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GEAP-3961

PREDICTION OF THE CRITICAL HEAT FLUX IN FORCED CONVECTION FLOW

By
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June 20, 1962

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PREDICTION OF THE CRITICAL HEAT FLUX
IN FORCED CONVECTION FLOW

By
S. Levy

June 20, 1962

Prepared Under
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ATOMIC POWER EQUIPMENT DEPARTMENT

GENERAL ELECTRIC

SAN JOSE, CALIFORNIA

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PREDICTION OF THE CRITICAL HEAT FLUX IN FORCED CONVECTION FLOW

SUMMARY

A superposition model is developed to predict the critical heat flux in forced convection flow. The model is applied to available experimental results in boiling water flows and good agreement is obtained between the model and test data over the multitude of geometries, flow rates, pressures, and fluid enthalpies tested to-date.

INTRODUCTION

Most predictions of the critical heat flux in forced convection flow have been empirical. A multitude of correlations have been proposed in the literature and typical examples are given in references (1), (2), and (3). By contrast, the number of analytical studies has been very limited. Only four models have been presented to-date, all of them published in recent years.

The earliest attempt to obtain an analytical solution in flowing systems was made by Griffith⁽⁴⁾ in 1958. Griffith postulated that the critical heat flux occurred when steam bubbles covered a certain fraction of the heater surface. Using dimensional analysis, he was able to specify this fraction in terms of flow rate, subcooling, and fluid properties. In 1959, Isbin, Vanderwater, Fauske, and Singh⁽⁵⁾ proposed an annular flow model consisting of a liquid film flowing along the channel walls and a central core of steam and dispersed water droplets. It was assumed that liquid was lost from the film by evaporation and entrainment and replenished by diffusion from the central core. Critical heat flux conditions were taken to correspond to disappearance of the liquid film. In 1960, Goldmann, Firstenberg, and Lombardi⁽⁶⁾ utilized the similarity between the mass transfer coefficient and friction factor to calculate the critical heat flux. Droplet diffusion through a steam boundary layer was taken as the dominant mechanism in this analysis. In 1962, Tippets⁽⁷⁾ presented the most comprehensive model yet developed. Starting from the flow model of Isbin and his co-workers⁽⁵⁾, he was able to obtain analytical expressions for the liquid film thickness from stability considerations and to determine the liquid diffusion rates from the central core by means of a turbulent mixing length model.

It is the purpose of this report to present another analytical solution to the problem of critical heat flux in forced convection flow. The proposed model utilizes a superposition method to predict the critical heat flux over the entire range of available test results. It covers both flowing and non-flowing systems under sub-cooled and net vapor generation conditions.

Description of the Method

The superposition method consists of subdividing a complex heat transfer problem into subproblems more amenable to analysis. Solutions are derived for the subproblems and these are added algebraically to obtain an analytical representation

of the overall process. This simplified approach is the basis of predictions proposed here for the critical heat flux in forced convection flow with subcooled fluid and net vapor generation.

In the case of subcooled flow, the critical heat flux under non-flow or pool boiling conditions is first specified. To this heat transfer rate are next added the heat conducted from the surface by the subcooled non-flowing fluid and the heat convected from it by the subcooled flowing fluid. Similarly, in the case of net vapor generation, heat convected from the surface is first added to the pool boiling critical heat flux and an equivalent mass transfer rate is next subtracted from it to account for the presence of vapor in the stream.

Analytical expressions for both cases are derived in the next two sections. It is, however, important before proceeding with the analysis to realize that

1. superposition methods are approximate because they neglect the interaction of the constituent solutions upon one another; and
2. the method of superposition is not new to the boiling field. It has been used previously by Rohsenow⁽⁸⁾ to predict the effects of forced convection in nucleate boiling heat transfer. More recently, it has been applied by Zuber, Tribus, and Westwater⁽⁹⁾ to determine the critical heat flux in subcooled pool boiling.

Critical Heat Flux in Subcooled Forced Convection

As previously indicated, the analysis is carried out in three steps. Addition of the solutions at each step gives the critical heat flux in pool boiling, subcooled pool boiling, and subcooled forced convection flow.

