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HEAT TRANSFER, FLOW INSTABILITY,
AND CRITICAL HEAT FLUX FOR WATER
IN A SMALL TUBE AT 200 psia

by

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June 1963

Operated by The University of Chicago
under
Contract W-31-109-eng-38
with the
U. S. Atomic Energy Commission

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NOMENCLATURE

c_p	Specific heat of liquid water	Btu/(lb)(°F)
D	Equivalent diameter of flow channel	ft
D_e	Equivalent diameter of flow channel (used in critical heat flux correlation only)	in.
G	Coolant mass velocity	lb/(hr)(ft ²)
h	Forced-convection heat transfer coefficient	Btu/(hr)(ft ²)(°F)
H_f	Enthalpy of saturated liquid water	Btu/lb
H_{fg}	Vaporization enthalpy of water	Btu/lb
H	Local enthalpy of subcooled water or wet steam	Btu/lb
k	Thermal conductivity of liquid water	Btu/(hr)(ft)(°F)
L	Length	ft
Q''	Heat flux	Btu/(hr)(ft ²)
Q_c''	Critical heat flux	Btu/(hr)(ft ²)
T_B	Local bulk coolant temperature	°F
T_{sat}	Saturation temperature at system pressure	°F
T_w	Local transfer surface temperature	°F
x	Local steam quality	w/o
(Gx)	Local steam mass velocity	lb/(hr)(ft ²)
$(H_f - H)$	Local subcooled enthalpy difference	Btu/lb
$(H - H_f)$	Local saturated enthalpy difference	Btu/lb
$(T_w - T_{sat})$	Local transfer surface superheat	°F
σ	Surface tension of liquid water	lb _f /in.
μ	Dynamic viscosity of liquid water	lb/(ft)(hr)
$N_{N\mu}$	Nusselt Number = hD/k	
N_{Pr}	Prandtl Number = $c_p \mu/k$	
N_{Re}	Reynolds Number = GD/μ	

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ABSTRACT

Heat transfer data from a vertical Type 304 stainless steel tube (0.045 in. in ID; 4.5 in. long) operating at 200 psia in forced-convection heating and subcooled nucleate boiling is compared with and agrees with the equations

$$h = \frac{(G)^{0.8}}{10^3(D)^{0.2}} (T_B)^{0.5} \text{ Btu/(hr)(ft}^2\text{)(}^\circ\text{F)} \quad ;$$

$$T_w - T_{\text{sat}} = 0.18 \times 10^6 \sigma (Q''/10^6)^{0.25} \text{ }^\circ\text{F} \quad .$$

Critical heat-flux data from the same system in the subcooled froth-flow regime is compared with and agrees with the correlation

$$Q_c''/10^6 = 1.75(H_{fg}/10^3)(De)^{-0.5} \left(1 + \frac{G}{10^6}\right)^{0.5} \left(\frac{H_f - H}{100}\right) \text{ Btu/(hr)(ft}^2\text{)} \quad .$$

The effects of developing two-phase flow regimes in a constant-pressure drop system are shown in recorded traces from a fast-response flow transducer: the mass velocity decreases and transfer-surface superheat increases with increasing steam quality. Examples of the hypothesized two-phase regimes of froth flow, slug flow, and stable and unstable annular flow are presented, and the critical heat flux occurrence is shown to have a hydrodynamic cause and distinctly characteristic trend in the unstable flow regimes.

I. INTRODUCTION

The core design of the Argonne Advanced Research Reactor (AARR)⁽¹⁾ poses thermal and hydrodynamic problems associated with closely spaced fuel plates, high coolant velocities, and the operational power limitations imposed by "burnout". Permissible temperatures of the core metal and economics of pressure vessels dictate operation at a system pressure of approximately 14 kg/cm² (200 psia) - a range where the occurrence of

"Subsequent increases in the rated design power require a current design pressure of 800 psia at the core inlet."

subcooled nucleate boiling might result in excessive local frictional losses in the core. Any resulting redistribution of flow within the core would tend to be autocatalytic, driving the local originating condition toward burnout.

