation that a closed form solution cannot be directly obtained from this set of equations. An approximate closed form solution was obtained indirectly, however which represents quite accurately the void fraction after dryout.
The procedure consists of solving the above set of equations for $V_g$ and $V_l$ on a digital computer using a Runge-Kutta integration scheme and applying a simple correlation technique to the resulting slip ratios in terms of the important variables. As $V_g$ and $V_l$ are each functions of $G$, $q/A$, $X_E$, $X_{DO}$ and $\kappa$, a parametric study of each of these variables was undertaken to ascertain their effect on the resulting slip ratio in post dryout. In the parametric study, $G$ was varied from $G_{\text{crit}}$ (defined in Appendix E) to 500,000 lbm/hr-ft$^2$. $X_{DO}$ was varied from 5% to 50%. $\kappa$ was varied from 1.0 to 0.2. $X_E$ is the dependent variable. Figures 17 and 18 give the results for nitrogen for $\kappa = 1.0$ and $\kappa = 0.5$ respectively. The slip ratio varies in a rather simple predictable manner. It starts at the dryout slip, $S_{DO}$ (determined from Hynek's procedure given $G$, $X_{DO}$ and $q/A$ at the dryout point), and decreases as $X_E$ increases until $S = 1$ at $X_A = 1$ (the point where $\kappa(X_E - X_{DO})$. A simple equation of the form

$$\frac{1-S}{1-S_{DO}} = [1 - \frac{\kappa(X_E - X_{DO})}{(1 - X_{DO})}]^D$$  \tag{4.12}$$

gives the observed trends for the slip ratio. The variables, $G$ and $q/A$ are hidden in the determination of $S_{DO}$. As $S$ does not physically remain at $S_{DO}$ for all values of $X_E$ where $\kappa = 0$ as equation (4.12) indicates, $D$ must be an inverse
FIGURE 17  COMPARISON OF COMPUTER GENERATED SLIP RATIOS WITH SLIP CORRELATION (EQN. 4.12)... $\kappa = 1$

NITROGEN

$P = 20$ psia

- $25,000$ lbm/hr-ft$^2$
- $75,000$
- $125,000$
- $200,000$

$q/A = 10,000$ btu/hr-ft$^2$

$x_{D0} = 5\%$

Equation (4.12)

with $\kappa = 1$

$D = 5$
NITROGEN

$P = 20$ psia

Computer Solution

- $25,000$ lbm/hr-ft$^2$
- $75,000$
- $125,000$
- $200,000$

$q/A = 10,000$ btu/hr-ft$^2$

$X_{D0} = 5\%$

---

Equation (4.12)

with $\kappa = .5$

$D = 7$

---

**FIGURE 18** COMPARISON OF COMPUTER GENERATED SLIP RATIOS
WITH SLIP CORRELATION (EQN. 4.12)... $\kappa = .5$
function of \( \kappa \) so that \( D \) will increase as \( \kappa \) decreases to the point where \( D = \infty \) at \( \kappa = 0 \). Equation (4.12) is plotted on Figure 17 using a value of \( D = 5 \) and predicts reasonably well the four mass fluxes given. A value of \( D = 7 \) gives a reasonable result for the case of \( \kappa = .5 \) given in Figure 18. This indicates that whatever \( G \) effect there might be on \( D \) is quite small and restricted to only the lowest values of \( G \). If one then proceeds to correlate \( S \) vs \( X_0 \) at the lowest value of \( S_{D0} \) for a given \( \kappa \) the resulting correlation will of necessity improve as \( G \) or \( X_{D0} \), hence \( S_{D0} \), increases.

Considering now that \( D \) is only a function of \( \kappa \) a series of curves similar to those presented in Figures 17 and 18 were developed for nitrogen, Freon 12 and water. The resulting values of \( D \) vs \( \kappa \) for each of the three fluids are tabulated in Table 2. Also given in Table 2 are the coefficients to the equation

\[
D = \frac{A}{\kappa B} \quad (4.13)
\]

which gives the correct functional dependence of \( \kappa \) on \( D \). For the case of nitrogen \( D \) must equal 5 when \( \kappa = 1.0 \), and \( D \) must approach infinity when \( \kappa \) approaches 0 . Equation (4.13) will give this dependence.

The void fraction in post dryout can now be obtained from the definition of void fraction. Substituting in (4.12)
<table>
<thead>
<tr>
<th>Fluid</th>
<th>$\kappa$</th>
<th>$D$</th>
<th>$A$</th>
<th>$B$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nitrogen</td>
<td>1.0</td>
<td>5.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>.5</td>
<td>7.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>.2</td>
<td>10.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.0</td>
<td>$\infty$</td>
<td>5.0</td>
<td>.486</td>
</tr>
<tr>
<td>Water</td>
<td>1.0</td>
<td>2.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>.5</td>
<td>3.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>.2</td>
<td>4.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.0</td>
<td>$\infty$</td>
<td>2.5</td>
<td>.264</td>
</tr>
<tr>
<td>Freon 12</td>
<td>1.0</td>
<td>1.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>.5</td>
<td>1.3</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>.2</td>
<td>1.9</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.0</td>
<td>$\infty$</td>
<td>1.0</td>
<td>.37</td>
</tr>
</tbody>
</table>

Tabulation of $\kappa$ vs $D$ for Correlating Post Dryout Slip Ratios

**TABLE 2**
and (4.13) the resulting void fraction becomes

$$
\alpha = \frac{X_A}{X_A + \frac{\rho_v}{\rho_\ell} (1 - (1 - S_{DO}) \left( \frac{1 - X_A}{1 - X_{DO}} \right) ) (1 - X_A}) \quad (4.14)
$$

where $X_A$ is determined from (4.1) and the values for A and B obtained from Table 1 for one of the three fluids correlated.

Figure 19 compares (4.14) against the computer solution of the void fraction for nitrogen, water, and Freon 12 at the conditions $G = 250,000 \text{ lbm/hr-ft}^2$ and $X_{DO} = 5\%$ with $\kappa = 0.5$. The temperatures dependence of the vapor density was included using equation (4.6) to determine $T_v$ as a function of $X_E$ and $\kappa$. As can be seen in Figure 19 the approximate closed form solution given by equation (4.14) is a very good representation of the computer solution for void fraction. The discrepancies associated with the approximate slip correlation are not wholly transferred to the void fraction equation. No effort will be made to quantify the errors associated with the use of (4.14) as indeed one does not know really how well the differential equations predict the behavior of an actual system. It is observed though that the approximate solution satisfies the end points of $X_A = X_{DO}$ and $X_A = 1$ and gives a reasonable reproduction of the exact solution of the differential
FIGURE 19 COMPARISON OF CORRELATED VOID FRACTION IN POST DRYOUT FOR SEVERAL FLUIDS AGAINST COMPUTER SOLUTION OF GOVERNING EQUATIONS

Void Fraction Correlation Eqn. (4.14)

- Nitrogen
- Water
- Freon 12

Computer Solution

\[ G = 250,000 \text{ lbm/hr-ft}^2 \]
\[ q/A = 10,000 \text{ btu/hr-ft}^2 \]
\[ X_{DO} = 5\% \]
\[ \kappa = 0.5 \]
equations describing the phenomenon. The accuracy of reproduction is increased as the void fraction increases to a point where the approximate calculation nearly equals the computer solution at void fractions above 50%.
4.4 Correlation of Post Critical Heat Transfer
For Nitrogen

Substituting, now, equations (4.6) and (4.14) back into equation (4.2) gives the basic framework for the post critical heat transfer correlation in terms of two as yet unspecified constants, \( \delta_{\text{film}} \) and \( \kappa \). The resulting substitution gives:

\[
q/A = 0.023 \frac{k}{D_T} \left( \frac{G_{X_A}D_T}{\mu_a} \right) \frac{Pr}{Pr} \left( \frac{\mu_v}{\mu_w} \right)^{1/3} \] 0.8 14

\[
[1 + 0.3\left( \frac{D_T}{L_{DO} + 0.1D_T} \right)^{0.7}] [ (T_w - T_{\text{sat}}) - \frac{h_f g(1 - \kappa)(X_E - X_{DO})}{C_p \rho_v X_A} ]
\]

\[
+ \frac{k_v f(1 - \alpha)}{\delta_{\text{film}}} \exp \left[ - \frac{2D_T}{L_{DO}} (T_w - T_{\text{sat}}) \right]
\]

(4.15)

where \( \alpha = \frac{X_A}{X_A + \frac{\rho_v}{\rho_g} \left( 1 - (1 - S_{DO}) \left( \frac{1 - X_A}{1 - X_{DO}} \right) \right)^{5/\kappa} - 0.486}

\[
X_A = X_{DO} + \kappa (X_E - X_{DO})
\]

\( S_{DO} \).....Iteration of equations (3.8) and (3.9)

[Subroutine DOCAL]

The general form of the post critical heat transfer equation is

\[
q/A = f(X_E, X_{DO}, L_{DO}, G, \Delta T_{\text{sat}}, \delta_{\text{film}}, \kappa)
\]

(4.16)
The dryout length, $L_{D0}$, is calculated by

$$L_{D0} = \frac{G h_f g D_T}{4} \int_{X_{D0}}^{X_E} \frac{dX'}{q/A}$$

(4.17)

where $q/A$ is the heat flux distribution between $X_{D0}$ and $X_E$. For uniformly heated tubes $L_{D0}$ is not an independent variable since it is calculated from $q/A$, $X_{D0}$, $X_E$, and $G$. For the test section used here $L_{D0}$ is an independent variable being determined by the heat flux in the preheater.

The determination of the two correlating parameters, $\delta_{film}$ and $\kappa$, in equation (4.18) is achieved in the following manner. First the value of $\kappa$ is selected via a technique to be described next. Anticipating the value of $\kappa$, $\delta_{film}$ will be determined from comparisons of equation (4.18) with transient nitrogen data. Although $\delta_{film}$ has been optimally determined in Chapter 3 such that the differential dispersed flow model affects the best
prediction of the temperature-length data, it is expected that a different value of $\delta_{\text{film}}$ will be necessary for the post critical heat transfer correlation to correct, somewhat, the errors involved in assuming a linear variation of $X_A$ vs $X_E$.

4.4.1 Evaluation of $\kappa$

To determine $\kappa$, the $X_A$ vs $X_E$ curve must be known. As physical measurements are cumbersome and involves a large amount of data scatter (see Forslund), alternative methods are required. There are two methods available. First a good estimation of the actual quality can be obtained from post critical heat transfer data by assuming that all heat is transferred by the vapor. This assumption together with the energy balance for obtaining the vapor temperature provide the two following equations for obtaining $X_A$ vs $X_E$ given $q/A$ and $T_w$.

$$q/A = 0.023 \frac{k_v}{D_T} \left( \frac{G X_A D_T}{\mu \alpha_v} \right)^{0.8} Pr_v^{1/3} (T_w - T_v)$$ (4.18)

where $\alpha = \frac{X_A}{X_A + \frac{\rho_v}{\rho_L} S(1 - X_A)}$ (4.19)

$$X_E h_{fg} = X_A [h_{fg} + C_{\rho v} (T_v - T_{sat})]$$

As a first approximation the slip, $S$, in equation (4.19)
can be obtained from the homogeneous model (i.e. S=1). A more exact method would be to use equations (4.13) and (4.14) for the cases where the correlating constants have been determined.

