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NONUNIFORM HEAT GENERATION
EXPERIMENTAL PROGRAM

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PRELIMINARY REPORT
U. S. - EURATOM EXCHANGE

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Lynchburg, Virginia

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PREFACE

The Babcock & Wilcox Company and the United States Atomic Energy Commission, acting for the Joint U.S. -Euratom Research and Development Board, signed an agreement for a nonuniform heat generation experimental program.

The program includes experimental and analytical investigations to determine how nonuniformly distributed heat flux along lengths of tubular and annular channels affects the onset of burnout (departure from nucleate boiling). The experimental program is being conducted in the 1.8-MW heat transfer facility at B&W's Research Center, Alliance, Ohio. The analytical program is being conducted by the Atomic Energy Division, Lynchburg, Virginia.

SUMMARY

Since the beginning of the program, complete sets of data have been obtained on four tubular test sections. One had uniform flux to serve as a reference condition, and three had nonuniform axial heat flux to show the effect of flux distribution on critical heat flux conditions. These data were compared with those of other investigators where comparable data were available. Empirical correlations were developed to express mathematically the effects of the various operating variables on critical heat flux. Before the present report period, complete sets of data were obtained on two annular test sections. In the first case, the inside surface was heated, and in the second, the outside surface was heated. Comparison with published burnout correlations revealed a need for improved correlations.

During this report period, data have been obtained on an annular test section having both surfaces heated. The experiments through December included all parameters, but the time permitted evaluating only the 1500 psia data before the end of the quarter. An additional task, a feasibility study of an analytical burnout model, was completed.

SCOPE

In an effort to clarify some of the areas of disagreement about the burnout phenomenon, Babcock & Wilcox, as part of the Joint U.S. - Euratom Research and Development Program, is conducting an experimental and analytical program to determine how nonuniform heat generation rates in tubular and annular flow channels affect the operating conditions causing burnout, under experimental conditions representative of water reactor systems. The objective of the program is to seek improved design correlations suitable for direct use by reactor designers; the technical approach to the work emphasizes the development of an improved understanding of the physical phenomena involved and their relation to design variables.

The scope of the experimental program includes data taken on tubular test sections with these conditions:

<u>Axial flux shape</u>	<u>Pressure, psia</u>	<u>Mass velocity, lb/hr-ft² × 10⁻⁶</u>	<u>Inlet subcooling, F</u>
Uniform	2000	0.5	20
Peak near inlet	1500	1.5	50
Peak near midpoint	1000	2.5	100
Peak near outlet	--	--	150

In addition, data have been taken on three annular test sections involving uniform axial flux shapes. Corresponding data will be taken on two more annular test sections involving cosine axial flux shapes.

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1. INTRODUCTION

Nuclear reactors and modern boilers require more and more knowledge about the limitations on the transfer rates of heat from channel walls to flowing liquids. Burnout (departure from nucleate boiling, or critical heat flux) is a serious design problem still facing the reactor designer. As a result of this phenomenon, the heat transfer conductance is reduced considerably, causing the metal and fuel temperatures to rise and initiate failure of the fuel element. Although considerable attention has been given to this problem in recent years, the predictability of the occurrence of burnout conditions is quite uncertain. In fact, there is still much disagreement as to how the channel geometry and operating conditions affect burnout.

In an effort to clarify some of the areas of disagreement, Babcock & Wilcox (B&W), as part of the Joint U.S. -Euratom Research and Development Program, is conducting an experimental and analytical program to determine the effects of nonuniform heat generation rates in tubular and annular flow channels on the operating conditions causing burnout under experimental conditions representative of water reactor systems. The technical approach to the work emphasizes the development of an improved understanding of the physical phenomena involved and their relation to design variables.

The experimental program, which was discussed in the first quarterly report, is being conducted on a heat transfer facility at B&W's Research Center in Alliance, Ohio.¹ The controlled variables for the tests are pressure, inlet subcooling, and mass velocity. The tests are being performed in 6-foot-long tubular and annular test sections having both uniform and nonuniform axial heat flux distributions. The Atomic Energy Division in Lynchburg, Virginia, is conducting the analytical program.