To confirm the applicability of standard forced-convection heat transfer and nucleate-boiling equations to the geometrical size and operational range of the proposed AARR core design, a series of slow-transient heat-transfer tests was conducted in the forced-convection-heating, subcooled nucleate-boiling, and net steam-generation regimes, up to the occurrence of the critical heat flux.

II. TEST APPARATUS AND PROCEDURE

The tests were conducted in a general-purpose, closed-system, recirculating flow loop incorporating a high-head (125-psi) centrifugal circulating pump and steam-dome pressurization. Effective de-aeration of the demineralized water coolant was accomplished during the warmup period of system stabilization.

A. Test Section

The heat-transfer test section was a Type 304 stainless steel tube (0.045 in. in ID; 0.010 in. wall), with an electric resistance-heated length of 4.5 in. The inside diameter conformed with the anticipated fuel-plate spacing in the AARR core. Extensions of the heated section ($L/D = 100$) included a calming-entrance section ($L/D = 68$) and a discharge section ($L/D = 54$) to reduce hydrodynamic entrance and exit effects.

Electric resistance heating resulted in essentially constant heat generation along the wall of the test section. The continuously variable voltage of the ac power supply was disengaged by a rate-of-change-of-resistance Wheatstone Bridge type of burnout control, with a total response time of approximately 50 ms.

The outer wall temperature was measured at a point $\frac{1}{8}$ in. from the exit end by butt-welded iron-constantan thermocouple leads, the outer tube surface serving as an intermediate metal section of the thermocouple.

B. Coolant Temperatures

Inlet and exit coolant temperatures were measured immediately before and after the test section by copper-tipped, iron-constantan immersion thermocouples. Coolant and metal surface temperatures were recorded on a six-point, direct-reading Brown temperature recorder, furnishing a semi-continuous thermal history of each test run. The coolant immersion thermocouples were steam calibrated over an appropriate range of pressures after installation, agreeing with standard steam-table values within the $\pm 1^\circ\text{F}$

resolution of the Brown recorder. Hot, isothermal high-velocity runs were made to compare the metal surface thermocouple with the inlet and exit coolant thermocouples. Again, the agreement was within the $\pm 1^\circ\text{F}$ resolution.

C. Coolant Flow Rates

Coolant flow rates were measured with a sharp-edged orifice plate ($\frac{1}{16}$ in. in diameter) mounted between pressure-tap flanges in a horizontal, $\frac{3}{8}$ -in., Schedule 40 pipe. Except for size and diameter ratio, ASME specifications were adhered to, and the calibrated accuracy of measurement was $\pm 2\%$. A continuous velocity record of each test run was made from the output of a differential pressure transducer connected across the metering orifice in parallel with a 60-in. mercury manometer.

D. Electric Power Inputs

Electrical power measurements were made with a $\frac{1}{4}\%$ wattmeter. Steady-state heat-balance runs showed heat losses to be less than the combined accuracy of measurement of the several variables involved.

E. System Pressure

System pressure was measured in an expanded flow-channel section immediately downstream from the exit calming length. The expansion pressure recovery compensated, to some degree, for the frictional pressure drop in the exit section. (Note: A possible pressure uncertainty of +5 to +10% at the exit of the heated length does not seriously affect the other measured and calculated values.)

F. Procedure

Operational procedure consisted of setting the desired values of mass velocity and heat flux in the vertical test section at a constantly maintained discharge pressure of 14.1 kg/cm^2 (200 psia). The coolant inlet subcooling was then slowly and continuously decreased from the initial stabilization level until the critical heat-flux power trip occurred. Throughout each test run, the fixed discharge pressure and fixed throttle-valve setting maintained a constant pressure drop across the tandem arrangement of the flow-metering orifice and test-section assembly. This operational control of constant pressure drop is considered to approximate closely a parallel channel condition, since, for any significant number of parallel channels, any changes in the flow rate in any one channel would affect the total flow rate rather than the driving force of the imposed pressure difference.

Use of a transient inlet subcooling as the operational control permits only one significant variable (local enthalpy) to change with time, and that in a continuous, deliberate, and readily controllable manner. Any resultant changes in other variables of interest are necessarily initiated by the exit enthalpy level and are easily compared thereto.

