SUBCOOLED CRITICAL HEAT FLUX FOR WATER IN ROUND TUBES

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ABSTRACT

A detailed examination of the literature pertaining to subcooled critical heat flux was conducted. Subcooling, system pressure, mass flow rate, system geometry and flow stability were determined to be the important parameters in determining critical heat flux. The effect of each of these is discussed in detail.

Various schemes for correlating experimental critical heat flux were subjected to critical examination. Form, content and comparison with experimental data were the bases for evaluation. The correlation of Griffith was found to compare most closely with the experimental data over a broad range of variables. A linearized correlating equation was developed for the MIT low pressure (30 to 90 psia) data, including that determined as part of this study.

Critical heat flux was determined experimentally for pressures from 30 to 90 psia, mass flow rates of 3.69×10^6 to 14.76×10^6 lbm/ft^2-hr and diameters of 0.047 to 0.242 inches over the subcooled region. The experimental data obtained is in general agreement with that of other investigations. This low pressure data, when combined with the results of previous investigations, provides a fairly complete picture of subcooled critical heat flux for tubes up to 0.25 inch diameter. A graphical interpolation scheme is presented by which it is possible to obtain predictions of critical heat flux which are generally superior to that of the mathematical correlations.

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I. INTRODUCTION

In the continuing search for devices of higher energy flux density, more and more emphasis is being placed upon boiling as a means of increasing heat transfer rates with moderate temperature differences. Current developments in high energy electromagnets, high performance electron tubes and rocket motors require very high heat fluxes in cooling applications. Although heat fluxes well in excess of One Million Btu per square foot per hour can be obtained, heat fluxes cannot be increased without limit. For a given set of flow conditions, there is a limiting Critical Heat Flux (CHF), at which the transition from nucleate to film boiling occurs. At this point, the wall temperature increases dramatically, usually resulting in catastrophic thermo-mechanical failure of the heat transfer surface. One also finds references in the literature to this as "burnout", and "the boiling crisis".

Often, the most desirable mode of boiling for heat transfer applications is that of surface, or subcooled boiling, where the average or bulk fluid temperature remains below that of saturation. In this case, vapor formed at the surface condenses when it comes in contact with the cooler main stream. The Pressurized Water Reactor uses water at high pressure (2000 psi) as coolant and moderator. The high pressure allows sufficient coolant temperatures for further heat transfer and power generation, while maintaining bulk temperatures below saturation. The maintenance of subcooling is important in reducing void fraction and retaining continuity of the moderator. The electromagnets and electron tubes are generally cooled by low pressure systems where the expense and complications of high system pressures can be avoided. Utilization of subcooled boiling eliminates the complexities of two-phase flow, as well as the mechanical complication of condensing equipment; however, the CHF problem is still present in subcooled boiling.

Over the past several years, numerous studies have been concerned with this CHF problem. A great deal of experimental data is available, under various conditions of flow. Investigators have usually selected certain parameters and investigated the effect of varying these within certain limits. The majority of the previous experimentation has been conducted with uniform heat flux and round tubes. This investigation has been limited to this region.

The first phase of this investigation consisted of a search of the literature to determine the flow conditions which had been investigated. Techniques, generally in the form of correlating equations, have been proposed for predicting CHF, by almost everyone who has conducted investigation in this field. One phase of this study has been concerned with a critical examination of the techniques and their form and agreement with available experimental data.

Another phase of this investigation consisted of completion of some previous experimental work. Experiments were conducted at selected tube diameters, at low pressures and various flow rates. Data thus obtained, when combined with that in the literature, provides a picture of CHF, for tubes of small diameter and flow rates of interest, over the complete range of pressures, from 30 psia to just below critical pressure.

II. PROCEDURE

This was a two part study; a survey of the current literature and correlating equations and experimental determination of Critical Heat Flux (CHF) under certain conditions.

First, citations of all of the current literature concerned with Subcooled Critical Heat Flux were obtained. From these, the majority of the literature was reviewed, and experimental data and proposed correlating equations were recorded for further investigation. A large part of the work in this field has been accomplished in Russia, consequently certain information is available only in the Russian Journals, untranslated. It was usually possible to extract data and proposed correlating equations without complete translation of the particular paper.

A listing of proposed correlating equations was prepared. These were compared with actual experimental data from various sources, including the experimental phase of this study, in an attempt to determine the validity of the correlation schemes.

Experiments were conducted to determine CHF for predetermined values of the important variables. The dependence of CHF upon subcooling was determined for three fixed values of mass flow rate, with constant system pressure, tube length and diameter. This data completes the low pressure end of the study of Ornatski and Kichigin (4, 5 and 6). Then, using one of the same flow rates and pressures, and holding exit subcooling constant, dependence of CHF upon tube diameter, and upon tube length-diameter ratio was determined. Next, the variation of CHF with changing entrance forms, with the other variables held constant, was investigated. Finally, tube diameter was changed to obtain comparison data, and to determine the maximum CHF available at the larger diameter with the experimental apparatus.

All of the experimental data relating to subcooled critical heat flux that has been obtained within recent years at MIT was collected, converted to similar format and placed on machine data cards in anticipation of future studies.

A detailed description of the experimental apparatus and procedures employed is contained in Appendix D and E.

III. RESULTS

FACTORS INFLUENCING CRITICAL HEAT FLUX

Although there has been a great deal of research and investigation in this field, particulary in the last decade, there exists today no clearly defined, universal, mathematical description of the process which results in a limiting CHF. Pool Boiling is itself a complex problem, the additional conditions of subcooled, forced flow tending to cloud the picture even more. Attempts at dimensional analysis, and treatment of the problem from basic equations have, in the main, produced little information, usually yielding long, complicated functional relationships and involving simplifying assumptions not necessarily borne out by experiment.

It is generally accepted, however, that subcooling, system pressure, mass flow rate, flow geometry and stability are the major factors in determining CHF. These variables evidence various degrees of interdependence, and in discussing the effects of each, this interdependence must be borne in mind.

Effect of Subcooling

Subcooling is essentially a measure of the amount of pre-heating required to obtain boiling. This is generally measured as the difference between outlet bulk temperature, or more precisely enthalpy, and that at saturated conditions. This is reasonable for uniform heating, in that failure normally occurs quite near the downstream end of the heated section. From another viewpoint, subcooling is a measure of the cooling effect of the main stream. Thus CHF increases with subcooling. There is a tendency in the literature to treat CHF as linear with subcooling. This is not the case, but it is in general, not a grossly inaccurate approximation. In the region of low values of subcooling the data shows a distinct non-linearity, as a minimum value of CHF is reached.

Effect of System Pressure

System pressure plays an important role in determining CHF, although the effect is less dramatic than in Pool Boiling. In a recent DuPont study, it is stated that a variation of system pressure from 60 psia to 1000 psia has little significant effect on CHF: this in tubes ranging from 0.3 inch to 0.89 inch in diameter. However, this is not borne out by the other experimental data (9). A pressure increase from 600 psia to 1000 psia results in a reduction of CHF of as much as 25%. Further increases in pressure above 1000 psia result in further reductions in CHF. At lower pressures there is disagreement over the effect of pressure. McAdams (32) and Gunther (33) report no effect up to 163 psia, while Mirshak, et al. (31) show a pressure dependence over the entire pressure range considered. Bergles (11), reports a 15% increase in CHF, for an increase in pressure from 20 to 85 psia. Chirkin and Iukin (24) demonstrate that increasing pressure, in the low pressure region, increases CHF. Their pressure effect is much stronger than that of other investigators, however, it is now felt that most of their data suffered from instability problems which could have clouded the real picture.

The experimental data does suggest, however, that for fixed geometry, flow rate and subcooling under conditions of stable flow, there is some pressure for which CHF attains a maximum value. The pressure at which this maximum value of CHF occurs is not clearly defined, being dependent upon mass flow rate, and possibly the other variables as well. It is safe to say though, that under conditions of reasonable flow rate, this maximum occurs somewhere between 400 and 600 psia. Unfortunately data in this region is scarce.

There is a tendency noted in the Russian papers to correlate data with straight lines whenever at all possible. In discussing the variation of CHF with diameter, Doroshchuck and Lantsman (9), have done this. However, a careful investigation of the data of this study shows that the variation of CHF with exit subcooling is indeed not linear, especially at the very high pressures. It is found, that at these high pressures CHF becomes less dependent upon subcooling as subcooling is increased. This suggests that at pressures near critical, and with very high values of subcooling (Δh_{sat} greater than 200 BTU/lb), CHF becomes somewhat independent of subcooling. Other high pressure data shows this same general trend (4, 5 and 6). This would, of course, require further investigation to be throughly substantiated.

<u>Effect of Mass Flow Rate</u>

Generally, in the subcooled region, increasing mass flow rate results in an increased CHF. This is easily visualized since the increased flow sweeps the bubbles away from the heat transfer surface more rapidly, replacing them with relatively cold water. However, the rate of increase in CHF due to increased flow rate is pressure dependent. At high pressures, the effect of increased mass flow rate becomes less pronounced.

<u>Effect of Geometry</u>

The majority of the research and study in this field has been limited to round tubes and annular fluid passages. The gross effects of the other variables however, are much the same for any cross section (8). In general, for given flow rate, pressure and subcooling, CHF is greatest for a circular tube and successively lower as the geometry becomes more complex. "Swirl flow" is one exception to this. The ratio of heated perimeter to total wetted perimeter has been suggested as important in the case of "swirl flow" (12).

Investigators disagree whether there is a range where diameter has no effect. Doroshchuck and Lantsman in a recent study (9), conducted tests for four tube diameters over most of the high pressure range, (735 psia to 2500 psia), with exit conditions from high subcooling (Δh_{sat}=200 BTU/lb_m) to bulk boiling (x_{exit}=0.2). This study showed CHF decreasing consistently with increasing channel diameter. Earlier investigations show decreasing CHF with decreasing diameter. Both Chirkin and Iukin (24) and Kafenhaus and Bocharov (26) report this. Again, this is now felt to have been in a large part due to flow instabilities in their systems. DuPont (8), Mirshak (3) and Zenkevich (14) note no diameter effect. Bergles (11) notes a strong inverse dependence of CHF on diameter for small diameter tubes, but he shows the Zenkevich Prediction (14) as a possible asymptote for diameters greater than 0.3 inches.

With flow rates high enough to eliminate the effects of natural convection within the passage, channel orientation has little effect on CHF.

The effect of heated length has not been completely determined. In general, the length, or length-diameter ratio, is not important if the length is long, or L/D is large. The popular conception is that CHF is determined by purely local phenomena, thus as long as the flow is fully developed and entrance effects play no part, the length is not significant. But the question of how long is long enough has not been satisfactorily resolved, partly because of the interaction of the variables. In general, the Russian studies consider an L/D ratio of 10 to be large enough to eliminate the length effect. This conclusion has not been completly verified. Bergles (11), in a low pressure study, showed an L/D dependence existing for L/D on the order of 40. Doroshchuck and Fried (18), present data for tubes of 0.12 inch diameter at 1500 psia that shows no difference in CHF for L/D of 14, 50 and 100, over a fairly braod range of exit conditions. DuPont (8) suggests consideration of the temperature gradient down the heated tube. The temperature difference is of course dictated by the First Law of Thermodynamics, so what this gradient really reflects is a length-diameter consideration.

In general, CHF decreases for increasing L/D, but for high pressures, the dependence of CHF upon L/D becomes less significant for values of L/D greater than 30.

Effect of Flow Stability

This refers to pulsations in flow and pressure, resulting from compressibility effects in the heated sections. If there is an expander or accumulator upsteam of the heated section, there is a distinct possibility of producing a flow oscillation which will periodically reduce the flow near the tube outlet and perhaps cause a premature failure of the heat transfer surface. Bergles (11) points out that there are two types of instability: flow oscillation due to compressibility and flow excursion due to system characteristics. Flow conditions under which these pulsations do effect CHF are considered unstable. The result of unstable flow is a drastic reduction in CHF, with unstable flow resulting in heat fluxes as low as 20% of the stable value. Stable flow usually can be insured for subcooled boiling by installing some sort of throttling device (orifice, valve, etc.) in the system just up-stream of the heated section. Doroshchuck and Lantsman used a pressure drop of 470 psi in their high pressure studies (9, 10). Bergles (11) reported a pressure drop of approximately 185 psi in his low pressure work.

Further discussions of the stability problem are contained in Ref. 7, 11 and 28. The stability problem is particulary acute in systems containing heat exchangers with parallel tubes from common inlet and exit headers, where a substantial amount of compressible volume may exist. In any event, the installation of a pressure reducing device, and the resultant loss of downstream system pressure, is more than justified by the increase in CHF.

<u>Effect of Secondary Factors</u>

Several other factors have been investigated to determine how they effect CHF.

Surface roughness, for reasonably smooth, (commercial quality) channels has little or no effect (7, 14).

Dissolved gas in the flow stream, in small amounts, has a negligibly small effect on CHF (17).

For heat flux unevenly distributed around the heated perimeter, peak heat flux was found to increase. For heat flux unevenly distributed along the length of the heated channel, qualitatively similar results have been obtained (7). Tests have been conducted under conditions where the heat flux was forced into a cosine distribution along the tube and CHF was shown to be reduced by some 30% (11).

CORRELATING EQUATIONS

General Discussion

Numerous schemes have been proposed as correlating CHF data, and by which CHF can supposedly be predicted. The earliest of these are of a power function form:

$$(q/A)_{cr} = A \cdot B^c D^e E^f$$

where B, D and F are the important variables, or combinations of them, and c, e and g are experimentally determined exponents. Usually correlations of this type tend to neglet the interaction between variables; in other cases the exponents themselves are functions of the controlling parameters. Another broad category includes the "superposition" correlations, those combining pool boiling data with forced convection heat transfer information. These correlations are often long and difficult to use.

Several attempts have been made at analytical treatment, Zenkevich (14) and Ryabov and Berzina (36) with dimensional analysis and Griffith (13, 17) with a theoretical analysis. Other equations were developed by "brute force" data analysis, using computing machines. DuPont favors a form:

$$(q/A)_{cr} = C\ (1 + C_1 \cdot A)\ (1 + C_2 \cdot B)\ (1 = C_3 \cdot D)$$

where A, B and D are the influencing variables, (pressure, subcooling, diameter, velocity,) and C, C_1, C_2, etc. are constants.

In general, the correlations are limited to specific ranges of the variables, departing radically from the experimental data in the excluded areas. In some cases, one finds correlations developed from a part of an experimental study, using certain data and excluding other. This technique is usually accompanied by a statement to the effect that so many percent of the experimental results agrees within plus or minus so much.

In cases such as this, the potential user is obliged to examine the filtering procedure as closely as he does the correlation itself.

As a result of this state of affairs, one phase of this study was concerned with examination of correlating equations. The listing of correlations is not all inclusive. Several correlations were examined and discarded, generally due to their marked similarity to others.

Of the correlation schemes examined, the following are those which agreed reasonably well with the experimental data, or illustrate some point.

Details of Particular Equations

Bell: (ref: 21)

$$(q/A)_{cr} = 0.427 \times 10^6 \times \left(2.59 + \frac{G}{10^6}\right)^{1.81} \times \left(\frac{h_g - h_b}{10^3}\right)^{1.77} \times \left(\frac{G}{10^6}\right)^{.406}$$

Range of values: $(q/A)_{cr}$ = Btu/hr-ft^2; p = 2000 psia; G = $.2 \times 10^6$ - 5.0×10^6 lbm/hr-ft^2; Δh_{sat} = 0 - 130 Btu/lbm

This correlation is compared with experimental data (figure 9) for conditions very nearly identical, from three different sources. In obtaining this data from the various references, a certain variation was unavoidable; two people do not conduct tests at precisely the same conditions. However, the variation was kept to less than the expected experimental error. The data from reference 15 was obtained in 1954, prior to the general acceptance of the need for stability, and may be of questionable value.

It can be seen that the correlation follows the general trend of the data, with variations of 15% to 30%.

Jens & Lottes: (ref: 17)

This correlation was based on data taken at UCLA and Purdue over a range of pressures from 500 - 3000 psia. The values of K and n were determined on the basis of the data such to make all deviations positive.

$$(q/A)_{cr} = K\,(G/10^6)^n \cdot (\Delta T_{sat})^{.22}$$

UCLA Data correlated to -0%, +23%, D = .26 in.

p	K	n
500	.817	.16
1000	.626	.275
2000	.445	.50

Purdue Data correlated to -0%, +60%, D = .143 in.

p	K	n
1000	.915	.275
2000	.545	.50
3000	.30	.725

The correlation was plotted for data (figure 9) of approximately D = .180 in., G = 2.1 x 10^6 lbm/hr-ft^2, at p = 2000 psia using the constants and exponents of the Purdue data.