Figure 16 presents the results Forslund obtained from his data upon applications of equations (4.13) and (4.19). A second method which was used extensively in this investigation consists of generating $X_A$ vs $X_E$ curves from the two step model using FILMBOIL. Using FILMBOIL to generate the $X_A$ vs $X_E$ curve is especially helpful to predict $\kappa$ for cases where no physical data is as yet available (i.e. water and Freon 12 data at low mass fluxes and qualities).

With the $X_A$ vs $X_E$ curve in hand the value of $\kappa$ is obtained by visually obtaining the best fit of the curve. This requires some explanation as we have to be consistent in the evaluation procedure if some quantification of variables affecting $\kappa$ is to be expected. In essence what we are trying to do is fit a straight line to a curve. Of necessity this approximation has a limited range of accuracy. In this investigation $\kappa$ was evaluated such that equation (4.1) approximated $X_A$ over the largest possible range of $X_E$ without introducing significant error in $X_A$. The general evaluation criterion used was that $\kappa$ be decreased from 1 to a value that caused $X_A$ calculated from equation
(4.1) to differ from the generated $X_A$ vs $X_E$ curve by
.02 to .03 in a range to the left of the intersection of
equation (4.1) with the calculated $X_A$ curve. A sample
of this graphical evaluation technique is given in
Figure 20 where $\kappa$ is evaluated from an $X_A$ vs $X_E$
generated from FILMBOIL for Nitrogen at $G = 70,000$
lbm/hr-ft$^2$ and $X_{DO} = 5\%$. The intersection point and
maximum allowable deviation point are clearly indicated.
For this case the approximate solution starts to deviate
significantly at an $(X_E - X_{DO})$ of 100\%.

Before going further into the evaluation procedure,
a general idea of what influences $\kappa$ is necessary in order
to localize the evaluation and eventual correlation pro-
cedures to the relevant variables. As $\kappa$ is in essence a
measure of the nonequilibrium of the flow in post dryout,
whatever tends to affect the degree of non equilibrium will
have an affect on $\kappa$. Returning again to Figure 16 one can
obtain considerable insight into the variables that
affect $\kappa$. $G$ is definitely a strong variable on $\kappa$. A
value of $\kappa = .55$ in equation (4.1) appears to predict $X_A$
for most of the range of $X_E$ for $G = 70,000$ lbm/hr-ft$^2$
whereas a value of $\kappa = .71$ predicts the case $G = 190,000$
lbm/hr-ft$^2$. Figure 16 also indicates that heat flux and
tube diameter have a negligible effect on the nonequilibrium.
The next and last variable considered to affect $\kappa$ is the
Figure 20  Best fit process for the determination of $\kappa$

Equation (4.1) with $\kappa = 0.58$

Approximation

Intersection point

Maximum allowable error point

NITROGEN

$P = 20$ psia
$G = 70,000$ lbm/hr-ft$^2$
$q/A = 10,000$ btu/hr-ft$^2$
$D_T = 0.323$ inches
dryout quality. This variable can be rationalized from the observation that at progressively higher dryout quantities there is less liquid to evaporate and cool the vapor. Because the vapor superheats faster the amount of non equilibrium increases proportionately. Figure 21 graphically depicts this variable affect. It consists of $X_A$ vs $X_E$ curves calculated from Filmboil for water at $G = 250,000$ lbm/hr-ft$^2$ at two values of dryout quality $X_{DO} = 20\%$ and $X_{DO} = 80\%$. The straight line approximation is also plotted for each of the two $X_A$ vs $X_E$ curves. $\kappa = .62$ closely approximate the $X_{DO} = 20\%$ case and $\kappa = .7$ for the $X_{DO} = 80\%$ case. This indicates almost a linear relationship between $\kappa$ and the quantity $(1-X_{DO})$. As $X_{DO}$ increases $\kappa$ decreases. Graphically this relationship results from use all $X_A$ vs $X_E$ curves with $X_{DO}$ greater than zero will eventually merge with the $X_A$ vs $X_E$ curve for $X_{DO} = 0$. In order to do this the slope of the linear approximation to the curves with higher dryout qualities must decrease. Including $X_{DO}$ as a parameter in some measure includes the $q/A$ affect which is not explicitly a variable.

With these two variables, $G$ and $(1-X_{DO})$, an extensive investigation was carried out to determine the best fit value of $\kappa$ for nitrogen using computer generated curves of $X_A$ vs $X_E$ and the evaluation technique just described.
Figure 21: Demonstration of dryout quality effect on $\kappa$

- $X_{DO} = 80\%$
- $X_{DO} = 20\%$
- $\kappa = 0.17$
- $\kappa = 0.62$

Water
- $P = 1000$ psia
- $G = 250,000$ lbm/hr-ft$^2$
- $q/A = 100,000$ btu/hr-ft$^2$
- $D_T = 0.497$ inches

Equation:
\[ X_A - X_{DO} = \kappa (X_E - X_{DO}) \]
Included, also, in this investigation were evaluations of $\kappa$ vs $G$ and $(1-X_{DO})$ for water and Freon 12. The results of this investigation are tabulated in Table 3.

As a tabulation is not the most effective way of using the results as well as anticipating a quantitative need of $\kappa$ for the final nitrogen correlation an attempt was made to develop a generalized curve for the best fit value of $\kappa$ in terms of a nondimensionalized form of the two independent parameters, $G$ and $X_{DO}$ based on the three fluids investigated. The following nondimensional equation was selected to represent the variation of $\kappa$ with $X_{DO}$ and $G$

$$\kappa = f\left(\frac{G^2 D_T}{\rho_v T} \left(1-X_{DO}\right)^m \right)$$  \hspace{1cm} (4.20)

The nondimensional form of $G$ was developed from the Weber number assuming the velocity difference could be estimated by $G/\rho_v$. This Weber number formation was initially chosen to account for the fact that small droplets reduce the nonequilibrium. However, the drop diameter vs length curve in post dryout was shown by Forslund to be relatively unaffected by $G$ whereas increasing $q/A$ significantly shifts the curve downward. The reason $G$ decreases nonequilibrium whereas $q/A$ does not is that for a given drop size a higher $G$ results in a larger number of drops per unit volume which increases the cooling area seen by the vapor.
<table>
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<th>Fluid</th>
<th>$D_T$ (feet)</th>
<th>$G \times 10^{-6}$ (lbm/hr-ft²)</th>
<th>$X_{DO}$</th>
<th>$\kappa$</th>
<th>$\Pi^+$</th>
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Tabulation of Best Fit Value of $\kappa$

TABLE 3

$\Pi^+ = \frac{G \sqrt{D_T}}{\rho_v \delta} \left(1 - X_{DO}\right)^5$
This discussion indicates that the parameter chosen to nondimensionalize $G$ has really no physical significance. The parameter was retained however for lack of a reasonable alternative*. The resulting plot of $\kappa$ vs

$$K \text{ vs } G \sqrt[5]{\frac{D_T}{\rho_v} \frac{1}{\sigma} (1-X_{D0})^5}$$

is given in Figure 22. Values of $m = 1/2$ and $n = 5$ were chosen to reduce the observed banding of the points as much as possible. The larger dependance of $(1-X_{D0})$ on $\kappa$ relative to $G$ is reasonable in light of discussion in Section 4.5. A distinction is made for those points whose dryout quality is less than 10% as those points will be unaffected by the power on the $(1-X_{D0})$ term. In fact looking at these points will give a good idea of how well the Weber number formulation for $G$ correlates the three fluids. There appears to be significant deviations between points for different fluids above and beyond that expected to be obtained from the subjective evaluations of $\kappa$ even though there is a definite trend with little data scatter for each individual fluids. This becomes more obvious when one takes into account all the data for each fluid.

The best fit values of $\kappa$ for water fall on one line which

* A Reynolds number formulation was discarded because $D_T$ would be given equal weight as $G$ where in fact it has little affect on $\kappa$. 
Figure 22: Correlation of best fit value of $\kappa$ vs $G \sqrt{D_T/\rho_v \sigma^3 (1 - X_{DO})^5}$.
is significantly above the data for the other two fluids. Freon 12 data falls on another line with Nitrogen data dividing itself among the two lines. At high G and low $X_{DO}$ the best fit values of $\kappa$ for nitrogen fall on the water line and at low G and high $X_{DO}$ the data falls on the Freon 12 line. To facilitate the evaluation of $\kappa$ two equations were developed for these lines.

$$\kappa = 0.0674 \log_e \left[ G \frac{D_T}{\rho_v \sigma} (1-X_{DO})^5 \right] + 0.402$$

(4.21)

Water

$$\kappa = 0.0811 \log_e \left[ G \frac{D_T}{\rho_v \sigma} (1-X_{DO})^5 \right] + 0.236$$

(4.22)

Freon 12

Equation (4.21) can be used to predict the best fit value for water and equation (4.22) for Freon 12. It is recommended that one use Figure 22 to determine the best fit value of $\kappa$ for liquid nitrogen. For the actual data comparison of nitrogen data with equation (4.15) to determine the correlating constant, $\delta_{film}$ as presented next the data for $\kappa$ in Table 3 is used.

4.4.2 Determination of $\delta_{film}$ from Transient and Steady State Nitrogen Data.

With the generated table of $\kappa$ vs G and $(1-X_{DO})$ for nitrogen an extensive comparison of the transient nitrogen data obtained in this investigation was undertaken. The
transient data of particular interest is that obtained in Runs 94-100 where the preheater tube was in the dry mode. The data for these runs will therefore have a considerable superheat effect. Figures 23-29 present the results of the evaluation procedure of δ_film for these particular data runs. These figures present data for two mass fluxes, G = 215,000 and 130,000 lbm/hr-ft^2 and a range of qualities for each mass flux. In all cases δ_film = 4x10^{-4} ft predicts the data extraordinarily well. A value of δ_film = 1x10^{-4} ft as obtained for the differential two steps model in the post critical heat transfer correlation gave too high a prediction for the transient data. It is extremely difficult to explain this discrepancy of δ_film between the two models as there are so many competing effects that are influenced by the approximation procedure used in the post critical heat transfer correlation. A cursory check of the approximation procedure embodied in equation (4.1) on the main variables affecting the total heat transfer (T_v, x_A, α) indicated that in applying the approximation to the differential two step model a smaller value of δ_film (i.e. a larger wall-to-drop heat flux) would be needed instead of the observed higher value. All computer codes used in the application of the equations for both models were scrupulously checked and found free of errors. The author concludes that an involved analysis of
NITROGEN
P = 20 psia

○ - Run 100
G = 123,600 lbm/hr-ft²
X_E = 20%
X_D0 = 2%
L_D0 = 48 inches
T_sat = -316°F
D_T = .4 inches

Prediction:
κ = .62
δ_film = 4 x 10⁻⁴ ft.