III. HEAT TRANSFER TO SUBCOOLED WATER

Representative data for clean and corroded heat-transfer surfaces are shown in Figs. 1 to 4. For surfaces at or near the original smooth-drawn surface condition, the convective-transfer data agree well with a simplified form of the Dittus-Boelter⁽²⁾ equation:

$$h = \frac{G^{0.8}}{10^3 D^{0.2}} T_B^{0.5} \quad (1)$$

The development of Eq. (1) is described in Appendix A.

Fig. 1. Forced-convection and Nucleate-boiling Heat Transfer to Liquid Water at 200 psia;
 $Q'' = 1.54 \times 10^6 \text{ Btu}/(\text{hr})(\text{ft}^2)$;
 $G_{\text{avg}} = 11.7 \times 10^6 \text{ lb}/(\text{hr})(\text{ft}^2)$.

Fig. 2. Forced-convection and Nucleate-boiling Heat Transfer to Liquid Water at 200 psia;
 $Q'' = 2.32 \times 10^6 \text{ Btu}/(\text{hr})(\text{ft}^2)$;
 $G_{\text{avg}} = 11.8 \times 10^6 \text{ lb}/(\text{hr})(\text{ft}^2)$.

Fig. 3.
 Forced-convection and Nucleate-boiling Heat Transfer to Liquid Water at 200 psia;
 $Q'' = 3.08 \times 10^6 \text{ lb}/(\text{hr})(\text{ft}^2)$;
 $G_{\text{avg}} = 11.8 \times 10^6 \text{ lb}/(\text{hr})(\text{ft}^2)$.

Fig. 4.

Nucleate-boiling Heat Transfer to Liquid Water
at 200 psia; $Q'' = 3.86 \times 10^6 \text{ Btu/(hr)(ft}^2\text{)}$.

For the same surface condition, the data range up to 30% higher than the nucleate-boiling transfer surface superheat predicted by a revised form⁽³⁾ of the Jens-Lottes⁽⁴⁾ equation:

$$T_w - T_{\text{sat}} = 0.18 \times 10^6 \sigma (Q''/10^6)^{0.25} \quad , \quad (2)$$

where

$$\sigma = (500 - 0.707T) 10^{-6} \text{ lb}_f/\text{in.} \quad . \quad (3)$$

The smooth transition from convective heating to simple nucleate boiling appears to be characteristic for smooth surfaces, and occurs at lower values of subcooling than predicted by Eqs. (1) and (2).

For the case of the flow channels which had been repeatedly subjected to high-power critical-flux operation, the resulting corrosion of the transfer surface caused predictably higher metal temperatures in the region of convective heating. Transition to the nucleate-boiling regime was abrupt: the characteristic temperature "overshoot" decreased precipitously to a relatively stable level approximately 20% below the superheat values predicted by Eq. (2). [The surface conditions cited represent rather extreme examples of nucleation capability, while Eq. (2) defines the mean value for "average" transfer surfaces.] Due to the increased temperatures of the corroded transfer surface in the convection-heating regime, this eruptive initiation of nucleate boiling occurred at higher values of bulk subcooling than those computed from Eqs. (1) and (2).

Also shown in Figs 1 to 4 are companion plots of the coolant temperature rise. For the test conditions of constant heat flux and constant imposed pressure drop these values are proportional to the coolant mass velocity. The generally insignificant changes in flow indicate that the frictional increase due to the subcooled nucleate-boiling turbulence is largely counterbalanced by compensating changes in the average coolant density and buoyant driving forces. Also to be considered is the small percentage of the total channel length in nucleate boiling and the probability that the boiling frictional loss occurs as an additive (rather than multiplying) factor to the predominant velocity frictional loss component.

Figure 5 presents a comparison of nucleate-boiling wall-superheat data for smooth and corroded transfer surfaces. The trends are very similar to the data of Berenson⁽⁵⁾ for saturated pool boiling of organic coolants.

Fig. 5.
Nucleate-boiling Surface-superheat Data for Smooth
and for Corroded Tube Surfaces.