Pool Boiling

Correlations for the critical heat flux in pool boiling have been proposed by Kutateladze⁽¹⁰⁾ and Borishanski⁽¹¹⁾. A similar relation was recently derived in reference (9) on theoretical grounds. Zuber, Tribus, and Westwater postulated that the critical heat flux occurred when the countercurrent flow of liquid and vapor at the heater surface became hydrodynamically unstable. Analysis of this unstable condition gave the following equation for the critical heat flux

$$\left(\frac{q}{A}\right)_p = 0.131 h_{fg} \rho_v \left[\frac{\sigma g^2 (\rho_l - \rho_v)}{\rho_v^2} \right]^{1/4} \quad (1)$$

Equation (1) has been checked against available experimental data and has been found to agree well with the test results.

Subcooled Pool Boiling

The critical heat flux in subcooled non-flowing systems was derived in reference (9). A heat conduction term was added to Equation (1) to account for the intermittent contact of subcooled fluid and heater surface. The conduction term is

given by

$$\left(\frac{q}{A}\right)_c = 0.696 \sqrt{k_L \rho_L c_{pL}} \left(\frac{\rho_L - \rho_v}{\sigma}\right)^{1/4} \left[\frac{\sigma g^2 (\rho_L - \rho_v)}{\rho_v^2}\right]^{1/8} \Delta T_{sub} \quad (2)$$

Addition of Equations (1) and (2) gives the critical heat flux for subcooled pool boiling and, as shown, in reference (9) good agreement is obtained between the sum of Equations (1) and (2) and the test results of Kutateladze and Schneidermann⁽¹²⁾.

Subcooled Forced Convection

The critical heat flux in this case is obtained by adding a convection term to the sum of Equations (1) and (2). The forced convection term is equal to

$$\left(\frac{q}{A}\right)_F = h_L (T_w - T_s) + h_L \Delta T_{sub} \quad (3)$$

The liquid heat transfer coefficient h_L in Equation (3) is calculated from a Colburn type relation

$$h_L = 0.023 \frac{k_L}{D} \left(\frac{GD}{\mu_L}\right)^{0.8} \left(\frac{\mu_c \rho}{\mu_L}\right)^{0.33} \quad (4)$$

The temperature difference ($T_w - T_s$) can be computed from one of the many nucleate boiling correlations available in the literature (8), (13), (14). In the case of water, the simplified Jens-Lottes⁽¹⁾ equation is recommended*

$$T_w - T_s = \frac{60}{e^{P/900}} \left(\frac{q}{A} \frac{1}{10^6}\right)^{1/4} \quad (5)$$

where q/A is the critical heat flux obtained by adding Equations (1), (2), and (3).

Critical heat flux values for forced convection flow of water were calculated from the sum of Equations (1), (2), and (3) and were compared to available test results. The predictions are based upon the use of Equations (4) and (5) with water properties evaluated at saturated conditions for simplification purposes. The ratio of predicted to measured critical heat flux at 2000 psia is plotted in Figure 1 for the test conditions reported in reference (2). It is seen from this figure that over 85 percent of the test points are correlated within ± 30 percent. A definite trend, with respect to mass-flow rates, is also noticeable. The experimental results tend to fall below the predictions at low flow rates and above them at very high flow rates.

*Equation (5) requires the use of a trial and error method. The contribution of the term ($T_w - T_s$) is usually small and a first approximation can be obtained by neglecting it.

This deviation from the superposition model can be explained.

At reduced flow rates, the measured values of critical heat flux are probably low due to hydraulic oscillations. This possibility is illustrated in Figure 2 where experimental data shown in Figure 1 and obtained at two relatively constant flow rates are plotted. The curves in Figure 2 exhibit characteristics noted in reference (15) when the test loop was hydrodynamically unstable. In the high flow range, the low values predicted by the analysis are attributed to the fact that it underpredicts the effects of forced convection. The forced convection term becomes particularly important at high flows and the values derived from Equation (4) are low because they neglect the presence of vapor in the stream. Vapor exists in the stream even at subcooled conditions and its presence will tend to accelerate the fluid in the channel*. The effects of subcooled vapor fraction are also expected to be important for small channel spacings since the vapor fraction in a subcooled fluid increases with reduced channel size (16), (17).

Additional comparisons of the proposed model to test results are shown in Figures 3 and 4. Figure 3 covers experimental results obtained at pressures other than 2000 psia (2), (18), while Figure 4 covers data measured in an annulus at 1000 psia (19). Again, correlation of more than 85 percent of the test points within ± 30 percent is noted in Figure 3, while in Figure 4, all the test data fall within this range. Figure 4 also exhibits an interesting reverse trend with flow rate. Because the results in Figure 4 were obtained with an annulus heated only on the inside surface, channeling of cold water on the unheated surface must take place. This channeling is expected to increase with increased flow rates and the predicted values should exceed the measurements in this range.