FIGURE 23 COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION
WITH TRANSIENT NITROGEN DATA - LONG DRYOUT LENGTH
NITROGEN

$P = 20 \text{ psia}$

- Run 99
  
  $G = 123,600 \text{ lbm/hr-ft}^2$
  
  $X_e = 41\%$
  
  $X_{DO} = 2\%$
  
  $L_{DO} = 48 \text{ inches}$
  
  $T_{sat} = -316 \degree F$
  
  $D_T = .4 \text{ inches}$

Prediction:

$\kappa = .62$

$\delta_{film} = 4 \times 10^{-4} \text{ ft.}$

$\left( T_{wall} - T_{sat} \right) \degree F$

FIGURE 24 COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH TRANSIENT NITROGEN DATA - LONG DRYOUT LENGTH
NITROGEN

\( P = 20 \text{ psia} \)

\( \triangle - \text{Run 98} \)

\( G = 123,000 \text{ lbm/hr-ft}^2 \)

\( X_E = 71\% \)

\( X_D = 2\% \)

\( L_D = 48 \text{ inches} \)

\( T_{sat} = 317^\circ F \)

\( D_T = 0.4 \text{ inches} \)

Prediction:

\( \kappa = 0.62 \)

\( \delta_{film} = 4 \times 10^{-4} \text{ ft} \)

FIGURE 25 COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH TRANSIENT NITROGEN DATA - LONG DRYOUT LENGTH
NITROGEN

P = 20 psia

Prediction:

\[ \kappa = 0.71 \]
\[ \delta_{\text{film}} = 4 \times 10^{-4} \text{ ft.} \]

- Run 97
  - \( G = 213,000 \text{ lbm/hr-ft}^2 \)
  - \( X_{E} = 22\% \)
  - \( X_{D0} = 2\% \)
  - \( L_{D0} = 48 \text{ inches} \)
  - \( T_{\text{sat}} = -315^\circ \text{F} \)
  - \( D_{T} = 0.4 \text{ inches} \)

**FIGURE 26** COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH TRANSIENT NITROGEN DATA - LONG DRYOUT LENGTH
NITROGEN

$P = 20$ psia

**PREDICTION:**

$\kappa = 0.71$

$\delta_{film} = 4 \times 10^{-4} \text{ft}$

- Run 96
  - $G = 214,600 \text{ lbm/hr-ft}^2$
  - $X_E = 40\%$
  - $X_D0 = 2\%$
  - $L_D0 = 96 \text{ inches}$
  - $T_{sat} = -316^\circ F$
  - $D_T = 0.4 \text{ inches}$

**FIGURE 27** COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH TRANSIENT NITROGEN DATA - LONG DRYOUT LENGTH
NITROGEN

P = 20 psia

Prediction:

\[ \kappa = 0.71 \]

\[ \delta_{\text{film}} = 4 \times 10^{-4} \text{ft.} \]

- Run 94
  - \( G = 213,000 \) lbm/hr-ft\(^2\)
  - \( X_E = 60\% \)
  - \( X_{DO} = 2\% \)
  - \( L_{DO} = 96 \) inches
  - \( T_{\text{sat}} = -315^\circ \text{F} \)
  - \( D_T = 0.4 \) inches

**FIGURE 28** COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH TRANSIENT NITROGEN DATA - LONG DRYOUT LENGTH
NITROGEN

$P = 20$ psia

Prediction:

$\kappa = 0.71$

$\delta_{\text{film}} = 4 \times 10^{-4}$ ft.

$\triangle$ - Run 95

$G = 216,000$ lbm/hr-ft$^2$

$X_E = 76\%$

$X_{DO} = 2\%$

$L_{DO} = 96$ inches

$T_{sat} = -315^\circ F$

$D_T = 0.4$ inches

FIGURE 29 COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH TRANSIENT NITROGEN DATA - LONG DRYOUT LENGTH
this discrepancy is not warranted once the view that $\delta_{\text{film}}$ is a correlating constant for the set of equations composing the post critical heat transfer correlation that is determined by data comparison. To be consistent in the idea that $\delta_{\text{film}}$ is a correlating constant no further adjustment should have to be made in it to predict other nitrogen data. This is a somewhat naive attitude if $\delta_{\text{film}}$ is taken in its physical sense of an average height which drops remain above the heated surface.

The most convincing support of $\delta_{\text{film}} = 4 \times 10^{-4}$ for nitrogen comes in comparing the prediction scheme with the transient nitrogen data for low values of $L_{DO}$. A low value of $L_{DO}$ means that very little superheat of the vapor can be affected and that the equilibrium value of $\kappa$ (i.e. $\kappa = 1$) should predict the data. This automatically reduces the problem of evaluating two correlating constants as $\delta_{\text{film}}$ is the only variable left to be correlated. As the bulk of the transient nitrogen data consists of low $L_{DO}$ data an extensive check on $\delta_{\text{film}}$ can be achieved. One must keep in mind that only the higher range of $(T_w - T_{\text{sat}})$ is certain of being free from heat losses as explained in Appendix A.

Figures 30-34 present the results of some selected runs where $L_{DO}$ equaled 3 inches or less. It is seen that $\delta_{\text{film}} = 4 \times 10^{-4}$ does a very good job of predicting this
NITROGEN

$P = 15\text{ psia}$

$\Delta - \text{Run } 127$

$\bigcirc - \text{Run } 102$

$G = 31,000 \text{ lbm/hr-ft}^2$

$X_E = X_{DO} = 30\%$

$L_{DO} = 2.\text{ inches}$

$T_{sat} = -320 \degree F$

$D_T = .4 \text{ inches}$

Prediction:

$\kappa = 1.$

$\delta_{film} = 4 \times 10^{-4} \text{ ft}$

**Figure 30** Comparison of postcritical heat transfer correlation with transient nitrogen data - short dryout length
NITROGEN

\[ P = 20 \text{ psia} \]

- Run 107
- \( G = 61,000 \ \text{lbm/hr-ft}^2 \)
- \( X_E = X_D = 12\% \)
- \( L_D = 3 \text{ inches} \)
- \( T_{sat} = -316 \ ^\circ F \)
- \( D_T = .4 \text{ inches} \)

Prediction:
- \( \kappa = 1.0 \)
- \( \delta_{film} = 4 \times 10^{-4} \text{ ft} \)

**FIGURE 31** COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH TRANSIENT NITROGEN DATA—SHORT DRYOUT LENGTH
NITROGEN
P = 18 psia

$G = 120,000 \text{ lbm/hr-ft}^2$
\[X_e = X_{DO} = 75\%\]
\[L_{DO} = 2 \text{ inches}\]
\[T_{sat} = -318^\circ F\]

Prediction:
$\kappa = 1$
\[\delta_{film} = 4 \times 10^{-4} \text{ ft}\]

FIGURE 32 COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH TRANSIENT NITROGEN DATA - SHORT DRYOUT LENGTH
FIGURE 33 COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH TRANSIENT NITROGEN DATA - SHORT DRYOUT LENGTH
NITROGEN
P = 20 psia

$\Delta$ - Run 126

$G = 201,000 \text{ lbm/hr-ft}^2$

$X_E = X_{DO} = 11\%$

$L_{DO} = 2 \text{ inches}$

$T_{sat} = -318 \, ^\circ F$

Prediction:

$\kappa = 1.$

$\delta_{film} = 4 \times 10^{-4} \text{ ft}$

$\delta_{film} = 1 \times 10^{-4} \text{ ft}$

FIGURE 34 COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH TRANSIENT NITROGEN DATA - SHORT DRYOUT LENGTH
data. Also shown on some of these curves is the value of heat flux for $\delta_{\text{film}} = 1 \times 10^{-4}$. This value is generally outside the data scatter observed. In Figure 33 the data at low values of $(T_w - T_{\text{sat}})$ is considerably below the prediction. This is a clear indication of heat gains affecting the data as discussed in Appendix A. As $(T_w - T_{\text{sat}})$ increases, the discrepancy decreases and appears to asymptote to the prediction at elevated $(T_w - T_{\text{sat}})$ values (hence towards less heat gains). Figure 34 indicates that more heat flux than that provided by $\delta_{\text{film}} = 1 \times 10^{-4}$ ft is needed to predict the data. As this is a high mass flux-low quality run, it is felt that heat is being conducted out the top of the transient section due to a wetted interface at the discharge to the transient test section (again refer to Appendix A for more complete description).

The concluding check of the post critical heat transfer correlation for nitrogen comes with the comparison of the steady state temperature-length tube data obtained by Forslund and Hynek. If the correlation predicts this data equally well with $\delta_{\text{film}} = 4 \times 10^{-4}$ ft then the uncertainties involved in basing the correlation on data suspected of heat losses is significantly reduced as the problems of insulating a tube are much less involved than that involved with the transient section used in this investigation. Figures 35-39 present the full range of steady state
NITROGEN

\[ P = 20 \text{ psia} \]
\[ G = 20,000 \text{ lbm/hr-ft}^2 \]
\[ X_{DO} = 2\% \]

Prediction:

\[ \delta_{film} = 4 \times 10^{-4} \text{ ft} \]
\[ \kappa = 1. \]
\[ \kappa = 0.4 \]

\[ X_E = 20\% \]
\[ X_E = 80\% \]

Hynek Data Fig 13 Ref 8

**FIGURE 35** COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH UNIFORMLY HEATED TUBE NITROGEN DATA -  
\[ G = 20,000 \text{ lbm/hr-ft}^2 \]
NITROGEN

\[ P = 20 \text{ psia} \]
\[ G = 40,700 \text{ lbm/hr-ft}^2 \]
\[ X_{D0} = 2\% \]

Prediction:

\[ \delta_{\text{film}} = 4 \times 10^{-4} \text{ft} \]

\[ \kappa = 1. \]

\[ \kappa = 0.45 \]

\[ X_E = 80\% \]

\[ X_E = 30\% \]

\[ D_T = 0.4 \text{ inches} \]

Hynek Data: Fig 14 Ref 8

FIGURE 36 COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH UNIFORMLY HEATED TUBE NITROGEN DATA -
\[ G = 41,000 \text{ lbm/hr-ft}^2 \]
NITROGEN

$P = 25$ psia

$G = 70,000$ lbm/hr-ft$^2$

$X_{DO} = 2\%$

$D_T = .323$ inches

\[ \delta_{film} = 4 \times 10^{-4} \text{ ft} \]

$\kappa = 0.58$

$\kappa = 1.0$

\[ (T_{wall} - T_{sat}) \text{ } ^\circ F \]

\[ X_E = 10\% \]

\[ X_E = 80\% \]

\[ - X_E = 10\% \]

\[ - X_E = 80\% \]

Forslund
Data: Fig 6
Ref 5

FIGURE 37 COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH UNIFORMLY HEATED TUBE NITROGEN DATA - $G = 70,000$ LB/HR-FT$^2$
NITROGEN

P = 25 psia
G = 130,000 lbm/hr-ft²
X_D0 = 2%
D_T = .228 inches

X_E = 75%
X_E = 15%
X_E = 5%

Prediction:
κ = .62
δ_film = 4 x 10⁻⁴ ft

Forslund
Data: Fig 10
Ref 5

FIGURE 38 COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION
WITH UNIFORMLY HEATED TUBE NITROGEN DATA -
G = 130,000 LB/HR-FT²
NITROGEN

P = 25 psia
G = 190,000 lbm/hr-ft²
X_{DO} = 2%
D_T = .228 inches

HEAT FLUX (BTU/hr-ft²)

\( X_E = 80\% \)

\( X_E = 10\% \)

\( \square - X_E = 10\% \)

\( \bigcirc - X_E = 80\% \)

Prediction:

\[ \kappa = .71 \]

\[ \delta_{film} = 4 \times 10^{-4} \text{ ft} \]

Forslund

Data: Fig 11

Ref 5

\((T_{wall} - T_{sat})^\circ F\)

FIGURE 39 COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH UNIFORMLY HEATED TUBE NITROGEN DATA -

G = 190,000 LB/HR-FT²
nitrogen data available in the literature. These figures encompass a G range of 20,000 - 190,000 lbm/hr-ft$^2$ and an $X_E$ range of 5-80%. Again $\delta_{film} = 4 \times 10^{-4}$ ft provides excellent correlation of the data. The accuracy of the prediction scheme decreases somewhat for the 20,000 and 40,000 lbm/hr-ft$^2$ cases. This is understandable considering that a significant portion of the data for these G's is probably in a froth flow regime. It should be noted that this is the same data from which a value of $\delta_{film} = 1 \times 10^{-4}$ was obtained for the differential two step model discussed in Chapter 3.