Ryabov and Berzina: (ref: 36)

1) $$(q/A)_{cr} = (q/A)_{pb} + \frac{403\, V^{.6}\, \Delta T_{sat}^{.67}\, k_s\, T_{sat}^{.33}\, (\rho_l - \rho_v)^{.515}\, \rho_l^{.3}}{(h_{fg}\,\rho_v)^{.33}\, \sigma^{.485}\, g^{.3}}$$

Range of values: D = 2 mm = .079 in., ΔT_{sat} = 18 - 590 deg F, L/D = 20, p = 10 - 200 atmospheres = 147 - 2940 psia, G = 3.69 x 10^6 - 22.14 x 10^6 lbm/hr-ft^2

2) $$(q/A)_{cr} = (q/A)_{pb} + \frac{257\, V^{.6}\, \Delta T_{sat}^{.67}\, k_s\, T_{sat}^{.33}\, (\rho_l - \rho_v)^{.515}\, \rho_l^{.3}}{(h_{fg}\,\rho_v)^{.33}\, \sigma^{.485}\, g^{.3}}$$

Range of values: D ≥ 4 mm (.158 in.), ΔT_{sat} = 18 - 430 deg F, L/D = 10 - 220, V = 3 - 48 ft/sec, p = 70 - 210 atm = 1030 - 3090 psia

Both of the above correlating equations were developed from dimensionless equations. In each case $(q/A)_{pb}$ is the same and is obtained from a plot of pool boiling CHF derived as a function of pressure. The correlating equations were derived from a dimensionless equation:

$$1)\ K_{\Delta q} = 3710\, K_w^{.6}\, K_{\Delta T}^{.67}\, Pr^{-.4} \quad (D = 2mm = .079 \text{ in.})$$

$$\text{or } 2)\ K_{\Delta q} = 2370\, K_w^{.6}\, K_{\Delta T}^{.67}\, Pr^{-.4} \quad (D \geq 4mm \geq .158 \text{ in.})$$

where $$K_{\Delta q} = \frac{(\Delta q/A)_{cr}\, h_{fg}\, \rho_v\, J\, g_o}{k_s\, T_{sat}\, (\rho_l - \rho_v) g}$$

$$K_w = V \sqrt{\frac{\rho_l}{g\sigma \sqrt{\frac{\rho_l - \rho_v}{\sigma}}}}$$

$$K_{\Delta T} = \frac{\Delta T_{sat}\, h_{fg}\, \rho_v\, J\, g_o}{g\, \sigma\, T_{sat} \sqrt{\frac{\rho_l - \rho_v}{\sigma}}}$$

It is assumed that $(q/A)_{cr} = (q/A)_{pb} + (\Delta q/A)_{cr}$

The 2 mm (.079 in.) correlation was derived from the data of Ornatski and Kichigin (4, 5, & 6) by the usual technique of letting the data determine the coefficients and exponents. Generally the data falls within $\pm$ 30% of the correlating line. The correlation was also computed for the 2 mm data taken during the experimental part of this study. In this case the correlation did not follow the basic trends of the data and was in error by as much as 100% at some subcoolings.

The correlation for $D \geq 4$ mm (.158 in.) was computed and plotted for $G = 2.1 \times 10^6$ lbm/hr-ft^2, p = 2000 psia and compared with the data for Doroshchuk and Lantsman's (9) 4 mm and 8 mm tubes. The correlation fit the 8 mm (.316 in.) data very closely , but was 15% - 30% low on the 4 mm data. It was not plotted on figure 9 to avoid confusion as it fell generally on top of the correlation of Zenkevich and Subbotin.

Zenkevich: (ref: 14)

$$(q/A)_{cr} = h_{fg}\left(\frac{G\,\rho_s\,g_o}{\nu}\right)^{\frac{1}{2}} \times \left[2.5 + \frac{184\,\Delta h_{sat}}{h_{fg}}\right] \times 10^{-5}$$

Range of values: $p = 14 - 165$ psia, $\Delta h_{sat} = 20 - 282$ Btu/lbm,

$G = .22 \times 10^6 - 9.96 \times 10^6$ lbm/hr-ft^2, $D = .16 - .50$ in.

This correlation is linear with subcooling as commonly noted for Russian analyses. Specific instructions as to the temperature to evaluate viscosity at are lacking. Using the fluid saturation temperature yields an equation linear in subcooling, while using fluid bulk temperature produces a slight curvature in the resulting plot. This correlation is plotted in figure 8 for both conditions and it can be seen that the equation does not really reflect the experimental results.

Zenkevich and Subbotin: (ref: 14)

$$(q/A)_{cr} = 397\ G^{\frac{1}{2}}\,\Delta T_{sat}^{\ .33}\,(v_g/v_{fg})^{-1.8} \quad \text{Btu/hr-ft}^2$$

Range of values: $\Delta T_{sat} = 18 - 180$ deg F, $D = 4 - 12$ mm $= .158 - .472$ in.

$p = 140 - 210$ atmospheres $= 2060 - 3090$ psia.

This correlation was plotted for $G = 2.1 \times 10^6$ lbm/hr-ft^2 for 2000 psia and 2500 psia and compared with the data of Doroshchuck and Lantsman (9) for tubes of 4 mm (.158 in.) and 8 mm (.316 in.). At 2000 psia the correlation does an almost remarkable job of predicting CHF for D = 8mm. At 2500 psia the correlation is very close to the experimental 8 mm data although approximately 20% low at low subcoolings. The correlation does not fit the 4 mm (.158 in.) data as well, with differences of 15% - 30% at both pressures. Figure 9 shows the correlation at 2000 psia for the 4 mm diameter.

Gambill: (ref: 30)

$$(q/A)_{cr} = K\, h_{fg}\, \rho_v \left(\frac{\sigma g_o\, a\, (\rho_l - \rho_v)}{v^2} \right)^{\frac{1}{4}} \left[1 + \left(\frac{\rho_l}{\rho_v} \right)^{.923} \frac{c_p\, \Delta T_{sat}}{25\, h_{fg}} \right]$$

$$+ K' (k/D)\, Re^m\, Pr^n\, (t_w - t_b)_{BO}$$

where K = .12 - .17, K', m, & n depend on the non-boiling turbulent flow correlation used.

Range of values: V = .05 - 174 ft/sec, p = 4.2 - 3000 psia, ΔT_{sat} = 0 - 506 deg F, a = 1 - 57,000 g's

This correlation is of the superposition form where a subcooled pool boiling burnout term is added to a non-boiling forced convection term, typically the well known McAdams correlation for turbulent flow.

Gambill leaves some latitude for the evaluation of the constant coeffiecients and exponents. Evaluation of the wall temperature (t_w) in the forced convection term is done by an auxiliary plot of ΔT_{sat} vs. T_{sat}/T_{crit} based on Bernath's correlation of wall temperature at burnout (16). This is explained in detail in the section on Bernath's correlation. The correlation was plotted under different conditions with the following results:

1) D = .242 in., G = 3.69 x 10^6 lbm/ft^{2}hr, p = 90 psia (figure 8) The correlation fell about 35% lower than the experimental data.

2) D = .079 in., G = 3.69 x 10^6 lbm/hr-ft^2, 7.38 x 10^6 lbm/hr-ft^2, & 14.76 x 10^6 lbm/hr-ft^2, p = 90 psia (figure 4): The correlation fell as much as 100% lower than the experimental data.

3) D = .180 in., p = 2000 psia, G = 2.1 x 10^6 lbm/hr-ft^2 (figure 9) The correlation is conservative (low), but does not show the trends of the data completely, and is in error by as much as 25%.

Bernath: (ref: 16)

$$(q/A)_{cr} = h_{bo}\ (t_w - t_b)_{bo}$$

where: $h_{bo} = 5710 + 48\ V\ D^{-.6}$ Btu/hr-ft^2-deg F

and: $t_w)_{bo} = 32 = (9/5)\left[57\ \ln(p) - \frac{54(p)}{p + 15} - \frac{V}{4}\right]$ deg F

Range of values: Round tubes, p = 500 - 3000 psia, V = 4.5 - 54 ft/sec, where D is measured in feet, and p in psia.

This correlation is based on a limiting heat transfer coefficient and a correlation of wall temperatures at burnout. The correlation was plotted (figure 9) for G = 2.1 x 10^6 lbm/hr-ft^2 at 2000 psia, and D = .180 in. for comparison with experimental data. The correlation is slightly conservative predicting this data within approximately 5%.

Bettis: (ref: 17)

$$(q/A)_{cr} = 10^6 \times 0.28\ (h_b/10^3\ )^{-2.5}\ (\ 1 + G/10^7\ )^2\ e^{-.0012L/D}\ \text{Btu/hr-ft}^2$$

Range of values: G = 0.2 x 10^6 - 8.0 x 10^6 lbm/hr-ft^2, p = 1850 - 2150 psia, Δh_{sat} = 0 - 180 Btu/lbm, L/D = 21 - 365.

This correlation is compared with experimental data on figure 9. It can be seen that it follows the general trend of experimental data, with variations of 15% to 40%.

Griffith: (ref: 13, 17)

$$(q/A_{cr} = f(\ p/P_c\ )\ (F)\rho_g\ (h_g - h_b)\left[\frac{g\ (\rho_1 - \rho_g\ )}{\mu_1}\left(\frac{k_1}{\rho_1 c_1}\right)^2\right]^{.33}$$

$$(F) = 1 = (Re_b \times 10^{-6}\ ) - 0.14\ K_1 + 0.5 \times 10^{-3}\ (Re_b\ K_1\ )^{\frac{1}{2}}$$

$$Re_b = \frac{V_b\ D\rho_1}{\mu_1}\ ; \qquad K_1 = \frac{\rho_1 c_1\ (T_{sat} - T_b\ )}{\rho_g\ \ h_{fg}};$$

and, $f(\ p/P_c)$ is presented graphically in the reference.

Range of values: p = 14.7 - 3000 psia, V = 0 - 110 ft/sec, ΔT_{sat} = 0 - 280 deg F.

The correlation was plotted and compared with data at two pressures: 90 psia (figure 8) and 2000 psia (figure 9). At 90 psia the correlation is outstanding and does an excellent job of predicting the data. At 2000 psia the correlation is higher than the data, yet only by 5%.

Mirshak, et al: (ref: 13)

$$(q/A)_{cr} = 480{,}000\ (1 + .0131\ p)(1 + .00508\ \Delta T_{sat})(1 + .0365\ V)$$

Range of values: p = 25 - 85 psia, V = 5 - 45 ft/sec, D = .21 - .46 in., where ΔT_{sat} is measured in deg F, and $(q/A)_{cr}$ in Btu/hr-ft^2.

This correlation was compared with experimental data for .242 in. diameter tubes at 90 psia (figure 8) and .305 in. diameter tubes at 30 psia (26). In both cases the correlation curve was conservative (lower than actual data).

DuPont: (ref: 8)

$$(q/A)_{cr} = 595{,}000\ (1 + .04V)(1 + .0055\ \Delta T_{sat})(1 + .0055\ \Delta T_{scg})$$

Range of values: V = 8.5 - 40 ft/sec, ΔT_{sat} = 40 - 120 deg F, D = .3 - .88 in., p = 60 - 1000 psia, ΔT_{scg} = Subcooling Gradient = $(T_{exit} - T_{inlet})$ per 2 foot length of tube.

DuPont's correlation is similar to the one developed earlier by Mirshak, et al (13); a combination of linear factors of velocity and subcooling. In the DuPont tests a two foot length of tube was used. Subcooling Gradient (ΔT_{scg}), the temperature difference through the tube, was the last factor used to correlate experimental data, using the form of a previous correlation for annular geometry.

Doroshchuck and Lantsman: (ref: 9)

$$(q/A)_{cr} = (q/A)_o + A(p)\left[(.316/D)^{.33} - 1\right] \quad \text{Btu/hr-ft}^2$$

where: $(q/A)_o$ is determined from a plot of $(q/A)_{cr}$ vs. Δh_{sat} for 8 mm (.316 in.) tubes for the pressure in question.

Range of values: D = .117 in. - .316 in., p = 50 - 170 atm = 738 - 2500 psia. A(p) is a tabulated function of pressure.

p - atm	psia	A(p) - Btu/hr-ft^2
50	738	2.55×10^6
80	1140	2.41×10^6
100	1470	2.31×10^6
140	2060	2.13×10^6
170	2500	1.99×10^6

Based on the 8 mm (.316 in.) data of Doroshchuck and Lantsman, the correlation predicts critical heat flux for the other diameters with a maximum deviation of 30%, for fixed values of the other variables. With other experimental data the variation is slightly greater.

MIT Small Diameter:

$$(q/A)_{cr} = 1.738 \times 10^6 \,(4.48 + .01\,\Delta h_{sat})(G/G_o)^{.35}\,(D/D_o)^{-.55}\left(\frac{L/D}{40}\right)^{-.12}$$

where: $G_o = 10^6$ lbm/hr-ft^2

$D_o = 0.01$ in.

Range of variables: D = .05 - .305 in., p = 30 - 90 psia, G = 2×10^6 - 15×10^6 lbm/hr-ft^2, L/D = 5 - 60, Δh_{sat} = 0 - 150 Btu/lbm

This equation was developed by the authors for the low pressure data from MIT and fits the majority of the data to ± 25%.

RESULTS OF EXPERIMENTAL STUDY

Scope of Experimental Study

As mentioned, there is a great deal of experimental data available in the literature. Although each investigator has attempted to be somewhat systematic in his own method of attack, there are very few cases where data from one investigation can be compared with that from another. Generally there are substantial differences in the variables, and lacking a good correlating equation, comparison, except in a very general way, is impossible. The experimental phase of this study was conceived with this situation in mind. Ornatski and Kichigin (4, 5 and 6) have conducted tests with tubes of two millimeter diameter, and four mass flow rates, over a range of pressures from 147 psia to 2840 psia. Part of the present study was to complete the low pressure end of the 2 mm map. Also, using the same size tube and flow rate, other parameters were varied. The results of this investigation, when combined with the data from Ornatski and Kichigin, present a fairly complete picture of CHF for tubes of two millimeter diameter.

Experiments were conducted to yield the following sets of data:

a. CHF at various subcooling for three mass flow rates ($G = 3.69 \times 10^6$, 7.37×10^6, 14.76×10^6 lbm/hr-ft^2 - identical to those used by Ornatski and Kichigin) with D = 2mm (.079 in.) for exit pressures of 30 psia and 90 psia.

b. CHF at various tube diameters, with diameters varying from 0.047 in. to 0.242 in., with mass flow rate held constant at 3.69×10^6 lbm/hr-ft^2, and at constant exit subcooling of 35 Btu/lbm.

i. With L/D held constant at a value of 40,

ii. With L held constant at a value of 3.15 inches

c. CHF for various L/D, for the same mass flow rate and exit subcooling as in part b. above with D = 0.079 in. L/D from 3.5 to 60.

d. CHF at various subcoolings for the same mass flow rate, with L/D of 40 and D = 0.242 in. This data was obtained in order to present a comparison with the corresponding D = 0.079 data, and as substantiation of the CHF verses diameter data.

e. CHF, for the same flow rate, exit subcooling and L/D, D = 0.094 in. with changing entrance conditions. In addition to the calming length form used in the majority of the study, four entrance forms were investigated. In all cases, the entrance was placed immediately up-stream of the heated length. Entrance forms investigated were:

i. Sharp Edge Entrance.

ii. 45-degree Chamfer.

iii. Rounded Entrance, radius of entrance equal to 2 diameters.

iv. Reentrant, length of reentrant section equal to 10 diameters.

f. Maximum Heat Flux. An investigation of the system capabilities, using 0.242 in. I.D. tubing, with L/D of 7.75 and the maximum available flow rate and exit subcooling, to determine the maximum obtainable CHF at this diameter.

Results

A total of 117 CHF data points were obtained in the experimental phase of this study. Of these, twelve were obtained using the Beryllium Copper tubing. A detailed description of the problems encountered with this tubing is contained in Appendix D.

Figures 3 and 4 show the CHF data obtained with D = 0.079 in., for the three mass flow rates. These figures show the same general dependence of CHF on mass flow and subcooling as the majority of the other experimental data. This low pressure CHF data does show the minimum CHF in the subcooled region more clearly than does the data at higher pressures. It can be seen that a minimum CHF is encountered at approximately 30 Btu/lbm subcooling, and that the subcooling for minimum CHF increases for increasing mass flow rate. CHF becomes nearly independent of subcooling at the lowest flow rate, at 30 psia.

The dependence of CHF upon tube diameter is shown in figure 5. Investigations were conducted with L/D held constant and with L held constant. The crossing of the two curves merely shows the diameter at which the selected values of L and L/D coincide. The data obtained with L held constant shows a greater CHF for the larger diameters, reaching a maximum increment of approximately 15% at the 0.242 in. diameter tube. The value of CHF can be seen to become less dependent upon diameter for the larger tubes. Limits of the apparatus precluded investigation of diameters larger than 0.242 in. at the flow rate selected for this study.

The comparison between CHF data at D = 0.079 in. and that at D = 0.242 in. can be seen in figure 6. This data provides an evaluation of the validity of the curve of CHF verses diameter (figure 5) which was determined for a particular value of exit subcooling. It can be seen that proportional difference in CHF between the two diameters is nearly constant over the range of exit subcoolings considered.

Investigation of the effect of changing length-diameter ratio upon CHF (figure 7) shows that CHF increases for decreasing L/D. This data shows considerable scatter at the low values of L/D. The test section construction procedure, discussed in detail in Appendix C, was such that very small fillets of solder remained at each end ot the heated length. In these small sections, this limited the accuracy of determining heated length. It can be seen that CHF approaches a minimum value as L/D is increased, and that CHF is essentially independent of L/D for values of L/D greater than 40. This is in agreement with the results of Bergles (1, 11). It should be noted that the value of CHF for L/D of 10 (the generally reported minimum length in the Russian papers) is some 15% greater than the minimum value.