Examination of Figures 1, 3, and 4 reveals that the superposition method gives a satisfactory prediction of the critical heat flux of water in forced convection flow. The comparison is all the more satisfying when it is realized that test points often exhibit a ± 30 percent experimental spread. Further verification of the superposition model by applying it to other fluids and conditions is not possible at the present time because insufficient test results are available. Some interesting qualitative trends can still be inferred from the proposed equation. The effects of subcooling and forced convection upon the critical heat flux of water at various pressures are illustrated in Figure 5. As shown in Figure 5, increased subcooling and flow tend to shift the peak critical heat flux to lower pressures. Similarly, the sum of Equations (1), (2), and (3) gives an increase in the critical heat flux as the hydraulic diameter is reduced. It finally makes it possible to evaluate the critical heat flux in fluids not previously tested. It should be noted again that Equation (5) applies only to water and that for other fluids the wall superheat can be evaluated from equations given in references (8), (13), and (14). Equation (4) should also be appropriately modified if fluids of low Prandtl number are utilized.

*This increased velocity could be estimated from reference (17). However, once this is done, one departs from the simple superposition model which allows no interaction between the constituent solutions.

Critical Heat Flux in Forced Convection
With Net Vapor Generation

With net vapor generation, the solution starts again with the non-flowing saturated liquid. The saturated liquid is next set into motion, and finally vapor is added to the main stream. The critical heat flux in the case of a non-flowing saturated liquid was previously specified [see Equation (1)] while motion of the saturated liquid requires the addition of a convection term. The convection heat transfer is obtained from Equation (2)*, except that the second term on the right side of Equation (2) now drops out since the fluid is no longer subcooled. This leaves only the effects of vapor in the main stream to be evaluated. The presence of vapor in the stream can be represented by an equivalent mass transfer rate. The mass transfer rate is obtained from two-phase density distributions derived in reference (20). According to reference (20), the density of a two-phase flowing mixture starts as liquid at the channel wall and decreases towards the center of the channel. Associated with this density distribution is a mass transfer rate M equal to

$$M = -C \left. \frac{d\rho}{dy} \right|_{y=0} \quad (6)$$

The term C in Equation (6) is an equivalent diffusion coefficient. The gradient $d\rho/dy$ can be calculated from the mixing length solutions of reference (20) and Equation (6) becomes

$$M = \frac{C \rho_L}{\mu_L} \frac{K^2 \beta^2}{1 - \rho_v / \rho_L} G \quad (7)$$

In Equation (7) K is the constant used in the mixing length expression (usually taken as 0.4) while β is a parameter dependent upon the channel geometry, the ratio of mean fluid to liquid density, and the liquid Reynolds number based upon the total mass flow rate G .

Part of the mass transfer rate M must be in the vapor stage and the corresponding vapor flow M_v is taken proportional to the average vapor fraction $\bar{\alpha}$ in the channel

$$M_v = M \bar{\alpha} = \frac{C \rho_L}{\mu_L} \frac{K^2 \beta^2}{1 - \rho_v / \rho_L} G \quad (8)$$

*A two-phase heat transfer convection term would be more appropriate here. Satisfactory correlations for two-phase flow heat transfer are not available and Equation (2) is utilized in the meantime. Note that in most practical applications, the convection term is small so that use of Equation (2) is acceptable.

To produce this mass transfer rate it would be necessary to have a heat flux $\left(\frac{q}{A}\right)_M$ equal to

$$\left(\frac{q}{A}\right)_M = - h_{fg} M_v = - h_{fg} \frac{C \rho_L}{\mu_L} \frac{K^2 \beta^2}{1 - \rho_v/\rho_L} G \quad (9)$$

A negative sign is used in Equation (9) since the vapor mass transfer rate is towards the channel wall rather than away from it. Equation (9) can now be added to Equations (1) and (2) to obtain the desired solution. Before proceeding with this addition, a small modification will first be made to Equation (9). The heat flux rate of Equation (9) assumes that there exists an infinitely thin liquid film against the channel wall. If the liquid film were actually finite, as reported in reference (7), the mass transfer rate will depend upon the ratio of liquid film thickness to channel spacing or diameter. The mass transfer rate would decrease as this ratio increases, tending to zero as the liquid film thickness approaches half of the channel spacing. This effect can be approximated by a power type expression, $(D/2\delta)^m$ where δ represents the liquid film thickness. If we assume that δ is proportional to the wave length at the interface of the liquid film and if the wave length is taken as the critical wave length obtained in the analysis of pool boiling, the expression for $\left(\frac{q}{A}\right)_M$ becomes