4.4.3 Discussion of Post Critical Heat Transfer Correlation for Nitrogen

From the two previous sections the final unknowns are added to the post critical heat transfer correlation for nitrogen given by equation (4.15). $\kappa$ is now evaluated from Figure 22 on Table 3. $\delta_{film}$ is set equal to $4 \times 10^{-4}$ feet. The purpose of this section is to discuss the application of the correlation; its properties and limitations.
This prediction scheme appears quite formidable as indeed the physics of post critical heat transfer dictates. However the application of these equations to a particular problem is considerably simpler than the differential two step model discussed in Chapter 3. The equations are completely determinant given $G$, $X_{D0}$ and $X_E$ for $q/A$ in terms of $(T_w - T_{Sat})$. These equations require iterations in a maximum of two places in the calculation procedure. For every application the dryout slip, $S_{D0}$, has to be iterated using equation (3.8) and (3.9). A second iteration is required between $u_w$ and $k_{v,f}$ and the wall temperature when heat flux is the independent variable. If wall temperature is the independent variable then this iteration is not needed.

From the data comparison presented in section (4.4.2) the ability of the correlation to predict both the quality and mass flux dependency on the post critical heat flux is evident. An increase in $G$ for a given set of system variables is readily seen to increase the post critical heat flux. Comparing Figures 37 and 39 demonstrate this. The effect of quality as represented by $X_E$ on the post critical heat flux is rather complicated and its trends
have not been until now wholly correlated. The particular trend that \( X_E \) will have on \( q/A \) in post dryout is basically dependent on \( X_{DO} \) and \( G \). Three possible trends of \( q/A \) vs \( X_E \) can occur. The first case is that \( q/A \) continually decreases after dryout as \( X_E \) increases. This occurs at a high value of \( X_{DO} \) and a corresponding moderate to low \( G \). Secondly \( q/A \) vs \( X_E \) can increase to a point, then subsequently decrease as \( X_E \) increases. This generally occurs at low values of \( X_{DO} \) for any \( G \) or at moderate to high values of \( X_{DO} \) for high values of \( G \). The final possibility is that \( q/A \) initially decreases, increases and finally decreases again as \( X_E \) increases. These three trends can be verified by returning to Figures 13-15 containing temperature length plots for various \( G \) and \( X_{DO} \) combinations. These trends are only partially observed in Figures 34-39 as the \( X_E \) variable is not high enough to demonstrate the eventual decrease in heat flux. The post critical heat transfer correlation presented here is capable of producing these trends because it is the sum of two terms which have opposite trends of \( q/A \) vs \( X_E \). \( q/A_{w,\delta} \) decreases with increasing \( X_E \). \( q/A_{w,v} \) increases at a decreasing rate due to vapor superheat. It is possible for this term to go negative if the decrease of the heat transfer due to vapor superheat is greater than the increase in \( h_{w,v} \) due to higher vapor velocity.
The sum of the two heat transfer mechanisms can produce the three trends discussed depending on the relative rates of change with $X_E$. The correlation automatically goes to the all vapor asymptote as $A$ goes to 1.

The final system variable affecting the post critical heat flux is the wall superheat defined as $(T_w - T_{sat})$. The heat flux from the wall, $q/A$, increases as $(T_w - T_{sat})$ increases. The particular shape that this curve takes depends on the quantity $(X_E - X_{DO})$. For $(X_E - X_{DO}) = 0$, the $q/A$ vs $(T_w - T_{sat})$ curve for any $G$ is nearly a straight line with a slope slightly higher than one as $(X_E - X_{DO})$ increases, $q/A$ vs $(T_w - T_{sat})$ deviates from a straight line and looks more like a curve that approaches a straight line. A straight line drawn through this curve would have a slope somewhat higher than one. Figure 38 demonstrates this effect very well.

A very useful property of the post critical heat transfer prediction scheme as presented in this thesis is its ability to give the upper and lower bound to $q/A$ vs $(T_w - T_{sat})$ in post dryout for any given system conditions. The upper bound is obtained immediately upon substituting $\kappa = 1$ into equation (4.15). The lower bound can be obtained by selecting a value of $\kappa$ from Figure 22 that is below the data for any given $G$ and $(1-X_{DO})$. Figure 40 demonstrates the ability of $\kappa = 1$ and $\kappa = 15$ to bound
FIGURE 40 COMPARISON OF BOUNDED SOLUTION WITH REPRESENTATIVE TRANSIENT DATA
the transient data given in Runs 95, 97, 98 and 100. One can see that as \((X_E - X_{DO})\) increases, the two bounds grow further apart. At \((X_E - X_{DO}) = 0\) the upper and lower bounds are the same.

It is interesting now to compare the results of this correlating procedure with the Groeneveld post critical heat transfer correlation for tubes as given in Chapter 1 [equation (1.1)]. The constants for the heat flux dependent case were used. This comparison is presented in Figure 41 for the case of \(G = 250,000 \text{ lbm/hr-ft}^2\), \(X_{DO} = 20\%\) and \(X_E = 40, 80\%\). From Figure 20 a value of \(\kappa = .62\) was selected given this set of conditions. For \(X_E = 40\%\) there appears to be very little discrepancy between the two correlations at least at high wall superheats. At low wall superheats the vapor superheat begins to have an affect on the nitrogen correlation whereas no affect is given by the Groeneveld correlation. The Groeneveld correlation shows very little quality effect on the heat flux which is understandable considering that the correlation was based on water data which has a \(\rho_v/\rho_{\ell}\) ratio almost an order of magnitude higher than nitrogen. This comparison vividly illustrates the inability of this type of post critical heat transfer correlation to predict accurately any data outside the range of data correlated. Just knowing \(X_E\) and \(G\) is not sufficient to quantify every conceivable post critical situation. The
NITROGEN

P = 20 psia
G = 250,000 lbm/hr-ft²
X_{DO} = 20%
D_T = .4 inches

\[ X_E = 80\% \]

\[ X_E = 40\% \]

Groeneveld Correlation
Equation (1.1)

Prediction:
\[ \kappa = .62 \]
\[ \delta_{film} = 4 \times 10^{-4} \text{ ft} \]

\( (T_{wall} - T_{sat}) \text{ °F} \)

FIGURE 41 COMPARISON OF THE GROENEVELD POST CRITICAL HEAT TRANSFER CORRELATION WITH THE PREDICTION SCHEME OF THIS REPORT
post critical heat transfer prediction scheme presented here for nitrogen does have the necessary variables that will allow it to be extrapolated outside the data on which it was based and as such this correlation should be able to predict data for different fluids. There is a possibility that $\delta_{\text{film}}$ will have to be modified but this can be taken care of by evaluating the constant with available data for the particular fluid in question. The next section presents a preliminary comparison of the nitrogen correlation with Water and Freon 12 data in the literature.
4.5 Comparison of Nitrogen Prediction Scheme with Other Fluids

Since the $\kappa$ vs $G$ and $(1 - X_{DO})$ relation and the post dryout $\alpha$ relation for Water and Freon 12 were determined simultaneously with the derivation of the post critical heat transfer correlation for nitrogen, it is a simple matter to apply these quantities to the general prediction scheme for nitrogen as given in equation (4.15) and ascertain its validity in predicting water and Freon 12 post critical heat transfer data. In the following comparison process the value of $\kappa$ was obtained from Figure 22 instead of using the appropriate correlation equation as specified in Section (4.4.1).

4.5.1 Water Comparison

Figures 42, 43, and 44 present the resulting comparison of the post critical heat transfer correlation with water data at 1000 psia for three mass fluxes $G =$ 485,000, 740,000 and 1,000,000 lbm/hr-ft$^2$ respectively. This data was obtained by Bennett$^6$ in a uniformly heated tube. The Heineman superheated steam correlation was used instead of the modified McAdams equation for the wall to vapor heat transfer coefficient.

The prediction scheme does suprisingly well considering the data is a cross plot of a number of runs having different values of $X_{DO}$. Figure 42 indicates an under-
Water

P = 1000 psia
G = 485,000 lbm/hr-ft²
X_{DO} = 75%
D_T = .497 inches

\[ \Delta T = \frac{Q}{U} \]

**Prediction:**

- \( \delta_{film} = 4 \times 10^{-4} \text{ ft} \)
- \( \kappa = .34 \)
- \( \kappa = 1. \)

**Bennett Data:**

- Runs: 5323, 5330, 5332, 5334, 5336
- Ref 6

**Figure 42** Comparison of Post Critical Heat Transfer Correlation with Uniformly Heat Tube Water Data - \( G = 485,000 \text{ lbm/hr-ft}² \)
WATER

\( P = 1000 \ \text{psia} \)
\( G = 740,000 \ \text{lbm/hr-ft}^2 \)
\( X_{DO} = 64\% \)
\( D_T = 0.497 \ \text{inches} \)

Prediction:

\[ \kappa = 0.4 \]
\[ \delta_{film} = 4 \times 10^{-4} \ \text{ft} \]

**Figure 43** Comparison of Post Critical Heat Transfer Correlation with Uniformly Heated Tube Water Data - G = 740,000 lbm/hr-ft²
WATER

$p = 1000$ psia

$G = 1,000,000$ lbm/hr-ft$^2$

$X_{DO} = 55\%$

$D_T = .497$ inches

Prediction:

$\kappa = .45$

$\delta_{film} = 4 \times 10^{-4}$ ft

$\circ - X_E = 60\%$

$\square - X_E = 70\%$

Bennett
Data: Runs 5242-5251
Ref 6

FIGURE 44  COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH UNIFORMLY HEATED TUBE WATER DATA -
$G = 1,000,000$ LBM/HR-FT$^2$
prediction of the data. The correct trend of the quality is observed in the prediction though. It appears that a \( \delta_{\text{film}} \) somewhat smaller than \( 4 \times 10^{-4} \) feet is needed. However, by reason of the high void decreasing \( \delta_{\text{film}} \) will not appreciably affect the resulting prediction. Also plotted on Figure 42 is the prediction for the 85% quality case using \( \kappa = 1 \). This does a reasonable job in predicting the data. The comparisons for the two higher mass fluxes are quite good with less than 15% deviation between the prediction and data.