IV. CRITICAL HEAT-FLUX AND HYDRODYNAMIC INSTABILITY

The subcooled critical heat-flux data shown in Fig. 6 follow the classic behavior of a precipitate and highly localized change from the

Fig. 6. Critical Heat-flux Occurrence during Froth Flow at 200 psia.

nucleate-boiling boundary layer to the film-boiling regime. As the liquid boundary layer vaporizes, its abrupt radial expansion results in the radical flow throttling shown in Fig. 7, and an equally abrupt excursion of the transfer-surface temperature. The data are compared in Fig. 6 with a previously developed correlation⁽³⁾ for the froth flow regime, showing an expected departure from the correlation with decreasing subcooling. Due to the low values of subcooling, the evidence of a surface effect similar to that of the nucleate-boiling wall superheat is minor and inconclusive.

Shortly before the subcooled critical heat-flux occurrence, there was a small, progressive decrease in mass velocity and a concurrent progressive increase in the transfer-surface superheat. This departure from the characteristic, simple-nucleate-boiling behavior is considered as the initiation of froth flow, one of several hypothesized subcooled bubble-flow regimes. This flow pattern is believed to be limited to low system pressures or small flow-geometries, the unquenched nucleate bubbles preempting the flow-stream cross section with high volumetric void fractions. As indicated by the flow traces, and also as reported by Gunther,⁽⁶⁾ this flow pattern precedes the subcooled critical heat-flux occurrence, being the most probable precipitating cause thereof.

Unless interrupted by the critical heat-flux occurrence, the increasing concentration of relatively large bubbles which constitutes the froth-flow regime eventually coalesce to form a vapor-phase flow stream. Depending upon the local rate of bubble accumulation in the flow stream, this coalescence may be abrupt and intermittent, resulting in slug flow or a gradual transition to annular flow. An example of slug-flow formation is reflected in the representative flow trace of Fig. 8. The increasing concentration of unquenched bubbles causes a progressive flow decrease and a shift in the exit enthalpy from subcooled to net steam generation. This gradual establishment of froth flow is terminated by the abrupt flow transients of the slug-flow regime, the oscillation limits of which can be bounded by smooth extrapolations from the froth-flow and annular-flow traces.

Concurrent with the flow oscillations, minor oscillatory increases in the transfer-surface superheat (of 5 - 15°F) were recorded; the higher temperature level coincided with the lower limit of the flow. This elevated temperature was maintained during the brief period of relatively stable annular flow, and decreased slightly with the small flow increase preceding the critical heat-flux occurrence. These modest increases in the transfer-surface superheat indicate that the vapor-phase flow stream, whether occurring intermittently in slug flow or continuously in annular flow, does not necessarily affect the nucleate-boiling boundary layer. The reduced effect of the nucleate-boiling turbulence acting upon a vapor-phase flow stream rather than on a bubble-laden liquid agglomerate seems obvious.

Fig. 7. Typical Flow Excursions Resulting from Critical Heat-flux Occurrence in Subcooled Froth Flow.

Fig. 8. Froth-flow, Periodic Vapor-slugging, and Stable Annular Flow during Net Steam Generation.

The duration of the upward flow excursion which accompanies the critical heat-flux occurrence (Fig. 8) is several times the total response time of the circuitry for burnout control. This, plus the high heat flux and low thermal storage capacity of the thin-wall test section, indicates that the flow excursion is the hydrodynamic cause of the critical heat-flux occurrence and not a resulting phenomenon.

Figure 9 shows flow traces at power levels below those of Fig. 8. The stable annular-flow regime is established with minor or no evidence of vapor slugging. The reduced rate of bubble production presumably permits more gradual and continuous bubble coalescence. The flow throttling resulting from the progressive establishment of froth and annular flow is again accompanied by an increasing transfer-surface superheat, the consistently parallel behavior of flow and wall temperature demonstrating a definite pattern of cause and effect. The upward flow excursion accompanying the critical heat-flux occurrence was observed without exception in the annular- and slug-flow regimes, apparently being a characteristic of these flow patterns. No explanation of the mechanics of this sudden release of the flow throttling can be offered at this time.