$$\left(\frac{q}{A}\right)_M = - \frac{B C \rho_L}{\mu_L} h_{fg} \frac{K^2 \beta^2}{1 - \rho_v/\rho_L} G \left[\frac{(\rho_L - \rho_v) D^2}{4 \pi^2 \sigma} \right]^{m/2} \quad (10)$$

where B is a proportionality constant. The exponent m is specified on the basis that we are dealing with a hydrodynamically unstable system. Stability analyses have shown (7), (9), that the critical velocity (proportional here to M_v/ρ_v) varies inversely with the square root of the wave length, and $m = 1/2$. The final expression for $\left(\frac{q}{A}\right)_M$ is

$$\left(\frac{q}{A}\right)_M = - \frac{B C \rho_L}{\mu_L} h_{fg} \frac{K^2 \beta^2}{1 - \rho_v/\rho_L} G \left[\frac{(\rho_L - \rho_v) D^2}{4 \pi^2 \sigma} \right]^{1/4} \quad (11)$$

The sum of Equations (1), (2) and (11) gives the critical heat flux in forced convection flow with net vapor generation. By contrast to subcooled flow, the solution now includes an undefined term, $B C \rho_L/\mu_L$ which must be obtained from experimental data. Before proceeding with calculation of the grouping $B C \rho_L/\mu_L$ let us qualitatively examine the trends predicted by the model. In order to facilitate the discussion, the dependence of the parameter β on total liquid Reynolds number $G D/\mu_L$ will first be expressed analytically. In reference (20), values of β are tabulated in terms of ρ_v/ρ_L and $G D/\mu_L$. For instance at a total liquid Reynolds number, $G D/\mu_L$, of 10^5 the following relation exists between β and mean density in a circular pipe

β	0.04	0.1	0.2	0.3	0.4	0.5	0.6	0.7
ρ_v/ρ_L	0.730	0.477	0.252	0.141	0.0836	0.0516	0.0335	0.0227

The corresponding values for flow between parallel plates are

β	0.04	0.1	0.2	0.3	0.4	0.5	0.6	0.7
ρ/ρ_L	0.715	0.453	0.227	0.122	0.0678	0.0397	0.0245	0.0158

Values at other Reynolds numbers are given in reference (20). It was, however, found that these could be readily predicted from the equation

$$\beta = \beta' \left(\frac{GD}{\mu_L} \frac{1}{10^5} \right)^{-1/6} \quad (12)$$

where β' is the value of the parameter β at $GD/\mu_L = 10^5$. Also, plots of β versus ρ/ρ_L show that the circular pipe values of β are about 8.3 percent larger than for parallel plates. Equation (11) can be rewritten

$$\left(\frac{q}{A} \right)_M = - \frac{BC\rho_L}{\mu_L} h_{fg} \frac{K^2 \beta'^2}{1 - \rho/\rho_L} \left[\frac{(\rho_L - \rho) D^2}{4\pi^2 \sigma} \right]^{1/4} \left(\frac{GD}{\mu_L} \frac{1}{10^5} \right)^{-1/3} \quad (13)$$

Effects of vapor fraction, flow rate, and geometry can now be evaluated from Equation (13). The most important results are as follows:

1. Increased vapor fraction reduces the mean fluid density. The only term affected in Equation (13) by this reduction is the parameter β' . According to the preceding tabulations, β' increases rapidly as the ratio ρ/ρ_L is reduced. In other terms $(q/A)_M$ becomes larger with increased void fraction and the critical heat flux drops sharply. This is without doubt the dominant effect in Equation (13).
2. The heat flux $(q/A)_M$ is proportional to $G^{2/3}$ according to Equation (13). Increased flow velocity means, therefore, a reduced value of critical heat flux. From a physical viewpoint, the trend can be explained if one realizes that an increased flow rate flattens the density profile and leads to higher density gradients at the wall.
3. There are compensating effects in the case of hydraulic diameter. According to Equation (13), the critical heat flux increases as the hydraulic diameter is reduced but its dependence upon hydraulic diameter is small.
4. Since the values of β' are higher for a circular pipe than in a parallel plate geometry, the circular pipe will give slightly lower critical heat flux values than a parallel plate system.

All the above qualitative trends are in agreement with test results. A detailed comparison of the analysis and data will now be presented for the case of water flow with net steam generation.