4.5.2 Freon 12 Comparison

The Freon 12 data for this comparison is again taken from Groeneveld\(^7\). It has been replotted in a \( q/A \) vs \( (T_w - T_{\text{sat}}) \) coordinates. Figures 45 and 46 present the resulting comparisons for \( G = 250,000 \) and \( 1.5 \times 10^6 \text{lbm/hr-ft}^2 \). \( \delta_{\text{film}} = 4 \times 10^{-4} \) ft appears to correlate this data fairly well. The McAdams equation was used for the wall to vapor heat transfer term. Figure 45 indicates that some superheating of the vapor has occurred as seen by the comparison of the \( \kappa = 1 \) prediction with that of \( \kappa = .4 \). \( X_E \) has not significantly increased above \( X_{D0} \) in Figure 46 with the result that any value of \( \kappa \) will give the same prediction for this case.
FREON 12
P = 155 psia
G = 250,000 lbm/hr-ft²
X_{DO} = 60%
D_T = .307 inches

Prediction:
\delta_{film} = 4 \times 10^{-4} \text{ft}
\kappa = 1.
\kappa = .4

\[ T_{wall} - T_{sat} \] °F

\[ \text{HEAT FLUX (KBTU/hr-ft}² \]

\[ \text{Groeneveld Data} \]
\[ \text{Reference 7} \]

FIGURE 45 COMPARISON OF POST-CRITICAL HEAT TRANSFER CORRELATION WITH UNIFORMLY HEATED TUBE FREON 12 DATA - G = 250,000 LBM/HR-FT²
FREON 12

\[ P = 155 \text{ psia} \]
\[ G = 1,500,000 \text{ lbm/hr-ft}^2 \]
\[ X_{DO} = (25-32\%) \]
\[ D_{T} = .307 \text{ inches} \]

Prediction:
\[ \kappa = .7 \]
\[ \delta_{film} = 4 \times 10^{-4} \text{ ft} \]
\[ X_{E} = 35\% \]
\[ X_{DO} = 30\% \]

\[ \triangle - X_{E} = (32-36\%) \]

GROENEVELD

DATA

Runs
8612
8613
8614
9614
Ref 7

\[ (T_{wall} - T_{sat}) \text{ \degree F} \]

**FIGURE 46** COMPARISON OF POST CRITICAL HEAT TRANSFER CORRELATION WITH UNIFORMLY HEATED TUBE FREON 12 DATA -

\[ G = 1,500,000 \text{ lbm/hr-ft}^2 \]
V. SUMMARY AND CONCLUSIONS

5.1 Summary

1. An experimental investigation of post critical heat transfer in forced vertical flow for a tube geometry was undertaken. A novel transient experimental technique was devised and implemented. The extreme versatility of this technique was demonstrated by its ability to generate, in one operation, post critical heat transfer data at one particular G and $X_E$ combination for various dryout lengths up to eight feet. The flexibility of the transient experiment was further increased with the ability to interchange transient test pieces for determining material, roughening and oxide effects on post critical heat transfer. Forced convection dry wall film boiling data was obtained using this experimental technique for the following range of parameters.

For a 0.4 inch inside diameter tube:

$G$ variation: $30,000 - 220,000$ lbm/hr-ft$^2$

$X_E$ variation: $0 - 90\%$

$L_{DO}$ variation: $2 - 96$ inches

Test material: Copper, Inconel 600, Aluminum 100

Surface Conditions: Smooth (10 - 50 microinches)

Roughened (approx. 400 microinches)
Oxidized (film approx. $10^{-5}$ inch thick)

$q/A$ variation: 1,000 - 25,000 btu/hr - ft$^2$

$(T_{wall} - T_{sat})$ variation: 50 - 550°F

This apparatus was capable of investigating dry wall film boiling heat transfer at such close proximity to the dryout point without axial conduction effects because of the careful insulation procedure used to isolate the transient test section from the preheater.

2. An extensive comparison of the Hynek and Groeneveld dispersed flow film boiling models with data in the post critical heat transfer regime was carried out. The original Hynek computer code was modified to include the Groeneveld droplet breakup mechanism. This code was compared with a large range of post critical heat transfer data for nitrogen, water and Freon 12.

3. A generalized post critical heat flux prediction scheme was developed upon application of the assumption that the $X_A$ vs $X_E$ curve can be approximated by the linear function $X_A - X_{D0} = \kappa (X_E - X_{D0})$ to a modified form of the differential two-step dispersed flow model used by Hynek. Considering, therefore, that heat is transferred in post dryout from the wall to superheated vapor and from the wall to drops via dry collisions, expressions for $T_v$ and $\alpha$ were
obtained in terms of \( \kappa \). \( T_v \) was obtained from direct application of the above assumption to the post dryout energy balance. The void fraction was obtained using a post dryout slip correlation based on computer generated slip ratios obtained from the solution of the momentum, energy and critical weber number equations. \( T_v \) and \( \alpha \) were subsequently substituted into the McAdams vapor heat transfer and Groeneveld droplet heat transfer to provide the total solution in terms of \( \kappa \). \( \kappa \) was evaluated using a visual best fit procedure to \( X_A \) vs \( X_E \) curves generated from modified Hynek computer code of the differential two-step dispersed flux mode. The final procedure in the development of the post critical heat transfer prediction scheme consisted of correlating \( \kappa \) vs \( G \) and \( X_{D0} \) using the following nondimensional parameter

\[
G \sqrt{\frac{D_T}{\rho_v \sigma}} (1 - X_{D0})^5
\]

The resulting post critical heat flux correlation given in Equation (4.15) was compared with the transient data obtained in this study as well as nitrogen, water and Freon 12 data found in the literature.

5.2 Conclusions

From the above investigation the following conclusions were made.

1. A transient boiling experiment will give the correct
representation of the post critical heat transfer as evidenced by the positive comparison between the transient data in this study and steady state data found in the literature.

2. No consistent trend for the effect of material on drywall film boiling heat transfer was observed.

3. Increasing the roughness of the heater surface increases its heat transfer capability. This results from increasing the wall-to-vapor heat transfer coefficient as demonstrated by the Colburn analogy. This effect is more pronounced as the vapor Reynolds number increases.

4. The addition of oxide to the heating surface increases the heat transfer characteristics of the surface. The reason for this effect is linked with the evidence that actual liquid contact can occur in dry wall film boiling even if the surface temperature is above the maximum liquid temperature. The increased wettability and porosity of the oxide allows the attached liquid to spread and adhere to the surface for a longer period of time than if the surface were clean. The combination of increased area of liquid contact, increase in time of contact and vigorous evaporation at the liquid-solid interface has the effect of increasing the heat transfer over and above any thermal resistance that the oxide might have. It is noted that the
augmentation of the heat transfer with oxide is more pronounced at low quality, high mass flux combinations where the liquid fraction is higher.

5. No major differences were noted in the ability of Hynek's and Groeneveld's models to predict data in post dryout conditions. The computer code, FILMBOIL, using a value of $\delta_{\text{film}} = 1. \times 10^{-4}$ feet for the correlating constant in Groeneveld's wall-to-droplet heat transfer coefficient predicts nitrogen, water and Freon 12 post dryout temperatures profiles with an accuracy of 10 - 20%.

6. The ability of the post critical heat transfer prediction scheme to correlate nitrogen data has been demonstrated. The effects of $G$, $X_E$, $X_{D0}$ and $T_{\text{sat}}$ on the post critical heat flux are properly accounted for through the implicit inclusion of nonequilibrium effects resulting from vapor superheating in post dryout. The only restrictions to the use of the post critical heat transfer correlation as presented in this thesis is that $G$ be greater than that required for liquid carryover.

5.3 Recommendations

1. It is recommended that a quantitative study of the effects of roughness and oxide on the post dryout heat transfer initiated. Included in this study should be a detailed analysis of liquid-solid contact at surface
temperatures above the maximum liquid temperature to more clearly understand the process.

2. The generalized post critical heat transfer correlation presented in this research can be further improved by reexamining the linear approximation to the \( X_A \) vs \( X_E \) curve with the possibility of using instead a polynomial curve fitting procedure. It was noted that the approximate vapor temperature was quite sensitive to how well the approximation resembled the actual curve. Luckily, this sensitivity was not wholly transferred to the predicted heat flux.
APPENDIX A

Estimation of Heat Losses for Transient Test Section
(Design 1 and 2)

From Fig 5 heat interactions between the test specimen and its environment can occur via three means: first, conduction radially through the cup, across the air gap to the exposed area of the test piece; second, conduction at contact points between the specimen and the cover assembly and third, at the contact point of the specimen and textolite spacer separating the specimen from the brass base. The following presents idealized models to estimate the heat flow resistance for the three cases.

A.1 Heat Interactions common to transient Design 1 and 2

radial heat interaction:

Figure A1 presents the model used to calculate this interaction. The model assumes heat can be transferred to the copper wall from outside through the insulation and through direct contact with the brass base which has a high heat capacity. The heat that reaches the copper wall is then transferred across the air gap to the test specimen. The gap is broken into two resistances: a conduction resistance and a convection resistance to account for the natural circulation.
Figure A1 Pictorial Representation and Resistance Network Model of Radial Heat Interaction with Test Piece and Environment
Substituting in the values for each resistance gives

\[
a) \quad \frac{1}{h_1A_1} \approx 5 \frac{0^\circ F - Hr}{\text{Btu}}
\]

\[
b) \quad \frac{\Delta X}{k_{cu}A_{cu}} \approx 0.09 \frac{0^\circ F - Hr}{\text{Btu}}
\]

\[
c) \quad \frac{\ln \left( \frac{D_4}{D_3} \right)}{2\pi L k_1} = 22 \frac{0^\circ F - Hr}{\text{Btu}} \quad (A.1)
\]

\[
d) \quad \frac{\ln \left( \frac{D_3}{D_2} \right)}{2\pi L k_a} = 41 \frac{0^\circ F - Hr}{\text{Btu}}
\]

\[
e) \quad \frac{1}{h_2A_2} \approx 20 \frac{0^\circ F - Hr}{\text{Btu}}
\]

As the resistance from $T_a$ to $T_{cu}$ is so much larger than the resistance from $T_A$ to $T_{cu}$ we can neglect it. Furthermore we can assume that $T_A = T_{cu}$. The radial heat transfer becomes

\[
q_{\text{Rad}} = \frac{(T_A - T_B)}{61} \text{ [Btu/hr]} \quad (A.2)
\]

Cover-Specimen Interaction:

The cover-specimen interaction is very difficult to estimate as there are a number of forces at play which could result in either heat gain or loss depending on the dominant force. The problem centers on whether the micarta-cap assembly quenches before the test specimen after the steam is shut off. This translates to a heat gain if the cover
assembly remains higher than the specimens or a heat loss if the cover assembly is always lower in temperature than the test piece. There is also a distinct possibility of continuous liquid contact of the micarta spacing even during the heat up period because the low conductivity causes the micarta area exposed to the nitrogen flow to act like a cold spot. This would cause a heat loss even during the initial portion of the transient where normally one would expect no heat losses or gains due to the uniformity of all temperatures. Figure A2 presents a conceptual model of the entire cover-specimen interaction which gives a clearer picture of the processes described.