Results of the investigation of Entrance Effect are not plotted. Holding all flow variables constant and changing only the form of the entrance to the heated length yielded the following data:

Entrance Form	Heat Flux
Sharp Edge	3.65×10^6 Btu/ft^{2}hr
45° Chamfer	3.86×10^6 Btu/ft^{2}hr
Rounded	3.82×10^6 Btu/ft^{2}hr
Reentrant	3.55×10^6 Btu/ft^{2}hr
20-D Calming Length	3.82×10^6 Btu/ft^{2}hr

The value for the sharp edge is a mean of two test values, the others are single tests. The 20-D Calming Length was the standard form used for the remainder of the experimental study, the value listed is the result of several tests at the same conditions. Complete data for these tests are listed under runs no. 33 through no. 37 in the listing of data, Appendix C.

In the study of maximum heat flux, subcoolings on the order of 200 Btu/lb_m and flow rates on the order of 5×10^6 lbm/ft^{2}hr were obtained with a test section pressure of 85 psia. These conditions lead to a CHF of almost 6×10^6 Btu/ft^{2}hr. Exact data on these tests are contained in the listing of data, Appendix C, runs through numbers 29, 30 and 31.

IV. CONCLUSIONS AND RECOMMENDATIONS

Discussion of Experimental Results

The data obtained in the experimental phase of this study shows the same general dependance of CHF upon the other variables as has been reported in similar studies. This data is perhaps a little more well ordered than that normally reported. This, of course, is heartening to the investigators, and is believed to be the result of careful experimental procedures and constant cognizance of the flow stability problem. It is felt that instrument and reading accuracies would combine to insure an experimental accuracy of five per-cent or better.

The curves of CHF verses exit subcooling terminate at the maximum subcooling obtained. This maximum is dependent upon the test section inlet temperature and pressure. The curves show the maximum subcooling available consistent with the capacity of the installed heat exchanger and the temperature of the city cooling water (minimum inlet temperature = 47° F). The additional 15 or so Btu/lbm of subcooling which could have been obtained by the installation of a complex test section pre-cooler did not seem to justify the additional complication.

The curves of CHF verses exit subcooling show a minimum value of CHF, signifying a change of flow regime. This minimum occurs in the subcooled region mainly because of the manner of determining conditions at the point of CHF. Fluid conditions were determined using standard thermodynamic variables, which are valid for equalibrium states, but which do not indicate the physical conditions with inherently non-equilibrium subcooled, forced flow boiling. What is needed is a better means of describing the physical state, one which would include some allowance for the non-equilibrium conditions as they actually exist.

The use of a void-fraction variable has been suggested for this purpose (35), but this is very difficult to evaluate and is dependent upon much the same variables as CHF. The use of subcooling, as determined by a difference in enthalpy or temperature, is convenient, but must be accompanied by a realization of its inherent inability to describe the actual physical phenomena.

At the very high values of exit subcooling, the curve of CHF seems to become less steep. That is to say that the rate of change of CHF with subcooling decreases with increasing subcooling. This is a very slight effect, not at all pronounced at low system pressures. The high pressure data of Doroshchuck and Lantsman (10) and Ornatski and Kichigin (5, 6) shows this effect more clearly. A possible explanation for this lies in analysis of the effect of subcooling upon the overall boiling mechanism. It appears that once high values of subcooling are reached, and a large recondensation potential is established, further increases in subcooling have less effect.

The increase of CHF with decreasing tube diameter is evident in figure 5. It has been suggested that void-fraction increases with decreasing diameter. This is due to the manner in which void-fraction is defined, rather than some physical effect. The increase in CHF is a direct result of the requirement for continuity of mass flow. The size of a vapor bubble is dependent mainly upon pressure. In small tubes, the bubble occupies a proportionately larger part of the cross section, requiring that the liquid phase assume a higher velocity to maintain continuity. It has been repeatedly shown, and can be easily explained, that CHF increases with increasing velocity; thus the increase in CHF with decreasing diameter.

The comparison of CHF with length-diameter ratio (figure 7), shows a leveling off at a value of L/D of approximately 40. This is a reasonable minimum value that can be employed in experimental work, where one is attempting to insure that the effects of the different variables are separated.

An interesting effect of changing L/D can be seen in the plot of CHF verses diameter (figure 5) The curve for constant tube length reflects the effect of simultaneously changing diameter and length-diameter ratio. At the larger diameter, the data for constant tube length reflects a substantial decrease in L/D, thus the greater CHF. Unfortunately, the limits of the apparatus were such that larger diameters could not be investigated at the mass flow rate selected for this study. It is altogether possible that the curve for constant length would show an increase in CHF for larger diameters; i.e. for even greater reductions in L/D.

Comparison of CHF data for two tube diameters (figure 6), shows that diameter is one of the more important variables. Within the experimental accuracy of this study, one can conclude that the variation of CHF with diameter is nearly independent of subcooling, at least in the low pressure region. This requires investigation at higher system pressure, where the effect of pressure becomes more pronounced, and where higher levels of subcooling are available.

The data from the entrance effect study shows that entrance geometry is unimportant in determining CHF. Here, as throughout this problem, there is a high degree of interaction between the variables. The investigation of changing entrance form in this study was done with a large length-diameter ratio to minimize the L/D effect. As a result, variations in flow due to the change in entrance form were likely damped out at the point of tube failure. Decreasing the heated length would have introduced the complication of the L/D effect. It is safe to conclude that changing entrance form has no effect for high values of L/D. The somewhat lower CHF for the reentrant form may reflect flow instability due to compressibility in the non-flow space around the reentrant section, but is more likely due to experimental scatter.

The maximum heat flux data is of interest only for future investigations using the particular apparatus employed in this study. This data has no value in analysis of the overall CHF problem because of the limitations imposed upon pressure and mass flow rate by the apparatus. Tests were conducted to determine the maximum CHF that could be obtained with working size tube diameter (0.242 in.) and this apparatus.

DISCUSSION AND EVALUATION OF CORRELATING EQUATIONS

The following comments on the various correlations and predicting schemes are based upon close examination and comparison with experimental data. The effect of the controlling parameters that are alluded to are described in detail in the section in Factors Influencing Critical Heat Flux (Chapter III).

Bell: (ref: 21)

The use of this correlation is restricted to pressures of 2000 psia. There is no restriction on the tube diameter, but from other results, it has been shown CHF has a strong inverse dependence on diameter; therefore, this correlation should be considered useful for large diameters (0.25 in. or larger) only. Although the accuracy is 15% to 30% in the case illustrated (figure 9), the correlation is relatively simple and easy to apply, and is generally conservative in its prediction.

Jens & Lottes: (ref: 17)

This correlation is of a power function form, and would indicate zero CHF at zero subcooling unless a realistic lower limit was established. Since the correlation was designed to have no negative deviation, it is one of the most conservative considered. Its main virtue is its ease of application.

Ryabov and Berzina: (ref: 36)

The 2 mm correlation is adequate for the data at pressures above 147 psia. It is not consistent with experiment for the lower pressure data (30 or 90 psia). There is a minimum subcooling, approximately 18 Btu/lbm below which the correlation fails due to the mathematical form.

The correlation for D = 4mm is adequate for larger tubes, (approximately .3in.), but for smaller tubes it suffers from lack of consideration of a diameter effect. Ryabov and Berzina considered the diameter effect in the derivation of this correlation by changing the leading coefficient, but they intimated that the diameter effect is negligible for larger diameters. This correlation could be improved by substituting a function of diameter for the leading constant.

Zenkevich: (ref: 14)

Evaluation of viscosity in this correlation at fluid bulk temperature produces a curve somewhat like the experimental data and is recommended if this correlation is to be used.

The absence of a tube diameter factor limits the usefulness of this correlation. The lower limit of 0.16 in. is too small to exclude all diameter effect. It is recommended that the use of this correlation be limited to tubes of 0.25 in. in diameter or larger, where the dependence of CHF upon diameter is quite small.

Zenkevich & Subbotin: (ref: 14)

This correlation, like so many others, neglects the effect of tube diameter. The lower limit of 0.16 in. is too small to neglect this effect entirely. For the larger diameter (.316 in.) data; this correlation is excellent. The lower limitation on subcooling is sound, due to the mathematical form of the equation. The upper limit has no physical validity and seems to be evidence of caution. In spite of the diameter consideration, the mathematical expression is convenient, and the results for larger tubes are quite good.

Gambill: (ref: 30)

The curve for wall temperature is suspect, and available data of wall superheat at burnout does not bear out the graph presented. There is too much latitude in choice of constants and exponents to be used in the correlation. Presumably, if the data is reasonably well ordered, the fit of the correlation can be improved by adjusting the values of K, K', m, and n. There are no easily applied guidelines for the adjustment of these constants included in the presentation. This correlations does include a diameter term, but it is only in the forced convection part, and then to only the (m - 1) power, where m is one of

Gambill's "variable" constants, generally on the order of 0.8. This is one of the few correlations which includes small tube diameters, yet clearly the correlated effect of diameter is less than that observed in the data of Bergles (1), Doroshchuck and Lantsman (9), and figure 4. The correlation is no better than that of Mirshak (13) at low pressures and is considerably clumsier to handle. At high pressures (figure 9) this correlation is no better or worse than many others.

Bernath: (ref: 16)

For a given pressure, tube diameter and flow rate, this correlation is linear with subcooling. This is a fairly good first approximation and typical of many correlations. This correlation is rather difficult to apply, requiring auxiliary calculations to determine heat transfer coefficients and wall temperatures. Of the seven high pressure correlations plotted on figure 9, this is one of the best, predicting CHF within 5% for the selected conditions.

Bettis: (ref: 17)

The use of this correlation is restricted to pressures near 2000 psia. There is no restriction on a minimum diameter; therefore, the correlation will yield low predicted values of CHF for diameters less than .25 in.

Griffith: (ref: 13, 17)

This correlation was ogiginally determined for several fluids including water and it is applicable throughout the subcooled boiling region and into net steam generation. Generally, over a wide range of variables, this correlation is the most consistently accurate one available. Its disadvantage lies in its extremely cumbersome and unwieldy form. Computation can be simplified somewhat by using the curves presented in WAPD-188 (17) for several of the recurring variable groups.

Mirshak, et al: (ref: 13)

The form of the correlation is linear with subcooling and it exhibits the approximate tendency of the experimental data. Within the range of diameters for which the correlation is defined, the dependence of CHF on diameter is neglected. The following general coments are appropriate:

1) The correlation gives a conservative indication of CHF, but errs by 25% - 50%
2) There is no consideration of the variation of CHF with diameter
3) The correlation is safe, quick and easy to apply

DuPont: (ref: 8)

This correlation was designed for correlating DuPont's data for tubes of a specific length. How to evaluate Subcooling Gradient for other tube lengths is unresolved. Subcooling gradient is not an independent variable in the equation, being tied to Heat Flux by the First Law of Thermodynamics, Diameter, and flow rate. No diameter effect is noted between 0.30 - 0.88 in. This is a reasonable approximation. Variation in pressure between 60 -1000 psia is considerable and should be taken into account. Since the equation was derived from a small sample of data and for tubes of only one length, its use is not recommended.

Doroshchuck & Lantsman: (ref: 9)

This is one of the few studies which has been devoted exclusively to the diameter effect problem. The general trend is correct, but comparing this correlating equation with experimental data - even that from the paper in which this correlation is presented, shows an error of up to 30%. Part of the difficulty is due to the interaction of the variables.

Comparing this scheme with other experimental data, notably that of Ornatski and Kichigin (4, 5, 6,) and Bergles (1, 11,) also showed the correlation to exceed a 30% error. It was found that a form of $A\left[(D/D_o)^{1.2} - 1\right]$ fit the Ornatski and Kichigin data (4, 5 and 6) quite well, but this failed in comparison with data from other studies.

It is not recommended that this correlation be used to predict CHF; however, it will serve to apply a diameter consideration to other correlations, within rather broad limits.

MIT Small Diameter

This equation was developed not so much as a prediction of CHF, but to provide a fairly concise mathematical expression for the low pressure MIT data. No pressure term is included as the variation of CHF with pressure is neither well ordered nor of large magnitude. The change in CHF with pressure over this range is much less than the error of this equation due to the assumption of CHF as a linear function of subcooling. CHF is, in fact, not linear with subcooling, this form has been assumed merely for convenience. It is recommended that the use of this equation be limited to determination of general levels, and that the actual data be used in any attempts at accurate prediction of CHF.

APPENDIX A

SMALL DIAMETER CRITICAL HEAT FLUX MAP, RECOMMENDED PROCEDURE FOR USE

Figures 3, 4, 10, 11 and 12 provide a graphical presentation of CHF for pressures from 30 psia to 2840 psia over a range of mass flow rates and exit subcoolings for tubes of two mm diameter. These curves in conjunction with the data of CHF dependence upon tube diameter provide a means of graphically determining CHF for tubes up to one-fourth inch diameter, flow rates from 2×10^6 lb_m/ft^2 hr. to 22 lb_m/ft^2 hr, at any pressure and exit subcooling. A procedure of linear and graphical interpolation is recommended for predicting CHF for any reasonable small diameter application. In cases where linear interpolation is recommended , auxiliary curves, have shown this technique to be satisfactory. The accuracy of this graphical prediction technique will exceed that of the few correlating equations applicable to small tubes, and in the majority of the cases tested, was within 15% of the experimental data considered.

The following procedure is recommended for predicting CHF:

1. Select desired system pressure, mass flow rate, tube diameter and length.
2. Locate the plots for pressures bracketing the desired pressure.
3. At the selected exit subcooling, read CHF for the three or four mass flow rates presented, for each pressure.
4. Prepare interpolation curves with coordinates of CHF and Mass Flow Rate, and determine CHF corresponding to the desired flow rate at each pressure.

5. Using the two values thus obtained, linear interpolation between the pressures under consideration will give a value of CHF for the desired pressure, flow rate and subcooling, for a diameter of two mm.

6. Refer to figure 5, and using the curve for constant L/D. determine the value of CHF for the diameter in question. The ratio of this last value, to that at two mm (a value of CHF of 4.15×10^6 Btu/ft^2 hr. at 2 mm diameter will provide satisfactory results) multiplied by the previously determined value of CHF, will yield a predicted value of CHF at the desired conditions of pressure, flow rate, subcooling and tube diameter.

7. In cases where the desired L/D is less than 20, the same sort of ratio correction should be performed, using figure 7. For values of L/D of 20 or greater, the error in CHF will be too small to warrant consideration.

For a rough, quick estimate of CHF, visual interpolation for mass flow rate on the plot for pressure nearest the desired pressure can be easily accomplished. The diameter correction should be included in this abbreviated procedure. This same sort of approach is useful in estimating the effect of gross change in the important variables.