1.1. Previously Reported Work

Fabrication progress on both tubular and annular test sections, along with experimental work and data analysis on the tubular test sections, has been reported previously.²⁻⁷ In addition, experimental work and preliminary data analysis on two annular test sections have been reported.^{8,9}

1.2. Current Work

This report concerns the experimental work on the third annular test section (test section 7, inside and outside tubes heated). Since final calibration data were not available when this report was written, the data plots, results, and conclusions must be considered tentative and subject to revision in the next quarterly report.

Test section 7 was reinstalled in October, and test runs were started in November. Testing was completed in December. Preliminary plots of the data collected to date on test section 7 are compared with those obtained from a uniformly heated tubular test section (test section 1).

The data from annular test section 7 will be compared with those from test sections 5 (inside tube heated) and 6 (outside tube heated) in the next quarterly report.

The feasibility of an analytical critical heat flux model was determined; the physical concepts, derivations, and supporting calculations are reported.

2. EXPERIMENTAL EQUIPMENT

The principal geometric measurements of the uniform annular test sections are as follows:

OD (inside tube), inches	0.607
ID (outside tube), inches	1.053
Heated length, inches	72

Reference 8 includes design and instrumentation details. Both surfaces of the annulus were heated for the experiments during the current report period.

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3. EXPERIMENTAL OPERATION

The experimental procedures used for uniform annular test sections were described in the eighth quarterly report.⁸ In one of these procedures pressure drops across the test section were taken to detect possible changes in the flow channel. Since pressure drops are taken daily, the test results have been updated to include new data. Figure 1 shows a single isothermal friction factor curve for all three annular test sections. This figure also shows the curve for the tubular test sections. The higher friction for the annular test sections is probably due to the additional pressure drop across the spacer pins in the flow channel.

Figure 1. Results of Pressure Drop Tests on Test Sections
1 Through 7

4. EXPERIMENTAL DATA

4.1. Data From Test Section 7

The results from test section 7 at 1500 psia are shown as plots of critical heat flux versus inlet subcooling in Figure 2 and critical heat flux versus exit steam quality in Figure 4. The data from test section 7 are compared with those from tubular test section 1 in Figures 3 and 5 using the same parameters. Conclusions from these figures are listed below.

4.2. Conclusions From Test Section 7

From Figures 2 and 4 the following conclusions are drawn:

1. At high subcoolings (>60 F) the critical heat flux always occurs on the outside tube (see Figure 2).
2. At low subcoolings (<60 F) the location of the critical heat flux depends on the mass velocity. From Figure 2 the relationship is:
 - a. $G \lesssim 1.0 \times 10^6$ lb/hr-ft² inside tube.
 - b. $1.0 \times 10^6 \leq G \leq 1.5 \times 10^6$ inside or outside tube.
 - c. $G \gtrsim 1.5 \times 10^6$ lb/hr-ft² outside tube.
3. The relationships between critical heat flux and the variables, inlet subcooling and exit steam quality, follow the same trends as those observed for annular test sections 5 (inside tube heated) and 6 (outside tube heated). For test section 7 see Figures 2 and 4 of this report, and for test sections 5 and 6 see the ninth quarterly report.⁹
4. The relationship between critical heat flux and inlet subcooling at constant mass velocity is approximately linear when both tubes are heated (see Figure 2).

4.3. Comparison of Annular and Tubular Results

From Figures 3 and 5 the following conclusions are drawn:

1. The annular data (test section 7) agree reasonably well with the tubular data (test section 1) at a mass velocity of 0.5×10^6 lb/hr-ft² (see Figure 3).
2. The magnitude of the critical heat fluxes obtained for a round tube is less than that obtained for a totally heated annulus at all mass velocities and inlet subcoolings; the relative difference increases with increased mass velocity (see Figure 3).
3. Plotted against the local parameter, steam quality, the magnitude of the critical heat flux obtained for a round tube is less than that obtained for a totally heated annulus at all mass velocities and exit qualities except at 0.5×10^6 lb/hr-ft² when the quality is less than 50% (see Figure 5).