Figure 6 shows a comparison of the predictions with experimental results in an 0.097 x 1 inch rectangular channel at 2000 psia (2). The predictions are based upon a value of 0.355 for the grouping $BC\rho_L/\mu_L$. It is seen that over 85 percent of the

test points are correlated within ± 30 percent and that over 95 percent fall within ± 40 percent. Experimental data obtained at Bettis in other channel sizes are shown in Figure 7. The same degree of correlation is obtained as in Figure 6. It can, therefore, be inferred from Figures 6 and 7 that the superposition model gives satisfactory answers over the large range of fluid enthalpy, flow rate, and channel geometries tested at 2000 psia. It is true that correlation of the data within a narrower band would be desirable. This, however, is rather difficult at the present time since the test points exhibit an inherent scatter of at least the same range. Typical experimental data obtained in a fixed channel and at a relatively constant mass flow rate are plotted in Figure 8. It is seen that for all three cases illustrated the spread in test points exceeds ± 30 percent and sometimes reaches up to ± 60 percent.

Additional comparisons of the model with test results are given in Figures 9, 10, and 11. Figures 9 and 10 cover data obtained at 1000 psia, while Figure 11 deals with 600 and 1200 psia. A satisfactory correlation is obtained in Figure 9 with CISE data (21) at different flow rates, heated lengths and pipe diameters. Similarly, the data of Tippets are well correlated in Figure 10. Test results at 600 and 1000 psia are also satisfactorily predicted in Figure 11. It should be noted here that only test results with heated length to diameter ratio greater than 60 are considered in Figures 6 to 11. Also, CISE data with inlet steam quality above 0.15 were not used as they produced a peak in the curve of critical heat flux versus steam quality.* Also, only the test data obtained by Tippets with a 0.010 inch heater ribbon are shown in Figure 10 because they are considered more reliable. Finally the following values of void fraction versus steam quality were used: (see Table I).

The values of the parameter $BC\rho/\mu_L$ used in Figures 6 to 11 are shown in Figure 12. A simple relation is obtained when this parameter is plotted against the density ratio ρ_L/ρ_v . An approximate equation for $BC\rho/\mu_L$ is

$$\frac{BC\rho}{\mu_L} = 2.75 \frac{\rho_v}{\rho_L} \quad (14)$$

This relation is valid only for the two geometries treated in reference (20), namely, pipe and parallel plate flow. It is expected that the parallel plate model can be applied to an annulus heated on both surfaces. In the special case of an annulus or parallel plate system heated on one side only, the previous analysis breaks down since the density distribution will no longer be symmetrical as assumed in reference (20). The effects of heating only one surface are schematically represented in Figure 13. A density distribution skewed towards the heated surface will result and if the average steam density is maintained the same as in the case of heating both surfaces, the density gradient next to the heated wall will be much larger for asymmetrical heating. Until solutions of the type derived in reference (20) are developed for asymmetrical heating, a simplified approach for taking it into account consists of changing the value of the grouping $BC\rho/\mu_L$ **

*This peak is probably due to flow oscillations as illustrated in Figure 2.

**Another method is to assume that the heated surface sees a lower mean density than exists over the entire channel.

TABLE I

STEAM QUALITY, x

	0	0.010	0.050	0.100	0.200	0.300	0.400	0.500	0.600	0.700	0.800	0.900	1.00	Pressure (psia)
Void Fraction	0	0.250	0.561	0.617	0.785	0.830	0.865	0.894	0.917	0.939	0.960	0.982	1.000	600
Void Fraction	0	0.150	0.445	0.590	0.722	0.785	0.830	0.865	0.896	0.922	0.950	0.976	1.000	1000
Void Fraction	0	0.118	0.395	0.552	0.690	0.760	0.813	0.854	0.885	0.916	0.945	0.974	1.000	1200
Void Fraction	0	0.068	0.278	0.448	0.647	0.758	0.830	0.879	0.917	0.944	0.968	0.985	1.000	2000

It was found that the experimental data of Janssen and Kervinen (19) at 1000 psia in an internally heated annulus could be predicted by multiplying the values in Figure 12 by 2.8. Test results given in reference (19) indicate that the constant 2.8 remains relatively the same as the pressure is varied. A recommended equation for annuli or parallel plates heated on one side only is, therefore,

$$\frac{BC\rho_L}{\mu_L} = 7.7 \frac{\rho_v}{\rho_L} \quad (15)$$

The use of Equations (13) and (15) is illustrated in Figure 14. Good correlation is obtained of the test results given in reference (19) at various flow rates, heated lengths and hydraulic diameters.