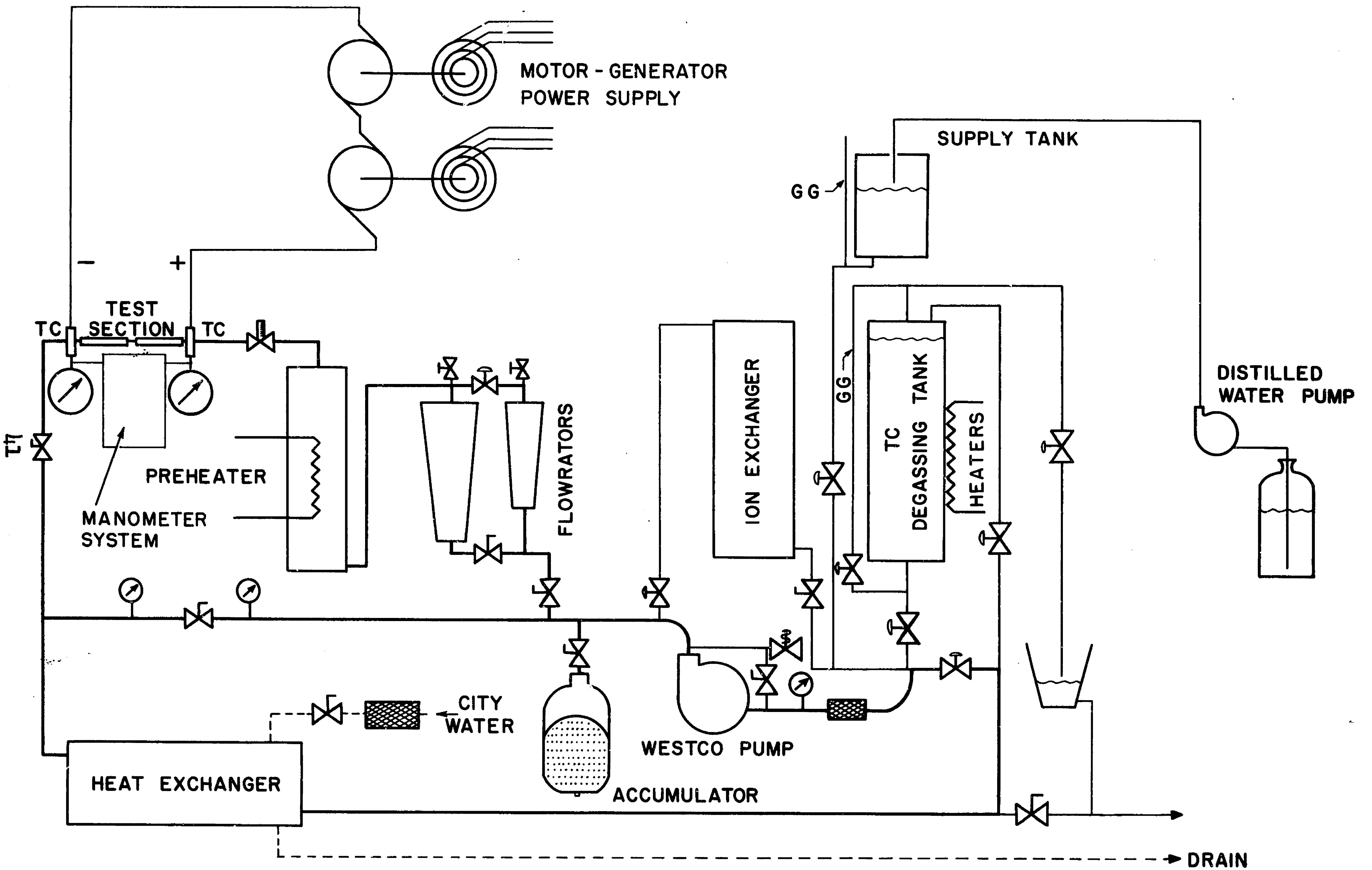

Figure (1) SCHEMATIC LAYOUT OF EXPERIMENTAL FACILITY

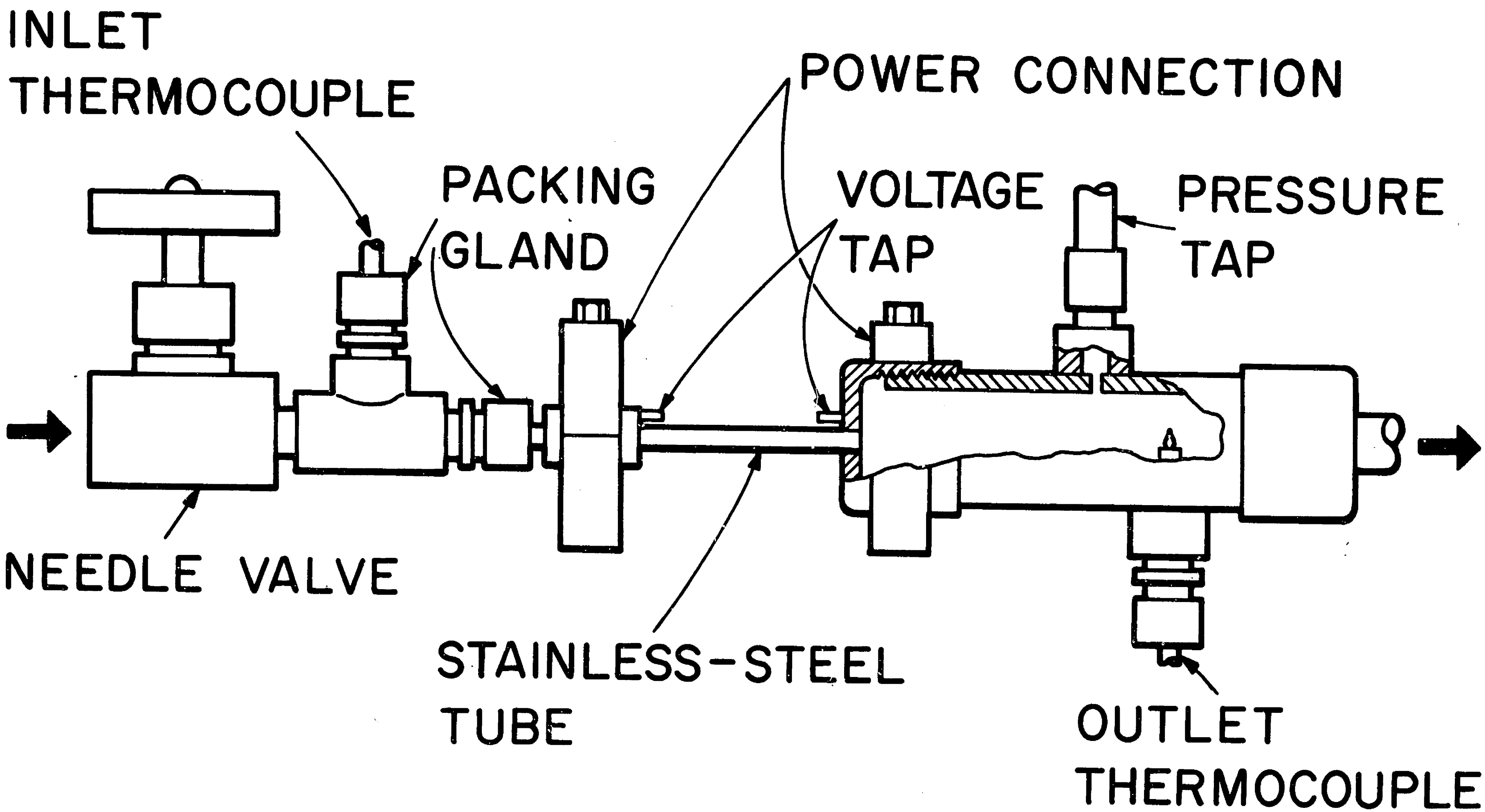

Figure (2) - Detail of Test Section Assembly

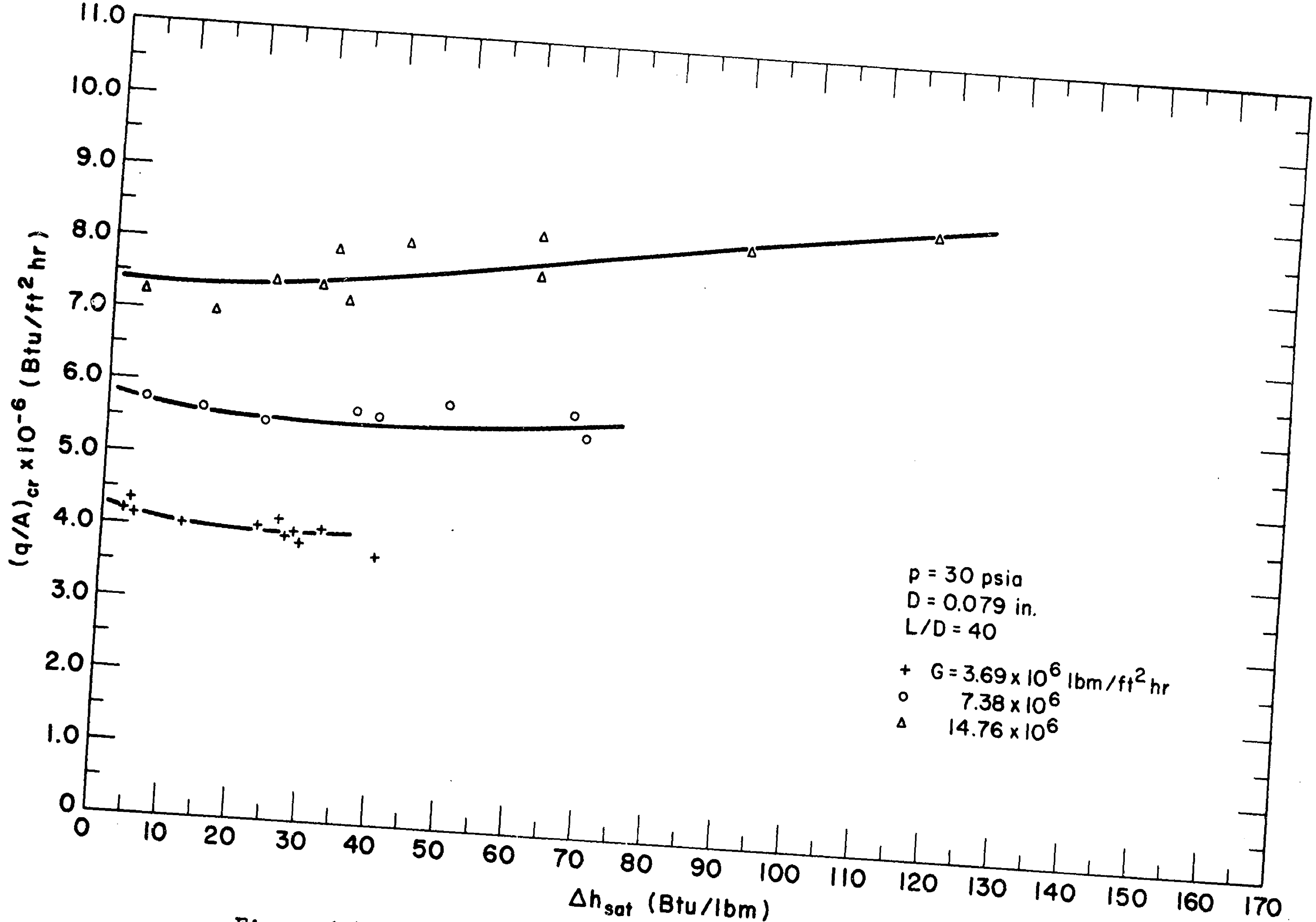

Figure (3) - Dependence of Critical Heat Flux on Subcooling

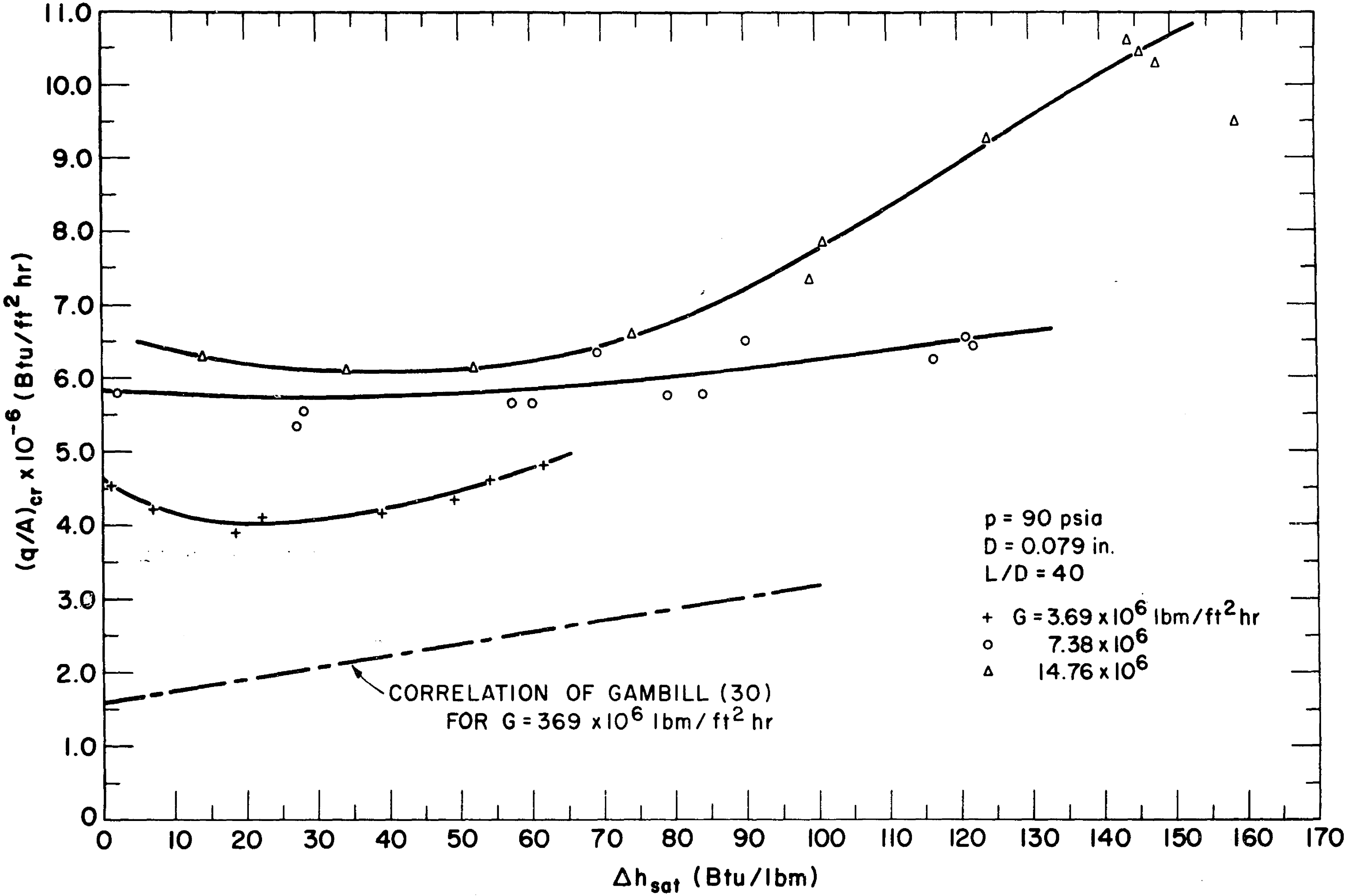

Figure (4) - Dependence of Critical Heat Flux on Subcooling

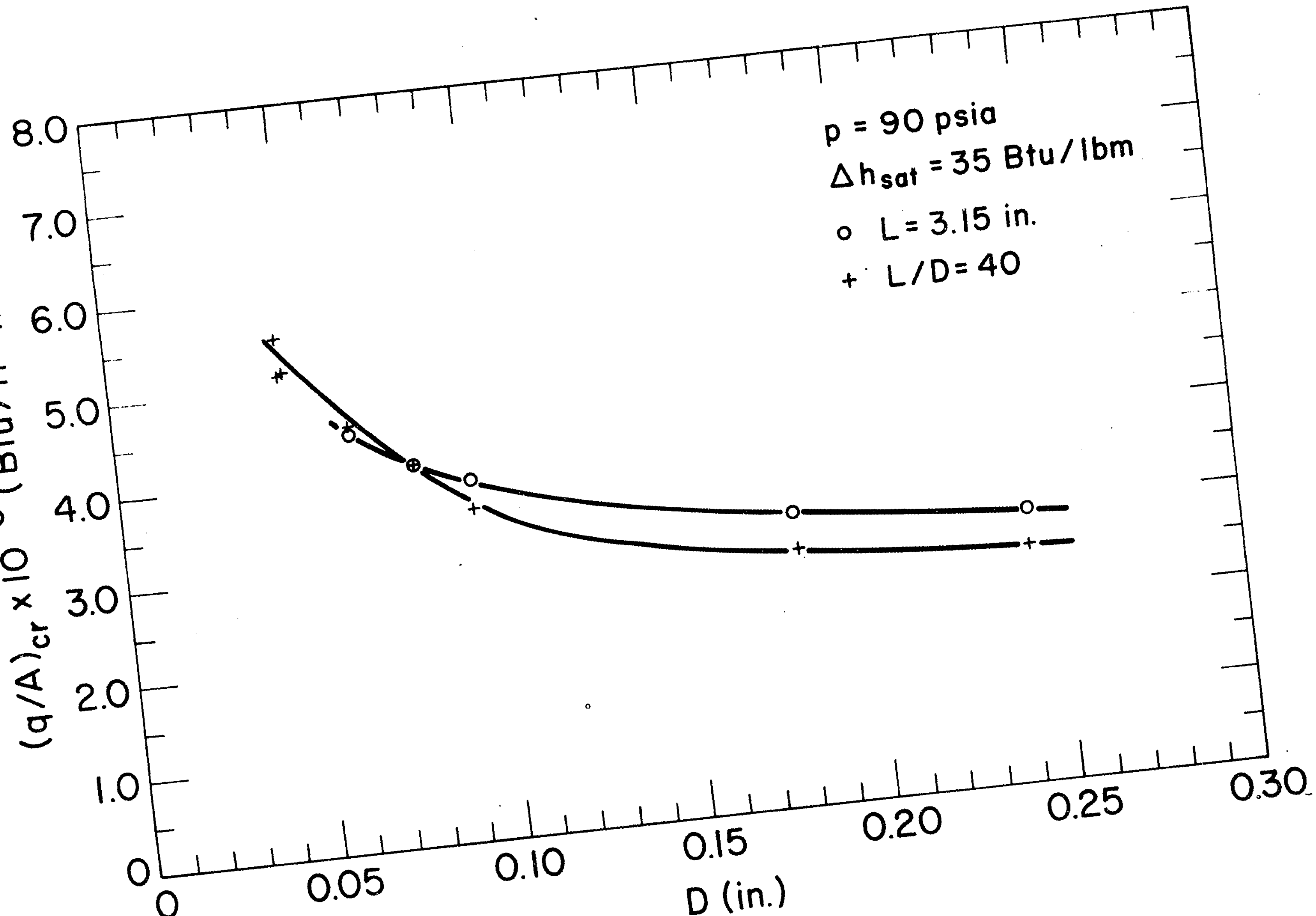

Figure (5) - Dependence of Critical Heat Flux on Tube Diameter

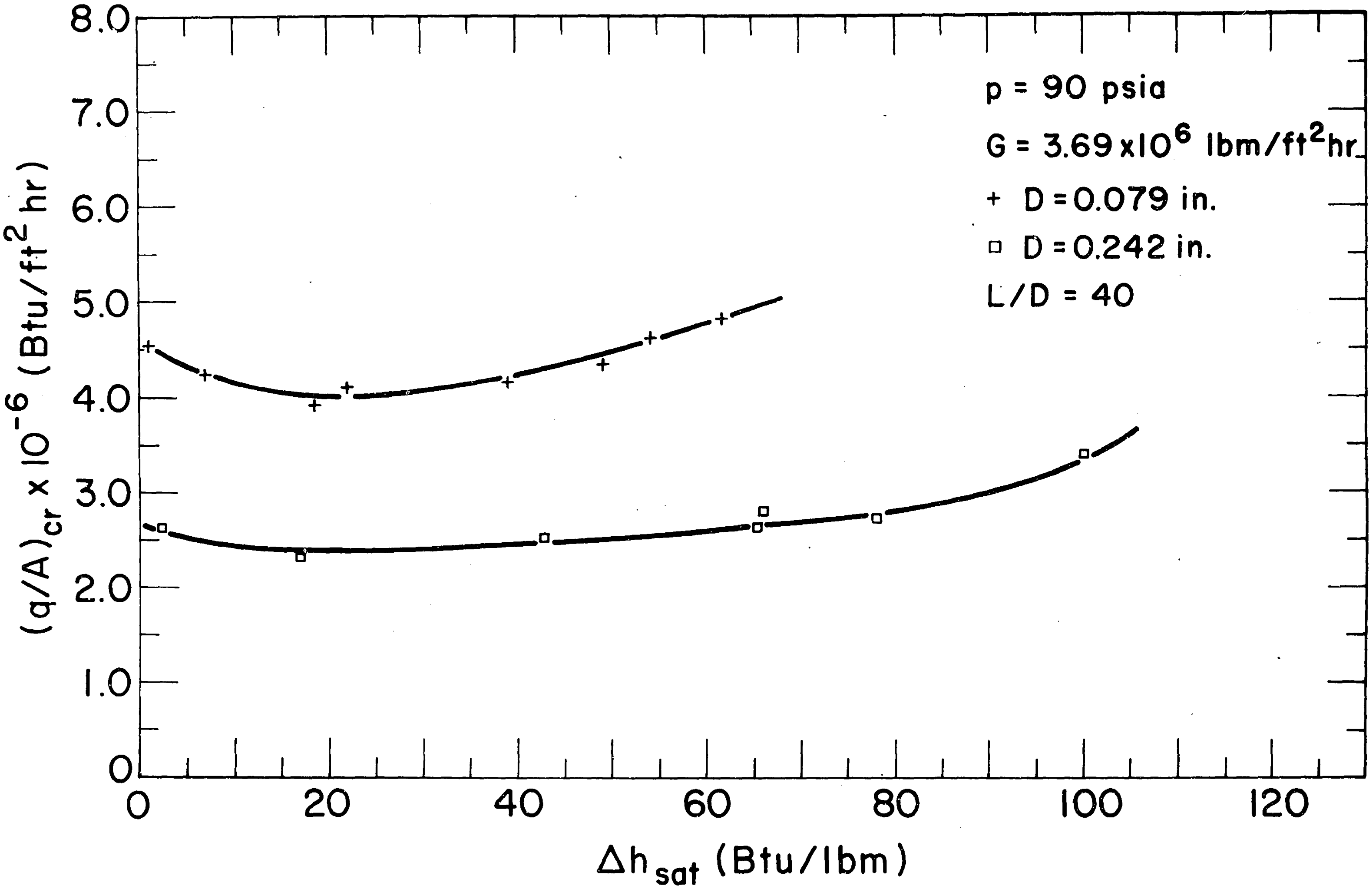

Figure (6) - Dependence of Critical Heat Flux on Subcooling

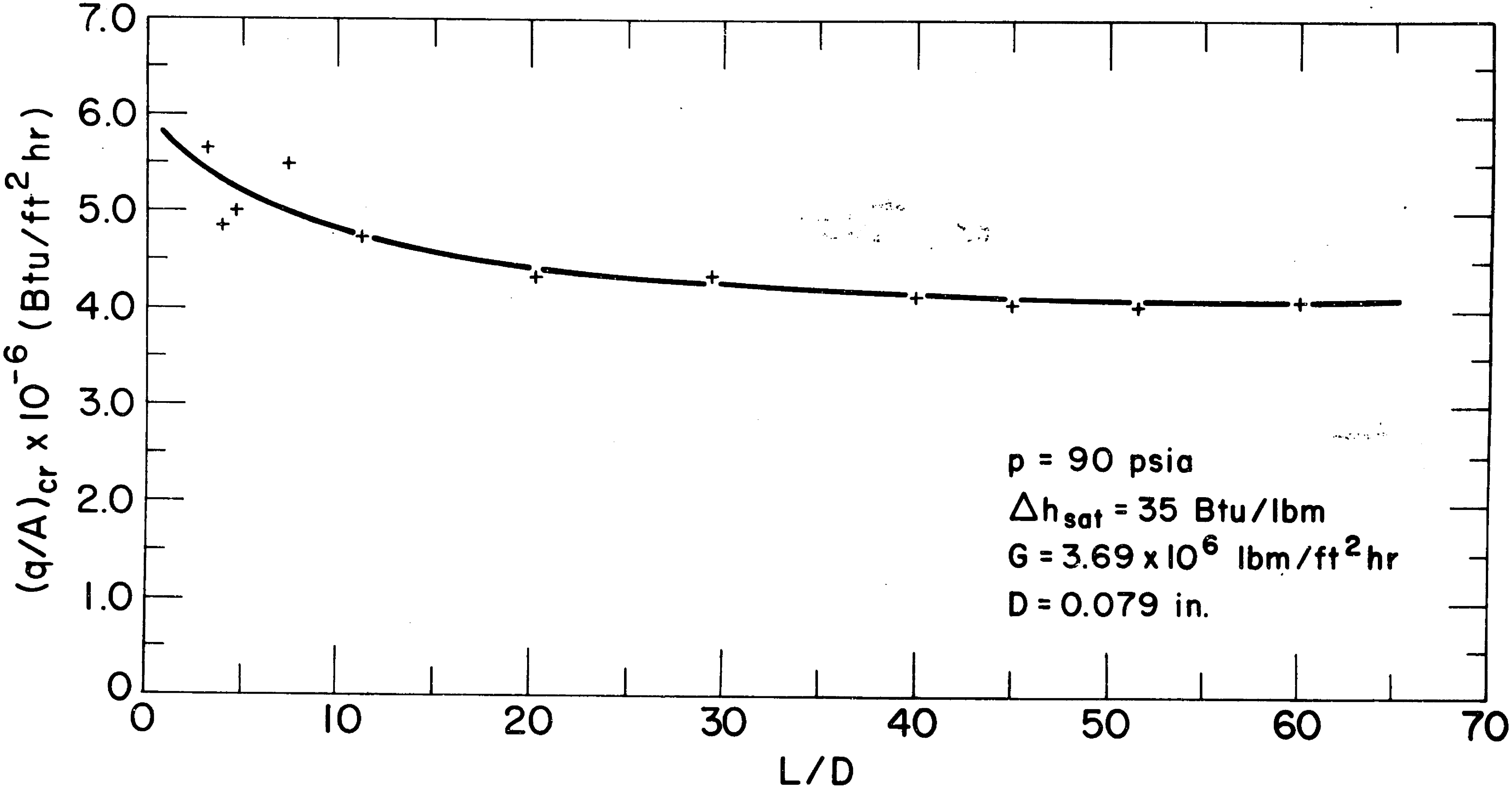

Figure (7) - Dependence of Critical Heat Flux on Heated Length

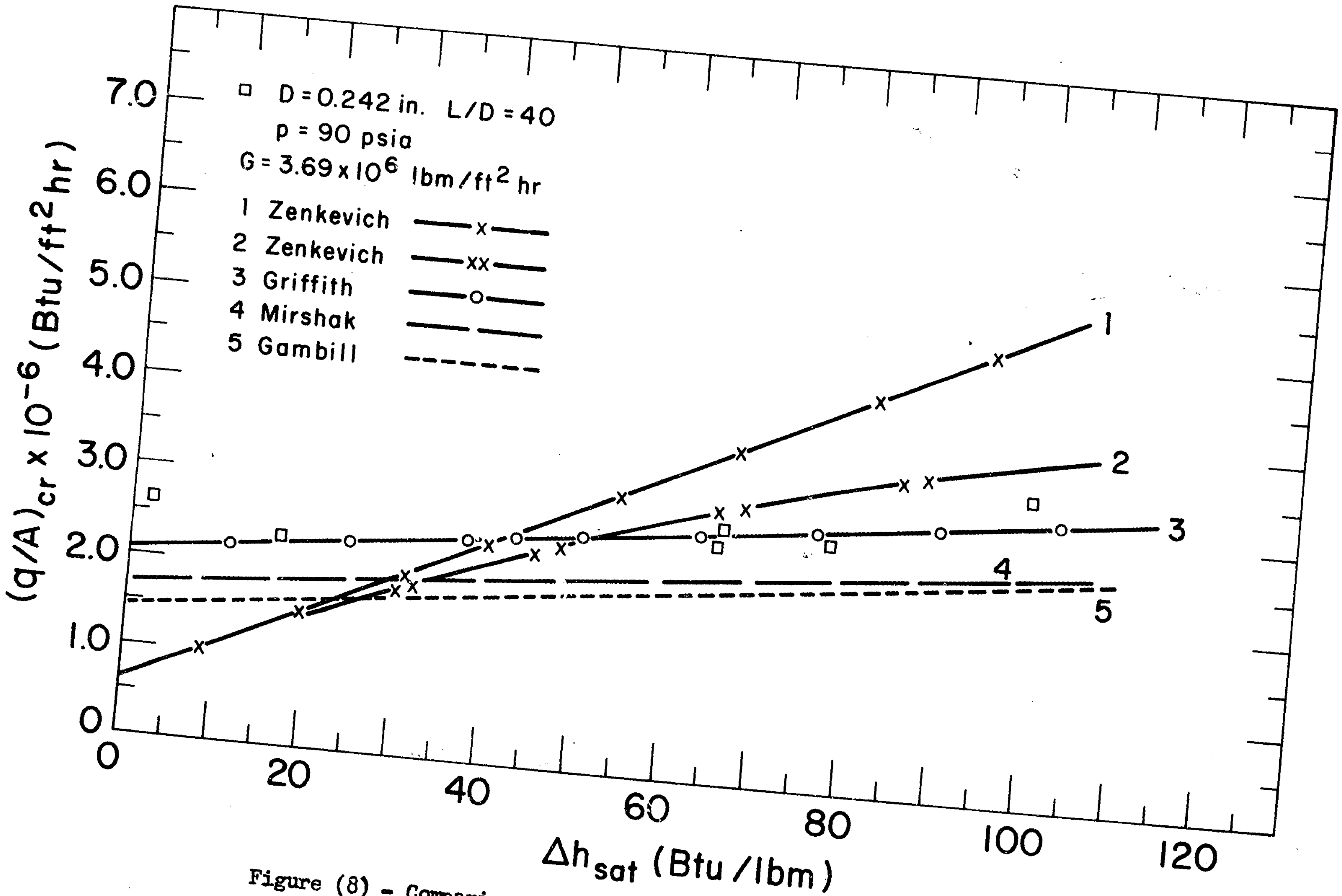

Figure (8) - Comparison of Correlations and Experimental Data

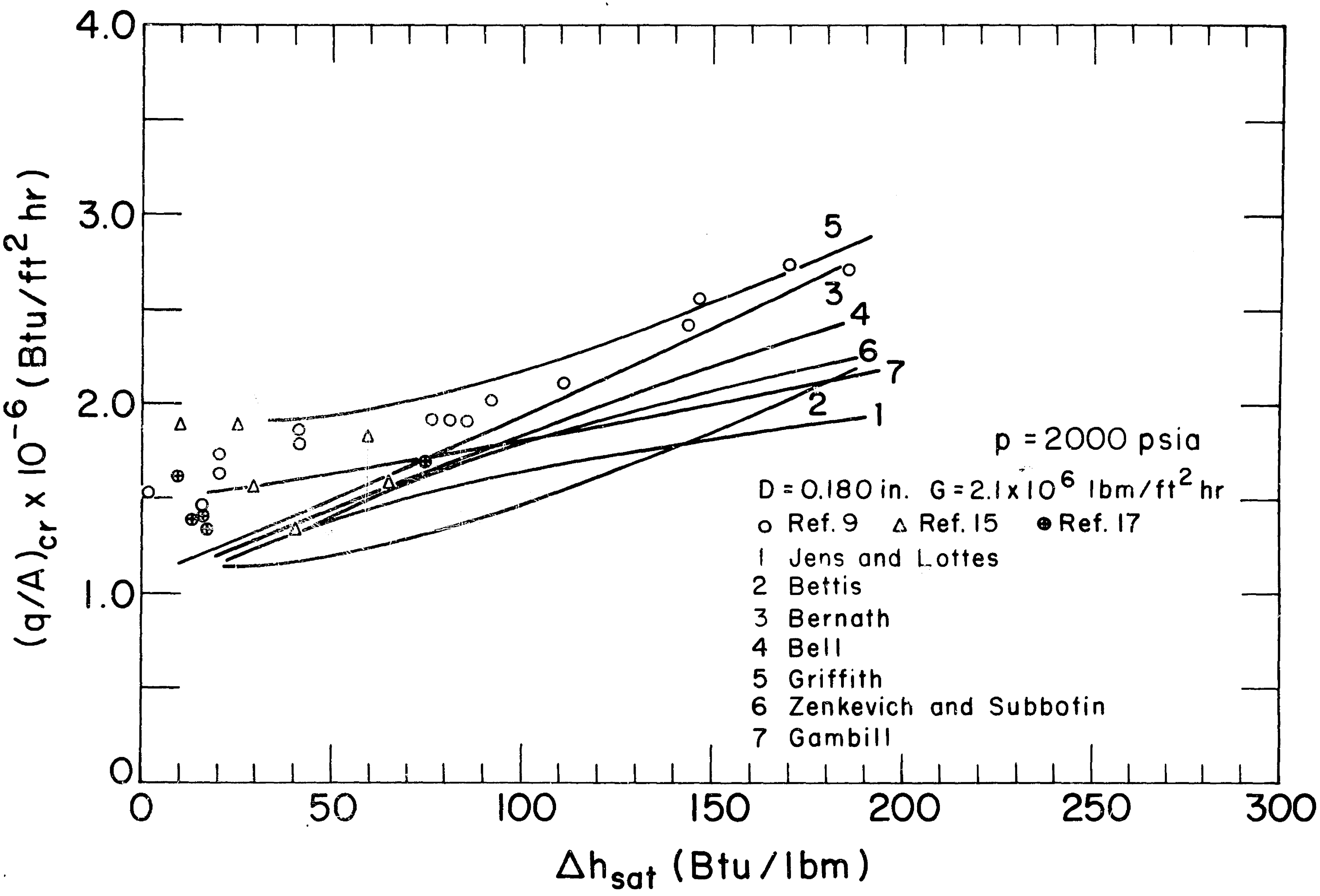

Figure (9) - Comparison of Correlations and Experimental Data

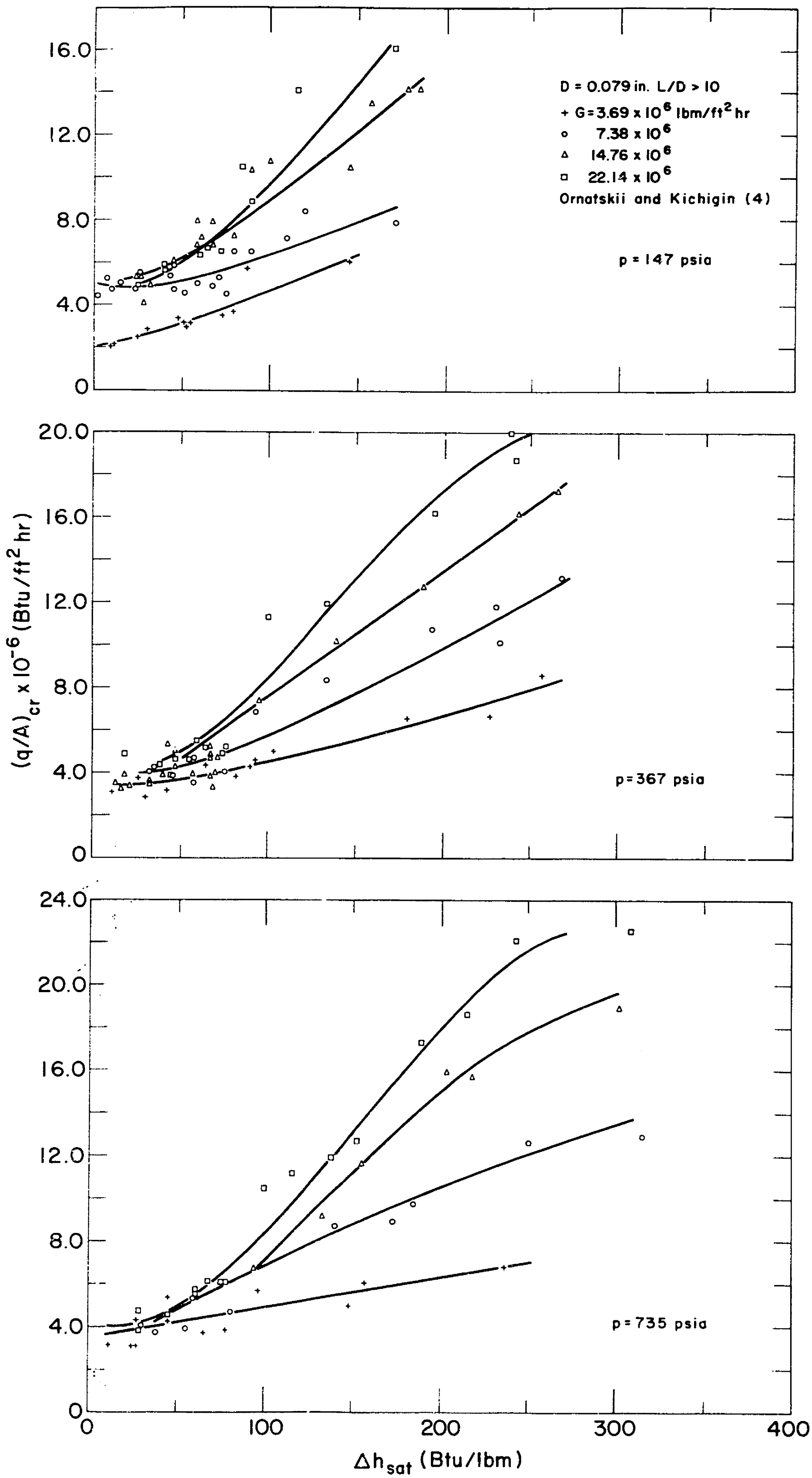

Figure (10) - Dependence of Critical Heat Flux on Subcooling

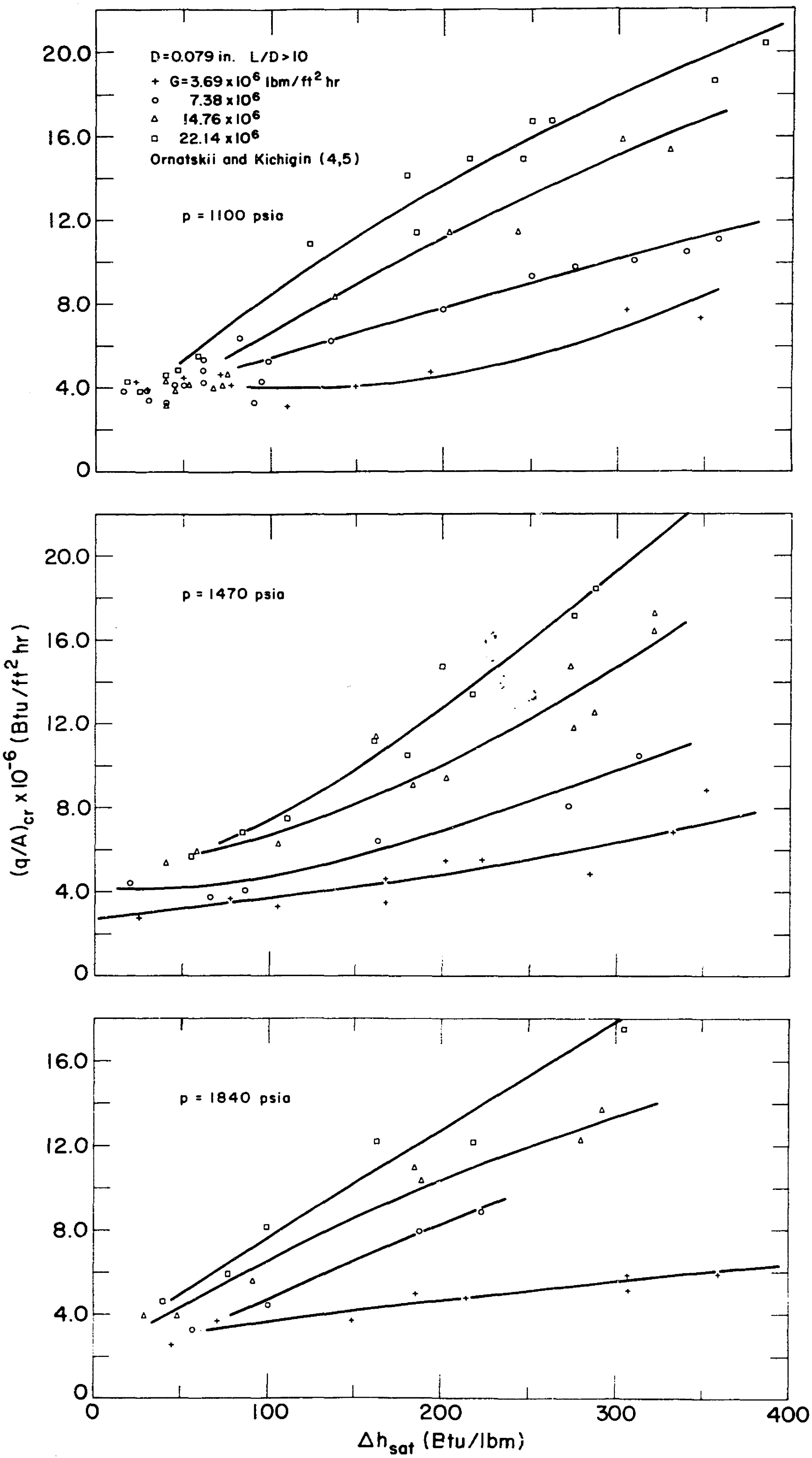

Figure (11) - Dependence of Critical Heat Flux on Subcooling

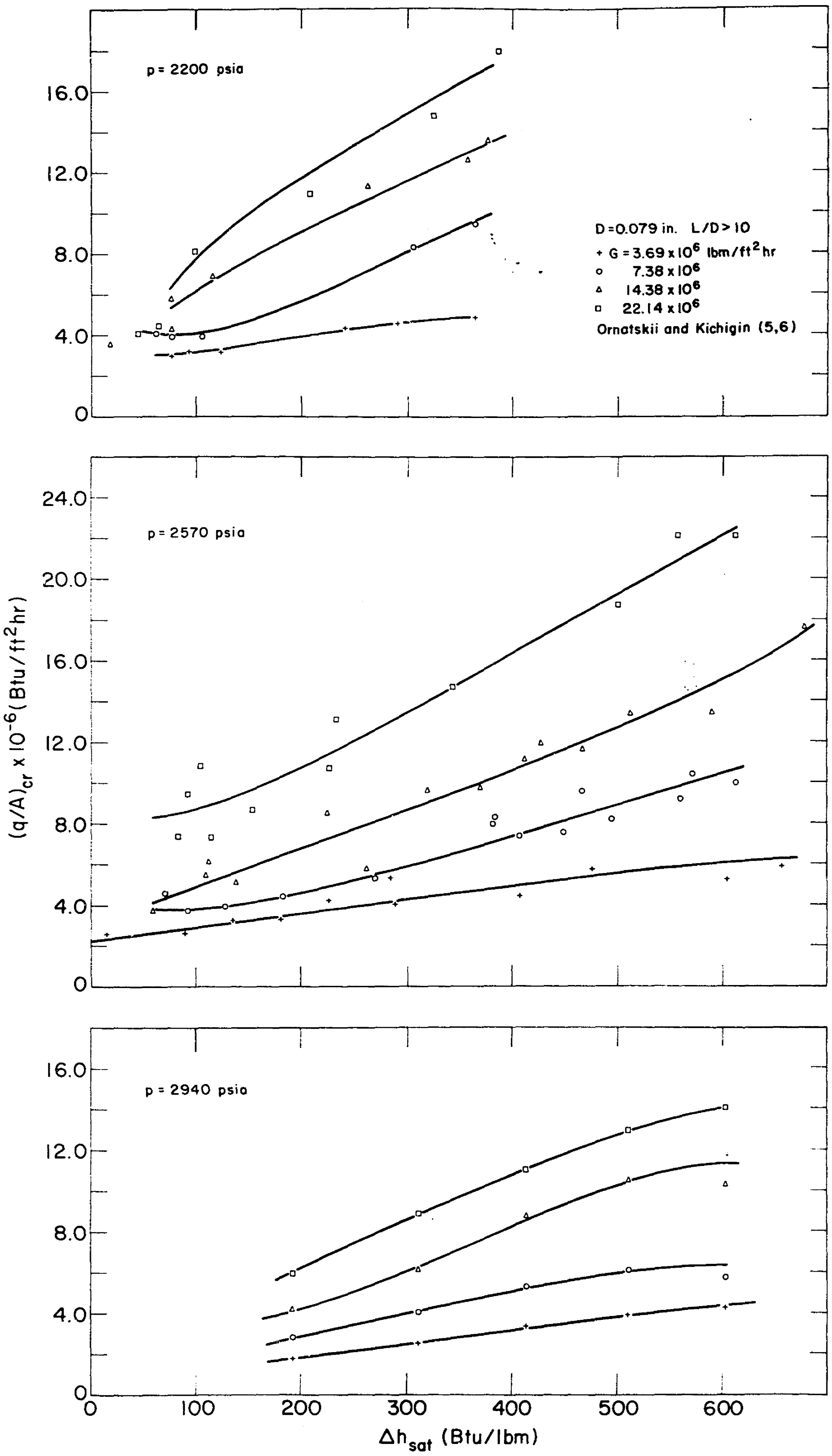

Figure (12) - Dependence of Critical Heat Flux on Subcooling

APPENDIX - C DATA

SUBCOOLED CRITICAL HEAT FLUX DATA

D	TUBE DIAMETER - IN.
L	HEATED LENGTH - IN.
P	PRESSURE - PSIA.
G	MASS FLOW RATE - LBM/HR-SQ.FT.
SUB	SUBCOOLING - BTU/LBM
Q/A	CRITICAL HEAT FLUX - BTU/HR-SQ.FT.
CLGL	CALMING LENGTH - IN.

D	L	P	G	SUB	Q/A	REF	YR	NO.	COMMENTS
.079	3.15	97.	5.23	66.	5.62	4	1965	2	BE-CU TUBE
.079	3.12	90.	5.23	79.	6.02	4	1965	3	BE-CU TUBE
.079	3.10	82.	5.23	82.	5.75	4	1965	4	BE-CU TUBE
.079	3.15	91.	5.23	87.	5.86	4	1965	5	BE-CU TUBE
.079	3.12	85.	5.23	95.	6.22	4	1965	6	BE-CU TUBE
.079	3.15	87.	18.7	122.	13.6	4	1965	7	BE-CU TUBE
.079	3.15	92.	5.23	45.	5.02	4	1965	8	BE-CU TUBE