The high (~ 110 psi) and constant pressure drop across the entire test section permitted operation with subcooled inlet temperatures appreciably above (30°F max) the exit saturation temperature. The principal effects of the resulting vapor flashing were increased local steam quality and accelerated establishment of the annular-flow regime. No contribution to flow instability or transfer-surface superheat was observed, indicating that the emerging or established flow patterns were undisturbed. Except for two identical runs, a few seconds duration of rapid, low-amplitude flow oscillation preceded, and was terminated by, the critical heat-flux occurrence. For the two exceptions noted, the developing annular flow was interrupted by a flow instability of greater amplitude and duration (see Fig. 10). Tentative classification of this flow pattern as slug flow is based primarily upon the higher mass velocity at which it occurs and the higher frequency of oscillation.

A more readily defined illustration of slug flow (see Fig. 11) was encountered at the lowest operational power level [$0.77 \times 10^6 \text{ Btu}/(\text{hr})(\text{ft}^2)$]. The inception of nucleate boiling, net steam generation, and vapor flashing were closely successive events, the flashing possibly determining a random establishment of flow pattern. Of three initially similar runs, one progressed from slug flow to stable annular flow; one "burned out" during the slug flow shown in Fig. 11; and the third (shown in Fig. 12) did not progress beyond stable froth flow. This single example of stable froth flow is characterized by the moderate (25%) flow throttling and the absence of any sizable flow excursion accompanying the critical heat-flux occurrence.

Fig. 9. Development of Stable Annular Flow and Terminal Flow Instability in a Vertical Boiling System with Vapor Flashing.

ANL 112-2758

Fig. 10. Possible Slug Flow Preceding and Causing Critical Heat-flux Occurrence at 200 psia and $Q'' = 1.54 \times 10^6 \text{ Btu}/(\text{hr})(\text{ft}^2)$, with Vapor Flashing.

ANL 112-2759

Fig. 11. Characteristic Flow Trace of Vapor-slugging Condition Preceding Critical Heat-flux Occurrence at 200 psia and $Q'' = 0.77 \times 10^8$ Btu/(hr)(ft²), with Vapor Flashing.

ANL 112-2757

Fig. 12. Critical Heat-flux Occurrence during Stable Froth Flow at 200 psia, with Vapor Flashing.

The terminal instability threshold preceding the critical heat-flux occurrence in annular and slug flow at several power levels is shown in Fig. 13. The distinguishing difference between the two regimes appears as a difference in mass velocity. Only at the highest power level does the terminal limit of flow stability coincide with the critical heat-flux occurrence. The locus of the data points defines a limiting hydrodynamic criterion with undefined pressure and geometry factors, the independence of heat flux over a sizable range indicating that the relationship is not simply thermodynamic.

The effect of flow stability upon the critical heat-flux occurrence can be seen in Fig. 14: the stable and unstable flow regimes differ sharply in trend as well as in magnitude. The critical heat-flux increase with increasing steam quality appears to be characteristic of, and generally peculiar to, flow instability. All the evidence presented indicates a preemptive hydrodynamic crisis in which the local heat flux is a requisite rather than a decisive factor. In unstable flow, it is difficult to determine the instantaneous mass velocity and the derivative steam-quality value. However, use of the vapor mass-velocity term (Gx) combines the compensatory uncertainties of both terms into a value of minimum uncertainty, assuming that the heat flux and inlet enthalpy are reliably known. Although it is beyond the scope of this discussion, the approach illustrated in Fig. 14 can be extended into a generalized criterion for the terminal hydrodynamic instability in axial-flow forced-convection boiling systems.

Fig. 13. Terminal Limit of Flow Stability Preceding Critical Heat-flux Occurrence during Annular and Slug Flow at 200 psia.

Fig. 14. Effect of Flow Regime and Degree of Hydrodynamic Instability upon Critical Heat-flux Occurrence during Net Steam Generation at 200 psia.