Figure 2. Critical Heat Flux Vs Inlet Subcooling for Annular Geometry — Both Tubes of Test Section 7 Uniformly Heated at 1500 psia

Figure 3. Comparison of Tubular Test Section 1 With Annular Test Section 7 at 1500 psia on Basis of Inlet Subcooling

Figure 4. Critical Heat Flux Vs Exit Steam Quality for Annular Geometry — Both Tubes of Test Section 7 Uniformly Heated at 1500 psia

Figure 5. Comparison of Tubular Test Section 1 With Annular Test Section 7 at 1500 psia on Basis of Exit Steam Quality

5. FUNDAMENTAL CRITICAL HEAT FLUX STUDY

As proposed at the beginning of this program, a suitable analytical model for critical heat flux calculations is being sought. The data from this program are being examined in an effort to discover a model suggested by the behavior of the data. Original ideas, along with information from the literature and available authorities, are being considered in an effort to produce a satisfactory theorem for the critical heat flux phenomenon. The progress on this latter task is presented in this section.

5.1. Background Information

The deterioration of boiling heat transfer coefficients under certain conditions has been observed since 1934.^{10,11} Since that time, especially in the last decade, efforts to understand the phenomenon have led to many studies and experiments. However, at the present time it is neither adequately explained nor adequately predicted.

According to one popular theory, when steam production at a liquid-solid interface reaches a certain high rate, the solid is blanketed by a layer of steam because the steam cannot escape from the wall at a sufficient rate. This theory seems inadequate because it neither explains the suddenness with which this phenomenon sometimes occurs, nor does it explain the stability of such a dry steam layer in highly turbulent channels at qualities well below 100%.

An effort to advance beyond the steam-blanket concept is offered in the proposed climbing-film theory.¹² This theory is based on the existence of a liquid film adjacent to the heated wall of a channel. The critical heat flux is produced by the breakup or the dryout of this film. This continuous film may exist, and one cannot dispute the assertion that its breakup would produce a heat transfer crisis. Granted this, the basic physical principle behind this phenomenon is still being sought.

The preceding theories recognize that the critical heat flux occurs when the heat transfer rate suddenly begins to approach that of dry steam. This phenomenon can occur when the bulk condition of the fluid is near 100% quality, which is not surprising. On the other hand it can occur at very low qualities or even during bulk subcooling. This is the part of the problem that makes prediction and explanation difficult. Most indicated burnout points measured in laboratories have occurred when considerable amounts of liquid have been present in the two-phase mixture. In fact, some observations have indicated that the crisis can occur and the heat transfer conditions can continue to deteriorate as a function of time without any discernable change in the liquid content of the flowing fluid.¹³ The apparent reason is that something causes the heated wall to prefer the vapor to the liquid. The observations indicate that critical heat flux may be heavily dependent on additional variables not usually considered in burnout studies. If this is true, a study of critical heat flux in the light of theories of interfacial phenomena is long overdue. In other words, the wettability of metals by liquids at high temperatures and pressures under conditions of heat and mass transfer should be studied seriously and carefully.

Studies of interfacial phenomena include several works covering conditions at atmospheric pressure and room temperature.¹⁴⁻¹⁸ Much study has been in connection with sprayed insecticides, which must cling effectively to plants and bugs. Wetting in high-temperature and low-pressure systems has been studied in the field of metallurgy with respect to intermetallic bonding or welding.^{19,20} On the other hand, very little work has been done on the wettability of heated solid surfaces in contact with power transport liquids, such as water, at high temperatures and pressures. The three phases of the system to be studied are solid metal, liquid coolant, and vapor coolant; the two components are solid metal and fluid coolant.

5.2. Criterion for Measuring Wettability

One of the most common ways of designating the wettability of a solid by a liquid in the presence of vapor or gas is to quote the contact angle, illustrated by θ in Figure 6. The theory of wetting depends on the concept of surface energy, which for ideal systems can be computed from

the equations by Gibbs¹⁵ or Helmholtz¹⁴. These equations are not quoted here. For the present, the equation of "wetting" or "nonwetting" is as follows:

$$\theta = \arccos \frac{F_{SV} - F_{SL}}{\gamma_{LV}} \quad (1)$$

where

θ = contact angle measured through the liquid between the tangent to the liquid and the plane of the solid at the solid-vapor-liquid intersection

F_{SV} = free energy of the solid-vapor interface

F_{SL} = free energy of the solid-liquid interface

γ_{LV} = surface tension at the liquid-vapor interface
= free energy of the liquid-vapor interface