.079	3.15	89.	5.23	61.	4.24	4	1965	9	BE-CU TUBE
.079	3.15	96.	5.23	83.	6.08	4	1965	10	BE-CU TUBE
.079	3.15	95.	5.23	31.	5.19	4	1965	11	BE-CU TUBE
.079	3.15	97.	5.23	10.	5.08	4	1965	12	BE-CU TUBE
.079	3.15	90.	5.23	42.	4.48	4	1965	13	BE-CU TUBE
.047	1.96	90.	3.69	55.	5.13	4	1965	15	
.047	1.96	87.	3.69	24.	5.52	4	1965	16	
.047	1.90	77.	3.69	35.	5.13	4	1965	17	
.094	3.15	90.	3.69	30.	3.83	4	1965	18	
.094	3.76	90.	3.69	36.	3.54	4	1965	20	
.180	3.15	89.	3.69	34.	3.13	4	1965	21	
.180	7.20	87.	3.69	30.	2.75	4	1965	22	
.242	9.68	80.	3.69	26.	2.52	4	1965	23	
.242	3.15	89.	3.69	53.	2.10	4	1965	24	
.242	3.15	84.	3.69	38.	2.89	4	1965	25	
.063	2.52	85.	3.69	26.	4.53	4	1965	27	
.063	3.15	86.	3.69	34.	4.45	4	1965	28	
.242	1.875	84.	4.89	218.	6.32	4	1965	29	T= .011 IN
.242	1.875	82.	4.89	217.	5.65	4	1965	30	T= .021 IN
.242	1.875	85.	4.84	214.	5.84	4	1965	31	T= .026 IN
.094	3.15	93.	3.69	36.	3.82	4	1965	32	
.094	3.15	93.	3.69	36.	3.62	4	1965	33	SHARP EDGE
.094	3.15	93.	3.69	36.	3.69	4	1965	34	SHARP EDGE
.094	3.15	94.	3.69	36.	3.86	4	1965	35	45 DEG BEV
.094	3.15	94.	3.69	35.	3.80	4	1965	36	ROUND ENT
.094	3.15	91.	3.69	37.	3.55	4	1965	37	10D REENT
.242	9.68	90.	3.69	78.	2.74	4	1965	38	
.242	9.68	90.	3.69	66.	2.79	4	1965	39	
.242	9.68	92.	3.69	100.	3.39	4	1965	40	
.242	9.68	92.	3.69	66.	2.63	4	1965	42	
.079	3.15	90.	3.69	62.	4.82	4	1965	45	
.079	3.15	90.	3.69	54.	4.62	4	1965	46	
.079	3.15	92.	3.69	49.	4.35	4	1965	47	
.079	3.15	86.	3.69	39.	4.17	4	1965	48	
.079	3.15	87.	3.69	22.	4.10	4	1965	49	

D	L	P	G	SUB	Q/A	REF	YR	NO.	COMMENTS
.079	3.15	90.	3.69	19.	3.90	4	1965	50	
.079	3.15	99.	3.69	1.	4.54	4	1965	51	
.079	3.15	93.	3.69	7.	4.23	4	1965	52	
.079	3.15	81.	7.38	122.	6.46	4	1965	53	
.079	3.15	90.	7.38	90.	6.52	4	1965	54	
.079	3.15	90.	7.38	69.	6.38	4	1965	55	
.079	3.15	90.	7.38	60.	5.66	4	1965	56	
.079	3.15	92.	7.38	28.	5.56	4	1965	57	