V. DISCUSSION AND CONCLUSIONS

As anticipated, either Eq. (1) or the original Dittus-Boelter⁽¹⁾ relationship was found to be satisfactory for prediction of values of convective heat transfer for clean surfaces. In the simple nucleate-boiling regime (nucleate bubbles rapidly quenched by the subcooled flow stream), Eq. (2) is recommended for the transfer surface temperatures for "average" surfaces and, in combination with Eq. (1), for determination of the inception of simple nucleate boiling. As pointed out in Section IV, surface-dependent deviations should be expected. Despite the low pressure and small diameter, it was found that the occurrence of simple nucleate boiling over short lengths of the vertical tube caused insignificant reductions in the high mass velocities. An increasing percentage of flow reduction should be expected as the mass velocity values are decreased.

For the subcooled enthalpy design conditions of AARR, the critical heat-flux equations shown in Fig. 6 are recommended; local subcoolings of less than 10°F should be avoided because of the strong probability of progressive flow instability.

Two-phase flow begins at low values of bulk subcooling, the transfer-surface superheat data showing a renewed influence of mass velocity in the froth-flow and annular-flow regimes. In froth flow, unlike other hypothesized bubble-flow regimes,⁽³⁾ the transfer-surface superheat increases with decreasing mass velocity. The same behavior is also evident in the annular-flow regime.

The transfer-surface superheat is affected to a greater extent by the flow-stream vapor content, the effect of the nucleate-boiling turbulence progressively decreasing as the vapor phase increases. For purposes of comparison, at constant pressure and heat flux, the inverse transfer-surface superheat can be considered a measure of the nucleate-boiling efficiency. On this basis, the efficiency is greatest in simple nucleate boiling, decreases in the froth-flow regime, and decreases still further in annular flow.

Changes in the internal flow resistance of the boiling channel parallel the changes in boiling efficiency and mass velocity effect - nonexistent or insignificant in the simple nucleate-boiling regime and increasing with increasing vapor content until stable annular flow is established. The progressively increasing length of flow channel in the several boiling regimes undoubtedly accentuates the effect, but the stable mass velocity of the annular-flow regime (see Fig. 9) indicates that the increasing boiling length is not the primary cause of the increased flow resistance.

The initiation of slug flow apparently requires some threshold rate of bubble accumulation in the flow stream, the flow pattern being either an intermittent or continuous oscillatory transition from the froth-flow to the annular-flow regime. The modest increases in transfer-surface superheat which prevailed during the formation of both the vapor slugs and the continuous

vapor stream of annular flow strongly indicate that the process was confined to the flow stream without materially affecting the nucleate-boiling boundary layer. No generalization of this statement is implied or recommended.

The limiting values of both stable and unstable annular flow conform to a well-defined linear mass velocity-steam quality relationship which is independent of heat flux over a wide range. This indicates that the critical heat-flux occurrence in annular flow results from a hydrodynamic crisis and is not an originating cause in itself.

The abrupt flow throttling caused by the critical heat-flux occurrence in incipient or developing froth flow is due to the large volumetric expansion of the flashing boundary layer. In stable froth flow, the flashing boundary layer supplements an already high volumetric void fraction, and no appreciable flow excursion (see Fig. 12) results from the critical heat-flux occurrence. In fully developed annular flow, an unexplained release of the accumulated flow resistance results in an abrupt and protracted upward flow excursion, the duration of which indicates that it precedes and causes the critical heat-flux occurrence. This hydrodynamic phenomenon was observed without exception for stable and unstable annular flow and, with somewhat less certainty of resolution, in the critical heat-flux occurrences in the slug-flow regime.

Stable annular flow with increasing steam quality characteristically terminated in a low-level flow instability, the succeeding upward flow excursion resulting in the critical heat-flux occurrence. (Pre-emptive interruption of this process by the critical heat-flux occurrence, as illustrated in Fig. 8, does not invalidate the statement.) The terminal flow instability causes uncertainties in the values of mass velocity and steam quality, but these uncertainties can be nullified by combining the terms into a vapor-phase mass velocity (Gx). Increases in the (Gx) term result largely from increasing steam quality (by boiling and/or vapor flashing), and the positive slope of the critical heat flux-steam mass velocity relationship (see Fig. 14), which is characteristic of unstable annular flow, is directly counter to the negative slope typical for stable flow. Since the slug-flow data follow a similar behavior, this characteristic may be applied to distinguish between stable and unstable flow conditions in systems using high-damping, slow-response secondary flowmeters, such as mercury manometers.