It is convenient to use the same kind of symbol for all of the free energies; hence equation 1 is often written as

$$\theta = \arccos \frac{\gamma_{SV} - \gamma_{SL}}{\gamma_{LV}} \quad (2)$$

If θ is 0° , the solid is considered to be completely wetted. If $0^\circ < \theta \leq 90^\circ$, the solid is considered by some authors to be "partially" wetted. If $90^\circ < \theta \leq 180^\circ$, the solid is considered to be nonwetted. This equation is valid for θ over the range of $0^\circ \leq \theta \leq 180^\circ$. If there is a transition from wetting to nonwetting in a system, the free energies must have certain limitations. For example,

$$|\gamma_{SV} - \gamma_{SL}| \leq \gamma_{LV} \quad (3)$$

or

$$-\gamma_{LV} \leq (\gamma_{SV} - \gamma_{SL}) \leq +\gamma_{LV} \quad (4)$$

If $\gamma_{SV} = \gamma_{SL}$, θ is 90° . The problem is determining the various free energies. Values for γ_{LV} are the easiest to obtain. They have been published for many liquids, especially water, but where they have not, laboratory methods for measuring surface tension are readily available.

The difficulty in the application of this wetting theory is the calculation or measurement of γ_{SV} and γ_{SL} . There is some indication in the literature that these values are large. In considering the wetting of metal by liquid metals, Gilliland¹⁹ used a γ_{SV} value of 1863 dynes/cm

for beryllium. Taylor²⁰ quotes values of 1400 dynes/cm for gold and 1670 dynes/cm for copper. Therefore, iron or steel probably has a value near 2000 dynes/cm. This points to another difficulty. If the value is this large, only a low-percentage error is allowable in the study of the steel-water-steam system, because γ_{LV} for water ranges from about 80 dynes/cm at room temperature to zero at the critical temperature, and the difference between γ_{SV} and γ_{SL} must be less than γ_{LV} (according to equation 3). Obtaining an accurate difference between two large, close values can be difficult.

5.3. Study of the Contact Angle by the Use of Observed Effects

Without considering how these numbers might be obtained, investigating the feasible ranges of equation 2 is useful to determine whether boiling phenomena can possibly be due to changes in the free energy of interfaces. Of first interest are the trends of these values as a function of some variable, such as temperature. The study of the Leidenfrost effect¹⁶ is related. Leidenfrost reported his work in 1756, but three modern sources of data on this subject are Gottfried,¹⁶ Poppendiek,¹⁷ and Borishansky. The data by Borishansky¹⁸ are used here, because his paper presents the most complete information about water in the nucleate boiling regime, which precedes the regime of the Leidenfrost phenomenon as surface temperature increases.

Borishansky performed rather simple droplet vaporization experiments, which yielded the results shown in Figure 7. The data were obtained by placing standard-size, known-volume droplets on a heated metal surface having known temperatures greater than the boiling point. The lifetimes of the droplets were measured to give an indication of the rate of heat transfer from the plate to a droplet. Figure 7 shows the extremes of heat transfer rates experienced during boiling processes. At the left side of the figure, the rate of heat transfer is good; as the surface temperature increases, the rate improves until a minimum vaporization time is reached, corresponding to the best boiling heat transfer coefficient. As the surface temperature increases further, the droplet vaporization time increases to a maximum, where the poorest heat transfer rate, corresponding to film boiling, occurs. The investigators quoted were most interested in film-boiling heat transfer, shown by the right-hand branch of the curve. Although this regime is not directly applicable

to the present study, certain interesting calculations concerning the transition from nucleate boiling to film boiling can be accomplished.

First, a model is chosen to represent a droplet on a surface. Experience seems to support the notion that high boiling heat transfer rates exist only when the liquid can contact the heated surface. Consequently, the area of contact between a liquid droplet and the heated solid is quite important. The droplet is assumed to be a segment of a sphere; this assumption is good if the volume is small. It is now possible to derive an expression for the contact area as a function of the contact angle and the volume of the droplet. Figure 8 shows a segment of a sphere having height Z and radius r . The volume of this spherical segment is given by

$$V = \frac{1}{3} \pi Z^2 (3r - Z) \quad (5)$$

From Figure 8,

$$Z = r + b$$

$$b = r \sin (\theta - 90^\circ) = -r \cos \theta$$

$$Z = r(1 - \cos \theta), \text{ where } 0^\circ \leq \theta \leq 180^\circ. \quad (6)$$