.079	3.15	87.	7.38	2.	5.78	4	1965	58	
.079	3.15	90.	7.38	27.	5.36	4	1965	59	
.079	3.15	92.	7.38	57.	5.67	4	1965	60	
.079	3.15	90.	7.38	79.	5.78	4	1965	61	
.079	3.15	90.	7.38	84.	5.79	4	1965	62	
.079	3.15	88.	14.76	159.	9.48	4	1965	63	
.079	3.15	88.	14.76	123.	9.25	4	1965	64	
.079	3.15	85.	14.76	99.	7.35	4	1965	65	
.079	3.15	90.	14.76	101.	7.83	4	1965	66	
.079	3.15	89.	14.76	74.	6.60	4	1965	67	
.079	3.15	89.	14.76	52.	6.14	4	1965	68	
.079	3.15	90.	14.76	34.	6.11	4	1965	69	
.079	3.15	93.	14.76	14.	6.30	4	1965	70	
.079	3.15	30.	3.69	4.	4.15	4	1965	71	
.079	3.15	30.	3.69	26.	4.18	4	1965	72	
.079	3.15	31.	3.69	22.	4.04	4	1965	73	
.079	3.15	33.	3.69	3.	4.20	4	1965	74	
.079	3.15	31.	3.69	40.	3.71	4	1965	75	
.079	3.15	30.	3.69	27.	3.93	4	1965	76	
.079	3.15	29.	3.69	28.	4.00	4	1965	77	
.079	3.15	29.	7.38	69.	5.55	4	1965	78	
.079	3.15	30.	7.38	49.	5.88	4	1965	79	
.079	3.15	30.	7.38	67.	5.85	4	1965	80	
.079	3.15	30.	7.38	39.	5.66	4	1965	81	
.079	3.15	30.	7.38	36.	5.70	4	1965	82	
.079	3.15	30.	7.38	23.	5.52	4	1965	83	
.079	3.15	30.	14.76	91.	8.30	4	1965	84	
.079	3.15	29.2	14.76	42.	8.08	4	1965	85	
.079	3.15	30.	14.76	32.	7.95	4	1965	86	
.079	3.15	30.	14.76	14.	7.05	4	1965	87	
.079	3.15	30.	14.76	4.	7.24	4	1965	88	
.079	3.15	30.	14.76	30.	7.45	4	1965	89	
.079	3.15	30.	14.76	34.	7.25	4	1965	90	
.079	3.15	30.	14.76	23.	7.48	4	1965	91	
.079	3.15	30.	14.76	61.	7.75	4	1965	92	
.079	3.15	30.	3.69	61.	8.30	4	1965	94	
.079	3.15	29.	14.76	118.	8.65	4	1965	93	
.079	3.15	30.	3.69	32.	4.03	4	1965	95	
.079	3.15	30.	3.69	4.	4.36	4	1965	96	
.079	3.15	30.	3.69	29.	3.84	4	1965	97	
.079	3.15	30.	3.69	12.	4.03	4	1965	99	
.079	3.15	29.5	7.38	14.	5.67	4	1965	100	
.079	3.15	29.5	7.38	5.4	5.76	4	1965	101	
.079	3.15	89.	7.38	121.	6.58	4	1965	102	
.079	3.15	88.	7.38	116.	6.26	4	1965	103	
.079	2.31	90.	3.69	34.	4.36	4	1965	104	L/D= 29.3