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APPENDIX A

Development of Simplified Forms of the Dittus-Boelter Equation

Derived from a dimensional analysis, the dimensionless Dittus-Boelter⁽²⁾ equation:

$$N_{Nu} = C (R_{Re})^m (N_{Pr})^n ,$$

is the accepted standard for computation of the forced-convection heat transfer coefficient. This equation relates the flow velocity and characteristic physical dimensions of the system to the significant physical properties of the coolant. It may be rearranged to the form

$$\frac{h(D)^{1-m}}{(G)^m} = C \frac{k}{(\mu)^m} (N_{Pr})^n .$$

Any consistent set of units may be employed with the most widely accepted values of the empirical constants:

$$C = 0.022; m = 0.8; n = 0.4 .$$

As rearranged, the right side of the equation contains only the coefficient C and a grouping of physical properties of the coolant. For liquid water, these properties are temperature dependent and insensitive to system pressure. Graphical comparison of the values of this grouping of liquid water properties with the associated temperatures shows that the values vary as the square root of the Fahrenheit temperature over the range from 50°F to 670°F, using property values for both 2000 psia and the saturation condition. Similar comparisons, using the Centigrade temperature scale and the saturated enthalpy, show that the values of the physical properties vary as the 0.4 power of these terms over the ranges from 20°C to 350°C and from 40 Btu/lb - 730 Btu/lb. Pressurewise, this spans the range from atmospheric to 2500 psia for liquid water.

Proceeding from the above, the following relations can be written for liquid water:

$$\begin{aligned} \frac{h(D)^{0.2}}{(G)^{0.8}} &= 0.022 \frac{k(N_{Pr})^{0.4}}{(\mu)^{0.8}} \\ &= 10^{-3} (T_B)^{0.5} \quad 50^\circ\text{F} \leq T_B \leq 670^\circ\text{F} \end{aligned} \quad (\text{A-1})$$

$$= 2.40 \times 10^{-3} (T_B)^{0.4} \quad 20^\circ\text{C} \leq T_B \leq 350^\circ\text{C} \quad (\text{A-2})$$

$$= 1.85 \times 10^{-3} (H_f)^{0.4} \quad 40 \text{ Btu/lb} \leq H_f \leq 730 \text{ Btu/lb.} \quad (\text{A-3})$$

For all cases, the agreement is within $\pm 5\%$, which compares well with the combined accuracy of the property values used.* Attention is called to the fact that the coefficients and factors are no longer dimensionless:

$$h = \text{Btu}/(\text{hr})(\text{ft}^2)(^\circ\text{F}); \quad D = \text{ft}; \quad G = \text{lb}/(\text{hr})(\text{ft}^2) \quad .$$

Error plots of Eqs. (A-1), (A-2), and (A-3), referenced to the original Dittus-Boelter equation, are shown in Figs. A-1, A-2, and A-3.

Fig. A-1. Comparison of Original with Simplified Dittus-Boelter Equation, on the Fahrenheit Scale, for Liquid Water.

Fig. A-2. Comparison of Original with Simplified Dittus-Boelter Equation, on the Centigrade Scale, for Liquid Water.

Fig. A-3.

Comparison of Original with Simplified Dittus-Boelter Equation as a Function of the Saturated Enthalpy of Liquid Water.

* Property values of E. J. Wellman, Reactor Handbook, AECD-3646 (May 1955), Vol. 2, Chap. 1.3, Tables 1.3.3, 1.3.5, 1.3.6, and 1.3.8.