Figure A2a indicates that the heat loss or gain is dependant on $T_m$, some average temperature inside the micarta spacing. This temperature is influenced by the surface temperature of the micarta exposed to the flowing nitrogen indicated as $T_{coldspot}$ as well as the average temperature of the brass cap, $T_A$. In order to estimate the behavior of this system it is assumed that only two mutually exclusive processes can occur. Either $T_{coldspot} = T_m$ resulting in the conduction path being brass cap to micarta to rubber O-ring to specimen or $T_{coldspot} = T_{sat}$ resulting in conduction path being from $T_{coldspot}$ through short length of micarta, through rubber O-ring to specimen. The first of these two cases is given in Figure A2b. The resistances
FIGURE A2  BRASS CAP - TEST SPECIMEN HEAT TRANSFER INTERACTION FOR DESIGN 1 and 2
are simply

\[ \frac{\Delta X_1}{k_m A_m} = \frac{\Delta X_1}{k_m \pi (r_2^2 - r_2^2)} = 18 \ \text{hr}^{-0_F} \text{Btu} \]  
(A.3)

\[ \frac{\Delta X_2}{k_r A_r} = \frac{\Delta X_2}{k_r \pi D_3 t} = 24 \ \text{hr}^{-0_F} \text{Btu} \]

This gives the heat gain for Case 1 as

\[ q_{cup_1} = \frac{(T_A - T_B)}{42} \ \text{[Btu/hr]} \]  
(A.4)

Figure A2c gives the circuit diagram for the second case where \( T_{coldspot} \) is assumed equal to \( T_{sat} \).

The resistances are

\[ \frac{\ln(D_2/D_1)}{2\pi k_m L} = 9 \ \text{hr}^{-0_F} \text{Btu} \]

\[ \frac{\Delta X_2}{k_r A_r} = 24 \ \text{hr}^{-0_F} \text{Btu} \] from (A.3.a)

This gives the heat loss for case 2 as

\[ q_{cup_2} = - \frac{(T_B - T_{sat})}{57} \]  
(A.5)
A.2 Heat interactions of test piece to base

A.2.1 Design 1

Figure A3a indicates how we tried to cut the heat losses down through the fin protruding from the base of the test specimen with a micarta sleeve.

An upper bound of the heat loss can be found by assuming that the two surfaces of the micarta contacting the inconel is at the temperature of the brass base, \( T_A \), and that the two surfaces touching the fin are at the test piece temperature. Figure A3b indicates the geometry.

The upper bound on the heat transfer from the base is approximately

\[
q_{\text{upper}} = \left( A_1 + A_2 \right) \frac{k_m (T_A - T_B)}{\delta} \quad (A.7)
\]

where \( A_1 = \pi D_1 L \)

\( A_2 = \pi (r_2^2 - r_1^2) \)

Substituting in the appropriate dimensions gives the upper limit as

\[
q_{\text{upper}} = .3 (T_A - T_B) \quad (A.8)
\]

Figure A3c gives an approximation for the calculation of the lower bound on the heat transfer up through the base. The lower bound becomes
**FIGURE A3** INTERACTION OF TEST PIECE WITH BRASS BASE FOR TRANSIENT DESIGN 1
\[
q_{\text{lower}} = \frac{(T_A - T_B)}{2 \pi L k_m} \ln \left(\frac{r_2/r_1}{1} \right)
\]  
(A.9)

Substituting in the appropriate values gives

\[
q_{\text{lower}} = 0.05 (T_A - T_B)
\]
(A.10)

A.2.2 Design 2

The only difficulty involved in estimating the heat interaction between the base and test piece for this design is the section where the copper sleeve, textolite and test piece come together as shown in Figure A4. Here it is assumed that the copper sleeve is at the temperature of the brass base and the conduction length separating it from the test piece is 1/3 of that used for each of the other two resistances. These resistances are

a) \[
\frac{\Delta x_1}{k_m A_1} = \frac{0.25 \pi D_1 t}{2 \pi D_1 t} = 95 \quad \frac{^\circ \text{F-Hr}}{\text{Btu}}
\]  
(A.11)

b) \[
\frac{\Delta x_1}{k_m A_2} = \frac{0.25 \pi D_2 t}{2 \pi D_2 t} = 110 \quad \frac{^\circ \text{F-Hr}}{\text{Btu}}
\]

c) \[
\frac{\Delta x_2}{k_m A_3} = \frac{0.25/3 \pi D_3 t}{2 \pi D_3 t} = 70 \quad \frac{^\circ \text{F-Hr}}{\text{Btu}}
\]

The heat gain up through the base becomes
FIGURE A4 INTERACTION OF TEST PIECE WITH BRASS BASE FOR TRANSIENT DESIGN 2
\[ q_{\text{base}} = (T_A - T_B) \left[ \frac{1}{95} + \frac{1}{110} + \frac{1}{70} \right] \]

\[ q_{\text{base}} = 0.023 \, (T_A - T_B) \]

(A.12)

A.3 Summary of heat interactions

A.3.1 Design 1

The results are separated into two categories. The first category includes the assumption that the brass cover and base quench down at the same rate so that their temperatures are equal and that the liquid does not attach itself to the micarta. The summation of heat gains calculated by (A.2), (A.4) and 1.5 times the heat transfer given by (A.10) results in the total heat gain as a function of the temperature difference between the test piece and brass base for the case of no liquid attachment to be

\[ (q_{\text{gain 1}})_{\text{N.L.A.}} = 0.106 \, (T_{\text{brass base}} - T_{\text{specimen}}) \]

(A.13)

Equation (A.13) is probably valid for all data runs except those of high mass flux and moderate to low quality.

The second category which takes care of the high mass flux and moderate to low quality runs contains the assumption
that the complete cover assembly is at $T_{\text{sat}}$. The summation of heat gains calculated by (A.2) and 1.5 times (A.10) with (A.6) results in equation (A.14) for the case of liquid attachment.

\[
(q_{\text{gain 1}})_{\text{L.A.}} = (0.09) (T_{\text{brass base}} - T_{\text{specimen}}) - 0.04(T_{\text{specimen}} - T_{\text{sat}}) \quad (A.14)
\]

These results indicate that the data from Design 1 transient section will be accurate at the initial portions of each transient where $(T_{\text{brass base}} - T_{\text{specimen}})$ is low if no rewetting of the micarta takes place. In most of the runs of this type the specimen quenched faster than the brass base with the difference never exceeding 200°F. The maximum initial error for the case where rewetting of the micarta is assumed is 20 Btu/hr obtained by substituting $T_{\text{specimen}} = T_{\text{brass base}} = 212^0\text{F}$ and $T_{\text{sat}} = -320^0\text{F}$ into equation (A.14). A general value for the film boiling heat transfer for this case is 200 Btu/hr. The error is 10%.

A.3.2 Design 2

This design cut the heat gains in through the test specimen by at least a factor of 3. The respective equations for the two cases described in Section A.3.1 for design 2 are

\[
(q_{\text{gain 2}})_{\text{N.L.A.}} = 0.063 (T_{\text{brass base}} - T_{\text{specimen}}) \quad (A.15)
\]
\[(q_{\text{gain}}^{2})_{\text{L.A.}} = 0.039 \left( T_{\text{brass base}} - T_{\text{specimen}} \right) \]
\[-0.04 \left( T_{\text{specimen}} - T_{\text{sat}} \right) \]

We are not able to quantify the heat losses very well from these equations as the brass base temperatures was not recorded during the data runs. The closest indication we have of the behavior of the brass base is from the thermocouples placed on the preheater at the exit as shown in Figure 5. The thermocouple was influenced to a much greater extent by the tube than by the brass base. It did show, however, a gradual decrease after steam was shut off.

A maximum value of the heat gains for design 1 and 2 can be estimated by substituting a value of 200 °F for \( T_{\text{brass base}} - T_{\text{specimen}} \) into equations (A.13) and (A.15). This gives a maximum heat gain for design 1 of 20 Btu/hr and a maximum heat gain for design 2 of 12 Btu/hr. These values of heat gain are present near the end of each transient quench test. The film boiling heat fluxes in this range (\( \Delta T_{\text{sat}} \) on the order of 50 - 100 °F) is 1000 Btu/hr-Ft\(^2\). The respective errors for the two designs are; therefore, 200 and 100%. These errors drop off sharply for higher wall superheat data where \( T_{\text{brass base}} - T_{\text{specimen}} \) is small. It is concluded that all transient data with wall superheats above 200 °F will be free from heat gains.
APPENDIX B

Determination of System Variables and Experimental Errors

B.1 System Variables

The important system variables calculated in the experimental program are mass flux, equilibrium quality to the transient section and the saturation temperature of liquid nitrogen at the transient section. The uncertainties in these are related to the independent quantities which go into their evaluation. This section presents the equation used in the determination of the quantities described above and tabulates the estimated errors for each. This is achieved using the principle of superposition of errors (Topping\textsuperscript{25}).