D	L	P	G	SUB	Q/A	REF	YR	NO.	COMMENTS
.079	1.59	91.	3.69	34.	4.36	4	1965	105	L/D= 20.2
.079	.88	90.	3.69	31.	4.74	4	1965	106	L/D= 11.2
.079	.37	90.	3.69	34.	5.00	4	1965	107	L/D= 4.7
.079	.31	87.	3.69	31.	4.85	4	1965	108	L/D= 3.9
.079	.575	91.	3.69	36.	5.50	4	1965	109	L/D= 7.3
.079	.25	90.	3.69	34.	5.65	4	1965	110	L/D= 3.2
.079	3.55	91.	3.69	37.	4.08	4	1965	111	L/D= 45.
.079	4.75	94.	3.69	26.	4.13	4	1965	112	L/D= 60.
.079	4.08	86.	3.69	28.	4.06	4	1965	113	L/D= 51.5
.242	9.68	92.	3.69	17.	2.32	4	1965	114	
.242	9.68	88.	3.69	43.	2.52	4	1965	115	
.242	9.68	91.	3.69	2.2	2.61	4	1965	116	
0.079	3.15	91.	14.76	145.5	10.45	4	1965	118	
0.079	3.15	90.	14.76	147.7	10.30	4	1965	119	
0.079	3.15	89.	14.76	143.8	10.60	4	1965	120	

SUBCOOLED CRITICAL HEAT FLUX DATA TAKEN AT MIT

REF 1 BERGLES, A.E. + ROHSENOW, W.M. (1)
REF 2 DORMER, T. + BERGLES, A.E. (2)
REF 3 BERGLES, A.E. (11)
REF 4 SKINNER, B.C. + LOOSMORE, C.S.
REF 5 REYNOLDS, J.M. (22)
REF 6 WESSEL, H.L. (26)
REF 7 SCAROLA, L.S. (27)

D	L	P	G	SUB	Q/A	REF	YR	NO.	COMMENTS
0.094	1.38	30.1	2.24	79.	4.68	1	1962	24	CLGL 1.6 IN
0.094	1.38	30.2	2.24	72.	4.52	1	1962	25	CLGL 1.6 IN
0.094	1.37	30.2	2.24	70.	4.22	1	1962	26	CLGL 1.6 IN
0.094	1.40	30.3	2.24	64.	3.68	1	1962	27	CLGL 1.6 IN
0.094	1.39	30.2	2.24	47.	3.18	1	1962	28	CLGL 1.6 IN
0.094	1.40	30.2	2.24	22.	3.08	1	1962	29	CLGL 1.6 IN
0.094	1.40	30.1	2.24	32.	3.04	1	1962	30	CLGL 1.6 IN
0.094	1.40	30.4	2.24	55.	3.76	1	1962	31	CLGL 1.6 IN
0.094	1.39	30.1	2.24	40.	3.05	1	1962	32	CLGL 1.6 IN
0.094	1.39	30.2	2.24	12.	3.20	1	1962	34	CLGL 1.6 IN
0.094	1.39	30.2	2.24	22.	2.66	1	1962	35	CLGL 1.6 IN
0.094	1.37	30.0	2.24	2.	3.35	1	1962	36	CLGL 1.6 IN
0.094	1.40	30.5	2.24	-23.	3.53	1	1962	37	CLGL 1.6 IN
0.094	1.39	31.7	2.24	-51.	3.62	1	1962	40	CLGL 1.6 IN
0.094	1.39	31.2	2.24	-51.	3.66	1	1962	41	CLGL 1.6 IN
0.094	3.25	29.9	2.24	32.	2.57	1	1962	48	L/D 34.6
0.094	3.25	29.7	2.24	31.	2.57	1	1962	49	L/D 34.7
0.094	2.77	30.6	2.24	32.	2.50	1	1962	50	L/D 29.6
0.094	2.75	30.5	2.24	31.	2.66	1	1962	51	L/D 29.2
0.094	1.88	30.1	2.24	31.	2.91	1	1962	53	L/D 20.0
0.094	1.98	29.9	2.24	32.	2.88	1	1962	54	L/D 21.0
0.094	1.15	30.1	2.24	32.	3.13	1	1962	55	L/D 12.3
0.094	0.49	30.2	2.24	32.	4.16	1	1962	56	L/D 5.2
0.094	0.47	30.6	2.24	32.	4.19	1	1962	57	L/D 4.7
0.094	0.72	30.6	2.24	34.	3.71	1	1962	58	L/D 7.6
0.094	1.18	30.2	2.24	39.	3.25	1	1962	14	CLGL 1.6 IN
0.094	1.16	30.5	2.24	36.	3.05	1	1962	15	CLGL 1.6 IN
0.094	2.32	30.6	2.24	51.	3.14	1	1962	22	CLGL 1.6 IN
0.094	2.35	30.2	2.24	53.	3.33	1	1962	23	CLGL 1.6 IN
0.094	4.70	30.6	4.48	34.	3.91	1	1962	3	CLGL 1.6 IN
0.094	2.33	29.9	2.24	32.	2.76	1	1962	6	CLGL 1.6 IN
0.094	2.33	30.4	2.24	35.	2.68	1	1962	7	CLGL 1.6 IN
0.094	2.33	30.5	4.48	34.	4.13	1	1962	8	CLGL 1.6 IN
0.094	2.33	30.1	4.48	31.	4.35	1	1962	21	CLGL 1.6 IN
0.180	4.50	29.9	2.24	38.	1.92	1	1962	4	CLGL 1.6 IN
0.180	4.50	30.7	2.24	35.	1.90	1	1962	5	CLGL 1.6 IN
0.180	4.45	30.6	4.48	46.	2.66	1	1962	47	CLGL 1.6 IN
0.047	1.18	30.6	2.24	34.	3.60	1	1962	9	CLGL 1.6 IN
0.047	0.93	30.6	2.24	32.	3.98	1	1962	46	CLGL 1.6 IN
0.047	1.14	30.2	4.48	39.	6.25	1	1962	10	CLGL 1.6 IN
0.047	1.13	30.6	4.48	36.	6.04	1	1962	18	CLGL 1.6 IN
0.023	0.54	30.1	2.24	42.	1.87	1	1962	12	PREMATURE

D	L	P	G	SUB	Q/A	REF	YR	NO.	COMMENTS
0.023	0.57	30.7	2.24	33.	1.54	1	1962	16	PREMATURE
0.023	0.56	29.8	4.48	30.	4.62	1	1962	19	
0.023	0.55	30.5	4.48	35.	4.83	1	1962	20	
0.023	0.39	29.8	2.24	33.	4.38	1	1962	45	
0.023	0.39	29.8	4.48	33.	7.97	1	1962	45	
0.023	0.56	30.1	4.48	36.	6.88	1	1962	43	
0.023	0.51	30.3	4.48	31.	6.35	1	1962	44	
0.180	9.0	30.5	3.36	30.	2.53	1	1962	1	CLGL 1.6 IN
0.180	9.0	30.2	3.36	39.	2.39	1	1962	2	CLGL 1.6 IN
0.094	4.69	30.0	8.96	50.0	5.72	2	1964	1	BO-30-1
0.094	4.66	30.0	8.96	82.	4.99	2	1964	21	
0.094	4.69	30.0	8.96	10.	5.48	2	1964	3	BO-30-3
0.094	4.70	30.0	8.96	0.	5.42	2	1964	4	BO-30-4
0.094	4.71	30.0	4.71	17.	3.88	2	1964	5	BO-30-5
0.094	4.69	30.0	8.96	32.0	5.58	2	1964	2	BO-30-2
0.094	4.54	29.0	8.96	77.	4.98	2	1964	22	
0.094	4.90	80.	6.72	79.	4.45	2	1964	30	
0.180	9.03	50.	2.24	29.	1.78	2	1964	50	B50(A)
0.180	4.41	50.	6.72	108.	4.76	2	1964	25	B25(C)
0.121	5.88	50.	4.48	56.	3.52	2	1964	50	C50(A)
0.121	5.89	50.	6.72	70.	4.42	2	1964	50	C50(B)
0.121	2.84	50.	2.24	68.	3.51	2	1964	25	C25(B)
.046	1.15	29.7	10.15	106.	9.98	3	1963	A1	
.046	1.17	30.	10.15	109.	9.19	3	1963	A2	
.046	1.14	29.7	10.15	97.	9.92	3	1963	A3	
.046	1.17	29.8	10.15	85.	8.64	3	1963	A4	
.046	1.16	30.5	10.15	9.	7.95	3	1963	A5	
.046	1.17	30.	10.15	-13.	8.05	3	1963	A6	
.046	1.15	30.	10.15	73..	8.05	3	1963	A7	
.046	1.17	30.	10.15	42.	9.15	3	1963	A8	
.046	1.18	30.	10.15	25.	8.50	3	1963	A9	
.046	1.15	30.	10.15	47.	9.64	3	1963	A10	
.046	1.17	30.	10.15	75.	8.78	3	1963	A11	
.046	1.17	30.	4.47	54.	6.38	3	1963	B1	
.046	1.14	30.	4.47	34.	6.00	3	1963	B2	
.046	1.15	30.	4.47	24.	5.67	3	1963	B3	
.046	1.15	30.	4.47	-7.0	6.13	3	1963	B4	
.046	1.15	30.	4.47	5.	6.07	3	1963	B5	
.046	1.14	30.	4.47	-21.0	6.22	3	1963	B6	
.046	1.15	30.	4.47	9.0	6.04	3	1963	B7	
.046	1.14	30.	4.47	26.	6.19	3	1963	B8	
.046	1.09	30.	4.47	-28.	6.04	3	1963	B9	
.046	1.15	30.	2.24	25.	3.91	3	1963	C1	
.046	1.16	30.	2.24	-11.	4.15	3	1963	C2	
.046	1.16	30.	2.24	-38.	3.91	3	1963	C4	
.046	1.16	30.	2.24	-15.	3.88	3	1963	C3	
.046	1.15	30.	17.9	119.	14.2	3	1963	E1	
.046	1.16	30.	17.9	93.	13.75	3	1963	E2	
.046	1.17	30.	17.9	81.	12.75	3	1963	E3	
.046	1.17	30.	17.9	62.	12.43	3	1963	E4	
.046	1.16	30.	17.9	54.	11.23	3	1963	E5	
.046	1.16	30.	17.9	32.	10.63	3	1963	E6	

D	L	P	G	SUB	Q/A	REF	YR	NO.	COMMENTS
0.094	1.38	30.2	2.24	-84.	4.14	3	1962	63	
0.094	4.64	30.7	2.24	-63.	2.86	3	1962	59	
0.094	1.4	30.2	4.48	105.	6.34	3	1962	64	CLGL .16 IN
0.094	1.4	29.7	4.48	67.	4.84	3	1962	65	CLGL .16 IN
0.094	1.39	30.2	4.48	30.	4.88	3	1962	66	CLGL .16 IN
0.094	1.25	29.9	4.48	99.	5.23	3	1962	67	CLGL .16 IN
0.094	1.28	29.7	4.48	97.	5.30	3	1962	68	CLGL .16 IN
0.094	1.28	29.7	4.48	106.	6.87	3	1962	69	CLGL .16 IN
0.094	1.26	30.2	4.48	90.	5.26	3	1962	70	CLGL .16 IN
0.094	1.33	30.1	4.48	52.	4.82	3	1962	71	CLGL .16 IN
0.094	1.33	29.8	4.48	101.	5.77	3	1962	72	CLGL .16 IN
0.094	1.41	30.7	4.48	-11.	4.92	3	1962	74	CLGL 1.6 IN
0.094	1.4	30.7	1.12	36.	2.68	3	1962	84	CLGL 1.6 IN
0.094	1.38	30.7	1.12	31.	2.57	3	1962	85	CLGL 1.6 IN
0.094	1.38	30.7	1.12	6.0	2.13	3	1962	86	CLGL 1.6 IN
0.094	1.38	30.7	1.12	23.	2.28	3	1962	87	CLGL 1.6 IN
0.094	1.38	31.7	1.12	-40.	2.55	3	1962	88	CLGL 1.6 IN
0.094	1.4	30.7	1.12	-25.	2.42	3	1962	92	CLGL 1.6 IN
0.094	1.36	30.7	1.12	-55.	2.61	3	1962	93	CLGL 1.6 IN
0.242	6.02	30.4	2.24	35.	1.62	3	1962	94	CLGL 3.15 IN
0.242	6.02	29.8	2.24	36.	1.635	3	1962	95	CLGL 3.15 IN
0.094	1.53	30.5	2.24	44.	3.16	3	1962	96	CLGL 1.6 IN
0.094	1.43	20.2	2.24	48.	3.00	3	1962	97	CLGL 1.6 IN
0.094	1.43	59.7	2.24	45.	3.48	3	1962	98	CLGL 1.6 IN
0.094	1.43	44.9	2.24	48.	3.14	3	1962	100	CLGL 1.6 IN
0.094	1.39	51.5	2.24	42.	3.50	3	1962	102	CLGL 1.6 IN
0.094	1.39	84.7	2.24	42.	3.56	3	1962	103	CLGL 1.6 IN
0.094	1.37	38.2	2.24	74.	4.33	3	1962	104	CLGL 1.6 IN
0.094	1.39	72.7	2.24	45.	3.54	3	1962	105	CLGL 1.6 IN
0.094	1.39	37.7	2.24	73.	3.88	3	1962	106	CLGL 1.6 IN
0.094	1.40	37.9	2.24	43.	3.20	3	1962	107	CLGL 1.6 IN
0.094	1.41	30.2	9.96	129.	9.25	3	1962	108	CLGL 1.6 IN
0.094	1.41	30.2	9.96	119.	7.10	3	1962	109	CLGL 1.6 IN
0.094	1.43	30.2	9.96	99.	5.62	3	1962	110	CLGL 1.6 IN
0.094	1.4	30.2	9.96	32.	6.18	3	1962	111	CLGL 1.6 IN
0.094	1.4	30.5	9.96	43.	6.58	3	1962	112	CLGL 1.6 IN
0.094	1.4	30.2	9.96	91.	6.28	3	1962	113	CLGL 1.6 IN
0.094	1.39	30.2	9.96	98.	5.88	3	1962	114	CLGL 1.6 IN
0.094	1.39	30.2	4.48	24.	4.78	3	1962	115	CLGL 1.6 IN
0.180	9.0	1035.	2.13	84.0	2.81	5	1957	1	
0.180	9.0	540.	2.10	-44.1	2.86	5	1957	2	
0.180	9.0	545.	2.12	-21.2	2.80	5	1957	3	
0.180	9.0	1010.	2.13	82.8	2.56	5	1957	4	
0.180	9.0	981.	2.08	103.3	2.55	5	1957	5	
0.180	9.0	1020.	2.05	29.8	2.39	5	1957	6	
0.180	9.0	1025.	2.02	11.1	2.18	5	1957	7	
0.180	9.0	1015.	1.91	-45.2	1.80	5	1957	8	
0.180	9.0	1525.	1.94	50.0	1.97	5	1957	9	
0.180	9.0	1525.	1.89	-2.0	1.89	5	1957	10	
0.305	10.57	29.5	1.12	35.6	1.205	6	1964	1	
0.305	10.55	29.5	1.12	17.5	1.015	6	1964	2	
0.305	10.53	30.5	1.12	31.6	1.02	6	1964	3	
0.305	10.55	29.2	2.24	51.	1.995	6	1964	4	

D	L	P	G	SUB	Q/A	REF	YR	NO.	COMMENTS
0.307	10.55	30.6	2.24	62.3	1.54	6	1964	5	
0.307	10.53	29.8	2.24	44.4	1.80	6	1964	6	
0.307	10.52	30.7	2.24	19.7	1.525	6	1964	7	
0.307	10.60	30.5	2.24	27.4	1.61	6	1964	8	
0.304	9.40	29.8	1.12	12.6	1.105	6	1964	10	
0.305	9.38	29.9	1.12	37.7	1.17	6	1964	9	
0.305	9.30	29.9	1.12	39.0	1.265	6	1964	11	
0.305	9.30	30.6	1.12	47.0	1.855	6	1964	12	
0.305	4.50	29.9	1.12	88.6	3.07	6	1964	13	
0.305	4.45	30.5	1.12	44.5	2.07	6	1964	14	
0.305	4.50	29.7	1.12	61.6	2.345	6	1964	15	
0.305	4.50	30.2	3.36	57.1	1.514	6	1964	16	
0.242	7.94	30.0	3.36	68.8	2.517	6	1964	17	
0.242	8.0	30.0	3.36	12.8	2.340	6	1964	18	
0.242	8.05	30.5	3.36	44.5	2.378	6	1964	19	
0.307	4.56	30.6	1.12	44.0	1.485	6	1964	22	
0.307	4.56	30.0	1.12	59.5	1.895	6	1964	23	
0.307	4.55	30.7	1.12	52.8	1.712	6	1964	24	
0.307	4.65	30.1	2.24	81.4	2.715	6	1964	25	
0.244	6.73	30.3	2.24	61.6	2.17	6	1964	28	
0.246	6.67	30.7	2.24	39.1	1.77	6	1964	29	
0.244	6.74	30.6	2.24	43.3	1.92	6	1964	30	
0.246	6.68	30.2	2.24	56.7	1.96	6	1964	31	
0.246	6.70	29.7	4.48	85.8	3.10	6	1964	32	
0.246	6.70	30.1	4.48	59.8	2.54	6	1964	33	
0.246	6.70	30.0	4.48	72.4	2.84	6	1964	33	NO. 33(A)
0.246	6.65	30.0	4.48	75.5	2.98	6	1964	34	
0.307	8.32	33.6	3.36	76.1	2.65	6	1964	35	
0.307	8.05	33.2	3.36	55.0	2.26	6	1964	36	
0.307	8.18	33.3	3.36	67.8	2.44	6	1964	37	
0.246		29.7	2.24	4.0	2.10	7	1964	1	
0.246		29.0	2.24	27.0	1.976	7	1964	2	
0.246		29.1	2.24	57.0	2.26	7	1964	3	
0.246		29.5	4.48	61.0	2.21	7	1964	4	
0.246		29.4	4.48	28.0	2.62	7	1964	5	
.047		64.9	2.24	75.	4.31	7	1964	7	
.047		64.7	2.24	58.	4.11	7	1964	8	
.047		64.2	2.24	45.	3.73	7	1964	9	
.047		62.2	2.24	33.	3.55	7	1964	11	

APPENDIX D

APPARATUS

Experimentation was conducted on existing apparatus which had been constructed for a previous study (1), and which had been used for several other boiling and CHF investigations. Only minor modification was necessary for this investigation. The apparatus is shown schematically in figure 1. The fluid loop consists of a closed main circulation loop, with a by-pass parallel test section line. All components are of corrosion resistant materials, The system contains a main circulating pump, accumulator, the test section line with flowmeter, preheater, test section with instrumentation, main loop by-pass line and heat exchanger with city water cooling. Auxiliary to this are a fill pump, supply tank, degassing tank and a continuously operating demineralizer. Power is supplied to the test section by a pair of a.c. motor - d.c. generator sets.