APPENDIX B

Critical Heat-flux Data

Type 304 Stainless Steel Vertical Tube: 0.045-in. ID x 0.010-in. wall
x 4.5 in. long; 200 psia

Run Number	Q_C'' , $10^6 \text{ Btu}/(\text{hr})(\text{ft}^2)$	G , ^a $10^6 \text{ lb}/(\text{hr})(\text{ft}^2)$	$(H-H_f)$, Btu/lb	Hypothesized Local Flow Regime
9-25	6.79	11.3	-28.5	Froth Flow ↓
9-8	6.16	11.5 ^c	-24.5	
9-20	6.16	11.4	-26.5	
9-6	5.40	11.4	-20	
9-7	5.40	11.2	-16	
9-26	5.40	10.9	-21	
9-3	4.62	10.1	-17	
9-4	4.62	11.1	-20	
9-5	4.62	11.0	-16	
9-27	4.62	11.0	-19	
9-28	4.62	11.2	-18	
9-1b	3.86	10.9	-17	
9-2b	3.86	11.1	-17	
9-29	3.86	11.5	-10.5	
7-10	3.86	10.5	-7.5	
7-11	3.86	11.0 ^c	-12.5	
7-12	3.86	11.1 ^c	-12.5	
7-8	3.86	10.5	-10.5	
7-9	3.86	10.8	-10.5	
9-11	3.08	11.3	-7.5	
9-30	3.08	11.1	-9.5	
7-13	3.08	10.9 ^c	-7.5	
7-14	3.08	10.9	-7.5	
7-6b	3.08	10.8	-12.5	
7-7b	3.08	11.1	-12.5	
8-3	3.08	11.1	-7.5	
9-12	2.32	3.82	177	Stable Annular Flow
9-31	2.32	11.6	-5	Froth Flow
9-32	2.32	3.33	202	Stable Annular Flow
7-17	2.32	3.22	224	↓
7-16	2.32	3.33	208	
7-15	2.32	3.37	204	↓
7-5b	2.32	3.44	215	
8-2b	2.32	11.4	-4	Froth Flow
6-2b	2.32	6.6	-12.5	↓
6-3b	2.32	4.8 ^d	-22.5	
9-24	1.93	2.74	278	Unstable Annular Flow ^e
5-2b	1.85	5.0	-16	Froth Flow
4-6	1.85	6.3	-4.5	↓
6-1b	1.85	6.6	-12.5	
1-7	1.85	6.3	-8.5	

Run Number	Q_C'' , $10^6 \text{ Btu}/(\text{hr})(\text{ft}^2)$	G , ^a $10^6 \text{ lb}/(\text{hr})(\text{ft}^2)$	$(H-H_f)$, Btu/lb	Hypothesized Local Flow Regime
9-13	1.54	3.46	191	Unstable Annular Flow ^e
9-18	1.54	5.74	142	Slug Flow(?) ^e
9-23	1.54	5.75	141	Slug Flow(?) ^e
9-33	1.54	2.93	247.5	Unstable Annular Flow ^e
7-18	1.54	3.18	206	↓ Froth Flow
7-19	1.54	3.28	199	
8-1 ^b	1.54	3.97	149	↓ Unstable Annular Flow ^e
2-2 ^b	1.41	5.85	-7.5	
1-5	1.35	6.85	-5	↓ Unidentified
1-6	1.35	6.80	-15	
9-14	1.15	4.95	114	↓ Stable Froth Flow ^e Slug Flow ^e
9-15	1.15	4.01	146	
9-21	1.15	3.94	144	↓ Unstable Annular Flow ^e
9-34	1.15	4.03	131	
1-2 ^b	0.950	4.36	101	↓ Unstable Annular Flow ^e
1-3 ^b	0.941	5.05	88.5	
1-4 ^b	0.926	3.98	124.5	↓ Unstable Annular Flow ^e
3-2 ^b	0.925	3.82	134	
9-16	0.771	8.85	61.5	↓ Unstable Annular Flow ^e
9-17	0.771	7.31	68.5	
9-22	0.771	4.52	68	Unstable Annular Flow ^e

^a Value at the critical heat-flux occurrence or at the initiation of terminal flow instability. (Test sections 1-4, no flow traces; values are final reading on mercury manometer.)

^b Clean, smooth heat-transfer surface.

^c Continuous low-level flow instability in subcooled froth flow.

^d Terminal flow instability during subcooled froth flow.

^e Vapor flashing in heated section.