Substituting equation 6 in equation 5 gives

$$V = \frac{1}{3} \pi r^3 [3(1 - \cos \theta)^2 - (1 - \cos \theta)^3] \quad (7)$$

Solving for r gives

$$r = \left\{ \frac{3V}{\pi [3(1 - \cos \theta)^2 - (1 - \cos \theta)^3]} \right\}^{1/3} \quad (8)$$

The contact area is given by

$$A = \pi a^2 \quad (9)$$

$$a = r \cos (\theta - 90^\circ)$$

$$a = r \sin \theta \quad (10)$$

Combining equations 9 and 10 gives

$$A = \pi r^2 \sin^2 \theta, \text{ where } 0^\circ \leq \theta \leq 180^\circ. \quad (11)$$

Substituting equation 8 in equation 11 gives

$$A = \frac{\pi^{1/3} 3^{2/3} V^{2/3} \sin^2 \theta}{[3(1 - \cos \theta)^2 - (1 - \cos \theta)^3]^{2/3}} \quad (12)$$

Let

$$\frac{[3(1 - \cos \theta)^2 - (1 - \cos \theta)^3]^{2/3}}{\pi^{1/3} 3^{2/3} \sin^2 \theta} = R \quad (13)$$

Then,

$$A = \frac{V^{2/3}}{R} \quad (14)$$

If, during time increment Δt , the droplet volume decreases by increment ΔV ,

$$q'' A \Delta t = -\rho H_{fg} \Delta V$$

where

$$\begin{aligned} q'' &= \text{heat flux} \\ \rho &= \text{liquid droplet density} \\ H_{fg} &= \text{latent heat of vaporization} \end{aligned}$$

Recalling that A is a function of V and allowing Δt and ΔV to become very small,

$$q'' dt = -\rho H_{fg} R \frac{dV}{V^{2/3}} \quad (15)$$

Integrating over the time interval, $0 \leq t \leq t_1$, and over the volume interval $V_1 \geq V \geq 0$,

$$q'' \int_0^{t_1} dt = -\rho H_{fg} R \int_{V_1}^0 V^{-2/3} dV$$

$$q'' t_1 = 3\rho H_{fg} R V_1^{1/3}$$

Solving for q'' gives

$$q'' = \frac{3\rho H_{fg} R V_1^{1/3}}{t_1} \quad (16)$$

The heat transfer coefficient h for area A can be defined by

$$q'' = h \Delta T \quad (17)$$

where ΔT is the difference between the plate temperature and the droplet saturation temperature. Eliminating the heat flux from equations 16 and 17 gives

$$h = \left[\frac{3\rho H_{fg} V_1^{1/3}}{\Delta T t_1} \right] R \quad (18)$$

The data used contain no information about the contact angle. However, Borishansky¹⁸ states one observation that leads to some relative calculations to test the feasibility of the model. Concerning the nucleate boiling portion of the curve (Figure 7 in the temperature range of 100 to 175 C) he states that the droplet spreads over an area equal to five times the projected area of the droplet. Assuming that the droplet is a sphere, the projected area is

$$A_p = \pi^{1/3} \left(\frac{3V}{4} \right)^{2/3} \quad (19)$$

The contact area for the spherical segment is

$$A = \frac{V^{2/3}}{R} \quad (14)$$

Therefore,

$$\frac{A}{A_p} = \frac{V^{2/3}}{R} \div \pi^{1/3} \left(\frac{3V}{4} \right)^{2/3} \quad (20)$$

and

$$\frac{A}{A_p} = 5 \quad (21)$$

Solving equations 20 and 21 gives

$$R = 0.1654$$

The most convenient solution for θ as a function of R is from a graph of R versus θ . Figure 9 indicates that θ is approximately 26° . This seems to be a reasonable contact angle for water on a metal surface. If it is assumed that this contact angle is maintained along the left-hand branch of the curve in Figure 7 until the minimum vaporization time is reached, the heat transfer coefficients can be computed with equation 18. The results of these calculations are shown in Table 1. The coefficient, 22,500 Btu/hr-ft², seems rather high, but it is considered attainable. McAdams²¹ quotes values in the range of 10,000 Btu/hr-ft². The validity of the higher coefficients in Table 1 is borne out by more computations with the assumed model.