The main supply pump is a two-stage turbine type pump, driven by a 3 hp induction motor. The pump is capable of delivering 3.6 gallons per minute at 260 psi. A bladder type accumulator located at the exit side of the pump serves to damp out pressure fluctuations. Plug valves at each end of the by-pass line provide control for the test section pressure. Test section flow rate is determined by a Fisher Porter flowmeter and controlled with a Hoke needle valve. This needle valve also provides the 150 psi up-stream pressure drop necessary to insure flow stability.

The motor-generator sets consist of 440 v., 3 phase synchronous motors and 36 kilowatt d.c. generators, connected in series. Each generator is nominally rated at 12 volt and 3000 amps. A total power of approximately 60 kilowatt is available at the test section.

Instrumentation is available for monitoring pressure levels, temperatures, test section flow rate and test section current and voltage. Test section exit pressures are read on Bourdon type pressure gauges with specified accuracy of 0.5 psi or better. The gauges are calibrated periodically with a dead weight tester. Test section flow rate is determined with the Flowrater Meters; the range of flow rate is varied by changing tubes and floats. Meters are calibrated in place and calibration curves are available. Test section power is obtained from the test section voltage and current. Voltage across the test section is read directly with a Weston variable scale, calibrated d.c. voltmeter. Current is read across a National Bureau of Standards shunt, with a specified calibration of 60.17 amps/mv.

All temperatures are determined by thermocouples manufactured from 30-gauge, duplex, Copper-Constantan thermocouple wire. Test section inlet and exit temperatures are measured on thermocouples inserted into the flow with Conax fittings. Output voltages of the thermocouples and current shunt are displayed upon a Minneapolis-Honneywell Brown, 0 - 26 mv., single channel recorder.

A much more detailed description of the apparatus is contained in ref. 1; however, certain modifications have been made. The more important of these include:

a. The braided copper flexible power connection and the rigid up-stream jumper from the test panel to test section bus connection were replaced during this study by Mackworth Reese Flexible Dry Jumpers. This requires that the test section be supported. Support is porvided by a laboratory support and clamp, with electrical insulation.

b. The braided copper power supply cables have been replaced with Mackworth Reese, water cooled cables. These cables are cooled by city water tapped off of the heat exchanger coolant supply line, discharged to a drain.

c. The preheater has been modified by the addition of three 5 kilowatt immersion heaters, with separate controls. These heaters are in series with the original 6 kilowatt variable control heater and provide any desired preheating from 0 to 21 kilowatt, with steady state operation.

d. The steam heater in the supply tank has been removed as it was found that sufficient heating and degassing could be obtained from the degas tank heaters alone.

e. The thermocouple selection switch now consists of two Leeds and Northrup, 12 position selector switches, in series. Thermocouple wires are connected through screw type terminal boards, thus reducing set-up time. This modification was made in order to increase the number of thermocouples which could be patched to the recorder. Resistance tests showed no appreciable change in resistance as a result of the added switch. In this study, only the primary switch was used.

f. When using the very short test sections, or those of type "A" Nickel, the test section resistance was so low as to make the motor-generator control highly erratic. To alleviate this problem, a series resistance was fabricated. This resistance consisted of a 10 inch length of 0.242 inch type 304 stainless steel tubing, with bus connections, placed electrically in series with the test section and cooled by city water. This added resistance caused an additional voltage drop of approximately 5 volts across the power supply, which improved the m.-g. set control considerably. Power to the test section was determined by reading voltage drop across the test section only, and by reading current through the series combination.

g. In the tests where very high power applications were required to reach CHF, the test section exit bus connection was found to heat up excessively, causing smoking and in one case a small fire, not to mention great inconvenience when changing test sections. In view of this, a bus connection coolant scheme was devised. A

$\frac{1}{2}$-inch copper nipple and two elbows were silver brazed to the brass bus connection. City water, from the power lead cooling system, was diverted through this nipple, drawing heat from the bus connection. Heat balance checks when operating, comparing power input to fluid temperature rise through the test section showed that this cooling device had no effect on the test data. No further overheating problems were encountered once this cooler was installed, and it was used whenever there was a requirement for high power input.

Test Section

Because each test destroyed the test section, construction was kept as simple as possible. A test section was constructed by facing a 3/4 inch brass pipe cap and drilling a hole in the center of it. A section of the test tubing was silver soldered into this hole. At the other end of the desired heated length, a 5/16 inch brass bushing, about 5/8 inch long, drilled to the O.D of the tubing was silver soldered to the tubing. Heated length was measured from fillet to fillet. A calming length of approximately 20 diameters extended beyond the up-stream bushing.

Sections were installed in the apparatus by threading the pipe cap, the down-stream end, to the down-stream plenum chamber which contained the exit pressure tap and exit thermocouple. The up-stream end was installed by inserting the calming length into a Conax fitting sealed with a neoprene washer. Brass bus connections were installed on the bushing and pipe cap completing the electrical circuit.

In the study of maximum CHF it was necessary to machine the outside diameter of the 0.242 in. stainless steel tubing, in order to obtain adequate impendance matching. Tube wall thickness used is noted in the listing of Data (Appendix C, runs #29, 30 and 31).

<u>Beryllium-Copper Tubing</u>

At the begining of the experimentation, tubing of 2% alloy, Be-Cu was ordered, with inside diameter of 0.079 inches (2 mm). It was felt that this tubing would be easier to work with than the type 304 stainless steel which had been used in previous studies. Several test sections were made up and tests were run. Unfortunately, considerable difficulty was encountered using the Be-Cu tubing. Some of the more important problems are discussed below.

About the only fabrication advantage that the Be-Cu tubing had was that it could be cut into desired lengths easily. Unfortunately, the alloy was not stable at temperatures in excess of 1100 degress F. If test sections were constructed by silver soldering, the tubing near the soldered connections underwent a phase change and this was accompanied by a change in electrical resistance of almost 40%. Since this change in resistance occured only in the region where heat was applied for soldering, the local change in resistance caused an uneven axial distribution of heat flux. Data obtained by measuring total test section voltage drop and current did not indicate the actual CHF.

Test sections were constructed using soft solder, but the high exit temperatures encountered during operation caused the soft solder to puddle just before reaching CHF. No leaks developed, but this was felt to be less than satisfactory. Higher temperature (90-10) soft solder was tried, with no better results.

The lower resistance Be-Cu tubing caused m.g. set control problems. The series resistance previously mentioned alleviated this problem, but tended to complicate the procedure.

CHF data was obtained using this Be-Cu tubing. Subsequent comparison with similar data obtained with type 304 stainless steel tubing showed that the Be-Cu data was inconsistent, for either the soft soldered or silver soldered sections.

A corrosion check was made with the Be-Cu tubing. After one hour at a heat flux of about one-third of CHF, the inside of the tube showed distinct blackening, using degassed and demineralized water.

In view of the various difficulties, it was concluded that the Be-Cu tubing was unsatisfactory for experiments of this type, and thick wall, type 304 stainless steel tubing, with inside diameter of 0.079 inches (2 mm) was obtained.

The stainless steel tubing could be silver soldered satisfactorily and the data obtained from it showed considerably more consistency. The data presented as the result of this investigation was obtained using type 304 stainless steel, except where noted to be Be-Cu (runs #2 through #13), and except for the D = 0.180 inch tubing which was type "A" Nickel.

APPENDIX E.

PROCEDURE

Once the test section was installed and determined to be free of leaks, the test loop and degas tank were filled with distilled water. Pet-cocks and air bleed spigots are located at various points throughout the apparatus, and air was bled from the system, both under the static head of the degas tank, and with the pump circulating water through the loop. Disolved air and other gas was then removed from the test water by electrically heating the degassing tank. As mentioned in ref. 1 and 2, about one half hour of degassing was necessary to insure satisfactory air removal.

Once the degassing procedure was completed, and the water in the loop cooled by the heat exchanger, the desired test section flow rate and pressure were established.

Since it was desired to present data on CHF at a particular exit subcooling, it was necessary to determine and establish a desired test section exit temperature. This was complicated by the fact that exit temperature is dependent upon the inlet temperature, which could be controlled by the preheater, and upon the heat flux in the test section. In order to obtain the desired exit temperature and pressure at the point of CHF, it was necessary to increase the power to the test section very slowly, and to continually adjust the test section pressure and the preheater controlling the inlet temperature. In some case, the 21 kilowatts of preheat available was not sufficient, in which case the flow of cooling water to the heat exchanger was reduced. This procedure was difficult in that the exit temperature showed a long (about 20 minutes) response time to adjustments in the flow of city water cooling the flow through the heat exchanger.

Often the process of obtaining the desired exit subcooling was a very delicate one. This was particulary true when the curve of CHF versus subcooling was approximately parallel to the curve of heat flux versus subcooling as determined by the First Law of Thermodynamics.

In the large majority of the test runs, it was possible to obtain the desired value of exit subcooling within 5% at the point of CHF. In those few cases where exit subcooling varied more than 5% from the desired value, an auxiliary plot of CHF versus subcooling was made, to determine the appropriate value, or to determine that the variation of CHF with subcooling was not significant in the range of subcooling in question.

For the purpose of this study, CHF was defined as the heat flux at tube failure. While the heat flux thus determined may exceed that at the departure from nucleate boiling (DNB), heat flux so defined should coincide closely with the heat flux at the point of transition from nucleate to film boiling. No burnout protection device was used in this study, as it was felt that less time and materials were involved in the fabrication of a new test section for each run than in the construction of such a device. Furthermore, there can be no doubt that a genuine maximum heat flux has been established when the tube ruptures.

All experiments were conducted with the test section in the horizontal position. System water was degassed only periodically. Care was taken to insure that no air was trapped in the system when replacing sections and extensive air bleeding was done before commencing a new run. The inside surface of the test section was given a thorough cleaning with acetone prior to installation. No additives were introduced into the water.

In almost all cases, the tube ruptured within three diameters of the exit connection. Thus conditions in the exit plenum chamber were considered to be representative of those at the point of failure. A recent study by Lopina (3) has shown that the pressure within the plenum is virtually identical with that at the down-stream end of the heated section throughout the subcooled region. The test section and plenum were wrapped with fiberglass insulation to greater than the critical radius to minimize heat loss.

Because the recorder used presents only one channel, it was necessary to estimate certain quantities at the point of CHF; however, near failure, very small increments of power were applied, and all quantities were checked frequently. Data not obtained right at failure could be readily inferred from that recorded just prior to reaching CHF. All changes in conditions were made very gradually during a run, and once the procedure was well established, it was possible to obtain a set of data, e.g. one data point, in approximately forty-five minutes.

D 0.079 in. P 90 psia Run # 45

L 3.15 in. L/D 40 Date 2/25

G 3.69 $\times 10^6$ lbm/ft^2 hr = w = 125.5 lbm/hr

= 74 % on Flowrater FP-1/2-21-G-10/80

Flow %	T_{in} mv	T_{out} mv	p psig	E v	I mv	Variac %
74	([illegible] °F) 0.42	(149 °F) 2.70	75	6.95	8.52	0
✓	✓	3.04	✓	7.50	9.06	✓
✓	✓	3.42	✓	8.00	9.56	✓
✓	✓	3.72	✓	8.40	9.92	✓
✓	✓	4.20	76	9.05	10.60	✓
✓	✓	4.42	73	9.36	10.90	✓
At Failure						
✓	0.46	5.54	75	10.55	12.05	✓

53 °F 260 °F 89.7 psia

Heat Balance

$w \cdot \Delta T = 125.5 \times 92 = 12300$

$E \cdot I(60.17)(3.41) = 6.95 \times 8.52 \times 60.17 \times 3.41 = 12150$

ok

h sat 290.3

$CHF = \dfrac{E \cdot I(60.17)(3.41)}{\pi D \cdot L/144}$

h bulk 228.6

h_{sat} 61.7 Btu/lbm

$CHF = \dfrac{10.55 \times 12.05 \times (60.17)(3.41)}{\pi \times 0.079 \times 3.15/144} = 4.82$ Btu/ft^2 hr

Obtain Temperature from Thermocouple Conversion Chart

h sat = h at p h bulk = h at T_{out}

APPENDIX G

DEVELOPMENT OF THE MIT SMALL DIAMETER CORRELATION

This equation was developed by the authors for the low pressure MIT data and fits the majority of it to ± 25%.

$$(q/A)_{cr} = 1.738 \times 10^6 \,(4.48 + .01\,\Delta h_{sat})\,(G/G_o)^{.35}\,(D/D_o)^{-.55}\left(\frac{L/D}{40}\right)^{-.12}$$

The correlation was developed in the following manner: The 30 and 90 psia plots (figures 3 & 4) for 2 mm data (.079 in.) were linearized to determine the slope and the intercept of the part of the equation dependent on subcoolong. The flow rate dependence was determined by plotting on log-log paper, cross-plots of the 2 mm data at constant subcoolings. Also plotted were cross-plots of Bergles'(11) and Wessel's (26) data for constant subcoolings. A mean value of the slopes was used to determine the exponent of the flow rate term. The same technique was used for diameter and L/D dependence, using the data of figures 5 and 7, and the data of Bergles (11). The leading constant was determined by the zero subcooling intercepts and the values of the constants G_o and D_o. No pressure term is included as the variation of CHF with pressure over this range is neither well ordered nor of large magnitude.

The correlation was plotted over the MIT data and small adjustments were made in the constants and exponents to get a better all around fit and balance out the positive and negative deviations.

A further refinement of the correlation would be to make the heat flux a weak 2nd or 3rd order dependence on subcooling, eliminating the linear approximation. A minimum could be introduced at about $\Delta h_{sat} = 20$ to 30 Btu/lbm.

APPENDIX H - NOMENCLATURE

a - Acceleration in g's

A - Heat Transfer Area, ft^2

c - Specific Heat, Btu/lbm-deg F

D - Tube Diameter, inches

g - Acceleration of gravity, ft/sec^2

g_o - 32.2 lb_f-ft/lbm-sec^2

h - Heat Transfer Coefficient, Btu/hr-ft^2-deg F

h - Enthalpy, Btu/lbm

J - Mechanical Equivalent of Heat, 778 ft-lb_f/Btu

k - Thermal Conductivity, Btu/hr-ft-deg F

L - Tube Length, inches

p - Pressure, psia

Pr - Prandtl Number

q/A - Heat Flux, Btu/hr-ft^2

t - Temperature, deg F

T - Temperature, degrees Rankine

ΔT_{sat} - (T_{sat} - T_b), deg F, Subcooling

v - Specific Volume, ft^3/lbm

V - Velocity, ft/sec

ν - Viscosity, ft^2/hr

μ - Viscosity, lbm/hr-ft

ρ - Density, lbm/ft^3

σ - Surface Tension, lb_f/ft

Δh_{sat} - (h_{sat} - h_b), Btu/lbm, Subcooling

Subscripts

bo	- At the point of burnout or critical heat flux
cr	- At the point of burnout or critical heat flux
b	- Bulk or mixed mean conditions
crit	- Critical
fg	- Saturated Condition - Vapor Condition
g	- Vapor Properties
l	- Liquid Properties
pb	- Pool Boiling
s	- Saturated Conditions
sat	- Saturated Conditions
v	- Vapor Properties
w	- Wall

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