**Mass Flux:**

The mass flux is obtained by measuring the flow rate of the evaporated nitrogen via two rotometers in parallel. The rotometers were calibrated with water over the entire scale of each meter. The following equation resulted.

\[
G = 1.143 \times 10^3 \left\{ - 1.0806 + .07796 \left( R_{\text{scale 1}} \right) \right\} + \\
[ .1116 + .199 \left( R_{\text{scale 2}} \right) ] \cdot \left[ (500 - \rho_n) \rho_n \right]^{1/2} \quad (B.1)
\]

where the scale range for each rotometer is
Rotometer | Range   | Increment
---|---------|---------
1    | 0-250   | 2       
2    | 0-100   | 1       

The density of gaseous nitrogen, $\rho_{n_2}$, is obtained from the perfect gas relationship.

$$\rho_{n_2} = 2.61 \frac{P_{n_2}}{T_{n_2}} \quad (B.2)$$

The estimated uncertainties for the independent quantities for $G$ are given in Table B.1

<table>
<thead>
<tr>
<th>Variable</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
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<td>$P_{n_2}$</td>
<td>6%</td>
</tr>
<tr>
<td>$T_{n_2}$</td>
<td>1%</td>
</tr>
<tr>
<td>Scale reading 1</td>
<td>2%</td>
</tr>
<tr>
<td>Scale reading 2</td>
<td>2%</td>
</tr>
</tbody>
</table>

Table B.1

From Topping (24) the error in $G$ due to the errors in its independent variables can be determined from

$$\delta G = \left| \frac{\partial G}{\partial P_{n_2}} \delta P_{n_2} \right| + \left| \frac{\partial G}{\partial T_{n_2}} \delta T_{n_2} \right| + \cdots \quad \text{etc} \quad (B.3)$$

taking representative values for $P_{n_2}$, $T_{n_2}$, Scale reading 1 and 2 and performing the operations indicated by (B.3) gives
\[
\frac{\delta G}{G} = 0.09 \quad \text{(B.4)}
\]

**Preheater Exit Quality:**

Preheater exit quality is calculated thermodynamically knowing the inlet subcool, preheater heat and unheated lengths, insulation losses, preheater power, and mass flux. The preheater exit quality is assumed to be the quality of the transient section. This quality is determined from (B.5).

\[
X = \frac{120}{84} G \left[ (Q_{IN} + DLOSS) (L_{heated} - DLENT) + 2.73 L_{unheated} \right] \quad \text{[power to preheater]}
\]

where

\[
Q_{IN} = \frac{2607.34 \text{ (Volt)(Amp)}}{L_{heated}} \quad \text{[insulation gains]}
\]

\[
DLOSS = \frac{1.23 (75 - T_{wall \ average})}{1.65} \quad \text{[position of zero quality]}
\]

\[
DLENT = 0.5(\Delta T_{s,c}) / 120 (Q_{IN} + DLOSS) \quad \text{[position of zero quality]}
\]

The estimated uncertainties for the independent quantities for X are given in Table B.2

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<th>Uncertainty</th>
</tr>
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<tr>
<td>Amp</td>
<td>1/2%</td>
</tr>
<tr>
<td>(L_{heated})</td>
<td>1/2%</td>
</tr>
</tbody>
</table>
\[ \Delta T_{s.c.} \quad 1/2\% \]

\[ G \quad 9\% \]

**Table B.2**

Using the method of superposition of errors gives

\[ \frac{\delta X}{X} = 0.01 \quad \text{(B.6)} \]

### B.2 Heat transfer data.

All heat transfer data is obtained from the transient section and consists of the surface heat flux vs. surface wall superheat. This is obtained from the temperature - time transient assuming lumped heat capacity. The heat flux and wall superheat assuming no losses are given by (B.7) and (B.8)

\[
\frac{q}{A} = \frac{V}{A} \rho (a + bT(i + n/2)) \left[ \frac{T(i) - T(i + n)}{\Delta t} \right]
\]

\[
(T_w - T_{sat}) = T(i + n/2) - T_{sat} \quad \text{(B.7)}
\]

where \( i = 1, 2, 3 + \)

\( n = \) number of points over which the slope is averaged.

\( a, b = \) coefficients for a linear temperature dependance of
\[ C_p \text{ vs } T(i) \]

\[ \Delta t = \text{time interval} \]

The assumption of no heat losses has been shown in Appendix A to be extremely bad in certain instances which has been discussed. For the data considered to be free of heat losses, thermocouple errors and measurement errors are the major sources of error. Table B.3 gives the thermocouple calibration check for the three thermocouples used in the measurement of transient thermal history.

The therocouples deviate from the standard thermocouples tables only near the saturation temperatures of liquid nitrogen. Thermocouples 1 & 2 are recorded on a Leeds & Northrup Speedomax W strip chart recorder which can be read to .05 millivolts or 5°F at liquid nitrogen temperatures (-300°F) and 2°F at steam temperatures (200°F). The data thermocouple (Φ2) is amplified, recorded on FM tape and subsequently digitized and converted to temperatures on the computer. Although there is no human interpolalian errors, the resolution of the digitizing process is only .01 millivolts. This is 1/5 of the error inherent in the strip chart recorder. From this discussion the errors associated with \( q/A \) and \( (T_w - T_{sat}) \) within the assumption of no heat losses is on the order of 3%. This includes errors of non uniformity of temperature resulting from the high Biot number associated with the Inconel-600 piece.
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<tr>
<th>Temperature Source</th>
<th>Thermocouple (See Fig. 5)</th>
<th>Thermocouple Reading (mV)</th>
<th>Temperature (°F) (Ref 26)</th>
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<td>-5.51</td>
<td>-325.</td>
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<td></td>
<td>2 (data)</td>
<td>-5.51</td>
<td>-325</td>
</tr>
<tr>
<td></td>
<td>3</td>
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<td>-326</td>
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<tr>
<td>ice (32.2 °F)</td>
<td>1</td>
<td>0.0</td>
<td>32.2</td>
</tr>
<tr>
<td></td>
<td>2 (data)</td>
<td>0.0</td>
<td>32.2</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>0.0</td>
<td>32.2</td>
</tr>
<tr>
<td>steam (212 °F)</td>
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<td>4.285</td>
<td>212.2</td>
</tr>
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<td>2 (data)</td>
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<tr>
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<td>3</td>
<td>4.285</td>
<td>212.2</td>
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</tbody>
</table>

Thermocouple Calibration

**TABLE B.3**
APPENDIX C

Tabulation of Transient Data

Explanatory remarks:

G ...............mass flux at entrance to transient section
X ..............X_E at entrance to transient section
TSAT ..........T_sat at entrance to transient section
LDO ............L_D0, length of dryout in preheater
PREHEATER Q/A .heat flux in preheater that generated X
(TW -TSAT).....(T_wall - T_sat) recorded during quench by
thermocouple 2
TIME ..........Time from start of quench
Q/A ..........Corresponding q/A calculated from
Equation (B.7)

Warning: Consider heat flux data at wall superheats of 250 °F
and less as faulty due to heat gains for Runs with
an LDO less than 5 inches. (see Appendix A)
**RUN 73**

<table>
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<tr>
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<tr>
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</tr>
<tr>
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</tr>
<tr>
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</tr>
<tr>
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**RUN 93**

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<td>RUN 95</td>
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<td>G: 214476; LBM/3430°F; TBAI = 314.8 (F)</td>
<td>G: 214329; LBM/3430°F; TBAI = 314.9 (F)</td>
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<td>LO: 11.6 INCHES</td>
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<td>PREHEATER G:A 128.3, 1 BTU/hr/3430°F</td>
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**Preheater G/A:** 2571.8 BTU/HR  
**Preheater Q/A:** 27.86 BTU/HR 80°F  
**Temperature:** 220°F  
**LOD:** 3 INCHES  
**Use:** SMOOTH APP. 10 U INCH

### RUN 119

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**Preheater Q/A:** 27.86 BTU/HR 80°F  
**Temperature:** 220°F  
**LOD:** 3 INCHES  
**Use:** SMOOTH APP. 10 U INCH

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**Specimen Material:** Inconel 688  
**Preheater G/A:** 2571.8 BTU/HR  
**Preheater Q/A:** 27.86 BTU/HR 80°F  
**Temperature:** 220°F  
**LOD:** 3 INCHES  
**Use:** SMOOTH APP. 10 U INCH
### Run 138

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189
APPENDIX D

Computer Listing of FILMBOIL.

INPUT:

QONA....................Heat Flux [btu/hr-ft$^2$]
G.........................Mass Flux [lbm/hr-ft$^2$]
XIN.......................Dryout Quality
FILMZ.....................$\delta_{\text{film}}$ [feet]
DT..........................Tube Diameter [feet]
MUL.......................$\mu_{\lambda}$ [lbm/ft-hr]
GAM.........................$\gamma$ [Coefficient of isentropic expansion]
GAS.......................R [Universal Gas Constant]
SIGMA.....................$\sigma$ [lbm/hr$^2$]
HFG......................$h_{fg}$ [btu/lbm]
TSAT......................$T_{\text{sat}}$ [$^\circ$F]
RHOL......................$\rho_{\lambda}$ [lbm/ft$^3$]
RHOGS...................$(\rho_v)_{\text{sat}}$ [lbm/ft$^3$]
CPGS.....................$(C_{pv})_{\text{sat}}$ [btu/lbm-$^\circ$F]
KGS.......................$(k_v)_{\text{sat}}$ [btu/hr-ft-$^\circ$F]
MUGS.....................$(\mu_v)_{\text{sat}}$ [lbm/hr-ft]
CPG1, CPG2, CPG3, CPG4...Curve Fit coefficients, $C_{pv}$ vs $T_v$
KG1, KG2, KG3............Curve Fit Coefficients, $k_v$ vs $T_v$
MUG1, MUG2, MUG3.........Curve Fit Coefficients, $\mu_v$ vs $T_v$

NODRP = 2             LOM = 5
NOIT = 2               NOREP = 3
NOSTP = 250            NQUIT = 5

ERHOG..................Curve Fit Coefficient, $\rho_v$ vs $T_v$
PROGRAM FILMBOIL

DISPERSED FLOW FILM BOILING MODEL

REAL MUGS,KGS,KGF,MUG,MUWG,MUG1,MUG2,MUG3,KG1,KG2,KG3,KG,MUL
DIMENSION XA(2),XE(2),TV(2),WE(2),DD(2),DELV(2)
DIMENSION VG(2),VL(2),Z(2),TW(2)
DIMENSION PLUM(3,200)
DIMENSION DAV(2,200)
READ(8,1) QO,NA,G,XIN,FILMZ,WEC,MUL,ERHOG
READ(8,1) DT,DZ,GAM,GAS,SIGMA,HFG,TSAT
READ(8,1) RHOL,RHOGS,CPGS,KGS,MUGS
READ(8,10) CPG1,CPG2,CPG3,CPG4
READ(8,10) KG1,KG2,KG3
READ(8,10) MUG1,MUG2,MUG3
READ(8,10) NODRP,NOIT,NOSTP,LOM
READ(8,15) NOREP,NQUIT,ISELT,INIT
1 FORMAT(7E10.2)
5 FORMAT(2I2,1I3,1I2)
10 FORMAT(4E15.0)
15 FORMAT(4I5)
NOSTP=30000
XDO = XIN

** ISEL **
(1)---GROENEVELD VAPOR HEAT TRANSFER COEF.
(2)---BENNETT VAPOR HEAT TRANSFER COEF.

** INIT **
(1)---GROENEVELD INITIALIZATION PROCEDURE
(2)---HYNEK INITIALIZATION PROCEDURE

GRAV=32.16*3600.*3600.
TSAT1=TSAT+459.75
SET UP THE INITIAL VALUES

CALL GSAL,GI,GOA,XDD,RHOL,RHOGS,SIGMA,HFG,HUGS,DT,WEC,VGDO,VLOD,

WRITE(35,F6.6,COMMENTS=""','QMA, FILEZ"

FORMAT(1,23K6,1A15)""GRONEV V D MODEL ""/5X,'G = 'F10.0,
1 = LBM/HR-SQ.FT.,/5X,'Q/A = 'F10.0,6 BTU/HR-SQ.FT.,/5X
2*FILM = "','F9.3,' FEET"/1)

WRITE(5,401)

FORMAT(20X,'CONDITIONS AT DRYOUT','SLIP',6.5X,'VOID',6.5X,'VG'
1,6.5X,'VL',6.5X,'DELV',6.5X,'DROP',6.5X,'CD',6.5X,'TWALL')

SET UP THIRD ITERATION FOR TW, FILM PROPERTIES
SUBROUTINE DOGALIG, QONA, XDO, RHOL, RHOGS, SIGMA, HFG, MUGS, DT, WEC, VGDO,
1 VECC, ODO, CD)

CALCULATION OF CONDITIONS AT DRYOUT USING HYNEK'S METHOD

REAL MUGS

XVE = VDO

GRAY = 2.16*3600.*3600.