Figure 14 illustrates, once again, the ability of the superposition model to grasp the most important experimental trends. This ability has now been repeatedly demonstrated for forced convection flow of water under subcooled and net steam generation conditions. An important characteristic of the model is that it agrees with the results of so many experimenters. There are, without doubt, some regions where the proposed correlation could still be improved. Such improvements must, however, await the following:

1. The reduction in the spread of experimental points. Such spread was illustrated in Figure 8, but it can also be observed in Figures 9, 11, and 13.
2. Better understanding of the phenomena involved. The problem of diffusion in a two-phase mixture deserves further analytical and experimental attention.
3. Additional test results with fluids other than water to establish the validity of the proposed equations and in particular to verify the proposed variation of $\frac{BC\rho_L}{\mu_L}$ with fluid properties.

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NOMENCLATURE

A	heat transfer area, ft^2
B	proportionality constant, non-dimensional
C	diffusion coefficient, ft^2/hr
c_p	specific heat, $\text{BTU}/\text{lb}^\circ\text{F}$
D	hydraulic diameter, ft.
G	mass flow rate per unit area, $\text{lb}/\text{hr}\text{-ft}^2$
g	gravitational constant, ft/hr^2 , $(32.2) (3600)^2$
h	heat transfer coefficient, $\text{BTU}/\text{hr}\text{-ft}^2\ ^\circ\text{F}$
h_{fg}	heat of vaporization, BTU/lb
K	mixing length constant, non-dimensional, (0.4)
k	thermal conductivity, $\text{BTU}/\text{hr}\text{-ft}^\circ\text{F}$
M	mass transfer rate, $\text{lb}/\text{hr}\text{-ft}^2$
m	exponent, non-dimensional
P	pressure, psia
q	heat flux, BTU/hr .
T_s	saturation temperature, $^\circ\text{F}$
T_w	wall temperature, $^\circ\text{F}$
x	steam quality, non-dimensional
y	distance from channel wall, ft.
$\bar{\alpha}$	average vapor fraction in channel, non-dimensional
β	non-dimensional constant
β_c	value of β at $\text{GD}/\mu_c = 10^5$
δ	liquid film thickness, ft.
μ	absolute viscosity, $\text{lb}/\text{hr}\text{-ft}$
ρ	density, lb/ft^3
σ	surface tension, lb/ft
ΔT_{sub}	subcooling, or saturation temperature less bulk fluid temperature, $^\circ\text{F}$

SUBSCRIPTS

a	average
c	conduction
F	forced convection
l	liquid
M	mass transfer
p	pool boiling
v	vapor

FIGURE 2 POSSIBILITY OF HYDRAULIC OSCILLATIONS AT LOW FLOWS

FIGURE 3 COMPARISON OF PREDICTION WITH SUBCOOLED WATER DATA AT PRESSURE OF 60-2750 PSIA

FIGURE 4 COMPARISON OF PREDICTION WITH SUBCOOLED WATER DATA IN ANNULUS AT 1000 PSIA

FIGURE 5 EFFECTS OF SUBCOOLING AND FORCED CONVECTION UPON CRITICAL HEAT FLUX OF WATER

FIGURE 6 COMPARISON OF PREDICTION WITH CRITICAL HEAT FLUX DATA IN RECTANGULAR CHANNELS AT 2000 PSIA

FIGURE 7 COMPARISON OF PREDICTION WITH OTHER CRITICAL HEAT FLUX DATA AT 2000 PSIA

FIGURE 8 TYPICAL SCATTER IN EXPERIMENTAL DATA AT 2000 PSIA

FIGURE 9 COMPARISON OF PREDICTION WITH CISE DATA AT 1000 PSIA

FIGURE 10 COMPARISON OF PREDICTION WITH RECTANGULAR CHANNEL DATA AT 1000 PSIA

FIGURE 11 COMPARISON OF PREDICTION WITH CISE DATA AT OTHER PRESSURES

FIGURE 12 VALUE OF PARAMETER $\frac{BC\rho_L}{\mu_L}$ FOR FLOW BETWEEN PARALLEL PLATES

FIGURE 13 EFFECTS OF ASSYMETRIC HEATING UPON DENSITY DISTRIBUTION

FIGURE 14 COMPARISON OF PREDICTION WITH INTERNALLY HEATED ANNULAR DATA AT 1000 PSIA