Borishansky¹⁸ states that, after the temperature of the plate is increased beyond that which produces the minimum vaporization time, the droplet begins to acquire a spheroidal shape; when the vaporization time is a maximum, the droplet shape is completely spheroidal. In this region of increasing vaporization time, one may assume that the heat transfer coefficient is constant and that the contact angle increases with increasing plate temperature. Using

$$h = 22,500 \text{ Btu/hr-ft}^2$$

equation 18 determines R and, indirectly, θ for data points between the minimum and the maximum vaporization times in Figure 7. These results are also shown in Table 1. Note that, for the maximum vaporization time,

$$\theta = 175^\circ$$

which is close to the contact angle of 180° for a perfectly spherical droplet.

5.4. Conclusions and Tentative Applications

The experimental information from Borishansky gives support to the proposed model. Of course, the single-droplet vaporization model is of limited use unless its information can be extended to practical situations, where surfaces are submerged or where the shear forces of circulating systems are involved. The effect of surface energies reflected in the contact angle may be important in all systems where liquids are heated by solids. Perhaps droplet vaporization experiments can determine all the variables reported by Borishansky and, in addition, can determine the contact angle. These contact angle results may then be valid in the more complicated situations of flooded surfaces and flow systems. The cited measurements on water are at atmospheric pressure. Experiments at pressures up to the critical pressure are needed for application to the systems being designed today.

Table 1. Contact Angles and Heat Transfer Coefficients

$\Delta T,$ F	t, sec	R	$\theta,$ degrees	h, Btu/hr-ft ²	Drop shape
34	12.2	0.1654	26	2,930
43	8.0	.1654	26	3,540	
50	4.2	.1654	26	5,790	
59	3.0	.1654	26	6,860	
65	1.8	.1654	26	10,400	
72	1.6	.1654	26	10,600	
76	0.8	.1654	26	20,000	
90	0.6	.1654	26	22,500	
124	1.6	0.608	98	22,500
126	2.6	1.004	121	22,500	
178	9.0	4.89	156	22,500	
184	10.0	5.62	158	22,500	
214	22.0	14.05	166	22,500	
259	135.4	107.1	175	22,500

Figure 6. Contact Angle and Surface Forces

Figure 7. Droplet Vaporization Data⁽¹⁸⁾

Figure 8. Droplet Model as Spherical Segment

Figure 9. Plot of R as a Function of θ

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6. CONCLUSIONS

1. At high subcoolings (>60 F) the critical heat flux always occurs on the outside tube.
2. At low subcoolings (<60 F) the location of the critical heat flux depends on the mass velocity. The relationship is:
 - a. $G \lesssim 1.0 \times 10^6$ lb/hr-ft² inside tube.
 - b. $1.0 \times 10^6 \leq G \leq 1.5 \times 10^6$ inside or outside tube.
 - c. $G \gtrsim 1.5 \times 10^6$ lb/hr-ft² outside tube.
3. The relationships between critical heat flux and the variables, inlet subcooling and exit quality, follow the same trends as those observed for annular test sections 5 (inside tube heated) and 6 (outside tube heated).
4. The relationship between critical heat flux and inlet subcooling at constant mass velocity is approximately linear when both tubes are heated.
5. The annular data (test section 7) agree reasonably well with the tubular data (test section 1) at a mass velocity of 0.5×10^6 lb/hr-ft² when plotted against the system parameter, inlet subcooling.
6. The magnitudes of the critical heat fluxes obtained for a round tube are less than those obtained for a totally heated annulus at all mass velocities and inlet subcoolings, but the relative difference increases with increased mass velocity.
7. Plotted against the local parameter, steam quality, the magnitude of the critical heat flux obtained for a round tube is less than that obtained for a totally heated annulus at all mass velocities and exit qualities except at 0.5×10^6 lb/hr-ft² when the quality is less than 50%.
8. A fundamental critical heat flux study has been performed as the first stage of the development of an analytical model. This initial analytical work is supported by data in the literature.

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7. FUTURE WORK

The data runs on test sections 8 and 9 are expected to be completed during the next quarter. Work on the analytical model and on the correlation of uniform, nonuniform, tubular, and annular data will continue.

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