CD = 40

I = 0

101 IF = 0

IF = 1

VVE = E0 + G*(1. - XIN)/RHOL

100 VCA = XIN/(1RHOGS*(1. - G)*(1. - XIN)/(RHOL+VVL))

VGL = VVL + SORT + SORT + RHOL*WEC*SIGMA*GRAY + 4.*QONA*XIN*VVL/(HFG*DT*

MUGS)**((0.75*CD*RHOGS**2)))

GO TO 102 (102, 103, 111)

102 IF(VGL-VGL) 106, 110, 107

106 VVE = VVL + L E2

GO TO 100

107 IF = 2

VVL = VVE - L E1

GO TO 100

109 IF(VV6 - VCA) 110, 110, 108

108 VVL = VVL - L E1

GO TO 100

110 CONTINUE

VLDO = VVL

VGGO = 0.5*(VCA + VG8)

DELVO = VGL0 - VLL0

DDD = SIGMA*WEC/(RHOGS*DELVO **2)

RED = RHOGS*DELVO *DDD / MUGS

IF(RED > 2000.) 24, 25, 25

24 CD = (24./RED)**(1.**0.142*RED**0.698)

IF(1.**5) 111, 25, 25

25 RETURN
APPENDIX E

Derivation of Criterion to Determine Droplet Carryover

In order that the dispersed flow model as well as the generalized post critical heat transfer prediction scheme function properly, one must be certain that the evaporating vapor at dryout is sufficient to carry the droplets out the tube. If the vapor velocity is below this value the liquid will collect at some level in the tube with the result that the models presented in this work for predicting post critical heat transfer will be invalid. In order to estimate this quantity, it is assumed that at dryout the liquid is in the form of spherical droplets and that the Weber number is critical. Figure El gives a physical representation of the problem. The momentum equation for a drop at the dryout point is

\[ \frac{dV_d}{dZ} = \frac{3\rho_v (V_g - V_d)^2 \mu}{4\rho_d \delta} - g \left[ \frac{\rho_d - \rho_v}{\rho_d} \right] \]  \hspace{1cm} (E.1)

Assuming the Weber number is critical allows one to obtain an expression for drop diameter, \( \delta \), to be

\[ \delta = \frac{(W_e)_{\text{crit}}}{\rho_v (V_g - V_d)^2} \]  \hspace{1cm} (E.2)

We can look at two cases where the acceleration term in Equation (E.1) is zero; that of a drop standing still just
FIGURE E1  PICTORAL REPRESENTATION OF FLOW REGIME FOR DETERMINING $G_{\text{crit}}$
ready to fall back down the tube represented by the equation

$$V_{\ell} = 0$$  \hspace{1cm} (E.3)

and that of a drop moving at some constant velocity just ready to begin deaccelerating represented by the equation

$$\frac{dV_{\ell}}{dz} = 0$$ \hspace{1cm} (E.4)

It is felt that Equation (E.3) forces the droplet to be too close to falling back down the tube to be a good cut off point for the determination of the criterion. Therefore, substituting Equation (E.4) and (E.2) into Equation (E.1) gives the minimum criterion for the velocity difference necessary for the droplet to remain at a constant velocity to be

$$V_{g} - V_{\ell} = \left[ \frac{4W_{e}}{3C_{D}} \right]^{1/4} \frac{1}{\sqrt{\rho}} \left[ \frac{\sigma (\rho_{\ell} - \rho_{v}) g}{\rho_{v}} \right]^{1/4}$$ \hspace{1cm} (E.5)

The liquid velocity is chosen such that it will not fall below its inlet velocity defined by

$$V_{\ell} \geq \frac{G}{\rho_{\ell}}$$ \hspace{1cm} (E.6)

which is obtained by assuming that the inlet void is zero. Substituting Equation (E.6) into Equation (E.5) gives the critical vapor velocity necessary to allow the liquid to continue moving at a constant velocity of $G/\rho_{\ell}$ as
\[ V_g > \frac{G}{\rho} + \left[ \frac{4(W_e)}{\rho^2} \right]^{1/4} \left[ \frac{\sigma (\rho - \rho_v) g}{3C_D} \right]^{1/4} \]

(E.7)

Equation (E.7) is not very useful as it stands because \( V_g \) is a dependant variable. From the continuity equation given as

\[ V_g = \frac{G X_A}{\rho_v \left[ 1 - \frac{G (1-X_A)}{\rho_v V \rho} \right]} \]  

(E.8)

one can obtain a second equation for \( V_g \) in terms of independent variables from substituting Equation (E.6) into Equation (E.8). This results in

\[ V_g = \frac{G}{\rho_v} \]  

(E.9)

Substituting Equation (E.9) into Equation (E.7) provides us with a criterion on \( G \) alone as

\[ G > \frac{G_{crit}}{\rho_v} = \left[ \frac{\rho \rho_v}{\rho - \rho_v} \right] \left[ \frac{4(W_e)}{\rho^2} \right]^{1/4} \left[ \frac{\sigma (\rho - \rho_v) g}{3C_D} \right]^{1/4} \]

(E.10)

which must be satisfied in order that \( V_g \) after dryout be greater than an equal to \( G/\rho \). Assuming that \( C_D = .45 \) and \( \langle W_e \rangle_{crit} = 7.5 \) this criterion for the three fluids considered in this work becomes
<table>
<thead>
<tr>
<th>Fluid</th>
<th>$G_{crit} \text{(lbm/hr-ft}^2\text{)}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nitrogen</td>
<td>11500</td>
</tr>
<tr>
<td>Water</td>
<td>33700</td>
</tr>
<tr>
<td>Freon 12</td>
<td>39000</td>
</tr>
</tbody>
</table>
APPENDIX F

Fluid Properties

This appendix presents the polynomial curve fit equations for $\mu_v$, $C_{pv}$, $k_v$, and $\rho_v$ as a function of $T_v$. The coefficients to these equations and all other fluid properties used in this thesis are tabulated in Table F.1. The computer notation used in FILMBOIL for denoting the various polynomial curve fit coefficients is retained. The equations are as follows.

$C_{pv}$ vs $T_v$

$$C_{pv} - C_{pg} = CPG1 + CPG2*T_v + CPG3*T_v^2 + CPG4*T_v^3$$

$k_v$ vs $T_v$

$$k_v - k_g = KG1 + KG2*T_v + KG3*T_v^2$$

$\mu_v$ vs $T_v$

$$\mu_v - \mu_g = MUG1 + MUG2*T_v + MUG3*T_v^2$$

$\rho_v$ vs $T_v$

$$\rho_v = \rho_g \left( \frac{T_{sat}}{T_v} \right)^{ERHOG} \text{ where temperatures are absolute}$$

These equations are also used to evaluate the above properties at the wall temperature and film temperature.
<table>
<thead>
<tr>
<th>Fluid Property</th>
<th>Nitrogen</th>
<th>Water</th>
<th>Freon 12</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P$ (psia)</td>
<td>20</td>
<td>1000</td>
<td>155</td>
</tr>
<tr>
<td>$T_{sat}$ ($^\circ$F)</td>
<td>-316.</td>
<td>544.</td>
<td>112.</td>
</tr>
<tr>
<td>$h_{fg}$ (Btu/lbm)</td>
<td>84.13</td>
<td>650.</td>
<td>53.98</td>
</tr>
<tr>
<td>$\sigma$ (lbm/hr$^2$)</td>
<td>2.34x10$^5$</td>
<td>5.17x10$^5$</td>
<td>1.87x10$^5$</td>
</tr>
<tr>
<td>$\mu_g$ (lbm/hr-ft)</td>
<td>.339</td>
<td>.23</td>
<td>.387</td>
</tr>
<tr>
<td>$\rho_g$ (lbm/ft$^3$)</td>
<td>49.6</td>
<td>46.3</td>
<td>77.1</td>
</tr>
<tr>
<td>$\rho_v$ (lbm/ft$^3$)</td>
<td>.381</td>
<td>2.24</td>
<td>3.84</td>
</tr>
<tr>
<td>$k_g$ (Btu/hr-ft-$^\circ$F, 00435)</td>
<td>.0325</td>
<td>.00635</td>
<td></td>
</tr>
<tr>
<td>$\mu_g$ (lbm/hr-ft)</td>
<td>.0145</td>
<td>.0475</td>
<td>.0349</td>
</tr>
<tr>
<td>$C_{pg}$ (Btu/lbm-$^\circ$F)</td>
<td>.253</td>
<td>1.25</td>
<td>.1925</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>1.4</td>
<td>1.366</td>
<td>1.31</td>
</tr>
<tr>
<td>$R$ (Btu/lbm-$^\circ$R)</td>
<td>.0709</td>
<td>1.31</td>
<td>.01642</td>
</tr>
<tr>
<td>CPG1</td>
<td>0.</td>
<td>1.274</td>
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</tr>
<tr>
<td>CPG2</td>
<td>0.</td>
<td>-4.68x10$^{-3}$</td>
<td>-5.96x10$^{-4}$</td>
</tr>
<tr>
<td>CPG3</td>
<td>0.</td>
<td>3.61x10$^{-6}$</td>
<td>1.78x10$^{-6}$</td>
</tr>
<tr>
<td>CPG4</td>
<td>0.</td>
<td>-8.99x10$^{-10}$</td>
<td>-1.68x10$^{-9}$</td>
</tr>
<tr>
<td>MUG1</td>
<td>2.29x10$^{-2}$</td>
<td>-3.6x10$^{-2}$</td>
<td>-5.78x10$^{-3}$</td>
</tr>
<tr>
<td>MUG2</td>
<td>6.5x10$^{-5}$</td>
<td>7.25x10$^{-5}$</td>
<td>5.16x10$^{-5}$</td>
</tr>
<tr>
<td>MUG3</td>
<td>-1.81x10$^{-8}$</td>
<td>-9.42x10$^{-9}$</td>
<td>0.</td>
</tr>
<tr>
<td>KG1</td>
<td>8.99x10$^{-3}$</td>
<td>-4.78x10$^{-3}$</td>
<td>-1.73x10$^{-3}$</td>
</tr>
<tr>
<td>KG2</td>
<td>2.47x10$^{-5}$</td>
<td>-3.32x10$^{-6}$</td>
<td>1.54x10$^{-5}$</td>
</tr>
<tr>
<td>KG3</td>
<td>-5.47x10$^{-9}$</td>
<td>2.1x10$^{-8}$</td>
<td>0.</td>
</tr>
<tr>
<td>ERH0G</td>
<td>1.07</td>
<td>1.6</td>
<td>1.7</td>
</tr>
</tbody>
</table>
REFERENCES


